Subject :vehicle technology : المادة

Weekly Hours : Theoretical:1 : النظري : 1

Tutorial :1 : التمارين : 1

Experimental : 3 : العملي : 3

Units:4 : الوحدات : 4

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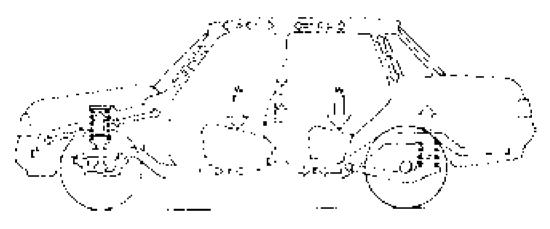
عدد ساعات التدريب	الموضوع	ألأسبوع
2ساعة	نبذة تاريخية عن المحركات ـ أجزاء وعمل المحرك فتح وتركيب	ألأول
2ساعة	الفروقات بين محركات البنزين والديزل ويبن المحركات الرباعية والثنائية الشوط	الثاني
2ساعة	منظومة الوقود /شرح نظري وعملي	الثالث
	فتح وتركيب مضخة الوقود والمغذي وكيفية صيانتهما	الرابع
2ساعة	منظومة التزييت	الخامس
	فتح وتركيب مضخة الزيت وكيفية صيانتها	السادس
2ساعة	منظومة التبريد	السابع
2ساعة	منظومة الأشتعال ألأعتيادية	الثامن
2ساعة	فتح وتركيب موزع الشرر وكيفية ضبط فتحة(اليلاتبن) وكيفية الصيانة	التاسع
2ساعة	منظومة الأشتعال ألألكترونية	العاشر
2ساعة	أجهزة نقل الحركة/ الفاصل	الحادي عشر
2ساعة	صندوق التروس	الثاني عشر
2ساعة	فتح وتركيب صندوق التروس وصيانته	الثالث عشر
2ساعة	المحور الخلفي/ فتح وتركيب	الرابع عشر
2ساعة	فتح وتركيب المحور الخلفي وصيانته	الخامش عشر

1 Vehicle Structure

1.1 ntegral body construction

The integral or unitary body structure of a car can be considered to be made in the form of three box compartments the middle and largest compartment stretching between the front and rear road wheel axles provides the passenger space, the extended front box built over and ahead of the front road wheels enclosing the engine and transmission units and the rear box behind the back axle providing boot space for luggage.

These box compartments are constructed in the form of a framework of ties (tensile) and struts (compressive), pieces (Fig. 1.1(a b)) made from rolled sheet steel pressed into various shapes such as rectangular, triangular, trapezium, top-hat or a combination of these to form closed box thin gauge sections. These sections are designed to resist direct tensile and compressive or bending and torsional loads, depending upon the positioning of the members within the structure.

Fig. 1.1 (a and b) Structural tensile and compressive loading of car body

1

1.1.1 Description and function of body components (Fig. 1.2)

The ma or individual components comprising the body shell will now be described separately under the following subheadings:

1 Window and door pillars
2 Windscreen and rear window rails
3 Cantrails
4 Roof structure
5 Upper quarter panel or window
6 Floor seat and boot pans
7 Central tunnel
8 Sills
9 Bulkhead
10 Scuttle
11 Front longitudinals
12 Front valance
13 Rear valance
14 Toe board
15 eel board

indow and door pillars (Fig. 1.2(3, 5, 6, and 8)) Windowscreen and door pillars are identified by a letter coding the front windscreen to door pillars are referred to as   , the centre side door pillars as   and the rear door to quarter panel as   . These are illustrated in Fig. 1.2.

These pillars form the part of the body structure which supports the roof. The short form A pillar and rear   pillar enclose the windscreen and quarter windows and provide the glazing side channels, whilst the centre BC pillar extends the full height of the passenger compartment from roof to floor and supports the rear side door hinges. The front and rear pillars act as struts (compressive members) which transfer a proportion of the bending effect, due to underbody sag of the wheelbase, to each end of the cantrails which thereby become reactive struts, opposing horizontal bending of the passenger compartment at floor level. The central BC pillar however acts as ties (tensile members), transferring some degree of support from the mid-span of the cantrails to the floor structure.

indscreen and rear window rails (Fig. 1.2(2)) These box-section rails span the front window pillars and rear pillars or quarter panels depending upon design, so that they contribute to the resistance opposing transverse sag between the wheel track by acting as compressive members. The other function is to support the front and rear ends of the roof panel. The undersides of the rails also include the glazing channels.

antrails (Fig. 1.2(4)) Cantrails are the horizontal members which interconnect the top ends of the vertical A and BC or BC and   door pillars (posts). These rails form the side members which make up the rectangular roof framework and as such are sub ected to compressive loads. Therefore, they are formed in various box-sections which offer the greatest compressive resistance with the minimum of weight and blend in with the roofing. A drip rail (Fig. 1.2(4)) is positioned in between the overlapping roof panel and the cantrails, the oins being secured by spot welds.

Roof structure (Fig. 1.2) The roof is constructed basically from four channel sections which form the outer rim of the slightly dished roof panel. The rectangular outer roof frame acts as the compressive load bearing members. Torsional rigidity to resist twist is maximized by welding the four corners of the channel-sections together. The slight curvature of the roof panel stiffens it, thus preventing winkling and the collapse of the unsupported centre region of the roof panel. With large cars, additional cross-rail members may be used to provide more roof support and to prevent the roof crushing in should the car roll over.

pper uarter panel or window (Fig. 1.2(6)) This is the vertical side panel or window which occupies the space between the rear side door and the rear window.   riginally the quarter panel formed an important part of the roof support, but improved pillar design and the desire to maximize visibility has either replaced them with quarter windows or reduced their width, and in some car models they have been completely eliminated.

loor seat and boot pans (Fig. 1.3) These constitute the pressed rolled steel sheeting shape to enclose the bottom of both the passenger and luggage compartments. The horizontal spread-out pressing between the bulkhead and the heel board is called the   , whilst the raised platform over the rear suspension and wheel arches is known as the   or   . This in turn oins onto a lower steel pressing which supports luggage and is referred to as the   .

To increase the local stiffness of these platform panels or pans and their resistance to transmitted vibrations such as drumming and droning, many narrow channels are swaged (pressed) into the steel sheet, because a sectional end-view would show a
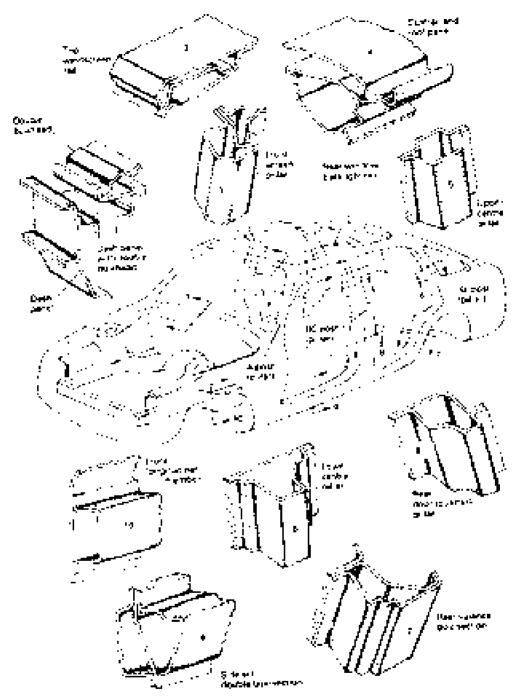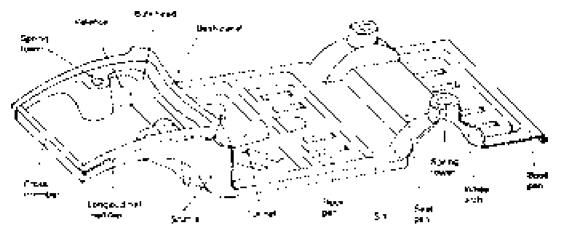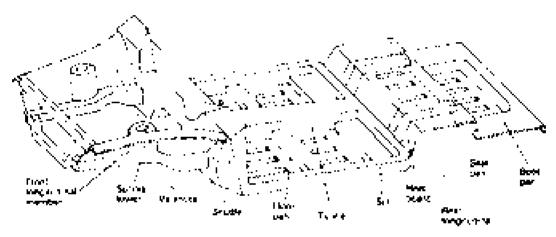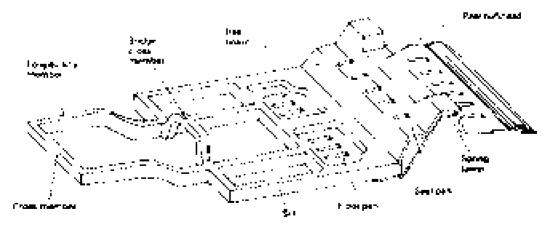
Fig. 1.2 oad bearing body box-section members

semi-corrugated profile (or ribs). These channels provide rows of shallow walls which are both bent and stretched perpendicular to the original flat sheet. In turn they are spaced and held together by the semicircular drawn out channel bottoms. Provided these swages are designed to lay the correct way and are not too long, and the metal is not excessively stretched, they will raise the rigidity

3

(a) Short central tunnel with front valance, rear wheel arches, bulkhead and front spring towers

(b) Large tunnel tunnel with front valance and spring tower with front box section reinforced seat pan

(c) Longitudinal frame members, on the front seat pan with bulkhead and spring towers

Fig. 1.3 (a–c) Platform chassis

of these panels so that they are equivalent to a sheet which may be several times thicker.

entral tunnel (Fig. 1.3(a and b)) This is the curved or rectangular hump positioned longitudinally along the middle of the floor pan. riginally it was a necessary evil to provide transmission space for the gearbox and propeller shaft for rear wheel drive, front-mounted engine cars, but since the chassis has been replaced by the integral box-section shell, it has been retained with front wheel drive, front-mounted engines as it contributes considerably to the bending rigidity of the floor structure. Its secondary function is now to house the exhaust pipe system and the hand brake cable assembly.

Sills (Figs 1.2(9) and 1.3(a, b and c)) These members form the lower horizontal sides of the car body which spans between the front and rear road-wheel wings or arches. To prevent body sag between the wheelbase of the car and lateral bending of the structure, the outer edges of the floor pan are given support by the side sills. These sills are made in the form of either single or double box-sections (Fig. 1.2(9)). To resist the heavier vertical bending loads they are of relatively deep section.

pen-top cars, such as convertibles, which do not receive structural support from the roof members, usually have extra deep sills to compensate for the increased burden imposed on the underframe.

ulkhead (Figs 1.2(1) and 1.3(a and b)) This is the upright partition separating the passenger and engine compartments. Its upper half may form part of the dash panel which was originally used to display the driver s instruments. Some body manufacturers refer to the whole partition between engine and passenger compartments as the dash panel. If there is a double partition, the panel next to the engine is generally known as the and that on the passenger side the or . The scuttle and valance on each side are usually oined onto the box-section of the bulkhead. This braces the vertical structure to withstand torsional distortion and to provide platform bending resistance support. Sometimes a bulkhead is constructed between the rear wheel arches or towers to reinforce the seat pan over the rear axle (Fig. 1.3(c)).

Scuttle (Fig. 1.3(a and b)) This can be considered as the panel formed under the front wings which spans between the rear end of the valance, where it meets the bulkhead, and the door pillar and wing. The lower edge of the scuttle will merge with the floor pan so that in some cases it may form part of the toe board on the passenger compartment side. Usually these panels form inclined sides to the bulkhead, and with the horizontal ledge which spans the full width of the bulkhead, brace the bulkhead wall so that it offers increased rigidity to the structure. The combined bulkhead dash panel and scuttle will thereby have both upright and torsional rigidity.

ront longitudinals (Figs 1.2(10) and 1.3(a and b)) These members are usually upswept box-section members, extending parallel and forward from the bulkhead at floor level. Their purpose is to withstand the engine mount reaction and to support the front suspension or subframe. A common feature of these members is their ability to support vertical loads in con unction with the valances. owever, in the event of a head-on collision, they are designed to collapse and crumble within the engine compartment so that the passenger shell is safeguarded and is not pushed rearwards by any great extent.

ront valance (Figs 1.2 and 1.3(a and b)) These panels pro ect upwards from the front longitudinal members and at the rear oin onto the wall of the bulkhead. The purpose of these panels is to transfer the upward reaction of the longitudinal members which support the front suspension to the bulkhead. Simultaneously, the longitudinals are prevented from bending sideways because the valance panels are shaped to slope up and outwards towards the top. The panelling is usually bent over near the edges to form a horizontal flanged upper, thus presenting considerable lateral resistance. Furthermore, the valances are sometimes stepped and wrapped around towards the rear where they meet and are oined to the bulkhead so that additional lengthwise and transverse stiffness is obtained.

If coil spring suspension is incorporated, the valance forms part of a semi-circular tower which houses and provides the load reaction of the spring so that the merging of these shapes compounds the rigidity for both horizontal lengthwise and lateral bending of the forward engine and transmission compartment body structure. Where necessary, double layers of sheet are used in parts of the spring housing and at the rear of the valance where they are attached to the bulkhead to relieve some of the concentrated loads.

Rear valance (Fig. 1.2(7)) This is generally considered as part of the box-section, forming the front half of the rear wheel arch frame and the panel immediately behind which merges with the heel board and seatpan panels. These side inner-side panels position the edges of the seat pan to its designed side profile and thus stiffen the underfloor structure above the rear axle and suspension. When rear independent coil spring suspension is adopted, the valance or wheel arch extends upwards to form a spring tower housing and, because it forms a semi-vertical structure, greatly contributes to the stiffness of the underbody shell between the floor and boot pans.

Toe board The toe board is considered to form the lower regions of the scuttle and dash panel near where they merge with the floor pan. It is this panelling on the passenger compartment side where occupants can place their feet when the car is rapidly retarded.

eel board (Fig. 1.3(b and c)) The heel board is the upright, but normally shallow, panel spanning beneath and across the front of the rear seats. Its purpose is to provide leg height for the passengers and to form a raised step for the seat pan so that the rear axle has sufficient relative movement clearance.

1.1. *latform chassis* (Fig. 1.3(a–c))
Most modern car bodies are designed to obtain their rigidity mainly from the platform chassis and to rely less on the upper framework of window and door pillars, quarter panels, windscreen rails and contrails which are becoming progressively slender as the desire for better visibility is encouraged.

The ma ority of the lengthwise (wheelbase) bending stiffness to resist sagging is derived from both the central tunnel and the side sill box-sections (Fig. 1.3(a and b)). If further strengthening is necessary, longitudinal box-section members may be positioned parallel to, but slightly inwards from, the sills (Fig. 1.3(c)). These lengthwise members may span only part of the wheelbase, or the full length, which is greatly influenced by the design of road wheel suspension chosen for the car, the depth of both central tunnel and side sills, which are built into the platform, and if there are subframes attached fore and aft of the wheelbase (Fig. 1.6 (a and b)).

Torsional rigidity of the platform is usually derived at the front by the bulkhead, dash pan and scuttle (Fig. 1.3(a and b)) at the rear by the heel board, seat pan, wheel arches (Fig. 1.3(a, b and c)), and if independent rear suspension is adopted, by the coil spring towers (Fig. 1.3(a and c)). Between the wheelbase, the floor pan is normally provided with box-section cross-members to stiffen and prevent the platform sagging where the passenger seats are positioned.

1.1.3 *Stiffening of platform chassis*
(Figs 1.4 and 1.5)
To appreciate the stresses imposed on and the resisting stiffness offered by sheet steel when it is sub ected to bending, a small segment of a beam greatly magnified will now be considered (Fig. 1.4(a)). As the beam deforms, the top fibres contract and the bottom fibres elongate. The neutral plane or axis of the beam is defined as the plane whose length remains unchanged during deformation and is normally situated in the centre of a uniform section (Fig. 1.4(a and b)).

The stress distribution from top to bottom within the beam varies from zero along the neutral axis (NA), where there is no change in the length of the fibres, to a maximum compressive stress on the outer top layer and a maximum tensile stress on the outer bottom layer, the distortion of the fibres being greatest at their extremes as shown in Fig. 1.4(b).

It has been found that bending resistance increases roughly with the cube of its distance from the neutral axis (Fig. 1.5(a)). Therefore, bending resistance of a given section can be greatly improved for a given weight of metal by taking metal away from the neutral axis where the metal fibres do not contribute very much to resisting distortion and placing it as far out as possible where the distortion is greatest. Bending resistance may be improved by using longitudinal or cross-member deep box-sections (Fig. 1.5(b)) and tunnel sections (Fig. 1.5(c)) to restrain the platform chassis from buckling and to stiffen the flat horizontal floor seat and boot pans. So that vibration and drumming may be reduced, many swaged ribs are pressed into these sheets (Fig. 1.5(d)).

1.1. *Body subframes* (Fig. 1.6)
Front or rear subframes may be provided to brace the longitudinal side members so that independent suspension on each side of the car receives adequate support for the lower transverse swing arms (wishbone members). Subframes restrain the two halves of the suspension from splaying outwards or the
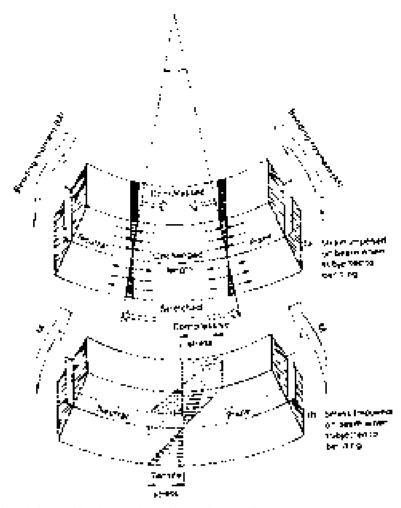
Fig. 1.4 Stress and strain imposed on beam when subjected to bending

longitudinal side members from lozenging as alternative road wheels experience impacts when travelling over the irregularities of a normal road surface.

It is usual to make the top side of the subframe the cradle for the engine or engine and transmission mounting points so that the main body structure itself does not have to be reinforced. This particularly applies where the engine, gearbox and final drive form an integral unit because any torque reaction at the mounting points will be transferred to the subframe and will multiply in proportion to the overall gear reduction. This may be approximately four times as great as that for the front mounted engine with rear wheel drive and will become prominent in the lower gears.

One advantage claimed by using separate subframes attached to the body underframe through the media of rubber mounts is that transmitted vibrations and noise originating from the tyres and road are isolated from the main body shell and therefore do not damage the body structure and are not relayed to the occupants sitting inside.

Cars which have longitudinally positioned engines mounted in the front driven by the rear wheels commonly adopt beam cross-member subframes at the front to stiffen and support the hinged transverse suspension arms (Fig. 1.6(a)). Saloon cars employing independent rear suspension sometimes prefer to use a similar subframe at the rear which provides the pivot points for the semi-trailing arms because this type of suspension requires greater support than most other arrangements (Fig. 1.6(a)).
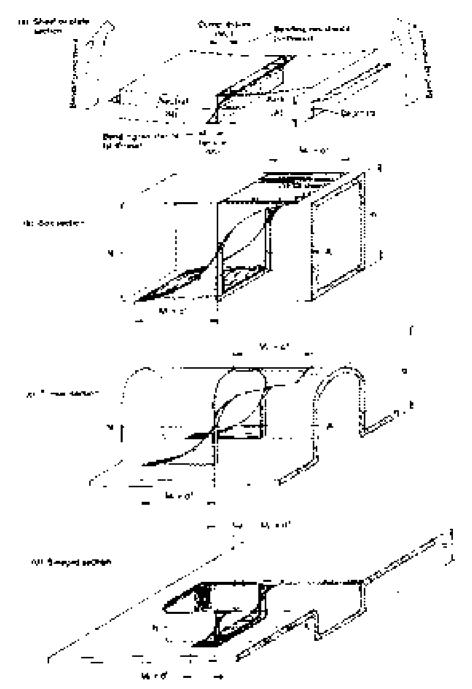
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Fig. 1.5 Bending resistance for various sheet sections

When the engine, gearbox and final drive are combined into a single unit, as with the front longitudinally positioned engine driving the front wheels where there is a large weight concentration, a subframe gives extra support to the body longitudinal side members by utilising a horseshoe shaped frame (Fig. 1.6(b)). This layout provides a platform for the entire mounting points for both the swing arm and anti-roll bar which between them make up the lower part of the suspension.

(a) Rectangular subframe

(b) Aerospace subframe

(c) Electro cross-member subframe
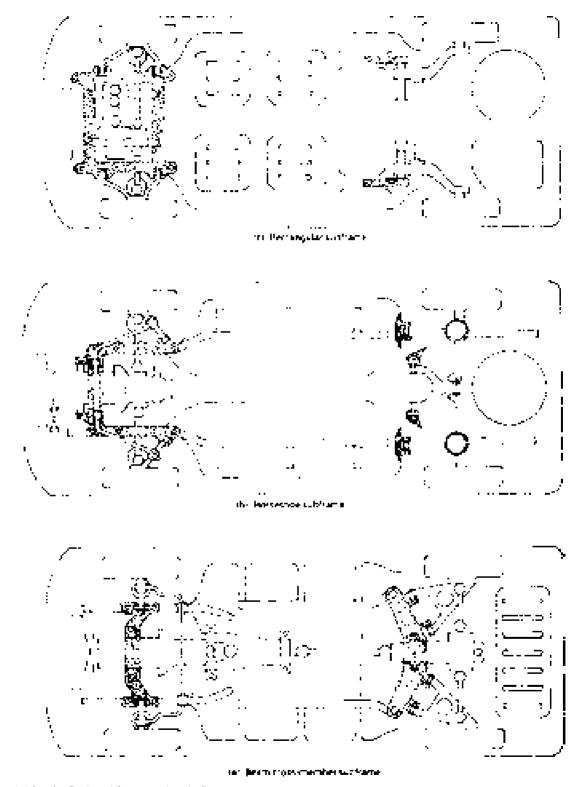
Fig. 1.6 (a–c) Body subframe and underfloor structure

Front wheel drive transversely positioned engines with their large mounting point reactions often use a rectangular subframe to spread out both the power and transmission unit s weight and their dynamic reaction forces (Fig. 1.6(c)). This configuration provides substantial torsional rigidity between both halves of the independent suspension without relying too much on the main body structure for support.

Soundproofing the interior of the passenger compartment (Fig. 1.7)

Interior noise originating outside the passenger compartment can be greatly reduced by applying layers of materials having suitable acoustic properties over floor, seat and boot pans, central tunnel, bulkhead, dash panel, toeboard, side panels, inside of doors, and the underside of both roof and bonnet etc. (Fig. 1.7).

Acoustic materials are generally designed for one of three functions:

a) Insulation from noise   This may be created by forming a non-conducting noise barrier between the source of the noises (which may come from the engine, transmission, suspension tyres etc.) and the passenger compartment.

b) Absorption of vibrations   This is the transference of excited vibrations in the body shell to a media which will dissipate their resultant energies and so eliminate or at least greatly reduce the noise.

c)   amping of vibrations   When certain vibrations cannot be eliminated, they may be exposed to some form of material which in some way
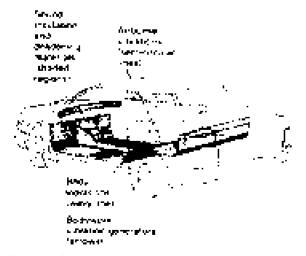

Fig. 1.7 Car body sound generation and its dissipation

modifies the magnitude of frequencies of the vibrations so that they are less audible to the passengers.

The installation of acoustic materials cannot completely eliminate boom, drumming, droning and other noises caused by resonance, but merely reduces the overall noise level.

nsulation   Because engines are generally mounted close to the passenger compartment of cars or the cabs of trucks, effective insulation is important. In this case, the function of the material is to reduce the magnitude of vibrations transmitted through the panel and floor walls. To reduce the transmission of noise, a thin steel body panel should be combined with a flexible material of large mass, based on PVC, bitumen or mineral wool. If the insulation material is held some distance from the structural panel, the transmissibility at frequencies above 400   z is further reduced. For this type of application the loaded PVC material is bonded to a spacing layer of polyurethane foam or felt, usually about 7 mm thick. At frequencies below 400   z, the use of thicker spacing layers or heavier materials can also improve insulation.

bsorption   For absorption, urethane foam or lightweight bonded fibre materials can be used. In some cases a vinyl sheet is bonded to the foam to form a roof lining. The required thickness of the absorbent material is determined by the frequencies involved. The minimum useful thickness of polyurethane foam is 13 mm which is effective with vibration frequencies above 1000   z.

amping   To damp resonance, pads are bonded to certain panels of many cars and truck cabs. They are particularly suitable for external panels whose resonance cannot be eliminated by structural alterations. Bituminous sheets designed for this purpose are fused to the panels when the paint is baked on the car. Where extremely high damping or light weight is necessary, a PVC base material, which has three times the damping capacity of bituminous pads, can be used but this material is rather difficult to attach to the panelling.

1.1.5 Collision safety (Fig. 1.8)

Car safety may broadly be divided into two kinds: Firstly the   , which is concerned with the car s road-holding stability while being driven, steered or braked and secondly the   ,

Fig. 1.8 Collision body safety

which depends upon body style and design structure to protect the occupants of the car from serious in ury in the event of a collision.

Car bodies can be considered to be made in three parts (Fig. 1.8) a central cell for the passengers of the welded bodywork integral with a rigid platform, acting as a floor pan, and chassis with various box-section cross- and side-members. This type of structure provides a reinforced rigid crush-proof construction to resist deformation on impact and to give the interior a high degree of protection. The extension of the engine and boot compartments at the front and rear of the central passenger cell are designed to form zones which collapse and crumble progressively over the short duration of a collision impact. Therefore, the kinetic energy due to the car s initial speed will be absorbed fore and aft primarily by strain and plastic energy within the crumble zones with very little impact energy actually being dissipated by the central body cell.

1.1.6 Body and chassis alignment chec s
(Fig. 1.9)
Body and chassis alignment checks will be necessary if the vehicle has been involved in a ma or

ab 1.1 Summary of function and application of soundproofing materials

Function	Acoustic materials	Application
nsulation	oaded C bitumen with or without foam or fibres base mineral wool	Floor bulkhead dash panel
amping	Bitumen or mineral wool	oors side panels underside of roof
Absorption	olyurethane foam mineral wool or bonded fibres	Side panels underside of roof engine compartment bonnet

collision, but overall alignment may also be necessary if the vehicle s steering and ride characteristics do not respond to the expected standard of a similar vehicle when being driven.

Structural misalignment may be caused by all sorts of reasons, for example, if the vehicle has been continuously driven over rough ground at high speed, hitting an obstacle in the road, mounting steep pavements or kerbs, sliding off the road into a ditch or receiving a glancing blow from some other vehicle or obstacle etc. Suspicion that something is wrong with the body or chassis alignment is focused if there is excessively uneven or high tyre wear, the vehicle tends to wander or pull over to one side and yet the track and suspension geometry appears to be correct.

Alignment checks should be made on a level, clear floor with the vehicle s tyres correctly inflated to normal pressure. A plumb bob is required in the form of a stubby cylindrical bar conical shaped at one end, the other end being attached to a length of thin cord. atum reference points are chosen such as the centre of a spring eye on the chassis mounting point, transverse wishbone and trailing arm pivot centres, which are attachment points to the underframe or chassis, and body cross-member to side-member attachment centres and subframe bolt-on points (Fig. 1.9).

Initially the cord with the plumb bob hanging from its end is lowered from the centre of each reference point to the floor and the plumb bob contact point with the ground is marked with a chalked cross. Transverse and diagonal lines between reference points can be made by chalking the full length of a piece of cord, holding it taut between reference centres on the floor and getting somebody to pluck the centre of the line so that it rebounds and leaves a chalked line on the floor.

A reference longitudinal centre line may be made with a strip of wood baton of length ust greater than the width between ad acent reference marks on the floor. A nail is punched through one end and this is placed over one of the reference marks. A piece of chalk is then held at the tip of the free end and the whole wood strip is rotated about the nailed end. The chalk will then scribe an arc between ad acent reference points. This is repeated from the other side. At the points where these two arcs intersect a straight line is made with a plucked, chalked cord running down the middle of the vehicle. This procedure should be followed at each end of the vehicle as shown in Fig. 1.9.

nce all the reference points and transverse and diagonal oining lines have been drawn on the
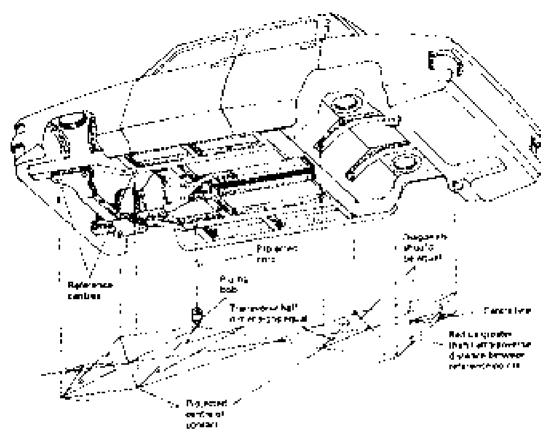
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Fig. 1.9 Body underframe alignment checks

floor, a rule or tape is used to measure the distances between centres both transversely and diagonally. These values are then chalked along their respective lines. Misalignment or error is observed when a pair of transverse or diagonal dimensions differ and further investigation will thus be necessary.

Note that transverse and longitudinal dimensions are normally available from the manufacturer s manual and differences between paired diagonals indicates lozenging of the framework due to some form of abnormal impact which has previously occurred.

1.2 Engine, transmission and body structure mountings

1. .1   nherent engine vibrations
The vibrations originating within the engine are caused by both the cyclic acceleration of the reciprocating components and the rapidly changing cylinder gas pressure which occurs throughout each cycle of operation.

Both the variations of inertia and gas pressure forces generate three kinds of vibrations which are transferred to the cylinder block:

1 Vertical and/or horizontal shake and rock
2 Fluctuating torque reaction
3 Torsional oscillation of the crankshaft

1. .   easons for fle ible mountings
It is the ob ective of flexible mounting design to cope with the many requirements, some having conflicting constraints on each other. A list of the duties of these mounts is as follows:

1 To prevent the fatigue failure of the engine and gearbox support points which would occur if they were rigidly attached to the chassis or body structure.
2 To reduce the amplitude of any engine vibration which is being transmitted to the body structure.
3 To reduce noise amplification which would occur if engine vibration were allowed to be transferred directly to the body structure.

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4 To reduce human discomfort and fatigue by partially isolating the engine vibrations from the body by means of an elastic media.

5 To accommodate engine block misalignment and to reduce residual stresses imposed on the engine block and mounting brackets due to chassis or body frame distortion.

6 To prevent road wheel shocks when driving over rough ground imparting excessive rebound movement to the engine.

7 To prevent large engine to body relative movement due to torque reaction forces, particularly in low gear, which would cause excessive misalignment and strain on such components as the exhaust pipe and silencer system.

8 To restrict engine movement in the fore and aft direction of the vehicle due to the inertia of the engine acting in opposition to the accelerating and braking forces.

1. .3 ubber fle ible mountings (Figs 1.10, 1.11 and 1.12)

A rectangular block bonded between two metal plates may be loaded in compression by squeezing the plates together or by applying parallel but opposing forces to each metal plate. n compression, the rubber tends to bulge out centrally from the sides and in shear to form a parallelogram (Fig. 1.10(a)).

To increase the compressive stiffness of the rubber without greatly altering the shear stiffness, an interleaf spacer plate may be bonded in between the top and bottom plate (Fig. 1.10(b)). This interleaf plate prevents the internal outward collapse of the rubber, shown by the large bulge around the sides of the block, when no support is provided, whereas with the interleaf a pair of much smaller bulges are observed.
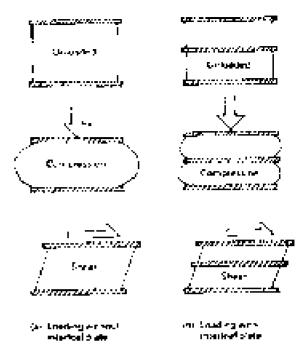

Fig. 1.10 (a and b) odes of loading rubber blocks

When two rubber blocks are inclined to each other to form a V mounting, see Fig. 1.11, the rubber will be loaded in both compression and shear shown by the triangle of forces. The magnitude of compressive force will be given by W_c and the much smaller shear force by W_S. This produces a resultant reaction force W_R. The larger the wedge angle Θ, the greater the proportion of compressive load relative to the shear load the rubber block absorbs.

The distorted rubber provides support under light vertical static loads approximately equal in both compression and shear modes, but with heavier loads the proportion of compressive stiffness
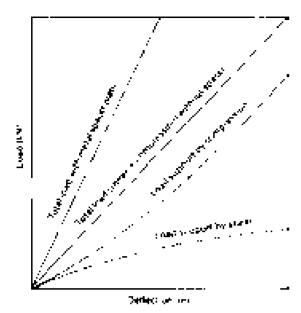

Fig. 1.11 rubber block mounting

Fig. 1.12 oad–deflection curves for rubber block

to that of shear stiffness increases at a much faster rate (Fig. 1.12). It should also be observed that the combined compressive and shear loading of the rubber increases in direct proportion to the static deflection and hence produces a straight line graph.

1. . A is of oscillation (Fig. 1.13)

The engine and gearbox must be suspended so that it permits the greatest degree of freedom when oscillating around an imaginary centre of rotation known as the principal axis. This principal axis produces the least resistance to engine and gearbox sway due to their masses being uniformly distributed about this axis. The engine can be considered to oscillate around an axis which passes through the centre of gravity of both the engine and gearbox (Figs 1.13(a, b and c)). This normally produces an axis of oscillation inclined at about 10–20 to the crankshaft axis. To obtain the greatest degree of freedom, the mounts must be arranged so that they offer the least resistance to shear within the rubber mounting.

1. .5 Si modes of freedom of a suspended body (Fig. 1.14)

If the movement of a flexible mounted engine is completely unrestricted it may have six modes of vibration. Any motion may be resolved into three linear movements parallel to the axes which pass through the centre of gravity of the engine but at right angles to each other and three rotations about these axes (Fig. 1.14).

These modes of movement may be summarized as follows:

inear motions	Rotational motions
1 orizontal longitudinal	4 Roll
2 orizontal lateral	5 Pitch
3 Vertical	6 aw

1. .6 ositioning of engine and gearbo mountings (Fig. 1.15)

If the mountings are placed underneath the combined engine and gearbox unit, the centre of gravity is well above the supports so that a lateral (side) force acting through its centre of gravity, such as experienced when driving round a corner, will cause the mass to roll (Fig. 1.15(a)). This condition is undesirable and can be avoided by placing the mounts on brackets so that they are in the same plane as the centre of gravity (Fig. 1.15(b)). Thus the mounts provide flexible opposition to any side force which might exist without creating a roll couple. This is known as a   condition.

An alternative method of making the natural modes of oscillation independent or uncoupled is achieved by arranging the supports in an inclined V position (Fig. 1.15(c)). Ideally the aim is to make the compressive axes of the mountings meet at the centre of gravity, but due to the weight of the power unit distorting the rubber springing the inter-section lines would meet slightly below this point. Therefore, the mountings are tilted so that the compressive axes converge at some focal point above the centre of gravity so that the actual lines of action of the mountings, that is, the direction of the resultant forces they exert, converge on the centre of gravity (Fig. 1.15(d)).

The compressive stiffness of the inclined mounts can be increased by inserting interleafs between the rubber blocks and, as can be seen in Fig. 1.15(e), the line of action of the mounts converges at a lower point than mounts which do not have interleaf support.

Engine and gearbox mounting supports are normally of the three or four point configuration. Petrol engines generally adopt the three point support layout which has two forward mounts (Fig. 1.13(a and c)), one inclined on either side of the engine so that their line of action converges on the principal axis, while the rear mount is supported centrally at the rear of the gearbox in approximately the same plane as the principal axis. arge diesel engines tend to prefer the four point support
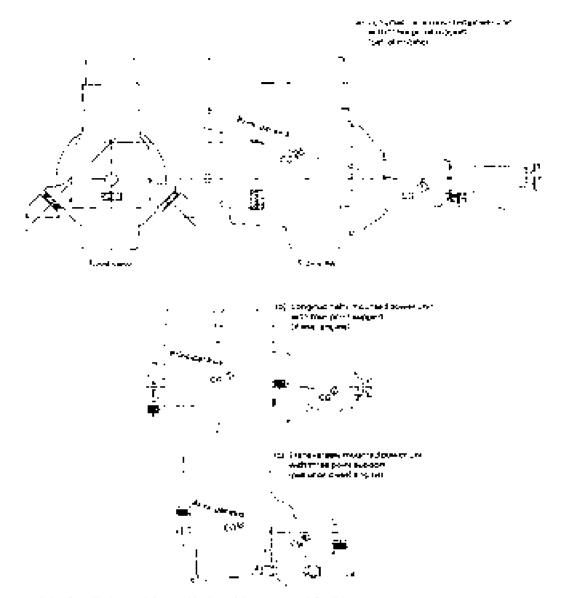
Fig. 1.13 Axis of oscillation and the positioning of the power unit flexible mounts

arrangement where there are two mounts either side of the engine (Fig. 1.13(b)). The two front mounts are inclined so that their lines of action pass through the principal axis, but the rear mounts which are located either side of the clutch bell housing are not inclined since they are already at principal axis level.

1. .7   *ngine and transmission vibrations*

atural fre uency of vibration (Fig. 1.16)   A sprung body when deflected and released will bounce up and down at a uniform rate. The amplitude of this cyclic movement will progressively decrease and the number of oscillations per minute of the rubber mounting is known as its   .

There is a relationship between the static deflection imposed on the rubber mount springing by the suspended mass and the rubber s natural frequency of vibration, which may be given by
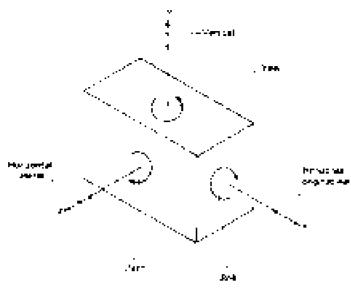
$$_0 = \frac{30}{}$$

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Fig. 1.14 Six modes of freedom for a suspended block

where $_0$ = natural frequency of vibration (vib/min)
= static deflection of the rubber (m)

This relationship between static deflection and natural frequency may be seen in Fig. 1.16.

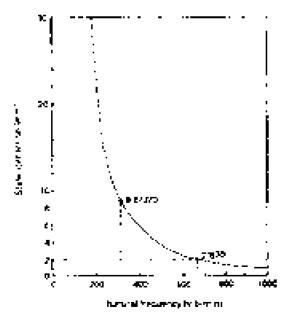**Resonance** Resonance is the unwanted synchronization of the disturbing force frequency imposed by the engine out of balance forces and the fluctuating cylinder gas pressure and the natural frequency of oscillation of the elastic rubber support mounting, i.e. resonance occurs when

$$\frac{}{_0} = 1$$

where = disturbing frequency
$_0$ = natural frequency

Transmissibility (Fig. 1.17) When the designer selects the type of flexible mounting the Theory of Transmissibility can be used to estimate critical resonance conditions so that they can be either prevented or at least avoided.

Transmissibility ( ) may be defined as the ratio of the transmitted force or amplitude which passes through the rubber mount to the chassis to that of the externally imposed force or amplitude generated by the engine:

$$= \frac{F_t}{F_d} = \frac{1}{1 \left(\dfrac{}{_0}\right)^2}$$

where F_t = transmitted force or amplitude
F_d = imposed disturbing force or amplitude

This relationship between transmissibility and the ratio of disturbing frequency and natural frequency may be seen in Fig. 1.17.
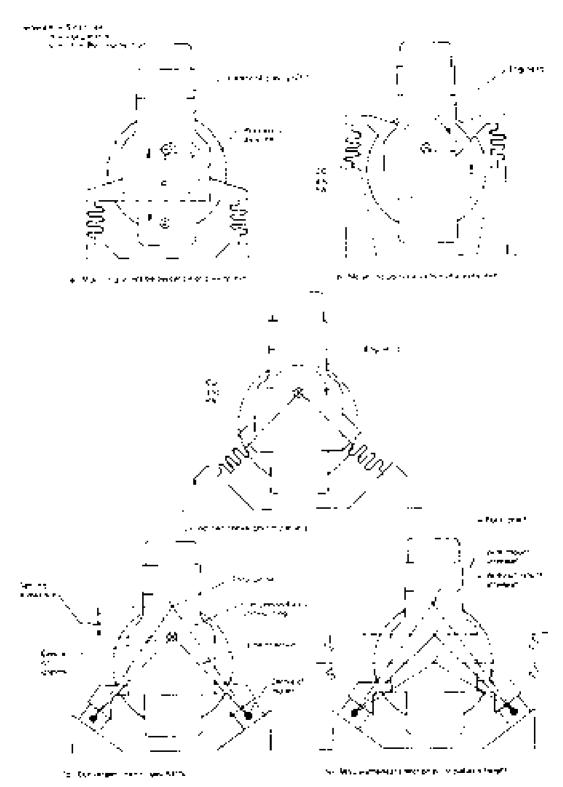

Fig. 1.16 Relationship of static deflection and natural frequency

Fig. 1.15 (a–) Coupled and uncoupled mounting points

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The transmissibility to frequency ratio graph (Fig. 1.17) can be considered in three parts as follows:

This is the resonance range and should be avoided. It occurs when the disturbing frequency is very near to the natural frequency. If steel mounts are used, a critical vibration at resonance would go to infinity, but natural rubber limits the transmissibility to around 10. If Butyl synthetic rubber is adopted, its damping properties reduce the peak transmissibility to about $2\frac{1}{2}$. Unfortunately, high damping rubber compounds such as Butyl rubber are temperature sensitive to both damping and dynamic stiffness so that during cold weather a noticeably harsher suspension of the engine results.

amping of the engine suspension mounting is necessary to reduce the excessive movement of a flexible mounting when passing through resonance, but at speeds above resonance more vibration is transmitted to the chassis or body structure than would occur if no damping was provided.

This is the recommended working range where the ratio of the disturbing frequency to that of the natural frequency of vibration of the rubber mountings is greater than $1\frac{1}{2}$ and the transmissibility is less than one. Under these conditions off-peak partial resonance vibrations passing to the body structure will be minimized.

This is known as the shock reduction range and only occurs when the disturbing frequency is lower than the natural frequency. enerally it is only experienced with very soft rubber mounts and when the engine is initially cranked for starting purposes and so quickly passes through this frequency ratio region.

An engine oscillates vertically on its flexible rubber mountings with a frequency of 800 vibrations per minute (vpm). With the information provided answer the following questions:

a) From the static deflection–frequency graph, Fig. 1.16, or by formula, determine the natural frequency of vibration when the static deflection of the engine is 2 mm and then find the disturbing to natural frequency ratio. Comment on these results.
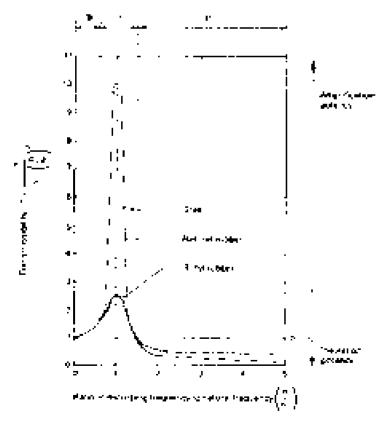b) If the disturbing to natural frequency ratio is increased to 2.5 determine the natural frequency

Fig. 1.17 Relationship of transmissibility and the ratio of disturbing and natural fre uencies for natural rubber Butyl rubber and steel

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of vibration and the new static deflection of the engine. Comment of these conditions.

a) $\quad f_0 = \frac{30}{\sqrt{x}} = \frac{30}{\sqrt{0.002}}$

$\qquad = \frac{30}{0.04472} = 670.84 \text{ vib/min}$

$\therefore \frac{n}{f_0} = \frac{800}{670.84} = 1.193$

The ratio 1.193 is very near to the resonance condition and should be avoided by using softer mounts.

b) $\quad \frac{n}{f_0} = \frac{800}{f_0} = 2.5$

$\therefore f_0 = \frac{800}{2.5} = 320 \text{ vib/min}$

Now $f_0 = \frac{30}{\sqrt{x}}$

thus $\sqrt{x} = \frac{30}{f_0}$

$\therefore x = \left(\frac{30}{f_0}\right)^2 = \left(\frac{30}{320}\right)^2$

$= 0.008789 \text{ m or } 8.789 \text{ mm}$

A low natural frequency of 320 vib/min is well within the insulation range, therefore from either the deflection–frequency graph or by formula the corresponding rubber deflection necessary is 8.789 mm when the engine s static weight bears down on the mounts.

1. . ngine to body/chassis mountings

Engine mountings are normally arranged to provide a degree of flexibility in the horizontal longitudinal, horizontal lateral and vertical axis of rotation. At the same time they must have sufficient stiffness to provide stability under shock loads which may come from the vehicle travelling over rough roads. Rubber sprung mountings suitably positioned fulfil the following functions:

1 Rotational flexibility around the horizontal longitudinal axis which is necessary to allow the impulsive inertia and gas pressure components of the engine torque to be absorbed by rolling of the engine about the centre of gravity.
2 Rotational flexibility around both the horizontal lateral and the vertical axis to accommodate any horizontal and vertical shake and rock caused by unbalanced reciprocating forces and couples.

1. . Subframe to body mountings
(Figs 1.6 and 1.19)

ne of many problems with integral body design is the prevention of vibrations induced by the engine, transmission and road wheels from being transmitted through the structure. Some manufacturers adopt a subframe (Fig. 1.6(a, b and c)) attached by resilient mountings (Fig. 1.19(a and b)) to the body to which the suspension assemblies, and in some instances the engine and transmission, are attached. The mass of the subframes alone helps to damp vibrations. It also simplifies production on the assembly line, and facilitates subsequent overhaul or repairs.

In general, the mountings are positioned so that they allow strictly limited movement of the subframe in some directions but provide greater freedom in others. For instance, too much lateral freedom of a subframe for a front suspension assembly would introduce a degree of instability into the steering, whereas some freedom in vertical and longitudinal directions would improve the quality of a ride.

1. .10 Types of rubber fle ible mountings
A survey of typical rubber mountings used for power units, transmissions, cabs and subframes are described and illustrated as follows:

ouble shear paired sandwich mounting (Fig. 1.18(a)) Rubber blocks are bonded between the aws of a U shaped steel plate and a flat interleaf plate so that a double shear elastic reaction takes place when the mount is sub ected to vertical loading. This type of shear mounting provides a large degree of flexibility in the upright direction and thus rotational freedom for the engine unit about its principal axis. It has been adopted for both engine and transmission suspension mounting points for medium-sized diesel engines.

ouble inclined wedge mounting (Fig. 1.18(b)) The inclined wedge angle pushes the bonded rubber blocks downwards and outwards against the bent-up sides of the lower steel plate when loaded in the vertical plane. The rubber blocks are sub ected to both shear and compressive loads and the proportion of compressive to shear load becomes greater with vertical deflection. This form of mounting is suitable for single point gearbox supports.

nclined interleaf rectangular sandwich mounting (Fig. 1.18(c)) These rectangular blocks are
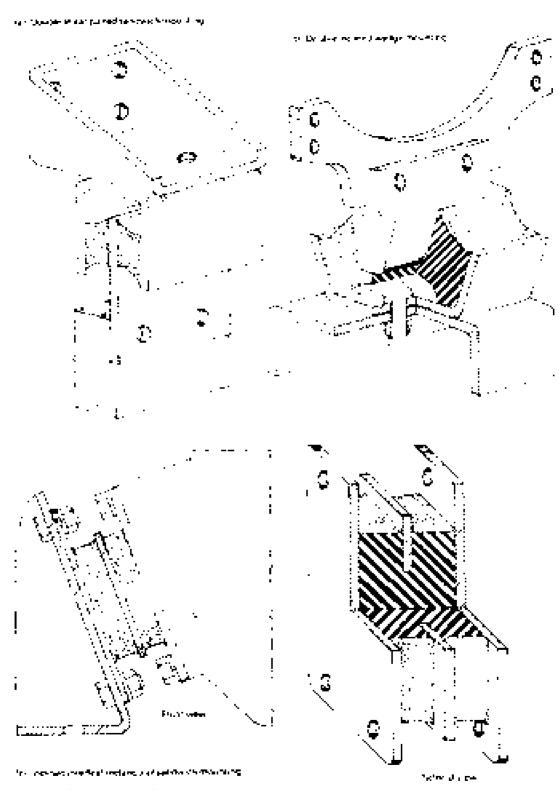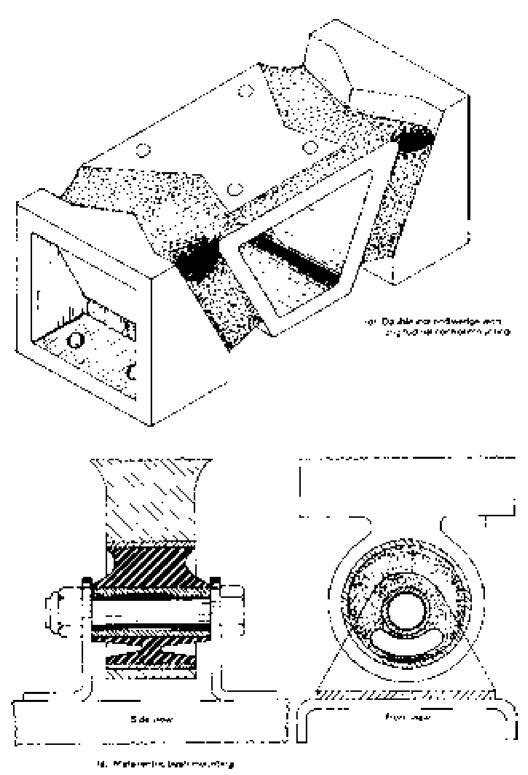
Fig. 1.18 (a–) Types of rubber flexible mountings

(c) Double row radial wedge seat ring hydraulic compaction plug

(d) Intermediate bush mounting
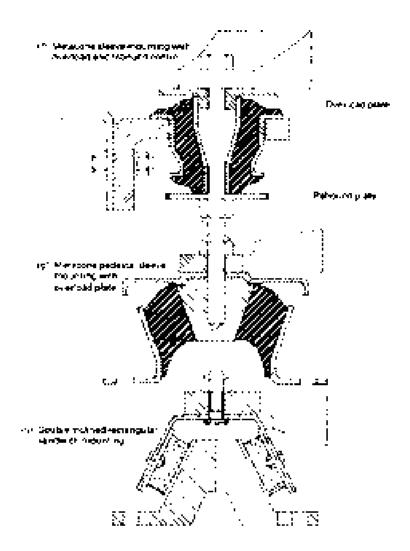
Fig. 1.18 *contd*

Fig. 1.18 *contd*

designed to be used with convergent V formation engine suspension system where the blocks are inclined on either side of the engine. This configuration enables the rubber to be loaded in both shear and compression with the ma ority of engine rotational flexibility being carried out in shear. Vertical deflection due to body pitch when accelerating or braking is absorbed mostly in compression. Vertical elastic stiffness may be increased without greatly effecting engine roll flexibility by having metal spacer interleafs bonded into the rubber.

ouble inclined wedge with longitudinal control mounting (Fig. 1.18(d)) Where heavy vertical loads and large rotational reactions are to be absorbed, double inclined wedge mounts positioned on either side of the power unit s bell housing at principal axis level may be used. ongitudinal movement is restricted by the double V formed between the inner and two outer members seen in a plan view. This V and wedge configuration provides a combined shear and compressive strain to the rubber when there is a relative fore and aft movement between the engine and chassis, in addition to that created by the vertical loading of the mount.

This mounting s ma or application is for the rear mountings forming part of a four point suspension for heavy diesel engines.

Meta entric bush mounting (Fig. 1.18(e)) When the bush is in the unloaded state, the steel inner sleeve is eccentric relative to the outer one so that

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there is more rubber on one side of it than on the other. Precompression is applied to the rubber expanding the inner sleeve. The bush is set so that the greatest thickness of rubber is in compression in the laden condition. A slot is incorporated in the rubber on either side where the rubber is at its minimum in such a position as to avoid stressing any part of it in tension.

When installed, its stiffness in the fore and aft direction is greater than in the vertical direction, the ratio being about 2.5 : 1. This type of bush provides a large amount of vertical deflection with very little fore and aft movement which makes it suitable for rear gearbox mounts using three point power unit suspension and leaf spring eye shackle pin bushes.

Metacone sleeve mountings (Fig. 1.18(f and g)) These mounts are formed from male and female conical sleeves, the inner male member being centrally positioned by rubber occupying the space between both surfaces (Fig. 1.18(f)). uring vertical vibrational deflection, the rubber between the sleeves is sub ected to a combined shear and compression which progressively increases the stiffness of the rubber as it moves towards full distortion. The exposed rubber at either end overlaps the flanged outer sleeve and there is an upper and lower plate bolted rigidly to the ends of the inner sleeve. These plates act as both overload (bump) and rebound stops, so that when the inner member deflects up or down towards the end of its movement it rapidly stiffens due to the surplus rubber being squeezed in between. Mounts of this kind are used where stiffness is needed in the horizontal direction with comparative freedom of movement for vertical deflection.

An alternative version of the Metacone mount uses a solid aluminium central cone with a flanged pedestal conical outer steel sleeve which can be bolted directly onto the chassis side member, see Fig. 1.18(g). An overload plate is clamped between the inner cone and mount support arm, but no rebound plate is considered necessary.

These mountings are used for suspension applications such as engine to chassis, cab to chassis, bus body and tanker tanks to chassis.

ouble inclined rectangular sandwich mounting (Fig. 1.18(h)) A pair of rectangular sandwich rubber blocks are supported on the slopes of a triangular pedestal. A bridging plate merges the resilience of the inclined rubber blocks so that they provide a combined shear and compressive

distortion within the rubber. Under small deflection conditions the shear and compression is almost equal, but as the load and thus deflection increases, the proportion of compression over the shear loading predominates.
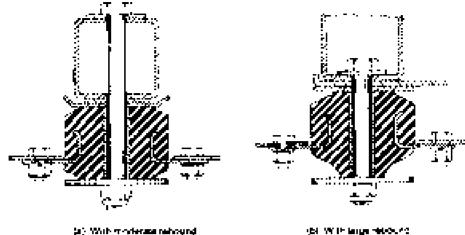
These mounts provide very good lateral stability without impairing vertical deflection flexibility and progressive stiffness control. When used for road wheel axle suspension mountings, they offer good insulation against road and other noises.
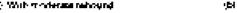
langed sleeve bobbin mounting with rebound control (Fig. 1.19(a and b)) These mountings have the rubber moulded partially around the outer flange sleeve and in between this sleeve and an inner tube. A central bolt attaches the inner tube to the body structure while the outer member is bolted on two sides to the subframe.
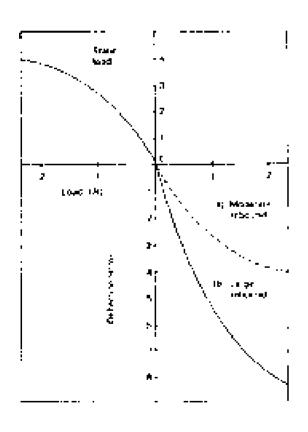
When loaded in the vertical downward direction, the rubber between the sleeve and tube walls will be in shear and the rubber on the outside of the flanged sleeve will be in compression.

There is very little relative sideway movement between the flanged sleeve and inner tube due to rubber distortion. An overload plate limits the downward deflection and rebound is controlled by the lower plate and the amount and shape of rubber trapped between it and the underside of the flanged sleeve. A reduction of rubber between the flanged sleeve and lower plate (Fig. 1.19(a)) reduces the rebound, but an increase in depth of rubber increases rebound (Fig. 1.19(b)). The load deflection characteristics are given for both mounts in Fig. 1.19c. These mountings are used extensively for body to subframe and cab to chassis mounting points.

ydroelastic engine mountings (Figs 1.20(a–c) and 1.21) A flanged steel pressing houses and supports an upper and lower rubber spring diaphragm. The space between both diaphragms is filled and sealed with fluid and is divided in two by a separator plate and small transfer holes interlink the fluid occupying these chambers (Fig. 1.20(a and b)). Under vertical vibratory conditions the fluid will be displaced from one chamber to the other through transfer holes. uring downward deflection (Fig. 1.20(b)), both rubber diaphragms are sub ected to a combined shear and compressive action and some of the fluid in the upper chamber will be pushed into the lower and back again by way of the transfer holes when the rubber rebounds (Fig. 1.20(a)). For low vertical vibratory frequencies,

(a) With moderate rebound

(b) With large rebound

(c) Relation between load and deflection for a flanged sleeve bobbin mounting

Fig. 1.19 (a–c) Flanged sleeve bobbin mounting with rebound control

the movement of fluid between the chambers is unrestricted, but as the vibratory frequencies increase, the transfer holes offer increasing resistance to the flow of fluid and so slow down the up and down motion of the engine support arm. This damps and reduces the amplitude of mountings vertical vibratory movement over a number of cycles. A comparison of conventional rubber and hydroelastic damping resistance over the normal operating frequency range for engine mountings is shown in Fig. 1.20(c).
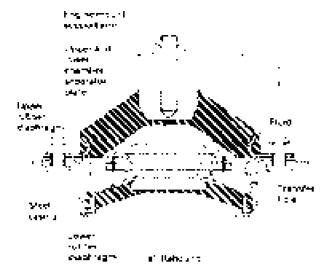
Instead of adopting a combined rubber mount with integral hydraulic damping, separate diagonally mounted telescopic dampers may be used in conjunction with inclined rubber mounts to reduce both vertical and horizontal vibration (Fig. 1.21).
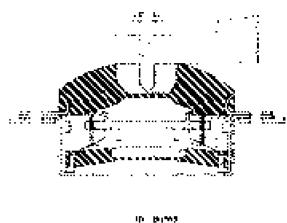
1.3 Fifth wheel coupling assembly
(Fig. 1.22(a and b))

The fifth wheel coupling attaches the semi-trailer to the tractor unit. This coupling consists of a semi-circular table plate with a central hole and a vee section cut-out towards the rear (Fig. 1.22(b)). Attached underneath this plate are a pair of pivoting coupling jaws (Fig. 1.22(a)). The semi-trailer has an upper fifth wheel plate welded or bolted to the underside of its chassis at the front and in the centre of this plate is bolted a kingpin which faces downwards (Fig. 1.22(a)).

When the trailer is coupled to the tractor unit, this upper plate rests and is supported on top of the tractor fifth wheel table plate with the two halves of the coupling jaws engaging the kingpin. To permit

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(a) (Rubber)

(b) (Fluid)

(c) Relationship of frequency of vibration and damping resistance with conventional rubber and hydroelastic input mountings

Fig. 1.20 (a–c) Hydroelastic engine mount

relative swivelling between the kingpin and jaws, the two interfaces of the tractor fifth wheel tables and trailer upper plate should be heavily greased. Thus, although the trailer articulates about the kingpin, its load is carried by the tractor table.

Flexible articulation between the tractor and semi-trailer in the horizontal plane is achieved by permitting the fifth wheel table to pivot on horizontal trunnion bearings that lie in the same vertical plane as the kingpin, but with their axes at right angles to that of the tractor s wheel base (Fig. 1.22(b)). Rubber trunnion rubber bushes normally provide longitudinal oscillations of about 10 .

The fifth wheel table assembly is made from either a machined cast or forged steel sections, or from heavy section rolled steel fabrications, and the upper fifth wheel plate is generally hot rolled steel welded to the trailer chassis. The coupling locking system consisting of the jaws, pawl, pivot pins and kingpin is produced from forged high carbon manganese steels and the pressure areas of these components are induction hardened to withstand shock loading and wear.

1.3.1 Operation of the fifth wheel coupling
(Fig. 1.23(a–d))
With the trailer kingpin uncoupled, the jaws will be in their closed position with the plunger withdrawn from the lock gap between the rear of the jaws, which are maintained in this position by the pawl contacting the hold-off stop (Fig. 1.23(a)). When coupling the tractor to the trailer, the jaws of the
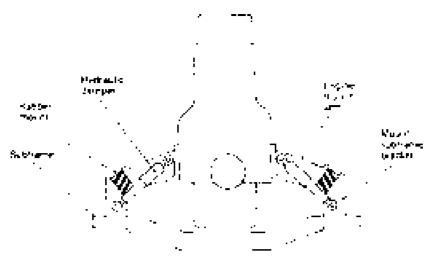
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Fig. 1.21   iagonally mounted hydraulic dampers suppress both vertical and hori ontal vibrations

fifth wheel strike the kingpin of the trailer. The aws are then forced open and the kingpin enters the space between the aws (Fig. 1.23(b)). The kingpin contacts the rear of the aws which then automatically pushes them together. At the same time, one of the coupler aws causes the trip pin to strike the pawl. The pawl turns on its pivot against the force of the spring, releasing the plunger, allowing it to be forced into the aws lock gap by its spring (Fig. 1.23(c)). When the tractor is moving, the drag of the kingpin increases the lateral force of the aws on the plunger.

To disconnect the coupling, the release hand lever is pulled fully back (Fig. 1.23(d)). This draws the plunger clear of the rear of the aws and, at the same time, allows the pawl to swing round so that it engages a pro ection hold-off stop situated at the upper end of the plunger, thus amming the plunger in the fully out position in readiness for uncoupling.

1.3.   peration of single a   and pa l coupling
(Fig. 1.24(a–d))
With the trailer kingpin uncoupled, the aw will be held open by the pawl in readiness for coupling (Fig. 1.24(a)). When coupling the tractor to the trailer, the aw of the fifth wheel strikes the kingpin of the trailer and swivels the aw about its pivot pin against the return spring, slightly pushing out the pawl (Fig. 1.24(b)). Further rearward movement of the tractor towards the trailer will swing the aw round until it traps and encloses the kingpin. The

spring load notched pawl will then snap over the aw pro ection to lock the kingpin in the coupling position (Fig. 1.24(c)). The securing pin should then be inserted through the pull lever and table eye holes. When the tractor is driving forward, the reaction on the kingpin increases the locking force between the aw pro ection and the notched pawl.

To disconnect the coupling, lift out the securing pin and pull the release hand lever fully out (Fig. 1.24(d)). With both the tractor and trailer stationary, the ma ority of the locking force applied to notched pawl will be removed so that with very little effort, the pawl is able to swing clear of the aw in readiness for uncoupling, that is, by ust driving the tractor away from the trailer. Thus the aw will simply swivel allowing the kingpin to pull out and away from the aw.

1.4 Trailer and caravan drawbar couplings

1. .1   ye and bolt dra bar coupling for heavy goods trailers (Figs 1.25 and 1.26)
rawbar trailers are normally hitched to the truck by means of an A frame drawbar which is coupled by means of a towing eye formed on the end of the drawbar (Fig. 1.25). When coupled, the towing eye hole is aligned with the vertical holes in the upper and lower aws of the truck coupling and an eye bolt passes through both coupling aws and drawbar eye to complete the attachment (Fig. 1.26).

ateral drawbar swing is permitted owing to the eye bolt pivoting action and the slots between the
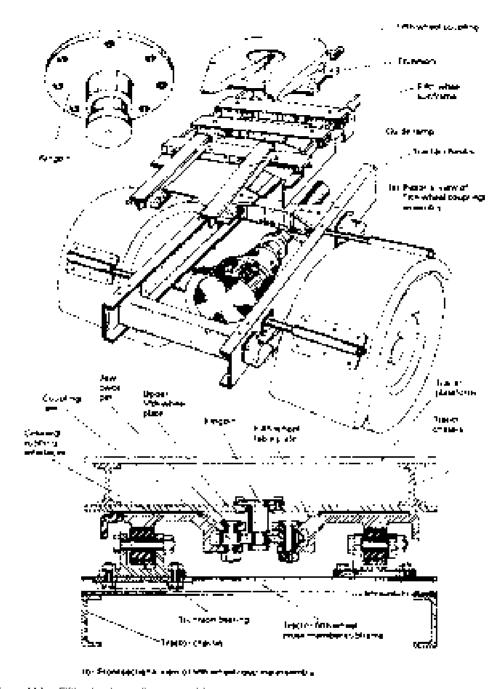
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Fig. 1.22 (a and b)   Fifth wheel coupling assembly

aws on either side. Aligning the towing eye to the aws is made easier by the converging upper and lower lips of the aws which guide the towing eye as the truck is reversed and the aws approach the drawbar. Isolating the coupling aws from the truck draw beam are two rubber blocks which act as a damping media between the towing vehicle and trailer. These rubber blocks also permit additional deflection of the coupling aw shaft relative to the draw beam under rough abnormal operating conditions, thus preventing over-straining the drawbar and chassis system.
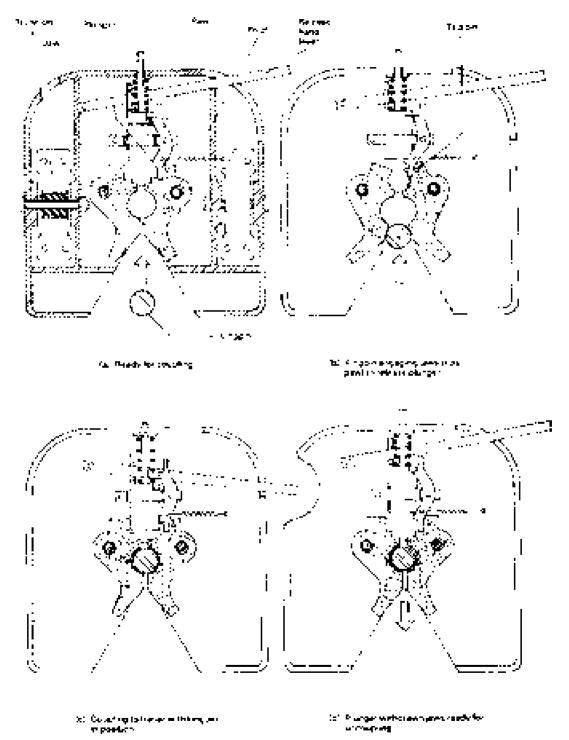
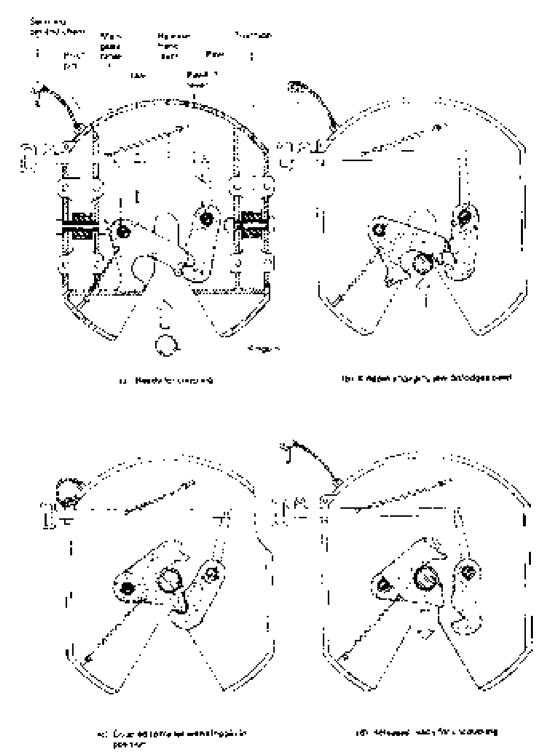**Fig. 1.23 (a–d)** Fifth wheel coupling with twin jaws plunger and pawl

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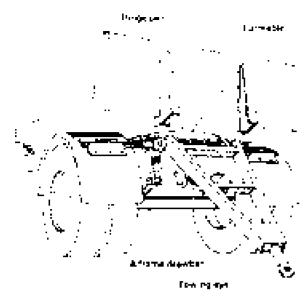**Fig. 1.24 (a–d)** Fifth wheel coupling with single aw and pawl

Fig. 1.25 Drawbar trailer

The coupling jaws, eye bolt and towing eye are generally made from forged manganese steel with induction hardened pressure areas to increase the wear resistance.

Operation of the automatic drawbar coupling (Fig. 1.26) In the uncoupled position the eyebolt is held in the open position ready for coupling (Fig. 1.26(a)). When the truck is reversed, the jaws of the coupling slip over the towing eye and in the process strike the conical lower end of the eye bolt (Fig. 1.26(b)). Subsequently, the eye bolt will lift. This trips the spring-loaded wedge lever which now rotates clockwise so that it bears down on the eye bolt. Further inward movement of the eye bolt between the coupling jaws aligns the towing eye with the eye bolt. The spring pressure now acts through the wedge lever to push the eye bolt through the towing eye and the lower coupling jaw (Fig. 1.26(c)). When the eye bolt stop-plate has been fully lowered by the spring tension, the wedge lever will slot into its groove formed in the centre of the eye bolt so that it locks the eye bolt in the coupled position.

To uncouple the drawbar, the handle is pulled upwards against the tension of the coil spring mounted on the wedge level operating shaft (Fig. 1.26(d)). This unlocks the wedge, freeing the eyebolt and then raises the eye bolt to the uncoupled position where the wedge lever jams it in the open position (Fig. 1.26(a)).

1.9 Ball and socket towing bar coupling for light caravan/trailers (Fig. 1.27)
Light trailers or caravans are usually attached to the rear of the towing car by means of a ball and socket type coupling. The ball part of the attachment is bolted onto a bracing bracket fitted directly to the boot pan or the towing load may be shared out between two side brackets attached to the rear longitudinal box-section members of the body.

A single channel section or pair of triangularly arranged angle-section arms may be used to form the towbar which both supports and draws the trailer.

Attached to the end of the towbar is the socket housing with an internally formed spherical cavity. This fits over the ball member of the coupling so that it forms a pivot joint which can operate in both the horizontal and vertical plane (Fig. 1.27).

To secure the socket over the ball, a lock device must be incorporated which enables the coupling to be readily connected or disconnected. This lock may take the form of a spring-loaded horizontally positioned wedge with a groove formed across its top face which slips underneath and against the ball. The wedge is held in the closed engaged position by a spring-loaded vertical plunger which has a horizontal groove cut on one side. An uncoupling lever engages the plunger's groove so that when the coupling is disconnected the lever is squeezed to lift and release the plunger from the wedge. At the same time the whole towbar is raised by the handle to clear the socket and from the ball member.

Coupling the tow bar to the car simply reverses the process, the uncoupling lever is again squeezed against the handle to withdraw the plunger and the socket housing is pushed down over the ball member. The wedge moves outwards and allows the ball to enter the socket and immediately the wedge springs back into the engaged position. Releasing the lever and handle completes the coupling by permitting the plunger to enter the wedge lock groove.

Sometimes a strong compression spring is interposed between the socket housing member and the towing (draw) bar to cushion the shock load when the car/trailer combination is initially driven away from a standstill.

1.10 Semi-trailer landing gear (Fig. 1.28)
Landing legs are used to support the front of the semi-trailer when the tractor unit is uncoupled.

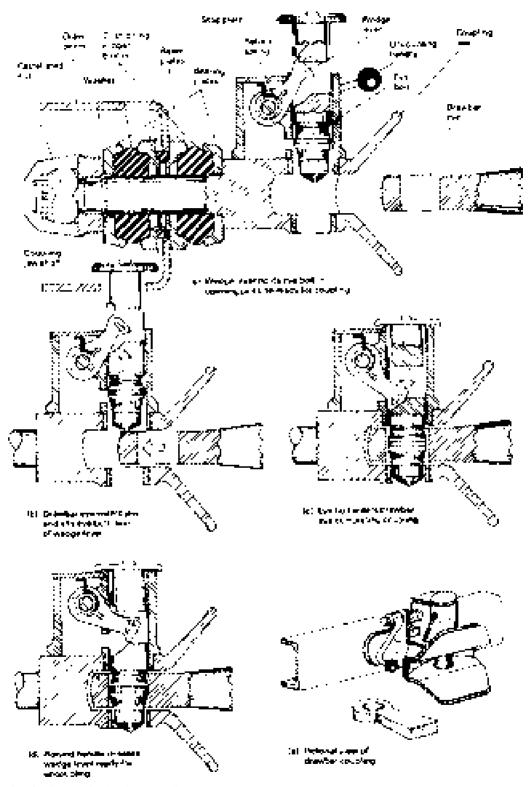Extendable landing legs are bolted vertically to each chassis side-member behind the rear wheels of

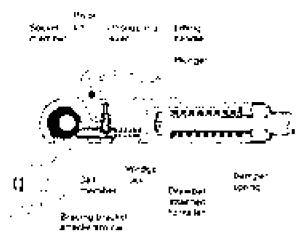**Fig. 1.26 (a–)** Automatic drawbar coupling

Fig. 1.27 Ball and socket caravan/trailer towing attachment

the tractor unit, ust sufficiently back to clear the rear tractor road wheels when the trailer is coupled and the combination is being manoeuvred (Fig. 1.28(a)). To provide additional support for the legs, bracing stays are attached between the legs and from the legs diagonally to the chassis cross-member (Fig. 1.28(b)).

The legs consist of inner and outer high tensile steel tubes of square section. A ackscrew with a bevel wheel attached at its top end supported by the outer leg horizontal plate in a bronze bush bearing. The awscrew fits into a nut which is mounted at the top of the inner leg and a taper roller bearing race is placed underneath the outer leg horizontal support plate and the upper part of the ackscrew to minimize friction when the screw is rotated (Fig. 1.28(b)). The bottom ends of the inner legs may support either twin wheels, which enable the trailer to be manoeuvred, or simply flat feet. The latter are able to spread the load and so permit greater load capacity.

To extend or retract the inner legs, a winding handle is attached to either the low or high speed shaft protruding from the side of the gearbox. The upper high speed shaft supports a bevel pinion which meshes with a vertically mounted bevel wheel forming part of the ackscrew.

Rotating the upper shaft imparts motion directly to the ackscrew through the bevel gears. If greater leverage is required to raise or lower the front of the trailer, the lower shaft is engaged and rotated. This provides a gear reduction through a compound gear train to the upper shaft which then drives the bevel pinion and wheel and hence the ackscrew.

1. utomatic chassis lubrication system

1.6.1 The need for automatic lubrication system
(Fig. 1.29)

wing to the heavy loads they carry commercial vehicles still prefer to use metal to metal oints which are externally lubricated. Such oints are kingpins and bushes, shackle pins and bushes, steering ball oints, fifth wheel coupling, parking brake linkage etc. (Fig. 1.29). These oints require lubricating in proportion to the amount of relative movement and the loads exerted. If lubrication is to be effective in reducing wear between the moving parts, fresh oil must be pumped between the oints frequently. This can best be achieved by incorporating an automatic lubrication system which pumps oil to the bearing s surfaces in accordance to the distance travelled by the vehicle.
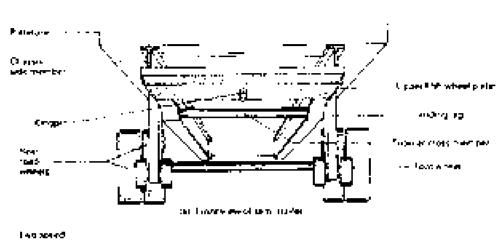
1.6. Description of airdromic automatic chassis lubrication system (Fig. 1.30)

This lubrication system comprises four ma or components a combined pump assembly, a power unit, an oil unloader valve and an air control unit.

Pump assembly (Fig. 1.30) The pump assembly consists of a circular housing containing a ratchet operated drive (cam) shaft upon which are mounted one, two or three single lobe cams (only one cam shown). Each cam operates a row of 20 pumping units disposed radially around the pump casing, the units being connected to the chassis bearings by nylon tubing.

Power unit (Fig. 1.30) This unit comprises a cylinder and spring-loaded air operated piston which is mounted on the front face of the pump assembly housing, the piston rod being connected indirectly to the drive shaft ratchet wheel by way of a ratchet housing and pawl.

il unloader valve (Fig. 1.30) This consists of a shuttle valve mounted on the front of the pump assembly housing. The oil unloader valve allows air pressure to flow to the power unit for the power stroke. uring the exhaust stroke, however, when air flow is reversed and the shuttle valve is lifted from its seat, any oil in the line between the power unit and the oil unloader valve is then discharged to atmosphere.
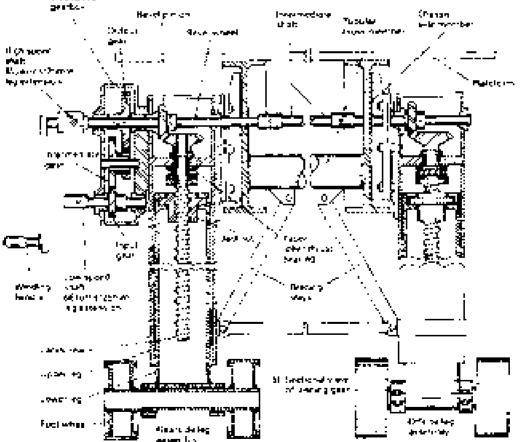
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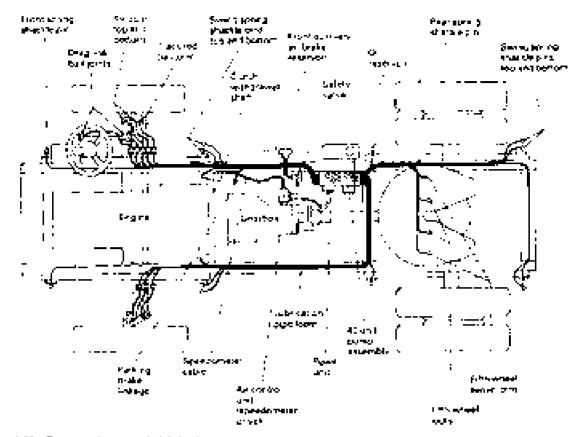**Fig. 1.28 (a and b)**   Semi-trailer landing gear

Fig. 1.29 Tractor unit automatic lubrication system

ir control unit (Fig. 1.30) This unit is mounted on the gearbox and is driven via the speedometer take-off point. It consists of a worm and wheel drive which operates an air proportioning control unit. This air proportioning unit is operated by a single lift face cam which actuates two poppet valves, one controlling air supply to the power unit, the other controlling the exhaust air from the power unit.

1.6.3 *peration of airdromic automatic chassis lubrication system* (Fig. 1.30)

Air from the air brake auxiliary reservoir passes by way of the safety valve to the air control (proportioning) unit inlet valve. Whilst the inlet valve is held open by the continuously rotating face cam lobe, air pressure is supplied via the oil unloader valve to the power unit attached to the multipump assembly housing. The power unit cylinder is supported by a pivot to the pump assembly casing, whilst the piston is linked to the ratchet and pawl

housing. Because the pawl meshes with one of the ratchet teeth and the ratchet wheel forms part of the camshaft, air pressure in the power cylinder will partially rotate both the ratchet and pawl housing and the camshaft clockwise. The cam (or cams) are in contact with one or more pump unit, and so each partial rotation contributes to a proportion of the erk plunger and barrel pumping cycle of each unit (Fig. 1.30).

As the control unit face cam continues to rotate, the inlet poppet inlet valve is closed and the exhaust poppet valve opens. Compressed air in the air control unit and above the oil control shuttle valve will now escape through the air control unit exhaust port to the atmosphere. Consequently the compressed air underneath the oil unloader shuttle valve will be able to lift it and any trapped air and oil in the power cylinder will now be released via the hole under the exhaust port. The power unit piston will be returned to its innermost position by the spring and in doing so will rotate the ratchet and pawl housing anti-clockwise. The pawl is thus
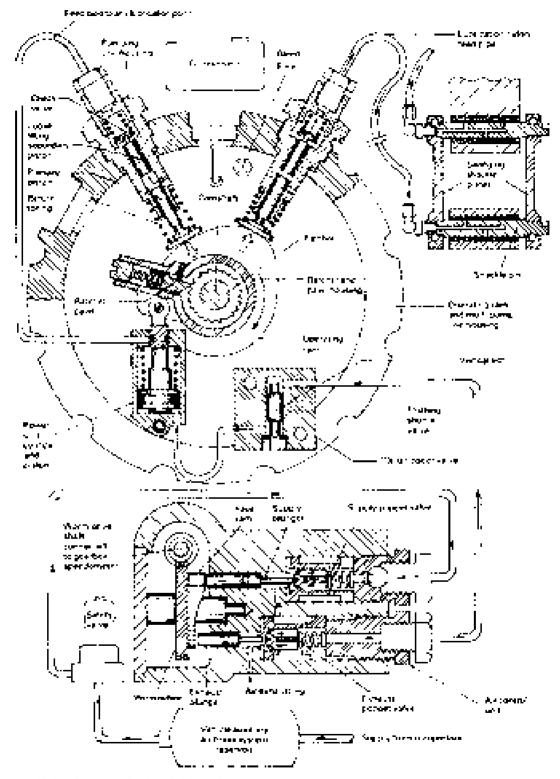
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Fig. 1.30 Airdromic automatic chassis lubrication system

able to slip over one or more of the ratchet teeth to take up a new position. The net result of the power cylinder being charged and discharged with compressed air is a slow but progressive rotation of the camshaft (Fig. 1.30).

A typical worm drive shaft to distance travelled relationship is 500 revolutions per 1 km. For 900 worm drive shaft revolutions the pumping cam revolves once. Therefore, every chassis lubrication point will receive one shot of lubricant in this distance.

When the individual lubrication pump unit s primary plunger is in its outermost position, oil surrounding the barrel will enter the inlet port, filling the space between the two plungers. As the cam rotates and the lobe lifts the primary plunger, it cuts off the inlet port. Further plunger rise will partially push out the secondary plunger and so open the check valve. Pressurised oil will then pass between the loose fitting secondary plunger and barrel to lubricate the chassis moving part it services (Fig. 1.30).

2   Friction clutch

2.1   Clutch fundamentals

Clutches are designed to engage and disengage the transmission system from the engine when a vehicle is being driven away from a standstill and when the gearbox gear changes are necessary. The gradual increase in the transfer of engine torque to the transmission must be smooth. Once the vehicle is in motion, separation and take-up of the drive for gear selection must be carried out rapidly without any fierceness, snatch or shock.

2.1.1   Driven plate inertia

To enable the clutch to be operated effectively, the driven plate must be as light as possible so that when the clutch is disengaged, it will have the minimum of spin, i.e. very little flywheel effect. Spin prevention is of the utmost importance if the various pairs of dog teeth of the gearbox gears, be they constant mesh or synchromesh, are to align in the shortest time without causing excessive pressure, wear and noise between the initial chamfer of the dog teeth during the engagement phase.

Smoothness of clutch engagement may be achieved by building into the driven plate some sort of cushioning device, which will be discussed later in the chapter, whilst rapid slowing down of the driven plate is obtained by keeping the diameter, centre of gravity and weight of the driven plate to the minimum for a given torque carrying capacity.

2.1.2   Driven plate transmitted torque capacity

The torque capacity of a friction clutch can be raised by increasing the coefficient of friction of the rubbing materials, the diameter and/or the spring thrust sandwiching the driven plate. The friction lining materials now available limit the coefficient of friction to something of the order of 0.35. There are materials which have higher coefficient of friction values, but these tend to be unstable and to snatch during take-up. Increasing the diameter of the driven plate unfortunately raises its inertia, its tendency to continue spinning when the driven plate is freed while the clutch is in the disengaged position, and there is also a limit to the clamping pressure to which the friction lining material may be subjected if it is to maintain its friction properties over a long period of time.

2.1.3   Multi pairs of rubbing surfaces (Fig. 2.1)

An alternative approach to raising the transmitted torque capacity of the clutch is to increase the number of pairs of rubbing surfaces. Theoretically the torque capacity of a clutch is directly proportional to the number of pairs of surfaces for a given clamping load. Thus the conventional single driven plate has two pairs of friction faces so that a twin or triple driven plate clutch for the same spring thrust would ideally have twice or three times the torque transmitting capacity respectively of that of the single driven plate unit (Fig. 2.1). However, because it is very difficult to dissipate the extra heat generated in a clutch unit, a larger safety factor is necessary per driven plate so that the torque capacity is generally only of the order 80% per pair of surfaces relative to the single driven plate clutch.

2.1.4   Driven plate wear (Fig. 2.1)

Lining life is also improved by increasing the number of pairs of rubbing surfaces because wear is directly related to the energy dissipation per unit area of contact surface. Ideally, by doubling the surface area as in a twin plate clutch, the energy input per unit lining area will be halved for a given slip time which would result in a 50% decrease in facing wear. In practice, however, this rarely occurs (Fig. 2.1) as the wear rate is also greatly influenced by the peak surface rubbing temperature and the intermediate plate of a twin plate clutch operates at a higher working temperature than either the flywheel or pressure plate which can be more effectively cooled. Thus in a twin plate clutch, half the energy generated whilst slipping must be absorbed by the intermediate plate and only a quarter each by the flywheel and pressure plate. This is usually borne out by the appearance of the intermediate plate and its corresponding lining faces showing evidence of high temperatures and increased wear compared to the linings facing the flywheel and pressure plate. Nevertheless, multiplate clutches do have a life expectancy which is more or less related to the number of pairs of friction faces for a given diameter of clutch.

For heavy duty applications such as those required for large trucks, twin driven plates are used, while for high performance cars where very
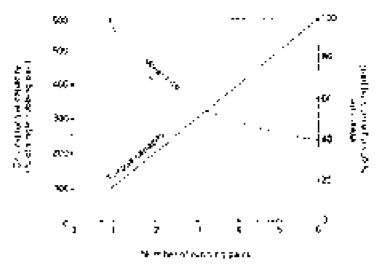
Fig. 2.1 Relationship of torque capacity wear rate and pairs of rubbing faces for multiplate clutch

rapid gear changes are necessary and large amounts of power are to be developed, small diameter multiplate clutches are preferred.

2.2 Angular driven plate cushioning and torsional damping (Figs 2.2–2.8)

2.2.1 Axial driven plate friction lining cushioning (Figs 2.2, 2.3 and 2.4)

In its simplest form the driven plate consists of a central splined hub. Mounted on this hub is a thin steel disc which in turn supports, by means of a ring of rivets, both halves of the annular friction linings (Figs 2.2 and 2.3).

Axial cushioning between the friction lining faces may be achieved by forming a series of evenly spaced T slots around the outer rim of the disc. This then divides the rim into a number of segments (Arcuate) (Fig. 2.4(a)). A horseshoe shape is further punched out of each segment. The central portion or blade of each horseshoe is given a permanent set to one side and consecutive segments have opposite sets so that every second segment is riveted to the same friction lining. The alternative set of these central blades formed by the horseshoe punch-out spreads the two half friction linings apart.

An improved version uses separately attached, very thin spring steel segments (borglite) (Fig. 2.4(b)), positioned end-on around a slightly thicker disc plate. These segments are provided with a wavy set so as to distance the two half annular friction linings.

Both forms of crimped spring steel segments situated between the friction linings provide
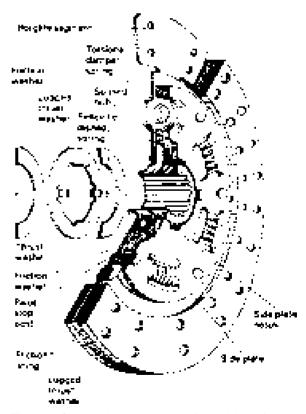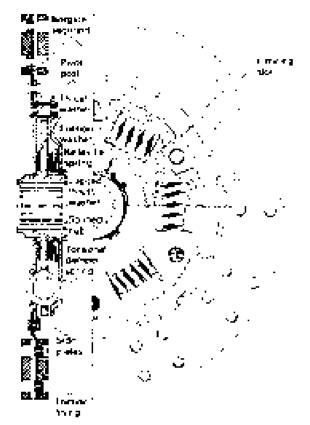

Fig. 2.2 Clutch driven centre plate (pictorial view)

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Fig. 2.3 Clutch driven centre plate (sectional view)

Fig. 2.4 (a and b) riven plate cushion take-up

progressive take-up over a greater pedal travel and prevent snatch. The separately attached spring segments are thinner than the segments formed out of the single piece driven plate, so that the squeeze take-up is generally softer and the spin inertia of the thinner segments is noticeably reduced.

A further benefit created by the spring segments ensures satisfactory bedding of the facing material and a more even distribution of the work load. In addition, cooling between the friction linings occurs when the clutch is disengaged which helps to sta-bilise the frictional properties of the face material.

The advantages of axial cushioning of the face linings provide the following:

a) Better clutch engagement control, allowing lower engine speeds to be used at take-up thus prolonging the life of the friction faces.
b) Improved distribution of the friction work over the lining faces reduces peak operating tempera-tures and prevents lining fade, with the resulting reduction in coefficient of friction and subse-quent clutch slip.
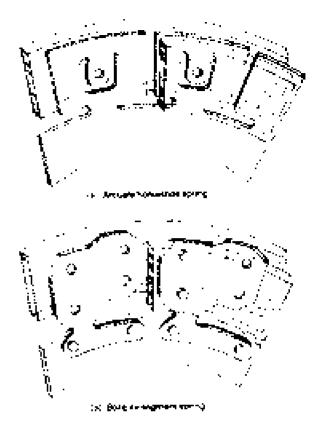
The spring take-up characteristics of the driven plate are such that when the clutch is initially engaged, the segments are progressively flattened so that the rate of increase in clamping load is provided by the rate of reaction offered by the spring segments (Fig. 2.5). This first low rate take-up period is followed by a second high rate engage-ment, caused by the effects of the pressure plate springs exerting their clamping thrust as they are allowed to expand against the pressure plate and so sandwich the friction lining between the flywheel and pressure plate faces.

. . *Torsional damping of driven plate*

rankshaft torsional vibration (Fig. 2.6) Engine crankshafts are sub ected to torsional wind-up and vibration at certain speeds due to the power impulses. Superimposed onto some steady mean rotational speed of the crankshaft will be additional fluctuating torques which will accelerate and decel-erate the crankshaft, particularly at the front pulley
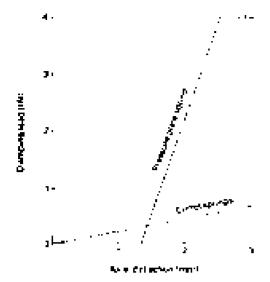
Fig. 2.5 Characteristics of driven plate axial clamping load to deflection take-up
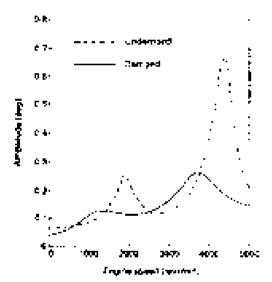

Fig. 2.6 Characteristics of crankshaft torsional vibrations undamped and damped

end and to a lesser extent the rear flywheel end (Fig. 2.6). If the flywheel end of the crankshaft were allowed to twist in one direction and then the other while rotating at certain critical speeds, the oscillating angular movements would take up the backlash between meshing gear teeth in the transmission system. Consequently, the teeth of the driving gears would be moving between the drive (pressure side) and non-drive tooth profiles of the driven gears. This would result in repeated shockloads imposed on

the gear teeth, wear, and noise in the form of gear clatter. To overcome the effects of crankshaft torsional vibrations a torsion damping device is normally incorporated within the driven plate hub assembly which will now be described and explained.

onstruction and operation of torsional damper springs (Figs 2.2, 2.3 and 2.7)   To transmit torque more smoothly and progressively during take-up of normal driving and to reduce torsional oscillations being transmitted from the crankshaft to the transmission, compressed springs are generally arranged circumferentially around the hub of the driven plate (Figs 2.2 and 2.3). These springs are inserted in elongated slots formed in both the flange of the splined hub and the side plates which enclose the hub s flange (Fig. 2.3). These side plates are riveted together by either three or six rivet posts which pass through the flanged hub limit slots. This thus provides a degree of relative angular movement between hub and side plates. The ends of the helical coil springs bear against both central hub flange and the side plates. Engine torque is therefore transmitted from the friction face linings and side plates through the springs to the hub flange, so that any fluctuation of torque will cause the springs to compress and rebound accordingly.

Multistage driven plate torsional spring dampers may be incorporated by using a range of different springs having various stiffnesses and spring location slots of different lengths to produce a variety of parabolic torsional load–deflection characteristics (Fig. 2.7) to suit specific vehicle applications.

The amount of torsional deflection necessary varies for each particular application. For example, with a front mounted engine and rear wheel drive vehicle, a moderate driven plate angular movement is necessary, say six degrees, since the normal transmission elastic wind-up is almost adequate, but with an integral engine, gearbox and final drive arrangement, the short transmission drive length necessitates considerably more relative angular deflection, say twelve degrees, within the driven plate hub assembly to produce the same quality of take-up.

onstruction and operation of torsional damper washers (Figs 2.2, 2.3 and 2.8)   The torsional energy created by the oscillating crankshaft is partially absorbed and damped by the friction washer clutch situated on either side of the hub flange (Figs 2.2 and 2.3). Axial damping load is achieved by a Belleville dished washer spring mounted between one of the side plates and a four lug thrust washer.
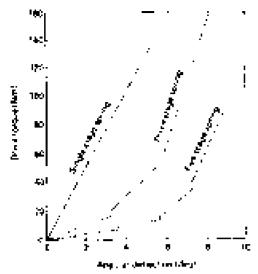
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Fig. 2.7 Characteristics of driven plate torsional spring tor ues to deflection take-up

Fig. 2.8 Characteristics of driven plate torsional damper thrust spring

The outer diameter of this dished spring presses against the side plate and the inner diameter pushes onto the lugged thrust washer. In its free state the Belleville spring is conical in shape but when assembled it is compressed almost flat. As the friction washers wear, the dished spring cone angle increases. This exerts a greater axial thrust, but since the distance between the side plate and lugged thrust washer has increased, the resultant clamping thrust remains almost constant (Fig. 2.8).

2.3   lutch friction materials

Clutch friction linings or buttons are sub ected to severe rubbing and generation of heat for relatively short periods. Therefore it is desirable that they have a combination of these properties:

a) Relatively high coefficient of friction under operating conditions,
b) capability of maintaining friction properties over its working life,
c) relatively high energy absorption capacity for short periods,
d) capability of withstanding high pressure plate compressive loads,
e) capability of withstanding bursts of centrifugal force when gear changing,
f) adequate shear strength to transmit engine torque,
g) high level of cyclic working endurance without the deterioration in friction properties,
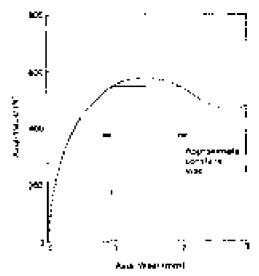h) good compatibility with cast iron facings over the normal operating temperature range,

i) a high degree of interface contamination tolerance without affecting its friction take-up and grip characteristics.

.3.1 Asbestos based linings (Figs 2.2 and 2.3)

 enerally, clutch driven plate asbestos-based linings are of the woven variety. These woven linings are made from asbestos fibre spun around lengths of brass or zinc wire to make lengths of threads which are both heat resistant and strong. The woven cloth can be processed in one of two ways:

a) The fibre wire thread is woven into a cloth and pressed out into discs of the required diameter, followed by stitching several of these discs together to obtain the desired thickness. The resultant disc is then dipped into resin to bond the woven asbestos threads together.
b) The asbestos fibre wire is woven in three dimensions in the form of a disc to obtain in a single stage the desired thickness. It is then pressed into shape and bonded together by again dipping it into a resin solution. Finally, the rigid lining is machined and drilled ready for riveting to the driven plate.

 evelopment in weaving techniques has, in certain cases, eliminated the use of wire coring so that asbestos woven lining may be offered as either non- or semi-metallic to match a variety of working conditions.

Asbestos is a condensate produced by the solidification of rock masses which cool at differential

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rates. When the moisture content of one layer is transferred to another, fibres are produced on solidification from which, as a result of high compression, these brittle, practically straight and exceptionally fine needle-like threads are made.

uring processing, these break down further with a diameter of less than 0.003 mm. They exhibit a length/thickness ratio of at least three to one. It is these fine fibres which can readily be inhaled into the lungs which are so dangerous to health.

The normal highest working temperature below which these asbestos linings will operate satisfactorily giving uniform coefficient of friction between 0.32 and 0.38 and a reasonable life span is about 260 C. Most manufacturers of asbestos-based linings quote a maximum temperature (something like 360 C) beyond which the lining, if operated continuously or very frequently, will suffer damage, with consequent alteration to its friction characteristics and deterioration in wear resistance.

.3.   Asbestos substitute friction material
(Figs 2.2 and 2.3)

The   uPont Company has developed a friction material derived from aromatic polyamide fibres belonging to the nylon family of polymers and it has been given the trade name   evlar aramid.

The operating properties relative to asbestos based linings are as follows:

1   igh endurance performance over its normal working pressure and temperature range.
2 It is lighter in weight than asbestos material therefore a reduction in driven plate spin shortens the time required for gear changing.
3   ood take-up characteristics, particularly with vehicles which were in the past prone to grab.
4 Weight for weight   evlar has five times the tensile strength of steel.
5   ood centrifugal strength to withstand lining disintegration as a result of sudden acceleration which may occur during the changing of gears.
6 Stable rubbing properties at high operating temperatures. It is not until a temperature of 425 C is reached that it begins to break down and then it does not simply become soft and melt, but steadily changes to carbon, the disintegration process being completed at about 500 C.

evlar exists in two states firstly as a 0.12 mm thick endless longitudinal fibre, which has a cut length varying between 6 and 100 mm, and secondly in the form of an amorphous structure of crushed and ground fibre known as   . In either form

these fibres are difficult to inhale because of their shape and size.

.3.3   etallic friction materials
Metallic and semi-metallic facings have been only moderately successful. The metallic linings are normally made from either sintered iron or copper-based sintered bronze and the semi-metallic facings from a mixture of organic and metallic materials. Metallic lining materials are made from a powder produced by crushing metal or alloy pieces into many small particles. They are then compressed and heated in moulds until sufficient adhesion and densification takes place. This process is referred to as   . The metallic rings are then ground flat and are then riveted back to back onto the driven plate.

enerally the metallic and semi-metallic linings have a higher coefficient of friction, can operate at higher working temperatures, have greater torque capacity and have extended life compared to that of the organic asbestos based linings. The ma or disadvantages of metallic materials are their relatively high inertia, making it difficult to obtain rapid gear changes high quality flywheel and pressure plate. Cast iron must be used to match their friction characteristics and these facings are more expensive than organic materials.

.3.   Cerametallic friction materials (Fig. 2.9)
Cerametallic button friction facings are becoming increasingly popular for heavy duty clutches. Instead of a full annular shaped lining, as with organic (asbestos or substitute) friction materials, four or six cerametallic trapezoidal-shaped buttons are evenly spaced on both sides around the driven plate. The cerametallic material is made from a powder consisting mainly of ceramic and copper. It is compressed into buttons and heated so that the copper melts and flows around each particle of solid ceramic. After solidification, the copper forms a strong metal-ceramic interface bond. These buttons are then riveted to the clutch driven plate and then finally ground flat.

The inherent advantages of these cerametallic-lined driven plates are:

1 A very low inertia (about 10% lower than the organic disc and 45% lower than a comparable sintered iron disc). Consequently it will result in quicker gear changes and, in the case of synchronized transmission, will increase synchronizer life.
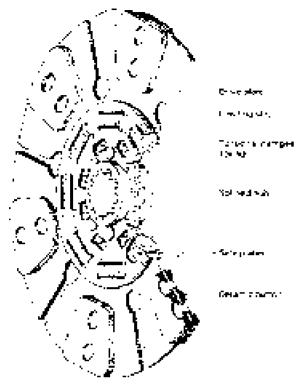2 A relatively high and stable coefficient of friction, providing an average value in the region of

Fig. 2.9 Clutch driven plate with ceramic facings

0.4, which increases the torque capacity of clutches using these driven plates.

3 The capability of operating at high working temperatures of up to 440 C for relatively long periods without showing signs of fade.

4 Button type driven plates expose more than 50% of the flywheel and pressure plate surfaces to the atmosphere during clutch engagement, so that heat transfer to the surrounding by convection may be improved by as much as 100%.

5 Button type friction pads do not suffer from warpage as do full ring metallic or organic linings and therefore are less prone to distort and cause clutch drag.

6 Button type friction pads permit the dust worn from the friction surfaces to be thrown clear of the clutch areas, thus preventing the possibility of any trapped work-hardened particles from scoring the friction faces.

7 Cerametallic materials are not as sensitive to grease and oil contamination as organic asbestos based linings.

8 The early ceramic-metallic friction buttons had a poor reputation as they tended to wear tracks in

flywheel and pressure plate facings. A prolonged development programme has virtually eliminated this problem and has considerably extended the driven plate life span compared to driven plates using organic (asbestos-based) annular disc linings.

2.4   lutch drive and driven member inspection

This inspection entails the examination of both the driven plate linings and the flywheel and pressure plate facings and will now be considered.

. .1 Driven plate lining face inspection

riven plate friction facings should, after a short period of service, give a polished appearance due to the frequent interface rubbing effect. This smooth and polished condition will provide the greatest friction grip, but it must not be confused with a glazed surface created by the formation of films of grease or oil worked into the rubbing surfaces, heated and oxidized.

A correctly bedded-in friction facing will appear highly polished through which the grain of the friction material can be clearly seen. When in perfect condition, these polished facings are of a grey or mid-brown colour. A very small amount of lubricant on the facings will burn off due to the generated heat. This will only slightly darken the facings, but providing polished facings remain so that the grain of the material can be clearly distinguished, it does not reduce its effectiveness.

arge amounts of lubricant gaining access to the friction surfaces may result in the following:

a) The burning of the grease or oil may leave a carbon deposit and a high glaze, this hides the grain of the material and is likely to cause clutch slip.

b) If the grease or oil is only partially burnt and leaves a resinous deposit on the facings it may result in a fierce clutch and may in addition produce clutch spin caused by the rubbing interfaces sticking.

c) If both carbon and resinous deposits are formed on the linings, clutch udder may develop during clutch take-up.

. .   ly heel and pressure plate facing inspection

Cast iron flywheel or pressure plate faces should have a smooth polished metallic appearance, but abnormal operating conditions or badly worn driven plate linings may be responsible for the following defects:

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a) verheated clutch friction faces can be identi-
fied by colouring of the swept polished tracks.
The actual surface temperatures reached can be
distinguished broadly by the colours straw,
brown, purple and blue which relate to 240 C,
260 C, 280 C and 320 C respectively.
b) Severe overheating will create thermal stresses
within the cast iron mass of the flywheel and
pressure plate, with the subsequent appearance
of radial hairline cracks.
c) Excessively worn driven plate linings with
exposed rivets and trapped work-hardened
dust particles will cause scoring of the rubbing
faces in the form of circular grooves.

2.   lutch misalignment (Fig. 2.10(a–d))
Clutch faults can sometimes be traced to mis-
alignment of the crankshaft to flywheel flange
 oint, flywheel housing and bell housing. Therefore,
if misalignment exists, the driven plate plane of
rotation will always be slightly skewed to that of
the restrained hub which is made to rotate about
the spigot shaft s axis. Misalignment is generally
responsible for the following faults:

1 Rapid wear on the splines of the clutch driven
plate hub, this being caused mainly by the tilted
hub applying uneven pressure over the interface
length of the splines.
2 The driven plate breaking away from the splined
hub due to the continuous cyclic flexing of the
plate relative to its hub.
3 Excessively worn pressure plate release mech-
anism, causing rough and uneven clutch
engagement.
4 Fierce chattering or dragging clutch resulting in
difficult gear changing.

If excessive clutch drag, backlash and poor
changes are evident and the faults cannot be
corrected, then the only remedy is to remove both
gearbox and clutch assembly so that the flywheel
housing alignment can be assessed (Fig. 2.10).

.5.1 Cran shaft end float (Fig. 2.10(a))
Before carrying out engine crankshaft, flywheel or
flywheel housing misalignment tests, ensure that
the crankshaft end float is within limits. (therwise
inaccurate run-out readings may be observed.)

To measure the crankshaft end float, mount the
magnetic dial gauge base to the back of the flywheel
housing and position the indicator pointer against
the crankshaft flanged end face. ero the dial gauge
and with the assistance of a suitable lever, force the
crankshaft back and forth and, at the same time,

observe the reading. Acceptable end float values
are normally between 0.08 and 0.30 mm.

.5.   Cran shaft fly heel flange runout
(Fig. 2.10(a))
The crankshaft flange flywheel oint face must be
perpendicular to its axis of rotation with no permis-
sible runout. To check for any misalignment, keep
the dial gauge assembly mounted as for the end
float check. ero gauge the dial and rotate the
crankshaft by hand for one complete revolution
whilst observing any dial movement. Investigate
further if runout exists.

.5.3 ly heel friction face and rim face runout
(Fig. 2.10(a and b))
When the flywheel is centred by the crankshaft axis,
it is essential that the flywheel friction face and rim
rotate perpendicularly to the crankshaft axis.

Mount the dial gauge magnetic base to the
engine flywheel housing. First set the indicator
pointer against the friction face of the flywheel
near the outer edge (Fig. 2.10(a and b)) and set
gauge to zero. Turn the flywheel one revolution
and observe the amount of variation. Secondly
reset indicator pointer against the flywheel rim
and repeat the test procedure (Fig. 2.10(b)). Max-
imum permissible runout in both tests is 0.02 mm
per 20 mm of flywheel radius. Thus with a 300 mm
diameter clutch fitted, maximum run-out would be
0.15 mm. Repeat both tests 2 or 3 times and com-
pare readings to eliminate test error.

.5.   ly heel housing runout (Fig. 2.10(c))
When the gearbox bell housing is centred by the
inside diameter and rear face of the engine flywheel
housing, it is essential that the inside diameter and
rear face of the housing should be concentric and
parallel respectively with the flywheel.

Mount the dial gauge magnetic base to the fly-
wheel friction face and position. Set the indicator
pointer against the face of the housing. Make sure
that the pointer is not in the path of the fixing holes
in the housing face or else incorrect readings may
result. ero the indicator and observe the reading
whilst the crankshaft is rotated one complete revolu-
tion. Reset the indicator pointer against the intern-
ally machined recess of the clutch housing and repeat
the test procedure. Maximum permissible runout is
0.20 mm. Repeat both tests two or three times and
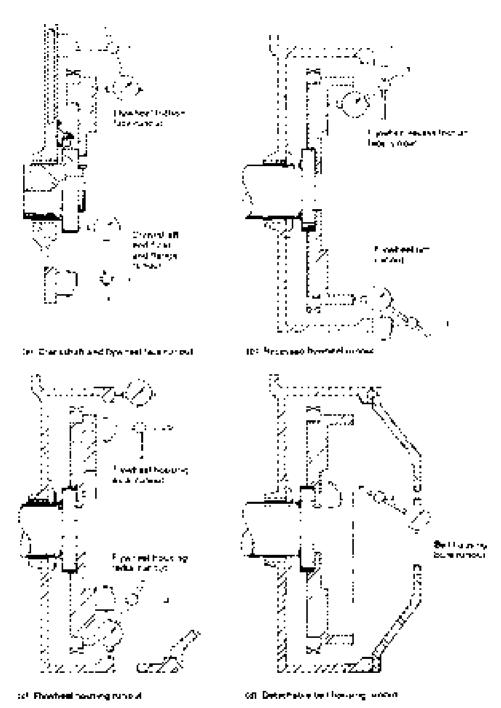compare readings to eliminate errors.

Fig. 2.10 (a–d) Crankshaft flywheel and clutch housing alignment

(Fig. 2.10(c and d))
When the gearbox bell housing is located by dowel pins instead of the inside diameter of the engine flywheel housing (Fig. 2.10(c)) (shouldered bell housing), it is advisable to separate the clutch bell housing from the gearbox and mount it to the flywheel housing for a concentric check.

Mount the dial gauge magnetic base onto the flywheel friction face and position the indicator pointer against the internal recess of the bell housing gearbox oint bore (Fig. 2.10(d)). Set the gauge to zero and turn the crankshaft by hand one complete revolution. At the same time, observe the dial gauge reading.

Maximum permissible runout should not exceed 0.25 mm.

2. Pull type diaphragm clutch (Fig. 2.11)
With this type of diaphragm clutch, the ma or components of the pressure plate assembly are a cast iron pressure plate, a spring steel diaphragm disc and a low carbon steel cover pressing (Fig. 2.11). To actuate the clutch release, the diaphragm is made to pivot between a pivot ring positioned inside the rear of the cover and a raised circumferential ridge formed on the back of the pressure plate. The diaphragm disc is divided into fingers caused by radial slits originating from the central hole. These fingers act both as leaf springs to provide the pressure plate thrust and as release levers to disengage the driven plate from the drive members.

When the driven and pressure plates are bolted to the flywheel, the diaphragm is distorted into a dished disc which therefore applies an axial thrust between the pressure plate and the cover pressing. This clutch design reverses the normal method of operation by pulling the diaphragm spring outwards to release the driven plate instead of pushing it.

wing to its configuration, the pull type clutch allows a larger pressure plate and diaphragm spring to be used for a given diameter of clutch. Advantages of this design over a similar push type clutch include lower pedal loads, higher torque capacity, improved take-up and increased durability. This clutch layout allows the ratio of the diaphragm finger release travel to pressure plate movement to be reduced. It is therefore possible to maintain the same pressure plate movement as that offered by a conventional push type clutch, and yet increase the ratio between clamp load and pedal load from 4:1 to 5:1.
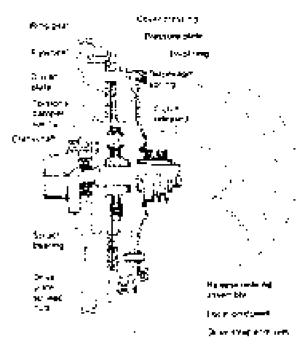

Fig. 2.11   iaphragm single plate pull type clutch

2.7 Multiplate diaphragm type clutch (Fig. 2.12)
These clutches basically consist of drive and driven plate members. The drive plates are restrained from rotating independently by interlocking lugs and slots which permit axial movement, but not relative rotational spin, whilst the driven plates are attached and supported by internally splined hubs to corresponding splines formed on the gearbox spigot shaft, see Fig. 2.12.

The diaphragm spring is in the form of a dished annular disc. The inner portion of the disc is radially slotted, the outer ends being enlarged with a circular hole to prevent stress concentration when the spring is distorted during disengagement. These radial slots divide the disc into a number of release levers (fingers).

The diaphragm spring is located in position with a shouldered pivot post which is riveted to the cover pressing. These rivets also hold a pair of fulcrum rings in position which are situated either side of the diaphragm.

Whilst in service, the diaphragm cone angle will change continuously as wear occurs and as the clutch is engaged and disengaged during operation. To enable this to happen, the diaphragm pivots and rolls about the fulcrum rings. When the clutch is engaged the diaphragm bears against the outer
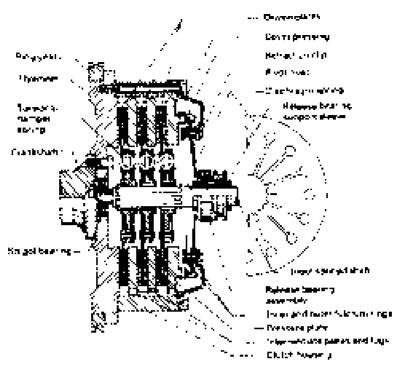
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Fig. 2.12 ultiplate diaphragm type clutch

ring, but when disengagement takes place the reaction load is then taken by the inner ring.

As the friction linings wear, the spring diaphragm will become more dished and subsequently will initially exert a larger axial clamping load. It is only when the linings are very worn, so that the distance between the cover pressing and pressure plate become excessive, that the axial thrust will begin to decline.

2. ipe rollway twin driven plate clutch (Fig. 2.13)
These clutches have two circular rows of helical coil springs which act directly between the pressure plate and the cover housing, see Fig. 2.13. The release mechanism is of the pull type in which a central release bearing assembly is made to withdraw (pull out) three release levers to disengage the clutch. The clutch pressure plate assembly is bolted to the flywheel and the driven plate friction linings are sandwiched between the flywheel, intermediate plate and pressure plate facings. The central hub of the driven plates is mounted on a splined gearbox spigot shaft (input shaft). The splined end of the input shaft is supported by a ball race bearing mounted inside the flywheel-crankshaft attachment flange. The other end of this shaft is supported inside the gearbox by either ball or taper roller

bearings. There are two types of pressure plate cover housings one with a deep extended cover rim which bolts onto a flat flywheel facing and the shallow cover type in which the pressure plate casting fits into a recessed flywheel.

The release mechanism is comprised of three lever fingers. The outer end of each lever pivots on a pin and needle race mounted inside each of the ad ustable eye bolt supports, which are attached to the cover housing through an internally and externally threaded sleeve which is secured to the cover housing with a lock nut. Inwards from the eye bolt, one-sixth of the release lever length, is a second pin which pivots on a pair of needle-bearing races situated inside the pressure plate lugs formed on either side of each layer.

elease lever ad ustment
Initially, setting up of the release levers is achieved by slackening the locknuts and then rotating each sleeve in turn with a two pronged fork adaptor tool which fits into corresponding slots machined out of the ad ustment sleeve end. Rotating the sleeve one way or the other will screw the eye bolts in or out until the correct dimension is obtained between the back of the release lever fingers and the outer cover rim edge. This setting dimension is provided by the
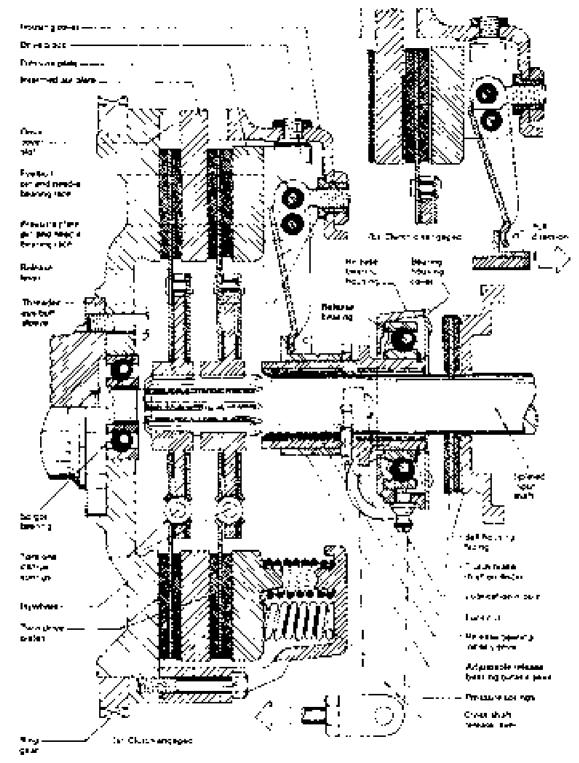
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Fig. 2.13 (a–b) Twin driven plate pull type clutch

manufacturers for each clutch model and engine application. Finally, tighten the locknuts of each eye bolt and re-check each lever dimension again.

elease bearing ad ustment
Slacken sleeve locknut with a C shaped spanner. Rotate the inner sleeve either way by means of the slotted ad usting nut until the recommended clearance is obtained between the bearing housing cover face and clutch brake.

i.e.   9.5 mm for 355 mm   1 P
         13 mm for 355 mm   2 P
         13 mm for 294 mm   2 P

Finally tighten sleeve locknut and re-check clearance.

2.9 Spicer twin driven plate angle spring pull type clutch (Fig. 2.14)
An interesting clutch engagement and release pressure plate mechanism employs three pairs of coil springs which are inclined to the axial direction of the driven plates. These springs are mounted between the pressure plate cover housing, which takes the spring reaction, and the release lever central hub (Fig. 2.14). The axial clamping thrust is conveyed by the springs to the six to one leverage ratio release levers (six of them) spaced evenly around the release hub. These release levers span between the release hub and a large annular shaped ad ustable pivot ring which is screwed inside the pressure plate cover housing. Towards the pivot pin end of the release levers a kink is formed so that it can bear against the pressure plate at one point. The pressure plate and intermediate plate are both prevented from spinning with the driven plates by cast-in drive lugs which fit into slots formed into the cover housing.

In the engaged position, the six springs expand and push the release hub and, subsequently, the release levers towards the pressure plate so that the driven plates are squeezed together to transmit the drive torque.

To release the clutch driven plates, the release bearing assembly is pulled out from the cover housing. This compels the release lever hub to compress and distort the thrust springs to a much greater inclined angle relative to the input shaft axis and so permits the pressure plate to be withdrawn by means of the retraction springs.

Because the spring thrust does not operate directly against the pressure plate, but is relayed through the release levers, the actual spring s stiffness is reduced by a factor of the leverage ratio in

this instance one-sixth of the value if the springs were direct acting.

The operating characteristics of the clutch mechanism are described as follows:

e engaged position (Fig. 2.14(a))
The spring thrust horizontal component of 2.2 kN, multiplied by the lever ratio, provides a pressure plate clamping load of 13.2 kN (Fig. 2.14(a)). The axial thrust horizontal component pushing on the pressure plate does not vary in direct proportion with the spring load exerted between its ends, but is a function of the angle through which the mounted springs operate relative to the splined input shaft.

orn engagement position (Fig. 2.14(b))
When the driven plate facings wear, the release bearing moves forward to the pressure plate so that the springs elongate. The spring load exerted between the spring ends is thus reduced. Fortunately, the inclined angle of spring axis to that of the thrust bearing axis is reduced so that as the spring load along its axis declines, the horizontal thrust component remains essentially the same. Therefore, the pressure plate clamping load remains practically constant throughout the life of the clutch (Fig. 2.14(b)).

elease position (Fig. 2.14(c))
When the clutch is released, that is when the bearing is pulled rearwards, the springs compress and increase in load, but the spring angle relative to the thrust bearing axis increases so that a greater proportion of the spring load will be acting radially instead of axially. Consequently, the horizontal component of axial release bearing load, caused by the spring thrust, gradually reduces to a value of about 1.7 kN as the bearing moves forwards, which results in the reduced pedal effort. This is shown in Fig. 2.14(c).

nternal manual ad ustment
Release bearing ad ustment is made by unscrewing the ring lock plate bolt and removing the plate. The clutch pedal is then held down to relieve the release levers and ad usting ring load. The ad usting ring is then rotated to screw it in or out so that it alters the release lever hub axial position.

Turning the ad usting ring clockwise moves the release bearing towards the gearbox (increasing free pedal movement). Conversely, turning the ad usting ring anticlockwise moves the release bearing towards the flywheel (decreasing free pedal movement).
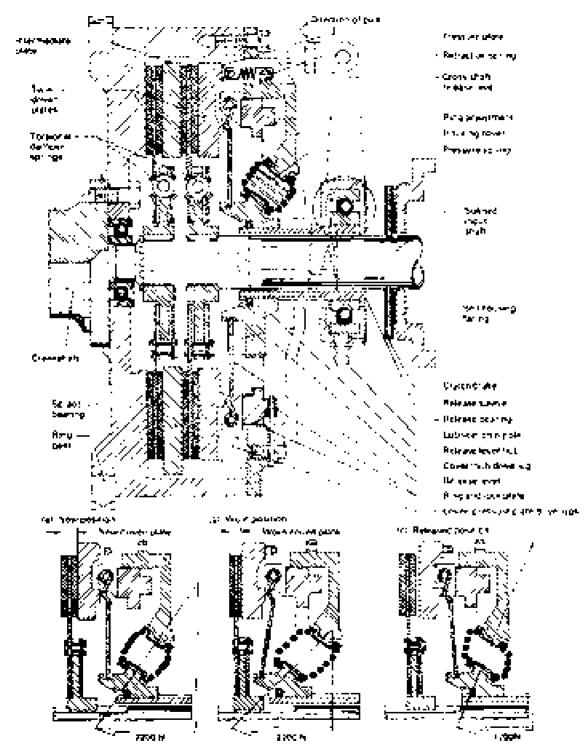
Fig. 2.14 (a–c) Twin driven plate angle spring pull type clutch

The ad usting ring outer face is notched so that it can be levered round with a screwdriver when ad ustment is necessary. The distance between each notch represents approximately 0.5 mm. Thus three notches moved means approximately 1.5 mm release bearing movement.

With the pedal released, there should be approximately 13 mm clearance between the release bearing face and clutch brake.

nternal self ad ustment

A clutch self-ad ustment version has teeth cut on the inside of the ad usting ring and a small worm and spring self-ad usting device replaces the lock plate. The worm meshes with the ad usting ring.

ne end of the spring is located in a hole formed in the release lever hub whilst the other end is in contact with the worm. Each time the clutch is engaged and disengaged, the release lever movement will actuate the spring. Consequently, once the driven plates have worn sufficiently, the increased release lever movement will rotate the worm which in turn will partially screw round the ad usting ring to compensate and so reset the position of the release levers.

2.1 lutch (upshift) brake (Fig. 2.15)

The clutch brake is designed primarily for use with unsynchronized (crash or constant mesh) gearboxes to permit shifting into first and reverse gear without severe dog teeth clash. In addition, the brake will facilitate making unshafts by slowing down the input shaft so that the next higher gear may be engaged without crunching of teeth.

The brake disc assembly consists of a pair of Belleville spring washers which are driven by a hub having internal lugs that engage machined slots in the input shaft. These washers react against the clutch brake cover with facing material positioned between each spring washer and outer cover (Fig. 2.15).

When the clutch pedal is fully depressed, the disc will be squeezed between the clutch release bearing housing and the gearbox bearing housing, causing the input spigot shaft to slow down or stop. The hub and spring washer combination will slip with respect to the cover if the applied torque load exceeds 34 Nm, thus preventing the disc brake being overloaded.

In general, the clutch brake comes into engagement only during the last 25 mm of clutch pedal
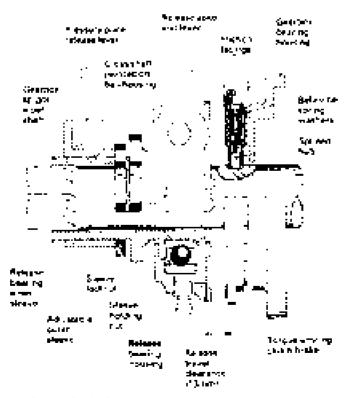

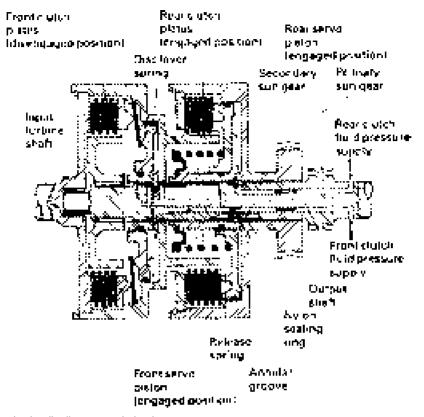**Fig. 2.15** Clutch upshift brake (tor ue limiting)

Fig. 2.16   ultiplate hydraulically actuated clutches

travel. Therefore, the pedal must be fully depressed to squeeze the clutch brake. The clutch pedal should never be fully depressed before the gearbox is put into neutral. If the clutch brake is applied with the gearbox still in gear, a reverse load will be put on the gears making it difficult to get the gearbox out of gear. At the same time it will have the effect of trying to stop or decelerate the vehicle with the clutch brake and rapid wear of the friction disc will take place. Never apply the clutch brake when making down shifts, that is do not fully depress the clutch pedal when changing from a higher to a lower gear.

2.11 Multiplate hydraulically operated automatic transmission clutches (Fig. 2.16)

Automatic transmissions use multiplate clutches in addition to band brakes extensively with epicyclic compound gear trains to lock different stages of the gearing or gear carriers together, thereby providing a combination of gear ratios.

These clutches are comprised of a pack of annular discs or plates, alternative plates being internally and externally circumferentially grooved to match up with the input and output splined drive members respectively (Fig. 2.16). When these plates are squeezed together, torque will be transmitted from the input to the output members by way of these splines and grooves and the friction torque generated between pairs of rubbing surfaces. These steel plates are faced with either resinated paper linings or with sintered bronze linings, depending whether moderate or large torques are to be transmitted. Because the whole gear cluster assembly will be submerged in fluid, these linings are designed to operate wet (in fluid). These clutches are hydraulically operated by servo pistons either directly or indirectly through a lever disc spring to multiplate, the clamping load which also acts as a piston return spring. In this example of multiplate clutch utilization hydraulic fluid is supplied under pressure through radial and axial passages drilled in the output shaft. To transmit pressurized fluid from one member to another where there is relative angular movement between components, the output shaft has machined grooves on either side of all the radial supply passages. Square sectioned nylon sealing rings are then pressed into these grooves so that

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when the shaft is in position, these rings expand and seal lengthwise portions of the shaft with their corresponding bore formed in the outer members.

ront clutch   C

When pressurized, fluid is supplied to the front clutch piston chamber. The piston will move over to the right and, through the leverage of the disc spring, will clamp the plates together with considerable thrust. The primary sun gear will now be locked to the input turbine shaft and permit torque to be transmitted from the input turbine shaft to the central output shaft and primary sun gear.

ear clutch   C

When pressurized, fluid is released from the front clutch piston chamber, and is transferred to the rear clutch piston chamber. The servo piston will be forced directly against the end plate of the rear clutch multiplate pack. This compresses the release spring and sandwiches the drive and driven plates together so that the secondary sun gear will now be locked to the input turbine shaft. Torque can now be transmitted from the input turbine shaft to the secondary sun gear.

2.12 Semicentrifugal clutch (Figs 2.17 and 2.18)

With this design of clutch lighter pressure plate springs are used for a given torque carrying capacity, making it easier to engage the clutch in the lower speed range, the necessary extra clamping thrust being supplemented by the centrifugal force at higher speeds.

The release levers are made with offset bob weights at their outer ends, so that they are centrifugally out of balance (Fig. 2.17). The movement due to the centrifugal force about the fixed pivot tends to force the pressure plate against the driven plate, thereby adding to the clamping load. While the thrust due to the clamping springs is constant, the movement due to the centrifugal force varies as the square of the speed (Fig. 2.18). The reserve factor for the thrust spring can be reduced to 1.1 compared to 1.4–1.5 for a conventional helical coil spring clutch unit. Conversely, this clutch design may be used for heavy duty applications where greater torque loads are transmitted.

2.13   ully automatic centrifugal clutch

(Figs 2.19 and 2.20)

Fully automatic centrifugal clutches separate the engine from the transmission system when the engine is stopped or idling and smoothly take up the drive with a progressive reduction in slip within a narrow rising speed range until sufficient engine power is developed to propel the vehicle directly. Above this speed full clutch engagement is provided.

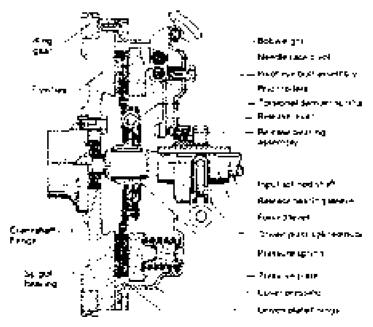To facilitate gear changes whilst the vehicle is in motion, a conventional clutch release

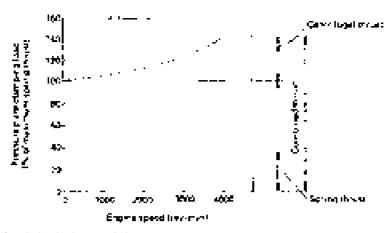**Fig. 2.17** Semicentrifugal clutch

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Fig. 2.18 Semicentrifugal clutch characteristics

lever arrangement is additionally provided. This mechanism enables the driver to disengage and engage the clutch independently of the flyweight action so that the drive and driven gearbox member speeds can be rapidly and smoothly unified during the gear selection process.

The automatic centrifugal mechanism consists of a reaction plate situated in between the pressure plate and cover pressing. Mounted on this reaction plate by pivot pins are four equally spaced bobweights (Fig. 2.19). When the engine s speed increases, the bobweight will tend to fly outward. Since the centre of gravity of their masses is to one side of these pins, they will rotate about their pins. This will be partially prevented by short struts offset to the pivot pins which relay this movement and effort to the pressure plate. Simultaneously, the reaction to this axial clamping thrust causes the reaction plate to compress both the reaction and pressure springs so that it moves backwards towards the cover pressing.

The greater the centrifugal force which tends to rotate the bobweights, the more compressed the springs will become and their reaction thrust will be larger, which will increase the pressure plate clamping load.

To obtain the best pressure plate thrust to engine speed characteristics (Fig. 2.20), ad ustable reactor springs are incorporated to counteract the main compression spring reaction. The initial compression length and therefore loading of these springs is set up by the ad usting nut after the whole unit has been assembled. Thus the resultant thrust of both lots of springs determine the actual take-up engine speed of the clutch.

ear changes are made when the clutch is disengaged, which is achieved by moving the release

bearing forwards. This movement pulls the reactor plate rearwards by means of the knife-edge link and also withdraws the pressure plate through the retractor springs so as to release the pressure plate clamping load.

2.14 lutch pedal actuating mechanisms
Some unusual ways of operating a clutch unit will now be described and explained.

.1 .1 *Clutch pedal ith over centre spring*
(Fig. 2.21)
With this clutch pedal arrangement, the over-centre spring supplements the foot pressure applied when disengaging the clutch, right up until the diaphragm spring clutch is fully disengaged (Fig. 2.21). It also holds the clutch pedal in the off position. When the clutch pedal is pressed, the master cylinder piston forces the brake fluid into the slave cylinder. The slave piston moves the push rod, which in turn disengages the clutch. After the pedal has been depressed approximately 25 mm of its travel, the over-centre spring change over point has been passed. The over-centre spring tension is then applied as an assistance to the foot pressure.

Ad ustment of the clutch is carried out by ad usting the pedal position on the master cylinder push rod.

.1 . *Clutch cable lin age ith automatic ad uster* (Fig. 2.22)
The release bearing is of the ball race type and is kept in constant contact with the fingers of the diaphragm spring by the action of the pedal self-ad ustment mechanism. In consequence, there is no pedal free movement ad ustment required (Fig. 2.22).
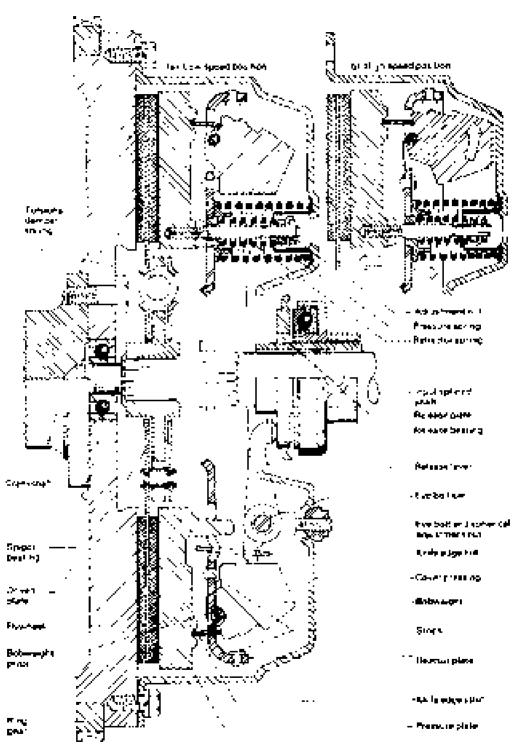
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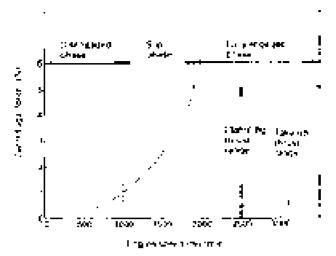**Fig. 2.19** Fully automatic centrifugal clutch

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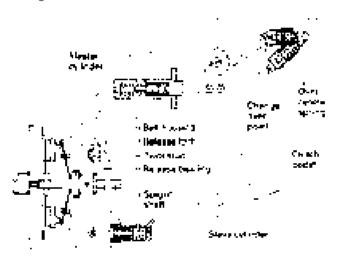**Fig. 2.20** Fully automatic centrifugal clutch characteristics

Fig. 2.21 Hydraulic clutch operating system with over-centre spring

When the pedal is released, the ad ustment pawl is no longer engaged with the teeth on the pedal quadrant. The cable, however, is tensioned by the spring which is located between the pedal and quadrant. As the pedal is depressed, the pawl engages in the nearest vee between the teeth. The particular tooth engagement position will gradually change as the components move to compensate for wear in the clutch driven plate and stretch in the cable.

.1 .3 Clutch air/hydraulic servo (Fig. 2.23)
In certain applications, to reduce the driver s foot effort in operating the clutch pedal, a clutch air/ hydraulic servo may be incorporated into the actuating linkage. This unit provides power assistance whenever the driver depresses the clutch pedal or maintains the pedal in a partially depressed position, as may be necessary under pull-away conditions. Movement of the clutch pedal is immediately relayed by way of the servo to the clutch in proportion to the input pedal travel.

As the clutch s driven plate wears, clutch actuating linkage movement is automatically taken up by the air piston moving further into the cylinder. Thus the actual servo movement when the clutch is being engaged and disengaged remains approximately constant. In the event of any interruption of the air supply to the servo the clutch will still operate, but without any servo assistance.

Immediately the clutch pedal is pushed down, the fluid from the master cylinder is displaced into
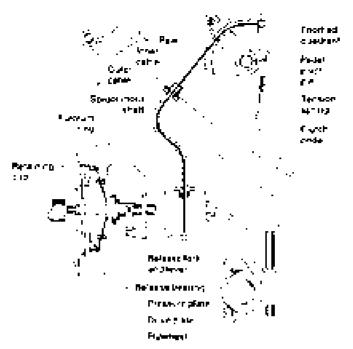
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Fig. 2.22 Clutch cable linkage with automatic ad uster

the servo hydraulic cylinder. The pressure created will act on both the hydraulic piston and the reaction plunger. Subsequently, both the hydraulic piston and the reaction plunger move to the right and allow the exhaust valve to close and the inlet valve to open. Compressed air will now pass through the inlet valve port and the passage connecting the reaction plunger chamber to the compressed air piston cylinder. It thereby applies pressure against the air piston. The combination of both hydraulic and air pressure on the hydraulic and air piston assembly causes it to move over, this movement being transferred to the clutch release bearing which moves the clutch operating mechanism to the disengaged position (Fig. 2.23(d)).

When the clutch pedal is held partially depressed, the air acting on the right hand side of the reaction plunger moves it slightly to the left which now closes the inlet valve. In this situation, further air is prevented from entering the air cylinder. Therefore, since no air can move in or out of the servo air cylinder and both valves are in the lapped position (both seated), the push rod will not move unless the clutch pedal is again moved (Fig. 2.23(c)).

When the clutch pedal is released fluid returns from the servo to the master cylinder. This permits the reaction plunger to move completely to the left

and so opens the exhaust valve. Compressed air in the air cylinder will now transfer to the reaction plunger chamber. It then passes through the exhaust valve and port where it escapes to the atmosphere. The released compressed air from the cylinder allows the clutch linkage return spring to move the air and hydraulic piston assembly back to its original position in its cylinder and at the same time this movement will be relayed to the clutch release bearing, whereby the clutch operating mechanism moves to the engaged drive position (Fig. 2.23(a)).

2.1 omposite flywheel and integral single plate diaphragm clutch (Fig. 2.24)

This is a compact diaphragm clutch unit built as an integral part of the two piece flywheel. It is designed for transaxle transmission application where space is at a premium and maximum torque transmitting capacity is essential.

The flywheel and clutch drive pressing acts as a support for the annular flywheel mass and functions as the clutch pressure plate drive member. The advantage of having the flywheel as a two piece assembly is that its mass can be concentrated more effectively at its outer periphery so that its overall mass can be reduced for the same cyclic torque and speed fluction which it regulates.
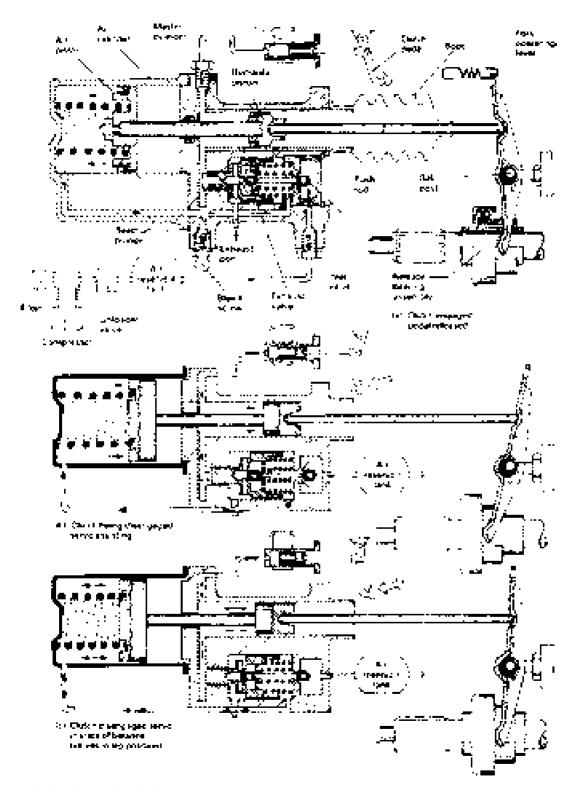
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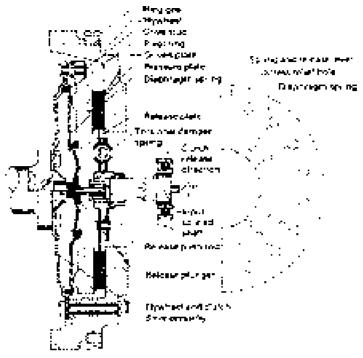**Fig. 2.23 (a–c)**   Clutch air/hydraulic servo

Fig. 2.24 ntegral single plate clutch and composite flywheel

The diaphragm spring takes the shape of a dished annular disc. The inner portion of the disc is radially slotted, the outer ends being enlarged with a circular hole to prevent stress concentration when the spring is deflected during disengagement. These radial slots divide the disc into many inwardly pointing fingers which have two functions, firstly to provide the pressure plate with an evenly distributed multileaf spring type thrust, and secondly to act as release levers to separate the driven plate from the sandwiching flywheel and pressure plate friction faces.

To actuate the clutch release, the diaphragm spring is made to pivot between a pivot spring positioned inside the flywheel/clutch drive pressing near its outer periphery and a raised circumferential rim formed on the back of the pressure plate. The engagement and release action of the clutch is similar to the pull type diaphragm clutch where the diaphragm is distorted into a dished disc when

assembled and therefore applies on axial thrust between the pressure plate and its ad acent flywheel/clutch drive pressing. With this spring leverage arrangement, a larger pressure plate and diaphragm spring can be utilised for a given overall diameter of clutch assembly. This design therefore has the benefits of lower pedal effort, higher transmitting torque capacity, a highly progressive engagement take-up and increased clutch life compared to the conventional push type diaphragm clutch.

The engagement and release mechanism consists of a push rod which passes through the hollow gearbox input shaft and is made to enter and contact the blind end of a recess formed in the release plunger. The plunger is a sliding fit in the normal spigot bearing hole made in the crankshaft end flange. It therefore guides the push rod and transfers its thrust to the diaphragm spring fingers via the release plate.

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3 Manual gearboxes and overdrives

3.1 The necessity for a gearbox

Power from a petrol or diesel reciprocating engine transfers its power in the form of torque and angular speed to the propelling wheels of the vehicle to produce motion. The object of the gearbox is to enable the engine's turning effect and its rotational speed output to be adjusted by choosing a range of under- and overdrive gear ratios so that the vehicle responds to the driver's requirements within the limits of the various road conditions. An insight of the forces opposing vehicle motion and engine performance characteristics which provide the background to the need for a wide range of gearbox designs used for different vehicle applications will now be considered.

3.1.1 *Resistance to vehicle motion*

To keep a vehicle moving, the engine has to develop sufficient power to overcome the opposing road resistance power, and to pull away from a standstill or to accelerate a reserve of power in addition to that absorbed by the road resistance must be available when required.

Road resistance is expressed as (kN). The propelling thrust at the tyre to road interface needed to overcome this resistance is known as (kN) (Fig. 3.1). For matching engine power output capacity to the opposing road resistance it is sometimes more convenient to express the opposing resistance to motion in terms of .

The road resistance opposing the motion of the vehicle is made up of three components as follows:

1 Rolling resistance
2 Air resistance
3 Gradient resistance

Rolling resistance (Fig. 3.1) Power has to be expended to overcome the restraining forces caused by the deformation of tyres and road surfaces and the interaction of frictional scrub when tractive effect is applied. Secondary causes of rolling resistance are wheel bearing, oil seal friction and the churning of the oil in the transmission system. It has been found that the flattening distortion of the tyre casing at the road surface interface consumes
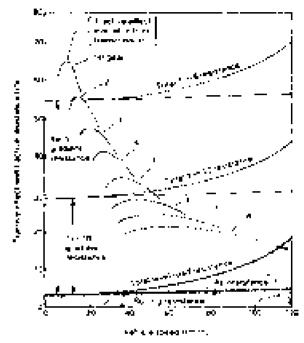
Fig. 3.1 Vehicle tractive resistance and effort performance chart

more energy as the wheel speed increases and therefore the rolling resistance will also rise slightly as shown in Fig. 3.1. Factors which influence the magnitude of the rolling resistance are the laden weight of the vehicle, type of road surface, and the design, construction and materials used in the manufacture of the tyre.

Air resistance (Fig. 3.1) Power is needed to counteract the tractive resistance created by the vehicle moving through the air. This is caused by air being pushed aside and the formation of turbulence over the contour of the vehicle's body. It has been found that the air resistance opposing force and air resistance power increase with the square and cube of the vehicle's speed respectively. Thus at very low vehicle speeds air resistance is insignificant, but it becomes predominant in the upper

speed range. Influencing factors which determine the amount of air resistance are frontal area of vehicle, vehicle speed, shape and streamlining of body and the wind speed and direction.

radient resistance (Fig. 3.1) Power is required to propel a vehicle and its load not only along a level road but also up any gradient likely to be encountered. Therefore, a reserve of power must be available when climbing to overcome the potential energy produced by the weight of the vehicle as it is progressively lifted. The gradient resistance opposing motion, and therefore the tractive effect or power needed to drive the vehicle forward, is directly proportional to the laden weight of the vehicle and the magnitude of gradient. Thus driving up a slope of 1 in 5 would require twice the reserve of power to that needed to propel the same vehicle up a gradient of 1 in 10 at the same speed (Fig. 3.1).

3.1.   o er to   eight ratio

When choosing the lowest and highest gearbox gear ratios, the most important factor to consider is not ust the available engine power but also the weight of the vehicle and any load it is expected to propel. Consequently, the power developed per unit weight of laden vehicle has to be known. This is usually expressed as the   .

i.e. $\text{Power to weight ratio} = \dfrac{\text{Brake power developed}}{\text{aden weight of vehicle}}$

There is a vast difference between the power to weight ratio for cars and commercial vehicles which is shown in the following examples.

etermine the power to weight ratio for the following modes of transport:

a) A car fully laden with passengers and luggage weighs 1.2 tonne and the maximum power produced by the engine amounts to 120 kW.
b) A fully laden articulated truck weighs 38 tonne and a 290 kW engine is used to propel this load.

a) $\text{Power to weight ratio} = \dfrac{120}{1.2} = 100 \text{ kW/tonne}$

b) $\text{Power to weight ratio} = \dfrac{290}{38} = 7.6 \text{ kW/tonne}.$

3.1.3   atio span

Another ma or consideration when selecting gear ratios is deciding upon the steepest gradient the vehicle is expected to climb (this may normally be taken as 20%, that is 1 in 5) and the maximum level road speed the vehicle is expected to reach in top gear with a small surplus of about 0.2% gradeability.

The two extreme operating conditions ust described set the highest and lowest gear ratios. To fix these conditions, the ratio of road speed in highest gear to road speed in lowest gear at a given engine speed should be known. This quantity is referred to as the   .

i.e. $\text{Ratio span} = \dfrac{\text{Road speed in highest gear}}{\text{Road speed in lowest gear}}$

(both road speeds being achieved at similar engine speed).

Car and light van gearboxes have ratio spans of about 3.5:1 if top gear is direct, but with overdrive this may be increased to about 4.5:1. arge commercial vehicles which have a low power to weight ratio, and therefore have very little surplus power when fully laden, require ratio spans of between 7.5 and 10:1, or even larger for special applications.

An example of the significance of ratio span is shown as follows:

Calculate the ratio span for both a car and heavy commercial vehicle from the data provided.

Type of vehicle	ear	Ratio	km/h/1000 rev/min
Car	Top	0.7	39
	First	2.9	9.75
Commercial vehicle (CV)	Top	1.0	48
	First	6.35	6

Car ratio span $= \dfrac{39}{9.75} = 4.0\text{:}1$

Commercial vehicle ratio span $= \dfrac{48}{6} = 8.0\text{:}1$

3.1.   ngine tor ue rise and speed operating range (Fig. 3.2)

Commercial vehicle engines used to pull large loads are normally designed to have a positive torque rise curve, that is from maximum speed to peak torque with reducing engine speed the available torque increases (Fig. 3.2). The amount of engine torque rise is normally expressed as a percentage of the peak torque from maximum speed (rated power) back to peak torque.

$\% \text{ torque rise} = \dfrac{\text{Maximum speed torque}}{\text{Peak torque}} \times 100$
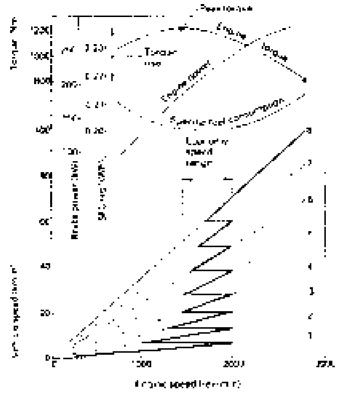
Fig. 3.2 Engine performance and gear split chart for an eight speed gearbox

The torque rise can be shaped depending upon engine design and taking into account such features as naturally aspirated, resonant induction tuned, turbocharged, turbocharged with intercooling and so forth. Torque rises can vary from as little as 5 to as high as 50%, but the most common values for torque rise range from 15 to 30%.

A large torque rise characteristic raises the engine s operating ability to overcome increased loads if the engine s speed is pulled down caused by changes in the road conditions, such as climbing steeper gradients, and so tends to restore the original running conditions. If the torque rise is small it cannot help as a buffer to supplement the high torque demands and the engine speed will rapidly fade. Frequent gear changes therefore become necessary compared to engines operating with high torque rise characteristics. nce the engine speed falls below peak torque, the torque rise becomes negative and the pulling ability of the engine drops off very quickly.

Vehicle driving technique should be such that engines are continuously driven between the speed range of peak torque and governed speed. The driver can either choose to operate the engine s speed in a range varying ust below the maximum rated power to achieve maximum performance and ourney speed or, to improve fuel economy, wear and noise, within a speed range of between 200 to 400 rev/min on the positive torque rise side of the engine torque curve that is in a narrow speed band ust beyond peak torque. Fig. 3.2 shows that the economy speed range operates with the specific fuel consumption at its minimum and that the engine speed band is in the most effective pulling zone.

3.2 ive speed and reverse synchromesh gearbo es

With even wider engine speed ranges (1000 to 6000 rev/min) higher car speeds (160 km/h and more) and high speed motorways, it has become desirable, and in some cases essential, to increase the number of traditional four speed ratios to five, where the fifth gear, and sometimes also the fourth gear, is an overdrive ratio. The advantages of increasing the number of ratio steps are several not only does the extra gear provide better acceleration response, but it enables the maximum engine rotational speed to be reduced whilst in top gear cruising, fuel

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ab 3.1 Typical four and five speed gearbox gear ratios

Five speed box		Four speed box	
ear	Ratio	ear	Ratio
top		top	1
	1		1
	1	2	2 1
2	2	1	
1		R	
R			

consumption is improved and engine noise and wear are reduced. Typical gear ratios for both four and five speed gearboxes are as shown in Table 3.1.

The construction and operation of four speed gearboxes was dealt with in   . The next section deals with five speed synchromesh gearboxes utilized for longitudinal and transverse mounted engines.

3. .1   ive speed and reverse double stage synchromesh gearbo   (Fig. 3.3)

With this arrangement of a five speed double stage gearbox, the power input to the first motion shaft passes to the layshaft and gear cluster via the first stage pair of meshing gears. Rotary motion is therefore conveyed to all the second stage layshaft and mainshaft gears (Fig. 3.3). Because each pair of second stage gears has a different size combination, a whole range of gear ratios are provided. Each mainshaft gear (whilst in neutral) revolves on the mainshaft but at some relative speed to it. Therefore, to obtain output powerflow, the selected mainshaft gear has to be locked to the mainshaft. This then completes the flow path from the first motion shaft, first stage gears, second stage gears and finally to the mainshaft.

In this example the fifth gear is an overdrive gear so that to speed up the mainshaft output relative to the input to the first motion shaft, a large layshaft fifth gear wheel is chosen to mesh with a much smaller mainshaft gear.

For heavy duty operations, a forced feed lubrication system is provided by an internal gear crescent type oil pump driven from the rear end of the layshaft (Fig. 3.3). This pump draws oil from the base of the gearbox casing, pressurizes it and then forces it through a passage to the mainshaft. The oil is then transferred to the axial hole along the centre of the mainshaft by way of an annular passage formed between two nylon oil seals.

ubrication to the mainshaft gears is obtained by radial branch holes which feed the rubbing surfaces of both mainshaft and gears.

3. .   ive speed and reverse single stage synchromesh gearbo   (Fig. 3.4)

This two shaft gearbox has only one gear reduction stage formed between pairs of different sized constant mesh gear wheels to provide a range of gear ratios. Since only one pair of gears mesh, compared to the two pairs necessary for the double stage gearbox, frictional losses are halved.

Power delivered to the input primary shaft can follow five different flow paths to the secondary shaft via first, second, third, fourth and fifth gear wheel pairs, but only one pair is permitted to transfer the drive from one shaft to another at any one time (Fig. 3.4).

The conventional double stage gearbox is designed with an input and output drive at either end of the box but a more convenient and compact arrangement with transaxle units where the final drive is integral to the gearbox is to have the input and output power flow provided at one end only of the gearbox.

In the neutral position, first and second output gear wheels will be driven by the corresponding gear wheels attached to the input primary shaft, but they will only be able to revolve about their own axis relative to the output secondary shaft. Third, fourth and fifth gear wheel pairs are driven by the output second shaft and are free to revolve only relative to the input primary shaft because they are not attached to this shaft but use it only as a supporting axis.

When selecting individual gear ratios, the appropriate synchronizing sliding sleeve is pushed towards and over the dog teeth forming part of the particular gear wheel required. Thus with first and second gear ratios, the power flow passes from the input primary shaft and constant mesh pairs of gears to the output secondary shaft via the first and second drive hub attached to this shaft.   ear engagement is completed by the synchronizing sleeve locking the selected output gear wheel to the output secondary shaft. Third, fourth and fifth gear ratios are selected when the third and fourth or fifth gear drive hub, fixed to the input primary shaft, is locked to the respective gear wheel dog clutch by sliding the synchronizing sleeve in to mesh with it. The power flow path is now transferred from the input primary shaft drive hub and selected pair of constant mesh gears to the output secondary shaft.
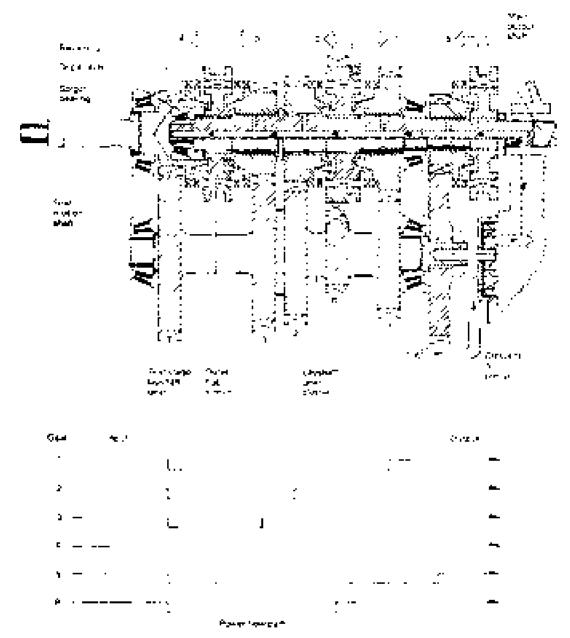
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Fig. 3.3 Five speed and reverse double stage synchromesh gearbox

Transference of power from the gearbox output secondary shaft to the differential left and right hand drive shafts is achieved via the final drive pinion and gear wheel which also provide a permanent gear reduction (Fig. 3.4). Power then flows from the differential cage which supports the final drive gear wheel to the cross-pin and planet gears where it then divides between the two side sun gears and accordingly power passes to both stub drive shafts.

3.3 ear synchroni ation and engagement
The gearbox basically consists of an input shaft driven by the engine crankshaft by way of the clutch and an output shaft coupled indirectly either
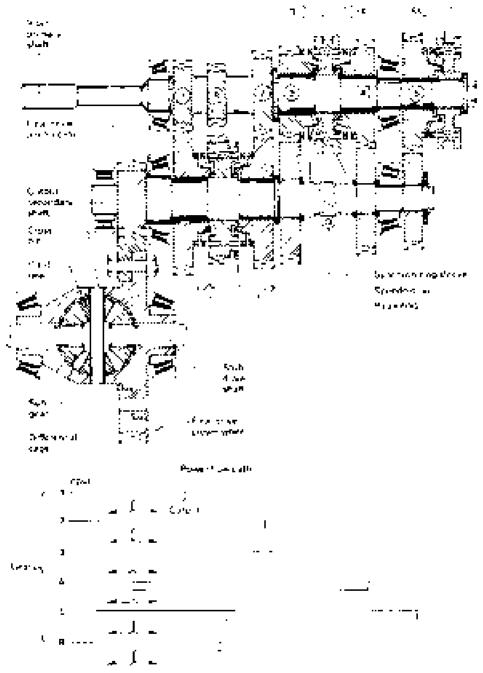
Fig. 3.4 Five speed and reverse single stage synchromesh gearbox with integral final drive (transaxle unit)

through the propellor shaft or intermediate gears to the final drive. Between these two shafts are pairs of gear wheels of different size meshed together.

If the gearbox is in neutral, only one of these pairs of gears is actually attached rigidly to one of these shafts while the other is free to revolve on the second shaft at some speed determined by the existing speeds of the input and output drive shafts.

To engage any gear ratio the input shaft has to be disengaged from the engine crankshaft via the

clutch to release the input shaft drive. It is then only the angular momentum of the input shaft, clutch drive plate and gear wheels which keeps them revolving. The technique of good gear changing is to be able to udge the speeds at which the dog teeth of both the gear wheel selected and output shaft are rotating at a uniform speed, at which point in time the dog clutch sleeve is pushed over so that both sets of teeth engage and mesh gently without grating.

Because it is difficult to know exactly when to make the gear change a device known as the
             is utilized. Its function is to apply a friction clutch braking action between the engaging gear and drive hub of the output shaft so that their speeds will be unified before permitting the dog teeth of both members to contact.

Synchromesh devices use a multiplate clutch or a conical clutch to equalise the input and output rotating members of the gearbox when the process of gear changing is taking place. Except for special applications, such as in some splitter and range change auxiliary gearboxes, the conical clutch method of synchronization is generally employed.

With the conical clutch method of producing silent gear change, the male and female cone members are brought together to produce a synchronizing frictional torque of sufficient magnitude so that one or both of the input and output members rotational speed or speeds ad ust automatically until they revolve as one.   nce this speed uniformity has been achieved, the end thrust applied to the dog clutch sleeve is permitted to nudge the chamfered dog teeth of both members into alignment, thereby enabling the two sets of teeth to slide quietly into engagement.

3.3.1   on positive constant load synchromesh unit (Fig. 3.5(a, b and c))

When the gear stick is in the neutral position the spring loaded balls trapped between the inner and outer hub are seated in the circumferential groove formed across the middle of the internal dog teeth (Fig. 3.5(a)). As the driver begins to shift the gear stick into say top gear (towards the left), the outer and inner synchromesh hubs move as one due to the radial spring loading of the balls along the splines formed on the main shaft until the female cone of the outer hub contacts the male cone of the first motion gear (Fig. 3.5(b)). When the pair of conical faces contact, frictional torque will be generated due to the combination of the axial thrust and the difference in relative speed of both input and output shaft members. If sufficient axial thrust is applied to the outer hub, the balls will be depressed inwards against the radial loading of the springs. Immedi-

ately the balls are pushed out of their groove, the chamfered edges of the outer hub s internal teeth will then be able to align with the corresponding teeth spacing on the first motion gear. Both sets of teeth will now be able to mesh so that the outer hub can be moved into the fully engaged position (Fig. 3.5(c)).

Note the bronze female cone insert frictional face is not smooth, but consists of a series of tramline grooves which assist in cutting away the oil film so that a much larger synchronizing torque will be generated to speed up the process.

3.3.   ositive baul ring synchromesh unit (Fig. 3.6(a, b and c))

The gearbox mainshaft rotates at propellor shaft speed and, with the clutch disengaged, the first motion shaft gear, layshaft cluster gears, and mainshaft gears rotate freely.

 rive torque will be transmitted when a gear wheel is positively locked to the mainshaft. This is achieved by means of the outer synchromesh hub internal teeth which slide over the inner synchromesh hub splines (Fig. 3.6(a)) until they engage with dog teeth formed on the constant mesh gear wheel being selected.

When selecting and engaging a particular gear ratio, the gear stick slides the synchromesh outer hub in the direction of the chosen gear (towards the left). Because the shift plate is held radially outwards by the two energizing ring type springs and the raised middle hump of the plate rests in the groove formed on the inside of the hub, the end of the shift plate contacts the baulking ring and pushes it towards and over the conical surface, forming part of the constant mesh gear wheel (Fig. 3.6(b)).

The frictional grip between the male and female conical members of the gear wheel and baulking ring and the difference in speed will cause the baulking ring to be dragged around relative to the inner hub and shift plate within the limits of the clearance between the shift plate width and that of the recessed slot in the baulking ring.   wing to the designed width of the shift plate slot in the baulking ring, the teeth on the baulking ring are now out of alignment with those on the outer hub by approximately half a tooth width, so that the chamfered faces of the teeth of the baulking ring and outer hub bear upon each other.

As the baulking ring is in contact with the gear cone and the outer hub, the force exerted by the driver on the gear stick presses the baulking ring female cone hard against the male cone of the gear. Frictional torque between the two surfaces will eventually cause these two members to equalize
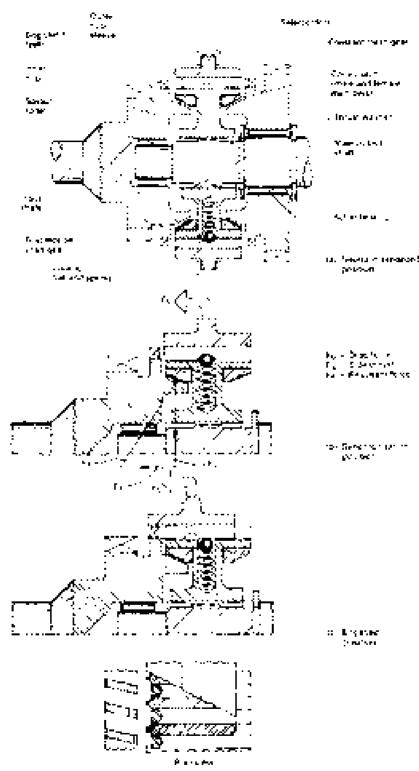
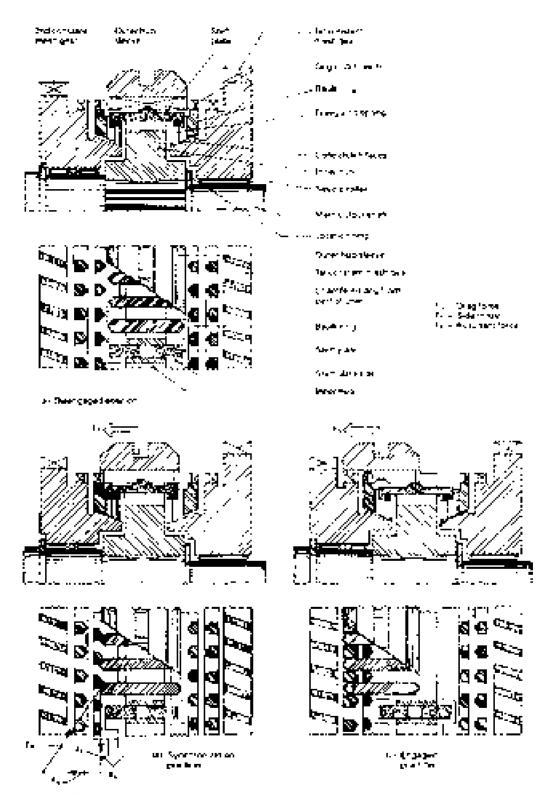**Fig. 3.5**   Non-positive constant load synchromesh unit

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Fig. 3.6 (a–c) ositive baulk ring synchromesh unit

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their speeds. Until this takes place, full engagement of the gear and outer hub dog teeth is prevented by the out of alignment position of the baulking ring teeth. When the gear wheel and main shaft have unified their speeds, the synchronizing torque will have fallen to zero so that the baulking ring is no longer dragged out of alignment. Therefore the outer hub can now overcome the baulk and follow through to make a positive engagement between hub and gear (Fig. 3.6(c)). It should be understood that the function of the shift plate and springs is to transmit ust sufficient axial load to ensure a rapid bringing together of the mating cones so that the baulking ring dog teeth immediately misalign with their corresponding outer hub teeth. nce the cone faces contact, they generate their own friction torque which is sufficient to flick the baulking ring over, relative to the outer hub. Thus the chamfers of both sets of teeth contact and oppose further outer hub axial movement towards the gear dog teeth.

3.3.3   ositive baul pin synchromesh unit
(Fig. 3.7(a, b, c and d))
Movement of the selector fork synchronizing sleeve to the left (Fig. 3.7(a and b)) forces the female (internal) cone to move into contact with the male (external) cone on the drive gear. Frictional torque will then synchronize (unify) the input and output speeds. Until speed equalization is achieved, the collars on the three thrust pins (only one shown) will be pressed hard into the enlarged position of the slots (Fig. 3.5(c)) in the synchronizing sleeve owing to the frictional drag when the speeds are dissimilar. Under these conditions, unless extreme pressure is exerted, the dog teeth cannot be crushed by forcing the collars into the narrow portion of the slots.   owever, when the speeds of the synchromesh hub and drive gear are equal (synchronized) the collars tend to float in the enlarged portion of the slots, there is only the pressure of the spring loaded balls to be overcome. The collars will then slide easily into the narrow portion of the slots (Fig. 3.5(d)) allowing the synchronizer hub dog teeth to shift in to mesh with the dog teeth on the driving gear.

3.3.   Split baul pin synchromesh unit
(Fig. 3.8(a, b, c and d))
The synchronizing assembly is composed of two thick bronze synchronizing rings with tapered female conical bores, and situated between them is a hardened steel drive hub internally splined with external dog teeth at each end (Fig. 3.8(a)). Three shouldered pins, each with a groove around its

centre, hold the bronze synchronizing cone rings apart. Alternating with the shouldered pins on the same pitch circle are diametrically split pins, the ends of which fit into blind bores machined in the synchronizing cone rings. The pin halves are sprung apart, so that a chamfered groove around the middle of each half pin registers with a chamfered hole in the drive hub.

If the gearbox is in the neutral position, both sets of shouldered and split pins are situated with their grooves aligned with the central drive hub (Fig. 3.8(a and b)).

When an axial load is applied to the drive hub by the gear stick, it moves over (in this case to the left) until the synchronizing ring is forced against the ad acent first motion gear cone. The friction (synchronizing) torque generated between the rubbing tapered surfaces drags the bronze synchronizing ring relative to the mainshaft and drive hub until the grooves in the shouldered pins are wedged against the chamfered edges of the drive hub (Fig. 3.8(c)) so that further axial movement is baulked.

Immediately the input and output shaft speeds are similar, that is, synchronization has been achieved, the springs in the split pins are able to expand and centralize the shouldered pins relative to the chamfered holes in the drive hub. The drive hub can now ride out of the grooves formed around the split pins, thus permitting the drive hub to shift further over until the internal and external dog teeth of both gear wheel hub mesh and fully engage (Fig. 3.8(d)).

3.3.5 Split ring synchromesh unit
(Fig. 3.9(a, b, c and d))
In the neutral position the sliding sleeve sits centrally over the drive hub (Fig. 3.9(a)). This permits the synchronizing ring expander band and thrust block to float within the constraints of the recess machine in the side of the gear facing the drive hub (Fig. 3.9(b)).

For gear engagement to take place, the sliding sleeve is moved towards the gear wheel selected (to the left) until the inside chamfer of the sliding sleeve contacts the bevelled portion of the synchronizing ring. As a result, the synchronizing ring will be slightly compressed and the friction generated between the two members then drags the synchronizing ring round in the direction of whichever member is rotating fastest, be it the gear or driven hub. At the same time, the thrust block is pulled round so that it applies a load to one end of the expander band, whilst the other end is restrained from moving by the anchor block (Fig. 3.9(c)).
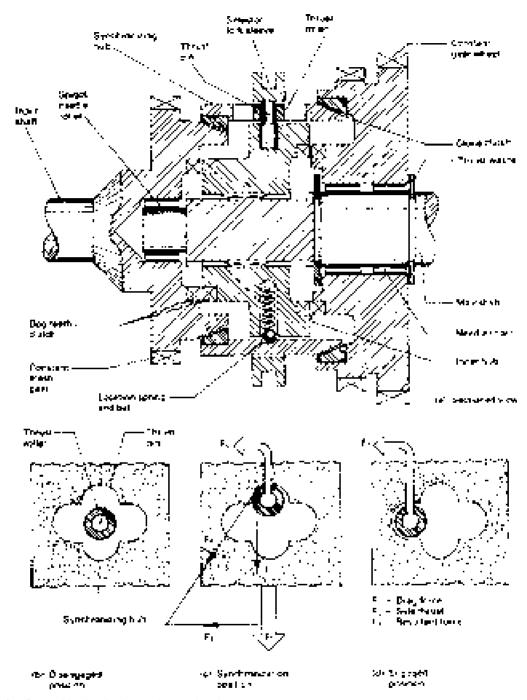
Fig. 3.7 (a–d) ositive baulk pin synchromesh unit

Whilst this is happening the expander is also pushed radially outwards. Consequently, there will be a tendency to expand the synchronizing slit ring, but this will be opposed by the chamfered mouth of the sliding sleeve. This energizing action attempting to expand the synchronizing ring prevents the sliding sleeve from completely moving over and engaging the dog teeth of the selected
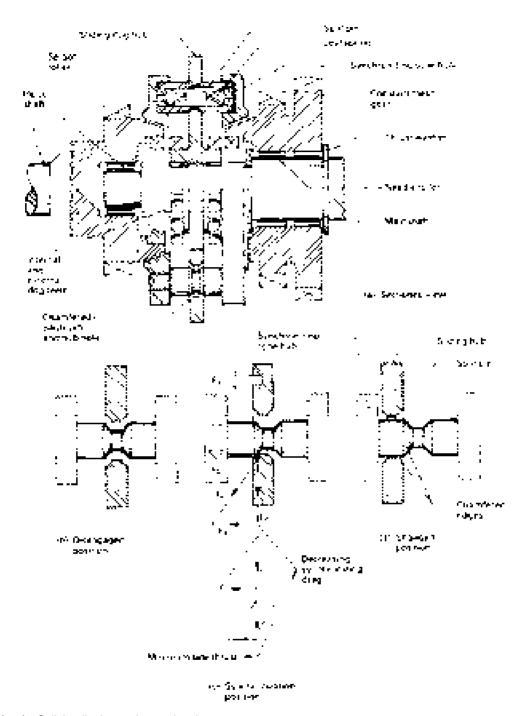
Fig. 3.8 (a–c) Split baulk pin synchromesh unit

gear wheel until both the drive hub and constant mesh gear wheel are revolving at the same speed.

When both input and output members are unified, that is, rotating as one, there cannot be any more friction torque because there is no relative speed to create the frictional drag. Therefore the expander band immediately stops exerting radial force on the inside of the synchronizing ring.
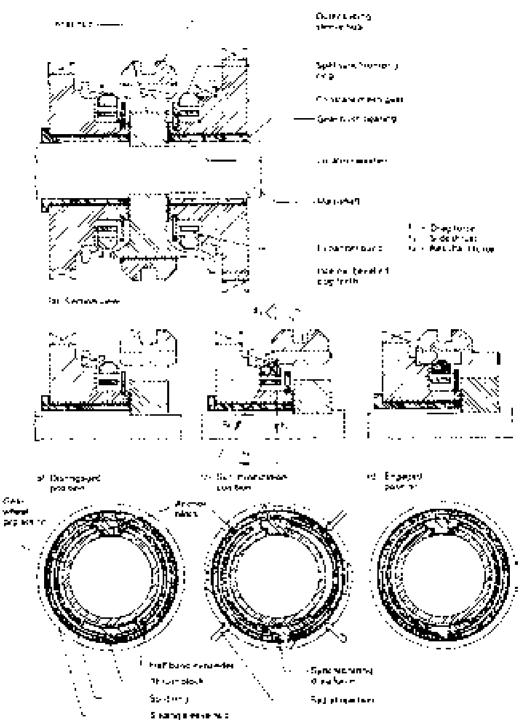
Fig. 3.9 (a–d) Split baulk ring synchromesh unit

The axial thrust applied by the gear stick to the sliding sleeve will now be sufficient to compress the split synchronizing ring and subsequently permits the sleeve to slide over the gear wheel dog teeth for full engagement (Fig. 3.9(d)).

3.4 Remote controlled gear selection and engagement mechanisms

ear selection and engagement is achieved by two distinct movements:

1 The selection of the required gear shift gate and the positioning of the engagement gate lever.
2 The shifting of the chosen selector gate rod into the engagement gear position.

These two operations are generally performed through the media of the gear shift lever and the remote control tube/rod. Any transverse movement of the gear shift lever by the driver selects the gear shift gate and the engagement of the gate is obtained by longitudinal movement of the gear shift lever.

Movement of the gear shift lever is conveyed to the selection mechanism via the remote control tube. Initially the tube is twisted to select the gate shift gate, followed by either a push or pull movement of the tube to engage the appropriate gear.

For the gear shift control to be effective it must have some sort of flexible linkage between the gear shift lever supported on the floor of the driver s compartment and the engine and transmission integral unit which is suspended on rubber mountings. This is essential to prevent engine and transmission vibrations being transmitted back to the body and floor pan and subsequently causing discomfort to the driver and passengers.

3. .1 emote controlled double rod and bell cran ed lever gear shift mechanism suitable for both four and five speed transverse mounted gearbo Talbot (Fig. 3.10)

Twisting the remote control tube transfers movement to the first selector link rod. This motion is then redirected at right angles to the transverse gate selector/engagement shaft via the selector relay lever (bell crank) to position the required gear gate (Fig. 3.10). A forward or backward movement of the remote control tube now conveys motion via the first engagement relay lever (bell crank), engagement link rod and second relay lever to rotate the transverse gate selector/engage-
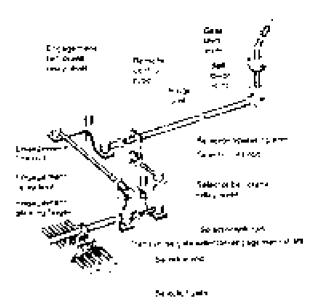

Fig. 3.10 Remote controlled double rod and bell crank lever gearshift mechanism suitable for both four and five speed transversely mounted gearbox

ment shaft. Consequently, this shifts the transverse selector/engagement shaft so that it pushes the synchronizing sliding sleeve into engagement with the selected gear dog teeth.

3. . emote controlled bell cran ed lever gear shift mechanism for a four speed transverse mounted gearbo ord (Fig. 3.11)

ear selection and engagement movement is conveyed from the gear shift lever pivot action to the remote control rod universal oint and to the control shift and relay lever guide (Fig. 3.11). Rocking the gear shift lever transversely rotates the control shaft and relay guide. This tilts the selector relay lever and subsequently the selection relay lever guide and shaft until the striker finger aligns with the chosen selector gate. A further push or pull movement to the gear shift lever by the driver then transfers a forward or backward motion via the remote control rod, control shaft and relay lever guide to the engagement relay lever. Movement is then redirected at right angles to the selector relay guide and shaft. Engagement of the gear required is finally obtained by the selector/engagement shaft forcing the striking finger to shift the gate and selector fork along the single selector rod so that the synchronizing sleeve meshes with the appropriate gear wheel dog clutch.
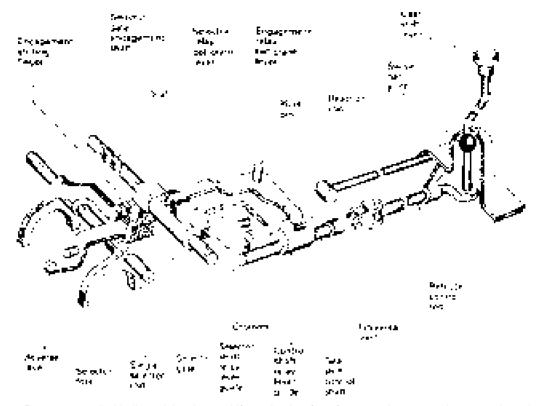
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Fig. 3.11 Remote controlled bell crank level gear shift mechanism for a four speed transversely mounted gearbox

3. .3   emote controlled sliding ball oint gear shift mechanism suitable for both four and five speed longitudinal or transverse mounted gearbo
(Fig. 3.12)

Selection and engagement of the different gear ratios is achieved with a swivel ball end pivot gear shift lever actuating through a sliding ball relay lever a single remote control rod (Fig. 3.12). The remote control rod transfers both rotary and push-pull movement to the gate selector and engagement shaft. This rod is also restrained in bushes between the gear shift lever mounting and the bulkhead. It thus permits the remote control rod to transfer both rotary (gate selection) and push-pull (select rod engagement shift) movement to the gate selector and engagement shaft. Relative movement between the suspended engine and transmission unit and the car body is compensated by the second sliding ball relay lever. As a result the gate engagement striking finger is able to select and shift into engagement the appropriate selector rod fork.

This single rod sliding ball remote control linkage can be used with either longitudinally or transversely mounted gearboxes, but with the latter
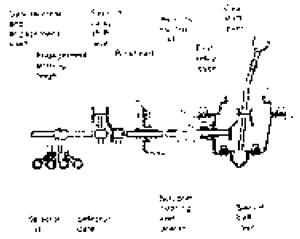

Fig. 3.12 Remote controlled sliding ball oint gear shift mechanism suitable for both four and five speed longitudinally or transversely mounted gearbox

an additional relay lever mechanism (not shown) is needed to convey the two distinct movements of selection and engagement through a right angle.

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3. . emote controlled double rod and hinged relay oint gear shift mechanism suitable for both four and five speed longitudinal mounted gearbo (Fig. 3.13)

With this layout the remote control is provided by a pair of remote control rods, one twists and selects the gear gate when the gear shift lever is given a transverse movement, while the other pushes or pulls when the gear shift lever is moved longitudinally (Fig. 3.13). Twisting movement is thus conveyed to the engagement relay lever which makes the engagement striking finger push the aligned selector gate and rod. Subsequently, the synchronizing sleeve splines mesh with the corresponding dog clutch teeth of the selected gear wheel. Relative movement between the gear shift lever swivel support and rubber mounted gearbox is absorbed by the hinged relay oint and the ball oints at either end of the remote control engagement rod.

3. .5 emote controlled single rod ith self aligning bearing gear shift mechanism suitable for both five and si speed longitudinal mounted gearbo ord (Fig. 3.14)

A simple and effective method of selecting and engaging the various gear ratios suitable for commercial vehicles where the driver cab is forward of the gearbox is shown in Fig. 3.14.
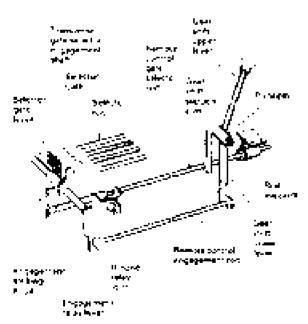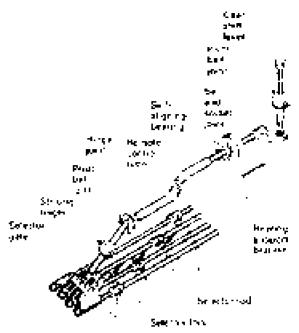

Fig. 3.14 Remote controlled single rod with self-aligning bearing gear shift mechanism suitable for both five and six speed longitudinally mounted gearbox

Movement of the gear shift lever in the usual transverse and longitudinal directions provides both rotation and a push-pull action to the remote control tube. Twisting the remote control tube transversely tilts the relay gear shift lever about its ball oint so that the striking finger at its lower end matches up with the selected gear gate. ear engagement is then completed by the driver tilting the gear shift lever away or towards himself. This permits the remote control tube to move axially through the mounted self-aligning bearing. As a result, a similar motion will be experienced by the relay gear shift lever, which then pushes the striking finger, selector gate and selector fork into the gear engaged position. It should be observed that the self-aligning bearing allows the remote control tube to slide to and fro. At the same time it permits the inner race member to tilt if any relative movement between the gearbox and chassis takes place.

Fig. 3.13 Remote controlled double rod and hinged relay oint gear shift mechanism suitable for both four and five speed longitudinally mounted gearbox

3. .6 emote controlled single rod ith s ing arm support gear shift mechanism suitable for five and si speed longitudinally mounted gearbo (Fig. 3.15)

This arrangement which is used extensively on commercial vehicles employs a universal cross-pin oint to transfer both the gear selection and
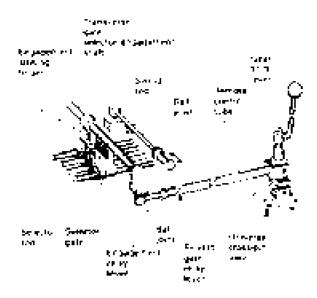
Fig. 3.15 Remote controlled single rod with swing arm support gear shift mechanism suitable for five and six speed longitudinally mounted gearbox

engagement motion to the remote control tube (Fig. 3.15). Twisting this remote control tube by giving the gear shift lever a transverse movement pivots the suspended selector gate relay lever so that the transverse gate selector/engagement shift moves across the selector gates until it aligns with the selected gate. The gear shift lever is then given a to or fro movement. This causes the transverse selector/engagement shaft to rotate, thereby forcing the striking finger to move the selector rod and fork. The synchronizing sleeve will now be able to engage the dog clutch of the appropriate gear wheel. Any misalignment between the gear shift lever support mounting and the gear shift mechanism forming part of the gearbox (caused by engine shake or rock) is thus compensated by the swing rod which provides a degree of float for the selector gate relay lever pivot point.

3. Splitter and range change gearboes

Ideally the tractive effect produced by an engine and transmission system developing a constant power output from rest to its maximum road speed would vary inversely with its speed. This characteristic can be shown as a smooth declining tractive effect curve with rising road speed (Fig. 3.16).

In practice, the transmission has a fixed number of gear ratios so that the ideal smooth tractive effect curve would be interrupted to allow for loss
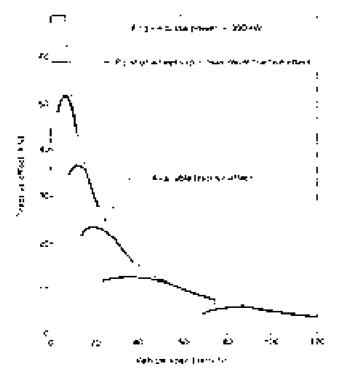

Fig. 3.16 deal and actual tractive effort-speed characteristics of a vehicle

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of engine speed and power between each gear change (see the thick lines of Fig. 3.16).

For a vehicle such as a saloon car or light van which only weighs about one tonne and has a large power to weight ratio, a four or five speed gearbox is adequate to maintain tractive effect without too much loss in engine speed and vehicle performance between gear changes.

Unfortunately, this is not the situation for heavy goods vehicles where large loads are being hauled so that the power to weight ratio is usually very low. Under such operating conditions if the gear ratio steps are too large the engine speed will drop to such an extent during gear changes that the engine torque recovery will be very sluggish (Fig. 3.17). Therefore, to minimize engine speed fall-off whilst changing gears, smaller gear ratio steps are required, that is, more gear ratios are necessary to respond to the slightest change in vehicle load, road conditions and the driver s requirements. Figs 3.2 and 3.18 show that by doubling the number of gear ratios, the fall in engine speed between gear shifts is much smaller. To cope with moderate payloads, conventional double stage compound gearboxes with up to six forward speeds are manufactured, but these boxes tend to be large and heavy. Therefore, if more gear ratios are essential for very heavy payloads, a far better way of extending the number of gear ratios is to utilize a two speed auxiliary gearbox in series with a three, four, five or six speed conventional compound gearbox. The function of this auxiliary box is to double the number of gear ratios of the conventional gearbox. With a three, four, five or six speed gearbox, the gear ratios are increased to six, eight, ten or twelve respectively (Figs 3.2 and 3.18). For very special
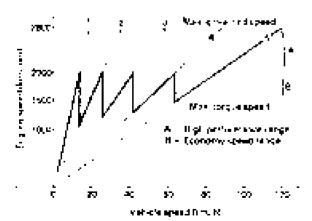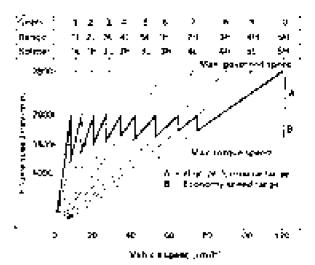

Fig. 3.18 Engine speed ratio chart for a vehicle employing either a ten speed range change or a splitter change gearbox

applications, a three speed auxiliary gearbox can be incorporated so that the gear ratios are trebled. Usually one of these auxiliary gear ratios provides a range of very low gear ratios known as   or   . The auxiliary gearbox may be situated either in front or to the rear of the conventional compound gearbox.

The combination of the auxiliary gearbox and the main gearbox can be designed to be used either as a splitter gear change or as a range gear change in the following way.

3.5.1 Splitter gear change (Figs 3.19 and 3.20)
With the splitter arrangement, the main gearbox gear ratios are spread out wide between ad acent gears whilst the two speed auxiliary gearbox has one direct gear ratio and a second gear which is either a step up or down ratio (Fig. 3.19). The auxiliary second gear ratio is chosen so that it splits the main gearbox ratio steps in half, hence the name   . The splitter indirect gear ratio normally is set between 1.2 and 1.4:1. A typical ratio would be 1.25:1.

A normal upchange sequence for an eight speed gearbox (Fig. 3.20), consisting of a main gearbox with four forward gear ratios and one reverse and a two speed auxiliary splitter stage, would be as follows:

Auxiliary splitter low ratio and main gearbox first gear selected results in first gear low (1) auxiliary splitter switched to high ratio but with the main gearbox still in first gear results in first gear high (1)

Fig. 3.17 Engine speed ratio chart for a vehicle employing a five speed gearbox
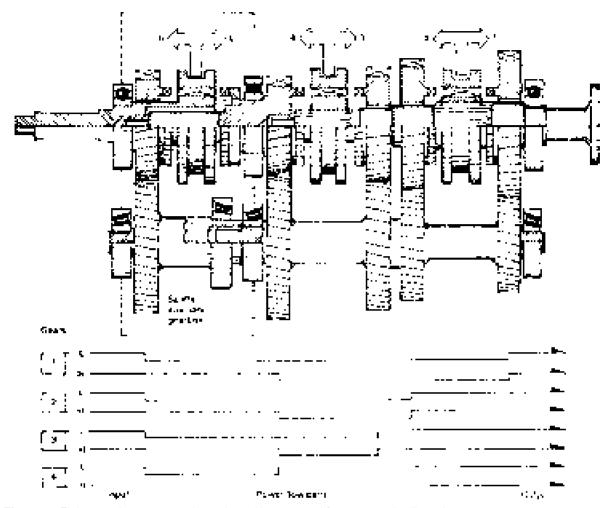
Fig. 3.19 Eight speed constant mesh gearbox with two speed front mounted splitter change

splitter switched again to low ratio and main gearbox second gear selected results in 2 splitter switched to high ratio, main gearbox gear remaining in second gives 2 splitter switched to low ratio main gearbox third gear selected gives 3 splitter switched to high ratio main gearbox still in third gives 3 . This procedure continues throughout the upshift from bottom to top gear ratio. Thus the overall upshift gear ratio change pattern would be:

ear ratio	1	2	3	4	5	6	7	8	Reverse
Upshift sequence	1	1	2	2	3	3	4	4	R	R

It can therefore be predicted that alternate changes involve a simultaneous upchange in the
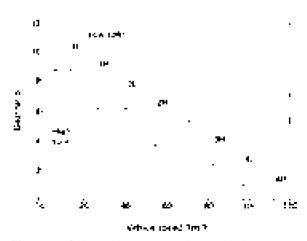
Fig. 3.20 Splitter change gear ratio–speed chart

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main gearbox and downchange in the splitter stage, or vice versa.

Referring to the thick lines in Figs 3.2, 3.17 and 3.18, these represent the recommended operating speed range for the engine for best fuel economy, but the broken lines in Fig. 3.17 represent the gear shift technique if maximum road speed is the criteria and fuel consumption, engine wear and noise become secondary considerations.

### 3.5.   ange gear change (Figs 3.21 and 3.22)

In contrast to the splitter gear change, the range gear change arrangement (Fig. 3.21) has the gear ratios between ad acent gear ratio steps set close together. The auxiliary two speed gearbox will have one ratio direct drive and the other one normally equal to ust over half the gear ratio spread from bottom to top. This is slightly larger than the main gearbox gear ratio spread.

To change from one gear ratio to the next with, for example, an eight speed gearbox comprising four normal forward gears and one reverse and a two speed auxiliary range change, the pattern of gear change would be as shown in Fig. 3.22.

Through the gear ratios from bottom to top low gear range is initially selected, the main gearbox order of upchanges are first, second, third and fourth gears. At this point the range change is moved to high gear range and the sequence of gear upchanges again becomes first, second, third and fourth. Therefore the total number of gear ratios is the sum of both low and high ranges, that is, eight. A tabulated summary of the upshift gear change pattern will be:

ear ratio	1	2	3	4	5	6	7	8	Reverse	
Upshift sequence	1	2	3	4	1	2	3	4	R	R

3.5.3 Si teen speed synchromesh gearbo ith range change and integral splitter gears
(Fig. 3.23)

This heavy duty commercial gearbox utilizes both a two speed range change and a two speed splitter gear change to enable the four speed gearbox to
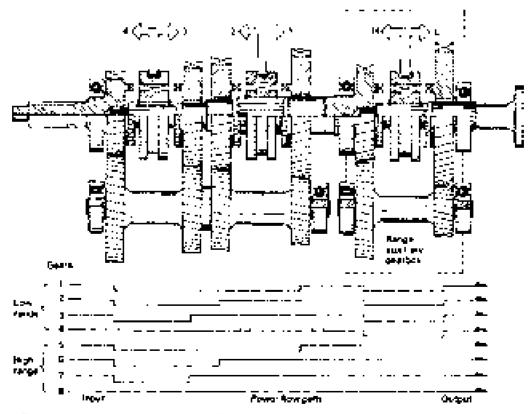

Fig. 3.21 Eight speed constant mesh gearbox with two speed rear mounted range change
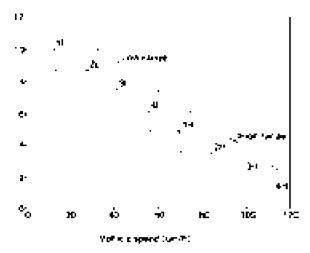
Fig. 3.22 Range change gear ratio–speed chart

extend the gear ratio into eight steps and, when required, to sixteen split (narrow) gear ratio intervals.

The complete gearbox unit can be considered to be divided into three sections the middle section (which is basically a conventional double stage four speed gearbox), and the first two pairs of gears at the front end which make up the two speed splitter gearbox. Mounted at the rear is an epicyclic gear train providing a two speed low and high range change (Fig. 3.23).

The epicyclic gear train at the rear doubles the ratios of the four speed gearbox permitting the driver to initially select the low ( ) gear range driving through this range 1, 2, 3 and 4 then selecting the high ( ) gear range. The gear change sequence is again repeated but the gear ratios now become 5, 6, 7 and 8.

If heavy loads are being carried, or if maximum torque is needed when overtaking on hills, much closer gear ratio intervals are desirable. This is provided by splitting the gear steps in half with the two speed splitter gears the gear shift pattern of 1st low, 1st high, 2nd low, 2nd high, 3rd low and so on is adopted.

ront end splitter two speed gearbo power flow (Fig. 3.23)   Input power to the gearbox is supplied to the first motion shaft. When the splitter synchronizing sliding sleeve is in neutral, both the splitter low and high input gear wheels revolve on their needle bearings independently of their supporting first motion shaft and mainshaft respectively.

When low or high splitter gears are engaged, the first motion shaft drive hub conveys power to the first or second pair of splitter gear wheels and hence to the layshaft gear cluster.

Mid-four speed gearbo power flow (Fig. 3.23) Power from the first motion shaft at a reduced speed is transferred to the layshaft cluster of gears and subsequently provides the motion to all the other mainshaft gear wheels which are free to revolve on the mainshaft, but at relatively different speeds when in the neutral gear position.

Engagement of one mid-gearbox gear ratio dog clutch locks the corresponding mainshaft drive hub to the chosen gear so that power is now able to pass from the layshaft to the mainshaft through the selected pair of gear wheels.

Reverse gear is provided via an idler gear which, when meshed between the layshaft and mainshaft, alters the direction of rotation of the mainshaft in the usual manner.

Rear end range two speed gearbo power flow (Fig. 3.23)   When the range change is in the neutral position, power passes from the mainshaft and sun gear to the planet gears which then revolve on the output shaft s carrier pin axes and in turn spin round the annular gear and synchronizing drive hub.

Engaging the low range gear locks the synchronizing drive hub to the gearbox casing. This forces the planet gears to revolve and walk round the inside of the annular gear. Consequently, the carrier and output shafts which support the planet gear axes will also be made to rotate but at a speed lower than that of the input shaft.

Changing to high range locks the annular gear and drive hub to the output shaft so that power flow from the planet gears is then divided between the carrier and annular, but since they need to rotate at differing speeds, the power flow forms a closed loop and ams the gearing. As a result, there is no gear reduction but ust a straight through drive to the output shaft.

3.5.   T in counter shaft ten speed constant mesh gearbo ith synchromesh t o speed rear mounted range change (Fig. 3.24)
With the quest for larger torque carrying capacity, closer steps between gear ratio changes, reduced gearbox length and weight, a unique approach to fulfil these requirements has been developed
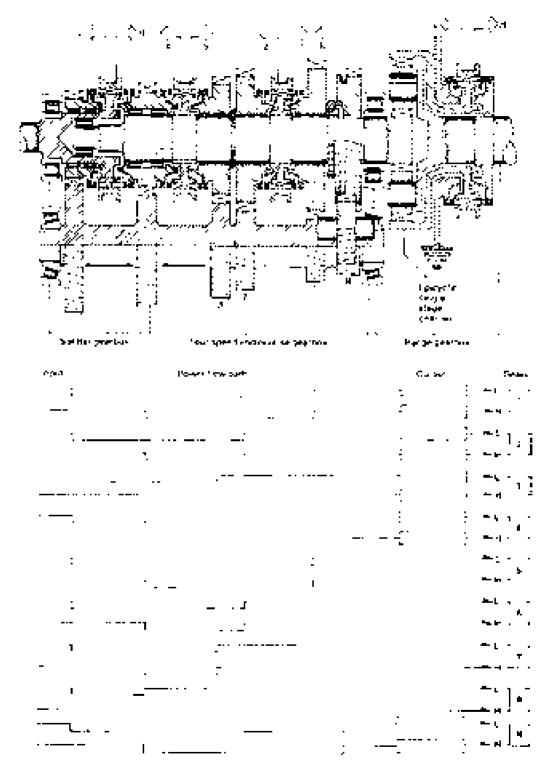
80

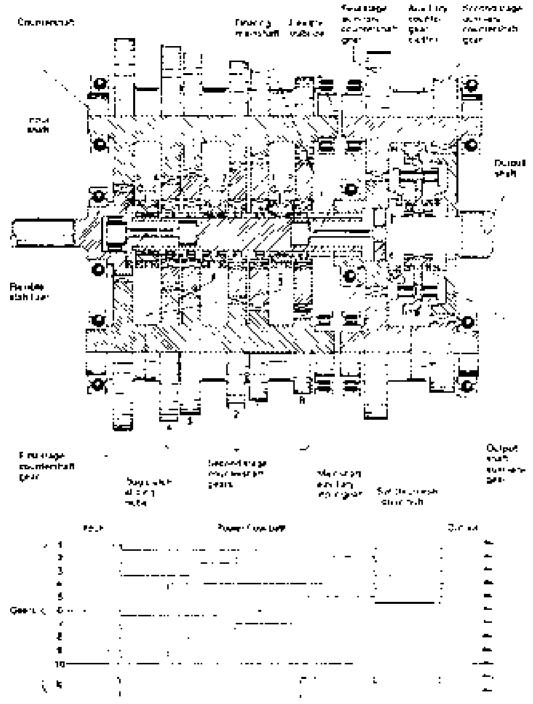**Fig. 3.23** Sixteen speed synchromesh with range change and integral splitter gears

Fig. 3.24 Twin countershaft ten speed constant mesh gearbox with synchromesh two speed range change

adopting the two countershaft constant mesh gearbox which incorporates a synchromesh two speed rear mounted range change (Fig. 3.24).

The main gearbox is in fact a double stage compound conventional gearbox using two countershafts (layshaft) instead of the normal single arrangement.

esign and construction   Referring to Fig. 3.24, there is a countershaft either side of the mainshaft and they are all in the same plane. What cannot be seen is that this single plane is inclined laterally at 19 to the horizontal to reduce the overall height of the gearbox.

The mainshaft is hollow and is allowed to float in the following manner: each end is counterbored, and into each counterbore is pressed a stabilizing rod. The front end of this rod pro ects into the rear of the input shaft which is also counterbored to house a supporting roller bearing for the stabilizer rod. The rear pro ecting stabilizer rod has a spherically shaped end which rests in a hole in the centre of a steel disc mounted inside the auxiliary drive gear immediately behind the mainshaft. This gear itself is carried by a ball bearing mounted in the gearbox housing. When torque is transmitted through the gearbox, the centrally waisted 11 mm diameter section of both stabilizers deflects until radial loads applied by the two countershaft gears to the mainshaft gear are equalized. By these means, the input torque is divided equally between the two countershafts and two diametrically opposite teeth on the mainshaft gear at any one time. Therefore, the face width of the gear teeth can be reduced by about 40% compared to gearboxes using single countershafts. Another feature of having a mainshaft which is relatively free to float in all radial directions is that it greatly reduces the dynamic loads on the gear teeth caused by small errors of tooth profile during manufacture. A maximum radial mainshaft float of about 0.75 mm has proved to be sufficient to permit the shaft to centralize and distribute the input torque equally between the two countershafts. To minimize end thrust, all the gears have straight spur teeth which run acceptably quietly due to the balanced loading of the gears.

Each of the five forward speeds and reverse are engaged by dog teeth clutches machined on both ends of the drive hubs. The ends of the external teeth on the drive hubs and the internal teeth in the mainshaft gears are chamfered at about 35 to provide some self-synchronizing action before engagement.

Power flow path   Power flows into the main gearbox through the input first motion shaft and gear wheel.   ere it is divided between the two first stage countershaft gears and is then conveyed via each countershaft gear wheel to the corresponding second stage mainshaft gears. Each of these rotate at relative speeds about the mainshaft. Torque is only transmitted to the mainshaft when the selected dog clutch drive hub is slid in to mesh with the desired gear dog teeth.

The power flow can then pass directly to the output shaft by engaging the synchromesh high range dog teeth. Conversely, a further gear reduction can be made by engaging the low range synchromesh dog teeth so that the power flow from the mainshaft auxiliary gear is split between the two auxiliary countershafts. The additional speed reduction is then obtained when the split power path comes together through the second stage auxiliary output gear. It should be observed that, unlike the mainshaft, the auxiliary gear reduction output shaft has no provision for radial float.

Reverse gear is obtained by incorporating an idle gear between the second stage countershaft reverse gears and the mainshaft reverse gear so that the mainshaft reverse gear is made to rotate in the opposite direction to all the other forward drive mainshaft gears.

3.   Transfer bo power take-off (PT) (Fig. 3.25)
A power take-off (PT) provides some shaft drive and coupling to power specialized auxiliary equipment at a specified speed and power output. Power take-offs (PT s) can be driven directly from the engine s timing gears, but it is more usual and practical to take the drive from some point off the gearbox. Typical power take-off applications are drives for hydraulic pumps, compressors, generators, hoists, derricks, capstain or cable winch platform elevators, extended ladders, hose reels, drain cleaning vehicles, tippers, road sweepers, snow plough blade and throwing operations and any other mechanical mechanism that needs a separate source of power drive output.

The power take-off can be driven either by one of the layshaft cluster gears, so that it is known as a   , or it may be driven from the back end of the layshaft, in which case it is known as a   (Fig. 3.25).

Transfer boxes can either be single or two speed arrangements depending upon the intended application. The gear ratios of the transfer box are so chosen that output rotational speeds may be anything from 50 to 150% of the layshaft input speed.
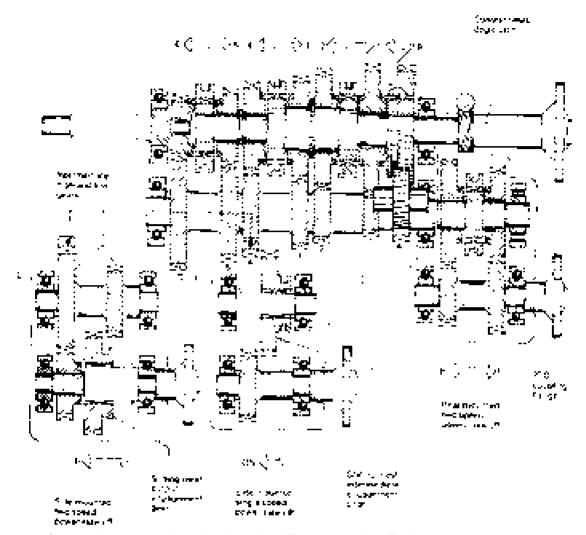
Fig. 3.25 Six speed constant mesh gearbox illustrating different power take-off point arrangement

3.6.1 Side mounted single speed transfer bo
(Fig. 3.25)

With the single speed side mounted transfer box, the drive is conveyed to the output gear and shaft by means of an intermediate gear mounted on a splined idler shaft which is itself supported by two spaced out ball bearings (Fig. 3.25). Engagement of the transfer output shaft is obtained by sliding the intermediate straight toothed gear into mesh with both layshaft gear and output shaft gear by a selector fork mounted on a gear shift not shown.

3.6. Side mounted t o speed transfer bo
(Fig. 3.25)

If a more versatile transfer power take-off is required, a two speed transfer box can be incorpor-

ated. With this gear train layout, the drive is conveyed to the intermediate shaft by a gear wheel which is in constant mesh with both the layshaft gear wheel and the high speed output gear (Fig. 3.25). The output shaft supports the high speed output gear which is free to revolve relative to it when the transfer drive is in neutral or low gear is engaged. Also attached to this shaft on splines is the low speed output gear.

igh transfer gear ratio engagement is obtained by sliding the low speed output gear towards the high speed output gear until its internal splines mesh with the dog teeth on the side of the gear. This then transfers the drive from the layshaft to the output shaft and coupling through a simple single stage gear reduction.

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ow transfer gear ratio engagement occurs when the low speed output gear is slid into mesh with the smaller intermediate shaft gear. The power flow then takes place through a double stage (compound) gear reduction.

ear mounted t o speed transfer bo (Fig. 3.25) In some gearbox designs, or where the auxiliary equipment requires it, a rear mounted transfer box may be more convenient. This transfer drive arrangement uses either an extended monolayshaft or a short extension shaft attached by splines to the layshaft so that it protrudes out from the rear of the gearbox (Fig. 3.25). The extended layshaft supports a pair of high and low speed gears which are in permanent mesh with corresponding gears mounted on the output shaft.

When the transfer box is in neutral, the gears on the extended layshaft are free to revolve independently on this shaft. Engagement of either high or low gear ratios is achieved by sliding the output drive hub sleeve in to mesh with one or other sets of ad acent dog teeth forming part of the transfer box layshaft constant mesh gears. Thus high gear ratio power flow passes from the layshaft to the constant mesh high range gears to the output shaft and coupling. Conversely, low gear ratio power transmission goes from the layshaft through the low range gears to the output drive.

3.7 verdrive considerations

Power is essential to propel a vehicle because it is a measure of the rate of doing work, that is, the amount of work being developed by the engine in unit time. With increased vehicle speed, more work has to be done by the engine in a shorter time.

The characteristic power curve over a speed range for a petrol engine initially increases linearly and fairly rapidly. Towards mid-speed the steepness of the power rise decreases until the curve reaches a peak. It then bends over and declines with further speed increase due to the difficulties experienced in breathing at very high engine speeds (Fig. 3.26).

A petrol engined car is usually geared so that in its normal direct top gear on a level road the engine speed exceeds the peak power speed by about 10 to 20% of this speed. Consequently, the falling power curve will intersect the road resistance power curve. The point where both the engine and road resistance power curves coincide fixes the road speed at which all the surplus power has been absorbed. Therefore it sets the maximum possible vehicle speed.

By selecting a 20% overdrive top gear, say, the transmission gear ratios can be so chosen that the engine and road resistance power curves coincide at peak engine power (Fig. 3.26). The undergearing has thus permitted the whole of the engine power curve to be shifted nearer the opposing road resistance power curve so that slightly more engine power is being utilized when the two curves intersect. As a result, a marginally higher maximum vehicle speed is achieved. In other words, the engine will be worked at a lower speed but at a higher load factor whilst in this overdrive top gear.

If the amount of overdrive for top gear is increased to 40%, the engine power curve will be shifted so far over that it intersects the road resistance power curve before peak engine power has been obtained (Fig. 3.26) and therefore the maximum possible vehicle speed cannot be reached.

Contrasting the direct drive 20% and 40% overdrive with direct drive top gear power curves with respect to the road resistance power curve at 70 km/h, as an example, it can be seen (Fig. 3.26) that the reserve of power is 59%, 47% and 38% respectively. This surplus of engine power over the power absorbed by road resistance is a measure of the relative acceleration ability for a particular transmission overall gear ratio setting.

A comparison of the three engine power curves shows that with direct drive top gear the area in the loop made between the developed and opposing power curves is the largest and therefore the engine would respond to the changing driving conditions with the greatest flexibility.

If top gear is overdriven by 20%, as shown in Fig. 3.26, the maximum engine power would be developed at maximum vehicle speed. This then provides the highest possible theoretical speed, but the amount of reserve power over the road resistance power is less, so that acceleration response will not be as rapid as if a direct drive top gear is used. perating under these conditions, the engine speed would never exceed the peak power speed and so the engine could not over-rev , and as a result engine wear and noise would be reduced. Benefits are also gained in fuel consumption as shown in Fig. 3.26. The lowest specific fuel consumption is shifted to a higher cruising speed which is desirable on motorway ourneys.

Indulging in an excessive 40% overdrive top gear prevents the engine ever reaching peak power so that not only would maximum vehicle speed be reduced compared to the 20% overdrive gearing, but the much smaller difference in power developed to power dissipated shown on the power curves would
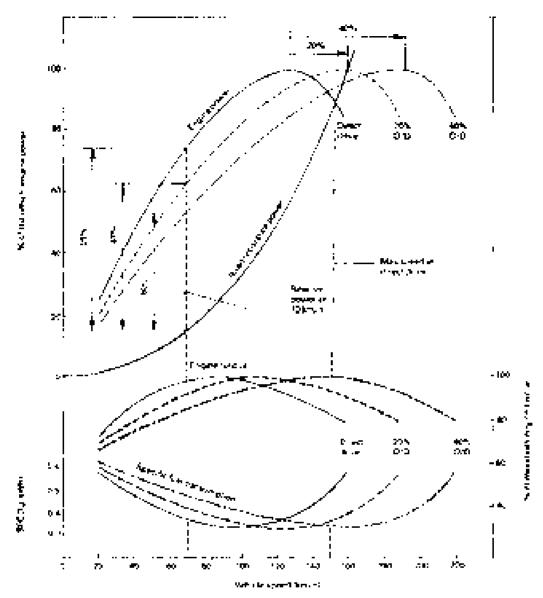
Fig. 3.26 Effect of over and undergearing on vehicle performance

severely reduce the flexibility of driving in this gear. It therefore becomes essential for more frequent down changes with the slightest fall-off in road speed. A further disadvantage with excessive overdrive is that the minimum specific fuel consumption would be shifted theoretically to the engine s upper speed range which in practice could not be reached.

An analysis of matching an engine s performance to suit the driving requirements of a vehicle shows that with a good choice of undergearing in top gear for motorway cruising conditions, benefits of

prolonged engine life, reduced noise, better fuel economy and less driver fatigue will be achieved. Another ma or consideration is the unladen and laden operation of the vehicle, particularly if it is to haul heavy loads. Therefore most top gear overdrive ratios are arrived at as a compromise.

3.7.1 picyclic overdrive gearing

Epicyclic gear train overdrives are so arranged that the input shaft drives the pinion carrier while the output shaft is driven by the annular gear ring

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(Figs 3.27 and 3.28). The gear train may be either of simple (single stage) or compound (double stage) design and the derived formula for each arrangement is as follows:

(Fig. 3.27)

$$\text{verdrive gear ratio} = \frac{}{}$$

also $= 2$

where $=$ number of annulus ring gear teeth

$=$ number of sun gear teeth

$=$ number of planet gear teeth

(Fig. 3.28)

$$\text{verdrive gear ratio} = \frac{(\quad\text{s}\quad)}{(\quad\text{s}\quad)\quad\text{s}}$$

also $= \text{s}$

where $=$ number of annulus ring gear teeth

$=$ number of sun gear teeth

$_s =$ number of small planet gear teeth

$=$ number of large planet gear teeth

The amount of overdrive (undergearing) used for cars, vans, coaches and commercial vehicles varies from as little as 15% to as much as 45%. This corresponds to undergearing ratios of between 0.87:1 and 0.69:1 respectively. Typical overdrive ratios which have been frequently used are 0.82:1 (22%), 0.78:1 (28%) and 0.75:1 (37%).

An overdrive simple epicyclic gear train has sun and annulus gears with 21 and 75 teeth respectively. If the input speed from the engine drives the planet carrier at 3000 rev/min, determine

a) the overdrive gear ratio,
b) the number of planet gear teeth,
c) the annulus ring and output shaft speed,
d) the percentage of overdrive.

a) $\quad\text{verdrive gear ratio} = \frac{}{} = \frac{75}{75 \quad 21}$

$$= \frac{75}{96} = 0.78125$$

b) $\quad = \quad 2$

Therefore $\quad = \dfrac{}{2}$

$$= \frac{75 \quad 21}{2}$$

$$= \frac{54}{2} = 27 \text{ teeth}$$
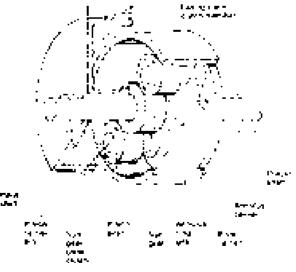

Fig. 3.27 Simple epicycle overdrive gear train

c) output speed $= \dfrac{3000}{0.78125} = 3840 \,\text{rev/min}$

d) Percentage of overdrive $= \left(\dfrac{3840 \quad 3000}{3000}\right)100$

$$= \dfrac{840 \times 100}{3000} = 28\%$$

A compound epicyclic gear train over-drive has sun, small planet and large planet gears with 21, 15 and 24 teeth respectively. etermine the following if the engine drives the input planet carrier at 4000 rev/min.

a) The overdrive gear ratio,
b) the number of annulus ring gear teeth,
c) the annulus ring and output shaft speed,
d) the percentage of overdrive.

a) $\begin{aligned}\text{verdrive} \\ \text{gear ratio}\end{aligned} = \dfrac{(\qquad \text{s}\qquad)}{(\quad \text{s}\quad) \quad \text{s}}$

$$= \dfrac{24\,(24\quad 15\quad 21)}{24\,(24\quad 15\quad 21)\quad(15 \times 21)}$$

$$= \dfrac{24 \times 60}{(24 \times 60)\quad 315} = \dfrac{1400}{1755}$$

$$= 0.82$$

b) $\quad = \qquad \text{s}$
$\quad = 21 \quad 24 \quad 15$
$\quad = 60 \,\text{teeth}$

c) output speed $= \dfrac{4000}{0.82} = 4878 \,\text{rev/min}$

d) Percentage of overdrive $= \dfrac{4878 \quad 4000}{4000} \times 100$

$$= \dfrac{878 \times 100}{4000}$$

$$= 21.95\%$$

3.7. *Simple epicyclic overdrive gear train*
(Fig. 3.27)

If the sun gear is prevented from rotating and the input shaft and planet carrier are rotated, the pinion gears will be forced to revolve around the fixed sun gear and these pinions will revolve simultaneously on their own axes provided by the carrier pins.

As a result, motion will be transferred from the carrier and pinion gears to the annulus ring gear due to the separate rotary movement of both the planet carrier and the revolving planet gears, thus
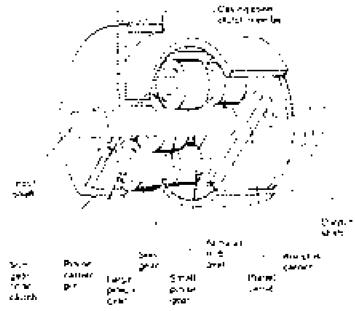

Fig. 3.28 Compound epicycle gear train

the annulus and therefore the output shaft will be compelled to revolve at a slightly faster speed.

3.7.3 Compound epicyclic overdrive gear train
(Fig. 3.28)
For only small degrees of overdrive (undergearing), for example 0.82:1 (22%), the simple epicyclic gearing would need a relatively large diameter annulus ring gear about 175 mm if the dimension of the gear teeth are to provide adequate strength. A way of reducing the diameter of the annulus ring gear for a similar degree of overdrive is to utilize a compound epicyclic gear train which uses double pinion gears on each carrier pin instead of one size of pinion. By this method, the annulus diameter is reduced to about 100 mm and there are only 60 teeth compared to the 96 teeth annulus used with the simple epicyclic gear train.

To transmit power, the sun gear is held still whilst the input shaft and planet carrier are rotated. This compels the large planet gear to roll around the stationary sun gear and at the same time forces each pair of combined pinion gears to revolve about their carrier pin axes.

Consequently the small pinion gear will impart both the pinion carrier orbiting motion and the spinning pinion gear motion to the annulus ring gear so that the output shaft will be driven at a higher speed to that of the input shaft.

3.7.   aycoc   simple gear train overdrive

escription (Fig. 3.29) The overdrive unit is attached to the rear of the gearbox and it consists of a constant mesh helical toothed epicyclic gear train which has a central sunwheel meshing with three planet gears which also mesh with an internally toothed annulus gear. The planet gears are supported on a carrier driven by the input shaft whilst the annulus is attached to the output shaft via a carrier forming an integral part of both members. A double cone clutch selects the different ratios when engaged one side of the clutch provides direct drive and when the other side is used, overdrive.

irect drive (Fig. 3.29)   irect drive is obtained when the inner cone clutch engages with the outer cone of the annular gear. Power will then be conveyed via the unidirectional clutch to the output shaft by means of the rollers which are driven up inclined ramps and wedged between the inner and outer clutch members. When the vehicle overruns

the engine, the output shaft will try to run faster than the input shaft and so tend to release the unidirectional clutch rollers, but this is prevented by the inner cone clutch locking the sun gear to the annulus, thereby amming the sun, planet and annular epicyclic gear train so that they cannot revolve relative to each other.

Engagement of the inner cone clutch to the external cone surface of the annulus gear is provided by four stationary thrust springs (only one shown) which are free to exert their axial load against a thrust plate. This in turn transfers thrust by way of a ball bearing to the rotating cone clutch support member splined to the sun gear sleeve. This overrun and reverse torque will be transmitted between the engine and transmission in direct drive.

wing to the helical cut teeth of the gear wheels, an end thrust exists between the planet gears and the sun gear during overrun and reverse which tends to push the latter rearwards. Therefore, additional clamping load between the cone clutch faces is necessary.

verdrive (Fig. 3.29)   When overdrive is engaged, the cone clutch, which is supported on the splined sleeve of the sun gear, is moved over so that its outer friction facing is in contact with the internal cone brake attached to the casing. Consequently the sun gear is held stationary. With the sun gear held still and the input shaft and planet carrier rotating, the planet gears are forced to rotate about their own axes and at the same time roll around the fixed sun gear, with the result that the annulus gear is driven at a faster speed than the input shaft. This causes the unidirectional clutch outer member (annular carrier) to overrun the inner member (planet carrier) so that the wedged rollers on their ramps are released. Pulling the cone clutch away from the annulus cone and into frictional contact with the brake casing cone against the axial load of the six thrust springs is achieved by means of hydraulic oil pressure. This pressure acts upon two slave pistons (only one shown) when a valve is opened by operating the driver controlled selector switch.

The outward movement of the slave pistons, due to the hydraulic pressure, draws the stationary thrust plate, ball bearing and rotating clutch member away from the annular cone and into engagement with the outer brake cone, thereby locking the sun gear to the casing. f the helix angle of the gear teeth, the torque reaction tends to push the sun gear forward so that extra end
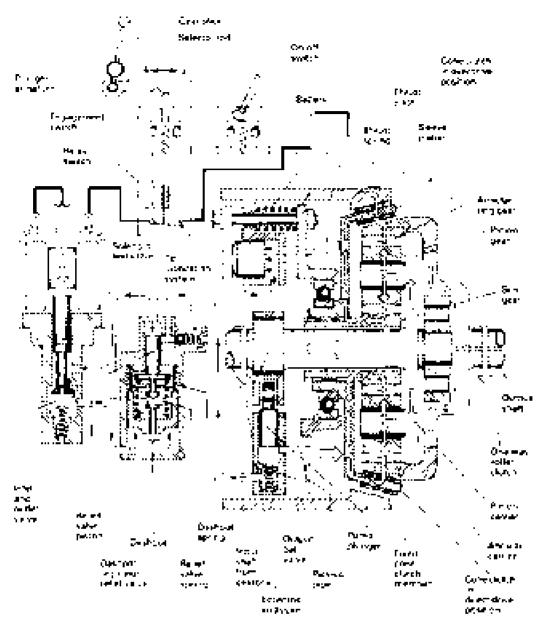
Fig. 3.29 aycock single epicycle overdrive

thrust is necessary to maintain sufficient clamping thrust between the frictional faces of the cones in the brake position.

irect and overdrive controlled gear change action (Fig. 3.29)   When direct drive is selected, hydraulic pressure is steadily increased and this gradually releases the double-sided cone clutch member from the cone brake fixed to the casing. The release of the cone clutch frees the sun gear and removes the load from the engine. The engine speed increases immediately until it catches up with the output shaft, at which point the unidirectional clutch rollers climb up their respective ramps and am. The input shaft s power coming from the engine is now permitted to drive the output shaft, which in turn transmits drive to the propellor shaft. At the same time the double-sided cone clutch completes its movement and engages the annular ring cone.

verdrive is engaged when the double-sided cone clutch moves away from the annulus gear cone and makes contact with the stationary cone brake, thus bringing the sliding cone clutch member and sun gear to rest. As a result of the sun gear being held stationary, the gears now operate as an epicyclic step up gear ratio transmission. uring the time the double-sided cone clutch member moves from the annulus cone to the brake cone the clutch will slip. This now permits the unidirectional roller clutch to transmit the drive. Whilst the input ramp member rotates as fast as the output ramp member the roller clutch drives. owever, as the annular ring gear speed rises above that of the input shaft, the rollers will disengage themselves from their respective ramps thereby diverting the drive to the epicyclic gear train.

Electrical system (Fig. 3.29) verdrive or direct drive gear ratio selection is controlled by an electrical circuit which includes an overdrive on/off switch, inhibitor switch and a relay switch. An inhibitor switch is incorporated in the circuit to prevent the engagement of overdrive in reverse and some or all of the indirect gears. A relay switch is also included in the circuit so that the overdrive on/off switch current rating may be small compared to the current draw requirements of the control solenoid. The overdrive may be designed to operate only in top gear, but sometimes the overdrive is permitted to be used in third or even second gear. Selection and engagement of overdrive by the driver is obtained by a steering column or fascia panel switch. When the driver selects overdrive in top gear or one of the permitted indirect gears, say third, the on/off switch is closed and the selected gearbox gear ratio selector rod will have pushed the inhibitor switch button into the closed switch position. Current will now flow from the battery to the relay switch, magnetizing the relay winding so that as the relay contacts close, a larger current will immediately energize the solenoid and open the control valve so that overdrive will be engaged.

ydraulic system (Fig. 3.29) A plunger type pump driven by an eccentric formed on the input shaft supplies the hydraulic pressure to actuate the slave pistons and thereby operates the clutch. The pump draws oil from the sump through a filter (not shown). It is then pressurized by the plunger and delivered through a non-return valve to both slave cylinders (only one shown) and also to the solenoid controlled valve and dashpot regulator relief valve. The dashpot pressure regulator ensures a smooth overdrive engagement and disengagement under differing operating conditions. When in direct drive the pump to slave cylinder s line pressure is determined by the regulator relief valve spring tension which controls the blow-off pressure of the oil escaping to the lubrication system. This line residual pressure in direct drive is normally maintained at about 2.8 bar, but when engaging overdrive it is considerably raised by the action of the dashpot to about 20–40 bar.

verdrive engagement Energizing the solenoid draws down the armature, thereby opening the inlet valve and closing the outlet valve. il at residual line pressure will now pass through the control orifice to the base of the dashpot regulator relief valve causing the dashpot to rise and compress both the dashpot spring and relief valve spring. Consequently, the pump to slave cylinder pressure circuit will gradually build up as the dashpot spring shortens and increases in stiffness until the dashpot piston has reached its stop, at which point the operating pressure will be at a maximum. It is this gradual increase in line pressure which provides the progressive compression of the clutch thrust springs and the engagement of the cone clutch with the fixed cone brake.

irect drive engagement e-energizing the solenoid closes the inlet valve and opens the outlet valve. This prevents fresh oil entering the dashpot cylinder and allows the existing oil under the dashpot to exhaust by way of the control orifice and the outlet valve back to the sump. The control orifice restricts the flow of escaping oil so that the pressure drop is progressive. This enables the clutch thrust springs to shift the cone clutch very gradually into contact with the annulus cone.

3.7.5 *aycoc compound gear train overdrive* (Fig. 3.30)

verdrive When overdrive is selected, the double-sided cone clutch contacts the brake cone which forms part of the casing. This brings the sun gear which is attached to the sliding clutch member to a standstill.

The input drive passes from the pinion carrier to the annulus ring and hence to the output shaft through the small planet gear. At the same time, the
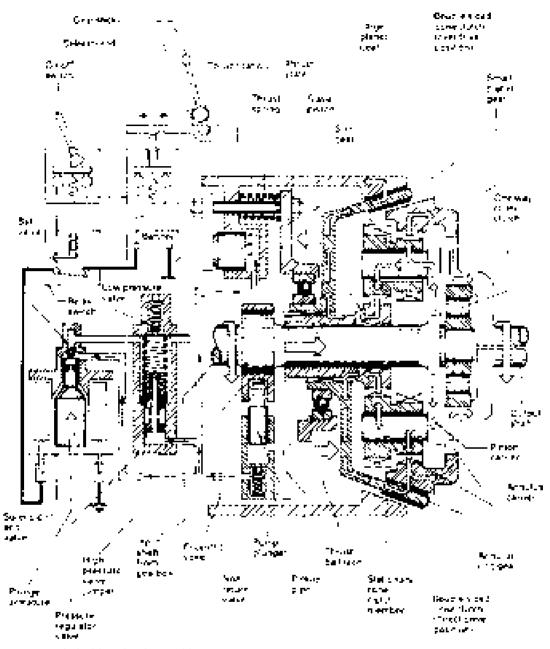
Fig. 3.30 aycock double epicycle overdrive

large planet gear absorbs the driving torque reaction and in the process is made to revolve around the braked sun gear. The overdrive condition is created by the large planet gears being forced to roll walk about the sun gear, while at the same time revolving on their own axes. As a result, the small planet gears, also revolving on the same carrier pins as the large planet gears, drive forward the annular ring gear at a faster speed relative to that of the input.

The overall gear ratio step up is achieved by having two stages of meshing gear teeth one between the large pinion and sun gear and the other between the small pinion and annulus ring gear. By using this compound epicyclic gear train, a

relatively large step up gear ratio can be obtained for a given diameter of annulus ring gear compared to a single stage epicyclic gear train.

irect drive (Fig. 3.30)   irect drive is attained by releasing the double-sided cone clutch member from the stationary conical brake and shifting it over so that it contacts and engages the conical frictional surface of the annulus ring gear. The power flow from the input shaft and planet carrier now divides into two paths   the small planet gear to annulus ring gear route and the large planet gear, sun gear and double-sided clutch member route, again finishing up at the annulus ring gear. With such a closed loop power flow arrangement, where the gears cannot revolve independently to each other, the gears am so that the whole gear train combination rotates as one about the input to output shaft axes. It thereby provides a straight through direct drive. It should be observed that the action of the unidirectional roller clutch is similar to that described for the single stage epicyclic overdrive.

lutch operating (Fig. 3.30) Engagement of direct drive and overdrive is achieved in a similar manner to that explained under single stage epicyclic overdrive unit.
irect drive is provided by four powerful springs holding the double-sided conical clutch member in frictional contact with the annulus ring gear. Conversely, overdrive is obtained by a pair of hydraulic slave pistons which overcome and compress the clutch thrust springs, pulling the floating conical clutch member away from the annulus and into engagement with the stationary conical brake.

ydraulic system (Fig. 3.30)   Pressure supplied by the hydraulic plunger type pump draws oil from the sump and forces it past the non-return ball valve to both the slave cylinders and to the solenoid valve and the relief valve.

irect drive engagement When direct drive is engaged, the solenoid valve opens due to the solenoid being de-energized.   il therefore flows not only to the slave cylinders but also through the solenoid ball valve to the overdrive lubrication system where it then spills and returns to the sump. A relatively low residual pressure will now be maintained within the hydraulic system. Should the oil pressure rise due to high engine speed or blockage,

the low pressure ball valve will open and relieve the excess pressure. Under these conditions the axial load exerted by the clutch thrust springs will clamp the double-sided floating conical clutch member to the external conical shaped annulus ring gear.

verdrive engagement To select overdrive the solenoid is energized. This closes the solenoid ball valve, preventing oil escaping via the lubrication system back to the sump.   il pressure will now build up to about 26–30 bar, depending on vehicle application, until sufficient thrust acts on both slave pistons to compress the clutch thrust springs, thereby permitting the double-sided clutch member to shift over and engage the conical surface of the stationary brake. To enable the engagement action to overdrive to progress smoothly and to limit the maximum hydraulic pressure, a high pressure valve umper is made to be pushed back and progressively open. This controls and relieves the pressure rise which would otherwise cause a rough, and possibly sudden, clutch engagement.

3.   Setting gear ratios
Matching the engine s performance characteristics to suit a vehicle s operating requirements is provided by choosing a final drive gear reduction and then selecting a range of gear ratios for maximum performance in terms of the ability to climb gradients, achievement of good acceleration through the gears and ability to reach some predetermined maximum speed on a level road.

3. .1 Setting top gear
To determine the maximum vehicle speed, the engine brake power curve is superimposed onto the power requirement curve which can be plotted from the sum of both the rolling ($_r$) and air ($_a$) resistance covering the entire vehicle s speed range (Fig. 3.31).
The total resistance   opposing motion at any speed is given by:

$$= \ _r \quad _a$$
$$= 10 \ _r W \qquad\qquad ^2$$

where C_r = coefficient of rolling resistance
W = gross vehicle weight (kg)
= coefficient of aerodynamic resistance (drag)
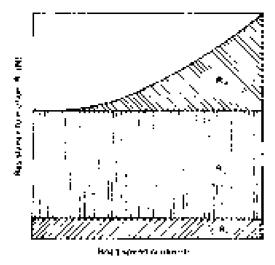= pro ected frontal area of vehicle (m^2)
= speed of vehicle (km/h)

Fig. 3.31 Forces opposing vehicle motion over its speed range

Fig. 3.32 Relationship of power developed and road power re uired over the vehicle s speed range

The top gear ratio is chosen so that the maximum road speed corresponds to the engine speed at which maximum brake power is obtained (or ust beyond) (Fig. 3.32).

earing is necessary to ensure that the vehicle speed is at a maximum when the engine is developing approximately peak power.

Thus   inear wheel speed =   inear road speed

$$\frac{}{_F} = \frac{1000}{60} \quad \text{(m/min)}$$

$\therefore$ Final drive gear ratio $_F = \dfrac{60}{100}$

$$= 0.06 \ \underline{\quad}$$

where $_F$ = final drive gear ratio
            = engine speed (rev/min)
            = effective wheel diameter (m)
            = road speed at which peak power is
               developed (km/h)

A vehicle is to have a maximum road speed of 150 km/h. If the engine develops its peak power at 6000 rev/min and the effective road wheel diameter is 0.54 m, determine the final drive gear ratio.

$$_F = \frac{0.06 \ \underline{\quad}}{}$$

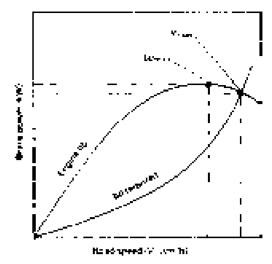$$= \frac{0.06 \times 3.142 \times 0.54 \times 6000}{150}$$

$$= 4.07 : 1$$

3. . *Setting bottom gear*

The maximum payload and gradient the vehicle is expected to haul and climb determines the necessary tractive effort, and hence the required overall gear ratio. The greatest gradient that is likely to be encountered is decided by the terrain the vehicle is to operate over. This normally means a maximum gradient of 5 to 1 and in the extreme 4 to 1. The minimum tractive effort necessary to propel a vehicle up the steepest slope may be assumed to be approximately equivalent to the sum of both the rolling and gradient resistances opposing motion (Fig. 3.31).

The rolling resistance opposing motion may be determined by the formula:

$$_r = 10 \ _r W$$

where $_r$ = rolling resistance (N)
            $_r$ = coefficient of rolling resistance
            W = gross vehicle weight (kg)

Average values for the coefficient of rolling resistance for different types of vehicles travelling at very slow speed over various surfaces have been determined and are shown in Table 3.2.

ikewise, the gradient resistance (Fig. 3.33) opposing motion may be determined by the formula:

$$_g = \frac{10W}{} \quad \text{or} \quad 10W \sin$$

where $_g$ = gradient resistance (N)
            W = gross vehicle weight ($10W$ kg = WN)
               = gradient (1 in x) = sin

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Tab 3.2 Average values of coefficient of rolling resistance

Vehicle type	Coefficient of rolling resistance (C_r)		
	Concrete	Medium hard soil	Sand
Passenger Car	1		
Trucks	12		2
Tractors	2		2

Note The coefficient of rolling resistance is the ratio of the rolling resistance to the normal load on the tyre
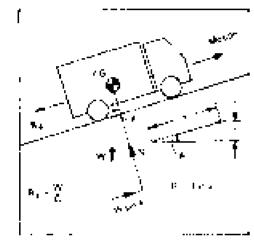
i.e. $C_r = \dfrac{r}{L}$

Fig. 3.33 Gradient resistance to motion

Tractive effort = Resisting forces opposing motion

$$= F_r + F_g \ (N)$$

where F = tractive effort (N)

R = resisting forces (N)

Once the minimum tractive effort has been calculated, the bottom gear ratio can be derived in the following way:

Driving torque = Available torque

$$= G_B \ G_F \ \eta_M$$

∴ Bottom gear ratio $G_B = \dfrac{F \, r}{T \, G_F \, \eta_M}$

where G_F = final drive gear ratio

G_B = bottom gear ratio

η_M = mechanical efficiency

F = tractive effort (N)

T = maximum engine torque (Nm)

r = effective road wheel radius (m)

A vehicle weighing 1500 kg has a coefficient of rolling resistance of 0.015. The transmission has a final drive ratio 4.07:1 and an overall mechanical efficiency of 85%.

If the engine develops a maximum torque of 100 Nm (Fig. 3.34) and the effective road wheel radius is 0.27 m, determine the gearbox bottom gear ratio.

Assume the steepest gradient to be encountered is a one in four.

$$F_r = 10 \, C_r W$$

$$= 10 \times 0.015 \times 1500 = 225 \, \text{N}$$

$$F_g = \frac{10 W}{G}$$

$$= \frac{10 \times 1500}{4} = 3750 \, \text{N}$$

$$F = F_r + F_g$$

$$= 3750 + 225 = 3975 \, \text{N}$$

$$G_B = \frac{F \, r}{T \, G_F \, \eta_M}$$

$$= \frac{3975 \times 0.27}{100 \times 4.07 \times 0.85} = 3.1:1$$
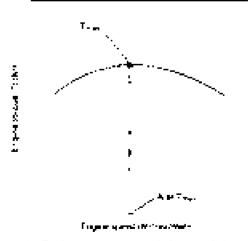

Fig. 3.34 Engine torque to speed characteristics

3. .3 Setting intermediate gear ratios

Ratios between top and bottom gears should be spaced in such a way that they will provide the tractive effort–speed characteristics as close to the ideal as possible. Intermediate ratios can be best selected as a first approximation by using a geometric progression. This method of obtaining the gear ratios requires the engine to operate within the same speed range in each gear, which is normally selected to provide the best fuel economy.

Consider the engine to vehicle speed characteristics for each gear ratio as shown (Fig. 3.35). When changing gear the engine speed will drop from the highest to the lowest without any change in road speed, i.e. G_1, G_2, G_3 etc.

Let G_1 = 1st overall gear ratio
G_2 = 2nd overall gear ratio
G_3 = 3rd overall gear ratio
G_4 = 4th overall gear ratio
G_5 = 5th overall gear ratio

where

$$\text{Overall gear ratio} = \frac{\text{Engine speed (rev/min)}}{\text{Road wheel speed (rev/min)}}$$

Wheel speed when engine is on the high limit N_H in first gear $n_1 = \dfrac{N_H}{G_1}$ (rev/min)

Wheel speed when engine is on the low limit N_L in second gear $n_2 = \dfrac{N_L}{G_2}$ (rev/min)

These wheel speeds must be equal for true rolling

Hence $\dfrac{N_H}{G_1} = \dfrac{N_L}{G_2}$

$$\therefore \quad G_2 = G_1\frac{N_L}{N_H}$$

Also $\dfrac{N_H}{G_2} = \dfrac{N_L}{G_3}$

$$\therefore \quad G_3 = G_2\frac{N_L}{N_H}$$

and $\dfrac{N_H}{G_3} = \dfrac{N_L}{G_4}$

$$\therefore \quad G_4 = G_3\frac{N_L}{N_H}$$

$$\dfrac{N_H}{G_4} = \dfrac{N_L}{G_5}$$

$$\therefore \quad G_5 = G_4\frac{N_L}{N_H}$$

The ratio $\dfrac{N_L}{N_H}$ is known as the minimum to maximum speed range ratio K for a given engine.

Now, gear $G_2 = G_1\dfrac{N_L}{N_H} = G_1 K$,

since $\dfrac{N_L}{N_H} = K$ (a constant)

gear $G_3 = G_2\dfrac{N_L}{N_H} = G_2 K = (G_1 K)K$
$$= G_1 K^2$$

gear $G_4 = G_3\dfrac{N_L}{N_H} = G_3 K = (G_1 K^2)K$
$$= G_1 K^3$$

gear $G_5 = G_4\dfrac{N_L}{N_H} = G_4 K = (G_1 K^3)K$
$$= G_1 K^4.$$

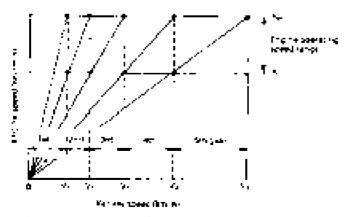Hence the ratios form a geometric progression.

Fig. 3.35 Gear ratios selected on geometric progression

The following relationship will also apply for a five speed gearbox:

$$\frac{R_2}{R_1} = \frac{R_3}{R_2} = \frac{R_4}{R_3} = \frac{R_5}{R_4} = \lambda =$$

and $\quad R_5 = R_1 \lambda^4 \quad$ or $\quad \lambda^4 = \dfrac{R_5}{R_1}$

ence $\quad \lambda = \left[\dfrac{R_5}{R_1}\right]^{\frac{1}{4}} \quad$ or $\quad \sqrt[4]{\dfrac{R_5}{R_1}}$

In general, if the ratio of the highest gear (R_T) and that of the lowest gear (R_B) have been determined, and the number of speeds (gear ratios) of the gearbox S is known, the constant λ can be determined by:

$$\lambda = \left[\frac{R_T}{R_B}\right]^{\frac{1}{S-1}}$$

So $\quad \dfrac{R_T}{R_B} = \lambda^{S-1}$

$\therefore \quad R_T = R_B \, \lambda^{S-1}$

For commercial vehicles, the gear ratios in the gearbox are often arranged in geometric progression. For passenger cars, to suit the changing traffic conditions, the step between the ratios of the upper two gears is often closer than that based on geometric progression. As a result, this will affect the selection of the lower gears to some extent.

A transmission system for a vehicle is to have an overall bottom and top gear ratio of 20:1 and 4.8 respectively. If the minimum to maximum speeds at each gear changes are 2100 and 3000 rev/min respectively, determine the following:

a) the intermediate overall gear ratios
b) the intermediate gearbox and top gear ratios.

$$\lambda = \frac{2100}{3000} = 0.7$$

a) 1st gear ratio $R_1 = R_B$.
2nd gear ratio $R_2 = R_1 \lambda = 20 \times 0.7 = 14.0{:}1$
3rd gear ratio $R_3 = R_1 \lambda^2 = 20 \times 0.7^2 = 9.8{:}1$
4th gear ratio $R_4 = R_1 \lambda^3 = 20 \times 0.7^3 = 6.86{:}1$
5th gear ratio $R_5 = R_1 \lambda^4 = 20 \times 0.7^4 = 4.8{:}1$

b) $\quad G_1 = \dfrac{20.0}{4.8} = 4.166{:}1$

$\quad G_2 = \dfrac{14.0}{4.8} = 2.916{:}1$

$\quad G_3 = \dfrac{9.8}{4.8} = 2.042{:}1$

$\quad G_4 = \dfrac{6.86}{4.8} = 1.429{:}1$

Top gear $\quad G_5 = \dfrac{4.8}{4.8} = 1.0{:}1$

4 Hydrokinetic fluid couplings and torque converters

A fluid drive uses hydrokinetic energy as a means of transferring power from the engine to the transmission in such a way as to automatically match the vehicle s speed, load and acceleration requirement characteristics. These drives may be of a simple two element type which takes up the drive smoothly without providing increased torque or they may be of a three or more element unit which not only conveys the power as required from the engine to the transmission, but also multiplies the output torque in the process.

4.1 Hydrokinetic fluid couplings
(Figs 4.1 and 4.2)

The hydrokinetic coupling, sometimes referred to as a fluid flywheel, consists of two saucer-shaped discs, an input impeller (pump) and an output turbine (runner) which are cast with a number of flat radial vanes (blades) for directing the flow path of the fluid (Fig. 4.1).

Owing to the inherent principle of the hydrokinetic coupling, there must be relative slip between the input and output member cells exposed to each
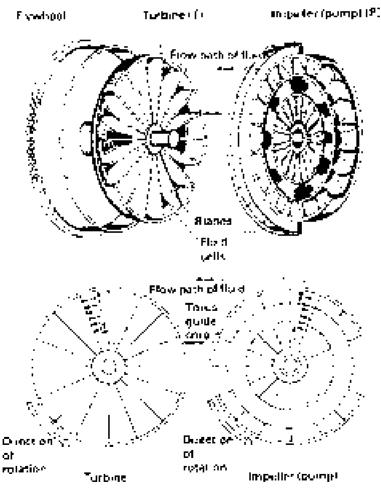

Fig. 4.1 Fluid coupling action

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other, and the vortex flow path created by pairs of adjacent cells will be continuously aligned and misaligned with different cells.

With equal numbers of cells in the two half members, the relative cell alignment of all the cells occurs together. Consequently, this would cause a jerky transfer of torque from the input to the output drive. By having differing numbers of cells within the impeller and turbine, the alignment of each pair of cells at any one instant will be slightly different so that the impingement of fluid from one member to the other will take place in various stages of circulation, with the result that the coupling torque transfer will be progressive and relatively smooth.

The two half-members are put together so that the fluid can rotate as a vortex. Originally it was common practice to insert at the centre of rotation a hollow core or guide ring (sometimes referred to as the) within both half-members to assist in establishing fluid circulation at the earliest moment of relative rotation of the members. These couplings had the disadvantage that they produced considerable drag torque whilst idling, this being due mainly to the effectiveness of the core guide in circulating fluid at low speeds. As coupling development progressed, it was found that turbine drag was reduced at low speeds by using only a core guide on the impeller member (Fig. 4.2). With the latest design
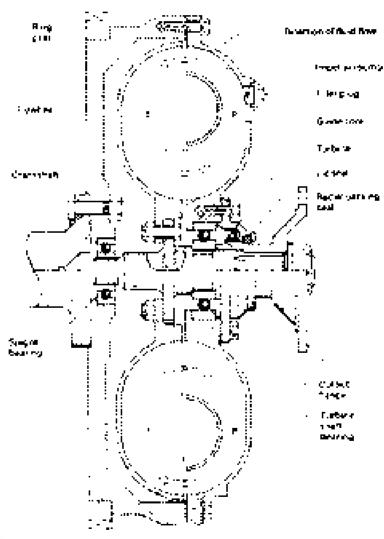

Fig. 4.2 Fluid coupling

these cores are eliminated altogether as this also reduces fluid interference in the higher speed range and consequently reduces the degree of slip for a given amount of transmitted torque (Fig. 4.6).

.1.1 ydro inetic fluid coupling principle of operation (Figs 4.1 and 4.3)

When the engine is started, the rotation of the (pump) causes the working fluid trapped in its cells to rotate with it. Accordingly, the fluid is sub ected to centrifugal force and is pressurized so that it flows radially outwards.

To understand the principle of the hydrokinetic coupling it is best to consider a small particle of fluid circulating between one set of impeller and turbine vanes at various points A, B, C and as shown in Figs 4.1 and 4.3.

Initially a particle of fluid at point A, when the engine is started and the impeller is rotated, will experience a centrifugal force due to its mass and radius of rotation, . It will also have acquired some kinetic energy. This particle of fluid will be forced to move outwards to point B, and in the process of increasing its radius of rotation from to , will now be sub ected to considerably more centrifugal force and it will also possess a greater amount of kinetic energy. The magnitude of the kinetic energy at this outermost position forces it to be e ected from the mouth of the impeller cell, its flow path making it enter one of the outer turbine cells at point C. In doing so it reacts against one side of the turbine vanes and so imparts some of its kinetic energy to the turbine wheel. The repetition of fluid particles being flung across the unction between the impeller and turbine cells will force the first fluid particle in the slower moving turbine member (having reduced centrifugal force) to move inwards to point . ence in the process of moving inwards from to , the fluid particle gives up most of its kinetic energy to the turbine wheel and subsequently this is converted into propelling effort and motion.

The creation and conversion of the kinetic energy of fluid into driving torque can be visualized in the following manner: when the vehicle is at rest the turbine is stationary and there is no centrifugal force acting on the fluid in its cells. owever, when the engine rotates the impeller, the working fluid in its cells flows radially outwards and enters the turbine at the outer edges of its cells. It therefore causes a displacement of fluid from the inner edges of the turbine cells into the inner edges of the impeller cells, thus a circulation of the fluid will be established between the two half cell members. The fluid has two motions firstly it is circulated

by the impeller around its axis and secondly it circulates round the cells in a vortex motion.

This circulation of fluid only continues as long as there is a difference in the angular speeds of the impeller and turbine, because only then is the centrifugal force experienced by the fluid in the faster moving impeller greater than the counter centrifugal force acting on the fluid in the slower moving turbine member. The velocity of the fluid around the couplings axis of rotation increases while it flows radially outwards in the impeller cells due to the increased distance it has moved from the centre of rotation. Conversely, the fluid velocity decreases when it flows inwards in the turbine cells. It therefore follows that the fluid is given kinetic energy by the impeller and gives up its kinetic energy to the turbine. ence there is a transference of energy from the input impeller to the output turbine, but there is no torque multiplication in the process.

.1. ydro inetic fluid coupling velocity diagrams (Fig. 4.3)

The resultant magnitude of direction of the fluid leaving the impeller vane cells, $_R$, is dependent upon the exit velocity, $_E$, this being a measure of the vortex circulation flow rate and the relative linear velocity between the impeller and turbine, .

The working principle of the fluid coupling may be explained for various operating conditions assuming a constant circulation flow rate by means of velocity vector diagrams (Fig. 4.3).

When the vehicle is about to pull away, the engine drives the impeller with the turbine held stationary. Because the stalled turbine has no motion, the relative forward (linear) velocity between the two members will be large and consequently so will the resultant entry velocity $_R$. The direction of fluid flow from the impeller exit to turbine entrance will make a small angle Θ_1, relative to the forward direction of motion, which therefore produces considerable drive thrust to the turbine vanes.

As the turbine begins to rotate and catch up to the impeller speed the relative linear speed is reduced. This changes the resultant fluid flow direction to Θ_2 and decreases its velocity. The net output thrust, and hence torque carrying capacity, will be less, but with the vehicle gaining speed there is a rapid decline in driving torque requirements.

At high turbine speeds, that is, when the output to input speed ratio is approaching unity, there will be only a small relative linear velocity and resultant entrance velocity, but the angle Θ_3 will be large. This implies that the magnitude of the fluid thrust will be very small and its direction ineffective in
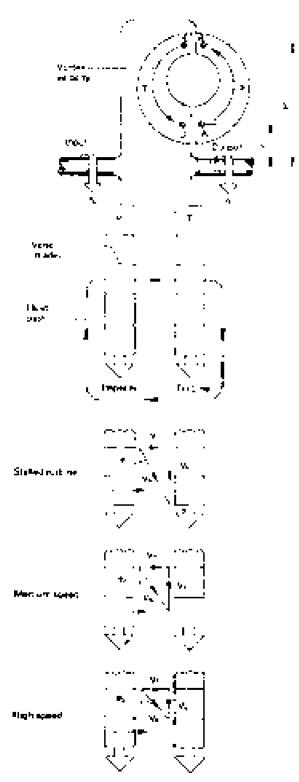
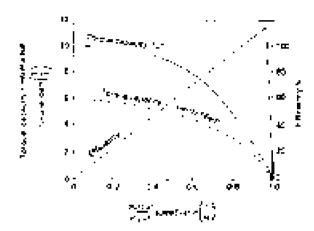**Fig. 4.3** Principle of the fluid coupling

Fig. 4.4 Relationship of torque capacity efficiency and speed ratio for fluid couplings
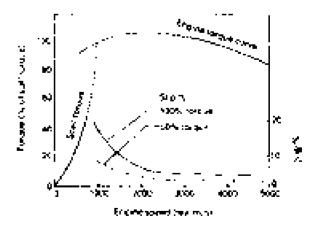

Fig. 4.5 Relationship of engine speed torque and slip for a fluid coupling

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rotating the turbine. Thus the output member will slip until sufficient circulating fluid flow imparts enough energy to the turbine again.

It can be seen that at high rotational speeds the cycle of events is a continuous process of output speed almost, but never quite, catching up to input speed, the exception being when the drive changes from engine driven to overrun transmission driven when the operating conditions will be reversed.

4.2 ydrokinetic fluid coupling efficiency and tor ue capacity (Figs 4.4 and 4.5)

Coupling efficiency is the ratio of the power available at the turbine to the amount of power supplied to the impeller. The difference between input and output power, besides the power lost by fluid shock, friction and heat, is due mainly to the relative slip between the two members (Fig. 4.4). A more useful term is the   , which is defined as the ratio of the difference in input and output speeds divided by the input speed and multiplied by 100.

i.e. $\% \text{ slip} = \left(\dfrac{\quad}{\quad} \right) \times 100$

The percentage slip will be greatly influenced by the engine speed and output turbine load conditions (Fig. 4.5). A percentage of slip must always exist to create a sufficient rate of vortex circulation which is essential to impart energy from the impeller to the turbine. The coupling efficiency is at best about 98% under light load high rotational speed conditions, but this will be considerably reduced as turbine output load is increased or impeller speed is lowered. If the output torque demand increases, more slip will occur and this will increase the vortex circulation velocity which will correspondingly impart more kinetic energy to the output turbine member, thus raising the torque capacity of the coupling. An additional feature of such couplings is that if the engine should tend to stall due to overloading when the vehicle is accelerated from rest, the vortex circulation will immediately slow down, preventing further torque transfer until the engine s speed has recovered.

Fluid coupling torque transmitting capacity for a given slip varies as the fifth power of the impeller internal diameter and as the square of its speed.

i.e. $\qquad ^5 \quad ^2$

where   = impeller diameter
          = impeller speed (rev/min)

Thus it can be seen that only a very small increase in impeller diameter, or a slight increase in impeller speed, considerably raises the coupling torque carrying capacity. A further controlling factor which affects the torque transmitted is the quantity of fluid circulating between the impeller and turbine. Raising or lowering the fluid level in the coupling increases or decreases the torque which can be transmitted to the turbine (Fig. 4.4).

4.3 luid friction coupling (Figs 4.6 and 4.7)

A fluid coupling has the take-up characteristics which particularly suit the motor vehicle but it suffers from two handicaps that are inherent in the system. Firstly, idling drag tends to make the vehicle creep forwards unless the parking brake is fully applied, and secondly there is always a small amount of slip which is only slight under part load (less than 2%) but becomes greater when transmitting anything near full torque.

These limitations have been overcome for large truck applications by combining a shoe and drum centrifugally operated clutch to provide a positive lock-up at higher output speeds with a smaller coreless fluid coupling than would be necessary if the drive was only to be through a fluid coupling. The reduced size and volume of fluid circulation in the coupling thereby eliminate residual idling drag (Fig. 4.6).

With this construction there is a shoe carrier between the impeller and flywheel attached to the output shaft. Mounted on this carrier are four brake shoes with friction material facings. They are each pivoted (hinged) to the carrier member at one end and a garter spring (coil springs shown on front view to illustrate action) holds the shoes in their retraction position when the output shaft is at rest.

When the engine is accelerated the fluid coupling automatically takes up the drive with maximum smoothness. Towards maximum engine torque speed the friction clutch shoes are thrown outwards by the centrifugal effect until they come into contact with the flywheel drum. The frictional grip will now lock the input and output drives together. Subsequently the fluid vortex circulation stops and the fluid coupling ceases to function (Fig. 4.7).

Relative slip between input and output member in low gear is considerably reduced, due to the automatic friction clutch engagement, and engine braking is effectively retained down to idling speeds.

4.4 ydrokinetic three element tor ue converter (Figs 4.8 and 4.9)

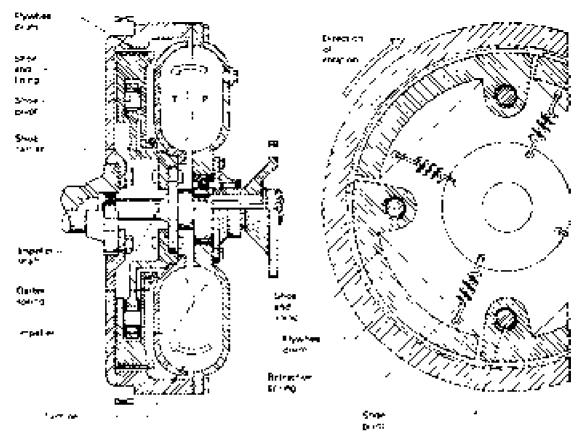A three element torque converter coupling is comprised of an input impeller casing enclosing the

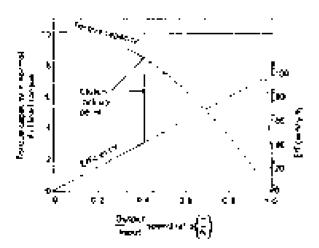**Fig. 4.6** Fluid friction coupling

Fig. 4.7 Relationship of torque carrying capacity efficiency and output speed for a fluid coupling

output turbine wheel. There are about 26 and 23 blades for the impeller and turbine elements respectively. Both of these elements and their blades are fabricated from low carbon steel pressings. The third element of the converter called the _____ is usually an aluminium alloy casting which may have something in the order of 15 blades (Figs 4.8 and 4.9).

The working fluid within a converter when the engine is operating has two motions:

1 Fluid trapped in the impeller and turbine vane cells revolves bodily with these members about their axis of rotation.
2 Fluid trapped between the impeller and turbine vane cells and their central torus core rotates in a circular path in the section plane, this being known as its _____.

When the impeller is rotated by the engine, it acts as a centrifugal pump drawing in fluid near the

centre of rotation, forcing it radially outwards through the cell passages formed by the vanes to the impeller peripheral exit. Here it is ejected due to its momentum towards the turbine cell passages and in the process acts at an angle against the vanes, thus imparting torque to the turbine member (Fig. 4.8).

The fluid in the turbine cell passages moves inwards to the turbine exit. It is then compelled to flow between the fixed stator blades (Fig. 4.9). The reaction of the fluids momentum as it glides over the curved surfaces of the blades is absorbed by the casing to which the stator is held and in the process it is redirected towards the impeller entrance. It enters the passages shaped by the impeller vanes. As it acts on the drive side of the vanes, it imparts a torque equal to the stator reaction in the direction of rotation (Fig. 4.8).

It therefore follows that the engine torque delivered to the impeller and the reaction torque transferred by the fluid to the impeller are both transmitted to the output turbine through the media of the fluid.

i.e. Engine torque + Reaction torque = Output turbine torque

4.8.1 Hydrokinetic three element torque converter principle of operation (Fig. 4.8)

When the engine is running, the impeller acts as a centrifugal pump and forces fluid to flow radially around the vortex passage made by the vanes and core of the three element converter. The rotation of the impeller by the engine converts the engine power into hydrokinetic energy which is utilized in
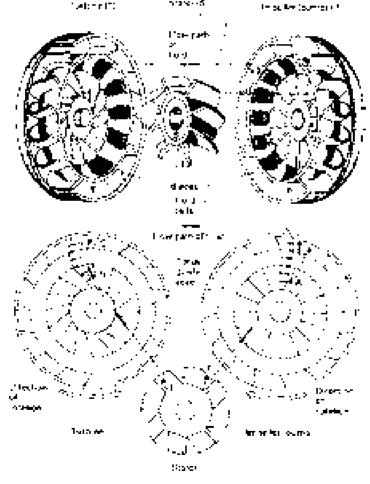
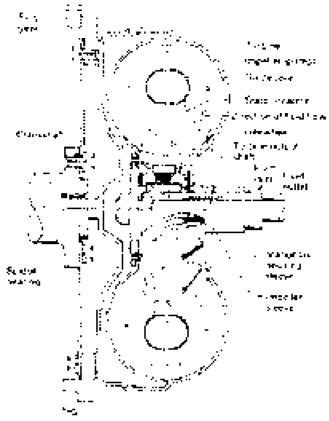

Fig. 4.8 Three element torque converter action

Fig. 4.9 Three element tor ue converter

providing a smooth engine to transmission take-up and in producing torque multiplication if a third fixed stator member is included.

An appreciation of the principle of the converter can be obtained by following the movement and events of a fluid particle as it circulates the vortex passage (Fig. 4.8).

Consider a fluid particle initially at the small diameter entrance point A in the impeller. As the impeller is rotated by the engine, centrifugal force will push the fluid particle outwards to the impeller s largest exit diameter, point B. Since the particle s circumferential distance moved every revolution will be increased, its linear velocity will be greater and hence it will have gained kinetic energy.

Pressure caused by successive particles arriving at the impeller outermost cell exit will compel the particle to be flung across the impeller–turbine unction where it acts against the side of cell vane it has entered at point C and thereby transfers some of its kinetic energy to the turbine wheel. Because the turbine wheel rotates at a lower speed relative to the impeller, the pressure generated in the impeller will be far greater than in the turbine. Subsequently the fluid particle in the turbine curved passage will be forced inwards to the exit point and in doing so will give up more of its kinetic energy to the turbine wheel.

The fluid particle, still possessing kinetic energy at the turbine exit, now moves to the stator blade s entrance side to point E. ere it is guided by the curvature of the blades to the exit point F.

From the fixed stator (reactor) blades the fluid path is again directed to the impeller entrance point A where it imparts its hydrokinetic energy to the impeller, this being quite separate to the kinetic energy produced by the engine rotating the impeller. Note that with the fluid coupling, the transfer of fluid from the turbine exit to the impeller entrance is direct. Thus the kinetic energy gained by the input impeller is that lost by the output turbine and there is no additional gain in output turning effort, as is the case when a fixed intermediate stator is incorporated.

4.8 Hydrokinetic three element torque converter velocity diagrams (Figs 4.9 and 4.10)
The direction of fluid leaving the turbine to enter the stator blades is influenced by the tangential exit velocity which is itself determined by the vortex circulating speed and the linear velocity due to the rotating turbine member (Fig. 4.10).

When the turbine is in the stalled condition and the impeller is being driven by the engine, the direction of the fluid leaving the impeller will be determined entirely by the curvature and shape of the turbine vanes. Under these conditions, the fluid's direction of motion, Θ_1, will make it move deep into the concave side of the stator blades where it reacts and is
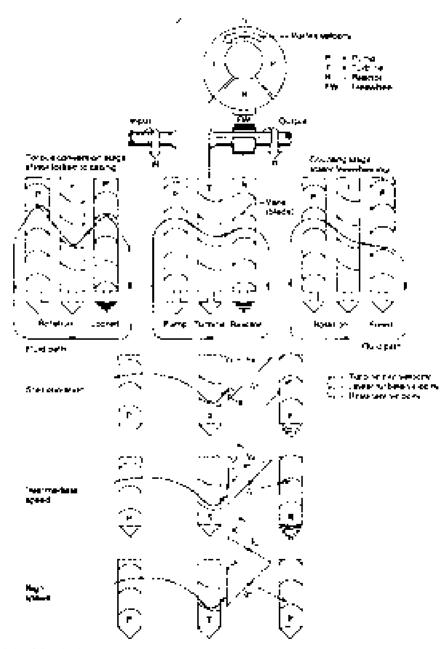
Fig. 4.10 Principle of the single stage torque converter

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made to flow towards the entrance of the impeller in a direction which provides the maximum thrust.

nce the turbine begins to rotate, the fluid will acquire a linear velocity so that the resultant effective fluid velocity direction will now be Θ_2. A reduced backward reaction to the stator will be produced so that the direction of the fluid s momentum will not be so effective.

As the turbine speed of rotation rises, the fluid s linear forward velocity will also increase and, assuming that the turbine s tangential exit velocity does not alter, the resultant direction of the fluid will have changed to Θ_3 where it now acts on the convex (back) side of the stator blades.

Above the critical speed, when the fluid s thrust changes from the concave to the convex side of the blades, the stator reaction torque will now act in the opposite sense and redirect the fluid. Thus its resultant direction towards the impeller entry passages will hinder instead of assist the impeller motion. The result of this would be in effect to cancel out some of the engine s input torque with further speed increases.

The inherent speed limitation of a hydrokinetic converter is overcome by building into the stator hub a one way clutch (freewheel) device (Fig. 4.9). Therefore, when the direction of fluid flow changes sufficiently to impinge onto the back of the blades, the stator hub is released, allowing it to spin freely between the input and output members. The freewheeling of the stator causes very little fluid interference, thus the three element converter now becomes a two element coupling. This condition prevents the decrease in torque for high output speeds and produces a sharp rise in efficiency at output speeds above the coupling point.

. .3 ydro inetic tor ue converter characteristics (Figs. 4.11 and 4.12)
Maximum torque multiplication occurs when there is the largest speed difference between the impeller and turbine. A torque output to input ratio of about 2:1 normally occurs with a three element converter when the turbine is stationary. Under such conditions, the vortex rate of fluid circulation will be at a peak. Subsequently the maximum hydrokinetic energy transfer from the impeller to turbine then stator to impeller again takes place (Figs 4.11 and 4.12). As the turbine output speed increases relative to the impeller speed, the efficiency rises and the vortex velocity decreases and so does the output to input torque ratio until eventually the circulation rate of fluid is so low that it can only support a 1:1 output to input torque ratio. At this point the reaction torque will be

zero. Above this speed the stator is freewheeled. This offers less resistance to the circulating fluid and therefore produces an improvement in coupling efficiency (Figs 4.11 and 4.12).

This description of the operating conditions of the converter coupling shows that if the transmission is suddenly loaded the output turbine speed will automatically drop, causing an increase in fluid circulation and correspondingly a rise in torque multiplication, but conversely a lowering of efficiency due to the increased slip between input and output members. When the output conditions have changed and a reduction in load or an increase in turbine speed follows the reverse happens the efficiency increases and the output to input torque ratio is reduced.

4. Tor ue converter performance terminology (Figs 4.11 and 4.12)
To understand the performance characteristics of a fluid drive (both coupling and converter), it is essential to identify and relate the following terms used in describing various relationships and conditions.

.5.1 luid drive efficiency (Figs 4.11 and 4.12)
A very convenient method of expressing the energy losses, due mainly to fluid circulation within a fluid drive at some given output speed or speed ratio, is
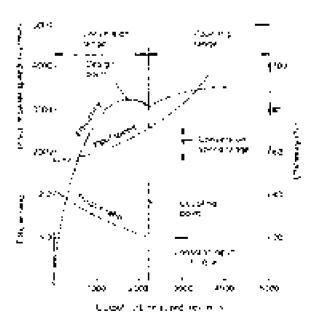

Fig. 4.11 Characteristic performance curves for a three element converted coupling
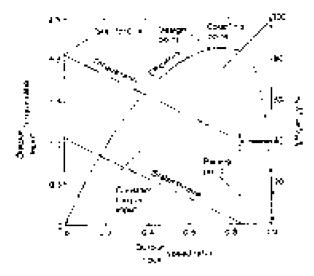
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Fig. 4.12 Characteristic performance curves for a converter coupling plotted to a base of output (turbine speed) to input (impeller speed)

to measure its efficiency, that is, the percentage ratio of output to input work done.

i.e. $\text{Efficiency} = \dfrac{\text{utput work done}}{\text{Input work done}} \times 100$

.5. *Speed ratio* (Fig. 4.12)

It is frequently necessary to compare the output and input speed differences at which certain events occur. This is normally defined in terms of a speed ratio of output (turbine) speed $_2$ to the input (impeller) speed $_1$.

i.e. $\text{Speed ratio} = \dfrac{_2}{_1}$

.5.3 *Tor ue ratio* (Fig. 4.12)

The torque multiplication within a fluid drive is more conveniently expressed in terms of a torque ratio of output (turbine) torque $_2$ to the input (impeller) torque $_1$.

i.e. $\text{Torque ratio} = \dfrac{_2}{_1}$

.5. *Stall speed* (Figs 4.11 and 4.12)

This is the maximum speed which the engine reaches when the accelerator pedal is fully down, the transmission in drive and the foot brake is fully applied. Under such conditions there is the greatest

impeller to turbine speed variation, with the result that the vortex fluid circulation and correspondingly torque conversion are at a maximum, conversely converter efficiency is zero. Whilst these stall conditions prevail, torque conversion loading drags the engine speed down to something like 60–70% of the engine s maximum torque speed, i.e. 1500–2500 rev/min. A converter should only be held in the stall condition for the minimum of time to prevent the fluid being overworked.

.5.5 *Design point* (Figs 4.11 and 4.12)

Torque converters are so designed that their internal passages formed by the vanes are shaped so as to make the fluid circulate with the minimum of resistance as it passes from one member to another member at definite impeller to turbine speed ratio, known as the   . A typical value might be 0.8:1.

Above or below this optimum speed ratio, the resultant angle and direction of fluid leaving one member to enter another will alter so that the flow from the exit of one member to the entry of another will no longer be parallel to the surfaces of the vanes, in fact it will strike the sides of the passage vanes entered. When the exit and entry angles of the vanes do not match the effective direction of fluid motion, some of its momentum will be used up in entrance losses and consequently the efficiency declines as the speed ratio moves further away on either side of the design point.   ther causes of momentum losses are internal fabrication finish, surface roughness and inter-vane or blade thickness interference. If the design point is shifted to a lower speed ratio, say 0.6, the torque multiplication will be improved at stall and lower speed conditions at the expense of an earlier fall-off in efficiency at the high speed ratio such as 0.8. There will be a reduction in the torque ratio but high efficiency will be maintained in the upper speed ratio region.

.5.6 *Coupling point* (Figs 4.11 and 4.12)

As the turbine speed approaches or exceeds that of the impeller, the effective direction of fluid entering the passages between the stator blades changes from pushing against the concave face to being redirected towards the convex (back) side of the blades. At this point, torque conversion due to fluid transfer from the fixed stator to the rotating impeller, ceases. The turbine speed when the direction of the stator reaction is reversed is known as the   and is normally between 80 and 90% of the impeller speed. At this point the stator is released by the freewheel device and is then driven

in the same direction as the impeller and turbine. At and above this speed the stator blades will spin with the impeller and turbine which then simply act as a fluid coupling, with the benefit of increasing efficiency as the turbine output speed approaches but never reaches the input impeller speed.

.5.7 acing or run a ay point (Fig. 4.12)

If the converter does not include a stator freewheel device or if the mechanism is ammed, then the direction of fluid leaving the stator would progressively change from transferring fluid energy to assist the impeller rotation to one of opposition as the turbine speed catches up with that of the impeller. Simultaneously, the vortex fluid circulation will be declining so that the resultant torque capacity of the converter rapidly approaches zero. Under these conditions, with the accelerator pedal fully down there is very little load to hold back the engine s speed so that it will tend to race or   . Theoretically racing or run-away should occur when both the impeller and turbine rotate at the same speed and the vortex circulation ceases, but due to the momentum losses caused by internal fluid resistance, racing will tend to begin slightly before a 1:1 speed ratio (a typical value might be 0.95:1).

.5. ngine bra ing transmitted through converter or coupling on overrun

Torque converters are designed to maximize their torque multiplication from the impeller to the turbine in the forward direction by adopting backward swept rotating member circulating passage vanes. Unfortunately, in the reverse direction when the turbine is made to drive the impeller on transmission overrun, the exit and entry vane guide angles of the members are unsuitable for hydrokinetic energy transference, so that only a limited amount of engine braking torque can be absorbed by the converter except at high output overrun vehicle speeds. Conversely, a fluid coupling with its flat radial vanes is able to transmit torque in either drive or overrun direction with equal effect.

4.   verrun clutches

Various names have been used for these mechanisms such as   and   , each one signifying the nature of the device and is therefore equally appropriate.

A freewheel device is a means whereby torque is transmitted from one stationary or rotating member to another member, provided that input torque (drive speed) is greater than that of the output member. If the conditions are reversed and the output member s applied torque (or speed) becomes greater than that of the input, the output member will overrun the input member (rotate faster). Thus the lock between the two members will be automatically released. Immediately the drive will be discontinued which permits the input and output members to revolve independently to one another.

 verrun clutches can be used for a number of applications, such as starter motor pre-engagement drives, overdrives, torque converter stator release, automatic transmission drives and final differential drives.

Most overrun clutch devices take the form of either the   to engage and disengage drive.

.6.1   verrun clutch ith single diameter rollers (Fig. 4.13)

A roller clutch is comprised of an inner and outer ring member and a series of cylindrical rollers spaced between them (see Fig. 4.13). Incorporated between the inner and outer members is a cage which positions the rollers and guides so that they roll up and down their ramps simultaneously. ne of the members has a cylindrical surface concentric with its axis, this is usually made the outer member. The other member (inner one) has a separate wedge ramp formed for each roller to react against. The shape of these wedge ramps may be flat or curved depending upon design. In operation each roller provides a line contact with both the outer internal cylindrical track and the external wedge ramp track of the inner member.

When the input wedge member is rotated clockwise and the output cylindrical member is prevented from rotating or rotates anticlockwise in the opposite direction, the rollers revolve and climb up the wedge ramps, and thereby squeeze themselves between the inner and outer member tracks. Eventually the elastic compressive and frictional forces created by the rollers against these tracks prevents further roller rotation. Torque can now transfer from the input inner member to the outer ring member by way of these ammed (locked) rollers.

If the output outer member tries to rotate in the same direction but faster than the inner member, the rollers will tend to rotate and roll down their ramps, thereby releasing (unlocking) the outer member from that of the input drive.
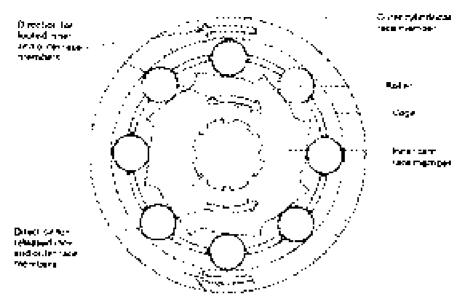
Fig. 4.13 verrun freewheel single diameter roller type clutch

.6. verrun clutch ith triple diameter rollers
(Fig. 4.14)

This is a modification of the single roller clutch in which the output outer member forms an internal cylindrical ring, whereas the input inner member has three identical external inclined plane profiles (see Fig. 4.14). Situated between the inner and outer tracks are groups of three different sized rollers. An anchor block and energizing shoe is arranged, between each group of rollers the blocks are screwed to the inner member while the shoes (with the assistance of the springs) push the rollers together and against their converging contact tracks. The inclined plane profile required to match the different diameter rollers provides a variable wedge angle for each size of roller. It is claimed that the take-up load of each roller will be progressive and spread more evenly than would be the case if all the rollers were of the same diameter.

When the input inner ring takes up the drive, the rollers revolve until they are wedged between the inclined plane on the inner ring and the cylindrical internal track of the outer member. Consequently the compressive load and the frictional force thus created between the rollers and tracks locks solid the inner and outer members enabling them to transmit torque.

If conditions change and the outer member overruns the inner member, the rollers will be compelled to revolve in the opposite direction to when the drive was established towards the diverging end of the tracks. It thus releases the outer member and creates the freewheel phase.

.6.3 Sprag overrun clutch (Fig. 4.15)

A very reliable, compact and large torque-carrying capacity overrun clutch is the sprag type clutch. This dispenses with the wedge ramps or inclined plane formed on the inner member which is essential with roller type clutches (see Fig. 4.15). The sprag clutch consists of a pair of inner and outer ring members which have cylindrical external and internal track surfaces respectively. Interlinking the input and output members are circular rows of short struts known as   . Both ends of the sprags are semicircular with their radius of curvature being offset to each other so that the sprags appear lopsided. In addition a tapered waste is formed in their mid-region. ouble cages are incorporated between inner and outer members. These cages have rectangular slots formed to equally space and locate the sprags around the inner and outer tracks. uring clutch engagement there will be a slight shift between relative positions of the two cages as the springs tilt, but the spacing will be
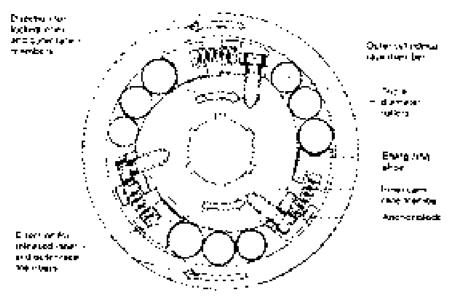
Fig. 4.14 verrun freewheel triple diameter roller type clutch

accurately kept. This ensures that each sprag equally contributes its share of wedge action under all operating conditions. In between the cages is a ribbon type spring which twists the sprags into light contact with their respective track when the clutch is in the overrun position.

When the inner ring member is rotated clockwise and the outer ring member is held stationary or is rotated anticlockwise, the spring tension lightly presses the sprags against their track. This makes the inner and outer members move in opposite directions. The sprags are thus forced to tilt anticlockwise, consequently wedging their inclined planes hard against the tracks and thereby locking the two drive and driven members together.

As conditions change from drive to overrun and the outer member rotates faster than the inner one, the sprags will rotate clockwise and so release the outer member: a freewheel condition is therefore established.

4.7 Three stage hydrokinetic tor ue converter
(Figs 4.16, 4.17 and 4.18)
A disadvantage with the popular three element torque converter is that its stall torque ratio is only in the region of 2:1, which is insufficient for some applications, but this torque multiplication can be doubled by increasing the number of

turbine and stator members within the converter, so that there are more stages of conversion (Fig. 4.16).

Consider the three stage torque converter. As shown in Fig. 4.17, it is comprised of one impeller, three interlinked output turbines and two fixed stator members.

Tracing the conversion vortex circuit starting from the input rotating member (Fig. 4.18), fluid is pumped from the impeller P by centrifugal force to the two velocity components $_t$ and $_r$, making up the resultant velocity $_p$ which enters between the first turbine blades $_1$ and so imparts some of its hydrokinetic energy to the output. Fluid then passes with a velocity $_1$ to the first fixed stator, S_1, where it is guided and redirected with a resultant velocity $_1$, made up from the radial and tangential velocities $_r$ and $_t$ to the second set of turbine blades T_2, so that momentum is given to this member. Fluid is now transferred from the exit of the second turbine T_2 to the entrance of the second stator S_2. ere the reaction of the curved blades deflects the fluid towards the third turbine blades T_3 which also absorb the fluid s thrust. Finally the fluid completes its circulation cycle by again entering the impeller passages.

The limitation of a multistage converter is that there are an increased number of entry and exit unctions between various members which raise
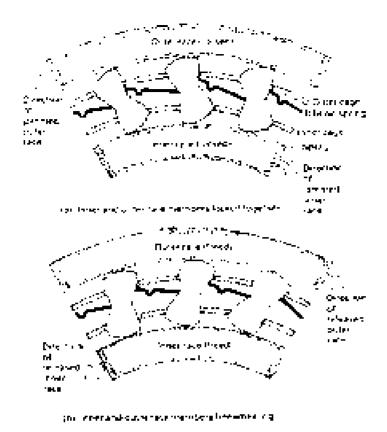
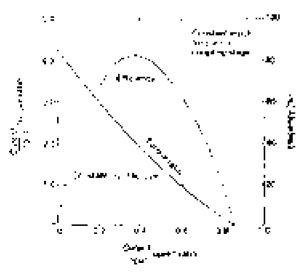**Fig. 4.15 (a and b)** verrun freewheel sprag type clutch

Fig. 4.16 Characteristic performance curves of a three stage converter

the fluid flow resistance around the torus passages. Subsequently, efficiency drops off fairly rapidly with higher speed ratios compared to the three element converter (Fig. 4.16).

4. Polyphase hydrokinetic tor ue converter
(Figs 4.19 and 4.20)

The ob ect of the polyphase converter is to extend the high efficiency speed range (Fig. 4.20) of the simple three element converter by altering the vane or blade shapes of one element. Normally the stator is chosen as the fluid entrance direction changes with increased turbine speed. To achieve this, the stator is divided into a number of separate parts, in this case three, each one being mounted on its own freewheel device built into its hub (Fig. 4.19). The turbine exit and linear velocities V_E and V produce an effective resultant velocity V_R which changes its direction of entry between stator blades as the impeller and turbine relative speeds
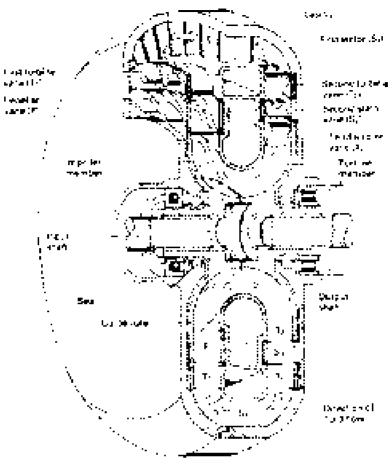
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Fig. 4.17   ultistage (six element) tor ue converter

approach unity. It is this direction of fluid entering between the stator blades which in phases releases the various stator members.

nitial phase
Under stall speed conditions, the fluid flow from the turbine to the stator is such as to be directed onto the concave (rear) side of all three sections of the divided stator blades, thus producing optimum stator reaction for maximum torque multiplication conditions.

Second phase
As the turbine begins to rotate and the vehicle is propelled forwards, the fluid changes its resultant direction of entry to the stator blades so that it impinges against the rear convex side of the first stator blades S_1. The reaction on this member is now reversed so that it is released and is able to

spin in the same direction as the input and output elements. The two remaining fixed stators now form the optimum blade curvatures for high efficiency.

Third phase
With higher vehicle and turbine speeds, the fluid s resultant direction of entry to the two remaining held stators changes sufficiently to push from the rear of the second set of stator blades S_2. This section will now be released automatically to enable the third set of stator blades to operate with optimum efficiency.

Coupling phase
Towards unity speed ratio when the turbine speed has almost caught up with the impeller, the fluid entering the third stator blades S_3 will have altered its direction to such an extent that it releases this
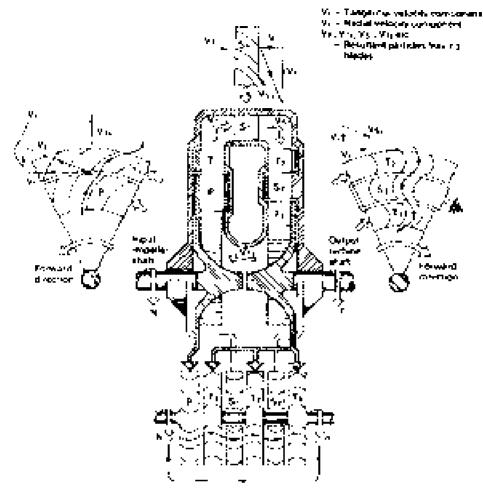
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Fig. 4.18 rinciple of the three stage tor ue converter

last fixed set of blades. Since there is no more reaction torque, conversion ceases and the input and output elements act solely as a fluid coupling.

4.9 Tor ue converter with lock-up and gear change friction clutches (Figs 4.21 and 4.22)

The two ma or inherent limitations with the torque converter drive are as follows:

Firstly, the rapid efficiency decline once the relative impeller to turbine speed goes beyond the design point, which implies higher input speeds for a given output speed and increased fuel consumption. Secondly, the degree of fluid drag at idle speed which would prevent gear changing with constant mesh and synchromesh gearboxes.

The disadvantage of the early fall in efficiency with rising speed may be overcome by incorporating a friction disc type clutch between the flywheel and converter which is hydraulically actuated by means of a servo piston (Fig. 4.21). This lock-up clutch is designed to couple the flywheel and impeller assembly directly to the output turbine shaft either manually, at some output speed decided by the driver which would depend upon the vehicle load and the road conditions or automatically, at a definite input to output speed ratio normally in the region of the design point here where efficiency is highest (Fig. 4.22).

To overcome the problem of fluid drag between the input and output members of the torque converter when working in con unction with either
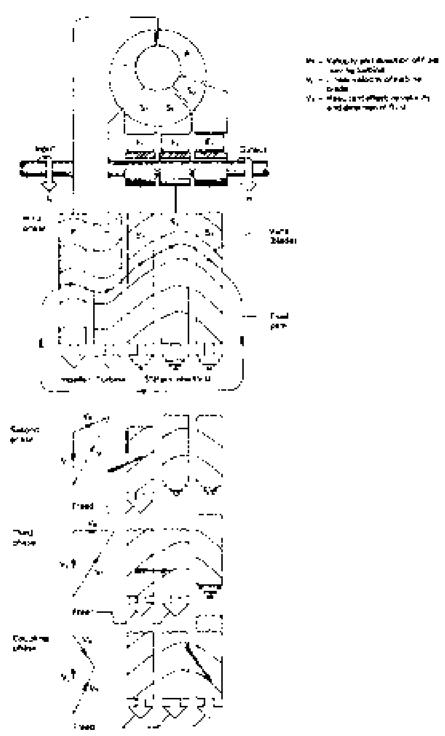
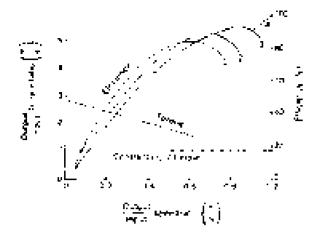**Fig. 4.19**   rinciple of a polystage tor ue converter

Fig. 4.20 Relationship of speed ratio, torque ratio and efficiency for a polyphase stator torque converter

Fig. 4.22 Characteristic performance curves of a three element converter with lock-up clutch
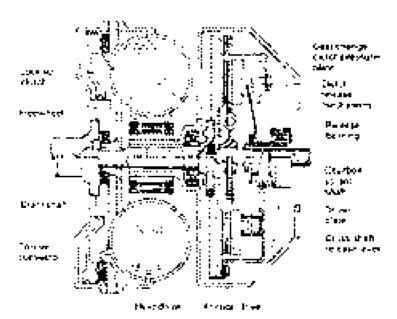

Fig. 4.21 Torque converter with lock-up and gear change function clutches

constant mesh or synchromesh gearboxes, a conventional foot operated friction clutch can be utilized between the converter and the gearbox. When the pedal is depressed and the clutch is in its disengaged position, the gearbox input primary shaft and the output main shaft may be unified, thereby enabling the gear ratio selected to be engaged both smoothly and silently.

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Semi- and fully automatic transmission

.1 utomatic transmission considerations

Because it is difficult to achieve silent and smooth gear ratio changes with a conventional constant mesh gear train, automatic transmissions commonly adopt some sort of epicyclic gear arrangement, in which different gear ratios are selected by the application of multiplate clutches and band brakes which either hold or couple various members of the gear train to produce the necessary speed variations. The problem of a gradual torque take-up when moving away from a standstill has also been overcome with the introduction of a torque converter between the engine and transmission gearing so that engine to transmission slip is automatically reduced or increased according to changes in engine speed and road conditions. Torque converter performance characteristics have been discussed in Chapter 3.

The actual speed at which gear ratio changes occur is provided by hydraulic pressure signals supplied by the governor valve and a throttle valve. The former senses vehicle speed whereas the latter senses engine load.

These pressure signals are directed to a hydraulic control block consisting of valves and pistons which compute this information in terms of pressure variations. The fluid pressure supplied by a pressure pump then automatically directs fluid to the various operating pistons causing their respective clutch, clutches or band brakes to be applied. Consequently, gear upshifts and downshifts are performed independently of the driver and are so made that they take into account the condition of the road, the available output of the engine and the requirements of the driver.

5.1.1 *The tor ue converter* (Fig. 5.1)

The torque converter provides a smooth automatic drive take-up from a standstill and a torque multiplication in addition to that provided by the normal mechanical gear transmission. The performance characteristics of a hydrokinetic torque converter incorporated between the engine and the gear train is shown in Fig. 5.1 for light throttle and full throttle maximum output conditions over a vehicle speed range. As can be seen, the initial torque multiplication when driving away from rest is con-

siderable and the large gear ratio steps of the conventional transmission are reduced and smoothed out by the converter s response between automatic gear shifts. Studying Fig. 5.1, whilst in first gear, the torque converter provides a maximum torque multiplication at stall pull away conditions which progressively reduces with vehicle speed until the converter coupling point is reached. At this point, the reaction member freewheels. With further speed increase, the converter changes to a simple fluid coupling so that torque multiplication ceases. In second gear the converter starts to operate nearer the coupling point causing it to contribute far less torque multiplication and in third and fourth gear the converter functions entirely beyond the coupling point as a fluid coupling. Consequently, there is no further torque multiplication.

.2 our speed and reverse longitudinally mounted automatic transmission mechanical power flow (Fig. 5.2)

(Similar gear trains are adopted by some F, Mercedes-Benz and Nissan transmissions)

The epicyclic gear train is comprised of three planetary gear sets, an overdrive gear set, a forward gear set and a reverse gear set. Each gear set consists of an internally toothed outer annular ring gear, a central externally toothed sun gear and a planet carrier which supports three intermediate planet gears. The planet gears are spaced evenly between and around the outer annular gear and the central sun gear.

The input to the planetary gear train is through a torque converter which has a lock-up clutch.

 ifferent parts of the gear train can be engaged or released by the application of three multiplate clutches, two band brakes and one first gear one way roller clutch.

Table 5.1 simplifies the clutch and brake engagement sequence for each gear ratio.

A list of key components and abbreviations used are as follows:

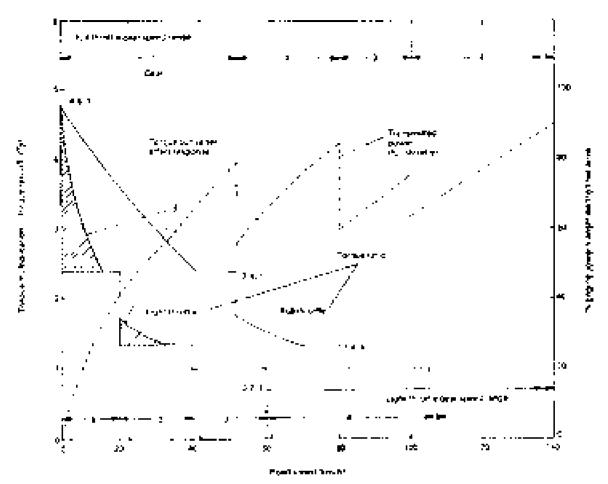|		
1	Manual valve	MV
2	Vacuum throttle valve	VTV
3	overnor valve	V
4	Pressure regulating valve	PRV
5	Torque converter	TC

Fig. 5.1 Tor ue multiplication and transmitted power performance relative to vehicle speed for a typical four speed automatic transmission

6	1–2 shift valve	(1–2)SV
7	2–3 shift valve	(2–3)SV
8	3–4 shift valve	(3–4)SV
9	Converter check valve	CCV
10	rive clutch	C
11	igh and reverse multiplate clutch	(R)C
12	Forward clutch	FC
13	verdrive band brake	B
14	Second gear band brake	2 B
15	ow and reverse multiplate brake	(R)B
16	First gear one way roller clutch	WC
17	Torque converter one way clutch	WC$_R$
18	Parking lock	P

5. .1 D drive range   first gear
(Figs 5.3(a) and 5.4(a))
With the selector lever in   range, engine torque is transmitted to the overdrive pinion gears via the out-put shaft and pinion carrier. Torque is then split between the overdrive annular gear and the sun gear, both paths merging due to the engaged direct clutch. Therefore the overdrive pinion gears are prevented from rotating on their axes, causing the overdrive gear set to revolve as a whole without any gear ratio reduction at this stage. Torque is then conveyed from the overdrive annular gear to the intermediate shaft where it passes through the applied forward clutch plates to the annular gear of the forward gear set. The clockwise rotation of the forward annular gear causes the forward planet gears to rotate clockwise, driving the double sun gear counter clockwise. The forward planetary car-rier is attached to the output shaft so that the planet gears drive the sun gear instead of walking around the sun gear. This anticlockwise rotation of the sun gear causes the reverse planet gears to rotate
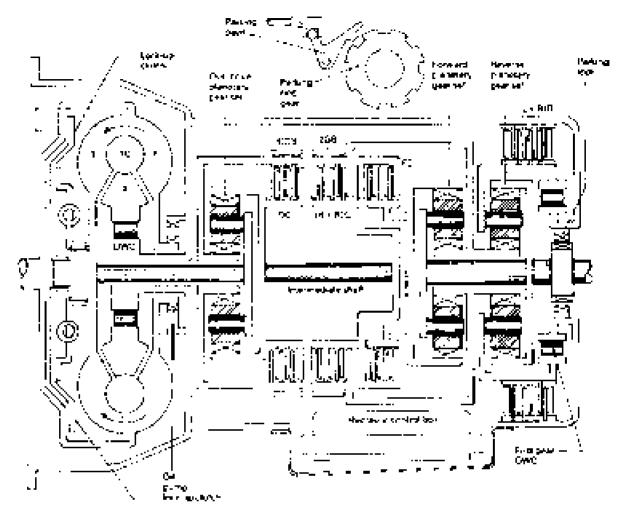
Fig. 5.2 ongitudinally mounted four speed automatic transmission layout

ab 5.1 Clutch and brake engagement se uence

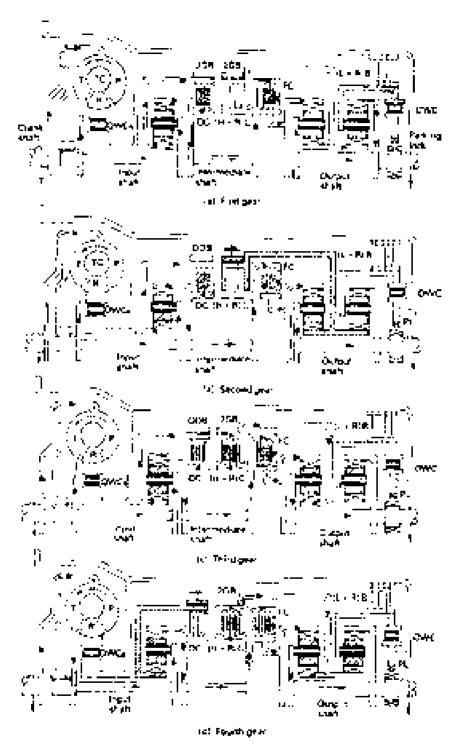| Range	rive clutch C	High and reverse clutch (H R)C	Second gear band brake 2 B	Forward clutch FC	verdrive brake B	ow and reverse brake (R)B	ne way clutch C	Ratio
and N	–	–	–	–	–	–	–	–
First	Applied	–	–	Applied	–	–	Applied	2 1
Second	Applied	–	Applied	Applied	–	Applied	–	1 1
Third	Applied	Applied	–	Applied	–	–	–	1 1
Fourth	–	Applied	–	Applied	Applied	–	–	1
Reverse R	Applied	Applied	–	–	–	Applied	–	2 1

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Fig. 5.3 (a–) Four speed and reverse automatic transmission for longitudinally mounted units
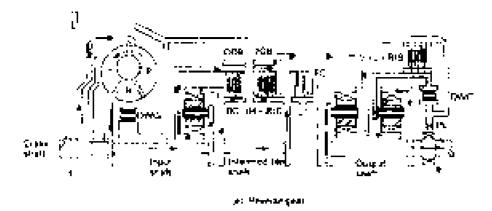
Fig. 5.3 *contd*

clockwise. With the one way roller clutch holding the reverse planet carrier, the reverse planetary gears turn the reverse annular gear and output shaft clockwise in a low speed ratio of approximately 2.46:1.

5. . D drive range   second gear
(Figs 5.3(b) and 5.4(b))
In   range in second gear, both direct and forward clutches are engaged. At the same time the second gear band brake holds the double sun gear and reverse pinion carrier stationary.

Engine torque is transmitted through the locked overdrive gear set similarly to first gear. It is then conveyed through the applied forward clutch via intermediate shaft to the forward annular gear. With the double sun gear held by the applied second gear band brake, the clockwise rotation of the forward annular gear compels the pinion gears to rotate on their own axes and roll walk around the stationary sun gear in a clockwise direction. Because the forward pinion gear pins are mounted on the pinion carrier, which is itself attached to the output shaft, the output shaft will be driven clockwise at a reduced speed ratio of approximately 1.46.

5. .3 D drive range   third or top gear
(Figs 5.3(c) and 5.4(c))
With the selector lever in   range, hydraulic line pressure will apply the direct clutch, high and reverse clutch and forward clutch.

As for first and second gear operating conditions, the engine torque is transmitted through the locked overdrive gear set to the high and reverse

multiplate clutch and the forward multiplate clutch, both of which are applied. Subsequently, the high and reverse clutch will rotate the double sun gear clockwise and similarly the forward clutch will rotate the forward annular gear clockwise. This causes both external and internal gears on the forward gear set to revolve in the same direction at similar speeds so that the bridging planet gears become locked and the whole gear set therefore revolves together as one. The output shaft drive via the reverse carrier therefore turns clockwise with no relative speed reduction to the input shaft, that is as a direct drive ratio 1:1.

5. . D drive range   fourth or overdrive gear
(Figs 5.3(d) and 5.4(d))
In   range in fourth gear, the overdrive band brake, the high and reverse clutch and the forward clutch are engaged. Under these conditions, torque is conveyed from the input shaft to the overdrive carrier, causing the planet gears to rotate clockwise around the held overdrive sun gear. As a result, the overdrive annular gear will be forced to rotate clockwise but at a higher speed than the input overdrive carrier. Torque is then transmitted via the intermediate shaft to the forward planetary gear set which are then locked together by the engagement of the high and reverse clutch and the forward clutch. Subsequently, the gear set is compelled to rotate bodily as a rigid straight through drive. The torque then passes from the forward planet carrier to the output shaft.   ence there is a gear ratio step up by the overdrive planetary gear set of roughly 30%, that is, the output to input shaft gear ratio is about 0.7:1.
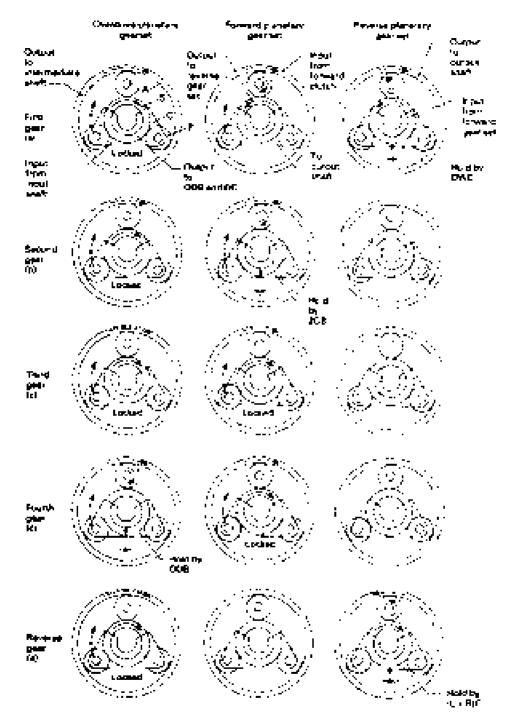
Fig. 5.4 (a–) Four speed and reverse epicycle gear set directional motion

5. .5   range   reverse gear
(Figs 5.3(e) and 5.4(e))
With the selector lever in reverse position all three clutches and the low and reverse multiplate brake are engaged. Subsequently, engine torque will be transmitted from the input shaft through the locked overdrive gear set through the locked forward gear set via the intermediate shaft to the reverse sun gear in a clockwise direction.

Because the reverse planet carrier is held by the low and reverse multiplate brake, the planet gears are forced to rotate counterclockwise on their axes, and in doing so compel the reverse annular gear to also rotate counterclockwise. As a result, the output shaft, which is attached to the reverse annular gear, rotates counterclockwise, that is, in the reverse direction, to the input shaft at a reduction ratio of approximately 2.18:1.

.3 The fundamentals of a hydraulic control system
The effective operation of an automatic transmission relies upon a hydraulic control circuit to actuate the gear changes relative to the vehicle s road speed and acceleration pedal demands with the engine delivering power.   nly a very small proportion of a transmission s operating time is spent in performing gear changes. In fact, the hydraulic system is operational for less than 1% of the driving time. The transition time from one gear ratio to the next takes roughly one second or less and therefore the hydraulic control valves must be designed to direct fluid pressure to the appropriate operating pistons which convert the fluid pressure into mechanical force and movement to energize the respective clutches and band brakes instantly and precisely.

An understanding of a basic hydraulic control system can best be considered under the four headings:

1 Pressure supply and regulating valves
2 Speed and load sensing valves
3   ear shift valves
4 Clutch and brake coupling and hold devices

5.3.1   ressure supply and regulating valve
(Fig. 5.5)
The essential input to the hydraulic control system is fluid pressure generated by a pump and driven by the engine. The pump s output pressure will increase roughly in proportion to the engine s speed.   owever, the pressure necessary to actuate

the various valves and to energize the clutch and band servo pistons will vary under different working conditions. Therefore the fluid pressure generated by the pump is unlikely to suit the many operating requirements. To overcome these difficulties, a pressure regulating valve is used which automatically ad usts the pump s output pressure to match the working requirements at any one time.   ne of the functions of the pressure regulating valve is to raise the line pressure reaching the clutch and brake when the vehicle is driven hard with large throttle opening to prevent the friction surfaces slipping. Conversely under light loads and with a small throttle opening, a much lower line pressure is adequate to clamp the friction plates or bands. By reducing the line pressure, fierce clutch and brake engagements are eliminated which promotes smooth and gentle gear changes. Power consumption, which is needed to drive the hydraulic pump, is also reduced as actuating pressures are lowered. The pressure regulating valve is normally a   , that is, a plunger with one or more reduced diameter sections along its length, positioned in a cylinder which has a number of passages intersecting the cylinder walls.

When the engine speed, and correspondingly pump pressure, is low, fluid flows via the inlet port around the wasted section of the plunger and out unrestricted along a passage leading to the manual valve where it is distributed to the various control valves and operating pistons. As the pump pressure builds up with rising engine speed, line pressure is conveyed to the rear face of the plunger and will progressively move the plunger forward against the control spring, causing the middle land to uncover an exhaust port which feeds back to the pump s intake.   ence as the pump output pressure tends to rise, more fluid is passed back to the suction intake of the pump. It therefore regulates the output fluid pressure, known as line pressure, according to the control spring stiffness. To enable the line pressure to be varied to suit the operating conditions, a throttle pressure is introduced to the spring end of the plunger which opposes the line pressure. Increasing the throttle pressure raises line pressure and vice versa.

In addition to the main pressure regulating valve there is a secondary regulating valve which limits the fluid flowing through to the torque converter. Raising the torque converter s fluid pressure increases its torque transmitting capacity which is desirable when driving in low gear or when the engine is delivering its maximum torque.

5.3. Speed and load sensing valves
(Figs 5.5 and 5.6)

For gear changes to take place effectively at the optimum engine and road speed, taking into account the driver s demands expressed in throttle opening, some means of sensing engine load and vehicle road speed must be provided. Engine output torque is simply monitored by a throttle valve which is linked to the accelerator pedal, either directly or indirectly, via a vacuum diaphragm operated linkage which senses the change in induction depression, which is a measure of the engine load. The amount the accelerator pedal or manifold vacuum alters is relayed to the throttle valve which accordingly raises or lowers the output pressure. This is then referred to as   .

Road speed changes are measured by a centrifugal force-sensitive regulating valve which senses transmission output shaft speed and transmits this information in the form of a fluid pressure, referred to as   , which increases or decreases according to a corresponding variation in road speed. Both throttle pressure and governor pressure are signalled to each gear shift valve so that these may respond to the external operating conditions (i.e. engine torque developed and vehicle speed) by permitting fluid pressure to be either applied or released from the various clutch and brake actuating piston chambers.

5.3.3   ear shift valves (Fig. 5.5)

Shift valves are of the spool plunger type, taking the form of a cylindrical plunger reduced in diameter in one or more sections so as to divide its length into a number of lands. When operating, these valves shift from side to side and cover or uncover passages leading into the valve body so that different hydraulic circuits are switched on and off under various operating conditions.

The function of a shift valve is to direct the fluid pressure to the various clutch and brake servo pistons to effect gear changes when the appropriate load and speed conditions prevail. Shift valves are controlled by line or throttle pressure, which is introduced into the valve at the spring end, and governor pressure, which is introduced directly to the valve at the opposite end.   enerally, the governor valve end is of a larger diameter than the spring end so that there will be a proportionally greater movement response due to governor pressure variation. Sometimes the shift valve plunger at the governor pressure end is referred to as the   .

The position of the shift valve at any instant depends upon the state of balance between the opposing end forces acting on the spool valve end faces.

$$\text{Spring load} + \text{Throttle pressure load} = \text{ overnor pressure load}$$

$$F_S + {}_T \cdot {}_T =$$

But $= F$

hence $F_S + F_T = F$

where $F_S = $ Spring load
$F_T = $ Throttle pressure load
${}_T = $ Throttle pressure
$F = $ overnor pressure load
${}_T = $ CSA of plunger at throttle pressure end
$= $ overnor pressure
$= $ CSA of plunger at governor pressure end

Thus increasing or decreasing the spring stiffness or enlarging or reducing the diameter of the spool valve at one end considerably alters the condition when the shift valve moves from one end to the other to redirect line pressure to and from the various clutch and brakes and so produce the necessary gear change.

Each shift valve control spring will have a particular stiffness so that different governor pressures, that is, road speeds, are required to cause either a gear upshift or downshift for a given opposing throttle pressure. Conversely, different engine power outputs will produce different throttle pressures and will alter the governor pressure accordingly when a particular gear shift occurs.   arge engine loads (high throttle pressure) will delay gear upshifts whereas light engine load demands (low throttle pressure) and high vehicle speeds (high governor pressure) will produce early upshifts and prevent early downshift.

To improve the quality of the time sequence of up or down gear shift, additional valves and components are included to produce a smooth transition from one gear to the next. Some of these extra devices are described in Section 5.6.

5.3.   Clutch and bra e coupling and hold devices
(Figs 5.5 and 2.16)

Silent gear change synchronization is made possible by engaging or locking out various members of the epicyclic gear train gear sets with the engine s power being transmitted continuously. It therefore requires a rapid and accurate gear change which is achieved by utilizing multiplate clutches and band brakes. A gear up- or downshift therefore occurs with the almost simultaneous energizing of one
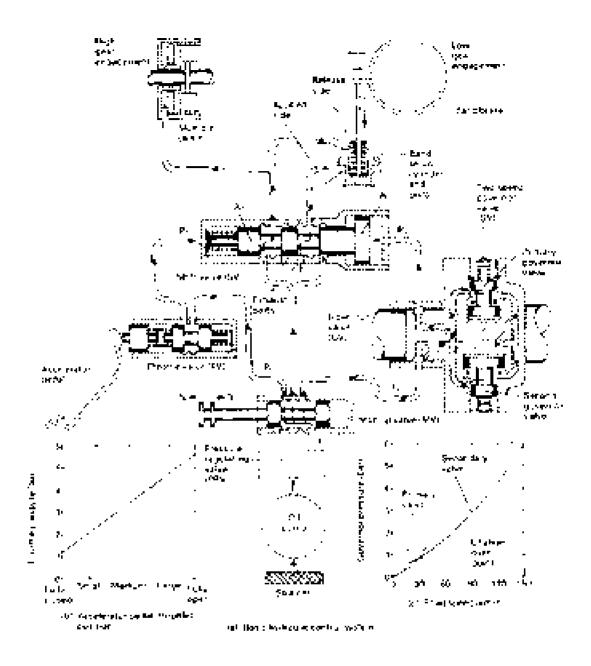
Fig. 5.5 (a and b)   Basic multiplate clutch and band brake transmission hydraulic control system

clutch or brake and a corresponding de-energizing of another clutch or brake.

Multiplate clutch (Figs 2.16 and 5.5)   Wet multi-plate type clutches are very compact for their torque transmitting and heat dissipating capacity. They are used to lock any two members of a planetary gear set together or to transfer drive from one shaft or member to another quickly and smoothly. The rotating and fixed friction plates can be energized by an annular shaped, hydraulically operated piston either directly or indirectly by a dished washer which acts also as a lever to multiply the operating clamping load. Return springs are used to separate the pairs of rubbing faces when the fluid pressure is released. Wear and

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ad ustment of the friction plate pack is automatically compensated by the piston being free to move further forward (see Chapter 2, Fig. 2.16).

and brake (Fig. 5.5) This form of brake consists of a friction band encompassing an external drum so that when the brake is applied the band contracts, thereby wrapping itself tightly around the drum until the drum holds. The application of the band is achieved through a double acting stepped servo cylinder and piston. Fluid line pressure is introduced to the small diameter end of the piston to energize the band brake. To release the band, similar line pressure is directed to the spring chamber side of the cylinder. Band release is obtained due to the larger piston area side producing a greater force to free the band. This method of applying and releasing the band enables a more prolonged and controllable energizing and de-energizing action to be achieved. This class of brake is capable of absorbing large torque reactions without occupying very much space, which makes the band brake particularly suitable for low gear high torque output gear sets. Band wear slackness can be taken up by externally ad usting the anchor screw.

.4 asic principle of a hydraulically controlled gearshift (Fig. 5.5)

Selecting the drive range positions the manual valve spool so that line pressure from the pressure regulator valve passes through to the shift valve, throttle valve and governor valve (Fig. 5.5(a)).

Throttle pressure will be introduced to the spring end of the shift valve via the throttle valve. epressing the accelerator pedal allows the spool valve to move outwards. This increases the valve opening so that a high throttle pressure will be delivered to the shift valve. Conversely, depressing the accelerator pedal partially restricts the flow of fluid and therefore reduces the throttle pressure reaching the shift valve (Fig. 5.5(b)).

At the same time, line pressure enters the governor valve, flows between the wasted region of both primary and secondary spool valves and reacts against the difference in the annular ad acent face areas of each spool valve. Both valves are forced inwards, covering up the two exits from the governor valve housing. As the vehicle moves forwards, the rotation of the governor causes a centrifugal force to act through the mass of each governor valve so that it tends to draw the valve spools outwards in opposition to the hydraulic pressure which is pushing each valve inwards (Fig. 5.5(a)).

With rising output shaft speed, the centrifugal force acting through the primary valve is sufficient to overcome the hydraulic line pressure, which is acting against the shouldered groove face area and will therefore progressively move outwards as the rotational speed increases until the valve borders on an end stop. The opening of the governor valve outlet passage now allows fluid to flow out from the governor, where it is then directed to the large diameter end of the shift valve. This output pressure is known as . With even higher rotational output shaft speed (vehicle speed), greater centrifugal force will be imposed on the secondary valve until it is able to overcome the much larger hydraulic inward load imposed on the large shoulder of this valve. The secondary valve will start to move out from the centre of rotation, uncovering the secondary valve outlet passage so that increased governor pressure passes to the shift valve.

This two stage governor valve action enables the governor to be more sensitive at the very low speeds but not oversensitive at the higher speeds (Fig. 5.5(c)). Sensitivity refers to the amount of fluid pressure increase or decrease for a unit change in rotational speed. If there is a large increase or decrease in governor pressure per unit charge in speed, then the governor is sensitive. If there is very little variation in governor pressure with a change in rotational speed (i.e. vehicle speed), then the governor is insensitive and therefore not suitable for signalling speed changes to the hydraulic control systems.

The reason a single stage governor would not perform satisfactorily over the entire output shift speed range is due to the : at low speeds the build-up in centrifugal force for a small increase in rotational speed is very small, whereas at higher speeds only a small rise in speed produces a considerable increase in centrifugal force. If the governor has the correct sensitivity at high speed it would be insensitive at low speed or if it has the desired sensitivity at low speed it would be far too responsive to governor pressure changes in the higher speed range.

nce the governor pressure end load ($\times$) equals the spring and throttle pressure load (F_S $_T \times$ $_T$) with rising vehicle speed, any further speed increase will push the shift valve plunger towards the spring end to the position shown in Fig. 5.5(a). The fluid on the applied side of the band brake servo piston will now exhaust (drain) through the shift valve to the inlet side of the oil pump. Simultaneously, line pressure from

the manual valve is directed via the shift valve to both the release side of the band servo piston and to the multiplate clutch piston which then energizes the friction plates.

Supply fluid to the spring side of the servo piston (known as the   ), provides a more progressive and controllable transition from one gear change to another which is not possible when relying only on the return spring.

When the vehicle s speed is reduced or the throttle pressure is raised sufficiently, the shift valve plunger will move to the governor pressure end of the valve (Fig. 5.5(a)). The line pressure transmitted to the shift valve is immediately blocked and both the multiplate clutch and the band brake hydraulic feed passages are released of fluid pressure by the middle plunger land uncovering the exhaust part. Simultaneously, as the same middle land covers the right hand exhaust port and uncovers the line pressure passage feeding from the manual valve, fluid will flow to the applied side of the band servo piston, causing the band to contract and so energize the brake.

.   asic four speed hydraulic control system

A simplified hydraulic control system for a four speed automatic transmission will now be examined for the reader to obtain an appreciation of the overall function of the hydraulic computer (control) system.

5.5.1   irst gear (Fig. 5.6)

With the manual valve in   , drive position, fluid is delivered from the oil pump to the pressure regulating valve. It then divides, some being delivered to the torque converter, the remainder passing out to the manual valve as regulating pressure (more commonly known as   ).   ine pressure from the manual valve is then channelled to the forward clutch, which is energized, and to the overdrive band servo on the applied side. At the same time, line pressure from the pressure regulating valve passes through the 3–4 shift valve where it is directed to energize the drive clutch and to the released side of the overdrive band servo, thus preventing the engagement of the band.   ine pressure is also directed to both the governor valve and to the vacuum throttle valve. The reduced pressure output from the governor valve which is known as governor pressure is directed to the end faces of each of the three shaft valves, whereas the output pressure from the throttle valve, known as throttle pressure, is conveyed to the spring end of the 2–3 and 3–4 shift valves.   n the other hand, the 1–2

shift valve spring end is sub ected to line pressure from the manual valve.

Whilst the transmission is in drive first gear the one way clutch will engage, so preventing the reverse planetary carrier from rotating (not shown in hydraulic system).

5.5.   Second gear (Fig. 5.7)

With the manual valve still in   , drive position, hydraulic conditions will be similar to first gear, that is, the overdrive and forward clutches are engaged, except that rising vehicle speed increases the governor pressure sufficiently to push the 1–2 shift valve against both spring and line pressure end loads. As a result, the 1–2 shift valve middle land uncovers the line pressure supply passage feeding from the manual valve.   ine pressure is now directed to the second gear band servo on the applied side, energizing the second gear brake and causing both the forward and reverse sun gears to hold.

If there is a reduction in vehicle speed or if the engine load is increased sufficiently, the resulting imbalance between the spring and throttle pressure load as opposed to governor pressure acting on the 1–2 shift valve at opposite ends causes the shift valve to move against the governor pressure. Consequently the hydraulic circuitry will switch back to first gear conditions, causing the transmission to shift down from second to first gear again.

5.5.3 Third gear (Fig. 5.8)

At even higher road speeds in   , drive position, the governor pressure will have risen to a point where it is able to overcome the spring and throttle pressure load of the 2–3 shift valve. This causes the spool valve to shift over so that the line pressure passage feed from the manual valve is uncovered.   ine pressure will now flow through the 2–3 shift valve where it is directed to the high and reverse clutch to energize the respective fixed and rotating friction plates. At the same time, line pressure passes to the second gear band servo on the release side to disengage the band. Consequently both overdrive and forward planetary gear sets lock-up, permitting the input drive from the torque converter to be transmitted directly through to the transmission s output shaft.

The actual vehicle speed at which the 2–3 shift valve switches over will be influenced by the throttle opening (throttle pressure). A low throttle pressure will cause an early gear upshift whereas a large engine load (high throttle pressure) will raise the upshift speed.
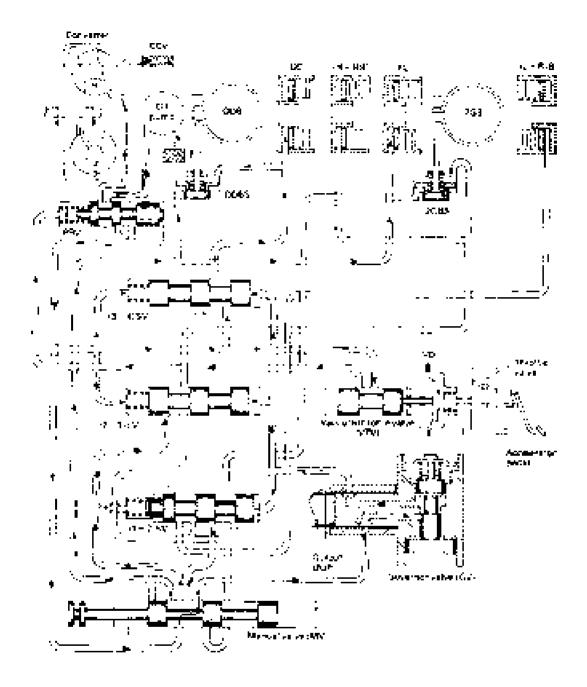
Fig. 5.6 Hydraulic control system ( ) range first gear

5.5. *ourth gear* (Fig. 5.9)

With still higher road speeds in , drive position, the increased governor pressure will actuate the 3–4 shift valve, forcing it to shift across so that it covers up the line pressure supply passage and at the same time uncovers the exhaust or drain port. As a result, the line pressure exhausts from the release side of the overdrive band servo which then permits the band to be energized. At the same time the drive clutch will be de-energized because of the collapse of line pressure as it is released through the 3–4 shift valve exhaust port.
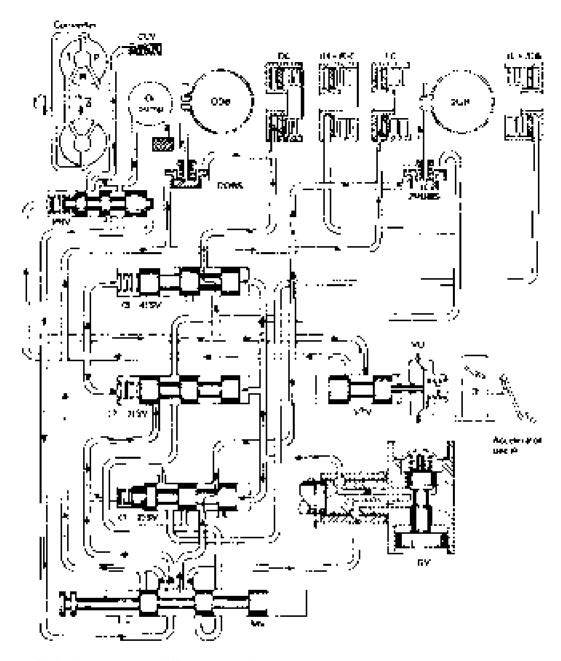
Fig. 5.7 Hydraulic control system ( ) range second gear

Under these operating conditions the overdrive shaft planetary gear set reduces the intermediate shift speed and, since the forward clutch is in a state of lock-up only, this speed step up is transmitted through to the output shaft.

5.5.5 *everse gear* (Fig. 5.10)

With the manual valve in R, reverse position, line pressure from the manual valve is directed via the 2–3 shift valve to the release side of the second gear band servo, causing the band to disengage. At the
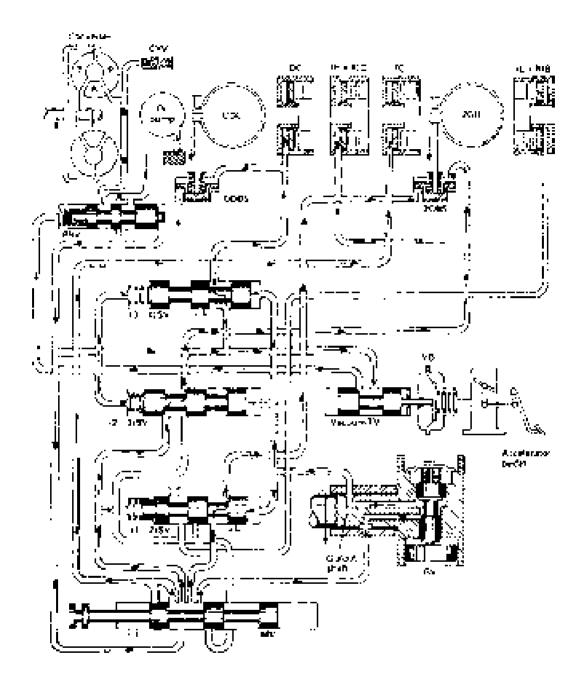
Fig. 5.8 Hydraulic control system ( ) range third gear

same time line pressure from the same supply passage engages the high and reverse clutch. The manual valve also supplies line pressure to the low and reverse band brake via the 1–2 shift valve to hold the reverse planetary carrier. In addition, line pressure from the pressure regulating valve output side is directed via the 3–4 shift valve to the release side of the overdrive brake servo to disengage the band and to the drive clutch piston to engage the friction plates. Note that both band brake servos on the applied sides have been exhausted of line pressure and so has the forward clutch piston chamber.
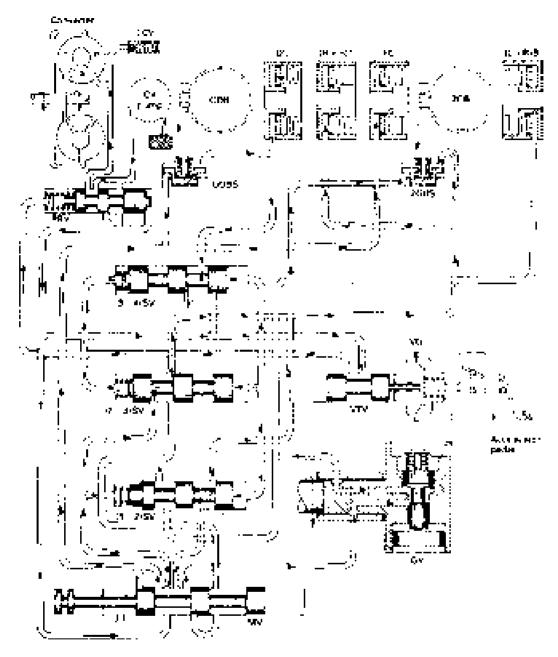
Fig. 5.9 Hydraulic control system ( ) range fourth gear

5.5.6 *oc up tor ue converter* (Fig. 5.11)

ntroduction To overcome the inherent relative slip which always occurs between the torque converter s pump impeller and the turbine runner, even driving at moderate speeds under light load conditions, a lock-up friction clutch may be incorporated between the input pump impeller and the turbine output shaft. The benefits of this lock-up can only be realised if the torque converter is allowed to operate when light torque demands are made on the engine and only when the converter is operating above its torque multiplication range that is beyond the coupling point. Consequently, converter
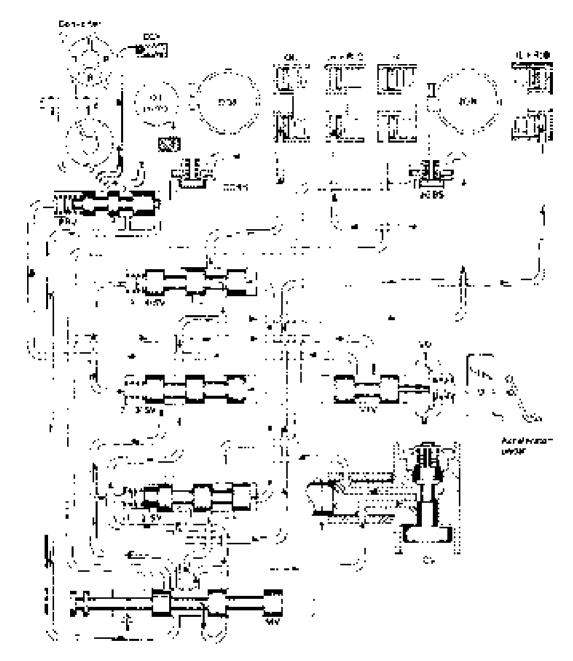
Fig. 5.10 Hydraulic control system ( ) reverse gear

lock-up is only permitted to be implemented when the transmission is in either third or fourth gear. The advantages of bypassing the power transfer through the circulating fluid and instead transmitting the engine s output directly to the transmission input shaft eliminates drive slippage, thereby increasing the power actually propelling the vehicle. ue to this net gain in power output, fuel wastage will be reduced.

ock-up clutch description The lock-up clutch consists of a sliding drive plate which performs two functions firstly to provide the friction coupling device and secondly to act as a hydraulic con-
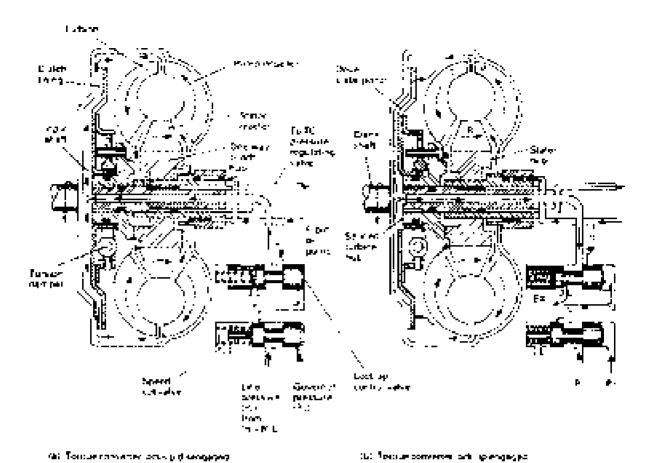
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Fig. 5.11 (a and b)   ock-up tor ue converter

trolled piston to energize or de-energize the clutch engagement facings. The lock-up drive plate/piston is supported by the turbine hub which is itself mounted on the transmission input shaft. A transmission damper device, similar to that used on a conventional clutch drive-plate, is incorporated in the lock-up plate to absorb and damp shock impacts when the lock-up clutch engages.

ock-up control   The automatic operation of the converter lock-up is controlled by a speed cut valve and a lock-up control valve. The function of these valves is to open and close fluid passages which supply and discharge fluid from the space formed between the torque converter casing and the lock-up drive plate/piston.

ock-up disengaged (Fig. 5.11(a))   With the vehicle driven in either first or second gear at relatively

low speeds, low governor pressure permits the speed cut and lock-up control valve return springs to push their respective plunger to the right. Under these conditions, pressurized fluid from the torque converter flows into the space separating the lock-up plate/piston from the turbine. At the same time, fluid from the oil pump is conveyed to the space formed between the torque converter s casing and the lock-up plate/piston via the lock-up control valve and the central axial passage in the turbine input shaft. Consequently, the pressure on both sides of the lock-up plate will be equalized and so the lock-up plate/piston cannot exert an engagement load to energize the friction contact faces.

ock-up engaged (Fig. 5.11(b))   As the speed of the vehicle rises, increased governor pressure will force the speed cut valve plunger against its spring until it uncovers the line pressure passage leading into the right hand end of the lock-up control

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valve. ine pressure fed from the high and reverse clutch is directed via the speed cut valve to the right hand end of lock-up control valve, thereby pushing its plunger to the left to uncover the lock-up clutch drain port. Instantly, pressurized fluid from the chamber created between the torque converter casing and lock-up plate/piston escapes via the central input shaft passage through the wasted region of the lock-up control valve plunger back to the inlet side of the oil pump. As a result, the difference of pressure across the two sides of the lock-up plate/piston causes it to slide towards the torque converter casing until the friction faces contact. This closes the exit for the converter fluid so that full converter fluid pressure is exerted against the lock-up plate/piston.

ence the input and output shafts are now locked together and therefore rotate as one.

Speed cut valve function The purpose of the speed cut valve is to prevent fluid draining from the space formed between the converter casing and lock-up plate/piston via the lock-up control valve if there is a high governor pressure but the transmission has not yet changed to third or fourth gear. Under these conditions, there is no line pressure in the high and reverse clutch circuit which is controlled by the shift valve. Therefore when the speed cut valve plunger moves to the left there is no line pressure to actuate the lock-up control valve so that the lock-up plate/ piston remains pressurized on both sides in the disengaged position.

. Three speed and reverse transa le automatic transmission mechanical power flow
(ear train as adopted by some Austin-Rover, VW and Audi 1.6 litre cars)
The operating principle of the mechanical power or torque flow through a transaxle three speed

automatic transmission in each gear ratio will now be considered in some depth, see Fig. 5.12.

The planetary gear train consists of two sun gears, two sets of pinion gears (three in each set), two sets of annular (internal) gears and pinion carriers which support the pinion gears on pins. elical teeth are used throughout.

For all forward gears, power enters the gear train via the forward annular gear and leaves the gear train by the reverse annular gear. In reverse gear, power enters the gear train by the reverse sun gear and leaves the gear train via the reverse annular gear.

First gear compounds both the forward gear set and the reverse gear set to provide the necessary low gear reduction. Second gear only utilizes the forward planetary gear set to produce the intermediate gear reduction. Third gear is achieved by locking the forward planetary gear set so that a straight through drive is obtained. With planetary gear trains the gears are in constant mesh and gear ratio changes are effected by holding, releasing or rotating certain parts of the gear train by means of a one way clutch, two multiplate clutches, one multiplate brake and one band brake.

The operation of the automatic transmission gear train can best be explained by referring to Table 5.2 which shows which components are engaged in each manual valve selection position.

5.6.1 *Selector lever* (Table 5.2)
The selector lever has a number of positions marked P R N 2 1 with definite functions as follows:

P   park When selected, there is no drive through the transmission. A mechanical lock actuated by a linkage merely causes a parking pawl to engage in the slots around a ring gear attached to the output shaft (Fig. 5.2). Thus the parking pawl

ab 5.2 anual valve selection position

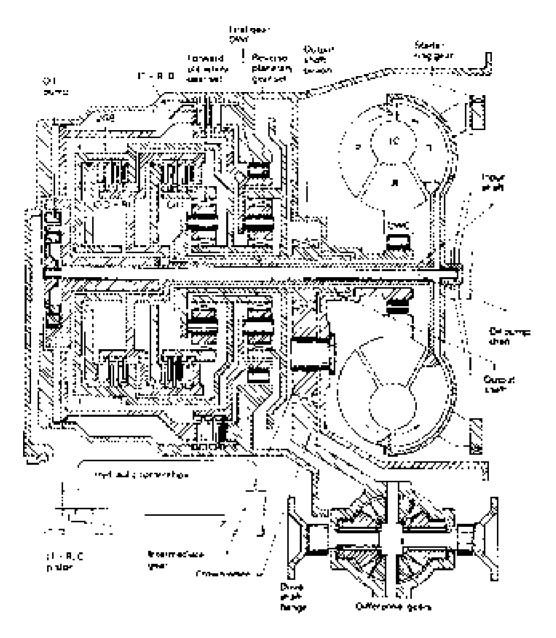| Range	Forward clutch FC	rive and reverse clutch (R)C	First and reverse brake (R)B	Second gear band 2 B	ne way clutch C	Ratio
and N	–	–	–	–	–	–
– 1st 2 – 1st	Applied	–	–	–	Applied	2 1 1
1 – 1st	Applied	–	Applied	–	–	2 1 1
– 2nd 2 – 2nd	Applied	–	–	Applied	–	1 1
– rd	Applied	Applied	–	–	–	1 1
R	–	Applied	Applied	–	–	2 1

Fig. 5.12 Transaxle three speed automatic transmission layout

locks the output shaft to the transmission casing so that the vehicle cannot roll backwards or forwards. This pawl must not be engaged whilst the vehicle is moving. The engine may be started in this position.

R reverse When selected, the output shaft from the automatic transmission is made to rotate in the opposite direction to produce a reverse gear drive.

The reverse position must only be selected when the vehicle is stationary. The engine will not start in reverse position.

neutral When selected, all clutches and band brake are disengaged so that there is no drive through the transmission. The engine may be started in N   neutral range.

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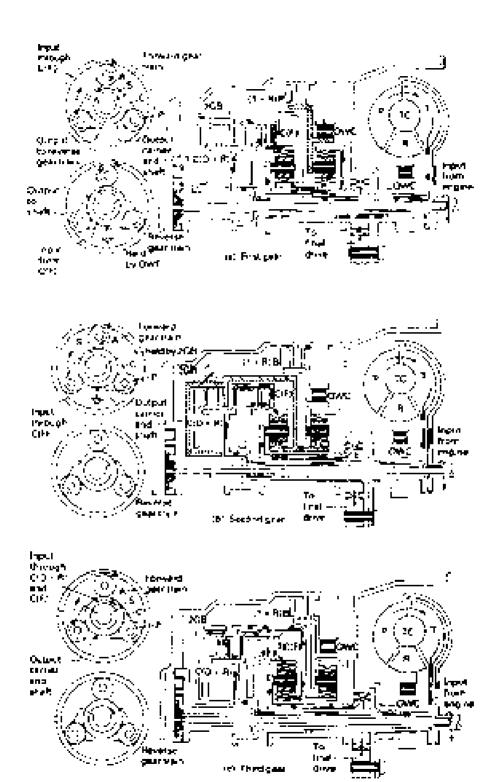
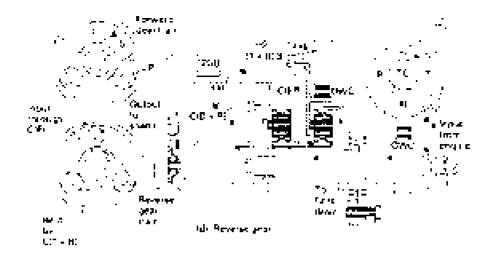**Fig. 5.13 (a–d)** Three speed and reverse automatic transmission transaxle units

Fig. 5.13 *contd*

drive This position is used for all normal driving conditions, automatically producing 1–2, 2–3 upshifts and 3–2, 2–1 downshifts at suitable road speeds or according to the position of the accelerator pedal. The engine will not start in drive range.

2 irst and second This position is selected when it is desired to restrict gear changes automatically from 1–2 upshift and 2–1 downshifts only. The selector must not be positioned in 2 range above 100 km/h (70 mph). The engine will not start in this range position.

1 irst gear When this range is selected, the transmission is prevented from shifting into second and third gear. A friction clutch locks out the one way roller clutch so that better control may be obtained when travelling over rough or wet ground or icy roads. Engine braking on overrun is available when descending steep hills.

5.6. *irst gear D 1st* (Fig. 5.13(a))
With the manual selector valve in range, engine torque is transmitted from the converter through the applied forward clutch to the annular gear of the forward planetary gear train. The clockwise rotation of the forward annular gear causes the forward planet gears to rotate clockwise, driving the double (compound) sun gear anticlockwise. The forward planetary carrier is splined to the output shaft. This causes the planet gears to drive the double sun gear

instead of rolling walking around the sun gears. This counterclockwise rotation of the sun gears causes the reverse planet gears to rotate clockwise. With the one way clutch holding the reverse planet carrier stationary, the reverse planetary gears turn the reverse annular gear and output shaft clockwise in a reduction ratio of something like 2.71:1.

When first gear is selected in the range, a very smooth transmission take-up is obtained when the one way clutch locks, but on vehicle overrun the one way clutch is released so that the transmission freewheels.

5.6.3 *irst gear manual 1 1st* (Fig. 5.13(a))
The power flow in first gear manual differs from the range in that the first and reverse brake are applied to hold the reverse planet carrier stationary. Under these conditions on vehicle overrun, engine braking is provided.

5.6. *Second gear D nd* (Fig. 5.13(b))
In range in second gear, the forward clutch and the second gear band brake are applied. The forward clutch then transmits the engine torque from the input shaft to the forward annular gear and the second gear band brake holds the double sun gear stationary. Thus engine torque is delivered to the annular gear of the forward planetary train in a clockwise rotation. Consequently, the planet gears are compelled to revolve on their axes and roll walk around the stationary sun gear in a clockwise direction. As a result the output shaft, which is splined to the forward planet carrier, is made to turn in a clockwise direction at a slower speed

relative to the input shaft with a reduction ratio of approximately 1.50:1.

5.6.5 Third gear D 3rd (Fig. 5.13(c))

In range engine torque is transmitted through both forward clutch and drive and reverse clutch. The drive and reverse clutch rotate the sun gear of the forward gear train clockwise and similarly the forward clutch turns the annular gear of the same gear set also clockwise. With both the annular gear and sun gear of the forward gear train revolving in the same direction at the same speed, the planet gear becomes locked in position, causing the forward gear train to revolve as a whole. The output shaft, which is splined to the forward planet carrier, therefore rotates at the same speed as the input shaft, that is as a direct drive ratio 1:1.

5.6.6 everse gear (Fig. 5.13(d))

With the manual selector valve in the R position, the drive and reverse multiplate brake is applied to transmit clockwise engine torque to the reverse gear set sun gear. With the first and reverse brake applied, the reverse planet gear carrier is held stationary. The planet gears are compelled to revolve on their own axes, thereby turning the reverse annular gear which is splined to the output shaft in an anticlockwise direction in a reduction ratio of about 2.43:1.

.7 ydraulic gear selection control components
(Fig. 5.24)
(Three speed and reverse transaxle automatic transmission)
An explanation of how the hydraulic control system is able to receive pressure signals which correspond to vehicle speed, engine load and the driver s requirements, and how this information produces the correct up or down gear shift through the action of the control system s various plunger (spool) valves will now be considered by initially explaining the function of each component making up the control system.

A list of key components and abbreviations used in the description of the hydraulic control system is as follows:

1	Manual valve	MV
2	ickdown valve	V
3	Throttle pressure valve	TPV
4	Valve for first gear manual range	V(1)MR

5	1–2 shift valve	(1–2)SV
6	1–2 governor plug	(1–2) P
7	Throttle pressure limiting valve	TP V
8	Main pressure limiting valve	MP V
9	Main pressure regulating valve	MPRV
10	Valve for first and reverse gear brake	V(1 R) B
11	Converter pressure valve	CPV
12	Soft engagement valve	SEV
13	2–3 shift valve	(2–3)SV
14	2–3 governor plug	(2–3) P
15	Valve for direct and reverse clutch	V(R)C
16	3–2 control valve	(3–2)CV
17	3–2 kickdown valve	(3–2) V
18	overnor valve	V
19	Forward clutch piston	FCP
20	il pump	P
21	Converter check valve	CCV
22	Second gear band servo	2 BS
23	Accumulator	A
24	Forward clutch piston	FCP
25	irect and reverse clutch piston	(R)CP
26	First and reverse brake piston	(1 R)BP
27	ne way clutch	WC

5.7.1 The pressure supply system
This consists of an internal gear crescent oil pump driven by the engine via a shaft splined to the torque converter impeller. The oil pressure generated by the oil pump is directed to the pressure regulating valve. By introducing limited throttle pressure into the regulator valve spring chamber, the thrust acting on the left hand end of the valve is increased during acceleration. This prevents the regulator valve being pushed back and spilling oil into the intake side of the oil pump. As a result, the line pressure will rise as the engine speed increases.

5.7. ain pressure regulator valve
(Fig. 5.14(a and b))
This valve controls the output pressure which is delivered to the brake band, multiplate brake and clutch servos. il pressure from the pump acts on the left hand end of the valve and opposes the return spring. This oil pressure moves the valve to the right, initially permitting oil to pass to the converter pressure valve and its circuit, but with further valve movement oil will be exhausted back to the pump intake passage. The line pressure build-up is also controlled by introducing limited throttle pressure into the regulator spring chamber

Fig. 5.14 (a and b)   ain pressure regulating valve (R)

which assists the spring in opposing the valve moving to the right. In addition, oil pressure from the manual valve passage, indirectly controlled by the governor, is imposed on the left hand end of the regulator valve. This modifies the valve movement to suit the various gear train and road condition requirements.

5.7.3 Throttle pressure valve   T
(Fig. 5.15(a and b))
The throttle pressure valve transmits regulated pressure based on engine throttle position.   pening or closing the engine throttle moves the kick-down valve spool so that the throttle valve spring tension is varied. The amount of intermediate pressure allowed through the throttle pressure valve is determined by the compression of the spring. The reduced pressure on the output side of the throttle valve is then known as throttle pressure. Throttle pressure is directed to the main pressure limiting valve, the   , and to one end of the shift valves in opposition to governor pressure,

which acts on the other end of the shift valves controlling upshift and downshift speeds.

5.7.   ain pressure limiting valve
(Fig. 5.16)
This valve is designed to limit or even cut off the variable throttle pressure passing through to the main regulating valve and the soft engagement valve. The pressure passing out from the valve to the main pressure regulator valve is known as   . As the pressure passes through the valve it reacts on the left hand end of the main pressure limiting valve so that the valve will progressively move to the right, until at some predetermined pressure the valve will close the throttle pressure port feeding the main pressure regulating valve circuit.

When the throttle pressure port closes, the high pressure in the regulator spring chamber is permitted to return to the throttle pressure circuit via the non-return ball valve.
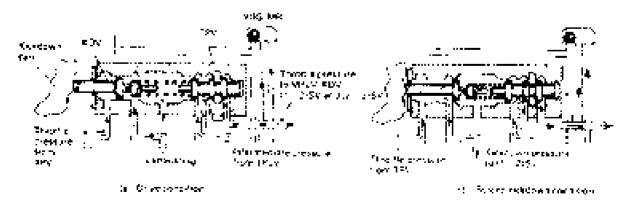

Fig. 5.15 (a and b)   ickdown valve (   ) throttle pressure valve (T   ) and valve for first gear manual range (1   R)
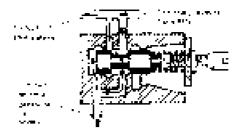
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Fig. 5.16 ain pressure limiting valve

5.7.5 *Converter pressure valve C* (Fig. 5.17)

This valve shuts off the oil supply to the torque converter once the delivery pressure reaches 6 bar.

ine pressure from the main pressure regulator valve passes through the valve to the torque converter and acts on its right hand end until the preset pressure is reached. At this point the valve is pushed back against its spring, closing off the oil supply to the torque converter until the converter pressure is reduced again.

The force on the output side of the converter pressure valve feeding into the converter is known as converter pressure.

5.7.6 *Converter chec valve CC*

This valve, which is located inside the stator support, prevents the converter oil drainage when the vehicle is stationary with the engine switched off. This valve is not shown in the diagrams.

5.7.7 *Throttle pressure limiting valve T*
(Fig. 5.18)

This valve converts line pressure, supplied by the pump and controlled by the main pressure regulator valve, into intermediate pressure. The pressure reduction is achieved by line pressure initially passing through the diagonal passage in the valve so that it reacts against the left hand end of the valve. Consequently the valve shifts over and partially reduces the line pressure port opening. The reduced output pressure now known as intermedi-

ate pressure then passes to the throttle pressure valve.

5.7. *ic do n valve D*
(Fig. 5.15(a and b))

This valve permits additional pressure to react on the shift valves and governor plugs when a rapid acceleration (forced throttle) response is required by the driver so that the governor pressure is compelled to rise to a higher value before a gear upshift occurs. When the throttle is forced wide open, the kickdown valve is moved over to the right, thus allowing throttle pressure to pass through the valve. The output pressure is known as kickdown pressure. The kickdown pressure feeds in between both 1–2 and 2–3 shift valves and governor plug combinations. As a result, this kickdown pressure opposes and delays the governor pressure movement of the governor plug and shift valve, thereby preventing a gear upshift occurring until a much higher speed is reached.

5.7. 1 *Shift valve and governor plug 1 S and 1* (Fig. 5.19)

This valve combination automatically controls and shifts the transmission from first to second or from second to first depending upon governor and throttle pressure. When governor pressure on the right hand governor plug side overcomes throttle pressure on the left hand 1–2 shift valve side, both
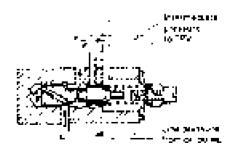

Fig. 5.17 Converter pressure valve (C)

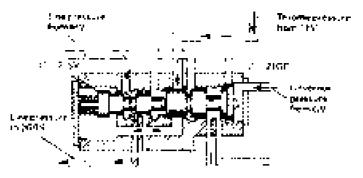**Fig. 5.18** Throttle pressure limiting valve (T)

Fig. 5.19 1–2 shift valve (1–2)S and 1–2 governor plug (1–2) in 1–2 upshift condition

1–2 governor plug and 1–2 shift valve move to the left thereby opening the line pressure port which delivers oil from the pump. ine pressure will now pass unrestricted through the valve to feed into the brake band servo. As a result an upchange occurs. If, in addition to the throttle pressure, kickdown pressure is introduced to the valve combination, gear upshifts will be prolonged. If 1 manual valve is selected, line pressure will be supplied to the governor plug chamber (large piston area) and the throttle spring chamber, preventing a 1–2 upshift. 1 manual position cannot be engaged at speeds above 72 km/h because the 1–2 shift valve cannot move across, due to the governor pressure.

5.7.10 3 Shift valve and governor plug
3 S and 3 (Fig. 5.20(a and b))

The 2–3 shift valve and governor plug control the gear change from second to top gear or from top to second depending upon governor and throttle pressure. As governor pressure exceeds throttle pressure, the shift valve and governor plug are pushed over to the left. This permits line pressure to pass through the valve so that it can supply pressure to the drive and reverse clutch piston, so that an upchange can now take place. When 2 manual valve position is selected, there is no pressure feeding to the shift valve which therefore prevents a 2–3 upshift.
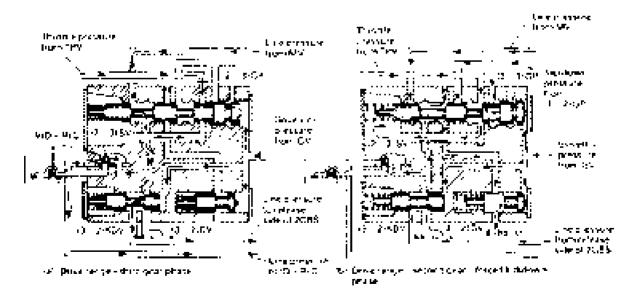

Fig. 5.20 (a and b) 2– shift valve (2–)S 2– governor plug (2–)   –2 control valve (–2)C   –2 kickdown valve (–2)   and valve for direct and reverse clutch   (R)C

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5.7.11 3   ic do n valve 3   D
(Fig. 5.20(a and b))
This valve is provided to prolong the downshift from third to second gear during rapid acceleration from above 90 km/h so that the change takes place relatively smoothly. With rising output shaft speed, the governor pressure acting on the right hand end of the valve moves it to the right, thus practically restricting the oil outflow from the servo spring chamber and therefore extending the second gear band engagement time.

5.7.1 3   Control valve 3   C
(Fig. 5.20(a and b))
This valve controls the expulsion of oil from the spring side of the second gear band servo piston at speeds in the region of 60 km/h. The time period for oil to exhaust then depends upon the governor pressure varying the effective exhaust port restriction.   ine pressure oil from the spring side of the second gear band servo piston passes through a passage leading to the 3–2 kickdown valve annular groove and from there to the 2–3 shift valve annular groove.   ere some oil exhausts out from a fixed restriction while the remainder passes via a passage

to the 3–2 control valve. As the vehicle speed approaches 60 km/h the governor pressure rises sufficiently to force back the 3–2 control valve piston, thus causing the wasted (reduced diameter) part of the control valve to complete the exhaustion of oil.

5.7.13   alves for direct and reverse clutch
D   C (Fig. 5.21(a and b))
When the manual selector valve is moved to reverse position the left hand ball valve drops onto its seat so that line pressure oil from the manual selector is compelled to move through a restriction. At the same time, the right hand ball valve is pushed to the right, immediately closing off the second gear servo piston spring side from the line pressure. The right hand ball valve is dislodged to the left when third gear is selected so that the manual reverse line pressure is prevented feeding the drive and reverse clutch piston.   uring the time third gear is selected the left hand ball serves no purpose.

5.7.1   alve for first gear manual range
1   (Figs 5.15(a and b) and 5.22)
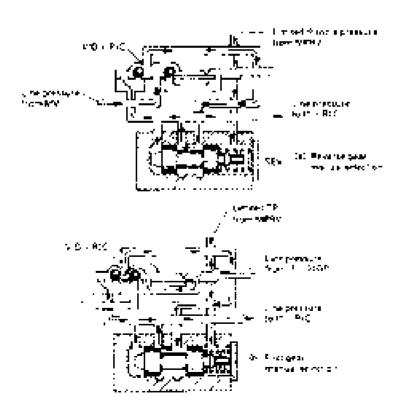The selection of first gear manual supplies line pressure to the underside passage to the ball valve,

Fig. 5.21 (a and b)   Soft engagement valve (SE) valve for first and reverse clutch   (1   R) C
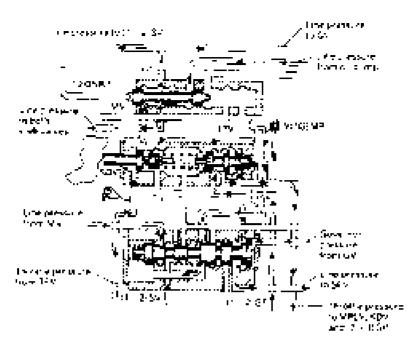
Fig. 5.22 anual valve ( ) kickdown valve ( ) throttle pressure valve (T) 1–2 shift valve (1–2)S and 1–2 governor plug (1–2) in first gear – manual selection

causing it to move to the right. ine pressure then fills the throttle pressure lines leading to the left hand end of the 1–2 shift valve and therefore a 1–2 upshift is prevented.

5.7.15 alves for first and reverse gear bra e
1 B (Fig. 5.20(a and b))

The selection of first gear manual position causes line pressure to dislodge the right hand ball valve to the left, thereby closing the reverse line passage from the selector valve. At the same time the left hand ball valve closes so that line pressure flow for the engagement of the first and reverse multiplate is slowed down.

The selection of reverse gear causes the right hand ball valve to be pushed by line pressure to the right and so the first gear line passage from the selector valve is closed. Similarly the left hand ball valve closes so that line pressure flow to the first and reverse brake is restricted, thus prolonging the clutch engagement period.

5.7.16 Soft engagement valve S
(Fig. 5.21(a and b))

This valve provides a cushioning effect for the engagement of first and reverse gear brake. This effect is achieved by line pressure acting on the left hand valve end pushing the valve to the right

against the opposing variable throttle pressure. The result of this movement is to restrict and slow down the pressure build-up on the first and reverse gear brake piston.

5.7.17 Clutches and bra es

ront clutch piston (P) (Fig. 5.24) This is an annular shaped piston which directs a clamping load to a multiplate clutch via a diaphragm type spring when line pressure is introduced behind the piston. The engagement of the clutch couples the output shaft from the torque converter turbine to the forward annular gear ring. The forward clutch is applied in all forward drive gear ranges.

irst and reverse brake piston (1 R) P (Fig. 5.24) Introducing line pressure to the first and reverse clutch piston cylinder engages the multiplate brake which locks the reverse planetary carrier to the transmission casing. The first and reverse brake is applied only in first and reverse range.

rive and reverse clutch piston (R) P (Fig. 5.24) irecting line pressure behind the drive and reverse clutch piston applies the clutch,

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thereby transmitting torque from the torque converter turbine output shaft to the forward sun gear. When the forward clutch is also applied, the forward planetary gears (annular, planet and sun gears) are locked together and they rotate bodily, thus producing a straight through 1:1 third gear drive. owever, when the first and reverse clutch is applied instead of the forward clutch, the reverse planetary carrier is held stationary causing the reverse gear reduction ratio to be engaged.

The timing of the release of one set of gears and the engagement of another to produce smooth up and down gear shifts between second and third gears is achieved by carefully controlling the delivery and exhaustion of hydraulic fluid from the clutch and band brake servo.

These operating conditions are explained under second gear band servo.

Second gear band servo (2 S) (Fig. 5.24) This is a double acting piston servo which has a small piston area to apply the band brake and a large piston area which is on the release spring chamber side of the servo.

irecting line pressure to the small piston area chamber of the servo applies the band brake against the resistance of the return spring and thereby holds stationary both sun gears. Introducing line pressure on the large piston area spring chamber side of the servo produces an opposing force which releases the grip on the band brake. The piston returns to the off position and the relaxing of the band brake is made possible by the difference in piston area on each side, both sides being sub ected to the same line pressure. The band brake is applied only in the second gear forward speed range.

uring upshift from 2–3 it is important that the second gear band brake does not release too quickly relative to the drive and reverse clutch engagement, in order to avoid (rapid engine speed surge) during the transition from 2nd to 3rd gear. uring downshift it is also important that the second gear band brake does not engage before the drive and rear clutch releases in order to avoid (gear amming) on the 3–2 shift.

The 3–2 control valve and the 3–2 kickdown valves therefore affect the timing relationship between the second gear band servo and the drive and reverse clutch to provide correct shift changes under all operating conditions.

irst gear one way clutch ( ) (Fig. 5.13(a)) When in drive range, the one way roller type clutch operates in place of the first and reverse multiplate brake to prevent the rotation of the reverse pinion carrier. This one way clutch enables the gear set to freewheel on overrun and to lock-up on drive, therefore preventing a erky gear ratio in 1–2 upshift and 2–1 downshift.

5.7.1 *The governor valve*
(Figs 5.23 and 5.24)
The governor revolving with the transmission output shaft is basically a pressure regulating valve which reduces line pressure to a value that varies with output vehicle speed. This variable pressure is known as governor pressure and is utilized in the control system to effect up and down gear shifts from 1–2 and 2–3 shift valves. overnor pressure opposes shift valve spring force, throttle pressure and kickdown pressure, and the resulting force acting on the governor plug and shift valve determines the vehicle s gear change speeds. The governor drive is achieved through a skew gear meshing with a ring gear mounted on the reverse annular gear carrier which is attached to the output pinion shaft.

The two types of governor valves used for this class of automatic transmission are the ball and pivot flyweight and the plunger and flyweight. These governors are described below.

Plunger and flyweight type governor (Fig. 5.23) Rotation of the governor at low speed causes the governor weight and valve to produce a centrifugal force. This outward force is opposed by an equal and opposite hydraulic force produced by governor pressure acting on the stepped annular area of the governor valve. Because the governor valve is a regulating valve, and will attempt to remain in equilibrium, governor pressure will rise in accordance with the increase in centrifugal force caused by increased rotational speed. As the output shaft speed increases, the governor weight moves outwards (due to the centrifugal force) to a stop in the governor body, when it can move no further. When this occurs, the governor spring located between the weight and the governor valve becomes effective. The force of this spring then combines with centrifugal force of the governor valve to oppose the hydraulic pressure, thus making the pressure less sensitive to output shaft speed variation. Therefore the governor provides two distinct phases of regulation, the first being used for accurate control of the low speed shift points.
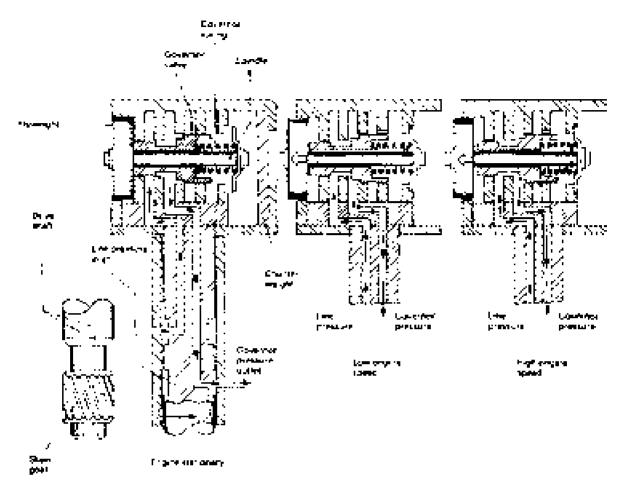
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Fig. 5.23 lunger and governor valve

all and pivot flyweight type governor (Fig. 5.24) This type of governor consists of a ball valve controlled by a hinged flyweight and a pressure relief ball valve. Fluid from the oil pump at line pressure is introduced via a restriction into an axial passage formed in the governor drive shaft. When the transmission output shaft stops rotating (vehicle stationary) with the engine idling, fluid pressure forces the governor ball valve off its seat, permitting fluid to escape back to the sump. Rotation of the output shaft as the vehicle accelerates from a standstill causes the flyweight centrifugal force to close the ball valve. Therefore fluid trapped in the governor drive shaft passage, known as governor pressure, has to reach a higher pressure before fluid exhausts through the valve. By these means the line pressure is regulated to a valve that varies with the output shaft and vehicle speed. A pressure relief valve is also included to safeguard the system

from excessively high pressure if the governor valve malfunctions.

5.7.1 *ydraulic accumulator* (Fig. 5.24)
This is a cylinder and spring loaded piston which is used to store a small amount of pressure energy to enable a rapid flow of fluid under pressure to one of the operating components or to absorb and smooth fluctuating fluid delivery. The piston is pushed back when the fluid pressure exceeds the spring load and fluid enters and fills up the space left behind by the displaced piston.

With the transmission in neutral or park, line pressure from the pressure pump enters the accumulator at the opposite end to the spring, thereby displacing the piston and compressing the spring. When the hydraulic control shifts into the second gear phase, line pressure from the 1–2 shift valve is

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directed to the second gear band servo applied end and the spring end of the accumulator.

When the accumulator spring is compressed, fluid from the supply can flow rapidly to the applied side of the band servo piston. As soon as the servo piston meets resistance (starts to apply its load), the fluid pressure increases and the accumulator piston spring is extended as the piston is pushed back by the spring. This is because there is equal line pressure acting on either side of the accumulator piston and so the spring is able to apply its load and extend. As a result, the supply of fluid is reduced to the applied side of the second gear band servo piston. The accumulator therefore smooths and times the application of the second gear band brake in order to reduce the risk of shock and a erky operation. In addition, the extra quantity of fluid in the system due to the accumulator leads to a slow rate of release of the servo piston and band.

. ydraulic gear selection control operation

5. .1 luid flo in neutral (Fig. 5.24)
Pressurized oil from the pump flows to the main pressure regulating valve. The valve shifts over due to the oil pressure, thus opening a passage supplying the torque converter via the converter pressure valve. Increased pump pressure moves the valve further until it uncovers the exhaust port dumping the oil back into the oil pump suction intake passage.

The oil pressure generated between the pump and main pressure limiting valve is known as line pressure and is directed to the throttle pressure limiting valve, accumulator, manual selection valve and through the latter valve to the 2–3 shift valve.

When the throttle foot released, the intermediate pressure exhausts so that there will be no throttle pressure. epressing the throttle pedal increases the throttle spring tension and creates a throttle pressure which is then directed to the kickdown valve, main pressure limiting valve, 1–2 shift valve and the 2–3 shift valve.

At the same time, a limited throttle pressure is created between the main pressure limiting valve and main pressure regulating valve. This pressure is also directed to the soft engagement valve.

With the manual gear selector valve in neutral, there is no line pressure to the rear of the main regulating valve and therefore the trapped line pressure will be at a maximum.

5. . luid flo in par (Fig. 5.24)
With the manual valve in park position, the hydraulic flow is the same as in neutral, except there is no line pressure to the 2–3 shift valve and the main pressure regulating valve provides maximum increase in line pressure.

5. .3 luid flo in drive range first gear
(Figs 5.24 and 5.19)
Both main pressure regulating valve and throttle pressure valve operate as for neutral and park.

With the manual selector valve in , line pressure is directed to the 1–2 shift valve, 2–3 shift valve and to the forward clutch which it engages.

The shift valve is sub ected to governor pressure at one end which opposes the spring tension and throttle pressure at the opposite end. As the car speed increases governor pressure will overcome throttle pressure causing a 1–2 upshift to take place.

Throughout this period a reduced line pressure reacts against the left hand end of the main pressure regulating valve. The valve movement then allows more oil to pass to the torque converter, thereby causing a reduction in line pressure to occur.

5. . luid flo in drive range second gear
(Figs 5.24 and 5.19)
As for drive range first gear, the main regulator valve and throttle pressure valve function as for neutral and park.

When the manual selector valve is positioned in , line pressure is directed to the 1–2 shift valve, 2–3 shift valve and to the forward clutch which it applies.

The rising governor pressure imposes itself against one end of the 1–2 governor plug counteracting throttle pressure until at some predetermined pressure difference the valve shifts across. This then permits line pressure to flow to the accumulator and the second gear servo piston thus causing the second gear brake band to be applied.

At the same time, full line pressure is applied to the left hand end of the main pressure regulating valve so that there will be a further decrease in line pressure relative to drive range – first gear operating conditions.

5. .5 luid flo in drive range third gear
(Figs 5.24 and 5.20(a))
As for drive range – first and second gears, the main regulator valve and throttle pressure valve perform as for neutral and park.

The manual selector valve will still be in position so that line pressure is directed to both 1–2 and
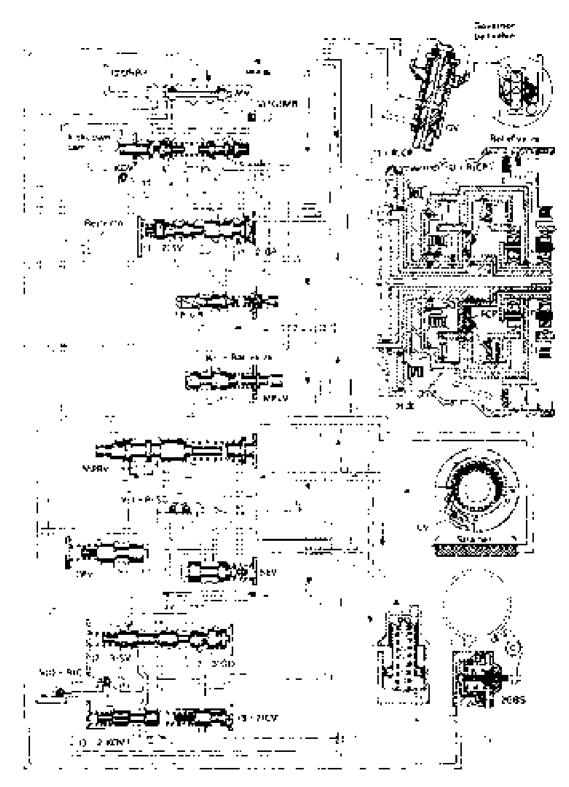
Fig. 5.24 Three speed automatic transmission hydraulic control system in neutral position

2–3 shift valves and to the forward clutch piston which clamps the friction clutch plates.

Increased governor pressure acting on the 2–3 governor plug moves the ad acent 2–3 shift valve over. This allows line pressure to flow around the wasted region of the shift valve to the 3–2 kickdown valve and from there to the second gear band servo spring chamber side (release) of the piston. Simultaneously, line pressure passes from the 2–3 shift valve through the right hand ball valve orifice for direct and reverse clutch to the direct and reverse clutch piston which is accordingly engaged.

The main pressure regulating valve will be sub ected to full line pressure acting on its left hand end so that line pressure reduction will be as in drive range second gear.

5. .6 luid flo in first gear manual selection (Figs 5.24 and 5.15)

With the manual selector in 1 position, line pressure passes to the forward clutch piston and accordingly applies the clutch plates. ine pressure from the manual selector valve moves the ball valve for the first gear manual range so that it cuts off throttle pressure to the 1–2 shift valve. ine pressure is therefore able to pass through the ball valve for first and reverse gear to the 1–2 governor plug, the soft engagement valve and finally passing to the first and reverse brake piston to engage the brake plates.

Consequently, line pressure will fill the normal throttle pressure lines of 1–2 shift valve and will react against the left hand end of the valve. This then prevents governor pressure at the opposite end acting on the governor plug moving the valve for a 1–2 upshift.

5. .7 luid flo in second gear manual selection (Figs 5.24 and 5.19)

uring this phase of gear change, the main pressure regulating valve and throttle pressure valve operate as for neutral and park.

When the manual selector valve is in 2 position, line pressure passes to the forward clutch piston to apply the clutch plates. Similarly, line pressure is also directed to the middle of the 1–2 shift valve. When road speed is high enough, governor pressure will be sufficient to push the valve to one side, thus uncovering the port feeding the accumulator and the second gear band servo piston on the applied side.

In the second gear manual selection position, line pressure passing to the 2–3 shift valve is blocked so that a 2–3 upshift is prevented.

In first gear a reduced pressure is applied at the end of the main pressure regulating valve, causing a

reduced line pressure to be created. ikewise in second gear full line pressure will act behind the main pressure regulating valve so that the line pressure is further reduced.

5. . luid flo in drive range second gear forced ic do n (Figs 5.24 and 5.15(b))

Similarly as for all other manual selector positions, the main pressure regulating valve and throttle pressure valve operate in the same way as for neutral and park.

With the manual selector valve in and the accelerator pedal fully depressed, the kickdown valve reduced waste aligns with the kickdown line outlet port thus causing throttle pressure to flow into the kickdown lines.

ickdown pressure now flows to 1–2 and 2–3 governor plugs assisting the throttle pressure applied on the 1–2 and 2–3 shift valves. When the governor pressure is low enough, throttle pressure and kickdown pressure overcomes governor pressure, causing the 2–3 governor plug to move to the right. As a result, the 2–3 shift valve moves to exhaust oil from the drive and reverse clutch piston chamber and the second gear band servo piston spring chamber causing a 3–2 downshift to occur.

5. . luid flo in reverse gear (Figs 5.24, 5.21(a) and 5.14(a))

In reverse gear the pressure regulator valve and throttle pressure valve operate in a similar way to neutral.

With the manual selection valve in the R position, line pressure is directed to the drive and reverse clutch piston by way of the ball valves for direct and reverse clutch. Similarly, line pressure is directed through the ball valves for first and reverse gear brake to the soft engagement valve and from there to the first and reverse gear brake piston thereby engaging the clutch plates.

Whilst in reverse gear, the main pressure regulating valve provides maximum line pressure increase, this being due to there being no pressure acting on the left hand end of the valve.

5. .10 Transmission po er train operating faults

The effective operation of an automatic transmission depends greatly upon clutch, band and one way clutch holding ability, torque converter one way clutch operation and engine performance. The method used in diagnosing faults in the engagement components of the transmission is known as the

. This test entails accelerating the engine with

the throttle wide open to maximum speed while the torque converter turbine is held stationary.

Stall test procedure
1 rive car or run engine until engine and transmission has attained normal working temperatures.
2 Check the level of fluid in the transmission box and correct if necessary.
3 Apply the hand brake and chock the wheels.
4 Connect a tachometer via leads to the coil ignition terminals.
5 Apply the foot brake, select   range and fully press the accelerator pedal down for a period not exceeding 10 seconds to avoid overheating the transmission fluid (this is very important).
6 uickly observe the highest engine speed reached on the tachometer and immediately release the throttle pedal.
7 Shift selector lever to N and allow transmission fluid to cool at least two minutes or more before commencing next test.
8 Repeat tests 5 and 6 in 1 and R range.

 nterpreting stall test results   A typical stall test maximum engine speed could be 2300 rev/min   100. If the actual stall speed differs from the recommended value (i.e. 2300 rev/min), Table 5.3 should be used as a guide to trace the fault. The stall test therefore helps to determine if the fault is due to the engine, the converter or the transmission assembly.
      The reason why a slipping torque converter stator drags down the engine s maximum speed is because the spinning stator makes the converter behave as a fluid coupling (no torque multiplication), causing the fluid to have a retarding effect on the impeller.
   By performing the stall test in   , 1 and R range, observing in which range or ranges the slippage occurs and comparing which clutch or band operates in the slipping range enables the effective components to be eliminated and the defective components to be identified (see Table 5.4).

ab 5.3 Table of stall tests

Test results	ossible causes
Below 1 rev/min	Stator slip
Approximately 21 rev/min	oor engine performance
Above 2 rev/min	Transmission slip

a) Slip in R can be the drive and reverse clutch or first and reverse brake. Engage 1 range. If slip still occurs, first and reverse brake must be slipping.
b) Slip in   can be forward clutch or one way clutch. Engage 1 . If slip still occurs, forward clutch must be slipping.
c) Slip in R can be the drive and reverse clutch or first and reverse brake. Engage 1 range. If there is no slip the drive and reverse clutch could be slipping.

Road test for defective tor ue converter   A road test enables a seized or slipping stator to be engaged, whereas the stall test can only indicate a possible slipping stator. The symptoms for a faulty stator one way clutch are shown in Table 5.5.
   If the converter one way clutch has seized, the vehicle will have poor high speed performance because the stator reaction above the coupling point speed hinders the circulation of fluid if it is not able to freewheel. Conversely, if the converter one way clutch is slipping there will be no stator reaction for the fluid and therefore no torque multiplication so that the acceleration will be sluggish up to about 50 km/h.

.9 The continuously variable belt and pulley transmission
The continuously variable transmission CVT, as used by Ford and Fiat, is based simply on the

ab 5.4 Table of possible faults

Selector position	ossible fault
R	(R)C or (1 R)B
	FC or C
1	FC C or (1 R)C

ab 5.5 Table of symptoms for a faulty one way clutch

Fault	ehicle response	
	– km/h	above km/h
Slipping stator	ery sluggish No hill start possible	rives normally
Sei ed stator	rives normally	oss of power Severe overheating

principle of a belt running between two V-shaped pulleys which is designed so that the effective belt contact diameter settings can be altered to produce a stepless change in the input to output pulley shaft speed.

Van oorne Transmissie in olland has been mainly responsible for the development of the steel belt which is the key component in the transmission. At present the steel belt power output capacity is suitable for engine sizes up to 1.6 litres but there does not appear to be any reason why uprated steel belts cannot be developed.

This type of transmission does not suffer from the limitations of the inefficient torque converter which is almost universely used by automatic transmissions incorporating epicyclic gear trains operated by multiplate clutches and band brakes.

5. .1 Stepless speed ratios (Figs 5.25 and 5.26)

The transmission consists basically of a pair of variable width vee-shaped pulleys which are interconnected by a composite steel belt. Each pulley consists of two shallow half cones facing each other and mounted on a shaft, one being rigidly attached to it whereas the other half is free to slide axially on linear ball splines (Fig. 5.25). The variable speed ratios are obtained by increasing or decreasing the effective wrap contact diameter of the belt with the primary input pulley producing a corresponding reduction or enlargement of the secondary output pulley working diameter. The belt variable wrap contact diameter for both primary and secondary pulleys is obtained by the wedge shaped belt being supported between the inclined ad acent walls of the two half pulleys.

When the primary input half pulleys are brought axially closer, the wedge or vee-shaped belt running between them is squeezed and is forced to ride up the tapered walls to a larger diameter. Conversely, since the belt is endless and inextensible, the secondary output half pulleys are compelled to separate, thus permitting the belt wrap to move inwards to a smaller diameter.

Alternatively, drawing the secondary output half pulleys closer to each other enlarges the belt s running diameter at that end. Accordingly it must reduce the primary input pulley wrap diameter at the opposite end. A one to one speed ratio is obtained when both primary and secondary pulleys are working at the same belt diameter (Fig. 5.26). A speed ratio reduction (underdrive) occurs when the primary input pulley operates at a larger belt contact diameter than the secondary output pulley (Fig. 5.26). Conversely, a speed ratio increase (over

drive) is achieved if the belt contact with the primary pulley is at a smaller diameter relative to the secondary pulley wrap diameter (Fig. 5.26).

In the case of the Ford Fiesta, the pulleys provide a continuously variable range of ratios from bottom 2.6:1 to a super overdrive top of 0.445:1.

An intermediate gear reduction of about 1.4:1 between the belt output pulley shaft and the final drive crownwheel is also provided so that the transmission can be made to match the engine s power output and the car s design expectation.

5. . Belt design (Figs 5.26 and 5.27)

Power is transmitted from the input to the output pulley through a steel belt which resembles a steel necklace of thin trapezoidal plates strung together between two multistrip bands made from flexible high strength steel (Fig. 5.27). There are 300 plates, each plate being roughly 2 mm thick, 25 mm wide and 12 mm deep, so that the total length of the endless belt is approximately 600 mm. Each band is composed of 10 continuous strips 0.18 mm thick. Also made of high strength steel, they fit into location slots on either side of the plates, their purpose being to guide the plates, whereas it is the plates function to transmit the drive by pushing. Another feature of the plates is that they are embossed in a dimple form to assist in the automatic alignment of the plates as they flex around the pulley.

Contact between the belt plates and pulley is provided by the tapered edges of the plates which match the inclination of the pulley walls. When in drive, both primary and secondary sliding half pulleys are forced against the belt so that different plates are in contact and are wedged between the vee profile of the pulley at any one time.

Consequently, the grip produced between the plates and pulley walls also forces the plates together so that in effect they become a continuous strut which transmits drive in compression (unlike the conventional belt which transfers power under tension (Fig. 5.26)). The function of the non-drive side of the belt, usually referred to as the slack side, is only to return the plate elements back to the beginning of the drive (compressive) side of the pulleys.

The relative movement between the band strips and the plates by this design is very small, therefore frictional losses are low. Nevertheless the transmission efficiency is only 92% with a one to one speed ratio dropping to something like 86% at pull-away, when the speed ratio reduction is 2.6:1.
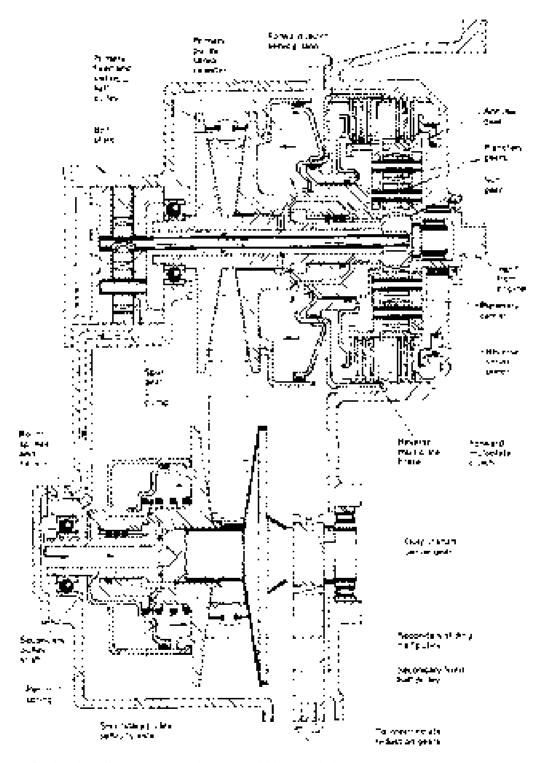
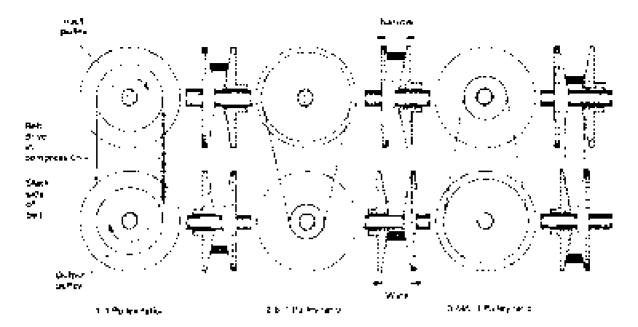**Fig. 5.25** Section view of a transverse continuously variable transmission

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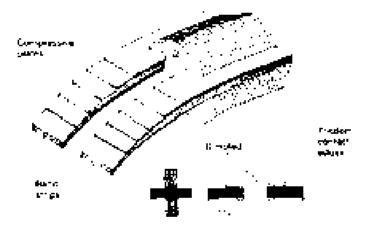**Fig. 5.26** Ilustration of pulley and belt under- and overdrive speed ratios

Fig. 5.27 Steel belt construction

5. .3 *ydraulic control system* (Fig. 5.28)

The speed ratio setting control is achieved by a spur type hydraulic pump and control unit which supplies oil pressure to both primary and secondary sliding pulley servo cylinders (Fig. 5.28). The ratio settings are controlled by the pressure exerted by the larger primary servo cylinder which accordingly moves the sliding half pulley axially inwards or outwards to reduce or increase the output speed setting respectively. This primary cylinder pressure causes the secondary sliding pulley and smaller secondary servo cylinder to move proportionally in the opposite direction against the resistance of both the return spring and the secondary cylinder pressure, this being necessary to provide the correct clamping loads between the belt and pulleys walls. The cylinder pressure necessary to prevent slippage of the belt varies from around 22 bar for the pull away lowest ratio setting to approximately 8 bar for the highest overdrive setting.

The speed ratio setting and belt clamping load control is achieved via a primary pulley position senser road assembly.

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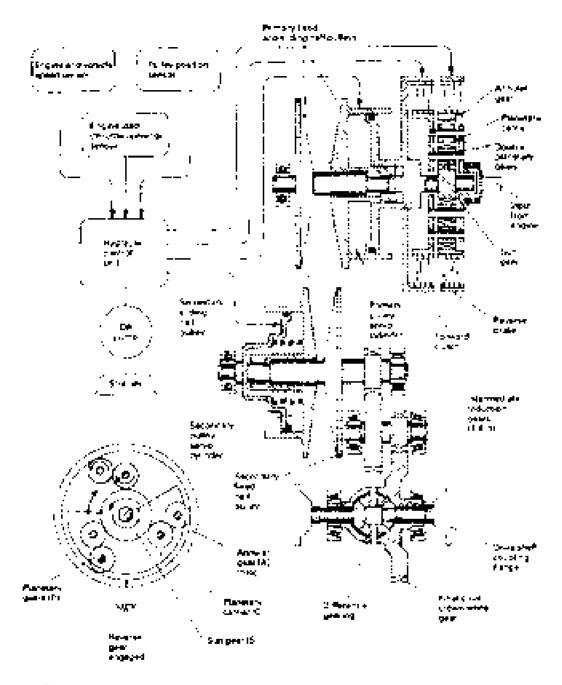
Fig. 5.28 Transaxle continuously variable belt and pulley transmission layout

owever, engine and road speed signals are provided by a pair of pitot tubes which sense the rate of fluid movement, this being a measure of speed, be it either under the influence of fluid flow caused by the engine s input or by the output drive relating to vehicle speed.

5. . *picyclic gear train construction and description* (Figs 5.25 and 5.28)

rive in both forward and reverse direction is obtained by a single epicyclic gear train controlled by a forward multiplate clutch and a reverse multiplate brake, both of which are of the wet type

(immersed in oil) (Fig. 5.25). The forward clutch is not only used for engagement of the drive but also to provide an initial power take-up when driving away from rest.

The epicyclic gear train consists of an input planetary carrier, which supports three sets of double planetary gears, and the input forward clutch plates. Surrounding the planetary gears is an internally toothed annulus gear which also supports the rotating reverse brake plates. In the centre of the planetary gears is a sun gear which is attached to the primary pulley drive shaft.

eutral or park (or P position) (Fig. 5.28) When neutral or park position is selected, both the multiple clutch and brake are disengaged. This means that the annulus gear and the planetary gears driven by the input planetary carrier are free to revolve around the sun gear without transmitting any power to the primary pulley shaft.

The only additional feature when park position is selected is that a locking pawl is made to engage a ring gear on the secondary pulley shaft, thereby preventing it from rotating and causing the car to creep forward.

orward drive (or position) (Fig. 5.28) Selecting or drive energizes the forward clutch so that torque is transmitted from the input engine drive to the right and left hand planetary carriers and planet pins, through the forward clutch clamped drive and driven multiplates. Finally it is transferred by the clutch outer casing to the primary pulley shaft. The forward gear drive is a direct drive causing the planetary gear set to revolve bodily at engine speed with no relative rotational movement of the gears themselves.

Reverse drive (R position) (Fig. 5.28) Selecting reverse gear disengages the forward clutch and energizes the reverse multiplate brake. As a result, the annular gear is held stationary and the input from the engine rotates the planetary carrier (Fig. 5.28).

The forward clockwise rotation of the carrier causes the outer planet gears to rotate on their own axes as they are compelled to roll round the inside of internally toothed annular gear in an anticlockwise direction.

Motion is then transferred from the outer planet gears to the sun gear via the inner planet gears. Because they are forced to rotate clockwise, the meshing sun gear is directionally moved in the opposite sense anticlockwise, that is in the reverse direction to the input drive from the engine.

5. .5 *erformance characteristics* (Fig. 5.29)
With drive selected and the car at a standstill with the engine idling, the forward clutch is ust sufficiently engaged to produce a small amount of transmission drag (point 1). This tends to make the car creep forwards which can be beneficial when on a slight incline (Fig. 5.29). pening the throttle slightly fully engages the clutch, causing the car to move positively forwards (point 2). epressing the accelerator pedal further sets the speed ratio according to the engine speed, road speed and the driver s requirements. The wider the throttle is opened the lower the speed ratio setting will be and the higher the engine speed and vice versa. With a light constant throttle opening at a minimum of about 1700 rev/min (point 3) the speed ratio moves up to the greatest possible ratio for a road speed of roughly 65 km/h which can be achieved on a level road. If the throttle is opened still wider (point 4) the speed ratio setting will again change up, but at a higher engine speed. Fully depressing the accelerator pedal will cause the engine speed to rise fairly rapidly (point 5) to about 4500 rev/min and will remain at this engine speed until a much higher road speed is attained. If the engine speed still continues to rise the pulley system will continue to change up until maximum road speed (point 6) has been reached somewhere near 5000 rev/min.

Partially reducing the throttle open then causes the pulley combination to move up well into the overdrive speed ratio setting, so that the engine speed decreases with only a small reduction in the car s cruising speed (point 7).

Even more throttle reduction at this road speed causes the pulley combination speed ratio setting to go into what is known as a backout upshift (point 8), where the overdrive speed ratio reaches its maximum limit. pening the throttle wide again brings about a kickdown downshift (point 9) so that there is a surplus of power for acceleration. A further feature which provides engine braking when driving fast on winding and hilly slopes is through the selection of range this changes the form of driving by preventing an upshift when the throttle is eased and in fact causes the pulley combination to move the speed ratio towards an underdrive situation (point 10), where the engine operates between 3000 and 4000 rev/min over an extensive road speed range.

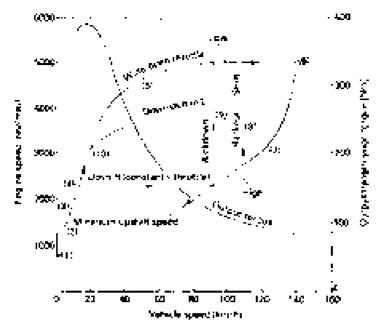The output torque developed by this continuously variable transmission approaches the ideal

Fig. 5.29 CVT speed and torque performance characteristics

constant power curve (Fig. 5.29) in which the torque produced is inversely proportional to the car's road speed.

5.1 Five speed automatic transmission with electronic-hydraulic control

5.10.1 Automatic transmission gear train system
(Fig. 5.30)
This five speed automatic transmission system is broadly based on a ZF design. Power is supplied though a hydrodynamic three element torque converter incorporating an integral disc type lock-up clutch. The power drive is then directed though a Ravigneaux type dual planetary gear train which provides five forward gears and one reverse gear; it then passes to the output side via a second stage single planetary gear train. The Ravigneaux planetary gear train has both large and small input sun gears, the large sun gears mesh with three long planet gears whereas the small sun gears mesh with three short planet gears; both the long and the short planet gears are supported on a single gear carrier. A single ring-gear meshing with the short planet gear forms the output side of the planetary gear train. Individual gear ratios are selected by applying the input torque to either the pinion carrier or one of the sun gears and holding various other members stationary.

5.10. Gear train power flow for individual gear ratios

drive range — first gear (Fig. 5.31) With the position selector lever in drive range, the one way clutch (OWC) holds the front planet carrier while multiplate clutch (B) and the multiplate brake ( ) are applied. Power flows from the engine to the torque converter pump wheel, via the fluid media to the output turbine wheel. It is then directed by way of the input shaft and the applied multiplate clutch (B) to the front planetary large sun gear (S). With the front planet carrier (C_F) held stationary by the locked one way clutch (OWC), power passes from the large sun gear (S) to the long planet gears (P) in an anticlockwise direction. The long planet gear (P) therefore drives the short planet gears (P_S) in a clockwise direction thus compelling the front annular ring gear (A_F) to move in a clockwise direction. Power thus flows from the front annular ring gear (A_F) though the rear intermediate shaft to the rear planetary gear annular ring gear (A_R) in a clockwise direction. With the rear sun gear (S_R) held stationary by the applied multiplate brake ( ), the rear planet gears (P_R) are forced to roll around the fixed sun gear in a clockwise direction, this in turn compels the rear planet carrier (C_R) and the output shaft also to rotate in a clockwise direction at a much reduced speed. Thus a two stage speed reduction produces an overall underdrive
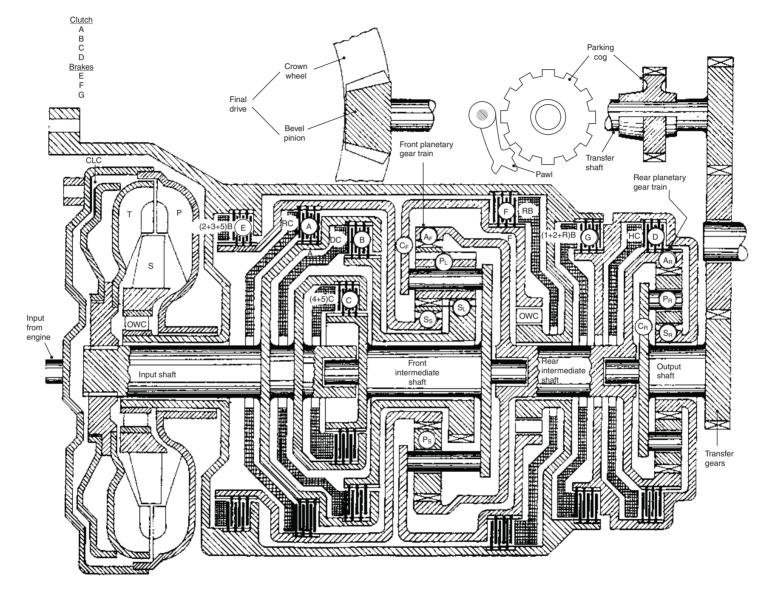
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Fig. 5.30 Five speed and reverse automatic transmission (transaxle/longitudinal) layout
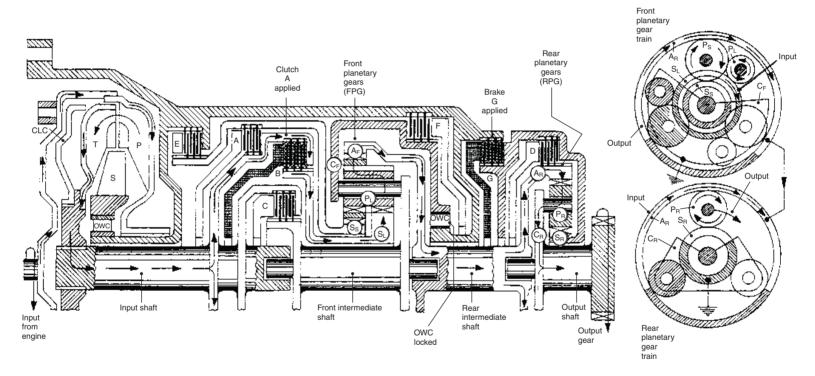
Fig. 5.31 Five speed and reverse automatic transmission power flow first gear

first gear. If the 2 first gear is selected multiplate brake (F) is applied in addition to the multiplate clutch (B) and multiplate brake ( ). As a result instead of the one way clutch (WC) allowing the vehicle to freewheel on overrun, the multiplate brake (F) locks the front planetary carrier (C_F) to the casing. Consequently a positive drive exists between the engine and transmission on both drive and overrun: it thus enables engine braking to be applied to the transmission when the transmission is overrunning the engine.

drive range   second gear (Fig. 5.32)   With the position selector lever in   drive range, multiplate clutch (C) and multiplate brakes (B) and ( ) are applied.

Power flows from the engine via the torque converter to the input shaft, it then passes via the multiplate clutch (B) to the first planetary large sun gear (S). With the multiplate brake (E) applied, the front planetary small sun gear (S_S) is held stationary. Consequently the large sun gear (S) drives the long plant gears (P) anticlockwise and the short planet gears (P_S) clockwise, and at the same time, the short planet gears (P_S) are compelled to roll in a clockwise direction around the stationary small sun gear (S_S).

The drive then passes from the front planetary annular ring gear (A_F) to the rear planetary annular ring gear (A_R) via the rear intermediate shaft. With the rear sun gear (S_R) held stationary by the applied multiplate brake ( ) the clockwise rotation of the rear annular ring gear (A_R) compels the rear planet gears (P_R) to roll around the held rear sun gear (S_R) in a clockwise direction taking with it the rear carrier (C_R) and the output shaft at a reduced speed. Thus the overall gear reduction takes place in both front and rear planetary gear trains.

drive range   third gear (Fig. 5.33)   With the position selector lever in   drive range, multiplate clutches (B) and ( ), and multiplate brake (E) are applied.

Power flows from the engine via the torque converter to the input shaft, it then passes via the multiplate clutch (B) to the front planetary large sun gear (S). With the multiplate brake (E) applied, the front planetary small sun gear (S_S) is held stationary. This results in the large sun gear (S) driving the long planet gears (P) anticlockwise and the short planet gears (P_S) clockwise, and simultaneously, the short planet gears (P_S) are compelled to roll in a clockwise direction around

the stationary small sun gear (S_S). Consequently, the annular ring gear (A_F) is also forced to rotate in a clockwise direction but at a reduced speed to that of the input large sun gear (S). The drive is then transferred from the front planetary annular ring gear (A_F) to the rear planetary annular ring gear (A_R) via the rear intermediate shaft. With the multiplate clutch ( ) applied the rear planetary sun gear (S_R) and rear annular ring gear (A_R) are locked together, thus preventing the rear planet gears from rotating independently on their axes. The drive therefore passes directly from the rear annular ring gear (A_R) to the rear carrier (C_R) and output shaft via the ammed rear planet gears. Thus it can be seen that the overall gear reduction is obtained in the front planetary gear train, whereas the rear planetary gear train only provides a one-to-one through drive.

drive range   fourth gear (Fig. 5.34)   With the positive selector lever in   drive range, multiplate clutches (B), (C) and ( ) are applied. Power flows from the engine via the torque converter to the input shaft, it then passes via the multiplate clutch (B) to the front planetary large sun gear (S) and via the multiplate clutch (C) to the front planetary planet-gear carrier (C_F). Consequently both the large sun gear and the planet carrier rotate at the same speed thereby preventing any relative planetary gear motion, that is, the gears are ammed.

 ence the output drive speed via the annular ring gear (A_F) and the rear intermediate shaft is the same as that of the input shaft speed. Power is then transferred to the rear planetary gear train by way of the front annular ring gear (A_F) and rear intermediate shaft to the rear planetary annular ring gear (A_R) and rear intermediate shaft to the rear planetary annular ring gear (A_R). owever, with the multiplate clutch ( ) applied, the rear annular ring gear (A_R) becomes locked to the rear sun gear (S_R) the drive therefore flows directly from the rear annular ring gear to the rear planet carrier (C_R) and output shaft via the ammed planet gears. Thus there is no gear reduction in both front and rear planetary gear trains, hence the input and output rotary speeds are similar.

drive range   fifth gear (Fig. 5.35)   With the position selector lever in   drive range, multiplate clutches (C) and ( ) and multiplate brake (E) are applied. Power flows from the engine via the torque converter to the input shaft, it then passes via the multiplate clutch (C) to the front planetary planet
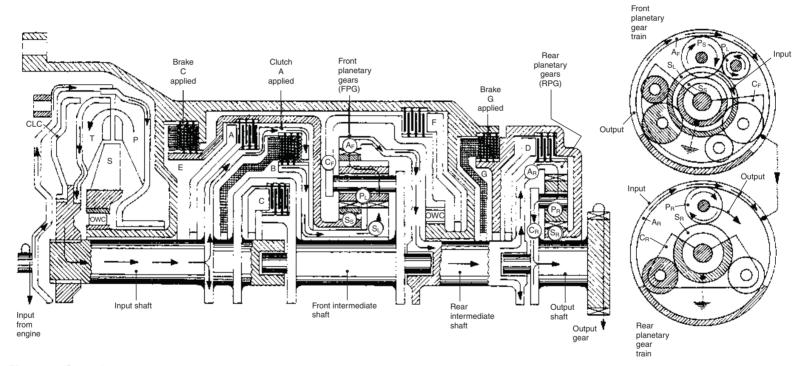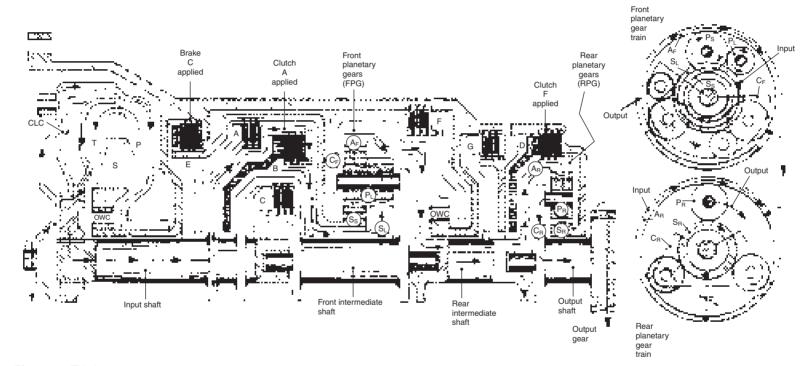
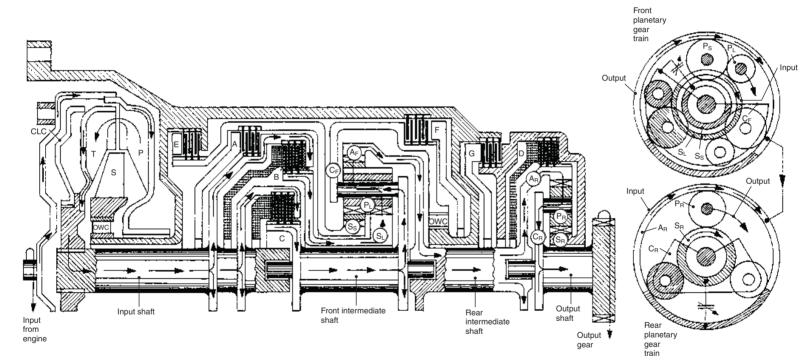**Fig. 5.32** Second gear

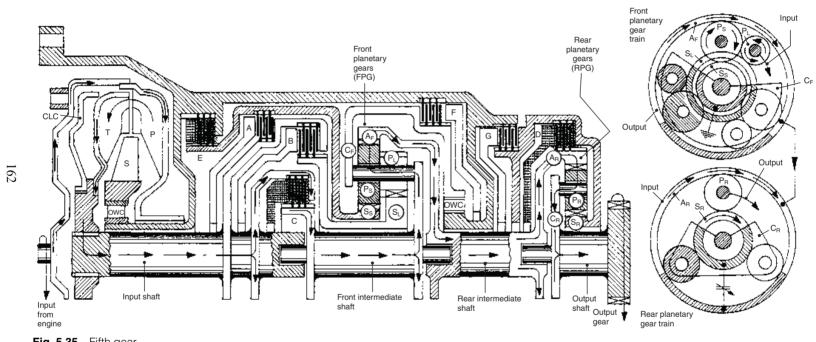**Fig. 5.33** Third gear

Fig. 5.34 Fourth gear

Fig. 5.35 Fifth gear

carrier (C_F). With the multiplate brake (E) applied the front planetary short sun gear (S_S) remains stationary. As a result the planet gear carrier (C_F) and both long and short planet gear pins are driven around in a clockwise direction it thus compels the short planet gears (P_S) to roll clockwise around the fixed small sun gear (S_S). It also causes the annulus ring gear (A_F) to revolve around its axis however, this will be at a speed greater than the input planet carrier (C_F). Note that the long planet gears (P) and large sun gear (S) revolve but are both inactive. The drive then passes from the front planetary annular ring gear (A_F) to the rear planetary annular ring gear (A_R) via the rear intermediate shaft. With the multiplate clutch ( ) applied both rear annular ring gear (A_R) and rear sun gear (S_R) are locked together.   ence the rear planet gears sandwiched between both the sun and the annular gears also am the drive therefore is passed directly though the ammed rear planetary gear train cluster to the output shaft without a change in speed. An overall speed step-up is thus obtained, that is, an overdrive fifth gear is achieved, the step-up taking place only in the first stage planetary gear train, the second planetary gear train providing only a through one-to-one drive.

R reverse gear (Fig. 5.36)   With the position selector in reverse R position, the multiplate brakes (F) and ( ) and the multiplate clutch (A) are applied. Power flows from the engine to the torque converter to the input shaft, it then passes via the multiplate clutch (A) to the front planetary small sun gear (S_S). With the multiplate clutch (F) applied, the front planet gear carrier (C_F) is held stationary, and the drive passes from the clockwise rotating small sun gear (S_S) to the short planet gears (P_S) making the latter rotate anticlockwise. As a result the internal toothed front annular ring gear (A_F) will also be compelled to rotate anticlockwise. The drive then passes from the front planetary annular ring gear (A_F) to the rear planetary annular ring gear (A_R) via the rear intermediate shaft. With the rear sun gear (S_R) held stationary by the applied multiplate brake ( ) the anticlockwise rotation of the rear annular ring gear (A_R) compels the rear planet gear (P_R) to roll around the held rear sun gear (S_R) in an anticlockwise direction taking with it the rear carrier (C_R) and the output shaft at a reduced speed.

Thus the direction of drive is reversed in the first planetary gear train, and there is an under-drive gear reduction in both planetary gear trains.

5.10.3 ear change hydraulic control (Fig. 5.30)
The shifting from one gear ratio to another is achieved by a sprag type one way clutch (when shifting from first to second gear and vice versa), four rotating multiplate clutches A, B, C and and three held multiplate brakes E, F and   . The multiplate clutches and brakes are engaged by electro-hydraulic control, hydraulic pressure being supplied by the engine driven fluid pump. To apply a clutch or brake, pressurized fluid from the hydraulic control unit is directed to an annular shaped piston chamber causing the piston to clamp together the drive and driven friction disc members of the multiplate clutch. Power therefore is able to be transferred from the input to the output clutch members while these members rotate at different speeds. Shifting from one ratio to another takes place by applying and releasing various multiplate clutches/brakes.   uring an up or down gear shift such as 2–3, 3–4, 4–5 or 5–4, 4–3, 3–2 one clutch engages while another clutch disengages. To achieve an uninterrupted power flow, the disengaging clutch remains partially engaged but at a much reduced clamping pressure, whereas the engaging clutch clamping pressure rise takes place in a phased pattern.

5.10. pshift clutch overlap control characteristics (Fig. 5.37 (a–c))
The characteristics of a gear ratio upshift is shown in Fig. 5.37(a), it can be seen with the vehicle accelerating, and without a gear change the engine speed steadily rises however, during a gear ratio upshift transition phase, there is a small rise in engine speed above that of the speed curve when there is no gear ratio change taking place. This slight speed upsurge is caused by a small amount of slip overlap between applying and releasing the clutches. Immediately after the load transference phase there is a speed decrease and then a steady speed rise, this being caused by the full transmitted driving load now pulling down the engine speed, followed by an engine power recovery which again allows the engine speed to rise.

When a gear upshift is about to commence the engaging clutch pressure Fig. 5.37(b) rises sharply from residual to main system pressure for a short period of time, it then drops rapidly to ust under half the main system pressure and remains at this value up to the load transfer phase.   ver the load transfer phase the engaging clutch pressure rises fairly quickly however, after this phase the pressure rise is at a much lower rate. Finally a small pressure
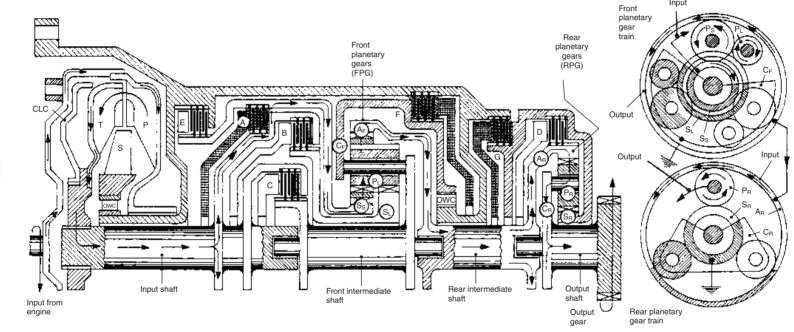
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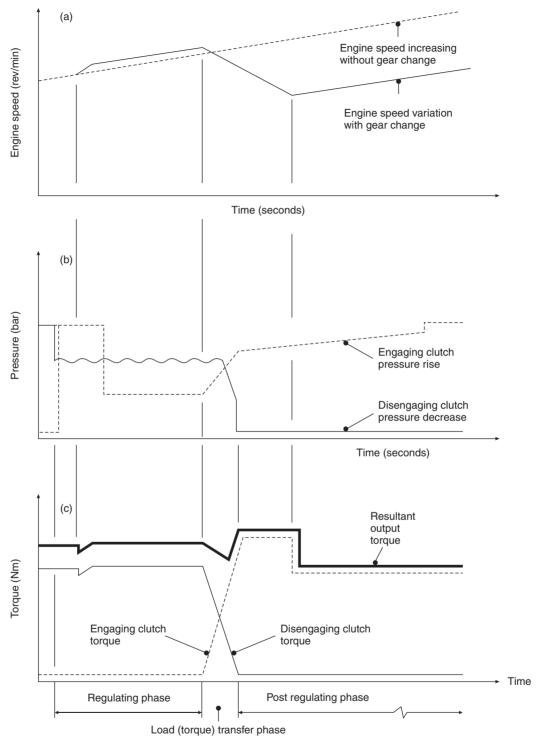**Fig. 5.36** Reverse gear

Fig. 5.37 (a–c) Upshift clutch overlap control characteristics

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ump brings it back to the main system pressure. Between the rise and fall of the engaging clutch pressure, the disengaging clutch pressure falls to something like two thirds of the main systems pressure, it then remains constant for a period of time. Near the end of the load transfer phase the pressure collapses to a very low residual pressure where it remains during the time the clutch is disengaged. Fig. 5.37(b) therefore shows a pressure overlap between the disengaging clutch pressure decrease and the engaging clutch pressure increase over the load transfer period. The consequence of too much pressure overlap would be to cause heavy binding of the clutch and brake multiclutch plate members and high internal stresses in the transmission power line, whereas insufficient pressure overlap causes the engine speed to rise when driving though the load transfer period. Fig. 5.37(c) shows how the torque load transmitted by the engaging and disengaging clutches changes during a gear ratio upshift. It shows a very small torque dip and recovery for the disengaging clutch after the initial disengaging clutch pressure drop, then during the load transfer phase the disengaging clutch output torque declines steeply while the engaging clutch output torque increases rapidly. The resultant transmitted output torque over the load transfer phase also shows a dip but recovers and rises very slightly above the previous maximum torque, this being due to the transmission now being able to deliver the full engine torque.

Finally the transmitted engine torque drops a small amount at the point where the engine speed has declined to its minimum, it then remains constant as the engine speed again commences to rise.

5.10.5 Description of ma or hydraulic and electronic components

ydraulic control unit (Fig. 5.38) The hydraulic control unit is housed in the oil-pan position underneath the transmission gears. A fluid pump operates the hydraulic circuitry it is driven directly from the engine via the torque converter casing, fluid is directed by way of a pressure regulation valve to the interior of the torque converter and to the various clutches and brakes via passages and valves. The hydraulic control circuit which operates the gear shifts are activated by three electromagnetic operated open/close valves (solenoid valves) and four electro-magnetic progressive opening and closing regulation valves (electronic pressure regulation valve EPRV). Both types of

valves are energized by the electronic transmission control unit ETCU which in turn receives input signals from various speed, load, temperature and accelerator pedal sensors all of these sensors are simultaneously and continuously monitoring the changing parameters. In addition a position selector lever or button operated by the driver relays the different driving programs to the electronic transmission control unit ETCU.

Electronic transmission control unit The function of the electronic transmission control unit ETCU is to collect, analyse and process all the input signals and to store program data so that the appropriate hydraulic circuit pressures will bring about transmission gear changes to match the engine speed and torque, vehicle s weight and load, driver s requirements and road conditions.

ontrol program The stored program provides data to give favourable shift characteristics for gears and the torque converter lock-up clutch, it co-ordinates parameters for pressure calculations, engine manipulation and synchronizing gear change phases, it provides regulation parameters for smooth gear shifts and the converter lock-up and finally it has built-in parameters for fault diagnoses.

Transmission input signal sensors The various signals which activate the electronic transmission control unit ETCU can be divided into three groups: (1) transmission, (2) engine and (3) vehicle:

1 Transmission
  (a) input turbine speed sensor
  (b) output drive speed sensor
  (c) transmission temperature sensor
  (d) position switch signalling-selector lever position to the electronic transmission control unit

2 Engine
  (a) engine speed sensor
  (b) engine load-in ector opening duration
  (c) throttle valve opening-potentiometer
  (d) engine temperature sensor

3 Vehicle
  (a) kickdown switch
  (b) position PRN 432 indicator
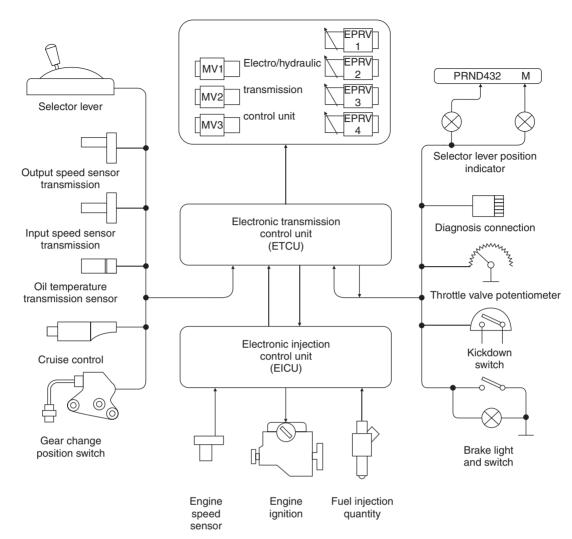 (c) manual gear selection program
  (d) brake light switch

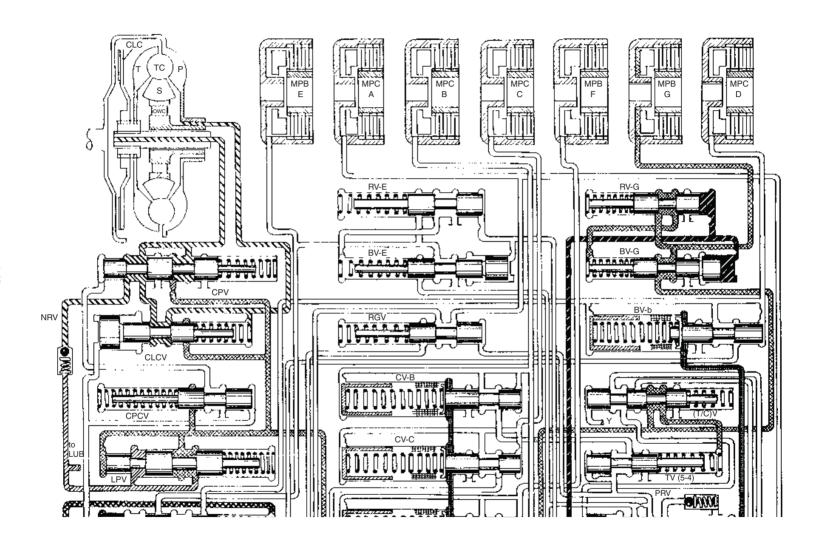**Fig. 5.38** Basic electronic control system layout

5.10.6 Description and function of the electro/hydraulic valves

Solenoid (electro-magnetic) valves (MV1, MV2 and MV3) (Fig. 5.39) The solenoid valves MV1, MV2 and MV3 are electro-magnetic disc armature operated ball-type valves which are energized by current supplied by the electronic transmission control unit. The ball-valve is either in the open or closed position, when the valve is de-energized the ball-valve closes the inlet port and vents the outlet port whereas when energized the ball-valve blocks the vent port and opens the inlet port. Solenoid valve MV1 when energized activates shift valve SV-1 and SV-3. Solenoid valve MV2 activates shift valve SV-2 and shuts down the switch over function of shift

valve SV-3. Solenoid valve MV3 activates the traction/coasting valve (T/C)V.

Electronic pressure regulating valves (EPRV-1, EPRV-2, EPRV-3 and EPRV-4) (Fig. 5.39) The electronic pressure regulating valves EPRV-(1–4) are variable pressure electro-magnetic cylindrical armature operated needle-type valves, the output pressure delivered being determined by the magnitude of the current supplied at any one time by the electronic transmission control unit to the electronic pressure regulating valves. With increasing current the taper-needle valve orifice is enlarged, this increases the fluid spill and correspondingly reduces the actuating control pressure delivered to the various valves responsible for gear shifts.
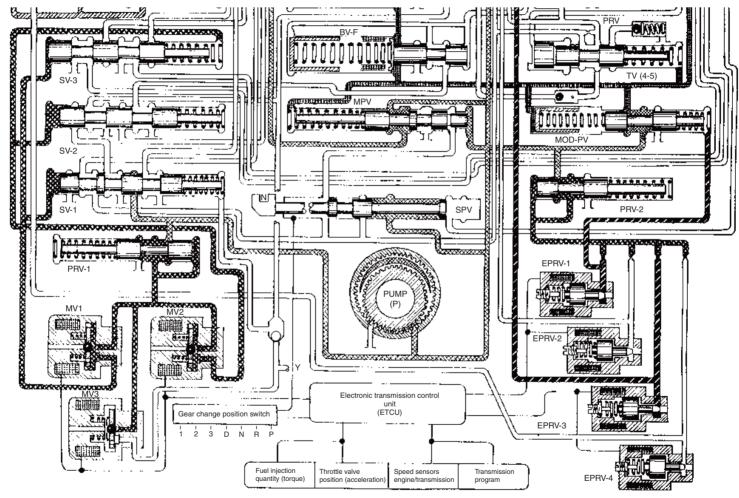
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SV-3

SV-2

SV-1

PRV-1

MV1

MV2

MV3

BV-F

MPV

SPV

N

PUMP
(P)

Y

PRV

TV (4-5)

MOD-PV

PRV-2

EPRV-1

EPRV-2

EPRV-3

EPRV-4

Gear change position switch

1 2 3 D N R P

Electronic transmission control
unit
(ETCU)

| Fuel injection quantity (torque) | Throttle valve position (acceleration) | Speed sensors engine/transmission | Transmission program |

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Fig. 5.39   Hydraulic/electronic transmission control system – Neutral position

ydraulic/electronic automatic transmission control system abbreviation key for Figs 5.39–5.46.

A list of key components and abbreviations used in the description of the electro/hydraulic control system is as follows:

1	fluid pressure pump	P
2	selector position valve	SPV
3	main pressure valve	MPV
4	pressure reducing valves	PRV-1 and PRV-2
5	modulation pressure valve	M -PV
6	shift valve	SV-1, SV-2 and SV-3
7	reverse gear valve	R V
8	clutch valves	CV-A, CV-B, CV-C and CV-
9	brake valves	BV-E, BV-F and BV-
10	retaining valves	RV-E and RV-
11	traction/coasting valve	(T/C)V
12	traction valves	TV (4–5) and TV (5–4)
13	converter pressure valve	CPV
14	converter pressure control valve	CPCV
15	converter lock-up clutch valve	C CV
16	lubrication pressure valve	PV
17	solenoid (electro-magnetic) valves	MV1, MV2 and MV3
18	electronic pressure regulating valves	EPRV-1, EPRV-2, EPRV-3 and EPRV-4
19	multiplate clutch/brake	MPC-A, MPC-B, MPC-C and MPC- /MPB-E, MPB-F and MPB-
20	pressure relief valve	PRV
21	non-return valve	NRV

ab 5.6 Hydraulic/electronic automatic transmission control system solenoid valve clutch and brake engagement se uence for different gear ratios for Figs   –

Solenoid valve – clutch – brake engagement se uence

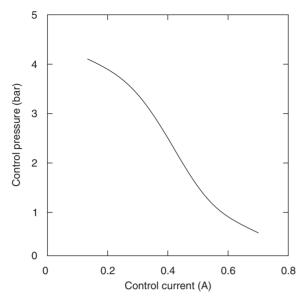| ear range	Solenoid valve logic						Clutch and brake logic					
	Solenoid valves		Pressure regulating valves				Clutch			Brake		
	1	2	1	2			A	B	C	E	F	C
N/ neutral/park	A	A	A		A							A
1st gear	A	A	A		A		A				A	A
2nd gear	A	A	A	A	A		A			A	A	
rd gear		A	A	A			A		A	A		
th gear			A			A	A	A	A			
th gear	A		A	A		A		A	A	A		
2 1st gear	A	A	A		A		A			A	A	(A)
R = reverse	A		A		A		A				A	A

Fig. 5.40 Electronic pressure regulating valve current-pressure characteristics

Thus control pressure delivered is inversely proportional to the amount of current supplied, that is, as the current rises the pressure decreases and vice versa. The characteristics of control pressure versus control current is shown in Fig. 5.40.

5.10.7 Description and function of pump and hydraulic valves

Pump (P) (Fig. 5.39) This internal gear crescent-type pump consists of an internal toothed-spur ring gear which runs outside but in mesh with a driving external toothed-spur gear, so that its axis of rotation is eccentric to that of the driving gear. ue to their eccentricity, there is a space between the external and internal gears which is occupied by a fixed spacer block known as the crescent whose function is to separate the inlet-output port areas. The rotation of the gears creates a low pressure area at the inlet suction end of the crescent which draws in fluid. As the gear wheels rotate, oil will be trapped between the teeth of the inner driver gear and the inside crescent side walls, and between teeth of the outer gear and the outside crescent side wall. These teeth will then carry this fluid around to the other end of the crescent where it will then be discharged at pressure by both set of teeth into the output port.

Selector position valve (SPV) (Fig. 5.39) This valve is indirectly operated by the driver to select the forward and reverse direction of drive and the neutral or park positions.

Main pressure valve (MPV) (Fig. 5.39) The main pressure valve MPV regulates the fluid pressure supply produced by the internal gear crescent pump it is a variable pressure limiting valve which relates to driving conditions and the driver s demands.

Pressure reduction valve (PRV-1) (Fig. 5.39) The pressure reduction valve PRV-1 reduces the main fluid pressure supply to an approximate constant 5 bar output pressure which is the necessary fluid pressure supply to operate the solenoid valves MV1, MV2 and MV3.

Pressure reduction valve (PRV-2) (Fig. 5.39) The pressure reduction valve PRV reduces the main fluid pressure supply to an approximate constant 5 bar output pressure which is the necessary fluid pressure supply to operate the electronic pressure regulation valves EPRV-1, EPRV-2, EPRV-3 and EPRV-4.

Modulation pressure value (M -PV) (Fig. 5.39) The modulation pressure valve is actuated by the electronic pressure regulator valve EPRV-1, it produces an output pressure which rises proportional to engine torque. The modulation pressure is conveyed to the main pressure valve and to each of the clutch valves, its purpose being to raise the system s pressure and to maximize the opening of the clutch valves with increased engine load so that a higher supply pressure reaches the appropriate multiplate clutch or/and brake.

Shift valves (SV-1, SV-2 and SV-3) (Fig. 5.39) The shift valves are actuated by the various solenoid valves MV1, MV2 and MV3: the function of a shift valve is to convey system pressure to the relevant operating circuit controlling the application or release of the various multiplate clutches/brakes.

Reverse gear valve (R V) (Fig. 5.39) The reverse gear valve functions as a shift valve for selecting reverse gear it also acts as safety valve for the forward gears by interrupting system pressure reaching clutch A , thus preventing the reverse gear being accidentally engaged whenever the vehicle is moving in a forward direction.

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lutch brakes (V- , V- , V- , V- V-E, V- and V-) (Fig. 5.39) The clutch valves control the engagement and disengagement of the multiplate clutches and brakes. These valves are variable pressure reduction valves which are actuated by the appropriate solenoid valves, electronic pressure regulator valves, traction valves and shift valves and are responsible for producing the desired clutch pressure variations during each gear shift phase. Clutch valves CV-B, CV-C and CV-F are influenced by modulation pressure which resists the partial closure of the clutch valves, hence it permits relatively high fluid pressure to reach these multiplate clutches and brake when large transmission torque is being transmitted.

Retaining valves (RV-E and RV-) (Fig. 5.39) In addition to the electronic pressure regulator valve which actuates the clutch valves, the retaining valves RV-E and RV- modify the opening and closing phases of the clutch valves in such a way as to cause a progressive build-up or a rapid collapse of operating multiplate clutch/brake fluid pressure during engagement or disengagement respectively.

Traction coasting valve (T -V) (Fig. 5.39) The traction coasting valve T/C-V cuts out the regulating action of the traction valve TV (5–4) and shifts the traction valve TV (4–5) into the shut-off position when required.

Traction valve (TV) (4) (Fig. 5.39) The traction valve TV (4–5) controls the main system fluid pressure to the multiplate-clutch MPC-B via the traction valve TV (5–4) and clutch valve CV-B and hence blocks the fluid pressure reaching the multiplate clutch CV-B when there is a upshift from fourth to fifth gear.

Traction valve (TV) (4) (Fig. 5.39) The traction valve TV (5–4) is another form of clutch valve, its function being to supply system pressure to the multiplate clutch MPC-B via clutch valve CV-B when there is a downshift from fifth to fourth gear.

onverter pressure valve (PV) (Fig. 5.39) The converter pressure valve CPV supplies the torque converter with a reduced system pressure to match the driving demands, that is, driving torque under varying driving conditions, it also serves as a pressure limiting valve to prevent excessive pressure build-up in the torque converter if the system pressure should become unduly high. The valve in addition vents the chamber formed on the drive-

plate side of the lock-up clutch when the torque converter pressure control valve is actuated.

onverter pressure control valve (P V) (Fig. 5.39) The converter pressure control valve CPCV is actuated by the electronic pressure regulation valve EPRV-4 , its purpose being to prevent the converter pressure valve CPV from supplying reduced system pressure to the chamber formed between the drive-plate and lock-up clutch and to vent this space. As a result the fluid pressure on the torque converter side of the lock-up clutch is able to clamp the latter to the drive-plate.

onverter lock-up clutch valve (V) (Fig. 5.39) The converter lock-up clutch valve C CV is actuated with the converter pressure control valve CPCV by the electronic pressure regulation valve EPRV-4 . The converter lock-up clutch valve C CV when actuated changes the direction of input flow at reduced system pressure from the drive-plate to the turbine wheel side of the lock-up clutch. Simultaneously the converter pressure valve CPV is actuated, this shifts the valve so that the space between the drive-plate and lock-up clutch face is vented. As a result the lock-up clutch is forced hard against the drive-plate thus locking out the torque converter function and replacing it with direct mechanical drive via the lock-up clutch.

ubrication pressure valve (PV) (Fig. 5.39) The lubrication pressure valve PV supplies fluid lubricant at a suitable reduced system pressure to the internal rubbing parts of the transmission gear train.

5.10. perating description of the electro/hydraulic control unit

To simplify the various solenoid valve, clutch and brake engagement sequences for each gear ratio Table 5.6 has been included.

eutral and park position (Fig. 5.39) With the selector lever in neutral or park position, fluid is delivered from the oil-pump to the selector position valve SPV , modulation pressure valve M -V , pressure reduction valves PRV-1 and PRV-2 , shift valve SV-1 , traction/coasting valve (T/C)V and clutch valve CV- . Regulating fluid pressure is supplied to the torque converter TC via the converter pressure valve CPV and to the lubrication system by way of the lubrication pressure valve PV . At the same time regulated constant fluid

pressure (5 bar) is supplied to the solenoid valves MV1, MV2 and MV3 via the pressure reduction valve PRV-1, and the electronic pressure regulating valves EPRV-(1–4) via the pressure reduction valve PRV-2. In addition controlling modulation pressure is relayed to the spring chamber of clutch valves CV-B, CV-C and CV- and brake valve CV-F via the modulation pressure valve M -PV. Neutral and parking position has the following multiplate clutch solenoid valves and electronic pressure regulator valves activated:

1 multiplate brake MPB- .
2 solenoid valves MV1 and MV3.
3 electronic pressure regulating valves EPRV-1 and EPRV-2.

 irst gear (Fig. 5.41) Engagement of first gear is obtained by applying the one way clutch WC and multiplate clutch and multiplate brake MPC-B and MPB- respectively. This is achieved in the following manner:

1   . Fluid pressure from the selector position valve SPV then passes via the traction valves TV (4–5) and TV (5–4) respectively to clutch valve CV-B, it therefore permits fluid pressure to apply the multiplate clutch MPC-B.

2   . Solenoid valve MV1 applies a reduced constant fluid pressure to the left-hand side of shift valves SV-1 and SV-3. Shift valve SV-1 shifts over to the right-hand side against the tension of the return spring blocking the fluid pressure passage leading to clutch valve CV- , however shift valve SV-3 cannot move over since a similar reduced constant pressure is introduced to the spring end of the valve via solenoid valve MV2. Solenoid valve MV2 applies reduced constant pressure to the left-hand side of shift valve SV-2 and the right-hand side of shift valve SV-3 this pushes the shift valve SV-2 to the right and so prevents shift valve SV-3 also being pushed to the right by fluid pressure from solenoid valve MV1 as previously mentioned.

3 Electronic pressure regulator valve EPRV-1 supplies a variable regulated fluid pressure to the modulation pressure valve M -PV, this pressure being continuously ad usted by the electronic transmission control unit ETCU to suit the operating conditions. Electronic pressure regulating valve EPRV-3 supplies a variable controlling fluid pressure to brake and retaining valves BV- and RV- respectively, enabling fluid pressure to apply the multiplate brake MPB- .
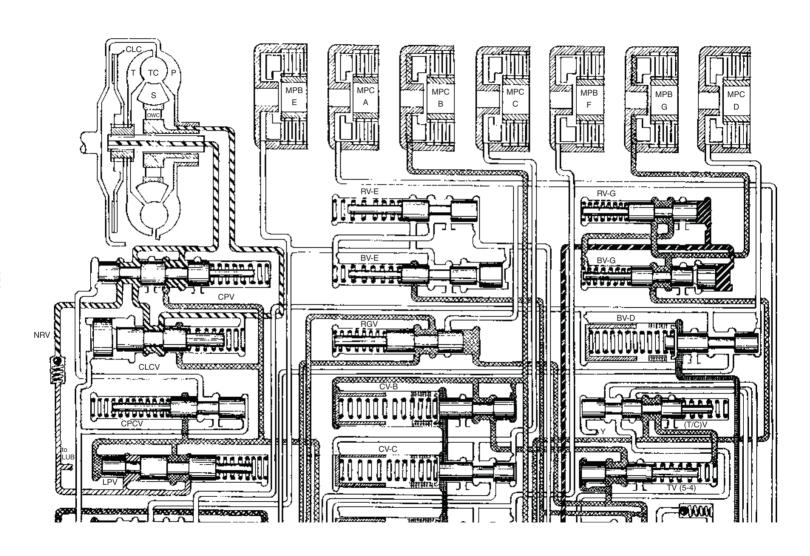
Second gear (Fig. 5.42) Engagement of second gear is obtained by applying multiplate clutch MPC-B and the multiplate brakes MPB-E and MPB- . This is achieved in the following manner with the selector position valve in the   drive range:

1 Multiplate clutch and brake MPC-B and MPB- respectively applied as for first gear.
2 Solenoid valves MV1 and MV2 are energized, thus opening both valves. Fluid pressure from MV1 is applied to the left-hand side of both SV-1 and SV-3 however, only valve SV-1 shifts over to the right-hand side. At the same time fluid pressure from solenoid valve MV2 shifts valve SV-2 over against the return-spring tension and also pressurizes the spring end of shift valve SV-3. This prevents shift valve SV-3 moving over to the right-hand side when fluid pressure from solenoid valve MV-1 is simultaneously applied at the opposite end.
3 The electronic pressure regulating valves EPRV-1 and EPRV-3 have their controlling current reduced, thereby causing an increase in line pressure to the modulation valve M -PV and to both brake and retaining valves BV- and RV-B respectively. Consequently line pressure continues to apply the multiplate brake MPB- .
4 The electronic pressure regulating valve EPRV-2 has its controlling current reduced, thus progressively closing the valve, consequently there will be an increase in fluid pressure acting on the right-hand side of both brake and retaining valves BV-E and RV-E respectively. As a result the brake valve BV-E opens to permit line pressure to actuate and apply the multiplate brake MPB-E.

Third gear (Fig. 5.43) Engagement of third gear is obtained by applying the multiplate clutches MPC-B and MPC- and the multiplate brake MPB-E.

The shift from second to third gear is achieved in the following manner with the selector position valve in the   drive range:

1 Multiplate clutch MPC-B and multiplate brake MPB-E are applied as for second gear.
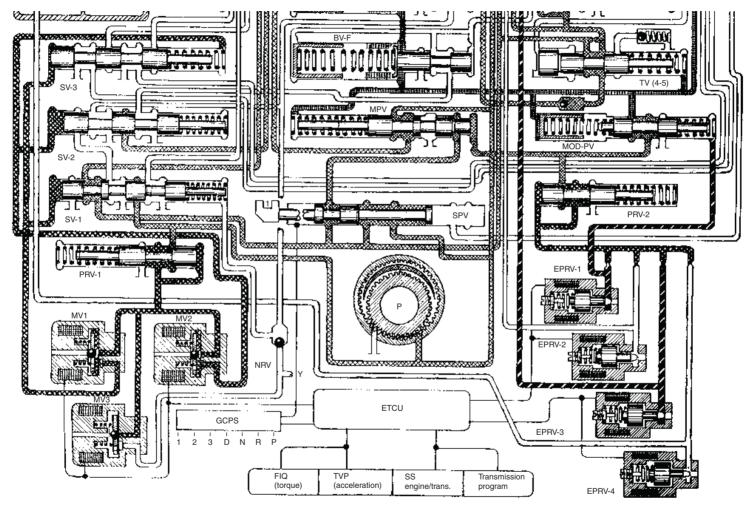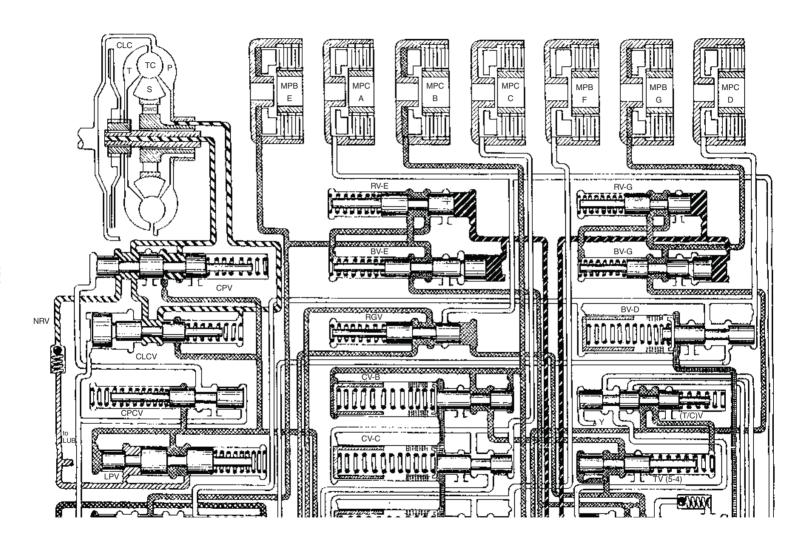2 Solenoid valve MV2 remains energized thus keeping the valve open as for first and second gear.

CLC

T | TC | P
S
OWC

MPB
E

MPC
A

MPC
B

MPC
C

MPB
F

MPB
G

MPC
D

RV-E

RV-G

BV-E

BV-G

CPV

BV-D

NRV

RGV

CLCV

CV-B

CPCV

Y

(T/C)V

to
LUB

CV-C

LPV

TV (5-4)

Fig. 5.41 Hydraulic/electronic transmission control system – first gear

CLC

T TC P
S
OWC

MPB
E

MPC
A

MPC
B

MPC
C

MPB
F

MPB
G

MPC
D

RV-E

RV-G

BV-E

BV-G

CPV

BV-D

NRV

RGV

CLCV

CV-B

CPCV

CV-C

to
LUB

(T/C)V

Y

LPV

TV (5-4)
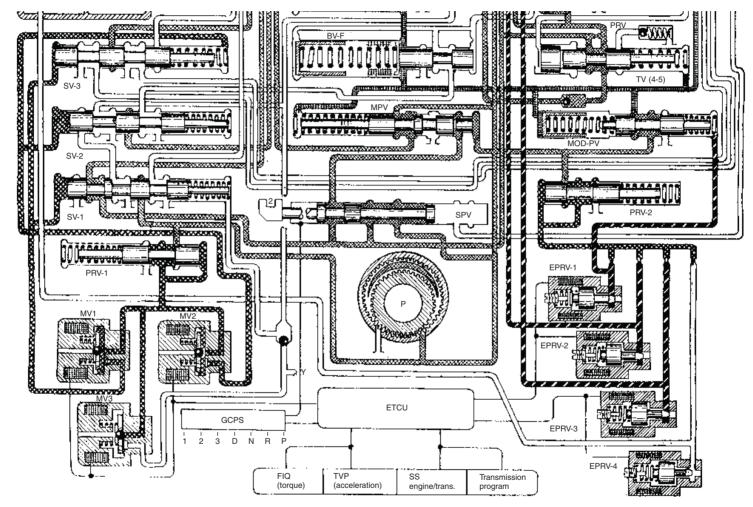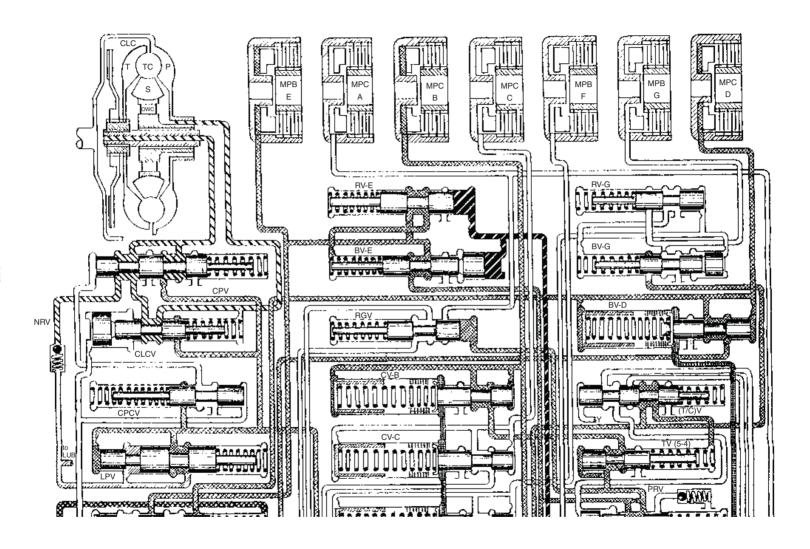
Fig. 5.42 Hydraulic/electronic transmission control system – second gear

CLC

T TC P

S

OWC

MPB
E

MPC
A

MPC
B

MPC
C

MPB
F

MPB
G

MPC
D

RV-E

RV-G

BV-E

BV-G

CPV

NRV

RGV

BV-D

CLCV

CPCV

CV-B

CV-C

(T/C)V

Y

to
LUB

LPV

TV (5-4)

PRV

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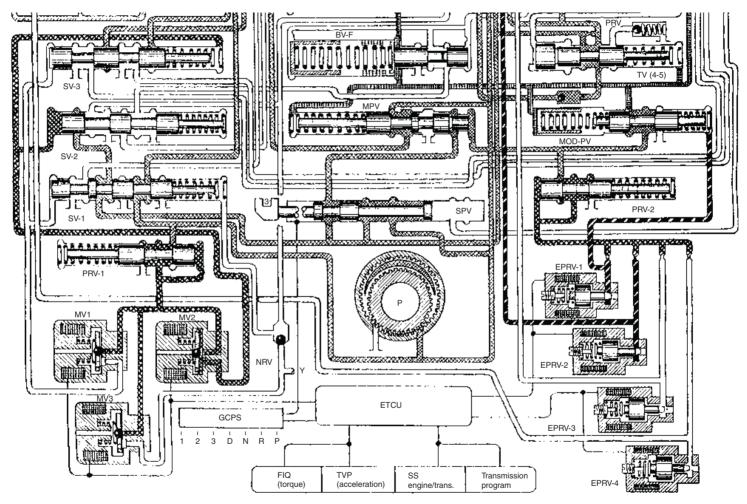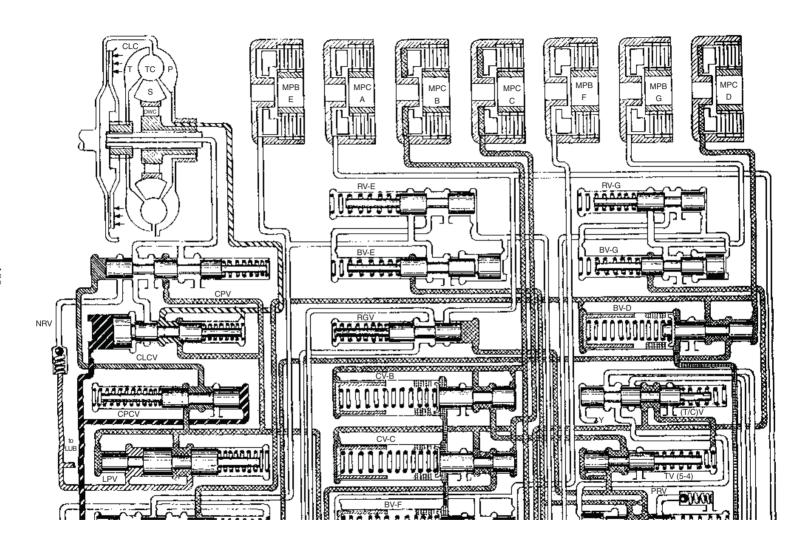
Fig. 5.43 Hydraulic/electronic transmission control system – third gear
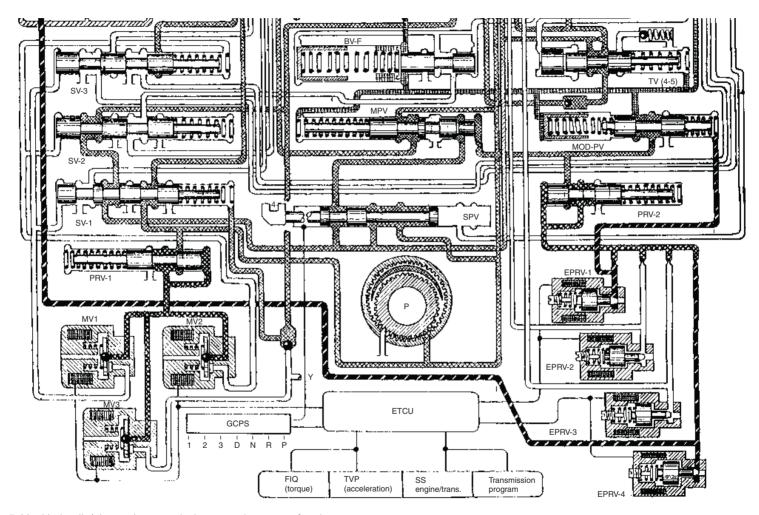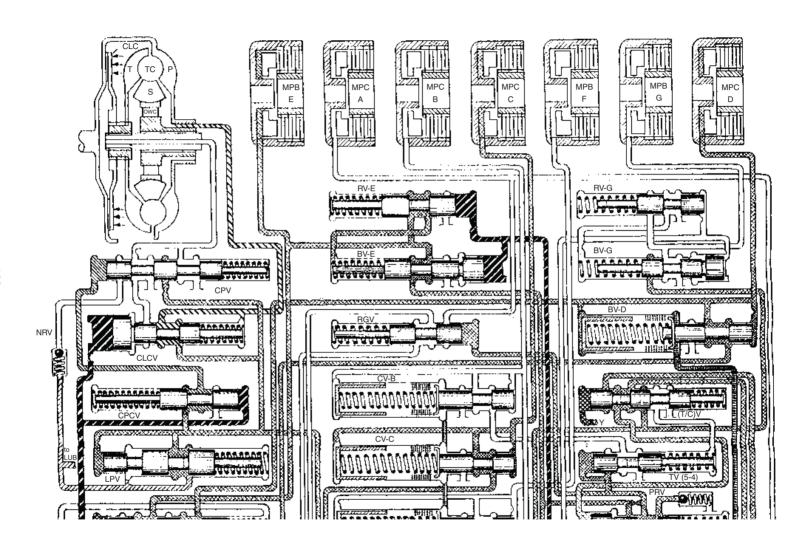
Fig. 5.44 Hydraulic/electronic transmission control system – fourth gear
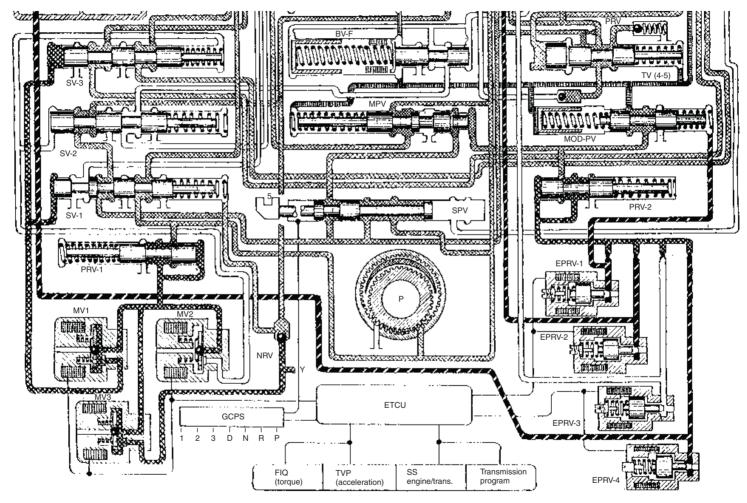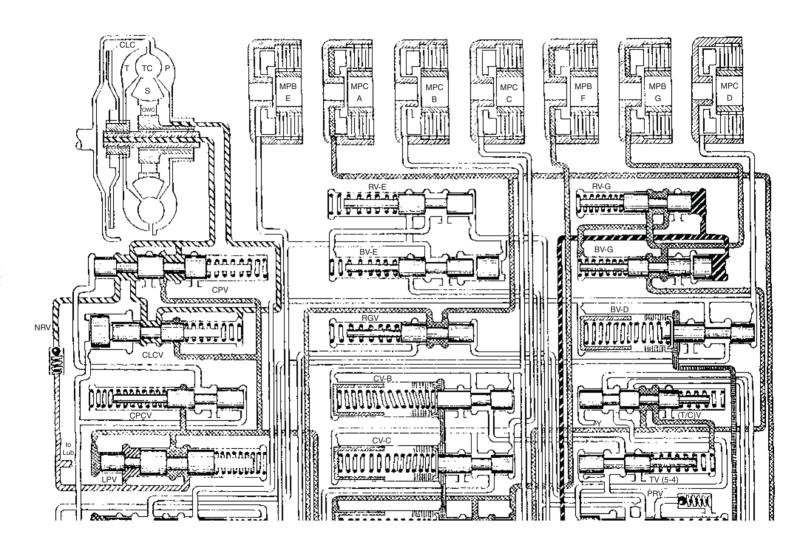
Fig. 5.45 Hydraulic/electronic transmission control system – fifth gear
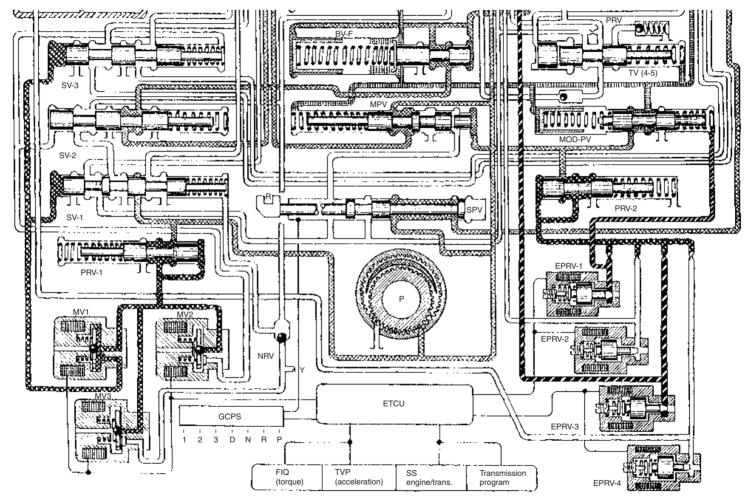
Fig. 5.46 Hydraulic/electronic transmission control system – reverse gear

3 Solenoid valve MV3 is in the de-energized state, it therefore blocks line pressure reaching traction/coasting valve (T/C)V via passage - .

4 Electronic pressure regulating valves EPRV-1 and EPRV-2 de-energized, this closes the valves and increases their respective regulating fluid pressure as for second gear.

5 Electronic pressure regulating valve EPRV-3 control current is increased, this causes the valve to open and the regulating fluid pressure to collapse. The returning spring now moves the clutch and retaining valves CV- and RV-respectively over to the right-hand side. Brake valve BV- now blocks the line pressure reaching the multiplate clutch MPB- and releases (exhausts) the line pressure imposed on the annular shaped brake piston the multiplate brake MPB- is therefore disengaged.

6 Solenoid valve MV1 is de-energized, this permits the shift valve SV-1 to return to the left-hand side. Subsequently line pressure now passes via the shift valve SV-1 to the clutch valve CV- and hence applies the multiplate clutch MPC- .

ourth gear (Fig. 5.44) Engagement of fourth gear is obtained by applying the multiplate clutches MPC-B, MPC-C and MPC- .

The shift from third to fourth gear is achieved in the following manner with the selector position valve in the drive range:

1 Multiplate clutches MPC-B and MPC-applied as for third gear.

2 Solenoid valves MV1 and MV3 de-energized and closed as for third gear.

3 Electronic pressure regulating valve EPRV-1 de-energized and partially closed, whereas EPRV-3 remains energized and open, both valves operating as for third gear.

4 Electronic pressure regulating valve EPRV-2 now progressively energizes and opens, this removes the control pressure from brake and retaining valves BV-E and RV-E respectively. ine pressure to brake valve BV-E is now blocked causing the release (exhausting) of fluid pressure via the brake valve BV-E and the disengagement of the multiplate brake MPB-E .

5 Fluid pressure now passes though to the multiplate clutch MPC-C via shift valves SV-1 and SV-2 , and clutch-valve CV-C . Subsequently, the multiplate clutch MPC-C is applied to complete the gear shift from third to fourth gear.

6 Electronic pressure regulating valve EPRV-4 de-energizes and progressively closes. Control pressure now shifts converter pressure control

valve CPCV to the left-hand side and converter lock-up clutch C CV to the right-hand side. Fluid pressure is thus supplied via the converter lock-up clutch valve C CV to the torque converter TC , whereas fluid pressure reaching the left-hand side of the torque converter lock-up clutch chamber is now blocked by the converter lock-up clutch valve C CV and exhausted by the converter pressure valve CPV . As a result fluid pressure within the torque converter pushes the lock-up clutch hard against the impeller rotor casing. Subsequently the transmission drive, instead of passing via fluid media from the impeller-rotor casing to the turbine-rotor output shaft, is now diverted directly via the lock-up clutch from the impeller-rotor casing to the turbine-rotor output shaft.

ifth gear (Fig. 5.45) Engagement of fifth gear is obtained by applying the multiplate clutches MPC-C and MPC- and the multiplate brake MPB-E .

The shift from fourth to fifth gear is achieved in the following manner with the selector position valve SPV in the drive range:

1 Multiplate clutches MPC-C and MPC-applied as for fourth gear.

2 Solenoid valve MV2 de-energized as for fourth gear.

3 Solenoid valve MV3 is energized, this allows fluid pressure via passage - to shift traction/coasting valve (T/C)V over to the right-hand side. As a result fluid pressure is released (exhausts) from the spring side of the traction valve TV (5–4) , hence fluid pressure acting on the left-hand end of the valve now enables it to shift to the right-hand side.

4 Solenoid valve MV1 is energized, this pressurizes the left-hand side of the shift valves SV-1 and SV-3 . owever, SV-1 cannot move over due to the existing fluid pressure acting on the spring end of the valve, whereas SV-3 is free to shift to the right-hand end. Fluid pressure from the clutch valve CV-E now passes via shift valve SV-3 and traction/coasting valve (T/C)V to the traction valve TV (4–5) causing the latter to shift to the right-hand side. Consequently traction valve TV (4–5) now blocks the main fluid pressure passing through the clutch valve CV-B and simultaneously releases the multiplate clutch MPC-B by exhausting the fluid pressure being applied to it.

5 Electronic pressure regulating valve EPRV-2 de-energized and partially closed. Controlled

fluid pressure now passes to the right-hand end of the clutch valve CV-E and retaining valve RV-E, thus causing both valves to shift to the left-hand end. Fluid pressure is now permitted to apply the multiplate brake MPB-E to complete the engagement of fifth gear.

6 Electronic pressure regulating valve EPRV-4 de-energized as for fourth gear. This causes the converter lock-up clutch C C to engage thereby by-passing the torque converter TC fluid drive.

Reverse gear (Fig. 5.46) Engagement of reverse gear is obtained by applying the multiplate clutch MPC-A and the multiplate brakes MPB-F and MPB- .

The shift from neutral to reverse gear is achieved in the following manner with the selector position valve SPV moved to reverse drive position R.

1 Multiplate brake MPB- applied as for neutral and park position.

2 Solenoid valve MV1 energized thus opening the valve. Constant fluid pressure now moves shift valves SV-1 and SV-3 over to the right-hand side.

3 Electronic pressure regulating valve EPRV-1 de-energized as for neutral position.

4 Electronic pressure regulating valves EPRV-3 de-energized and closed. Controlling fluid pressure is relayed to the brake valve BV- and retaining valve RV- . Both valves shift to the left-hand side thus permitting fluid pressure to reach and apply the multiplate brake MPB- .

5 Selector position valve SPV in reverse position diverts fluid pressure from the fluid pump, directly to multiplate clutch MPC-A and indirectly to multiplate brake MPB-F via the selector position valve SPV, reverse gear valve R V, shift valve SV-2 and the clutch valve CV-F. Both multiplate clutch MPC-A and multiplate brake MPB-F are therefore applied.

.11 Semi-automatic (manual gear change two pedal control) transmission system

5.11.1 Description of transmission system
(Fig. 5.48)

The system being described is broadly based on the F Man Tip Matic/ F AS Tronic 12 speed twin countershaft three speed constant mesh gearbox with a front mounted two speed splitter gear change and a rear positioned single stage two speed epicyclic gear range change however, the basic concept has been modified and considerably simplified in this text.

ear changes are achieved by four pneumatically operated power cylinders and pistons which are attached to the ends of the three selector rods, there being one power cylinder and piston for each of the splitter and range selector rods and two for the three speed and reverse constant mesh two piece selector rod. ear shifts are actuated by inlet and exhaust solenoid control valves which supply and release air to the various shift power cylinders as required (see Fig. 5.48).

A multiplate transmission brake with its inlet and exhaust solenoid control valves are provided to shorten the slow down period of the clutch, input shaft and twin countershaft assembly during the gear change process.

A single plate dry friction clutch is employed but instead of having a conventional clutch pedal to control the engagement and disengagement of the power flow, a pneumatic operated clutch actuator with inlet and exhaust solenoid control valves are used. Thus the manual foot control needed for driving away from rest and changing gear is eliminated.

radual engagement of the power flow via the clutch when pulling away from a standstill and smooth gear shift changes are achieved via the wheel speed and engine speed sensors, air pressure sensors and the electronic diesel control unit (E CU): this being part of the diesel engine management equipment, they all feed signals to the electronic transmission control unit (ETCU). This information is then processed so that commands to the various solenoid control valves can be made to produce the appropriate air pressure delivery and release to meet the changing starting and driving conditions likely to be experienced by a transmission system. A gear selector switch control stick provides the driver with a hand control which instructs the electronic transmission control unit (ETCU) to make an up and down gear shift when prevailing engine torque and road resistance conditions are matched.

5.11. Splitter gear change stage (Fig. 5.47)
Power flows via the clutch and input shaft to the splitter synchromesh dog clutch. The splitter synchromesh dog clutch can engage either the left or right hand matching dog clutch teeth on the central splitter gear mounted on the input shaft to obtain a low splitter gear ratio, or to the central third gear
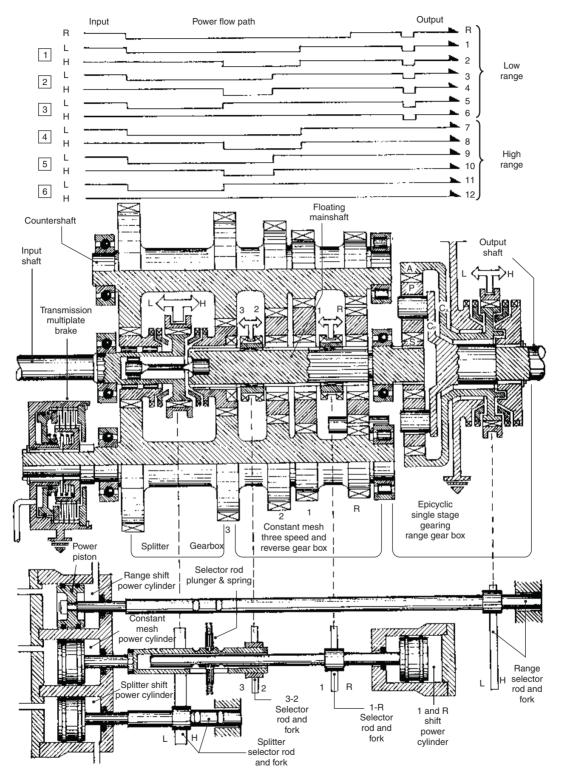
Fig. 5.47 Twin countershaft 12 speed constant mesh gearbox with synchromesh two speed splitter and range changes

mounted on the mainshaft to obtain the high splitter gear ratio. Power is now able to pass via the twin countershafts to each of the mainshaft constant mesh central gears by way of the constant mesh gears 1, 2, 3 and R.

5.11.3 Constant mesh 1 3 and gear stage
(Fig. 5.47)
The selection and engagement of one of the sliding dog clutch set of teeth either with R, 1, 2 or 3 floating mainshaft central constant mesh gears permits the drive path to flow from the twin countershaft gears via the mainshaft to the epicyclic range change single stage gear train.

5.11. ange change gear stage (Fig. 5.47)

ow range gear selection With the synchromesh dog clutch hub moved to the left-hand side, the internal toothed annular gear (A) will be held stationary the drive from floating mainshaft is therefore compelled to pass from the central sun gear (S) to the output shaft via the planet gear carrier (C_P) (see Fig. 5.47). Now since the annular gear is held stationary, the planet gears (P) are forced to rotate on their axes and also to roll around the internal teeth of the annular gear (A), consequently the planet carrier (C_P) and output shaft will now rotate at a lower speed than that of the sun gear (S) input.

igh range gear selection With the syncromesh dog clutch hub moved to the right-hand side, the annular gear (A) becomes fixed to the output shaft, therefore the drive to the planet gears (P) via the floating mainshaft and sun gear (S) now divides between the planet gear carrier (C_P) and the annular gear carrier (C_A) which are both fixed to the output shaft (see Fig. 5.47). As a result the planet gears (P) are prevented from rotating on their axes so that while the epicyclic gear train is compelled to revolve as one rigid mass, it therefore provides a one-to-one gear ratio stage.

5.11.5 Clutch engagement and disengagement
(Fig. 5.48)
With the ignition switched on and the first gear selected the clutch will automatically and progressively take up the drive as the driver depresses the accelerator pedal. The three basic factors which determine the smooth engagement of the transmission drive are vehicle load, which includes pulling away from a standstill and any road gradient, vehicle speed and engine speed. Thus the vehicle s resistance to move is monitored in terms of engine

load by the electronic diesel control unit E CU which is part of the diesel engine s fuel in ection equipment, and engine speed is also monitored by the E CU, whereas vehicle speed or wheel speed is monitored by the wheel brake speed sensors. These three factors are continuously being monitored, the information is then passed on to the electronic transmission control unit ETCU which processes it so that commands can be transferred in the form of electric current to the inlet and exhaust clutch actuator solenoid control valves.

Engagement and disengagement of clutch when pulling away from a standstill (Fig. 5.48) With the vehicle stationary, the ignition switched on and first gear selected, the informed ETCU energizes and opens the clutch solenoid inlet control valve whereas the exhaust control valve remains closed (see Fig. 5.48). Compressed air now enters the clutch cylinder actuator, this pushes the piston and rod outwards causing the clutch lever to pivot and to pull back the clutch release bearing and sleeve. As a result the clutch drive disc plate and input shaft to the gearbox will be disengaged from the engine. As the driver depresses the accelerator pedal the engine speed commences to increase (monitored by the engine speed sensor), the ETCU now progressively de-energizes the solenoid controlled clutch inlet valve and conversely energizes the solenoid controlled exhaust valve. The steady release of air from the clutch actuator cylinder now permits the clutch lever, release bearing and sleeve to move towards the engagement position where the friction drive plate is progressively squeezed between the flywheel and the clutch pressure plate. At this stage the transmission drive can be partially or fully taken up depending upon the combination of engine speed, load and wheel speed.

As soon as the engine speed drops below some predetermined value the ETCU reacts by de-energizing and closing the clutch exhaust valve and energizing and opening the clutch inlet valve, thus compressed air will again enter the clutch actuator cylinder thereby causing the friction clutch drive plate to once more disengage.

Note a built-in automatic clutch re-ad ustment device and wear travel sensor is normally incorporated within the clutch unit.

Engagement and disengagement of the clutch during a gear change (Fig. 5.48) When the driver moves the gear selector stick into another gear position
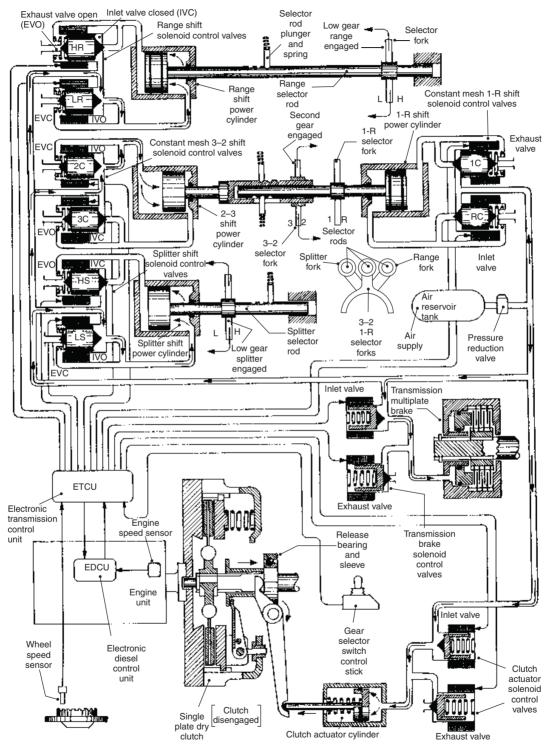
Fig. 5.48 A simplified electro/pneumatic gear shift and clutch control

with the vehicle moving forwards, the ETCU immediately signals the clutch solenoid control valves to operate so that the compressed air can bring about the disengagement and then engagement of the clutch drive plate for sufficient time (programmed time setting) for the gear shift to take place (see Fig. 5.48). This is achieved in the first phase by de-energizing and closing the clutch solenoid exhaust valve and correspondingly energizing and opening the inlet valve, thus permitting the compressed air to enter the clutch actuator cylinder and to release the clutch. The second phase de-energizes and closes the inlet valve and then energizes and opens the exhaust valve so that the clutch release mechanism allows the clutch to engage the transmission drive.

5.11.6 Transmission bra e (Figs 5.47 and 5.48)
This is a compressed air operated multiplate brake. Its purpose is to rapidly reduce the free spin speed of the driven disc plate, input shaft and twin countershaft masses when the clutch is disengaged thus enabling fast and smooth gear shifts to be made.

When a gear shift change is about to be made the driver moves the gear selector stick to a new position. This is signalled to the ETCU, and one outcome is that the transmission brake solenoid control inlet valve is energized to open (see Fig. 5.48). It thus permits compressed air to enter the piston chamber and thereby to squeeze together the friction disc plate so that the freely spinning countershafts are quickly dragged down to the main shaft s speed, see Fig. 5.47. nce the central gears wedged in between the twin countershafts have unified their speed with that of the mainshaft, then at this point the appropriate constant mesh dog clutch can easily slide into mesh with it ad acent central gear dog teeth. Immediately after the gear shift the transmission brake inlet valve closes and the exhaust valve opens to release the compressed air from the multiplate clutch cylinder thereby preventing excessive binding and strain imposed to the friction plates and assembly.

5.11.7 Splitter gear shifts (Fig. 5.48)
The splitter gear shift between low and high gear ratio takes place though a synchromesh type dog clutch device. Note for all the gear changes taking place in the gearbox, the splitter gears are constantly shifted from low to high going up the gear ratios or from high to low going down the gear ratios. With ignition switched on and the gear selector stick positioned say in low gear, the ETCU signals the splitter solenoid control to

close and open the inlet and exhaust valves respectively for the high splitter gear solenoid control, and at the same time to close and open the exhaust and inlet valves respectively for the low splitter gear solenoid control (see Fig. 5.48). The splitter shift power cylinder will now operate, compressed air will be released from the left-hand side and simultaneously compressed air will be introduced to the right-hand side of the splitter shift power cylinder the piston and selector rod now smoothly shift to the low splitter gear position. Conversely if high splitter gear was to be selected, the reverse would happen to the solenoid control valves with regards to their opening and closing so that the piston and selector rod would in this case move to the right.

5.11. ange gear shifts (Figs 5.47 and 5.48)
The range gear shift takes place though a single stage epicyclic gear train and operates also via a synchromesh type dog clutch mechanism.

oing though the normal gear change sequence from 1 to 12 the first six gear ratios one to six are obtained with the range gear shift in the low position and from seventh to twelfth gear in high range shift position, see Fig. 5.47.

With the ignition switched on and the gear selector stick moved to gear ratios between one and six the low range gear shift will be selected, the ETCU activates the range shift solenoid control valves such that the high range inlet and exhaust valves are closed, and opened respectively, whereas the low range inlet valve is opened and exhaust valve is closed, see Fig. 5.48. ence compressed air is exhausted from the left hand side of the range shift power cylinder and exposed to fresh compressed air on the right-hand side. Subsequently the piston and selector rod is able to quickly shift to the low range position.

A similar sequence of events takes place if the high range gear shift is required except the opening and closing of the valves will be opposite to that needed for the low range shift.

5.11. Constant mesh three speed and reverse gear shift (Figs 5.47 and 5.48)
These gear shifts cover the middle section of the gearbox which involves the engagement and disengagement of the various central mainshaft constant mesh gears via a pair of sliding dog clutches. There is a dog clutch for engagement and disengagement for gears 1-R and similarly a second dog clutch for gears 2–3.

To go though the complete gear ratio steps, the range shift is put initially into low, then the splitter gear shifts are moved alternatively into low and high as the constant mesh dog clutch gears are shifted progressively up this is again repeated but the second time with the range shift in high (see Fig. 5.47). This can be presented as range gear shifted into low, 1 gear constant mesh low and high splitter, 2 gear constant mesh low and high splitter, and 3 gear constant mesh low and high splitter gear at this point the range gear is shifted into high and the whole sequence is repeated, 1 constant mesh gear low and high splitter, 2 constant mesh gear low and high splitter and finally third constant mesh gear low and high splitter thus twelve gear ratios are produced thus:

First six overall gear ratios = splitter gear (and)S × constant mesh gears (1, 2 and 3) × range gear low (R)

Second six overall gear ratios = splitter gear (and) × constant mesh gears (1, 2 and 3) × range gear high (R).

$$1 \quad R = S \times CM\ 1\ \times\ R$$
$$2 \quad R = S \times CM\ 1\ \times\ R$$
$$3 \quad R = S \times CM\ 2\ \times\ R$$
$$4 \quad R = S \times CM\ 2\ \times\ R$$
$$5 \quad R = S \times CM\ 3\ \times\ R$$
$$6 \quad R = S \times CM\ 3\ \times\ R$$

$$7 \quad R = S \times CM\ 1\ \times\ R$$
$$8 \quad R = S \times CM\ 1\ \times\ R$$
$$9 \quad R = S \times CM\ 2\ \times\ R$$
$$10 \quad R = S \times CM\ 2\ \times\ R$$
$$11 \quad R = S \times CM\ 3\ \times\ R$$
$$12 \quad R = S \times CM\ 3\ \times\ R$$

where R = overall gear ratio
CM = constant mesh gear ratio
S/ S = low or high splitter gear ratio
S/ R = low or high range gear ratio

Assume that the ignition is switched on and the vehicle is being driven forwards in low splitter and low range shift gear positions (see Fig. 5.48). To engage one of the three forward constant mesh gears, for example, the second gear, then the gear selector stick is moved into 3 gear position (low splitter, low range 2 gear). Immediately the ETCU signals the constant mesh 3–2 shift solenoid control valves by energizing the 2 constant mesh solenoid control so that its inlet valve opens and its exhaust valve closes at the same time, the 3 constant mesh solenoid control is de-energized so that its inlet valve closes and the exhaust valve opens (see Fig. 5.48). Accordingly, the 2–3 shift power cylinder will be exhausted of compressed air on the right-hand side, while compressed air is delivered to the left-hand side of the cylinder, the difference in force between the two sides of the piston will therefore shift the 3–2 piston and selector rod into the second gear position. It should be remembered that during this time period, the clutch will have separated the engine drive from the transmission and that the transmission brake will have slowed the twin countershafts sufficiently for the constant mesh central gear being selected to equalize its speed with the mainshaft speed so that a clean engagement takes place. If first gear was then to be selected, the constant mesh 3–2 shift solenoid control valves would both close their exhaust valves so that compressed air enters from both ends of the 2–3 shift power cylinder, it therefore moves the piston and selector rod into the neutral position before the 1-R shift solenoid control valves are allowed to operate.

Transmission bearings and constant velocity oints

.1 Rolling contact bearings

Bearings which are designed to support rotating shafts can be divided broadly into two groups the plain lining bearing, known as the
and the   The fundamental difference between these bearings is how they provide support to the shaft. With plain sleeve or lining bearings, metal to metal contact is prevented by the generation of a hydrodynamic film of lubricant (oil wedge), which supports the shaft once operating conditions have been established. owever, with the rolling contact bearing the load is carried by balls or rollers with actual metal to metal contact over a relatively small area.

With the conventional ournal bearing, starting friction is relatively high and with heavy loads the coefficient of friction may be in the order of 0.15.

 owever, with the rolling contact bearing the starting friction is only slightly higher than the operating friction. In both groups of bearings the operating coefficients will be very similar and may range between 0.001 and 0.002. ydrodynamic ournal bearings are sub ected to a cyclic pro ected pressure loading over a relatively large surface area and therefore en oy very long life spans. For example, engine big-ends and main ournal bearings may have a service life of about 160 000 kilometres (100 000 miles). Unfortunately, the inherent nature of rolling contact bearing raceway loading is of a number of stress cycles of large magnitude for each revolution of the shaft so that the life of these bearings is limited by the fatigue strength of the bearing material.

 ubrication of plain ournal bearings is very important. They require a pressurized supply of consistent viscosity lubricant, whereas rolling contact bearings need only a relatively small amount of lubricant and their carrying capacity is not sensitive to changes in lubricant viscosity. Rolling contact bearings have a larger outside diameter and are shorter in axial length than plain ournal bearings.

Noise levels of rolling contact bearings at high speed are generally much higher than for plain ournal bearings due mainly to the lack of a hydrodynamic oil film between the rolling elements and their tracks and the windage effects of the ball or roller cage.

6.1.1   inear motion of a ball bet een t o flat trac s (Fig. 6.1)

Consider a ball of radius $_b$ placed between an upper and lower track plate (Fig. 6.1). If the upper track plate is moved towards the right so that the ball completes one revolution, then the ball has rolled along the lower track a distance 2 $_b$ and the upper track has moved ahead of the ball a further distance 2 $_b$. Thus the relative movement,   between both track plates will be 2 $_b$ 2 $_b$, which is twice the distance,   travelled forward by the centre of the ball. In other words, the ball centre will move forward only half that of the upper to lower relative track movement.

i.e. $\qquad - = \dfrac{2 \quad _b}{4 \quad _b} = \dfrac{1}{2}$

$\qquad \therefore \quad = \dfrac{}{2}$

6.1. Ball rolling contact bearing friction

(Fig. 6.2(a and b))

When the surfaces of a ball and track contact under load, the profile a–b–c of the ball tends to flatten out and the profile a–e–c of the track becomes concave (Fig. 6.2(a)). Subsequently the pressure between the contact surfaces deforms them into a common elliptical shape a–d–c. At the same time, a bulge will be established around the contact edge of the ball due to the displacement of material.

If the ball is made to roll forward, the material in the front half of the ball will be sub ected to increased compressive loading and distortion whilst that on the rear half experiences pressure release (Fig. 6.2(b)). As a result, the stress distribution over the contact area will be constantly varying.

The energy used to compress a perfect elastic material is equal to that released when the load is removed, but for an imperfect elastic material (most materials), some of the energy used in straining the material is absorbed as internal friction (known as   ) and is not released when the load is removed. Therefore, the energy absorbed by the ball and track when sub ected to a compressive load, causing the steel to distort, is greater than that released as the ball moves forward. It is this missing
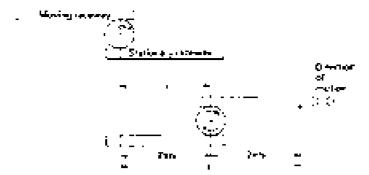
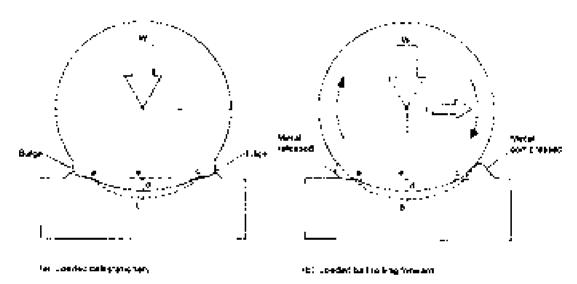**Fig. 6.1** Relationship of rolling element and raceway movement

Fig. 6.2 (a and b) Ilustration of rolling ball resistance against motion

energy which creates a friction force opposing the forward motion of the ball.

wing to the elastic deformation of the contact surfaces of the ball and track, the contact area will no longer be spherical and the contact profile arc will therefore have a different radius to that of the ball (Fig. 6.2(b)). As a result, the line a–e–c of the undistorted track surface is shorter in length than the rolling arc profile a–d–c. In one revolution the ball will move forward a shorter distance than if the ball contact contour was part of a true sphere. ence the discrepancy of the theoretical and actual forward movement of the ball is accommodated by slippage between the ball and track interfaces.

6.1.3 *adial ball bearings* (Fig. 6.3)
The essential elements of the multiball bearing is the inner externally grooved and the outer intern-ally grooved ring races (tracks). odged between these inner and outer members are a number of balls which roll between the grooved tracks when relative angular motion of the rings takes place (Fig. 6.3(a)). A fourth important component which is not sub ected to radial load is the ball cage or retainer whose function is to space the balls apart so that each ball takes its share of load as it passes from the loaded to the unloaded side of the bearing. The cage prevents the balls piling up and rubbing together on the unloaded bearing side.

ontact area The area of ball to track groove con-tact will, to some extent, determine the load carry-ing capacity of the bearing. Therefore, if both ball and track groove profiles more or less conform, the bearing load capacity increases. Most radial ball bearings have circular grooves ground in the inner

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and outer ring members, their radii being 2–4% greater than the ball radius so that ball to track contact, friction, lubrication and cooling can be controlled (Fig. 6.3(a)). An unloaded bearing produces a ball to track point contact, but as the load is increased, it changes to an elliptical contact area (Fig. 6.3(a)). The outer ring contact area will be larger than that of the inner ring since the track curvature of the outer ring is in effect concave and that of the inner ring is convex.

Bearing failure The inner ring face is subjected to a lesser number of effective stress cycles per revolution of the shaft than the corresponding outer ring race, but the maximum stress developed at the inner race because of the smaller ball contact area as opposed to the outer race tends to be more critical in producing earlier fatigue in the inner race than that at the outer race.

Lubrication Single and double row ball bearings can be externally lubricated or they may be pre-packed with grease and enclosed with side covers to prevent the grease escaping from within and at the same time stopping dust entering the bearing between the track ways and balls.

6.1. Relative movement of radial ball bearing elements (Fig. 6.3(b))

The relative movements of the races, ball and cage may be analysed as follows:

Consider a ball of radius r_b revolving N_b rev/min without slip between an inner rotating race of radius r_i and outer stationary race of radius r_o (Fig. 6.3(b)). Let the cage attached to the balls

be at a pitch circle radius r_p and revolving at N_c rev/min.

$$\text{linear speed of ball} = 2\pi r_b N_b \,(\text{m/s}) \quad (1)$$
$$\text{linear speed of inner race} = 2\pi r_i N_i \,(\text{m/s}) \quad (2)$$
$$\text{linear speed of cage} = 2\pi r_p N_c \,(\text{m/s}) \quad (3)$$

$$\text{Pitch circle radius} \quad r_p = \frac{r_i + r_o}{2} \,(\text{m}) \quad (4)$$

But the linear speed of the cage is also half the speed of the inner race

i.e. $\dfrac{2\pi r_i N_i}{2}$

Hence linear speed of

cage $= \pi r_i N_i \,(\text{m/s}) \quad (5)$

If no slip takes place,

linear speed of ball = linear speed of inner race

$$2\pi r_b N_b = 2\pi r_i N_i$$

$$\therefore N_b = \frac{r_i}{r_b} N_i \,(\text{rev/min}) \quad (6)$$

linear speed of cage = half inner speed of inner race

$$2\pi r_p N_c = \pi r_i N_i$$

Hence $N_c = \dfrac{r_i}{2 r_p} N_i$

$$\therefore N_c = \frac{r_i}{r_p}\frac{N_i}{2} \,(\text{rev/min}) \quad (7)$$

A single row radial ball bearing has an inner and outer race diameter of 50 and 70 mm respectively.

If the outer race is held stationary and the inner race rotates at 1200 rev/min, determine the following information:
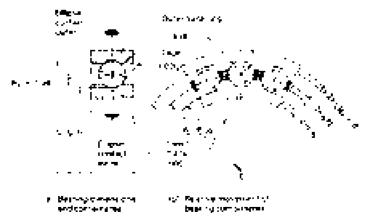

Fig. 6.3 (a and b) Deep groove radial ball bearing terminology

(a) The number of times the balls rotate for one revolution of the inner race.

(b) The number of times the balls rotate for them to roll round the outer race once.

(c) The angular speed of balls.

(d) The angular speed of cage.

(a) Diameter of balls $= d_o - d_i$

$$= 35 - 25 = 10 \text{ mm}$$

Assuming no slip,

$$\begin{array}{c}\text{Number of} \\ \text{ball rotations}\end{array} \times \begin{array}{c}\text{Ball} \\ \text{circumference}\end{array} = \begin{array}{c}\text{Inner race} \\ \text{circumference}\end{array}$$

Number of ball rotations, $2\pi d_b = 2\pi d_i$

∴ Number of ball revolutions

$$= \frac{2\pi d_i}{2\pi d_b} = \frac{d_i}{d_b}$$

$$= \frac{25}{5} = 5 \text{ revolutions}$$

(b)
$$\begin{array}{c}\text{Number of} \\ \text{ball rotations}\end{array} \times \begin{array}{c}\text{Ball} \\ \text{circumference}\end{array} = \begin{array}{c}\text{Outer race} \\ \text{circumference}\end{array}$$

Number of ball rotations, $2\pi d_b = 2\pi d_o$

∴ Number of rotations $= \dfrac{2\pi d_o}{2\pi d_b} = \dfrac{d_o}{d_b}$

$$= \frac{35}{5} = 7 \text{ revolutions}$$

(c) Ball angular speed $\omega_b = \dfrac{d_i}{d_b}\,\omega_i$

$$= \frac{25}{5} \times 1200$$

$$= 6000 \text{ rev/min}$$

(d) $d_p = \dfrac{d_i + d_o}{2} = \dfrac{25 + 35}{2}$

$$= 30 \text{ mm}$$

Cage angular speed $\omega_c = \dfrac{d_i}{d_p}\,\dfrac{\omega_1}{2}$

$$= \frac{25}{30} \times \frac{1200}{2}$$

$$= 500 \text{ rev/min}$$

6.1.5 Bearing loading

Bearings used to support transmission shafts are generally subjected to two kinds of loads:

1 A load (force) applied at right angles to the shaft and bearing axis. This produces an outward
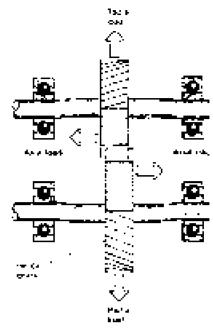
Fig. 6.4 Ilustration of radial and axial bearing loads

force which is known as a radial force. This kind of loading could be caused by pairs of meshing spur gears radially separating from each other when transmitting torque (Fig. 6.4).

2 A load (force) applied parallel to the shaft and bearing axis. This produces an end thrust which is known as an axial force. This kind of loading could be caused by pairs of meshing helical gears trying to move apart axially when transmitting torque (Fig. 6.4).

When both radial and axial loads are imposed on a ball bearing simultaneously they result in a single combination load within the bearing which acts across the ball as shown (Fig. 6.6).

6.1.6 Ball and roller bearing internal clearance

Internal bearing clearance refers to the slackness between the rolling elements and the inner and outer raceways they roll between. This clearance is measured by the free movement of one raceway ring relative to the other ring with the rolling elements in between (Fig. 6.5). For ball and cylindrical (parallel) roller bearings, the radial or diametrical clearance is measured perpendicular to the axis of the bearing. Deep groove ball bearings also have axial clearance measured parallel to the axis of the bearing. Cylindrical (parallel) roller bearings without inner and outer ring end flanges do not have axial clearance. Single row angular contact bearings and
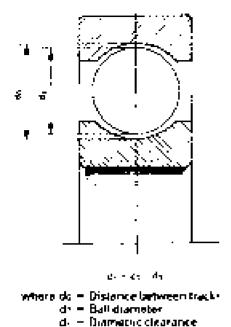
$d_c = d_t - d_b$

where d_t = Distance between track
      d_b = Ball diameter
      d_c = Diametric clearance

Fig. 6.5 Internal bearing diametric clearance

taper roller bearings do have clearance slackness or tightness under operating conditions but this cannot be measured until the whole bearing assembly has been installed in its housing.

A radial ball bearing working at operating temperature should have little or no diametric clearance, whereas roller radial bearings generally operate more efficiently with a small diametric clearance.

Radial ball and roller bearings have a much larger initial diametric clearance before being fitted than their actual operating clearances.

The difference in the initial and working diametric clearances of a bearing, that is, before and after being fitted, is due to a number of reasons:

1 The compressive interference fit of the outer raceway member when fitted in its housing slightly reduces diameter.
2 The expansion of the inner raceway member when forced over its shift minutely increases its diameter.

The magnitude of the initial contraction or expansion of the outer and inner raceway members will depend upon the following:

a) The rigidity of the housing or shaft is it a low strength aluminium housing, moderate strength

cast iron housing or a high strength steel housing Is it a solid or hollow shaft are the inner and outer ring member sections thin, medium or thick
b) The type of housing or shaft fit is it a light, medium or heavy interference fit

The diametric clearance reduction when an inner ring is forced over a solid shaft will be a proportion of the measured ring to shaft interference.

The reductions in diametric clearance for a heavy and a thin sectioned inner raceway ring are roughly 50% and 80% respectively.   iametric clearance reductions for hollow shafts will of course be less.

Working bearing clearances are affected by the difference in temperature between the outer and inner raceway rings which arise during operation. Because the inner ring attached to its shaft is not cooled so effectively as the outer ring which is supported in a housing, the inner member expands more than the outer one so that there is a tendency for the diametric clearance to be reduced due to the differential expansion of the two rings.

Another reason for having an initial diametric clearance is it helps to accommodate any inaccuracies in the machining and grinding of the bearing components.

The diametric clearance affects the axial clearance of ball bearings and in so doing influences their capacity for carrying axial loads. The greater the diametric clearance, the greater the angle of ball contact and therefore the greater the capacity for supporting axial thrust (Fig. 6.6).

Bearing internal clearances have been so derived that under operating conditions the existing clearances provide the optimum radial and axial load carrying capacity, speed range, quietness of running and life expectancy. As mentioned previously, the diametric clearance is greatly influenced by the type of fit between the outer ring and its housing and the inner ring and its shaft, be they a slip, push, light press or heavy press interference.

The tightness of the bearing fit will be determined by the extremes of working conditions to which the bearing is sub ected. For example, a light duty application will permit the bearing to be held with a relatively loose fit, whereas for heavy conditions an interference fit becomes essential.

To compensate for the various external fits and applications, bearings are manufactured with different diametric clearances which have been standardized by BSI and IS . ournal bearings are made with a range of diametrical clearances, these clearances being designated by a series of codes shown below in Table 6.1.
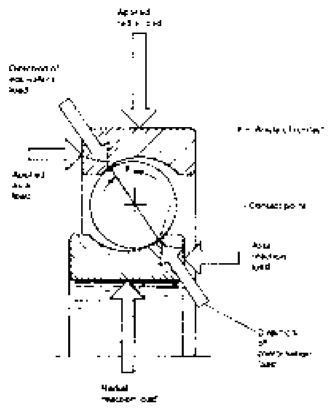
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Fig. 6.6 Effects of diametric clearance and axial load on angle of contact

ab 6.1 ournal bearing diametrical clearances

BS esignation	S roup	S F esignation	Hoffmann esignation
	roup 2	C2	
C2	Normal group	Normal	
C	roup	C	
	roup	C	

The lower the number the smaller is the bearing s diametric clearance. In the new edition of BS 292 these designations are replaced by the IS groups. For special purposes, bearings with a smaller diametric clearance such as roup 1 and larger roup 5 are available.

The diametrical clearances 0, 00, 000 and 0000 are usually known as or fits. These clearances are identified by the appropriate code or number of polished circles on the stamped side of the outer ring.

The applications of the various diametric clearance groups are compared as follows:

roup 2 These bearings have the least diametric clearance. Bearings of this group are suitable when freedom from shake is essential in the assembled bearing. The fitting interference tolerance prevents the initial diametric clearance being eliminated. Very careful attention must be given to the bearing housing and shaft dimensions to prevent the expansion of the inner ring or the contraction of the outer ring causing bearing tightness.

ormal group Bearings in this group are suitable when only one raceway ring has made an interference fit and there is no appreciable loss of clearance due to temperature differences. These diametric clearances are normally adopted with radial ball bearings for general engineering applications.

roup 3 Bearings in this group are suitable when both outer and inner raceway rings have made an interference fit or when only one ring has an interference fit but there is likely to be some loss of clearance due to temperature differences. Roller

bearings and ball bearings which are sub ected to axial thrust tend to use this diametric clearance grade.

roup 4 Bearings in this group are suitable when both outer and inner bearing rings are an interference fit and there is some loss of diametric clearance due to temperature differences.

6.1.7 Taper roller bearings

escription of bearing construction (Fig. 6.7) The taper roller bearing is made up of four parts the inner raceway and the outer raceway, known respectively as the and the taper rollers shaped as frustrums of cones and the cage or roller retainer (Fig. 6.8). The taper rollers and both inner and outer races carry load whereas the cage carries no load but performs the task of spacing out the rollers around the cone and retaining them as an assembly.

Taper roller bearing true rolling principle (Fig. 6.8(a and b)) If the axis of a cylindrical (parallel) roller is inclined to the inner raceway axis, then the relative rolling velocity at the periphery of both the outer and inner ends of the roller will tend to be different due to the variation of track diameter (and therefore circumference) between the two sides of the bearing. If the mid position of the roller produced true rolling without slippage, the portion of the roller on the large diameter side of the tracks would try to slow down whilst the other half of the roller on the smaller diameter side of the tracks would tend to speed up. Consequently both ends would slip continuously as the central raceway
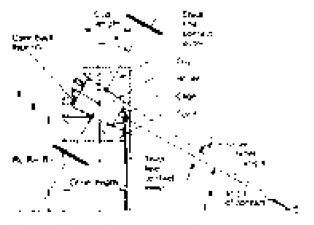
member rotated relative to the stationary outer race members (Fig. 6.8(a)).

The design geometry of the taper roller bearing is therefore based on the cone principle (Fig. 6.8(b)) where all pro ection lines, lines extending from the cone and cup working surfaces (tracks), converge at one common point on the axis of the bearing.

With the converging inner and outer raceway (track) approach, the track circumferences at the large and small ends of each roller will be greater and smaller respectively. The different surface velocities on both large and small roller ends will be accommodated by the corresponding change in track circumferences. ence no slippage takes place, only pure rolling over the full length of each roller as they revolve between their inner and outer tracks.

ngle of contact (Fig. 6.7) Taper roller bearings are designed to support not only radial bearing loads but axial (thrust) bearing loads.

The angle of bearing contact Θ, which determines the maximum thrust (axial) load, the bearing can accommodate is the angle made between the perpendiculars to both the roller axis and the inner cone axis (Fig. 6.7). The angle of contact Θ is also half the pitch cone angle. These angles can range from as little as 7 to as much as 30 . The standard or normal taper roller bearing has a contact angle of 12–16 which will accommodate moderate thrust (axial) loads. For large and very large thrust loads, medium and steep contact angle bearings are available, having contact angles in the region of 20 and 28 respectively.

rea of contact (Fig. 6.7) Contact between roller and both inner cone and outer cup is of the line form without load, but as the rollers become loaded the elastic material distorts, producing a thick line contact area (Fig. 6.7) which can support very large combinations of both radial and axial loads.

age (Fig. 6.7) The purpose of the cage container is to equally space the rollers about the periphery of the cone and to hold them in position when the bearing is operating. The prevention of rolling elements touching each other is important since ad acent rollers move in opposite directions at their points of closest approach. If they were allowed to touch they would rub at twice the normal roller speed.

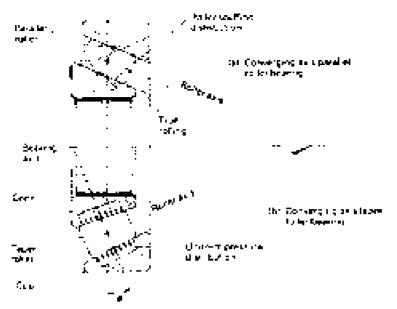The cage resembles a tapered perforated sleeve (Fig. 6.7) made from a sheet metal stamping which

Fig. 6.7 Taper rolling bearing terminology

Fig. 6.8 rinciple of taper rolling bearing

has a series of roller pockets punched out by a single impact of a multiple die punch.

Although the back cone rib contributes most to the alignment of the rollers, the bearing cup and cone sides furthest from the point of bearing loading may be slack and therefore may not be able to keep the rollers on the unloaded side in their true plane. Therefore, the cage (container) pockets are precisely chamfered to conform to the curvature of the rollers so that any additional corrective alignment which may become necessary is provided by the individual roller pockets.

Positive roller alignment (Fig. 6.9)   Both cylindrical parallel and taper roller elements, when rolling between inner and outer tracks, have the tendency to skew (tilt) so that extended lines drawn through their axes do not intersect the bearing axis at the same cone and cup pro ection apex. This problem has been overcome by grinding the large end of each roller flat and perpendicular to its axis so that all the rollers square themselves exactly with a shoulder or rib machined on the inner cone (Fig. 6.9). When there is any relative movement between the cup and cone, the large flat ends of the rollers make contact with the ad acent shoulder (rib) of the cone, compelling the rollers to positively align themselves

between the tapered faces of the cup and cone without the guidance of the cage. The magnitude of the roller-to-rib end thrust, known as the

will depend upon the taper roller contact angle.
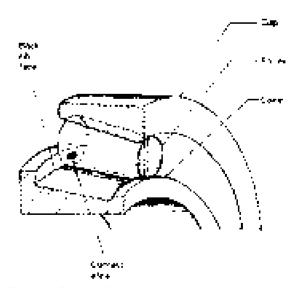
Fig. 6.9 Roller self-alignment

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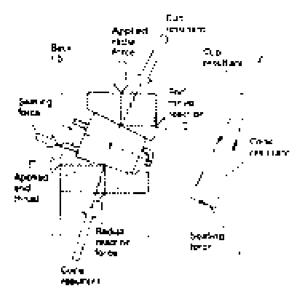
Fig. 6.10 Force diagram illustrating positive roller alignment seating force

Self-alignment roller to rib seating force (Fig. 6.10) To make each roller do its full share of useful work, positive roller alignment is achieved by the large end of each roller being ground perpendicular to its axis so that when assembled it squares itself exactly with the cone back face rib (Fig. 6.10).

When the taper roller bearing is running under operating conditions it will generally be sub ected to a combination of both radial and axial loads. The resultant applied load and resultant reaction load will be in apposition to each other, acting perpendicular to both the cup and cone track faces. Since the rollers are tapered, the direction of the perpendicular resultant loads will be slightly inclined to each other, they thereby produce a third force parallel to the rolling element axis. This third force is known as the   and it is this force which provides the rollers with their continuous alignment to the bearing axis. The magnitude of this   is a function of the included taper roller angle which can be obtained from a triangular force diagram (Fig. 6.10). The diagram assumes that both the resultant applied and reaction loads are equal and that their direction lies perpendicular to both the cup and cone track surface. A small roller included angle will produce a small rib seating force and vice versa.

6.1. Bearing materials
Bearing inner and outer raceway members and their rolling elements, be they balls or rollers, can

be made from either a case hardening alloy steel or a through hardened alloy steel.

a) The case hardened steel is usually a low alloy nickel chromium or nickel-chromium molybdenum steel, in which the surface only is hardened to provide a wear resistance outer layer while the soft, more ductile core enables the bearing to withstand extreme shock and overloading.
b) The through hardened steel is generally a high carbon chromium steel, usually about 1.0% carbon for adequate strength, together with 1.5% chromium to increase hardenability. (This is the ability of the steel to be hardened all the way through to a 60–66 Rockwell C scale.)

The summary of the effects of the alloying elements is as follows:

Nickel increases the tensile strength and toughness and also acts as a grain refiner. Chromium considerably hardens and raises the strength with some loss in ductility, whilst molybdenum reduces the tendency to temper-brittleness in low nickel low chromium steel.

Bearing inner and outer raceways are machined from a rod or seamless tube. The balls are produced by closed die forging of blanks cut from bar stock, are rough machined, then hardened and tempered until they are finally ground and lapped to size.

Some bearing manufacturers use case-hardened steel in preference to through-hardened steel because it is claimed that these steels have hard fatigue resistant surfaces and a tough crack-resistant core. Therefore these steels are able to withstand impact loading and prevent fatigue cracks spreading through the core.

6.1. Bearing friction
The friction resistance offered by the different kinds of rolling element bearings is usually quoted in terms of the coefficient of friction so that a relative comparison can be made. Bearing friction will vary to some extent due to speed, load and lubrication. ther factors will be the operating conditions which are listed as follows:

1 Starting friction will be higher than the dynamic normal running friction.
2 The quantity and viscosity of the oil or grease a large amount of oil or a high viscosity will increase the frictional resistance.
3 New unplanished bearings will have higher coefficient of friction values than worn bearings which have bedded down.

4 Preloading the bearing will initially raise the coefficient of friction but under working conditions it may reduce the overall coefficient value.

5 Pre-lubricated bearings may have slightly higher coefficients than externally lubricated bearings due to the rubbing effect of the seals.

oefficient of friction average values for various bearing arrangements

Self-alignment ball bearings	=	0.001
Cylindrical roller bearing	=	0.0011
Thrustball bearings	=	0.0013
Single row deep grooveball bearings	=	0.0015
Taper and spherical roller bearings	=	0.0018

6.1.10 Ball and roller bearing load distribution
(Fig. 6.11)

When either a ball or roller bearing is sub ected to a radial load, the individual rolling elements will not be loaded equally but will be loaded according to their disposition to the direction of the applied load. Applying a radial load to a bearing shaft pushes the inner race towards the outer race in the direction of the load so that the balls or rollers in one half of the bearing do not support any load whereas the other half of the bearing reacts to the load (Fig. 6.11(a)). The distribution of load on the reaction side of the bearing will vary considerably with the diametrical rolling element clearance and the mounting rigidity preventing deformation of the bearing assembly.

If the internal radial clearances of the rolling elements are zero and the inner and outer bearing races remain true circles when loaded, the load distribution will span the full 180 so that approximately half the balls or rollers will, to some extent, share the radial load (Fig. 6.11(b)). Conversely slackness or race circular distortion under load will reduce the pro ected load zone so that the rolling elements which provide support will be very much more loaded resulting in considerably more shaft deflection under load. ightly preloaded bearings may extend the radial load zone to something greater than 180 but less than 360 (Fig. 6.11(c)). This form of initial bearing loading will eliminate gear mesh teeth misalignment due to shaft deflection under operating conditions. eavy bearing preloading may extend the load zone to 360 (Fig. 6.11(d)) this degree of preloading should only be used for severe working conditions or where large end thrust is likely to be encountered and must be absorbed without too much axial movement.

End thrusts (axial loads), unlike radial loads, produce a uniform load distribution pattern around the bearing (Fig. 6.11(e)). eep groove radial ball bearings can tolerate light end thrust. Angular contact ball bearings are capable of supporting medium axial loads. Taper roller bearings, be they normal or steep angled, can operate continuously under heavy and very heavy end loads respectively. nly if the shaft being supported deflects will the end load distribution become uneven.

6.1.11 Bearing fatigue

Fatigue in ball or roller bearings is caused by repeated stress reversals as the rolling elements move around the raceways under load. The periodic elastic compression and release as the rolling elements make their way around the tracks will ultimately overwork and rupture the metal ust below the surface. As a result, tiny cracks propagate almost parallel to the surface but ust deep enough to be invisible. With continuous usage the alternating stress cycles will cause the cracks to extend, followed by new cracks sprouting out from the original ones. Eventually there will be a network of minute interlinking cracks rising and merging together on the track surface. Subsequently, under further repeating stress cycles, particles will break away from the surface, the size of material leaving the surface becoming larger and larger. This process is known as of the bearing and eventually the area of metal which has come away will end the effective life of the bearing. If bearing accuracy and low noise level is essential the bearing will need to be replaced, but if bearing slackness and noise can be accepted, the bearing can continue to operate until the rolling elements and their tracks find it impossible to support the load.

6.1.1 olling contact bearing types

Single row deep grooved radial ball bearing (Fig. 6.12) These bearings are basically designed for light to medium radial load operating conditions. An additional feature is the depth of the grooves combined with the relatively large size of the balls and the high degree of conformity between balls and grooves which gives the bearing considerable thrust load carrying capacity so that the bearing will operate effectively under both radial and axial loads.

These bearings are suitable for supporting gearbox primary and secondary shafts etc..

Single row angular contact ball bearing (Fig. 6.13) Bearings of this type have ball tracks which are so
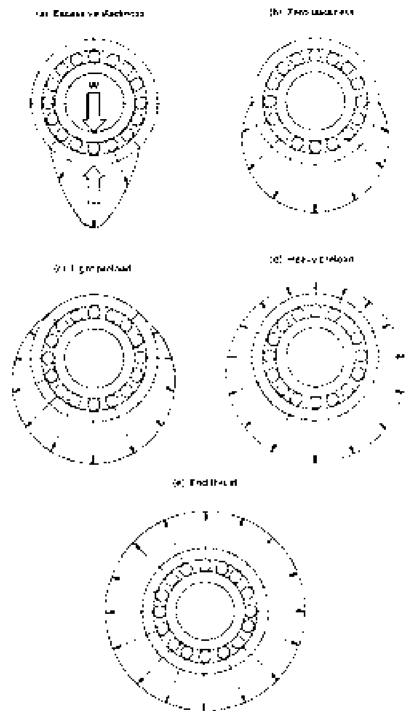
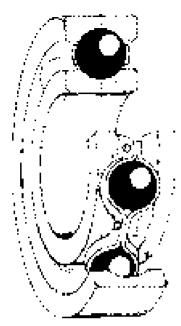**Fig. 6.11 (a–)** Bearing radial and axial load distribution

Fig. 6.12 Single row deep groove radial ball bearing

Fig. 6.13 Single row deep angular contact ball bearing

disposed that a line through the ball contact forms an acute angle with the bearing shaft axes. Ball to track ring contact area is elliptical and therefore with the inclined contact angle this bearing is particularly suitable for heavy axial loads. Ad ustment of these bearings must always be towards another bearing capable of dealing with axial loads in the opposite direction. The standard contact angle is 20 , but for special applications 12, 15, 25 and 30 contact angle bearings are available. These bearings are particularly suited for supporting front and rear wheel hubs, differential cage housings and steering box gearing such as the rack and pinion.

ouble row angular contact ball bearings (Fig. 6.14) With this double row arrangement, the ball tracks are ground so that the lines of pressure through the balls are directed towards two comparatively widely separated points on the shaft. These bearings are normally preloaded so that even when sub ected to axial loads of different magnitudes, axial deflection of the shaft is minimized. End thrust in both axial directions can be applied and at the same time very large radial loads can be carried for a relatively compact bearing assembly.

A typical application for this type of bearing would be a semi- or three-quarter floating outer

Fig. 6.14 ouble row angular contact ball bearing
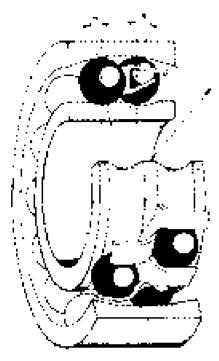
Fig. 6.15 ouble row self-alignment ball bearing

half shaft bearing, gearbox secondary output shaft bearing etc.

ouble row self-aligning ball bearing (Fig. 6.15) This double row bearing has two rows of balls which operate in individual inner raceway grooves in con unction with a common spherical outer raceway ring. The spherical outer track enables the inner ring and shaft to deflect relative to the outer raceway member, caused by the balls not only rolling between and around their tracks but also across the common outer circular track. Thus the self-aligning property of the bearing automatically ad usts any angular deflection of the shaft due to mounting errors, whip or settlement of the mounting. It also prevents the bearing from exerting a bending influence on the shaft. The radial load capacity for a single row self-aligning bearing is considerably less than that for the deep groove bearing due to the large radius of the outer spherical race providing very little ball to groove contact. This limitation was solved by having two staggered rows of balls to make up for the reduced ball contact area.

Note that double row deep groove bearings are not used because radial loads would be distributed unevenly between each row of balls with a periodic shaft deflection. They are used for intermediate

propellor shaft support, half shafts and wherever excessive shaft deflection is likely to occur.

Single row cylindrical roller bearing (Fig. 6.16) In this design of roller bearing, the rollers are guided by flanges, one on either the inner or outer track ring. The other ring does not normally have a flange. Consequently, these bearings do not take axial loads and in fact permit relative axial deflection of shafts and bearing housing within certain limits. These bearings can carry greater radial loads than the equivalent size groove bearing and in some applications both inner and outer ring tracks are flanged to accommodate very light axial loads.

Bearings of this type are used in gearbox and final drive transmissions where some axial alignment may be necessary.

Single row taper roller bearing (Fig. 6.17) The geometry of this class of bearing is such that the axes of its rollers and conical tracks form an angle with the shaft axis. The taper roller bearing is therefore particularly adaptable for applications where large radial and axial loads are transmitted simultaneously. For very severe axial loads, steep taper angle bearings are available but to some extent this is at the expense of the bearing s radial load carrying capacity. With taper bearings, ad ustment must always be towards another bearing capable of dealing with axial forces acting in the opposite direction. This is a popular bearing for medium and heavy duty wheel hubs, final drive pinion shafts, the differential cage and crownwheel bearings, for heavy duty gearbox shaft support and in-line in ection pump camshafts.

ouble row taper roller bearing (Fig. 6.18) These bearings have a double cone and two outer single cups with a row of taper rollers filling the gap between inner and outer tracks on either side. The compactness of these bearings makes them particularly suitable when there is very little space and where large end thrusts must be supported in both axial directions. Thus in the case of a straddled final drive pinion bearing, these double row taper bearings are more convenient than two single row bearings back to back. Another application for these double tow taper roller bearings is for transversely mounted gearbox output shaft support.

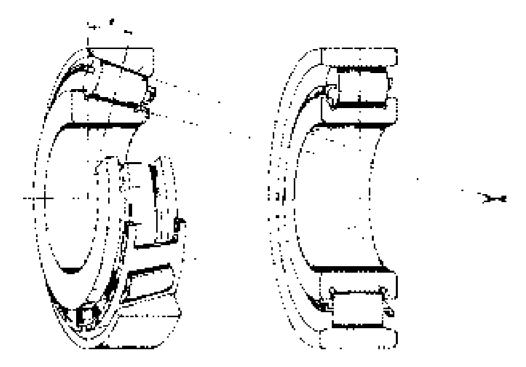**ouble row spherical roller bearing** (Fig. 6.19) Two rows of rollers operate between a double

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Fig. 6.17 Single row taper roller bearing

Fig. 6.16 Single row cylindrical roller bearing
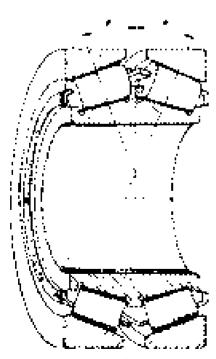

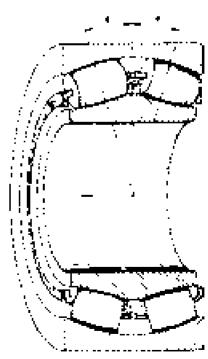**Fig. 6.18** ouble row taper roller bearing

Fig. 6.19 ouble row spherical roller bearing

grooved inner raceway and a common spherically shaped outer raceway ring. With both spherical rollers having the same radii as the outer spherical raceway, line contact area is achieved for both inner and outer tracks. The inner double inclined raceway ring retains the two rows of rollers within their tracks, whereas the outer spherical track will accommodate the rollers even with the inner track ring axis tilted relative to the outer track ring axis. This feature provides the bearing with its self-alignment property so that a large amount of shaft deflection can be tolerated together with its capacity, due to roller to track conformity, to operate with heavy loads in both radial and axial directions. This type of bearing finds favour where both high radial and axial loads are to be supported within the constraints of a degree of shaft misalignment.

Single row thrust ball bearing (Fig. 6.20) These bearings have three load bearing members, two grooved annular disc plates and a ring of balls lodged between them. A no-load-carrying cage fourth member of the bearing has two functions firstly to ease assembly of the bearing when being installed and secondly to evenly space the balls around their grooved tracks. Bearings of this type operate with one raceway plate held stationary while the other one is attached to the rotating shaft.

In comparison to radial ball bearings, thrust ball bearings suffer in operation from an inherent increase in friction due to the balls sliding between the grooved tracks. To minimize the friction, the groove radii are made 6–8% larger than the radii of the balls so that there is a reduction in ball contact area. Another limitation of these bearings is that they do not work very satisfactorily at high rotative speeds since with increased speed the centrifugal force pushes the balls radially outwards, so causing the line of contact, which was originally in the middle of the grooves, to shift further out. This in effect increases ball to track sliding and subsequently the rise in friction generates heat. These bearings must deal purely with thrust loads acting in one direction and they can only tolerate very small shaft misalignment. This type of bearing is used for in ection pump governor linkage axial thrust loads, steering boxes and auxiliary vehicle equipment.

eedle roller bearings (Fig. 6.21) Needle roller bearings are similar to the cylindrical roller bearing but the needle rollers are slender and long and there is no cage (container) to space out the needles around the tracks. The bearing has an inner plain
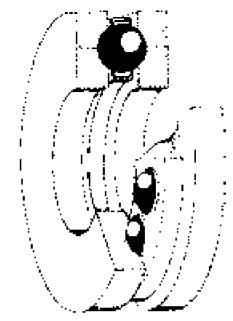
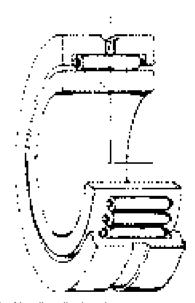**Fig. 6.20** Single row thrust bearing

Fig. 6.21 Needle roller bearing

raceway ring. The outer raceway is shouldered either side to retain the needles and has a circular groove machined on the outside with two or four radial holes to provide a passageway for needle lubrication. The length to diameter ratio for the needles usually lies between 3 to 8 and the needle roller diameter normally ranges from 1.5 to 4.5 mm. Sometimes there is no inner raceway ring and the

needles operate directly upon the shaft. To increase the line contact area, and therefore the load carrying capacity, needles are made relatively long, but this makes the needle sensitive to shaft misalignment which may lead to unequal load distribution along the length of each needle. Another inherent limitation with long needles is that they tend to skew and slide causing friction losses and considerable wear. The space occupied by a complete needle roller bearing is generally no more than that of a hydrodynamic plain ournal bearing. Bearings of this type are well suited for an oscillating or fluctuating load where the needles operate for very short periods before the load or motion reverses, thereby permitting the needles to move back into their original position, parallel to the shaft axis.

Because the needles are not separated from each other, there is a tendency for them to rub together so that friction can be relatively high. To improve lubrication, the needle ends are sometimes tapered or stepped so that oil or grease may be packed between the ends of the needles and the ad acent raceway shoulders.

Needle roller bearings are used for universal oints, gearbox layshafts, first motion shafts, mainshaft constant mesh gear wheel bearings, two stroke heavy duty connecting rod small bearings etc.

ater pump spindle double row ball bearing (Fig. 6.22) In cases where it is necessary to have two bearings spaced apart to support a lightly loaded shaft, such as that used for a water pump and fan, it is sometimes convenient to dispense with the inner raceway ring and mount the balls directly onto the raceway grooves formed on the shaft itself. The space between the shaft and housing is then charged with grease and is fully sealed so that no further attention is required during the working life of the bearings. Such arrangements not only
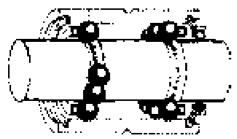
Fig. 6.22 ater pump spindle double row ball bearing
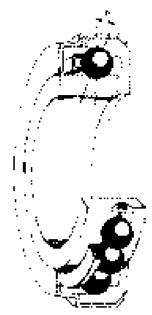
Fig. 6.23 Clutch release thrust ball bearing

reduce the size of the bearings and housing but also reduce the cost of the assembly.

lutch release thrust ball bearing (Fig. 6.23)
Where only pure thrust loads are to be coped with for relatively short periods at high speeds, the use of angular contact ball bearings is generally preferred to the single row parallel disc type ball thrust bearing. This configuration has a deep grooved inner race ring and an angular contact outer race ring with a thrust flange on one side. These inner and outer ball track bearing arrangements do not suffer from high rotative speed effects and will operate over a long period of time under moderate random thrust loading, such as when a clutch release mechanism is engaged and released. Sealing the assembled bearing is a steel cover pressing which is designed to retain the pre-packed grease and to exclude dirt getting to the balls and grooved tracks.

6.1.13 *Structural rigidity and bearing preloading* (Fig. 6.24)
The universal practice of preloading bearing assembly supporting a shaft or hub (Fig. 6.24) raises the rigidity of the bearing assembly so that its deflection under operating conditions is minimized. Insufficient structural rigidity of a shaft or hub assembly may be due to a number of factors which can largely be overcome by preloading the bearings. Bearing
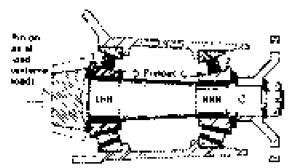
Fig. 6.24 Final drive pinion bearing spring preload analogy

Fig. 6.25 oad-deflection characteristics of several rolling contact bearings (mm inner ring shaft diameter)

preload goes a long way towards compensating for the following inherent side effects which occur during service.

1 The actual elasticity of the roller elements and their respective tracks cause the bearings to deflect both radially and axially in proportion to the applied load and could amount to a considerable increase in shaft movement under working conditions.
2 As the rolling elements become compressed between the inner and outer tracks, the minute surface irregularities tend to deform under the loaded half of the bearing so that the inner raceway ring or cone member centre axis becomes eccentric to that of the outer ring or cup member.
3 If the structure of the housing which contains the bearing is not sufficiently substantial or is made from soft or low strength aluminium alloy, it may yield under heavy loads so that the bearing roller elements become loose in their tracks.
4 Temperature changes may cause the inner or outer track members to become slack in their housings once they have reached operating temperature conditions even though they may have had an interference fit when originally assembled.
5 ver the working life of a bearing the metal to metal contact of the rolling elements and their raceways will planish the rolling surfaces so that bearing slackness may develop.

6.1.1 Bearing selection (Fig. 6.25)
The rigidity of a rolling contact bearing to withstand both radial and axial loads simultaneously is a ma or factor in the type of bearing chosen for a particular application. With straight cut gear teeth, pairs of meshing gears are forced apart due to the leverage action when torque is applied so that radial loads alone are imposed onto the bearings. owever the
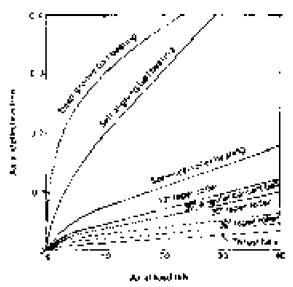
ma ority of transmission gear trains have either helical cut teeth or are bevel gears. In either case, end thrust is generated which must be absorbed by the bearings to prevent the gears separating in an axial direction. Bearings are therefore designed not only to carry radial loads but also to support various amounts of axial thrust. As can be seen in Fig. 6.25 the various types of rolling contact bearings offer a range of axial load–deflection characteristics. The least rigid bearing constructions are the deep groove ball and the self-alignment ball bearings, whereas the roller type bearings, with the exception of the angular contact ball and pure thrust ball bearings, provide considerably more axial stiffness. Furthermore, the ability for taper roller bearings to increase their axial load capacity depends to some extent on the angle of bearing contact. The larger the angle, the greater the axial load carrying capacity for a given axial deflection will be. The radial load–deflection characteristics follow a very similar relationship as the previous axial ones with the exception of the pure thrust ball bearing which cannot support radial loads.

6.1.15 reloading ball and taper roller bearings
An understanding of the significance of bearing preloading may be best visualized by considering a final drive bevel pinion supported between a pair of taper roller bearings (Fig. 6.24). Since the steel of which the rollers, cone and cup are made to obey ooke s
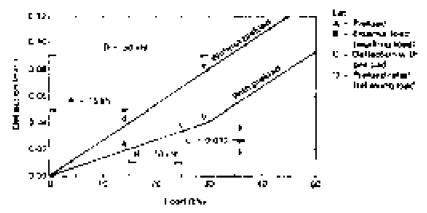
Fig. 6.26 Comparison of bearing load–deflection graph with and without preload

law, whereby strain is directly proportional to the stress producing it within the elastic limit of the material, the whole bearing assembly can be given the spring analogy treatment (Fig. 6.25). The ma or controlling factor for shaft rigidity is then the stiffness (elastic rate) of the bearing which may be defined as the magnitude of the force exerted per unit of distortion,

i.e.   = W (N/m)
where   =   (N/m)
        W =   applied force (N)
          =   deflection (mm)

et the bevel pinion nut be tightened so that each roller bearing is squeezed together axially 0.04 mm when sub ected to a preload of 15 kN. Each bearing will have a stiffness of

$$= W$$
$$= 15 \ 0.04 = 375 \ \text{kN} \ \text{mm}$$

Now if an external force (crownwheel tooth load) exerts an outward axial thrust to the right of magnitude 15 kN, the left hand bearing (1), will be compressed a further 0.04 mm whereas the preloaded right hand bearing (2) will be released 0.04 mm, ust to its unloaded free position. Thus the preloaded assembly has increased its bearing stiffness to $\frac{30}{0.04} = 750$ kN/mm, which is twice the stiffness of the individual bearings before they were preloaded.

Fig. 6.26 shows the relationship between axial load and deflection for bearings with and without preload. With the preloaded bearing assembly, the steepness of the straight line (–a–c–b) for the left hand bearing and shaft is only half of the

unpreloaded assembly (–d–e) and its stiffness is 750 kN/mm (double that of the unpreloaded case).

nce the outer right hand bearing has been relieved of all load, the stiffness of the whole bearing assembly reverts back to the nil preload assembly stiffness of 375 kN/mm. The slope now becomes parallel to the without preload deflection load line. The deflection of the bearing assembly for an external axial force of 10 kN when imposed on the pinion shaft in the direction towards the right hand side can be read off the graph vertically between the preload and working load points (a and c) giving a resultant deflection of 0.012 mm.

For the designer to make full use of bearing preload to raise the rigidity of the pinion shaft bearing assembly, the relieving load (outer bearing ust unloaded) should exceed the working load (external force) applied to the shaft and bearing assembly.

The technique of reducing bearing axial deflection against an applied end thrust so that the bearing assembly in effect becomes stiffer can be appreciated another way by studying Fig. 6.27.

Suppose a pair of taper bearings (Fig. 6.24) are sub ected to a preload of 15 kN. The corresponding axial deflection will be 0.04 mm according to the linear deflection–load relationship shown. If an external axial load of 10 kN is now applied to the pinion shaft so that it pushes the shaft towards the bearings, the load on the left hand bearing (1) will instantaneously increase to 25 kN. The increased deflection of bearing (1) accompanying the increase in load will cause the right hand bearing (2) to lose some of its preload and hence some of its deflection. Simultaneously the change of preload on bearing (2) will influence the load acting on bearing (1) and hence the deflection of this bearing. nce
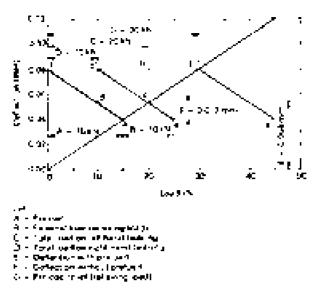
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Fig. 6.27 Bearing load–deflection graph using inverted preload curve to obtain working load deflection

equilibrium between the two bearings has been established, the rollers of bearing (1) will support a load of less than 25 kN and those of bearing (2) will carry a load of less than 15 kN.

The distribution of the applied load between the two bearings at equilibrium may be determined by inverting the deflection–load curve from zero to pre-load deflection, a–b. This represents the reduction in preload of the right hand bearing (2) as the external axial load forces the pinion shaft towards the left hand bearing. The inverted right hand bearing pre-load curve can be shifted horizontally from its original position a–b where it intersects the full curve at point a to a new position a –b , the distance a to a being equal to the external load applied to the pinion shaft (working load). Note that point a is the instant-aneous load on bearing (1) before equilibrium is established. The intersection of the shifted inverted curve with the full curve point c represents the point of equilibrium for bearing (1), the total left hand bearing load. The axial deflection of the pinion shaft under the applied load of 10 kN is thus equal to the vertical reading between point a and c, that is 0.012 mm. The equilibrium point for bearing (2) can be found by drawing a horizontal line from point c to intersect at point d on the original inverted curve, so that point (d) becomes the total right hand bearing load.

6.1.16 elationship bet een bearing tightness and life e pectancy (Fig. 6.28)
Taper roller and angular contact ball bearing life is considerably influenced by the slackness or tightness
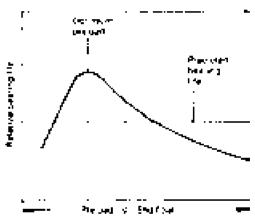
Fig. 6.28 Effect of bearing tightness or slackness on life expectancy

to which the bearings are originally set (Fig. 6.28). The graph shows that if the bearings are heavily preloaded the excessive elastic distortion, and possibly the breakdown in lubrication, will cause the bearings to wear rapidly. ikewise excessive end float causes roller to track misalignment and end to end shock loading with much reduced ser-vice life. owever it has been found that a small degree of bearing preload which has taken up all the free play when stationary loosens off under working conditions so that the rollers will have light positive contact with their tracks. This results in pure rolling and hence optimum bearing life.

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.2 The need for constant velocity oints

Universal oints are necessary to transmit torque and rotational motion from one shaft to another when their axes do not align but intersect at some point. This means that both shafts are inclined to each other by some angle which under working conditions may be constantly varying.

Universal oints are incorporated as part of a vehicle s transmission drive to enable power to be transferred from a sprung gearbox or final drive to the unsprung axle or road wheel stub shaft.

There are three basic drive applications for the universal oint:

1 propellor shaft end oints between longitudinally front mounted gearbox and rear final drive axle,
2 rear axle drive shaft end oints between the sprung final drive and the unsprung rear wheel stub axle,
3 front axle drive shaft end oints between the sprung front mounted final drive and the unsprung front wheel steered stub axle.

Universal oints used for longitudinally mounted propellor shafts and transverse rear mounted drive shafts have movement only in the vertical plane. The front outer drive shaft universal oint has to cope with movement in both the vertical and horizontal plane it must accommodate both vertical suspension deflection and the swivel pin angular movement to steer the front road wheels.

The compounding of angular working movement of the outer drive shaft steering oint in two planes imposes abnormally large and varying working angles at the same time as torque is being transmitted to the stub axle. Because of the severe working conditions these oints are sub ected to special universal oints known as

These have been designed and developed to eliminate torque and speed fluctuations and to operate reliability with very little noise and wear and to have a long life expectancy.

6. .1 oo e s universal oint (Figs 6.29 and 6.30)

The ooke s universal oint comprises two yoke arm members, each pair of arms being positioned at right angles to the other and linked together by an intermediate cross-pin member known as the spider. When assembled, pairs of cross-pin legs are supported in needle roller caps mounted in each yoke arm, this then permits each yoke member to swing at right angles to the other.

Because pairs of yoke arms from one member are situated in between arms of the other member, there will be four extreme positions for every revolution when the angular movement is taken entirely by only half of the oint. As a result, the spider cross-pins tilt back and forth between these extremes so that if the drive shaft speed is steady throughout every complete turn, the drive shaft will accelerate and decelerate twice during one revolution, the magnitude of speed variation becoming larger as the drive to driven shaft angularity is increased.

ooke s oint speed fluctuation may be better understood by considering Fig. 6.29. This shows the drive shaft horizontal and the driven shaft inclined downward. At zero degree movement the input yoke cross-pin axis is horizontal when the drive shaft and the output yoke cross-pin axis are vertical. In this position the output shaft is at a minimum. Conversely, when the input shaft has rotated a further 90 , the input and output yokes and cross-pins will be in the vertical and horizontal position respectively. This produces a maximum output shaft speed. A further quarter of a turn will move the oint to an identical position as the initial position so that the output speed will be again at a minimum. Thus it can be seen that the cycle of events repeat themselves every half revolution.

Table 6.2 shows how the magnitude of the speed fluctuation varies with the angularity of the drive to driven shafts.

ab **6.2** ariation of shaft angle with speed fluctuation

Shaft angle (deg)	1	1	2	2	
speed fluctuation			12	1	2	1

The consequences of only having a single ooke s universal oint in the transmission line can be appreciated if the universal oint is considered as the link between the rotating engine and the vehicle in motion, moving steadily on the road. Imagine the engine s revolving inertia masses rotating at some constant speed and the vehicle itself travelling along uniformly. Any cyclic speed variation caused by the angularity of the input and output shafts will produce a correspondingly periodic driving torque fluctuation. As a result of this torque variation, there will be a tendency to wind and unwind the drive in proportion to the working angle of the oint, thereby imposing severe stresses upon the transmission system. This has been found to produce uneven wear on the driving tyres.

To eliminate torsional shaft cyclic peak stresses and wind-up, universal oints which rotate uniformly during each revolution become a necessity.
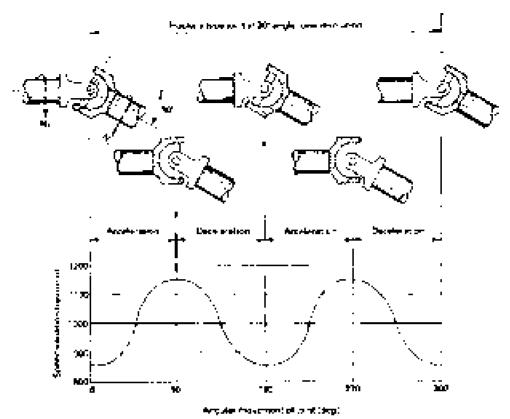
Fig. 6.29 Hooke s oint cycle of speed fluctuation for shaft angularity

6. . oo e s oint cyclic speed variation due to drive to driven shaft inclination (Fig. 6.30)

Consider the ooke s oint shown in Fig. 6.30(a) with the input and output yokes in the horizontal and vertical position respectively and the output shaft inclined Θ degrees to the input shaft.

et ω_i = input shaft angular velocity (rad/sec)
ω_o = output shaft angular velocity (rad/sec)
Θ = shaft inclination (deg)
R = pitch circle oint radius (mm)

Then
  inear velocity of point ( ) = $\omega_i R$
and
  inear velocity of point ( ) = $\omega_o r$.

Since these velocities are equal,

$$\omega_o r = \omega_i R$$

$$\therefore \quad \omega_o = \omega_i \frac{R}{r}$$

but $\frac{r}{R} = \cos\Theta$.

Thus $\quad \omega_o = \omega_i \cos\Theta$

but $\quad \omega_i = \frac{2\pi N_i}{60}$.

So $\frac{2\pi}{60} N_o = \frac{2\pi}{60} N_i \cos\Theta$

ence $\quad N_o = N_i \cos\Theta$ (this being a minimum) (1).

If now the oint is rotated a quarter of a revolution (Fig. 6.30(b)) the input and output yoke positions will be vertical and horizontal respectively.

Then
  inear velocity of point ( ) = $\omega_o R$
also
  inear velocity of point ( ) = $\omega_i r$.

Since these velocities are equal,

$$\omega_o R = \omega_i r$$

$$\omega_o = \omega_i \frac{r}{R}$$

but $\frac{R}{r} = \frac{1}{\cos\Theta}$.

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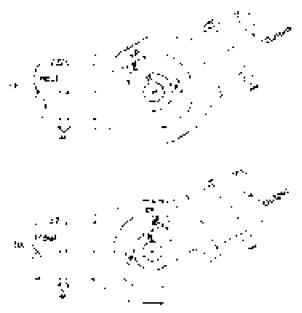
Fig. 6.30 (a and b) Hooke s oint geometry

Thus
$$\omega_o = \frac{\omega_i}{\cos \Theta}$$

$$\frac{2\pi}{60} \omega_o = \frac{2\pi}{60} \frac{\omega_i}{\cos \Theta}$$

$$\omega_o = \frac{\omega_i}{\cos \Theta} \text{(this being a maximum)} \quad (2)$$

1 When $\omega = \phi$ the angular instantaneous velocities will be equal.
2 When ϕ is smaller than ω the output instantaneous velocity will be less than the output.
3 When ϕ is larger than ω the output instantaneous velocity will be greater than the input.

A ooke s universal oint connects two shafts which are inclined at 30 to each other. If the driving shaft speed is 500 rev/min, determine the maximum and minimum speeds of the driven shaft.

Minimum speed $\quad \omega_o = \omega_i \cos 30$
$$= 500 \times 0.866$$
$$= 433 \text{ (rev min)}$$

Maximum speed $\quad \omega_o = \frac{\omega_i}{\cos 30}$
$$= \frac{500}{0.866}$$
$$= 577 \text{ (rev min)}$$

A   ooke s universal oint connects two shafts which are inclined at some angle. If the input and output oint speeds are 500 and 450 rev/min respectively, find the angle of inclination of the output shaft.

$$\omega_o = \omega_i \cos \Theta$$

$$\cos \Theta = \frac{\omega_o}{\omega_i}$$

ence $\quad \cos \Theta = \frac{450}{500} = 0.9$

Therefore $\Theta = 25°50$

6. .3 Constant velocity oints
Constant velocity oints imply that when two shafts are inclined at some angle to one another and they are coupled together by some sort of oint, then a uniform input speed transmitted to the output shaft produces the same angular output speed throughout one revolution. There will be no angular acceleration and deceleration as the shafts rotate.

6. . Double oo e s type constant velocity oint
(Figs 6.31 and 6.32)
ne approach to achieve very near constant velocity characteristics is obtained by placing two ooke s type oint yoke members back to back with their yoke arms in line with one another (Fig. 6.31). When assembled, both pairs of outer yoke arms will be at right angles to the arms of the central double yoke member. Treating this double oint combination in two stages, the first stage hinges the drive yoke and driven central double yoke together, whereas the second stage links the central double yoke (now drive member) to the driven final output yoke. Therefore the second stage drive half of the central double yoke is positioned a quarter of a revolution out of phase with the first stage drive yoke (Fig. 6.32).

Consequently when the input and output shafts are inclined to each other and the first stage driven central double yoke is speeding up, the second stage driven output yoke will be slowing down. Conversely when the first stage driven member is reducing speed the second stage driven member increases its speed the speed lost or gained by one half of the oint will equal that gained or lost by the second half of the oint respectively. There will therefore be no cyclic speed fluctuation between input and output shafts during rotation.

An additional essential feature of this double oint is a centring device (Fig. 6.31) normally of the ball and socket spring loaded type. Its function is to maintain equal angularity of both the input and
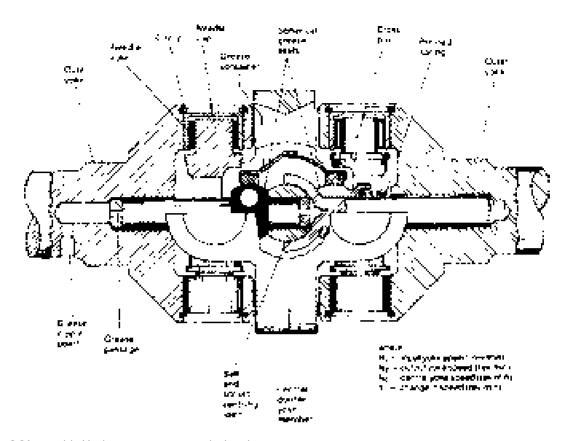
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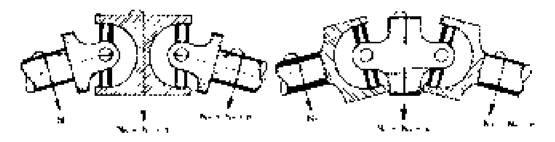**Fig. 6.31** Double Hooke's type constant velocity joint

Fig. 6.32 Double Hooke's type joint shown in two positions out of phase

output shafts relative to the central double yoke member. This is a difficult task due to the high end loads imposed on the sliding splined joint of the drive shaft when repeated suspension deflection and large drive torques are being transmitted simultaneously.

However, the accuracy of centralizing the double yokes is not critical at the normal relatively low drive shaft speeds.

This double Hooke's joint is particularly suitable for heavy duty rigid front wheel drive live axle vehicles where large lock-to-lock wheel swivel is necessary. A major limitation with this type of joint is its relatively large size for its torque transmitting capacity.

6.7.5 Birfield joint based on the Rzeppa principle (Fig. 6.33)

Alfred Hans Rzeppa (pronounced), a Ford engineer in 1926, invented one of the first practical

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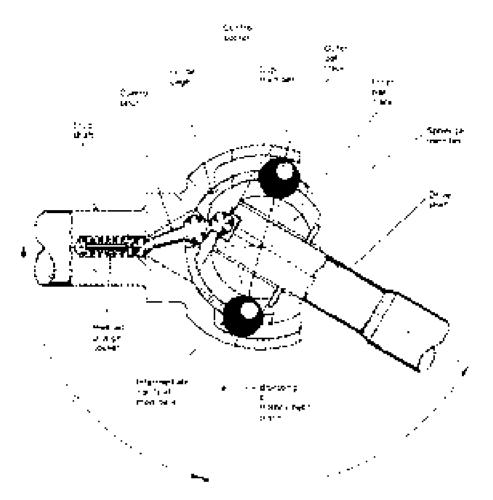
Fig. 6.33 Early R eppa constant velocity oint

constant velocity oints which was able to transmit torque over a wide range of angles without there being any variation in the rotary motion of the output shafts. An improved version was patented by Rzeppa in 1935. This oint used six balls as intermediate members which where kept at all times in a plane which bisects the angle between the input and output shafts (Fig. 6.33). This early design of a constant velocity oint incorporated a controlled guide ball cage which maintained the balls in the bisecting plane (referred to as the median plane) by means of a pivoting control strut which swivelled the cage at an angle of exactly half that made between the driving and driven shafts. This control strut was located in the centre of the enclosed end of the outer cup member, both ball ends of the strut being located in a recess and socket formed in the ad acent ends of the driving

and driven members of the oint respectively. A large spherical waist approximately midway along the strut aligned with a hole made in the centre of the cage. Any angular inclination of the two shafts at any instant deflected the strut which in turn proportionally swivelled the control ball cage at half the relative angular movement of both shafts. This method of cage control tended to am and suffered from mechanical wear.

oint construction (Fig. 6.34) The Birfield oint, based on the Rzeppa principle and manufactured by ardy Spicer imited, has further developed and improved the oint s performance by generating converging ball tracks which do not rely on a controlled ball cage to maintain the intermediate ball members on the median plane (Fig. 6.34(b)). This

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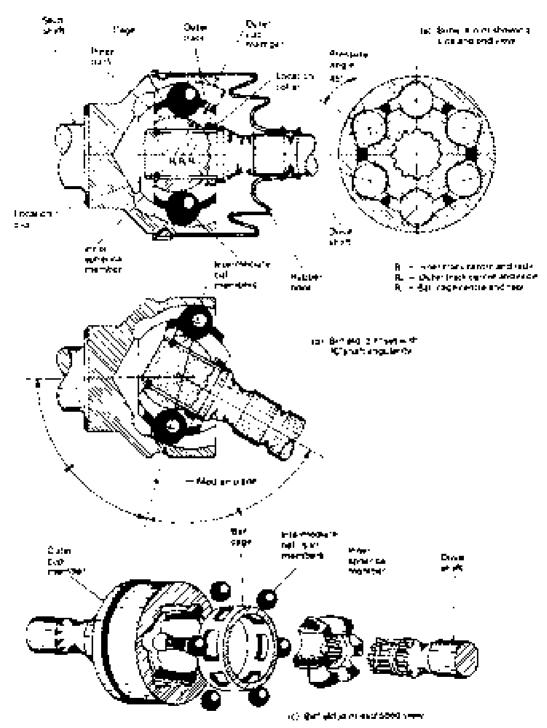
Fig. 6.34 (a–c) Birfield Rzeppa type constant velocity joint

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oint has an inner (ball) input member driving an outer (cup) member. Torque is transmitted from the input to the output member again by six intermediate ball members which fit into curved track grooves formed in both the cup and spherical members. Articulation of the oint is made possible by the balls rolling inbetween the inner and outer pairs of curved grooves.

all track convergence (Figs 6.34 and 6.35) Constant velocity conditions are achieved if the points of contact of both halves of the oint lie in a plane which bisects the driving and driven shaft angle, this being known as the median plane (Fig. 6.34(b)). These conditions are fulfilled by having an intermediate member formed by a ring of six balls which are kept in the median plane by the shape of the curved ball tracks generated in both the input and output oint members.

To obtain a suitable track curvature in both half, inner and outer members so that a controlled movement of the intermediate balls is achieved, the tracks (grooves) are generated on semicircles. The centres are on either side of the oint s geometric centre by an equal amount (Figs 6.34(a) and 6.35). The outer half cup member of the oint has the centre of the semicircle tracks offset from the centre of the oint along the centre axis towards the open mouth of the cup member, whilst the inner half spherical member has the centre of the semicircle track offset an equal amount in the opposite direction towards the closed end of the oint (Fig. 6.35).

When the inner member is aligned inside the outer one, the six matching pairs of tracks form grooved tunnels in which the balls are sandwiched.

The inner and outer track arc offset centre from the geometric oint centre are so chosen to give an angle of convergence (Fig. 6.35) marginally larger than 11 , which is the minimum amount necessary to positively guide and keep the balls on the median plane over the entire angular inclination movement of the oint.

Track groove profile (Fig. 6.36) The ball tracks in the inner and outer members are not a single semicircle arc having one centre of curvature but instead are slightly elliptical in section, having effectively two centres of curvature (Fig. 6.36). The radius of curvature of the tracks on each side of the ball at the four pressure angle contact points is larger than the ball radius and is so chosen so that track contact occurs well within the arc grooves, so that groove edge overloading is eliminated. At the same time the ball contact load is taken about one third below and above the top and bottom ball tips so that compressive loading of the balls is considerably reduced. The pressure angle will be equal in the inner and outer tracks and therefore the balls are all under pure compression at all times which raises the limiting stress and therefore loading capacity of the balls.

The ratio of track curvature radius to the ball radius, known as the   is selected so that a 45 pressure angle point contact is achieved, which has proven to be effective and durable in transmitting the torque from the driving to the driven half members of the oint (Fig. 6.36).

As with any ball drive, there is a certain amount of roll and slide as the balls move under load to and fro along their respective tracks. By having a pressure angle of 45 , the roll to sliding ratio is roughly 4:1.
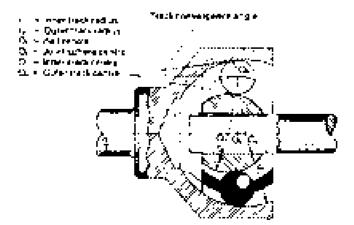

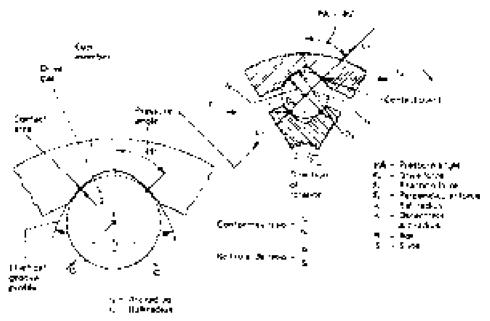**Fig. 6.35** Birfield R eppa type oint showing ball track convergence

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Fig. 6.36 Birfield joint rack groove profile

This is sufficient to minimize the contact friction during any angular movement of the joint.

Ball cage (Fig. 6.34(b and c)) Both the inner drive and outer driven members of the joint have spherical external and internal surfaces respectively. Likewise, the six ball intermediate members of this joint are positioned in their respective tracks by a cage which has the same centre of arc curvature as the input and output members (Fig. 6.34(c)). The cage takes up the space between the spherical surfaces of both male inner and female outer members. It provides the central pivot alignment for the two halves of the joint when the input and output shafts are inclined to each other (Fig. 6.34(b)).

Although the individual balls are theoretically guided by the grooved tracks formed on the surfaces of the inner and outer members, the overall alignment of all the balls on the median plane is provided by the cage. Thus if one ball or more tends not to position itself or themselves on the bisecting plane between the two sets of grooves, the cage will automatically nudge the balls into alignment.

Mechanical efficiency The efficiency of these joints is high, ranging from 100% when the joint working angle is zero to about 95% with a 45° joint working angle. Losses are caused mainly by internal friction between the balls and their respective tracks, which is affected by ball load, speed and working angle and by the viscous drag of the lubricant, the latter being dependent to some extent by the properties of the lubricant chosen.

Fault diagnoses Symptoms of front wheel drive constant velocity joint wear or damage can be narrowed down by turning the steering to full lock and driving round in a circle. If the steering or transmission now shows signs of excessive vibration or clunking and ticking noises can be heard coming from the drive wheels, further investigation of the front wheel joints should be made. Split rubber gaiters protecting the constant velocity joints can considerably shorten the life of a joint due to exposure to the weather and abrasive grit finding its way into the joint mechanism.

6.7.6 Rot type constant velocity joint (Fig. 6.37)
This joint manufactured by both the Birfield and Bendix companies has been designed to provide a solution to the problem of transmitting torque with varying angularity of the shafts at the same time as accommodating axial movement.

There are four basic parts to this joint which make it possible to have both constant velocity characteristics and to provide axial plunge so that the effective drive shaft length is able to vary as the angularity alters (Fig. 6.37):
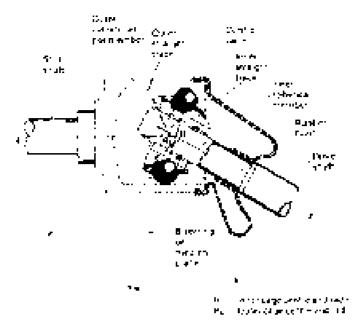
Fig. 6.37 Birfield Rzeppa pot type joint

1 A pot input member which is of cylindrical shape forms an integral part of the final drive stub shaft and inside this pot are ground six parallel ball grooves.
2 A spherical (ball) output member is attached by splines to the drive shaft and ground on the external surface of this sphere are six matching straight tracked ball grooves.
3 Transmitting the drive from the input to the output members are six intermediate balls which are lodged between the internal and external grooves of both pot and sphere.
4 A semispherical steel cage positions the balls on a common plane and acts as the mechanism for automatically bisecting the angle between the drive and driven shafts (Fig. 6.38).

It is claimed that with straight cut internal and external ball grooves and a spherical ball cage which is positioned over the spherical (ball) output member that a truly homokinetic (bisecting) plane is achieved at all times. The joint is designed to have a maximum operating angularity of 22°, 44° including the angle, which makes it suitable for independent suspension inner drive shaft joints.

6.3.7 Carl Weiss constant velocity joint
(Figs 6.38 and 6.40)
A successful constant velocity joint was initially invented by Carl W. Weiss of New York, USA,

and was patented in 1925. The Bendix Products Corporation then adopted the Weiss constant velocity principle, developed it and now manufacture this design of joint (Fig. 6.38).

Joint construction and description With this type of time constant velocity joint, double prong (arm) yokes are mounted on the ends of the two shafts transmitting the drive (Fig. 6.37). Ground inside each prong member are four either curved or straight ball track grooves (Fig. 6.39). Each yoke arm of one member is assembled inbetween the prong of the other member and four balls located in adjacent grooved tracks transmit the drive from one yoke member to the other. The intersection of each matching pair of grooves maintains the balls in a bisecting plane created between the two shafts, even when one shaft is inclined to the other (Fig. 6.40). Depending upon application, some joint models have a fifth centralizing ball inbetween the two yokes while the other versions, usually with straight ball tracks, do not have the central ball so that the joint can accommodate a degree of axial plunge, especially if, as is claimed, the balls roll rather than slide.

Carl Weiss constant velocity principle (Fig. 6.41) Consider the geometric construction of the upper half of the joint (Fig. 6.41) with ball track
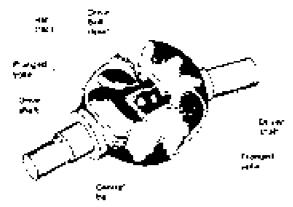
Fig. 6.38 Pictorial view of Bendix Weiss constant velocity type joint

curvatures on the left and right hand yokes to be represented by circular arcs with radii (r) and centres of curvature L and R on their respective shaft axes when both shafts are in line. The centre of the joint is marked by point O and the intersection of both the ball track arc centres occurs at point P. Triangle LOP equals triangle ROP with sides LP and R P being equal to the radius of curvature. The offset of the centres of track curvature from the joint centre are OL and OR, therefore sides LP and R P are also equal. Now, angles OLP and

R P are two right angles and their sum of 90 90 is equal to the angle R, that is 180, so that point P lies on a perpendicular plane which intersects the centre of the joint. This plane is known as the or

If the right hand shaft is now swivelled to a working angle its new centre of track curvature will be R and the intersection point of both yoke ball track curvatures is now P (Fig. 6.41). Therefore triangle P and R P are equal because both share the same bisecting plane of the left and right hand shafts. Thus it can be seen that sides P and R P are also equal to the track radius of curvature r and that the offset of the centres of R and R are equal to . Consequently, angle P equals angle R P and the sum of the angles P and R P equals angle R of 180 – Θ. It therefore follows that angle P equals angle R P which is (180 – Θ) 2. Since P bisects the angle made between the left and right hand shaft axes it must lie on the median (homokinetic) plane.

The ball track curvature intersecting point line projected to the centre of the joint will always be half the working angle Θ made between the two shaft axes and fixes the position of the driving balls. The geometry of the intersecting circular arcs therefore constrains the balls at any instant to be in the median (homokinetic) plane.
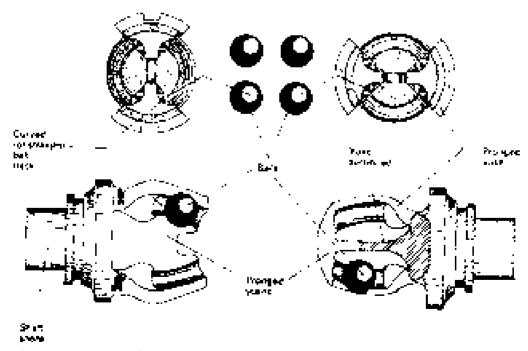
Fig. 6.39 Side and end views of Carl Weiss type joint
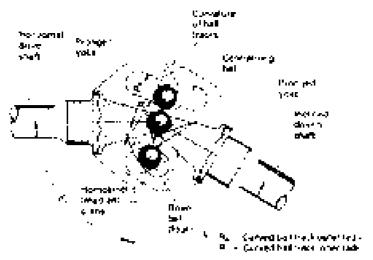
Fig. 6.40 Principle of Bendix Weiss constant velocity type joint
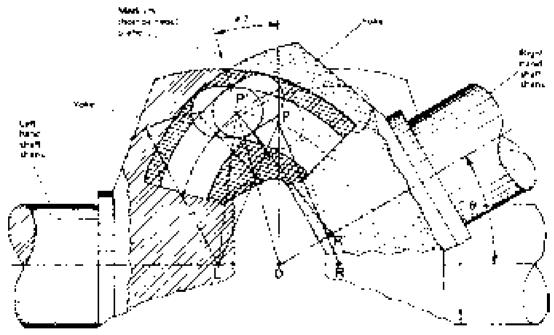

Fig. 6.41 Geometry of Carl Weiss type joint

6.7. *Tracta constant velocity joint* (Fig. 6.42)
The tracta constant velocity joint was invented by
Fennille in France and was later manufactured in
England by Bendix Ltd.

With this type of joint there are four main components: two outer yoke jaw members and two
intermediate semispherical members (Fig. 6.41(a)).
Each yoke jaw engages a circular groove machined

on the intermediate members. In turn both intermediate members are coupled together by a swivel
tongue (spigot joint) and grooved ball (slotted joint).

In some ways these joints are very similar in
action to a double Hooke's type constant velocity
joint.

Relative motion between the outer jaw yokes
and the intermediate spherical members is via the

yoke jaw fitting into circular grooves formed in each intermediate member. Relative movement between adjacent intermediate members is provided by a double tongue formed on one member slotting into a second circular groove and cut at right angles to the jaw grooves (Fig. 6.42(b)).

When assembled, both the outer yoke jaws are in alignment, but the central tongue and groove part of the joint will be at right angles to them (Fig. 6.43 (a and b)). If the input and output shafts are inclined at some working angle to each other, the driving intermediate member will accelerate and decelerate during each revolution. Owing to the central tongue and groove joint being a quarter of a revolution out of phase with the yoke jaws, the corresponding speed fluctuation of the driven intermediate and output jaw members exactly counteract and neutralize the input half member's speed variation. Thus the output speed changes will be identical to that of the input drive.

Relative motion between members of this type of joint is not of a rolling nature but one of sliding. Therefore friction losses will be slightly higher than for couplings which incorporate intermediate ball members, but the large flat rubbing surfaces in contact enables large torque loads to be transmitted. The size of these joints are fairly large compared to other types of constant velocity joint

arrangements but it is claimed that these joints provide constant velocity rotation at angles up to 50°. A tracta joint incorporated in a rigid front wheel drive axle is shown in Fig. 6.42(c and d).

6.9. Tripot universal joint (Fig. 6.43)
Instead of having six or four ball constant velocity joints, a low cost semi-constant velocity joint providing axial movement and having only three bearing contact points has been developed. This joint is used at the inner final drive end of a driving shaft of independent suspension as it not only accommodates continuous variations in shaft working angles, but also longitudinal length changes both caused by road wheel suspension vertical flexing.

One version of the tripot joint incorporates a three legged spider (tripole) mounted on a splined hub which sits on one end of the drive shaft (Fig. 6.43(a and b)). Each of the spider legs supports a semispherical roller mounted on needle bearings. The final drive stub shaft is integral with the pot housing and inside of this pot are ground roller track grooves into which the tripole rollers are lodged.

In operation, the stub shaft and pot transfers the drive via the grooves, rollers and spider to the output drive shaft.
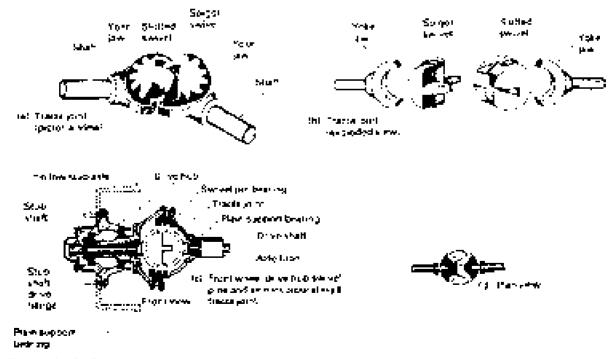

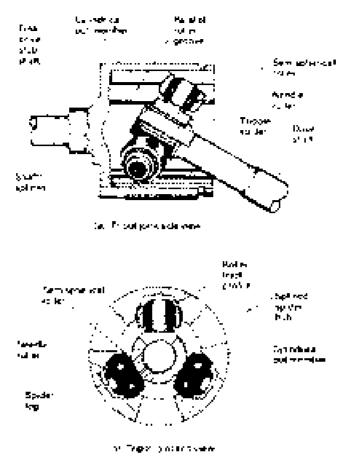**Fig. 6.42 (a–d)** Bendix tracta joint

Fig. 6.43 (a and b) Tripot type universal joint

When there is angularity between the final drive stub shaft and drive shaft, the driven shaft and spider will rotate on an inclined axis which intersects the stub shaft axis at some point. If the motion of one roller is followed (Fig. 6.43(a)), it will be seen that when the driven shaft is inclined downwards, when one spider leg is in its lowest position, its rollers will have moved inwards towards the blind end of the pot, but as the spider leg rotates a further 180 and approaches its highest position the roller will have now moved outwards towards the mouth of the pot. Thus as the spider revolves each roller will roll to and fro in its deep groove track within the pot. At the same time that the rollers move along their grooves, the rollers also slide radially back and forth over the needle bearings to take up the extended roller distance from the centre of rotation as the angularity between the shafts becomes greater and vice versa as the angle between the shafts decreases.

Because the rollers are attached to the driven shaft through the rigid spider, the point of contact between the three rollers and their corresponding grooves do not produce a plane which bisects the angle between the driving and driven shafts. Therefore this coupling is not a true constant velocity joint.

6. .10 Tripronged universal joint (Fig. 6.44)
Another version of the three point contact universal joint consists of a triple prong input member (Fig. 6.45(b)) forming an integral part of the drive shaft and an output stub axle cup member inside which a tripole spider is located Fig. 6.44(a). Three holes are drilled in the circumference of the cup member to accommodate the ends of the spider legs, these being rigidly attached by welds (Fig. 6.44(a and c)). Mounted over each leg is a roller spherical ring which is free to both revolve and slide.
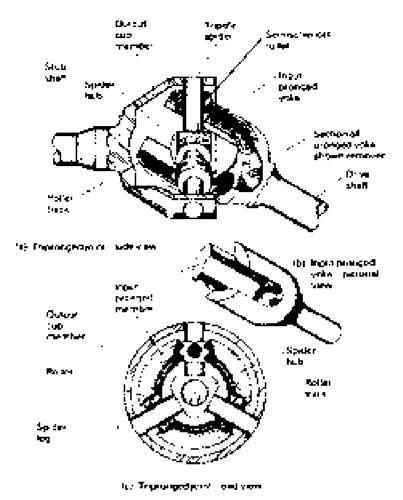
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Fig. 6.44 (a–c) Tripronged type universal joint

When assembled, the input member prongs are located in between adjacent spider legs and the roller aligns the drive and driven joint members by lodging them in the grooved tracks machined on each side of the three projecting prongs (Fig. 6.44(c)).

The input driveshaft and pronged member imparts driving torque through the rollers and spider to the output cup and stub axle member. If there is an angle between the drive and driven shafts, then the input drive shaft will swivel according to the angularity of the shafts. Assuming that the drive shaft is inclined downwards (Fig. 6.44(a)), then the prongs in their highest position will have moved furthest out from the engaging roller, but the rollers in their lowest position will be in their deepest position along the supporting tracks of the input member.

As the shaft rotates, each roller supported and restrained by adjacent prong tracks will move radially back and forth along their respective legs to accommodate the orbiting path made by the rollers about the output stub axle axis. Because the distance of each roller from the centre of rotation varies from a maximum to a minimum during one revolution, each spider leg will produce an acceleration and deceleration over the same period.

This type of joint does not provide true constant velocity characteristics with shaft angularity since the roller plane does not exactly bisect the angle made between the drive and driven shaft, but the joint is tolerant to longitudinal plunge of the drive shaft.

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7   inal drive transmission

7.1   rownwheel and pinion a le ad ustments

The setting up procedure for the final drive crown-
wheel and pinion is explained in the following
sequence:

1 Remove differential assembly with shim pre-
loaded bearings.
2 Set pinion depth.
3 Ad ust pinion bearing preload.
  a) Set pinion bearing preloading using spacer
     shims.
  b) Set pinion bearing preloading using collaps-
     ible spacer.
4 Ad ust crownwheel and pinion backlash and dif-
  ferential bearing preload.
  a) Set differential cage bearing preload using
     shims.
  b) Set crownwheel and pinion backlash using
     shims.
  c) Set crownwheel and pinion and preloading
     differential bearing using ad usting nuts.
5 Check crownwheel and pinion tooth contact.

7.1.1   emoving differential assembly   ith shim preloaded bearings (Fig. 7.1)

Before removing the differential assembly from the
final drive housing, the housing must be expanded
to relieve the differential cage bearing preload.
Spreading the housing is achieved by assembling
the housing stretcher plates (Fig. 7.1) to the hous-
ing, taking up the turnbuckle slack until it is hand
tight and tightening the turnbuckle with a spanner
by three to four flats of the hexagonal until the
differential cage bearing end thrust is removed.
Never stretch the housing more than 0.2 mm,
otherwise the distortion may become permanent.
The differential cage assembly can then be with-
drawn by levering out the unit.

7.1.   Setting pinion depth (Fig. 7.2)

Press the inner and outer pinion bearing cups into
the differential housing and then lubricate both
bearings. Slip the standard pinion head spacer
(thick shim washer) and the larger inner bearing
over the dummy pinion and align assembly into the
pinion housing (Fig. 7.2). Slide the other bearing
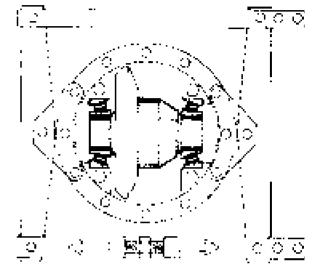and centralizing cone handle over the pinion shank,

Fig. 7.1   Stretching axle housing to remove differential
cage assembly

then screw on the preloading sleeve.   old the han-
dle of the dummy pinion while winding round the
preload sleeve nut until the sleeve is screwed down
to the first mark for re-used bearings or second for
new bearings. Rotate dummy pinion several times
to ensure bearings seat properly. Check the bearing
preload by placing a preload gauge over the pre-
load sleeve nut and read off the torque required to
rotate the dummy pinion. (A typical preload torque
would be 2.0–2.4 Nm.)

Place the stepped gauge block and dial indicator
magnetic stand onto the surface plate then swing
the indicator spindle over selected gauge step and
zero indicator gauge.

Clean the driving pinion head and place the
magnetic dial gauge stand on top of the pinion
head. Move the indicator arm until the spindle of
the gauge rests on the centre of one of the differ-
ential bearing housing bores. Slightly swing the
gauge across the bearing housing bore until the
minimum reading at the bottom of the bore is
obtained. Repeat the check for the opposite bear-
ing bore. Add the two readings together and divide
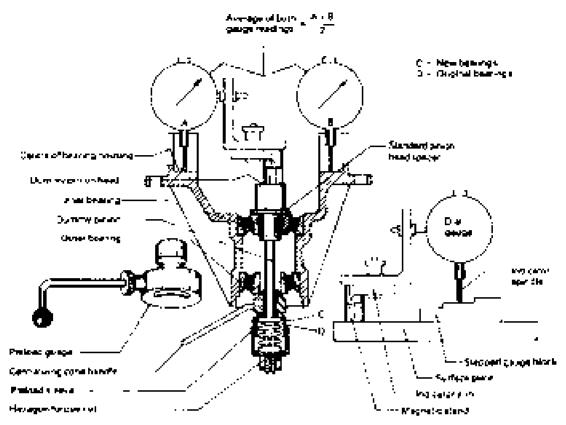by two to obtain a mean reading. This is the

Fig. 7.2 Setting pinion depth dummy pinion

Etched on the pinion head is either the letter N (normal) or a number with either a positive or negative sign in front which provides a correction factor for deviations from the normal size within the production tolerance for the pinion cone distance.

If the etched marking on the pinion face is N (normal), there should be no change in pinion head washer thickness.

If the etched marking on the pinion face is positve ( ) (pinion head height oversize), reduce the size of the required pinion head washer by the amount marked.

If the etched marking on the pinion face is negative ( ) (pinion head height undersize), increase size of the required pinion head washer by the amount marked.

The numbers range between 5 and 30 (units are hundredths of millimetres). So, 20 means subtracting 20/100 mm, i.e. 0.2 mm subtracted from pinion head washer thickness, or 5 means adding 5/100 mm, i.e. 0.05 mm added to pinion head washer thickness.

For example,

Average clock bearing bore reading	=	0.05 mm
Pinion head standard washer thickness	=	1.99 mm
Pinion cone distance correction factor	=	0.12 mm
Required pinion head washer thickness	=	2.12 mm

7.1.3 Ad usting pinion bearing preload

Setting pinion bearing preload using spacer shims (Fig. 7.3(a)) Slip the correct pinion head washer over pinion shank and then press on the inner bearing cone. il the bearing and fit the pinion assembly into the housing. Slide on the bearing spacer with the small end towards the drive flange. Fit the old preload shim next to the spacer, oil and fit outer

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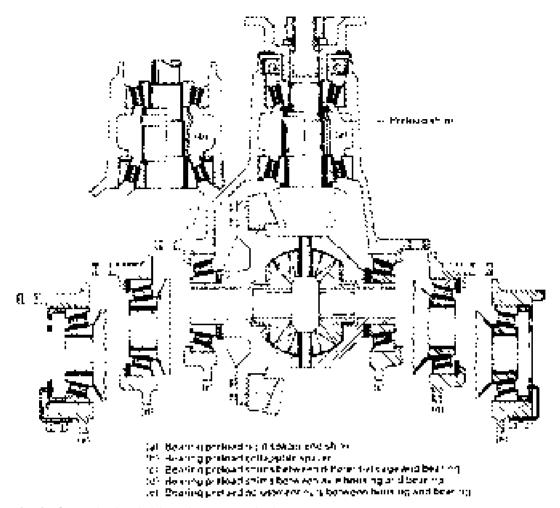
Fig. 7.3 (a–) Crownwheel and pinion adjustment methods

bearing to pinion shank. Assemble the pinion drive flange, washer and nut (Fig. 7.3(a)).

Using a torque wrench gradually tighten the nut to the correct torque (about 100–130 Nm). Rotate the pinion several times so that the bearings settle to their running conditions and then check the preload resistance using the preload gauge attached to the pinion nut or drive flange. Typical bearing preload torque ranges 15–25 Nm. If necessary, increase or decrease the spacer shim thickness to keep within the specified preload.

If the preload is high, increase spacer shim thickness. Alternatively if the preload is low, decrease the shim thickness. Note that 0.05 mm shim thickness is approximately equivalent to 0.9 Nm pinion preload torque.

To alter pinion preload, remove pinion nut, flange, washer and pinion outer bearing. If pre-

loading is high, add to the original spacer shim thickness, but if preload is too low, remove original shim and fit a thinner one.

nce the correct pinion preload has been obtained, remove pinion nut, washer and drive flange. Fit a new oil seal and finally reassemble. Retighten drive flange nut to the fully tight setting (i.e. 120 Nm) if a castlenut is used instead of a self-locking nut fit split pin.

Setting pinion bearing preload using collapsible spacer (Fig. 7.3(b)) Fit the selected pinion head spacer washer to the pinion and press on inner pinion bearing cone. Press both pinion bearing cones into housing. Fit the outer bearing cone to its cup in the pinion housing and locate a new oil seal in the housing throat with the lip towards the bearing

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and press it in until it contacts the inner shoulder. ightly oil the seal.

Install the pinion into the final drive housing with a new collapsible spacer (Fig. 7.3(b)). Fit the drive flange and a new retainer nut. Tighten the nut until a slight end float can be felt on the pinion.

Attach the pinion preload gauge to the drive flange and measure the oil seal drag (usually around 0.6 Nm). To this oil sealed preload drag add the bearing preload torque of 2.2.–3.0 Nm.

i.e. Total preload = il seal drag Bearing drag
= 0.6 2.5 = 3.1 Nm

radually and carefully tighten the drive flange nut, twisting the pinion to seat the bearings, until the required preload is obtained. Frequent checks must be taken with the preload gauge and if the maximum preload is exceeded the collapsible spacer must be renewed. Note that slackening off the drive flange nut will only remove the established excessive preload and will not reset the required preload.

7.1. Ad ust cro n heel and pinion bac lash and differential bearing preload

Setting differential cage bearing preload using shims (Figs 7.3(c and d) and 7.4) ifferential bearing preload shims may be situated between the differ-
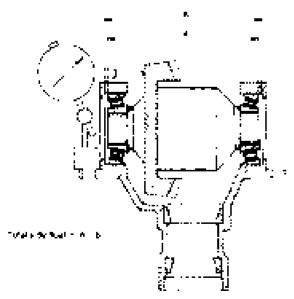
Fig. 7.4 Setting differential cage bearing preload using shims

ential cage and bearings (Fig. 7.3(c)) or between axle housing and bearings (Fig. 7.3(d)). The method of setting the differential bearing preload is similar in both arrangements, but only the case of shims between the axle housing and bearing will be described.

With the pinion removed, press both differential bearing cones onto the differential cage and slip the bearing cups over rollers and cones. ower the differential and crownwheel assembly with bearing cups but without shims into the final drive axle housing. Install the dial indicator on the final drive housing with the spindle resting against the back face of the crownwheel. Insert two levers between the final drive housing and the differential cage assembly, fully moving it to one side of the housing. Set the indicator to zero and then move the assembly to the other side and record the reading, which will give the total side float between the bearings as now assembled and the abutment faces of the final drive housing. A preload shim thickness is then added to the side float between the differential bearings and final drive housing. This normally amounts to a shim thickness of 0.06 mm added to both sides of the differential. The total shim thickness required between the differential bearings and final drive housing can then be divided according to the crownwheel and pinion backlash requirements as under setting backlash with shims.

For example,

ifferential side float	=	1.64 mm
ifferential bearing preload allowance (2 × 0.06)	=	0.12 mm
Total differential bearing pack thickness	=	1.76 mm

Setting crownwheel and pinion backlash using shims (Figs 7.5 and 7.6) After the pinion depth has been set, place the differential assembly with the bearing cups but without shims into the final drive housing, being sure that all surfaces are absolutely clean. Install a dial indicator on the housing with spindle resting against the back of the crownwheel (Fig. 7.5). Insert two levers between the housing and the crownwheel side of the differential assembly. Move the differential away from the pinion until the opposite bearing cup is seated against the housing. Set the dial indicator to zero. The levers are now transferred to the opposite side of the differential cage so that the whole unit can now be pushed
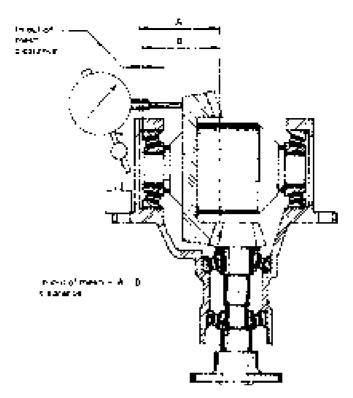
Fig. 7.5 Setting crownwheel and pinion backlash using shims

towards the pinion until both crownwheel and pinion teeth fully mesh. bserve the dial indicator reading, which is the of mesh clearance between the crownwheel and pinion teeth (shims removed). This denotes the thickness of shims minus the backlash allowance to be placed between the final drive housing and the bearing cone on the crownwheel side of the differential cage to obtain the correct backlash.

Backlash allowance is either etched on the crownwheel or it may be assumed a movement of 0.05 mm shim thickness from one differential bearing to the other will vary the backlash by approximately 0.05 mm.

From the following data determine the shim pack thickness to be placed on both sides of the crownwheel and differential assembly between the bearing and axle housing.

ifferential side float with shims	
removed	= 1.64 mm
ifferential bearing preload	
allowance each side	= 0.06 mm
In-out of mesh clearance	= 0.62 mm
Backlash allowance	= 0.12 mm

In-out of mesh clearance	= 0.62 mm
ifferential bearing	
preload allowance (add)	= 0.06 mm
Backlash allowance (subtract)	= 0.12 mm
Required shim pack crownwheel side	= 0.56 mm

Total differential side float	= 1.64 mm
ifferential bearing	
preload allowance (add)	= 0.06 mm
Crownwheel side shim pack without	
preload and allowance (subtract)	= 0.50 mm
Required shim pack opposite	
crownwheel side	= 1.20 mm

Alternatively,

Required shim	= Total differential	Shim pack
pack opposite	bearing pack	crownwheel
crownwheel	thickness	side
side		
	= (1.64 0.12) 0.56
	= 1.76 0.56
	= 1.20 mm

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To check the crownwheel and pinion backlash, attach the dial gauge magnetic stand on the axle housing flange with the dial gauge spindle resting against one of the crownwheel teeth so that some sort of gauge reading is obtained (Fig. 7.6). old the pinion stationary and rock the crownwheel backwards and forwards observing the variation in gauge reading, this valve being the backlash between the crownwheel and pinion teeth. A typical backlash will range between 0.10 and 0.125 mm for original bearings or 0.20 and 0.25 mm for new bearings.

Setting crownwheel and pinion backlash and pre-loading differential bearings using ad usting nuts (Figs 7.6 and 7.7) ocate the differential bearing caps on their cones and position the differential assembly in the final drive housing. Refit the bearing caps with the mating marks aligned and replace the bolts so that they ust nip the caps in position. Screw in the ad usting nuts whilst rotating the crownwheel until there is ust a slight backlash. Bolt the spread gauge to the centre bolt hole of the bearing cap and fit an inverted bearing cap lock tab to the other cap (Fig. 7.7). Ensure that the dial gauge spindle rests against the lock tab and set the gauge to zero. Mount the backlash gauge magnetic base stand on the final drive housing flange so that the dial spindle rests against a tooth at right angles to it
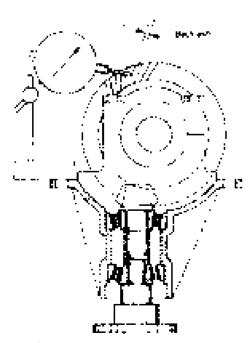
and zero the gauge (Fig. 7.6). Screw in the ad usting nuts until a backlash of 0.025 to 0.05 mm is indicated when rocking the crownwheel. Swing the backlash gauge out of position.

Screw in the ad usting nut on the differential side whilst rotating the crownwheel until a constant cap spread (preload) of 0.20–0.25 mm is indicated for new bearings, or 0.10–0.125 mm when re-using the original bearings.

Swing the backlash gauge back into position and zero the gauge. old the pinion and rock the crownwheel. The backlash should now be 0.20–0.25 mm for new bearings or 0.10–0.125 mm with the original bearings.

If the backlash is outside these limits, ad ust the position of the crownwheel relative to the pinion by slackening the ad usting nut on one side and tightening the nut on the other side so that the cap spread remains unaltered. The final tightening must always be made to the nut on the crownwheel side.

Refit the lock tabs left and right hand and torque down the cap bolts a typical value for a car axle is 60–70 Nm.

7.1.5 *ear tooth terminology* (Fig. 7.8(a))

The halfway point on the tooth profile between the face and the flank is called the .

The bottom of the tooth profile is known as the .

The upper half position of the tooth profile between the pitch line and the tooth tip is called the .

The lower half position of the tooth profile between the pitch line and the tooth root is called the .

The outer large half portion of the crownwheel tooth length is known as the .

The inner half portion of the crownwheel tooth length is known as the .

This is the convex side of each crownwheel tooth wheel which receives the contact pressure from the pinion teeth when the engine drives the vehicle forward.

This is the concave side of each crownwheel tooth which contacts the
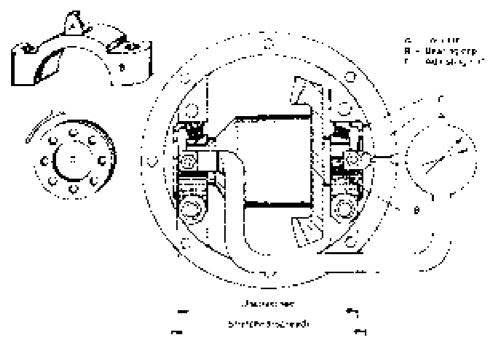

Fig. 7.6 Checking crownwheel and pinion backlash

Fig. 7.7 Setting differential cage bearing preload using adjusting nuts

pinion teeth when the transmission overruns the engine or the vehicle is being reversed.

The free clearance between meshing teeth is known as .

7.1.6 Checking crown wheel and pinion tooth contact

Prepare crownwheel for examining tooth contact marks (Fig. 7.8) After setting the correct backlash, the crownwheel and pinion tooth alignment should be checked for optimum contact. This may be achieved by applying a marking cream such as Prussian blue, red lead, chrome yellow, red or yellow ochre etc. to three evenly spaced groups of about six teeth round the crownwheel on both drive coast sides of the teeth profiles. Apply a load to the meshing gears by holding the crownwheel and allowing it to slip round while the pinion is turned a few revolutions in both directions to secure a good impression around the crownwheel. Examine the tooth contact pattern and compare it to the recommended impression.

Understanding tooth contact marks (Fig. 7.8(a–f)) If the crownwheel to pinion tooth contact pattern is incorrect, there are two adjustments that can be made to change the position of tooth contact. These adjustments are of backlash and pinion depth.

The adjustment of backlash moves the contact patch lengthwise back and forth between the toe heel of the tooth. Moving the crownwheel nearer the pinion decreases the backlash, causing the contact patch to shift towards the toe portion of the tooth. Increasing backlash requires the crownwheel to be moved sideways and away from the pinion. This moves the contact patch nearer the heel portion of the tooth.

When adjusting pinion depth, the contact patch moves up and down the face–flank profile of the tooth. With insufficient pinion depth (pinion too far out from crownwheel) the contact patch will be concentrated at the top (face zone) of the tooth. Conversely, too much pinion depth (pinion too near crownwheel) will move the contact patch to the lower root (flank zone) of the tooth.

(Fig. 7.8(b)) The area of tooth contact should be evenly distributed over the working depth of the tooth profile and should be nearer to the toe than the heel of the crownwheel tooth. The setting of the tooth contact is initially slightly away from the heel and nearer the root to compensate for any deflection of the bearings,
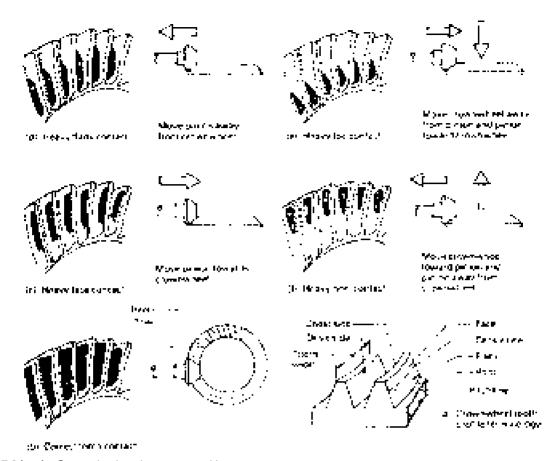
Fig. 7.8 (a–)   Crownwheel tooth contact markings

crownwheel, pinion and final drive housing under operating load conditions, so that the pressure contact area will tend to spread towards the heel towards a more central position.

(Fig. 7.8(c)) Tooth contact area is above the centre line and on the face of the tooth profile due to the pinion being too far away from the crownwheel (insufficient pinion depth). To rectify this condition, move the pinion deeper into mesh by using a thicker pinion head washer to lower the contact area and reset the backlash.

(Fig. 7.8(d)) Tooth contact area is below the centre line and on the flank of the tooth profile due to the pinion being too far in mesh with the crownwheel (too much pinion depth). To rectify this condition, move the pinion away from the crownwheel using a thinner washer between the pinion head and inner

bearing cone to raise the contact area and then reset the backlash.

(Fig. 7.8(e)) Tooth contact area is concentrated at the small end of the tooth (near the toe). To rectify this misalignment, increase backlash by moving the crownwheel and differential assembly away from the pinion, by transferring shims from the crownwheel side of the differential assembly to the opposite side, or slacken the ad usting nut on the crownwheel side of the differential and screw in the nut on the opposite side an equal amount. If the backlash is increased above the maximum specified, use a thicker washer (shim) behind the pinion head in order to keep the backlash within the correct limits.

(Fig. 7.8(f)) Tooth contact area is concentrated at the large end of the tooth which is near the heel. To rectify this misalignment, decrease backlash by moving the crownwheel nearer

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the pinion (add shims to the crownwheel side of the differential and remove an equal thickness of shims from the opposite side) or slacken the differential side ad usting nut and tighten the crownwheel side nut an equal amount. If the backlash is reduced below the minimum specified, use a thinner washer (shim) behind the pinion head.

7.1.7   inal drive a le noise and defects

Noise is produced with all types of meshing gear teeth such as from spur, straight or helical gears and even more so with bevel gears where the output is redirected at right angles to the input drive.

Vehicle noises coming from tyres, transmission, propellor shafts, universal oints and front or rear wheel bearings are often mistaken for axle noise, especially tyre to road surface rumbles which can sound very similar to abnormal axle noise.   istening for the noise at varying speeds and road surfaces, on drive and overrun conditions will assist in locating the source of any abnormal sound.

nce all other causes of noise have been eliminated, axle noise may be suspected. The source of axle noise can be divided into gear teeth noises and bearing noise.

ear noise   ear noise may be divided into two kinds:

1 Broken, bent or forcibly damaged gear teeth which produce an abnormal audible sound which is easily recognised over the whole speed range.
   a) Broken or damaged teeth may be due to abnormally high shock loading causing sudden tooth failure.
   b) Extended overloading of both crownwheel and pinion teeth can be responsible for eventual fatigue failure.
   c)   ear teeth scoring may eventually lead to tooth profile damage. The causes of surface scoring can be due to the following:

   i) Insufficient lubrication or incorrect grade of oil
   ii) Insufficient care whilst running in a new final drive
   iii) Insufficient crownwheel and pinion backlash
   iv)   istorted differential housing
   v) Crownwheel and pinion misalignment
   vi)   oose pinion nut removing the pinion bearing preload.
2 Incorrect meshing of crownwheel and pinion teeth. Abnormal noises produced by poorly

meshed teeth generate a very pronounced cyclic pitch whine in the speed range at which it occurs whilst the vehicle is operating on either drive or overrun conditions.

If a harsh cyclic pitch noise is heard when the engine is driving the transmission it indicates that the pinion needs to be moved slightly out of mesh.

If a pronounced humming noise is heard when the vehicle s transmission overruns the engine, this indicates that the pinion needs to be moved further into mesh.

A pronounced time lag in taking the drive up accompanied by a knock when either accelerating or decelerating may be traced to end play in the pinion assembly due possibly to defective bearings or incorrectly set up bearing spacer and shim pack.

earing noise   Bearings which are defective produce a rough growling sound that is approximately constant in volume over a narrow speed range.

riving the vehicle on a smooth road and listening for rough transmission sounds is the best method of identifying bearing failure.

A distinction between defective pinion bearings or differential cage bearings can be made by listening for any constant rough sound. A fast frequency growl indicates a failed pinion bearing, while a much slower repetition growl points to a defective differential bearing. The difference in sound is because the pinion revolves at about four times the speed of the differential assembly.

To distinguish between differential bearing and half shaft bearing defects, drive the vehicle on a smooth road and turn the steering sharply right and left. If the half shaft bearings are at fault, the increased axle load imposed on the bearing will cause a rise in the noise level, conversely if there is no change in the abnormal rough sound the differential bearings should be suspect.

efective differential planet and sun gears   The sun and planet gears of the differential unit very rarely develop faults. When differential failure does occur, it is usually caused by shock loading, extended overloading and seizure of the differential planet gears to the cross-shaft resulting from excessive wheel spin and consequently lubrication breakdown.

A roughness in the final drive transmission when the vehicle is cornering may indicate defective planet/sun gears.

7.2 ifferential locks

A differential lock is desirable, and in some cases essential, if the vehicle is going to operate on low traction surfaces such as sand, mud, wet or water-logged ground, worn slippery roads, ice bound roads etc. at relatively low speeds.

rive axle differential locks are incorporated on heavy duty on/off highway and cross-country vehicles to provide a positive drive between axle half shafts when poor tyre to ground traction on one wheel would produce wheel spin through differential bevel gear action.

The differential lock has to be engaged manually by cable or compressed air, whereas the limited slip or viscous coupling differential automatically operates as conditions demand.

All differential locks are designed to lock together two or more parts of the differential gear cluster by engaging ad acent sets of dog clutch teeth. By this method, all available power transmitted to the final drive will be supplied to the wheels. Even if one wheel loses grip, the opposite wheel will still receive power enabling it to produce torque and therefore tractive effect up to the limit of the tyres ability to grip the road. Axle wind-up will be dissipated by wheel bounce, slippage or scuffing.

These unwanted reactions will occur when travelling over slippery soft or rough ground where true rolling will be difficult. Since the tyre tread cannot exactly follow the contour of the surface it is rolling over, for very brief periodic intervals there will be very little tyre to ground adhesion. As a result, any build up of torsional strain between the half shafts will be continuously released.

7. .1 Differential loc mechanism
(Figs 7.9 and 7.10)

ne example of a differential lock is shown in Fig. 7.9. In this layout a hardened and toughened flanged side toothed dog clutch member is clamped and secured by dowls between the crownwheel and differential cage flanges. The other dog clutch member is comprised of a sleeve internally splined to slot over the extended splines on one half shaft. This sleeve has dog teeth cut at one end and the double flange formed at the end to provide a guide groove for the actuating fork arm.

Engagement of the differential lock is obtained when the sleeve sliding on the extended external splines of the half shaft is pushed in to mesh with corresponding dog teeth formed on the flanged member mounted on the crownwheel and cage.

ocking one half shaft to the differential cage prevents the bevel gears from revolving independently within the cage. Therefore, the half shafts and cage
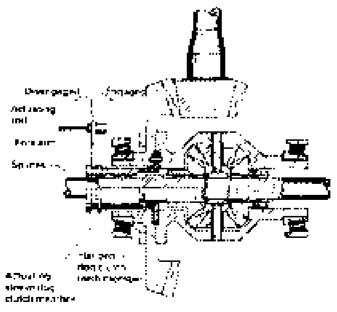

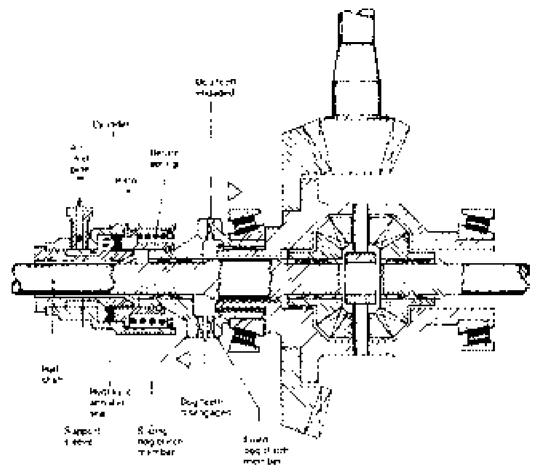**Fig. 7.9** ifferential lock mechanism

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Fig. 7.10 ifferential lock mechanism with air control

will be compelled to revolve with the final drive crownwheel as one. The lock should be applied when the vehicle is ust in motion to enable the tooth to align, but not so fast as to cause the crashing of misaligned teeth. The engagement of the lock can be by cable, vacuum or compressed air, depending on the type of vehicle using the facility. An alternative differential lock arrangement is shown in Fig. 7.10 where the lock is actuated by compressed air operating on an annulus shaped piston positioned over one half shaft. When air pressure is supplied to the cylinder, the piston is pushed outwards so that the sliding dog clutch member teeth engage the fixed dog clutch member teeth, thereby locking out the differential gear action.

When the differential lock is engaged, the vehicle should not be driven fast on good road surfaces to prevent excessive tyre scrub and wear. With no differential action, relative speed differences between

inner and outer drive wheels can only partially be compensated by the tyre tread having sufficient time to distort and give way in the form of minute hops or by permitting the tread to skid or bounce while rolling in slippery or rough ground conditions.

7.3 Skid reducing differentials

7.3.1 Salisbury o r o limited slip differential
(Fig. 7.11)
This type of limited slip differential is produced under licence from the American Thornton Axle Co.

The Powr- ok limited slip differential essentially consists of an ordinary bevel gear differential arranged so that the torque from the engine engages friction clutches locking the half shafts to the differential cage. The larger the torque, the greater the locking effect (Fig. 7.11).

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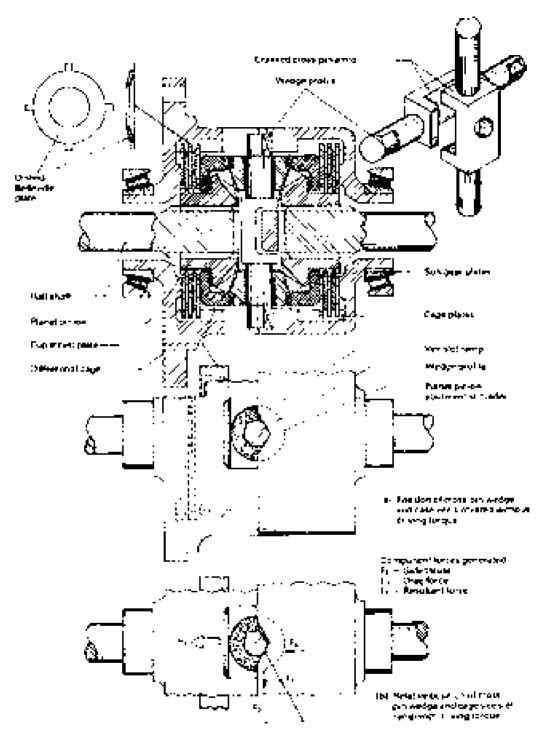
Fig. 7.11 ulticlutch limited slip differential

There are three stages of friction clutch loading:

1 Belleville spring action,
2 Bevel gear separating force action,
3 Vee slot wedging action.

elleville spring action (Fig. 7.11) This is achieved by having one of the clutch plates dished to form a Belleville spring so that there is always some spring axial loading in the clutch plates. This then produces a small amount of friction which tends to lock the half shaft to the differential cage when the torque transmitted is very low. The spring thus ensures that when adhesion is so low that hardly any torque can be transmitted, some drive will still be applied to the wheel which is not spinning.

evel gear separating force action (Fig. 7.11) This arises from the tendency of the bevel planet pinions in the differential cage to force the bevel sun gears outwards. Each bevel sun gear forms part of a hub which is internally splined to the half shaft so that it is free to move outwards. The sun gear hub is also splined externally to align with one set of clutch plates, the other set being attached by splines to the differential cage. Thus the extra outward force exerted by the bevel pinions when one wheel tends to spin is transmitted via cup thrust plates to the clutches, causing both sets of plates to be camped together and thereby preventing relative movement between the half shaft and cage.

Vee slot wedging action (Fig. 7.11(a and b)) When the torque is increased still further, a third stage of friction clutch loading comes into being. The bevel pinions are not mounted directly in the differential cage but rotate on two separate arms which cross at right angles and are cranked to avoid each other. The ends of these arms are machined to the shape of a vee wedge and are located in vee-shaped slots in the differential cage. With engine torque applied, the drag reaction of the bevel planet pinion cross-pin arms relative to the cage will force them to slide inwards along the ramps framed by the vee-shaped slots in the direction of the wedge (Fig. 7.11(a and b)). The abutment shoulder of the bevel planet pinions press against the cup thrust plates and each set of clutch plates are therefore squeezed further together, increasing the multiclutch locking effect.

Speed differential and traction control (Fig. 7.12) Normal differential speed ad ustment takes place
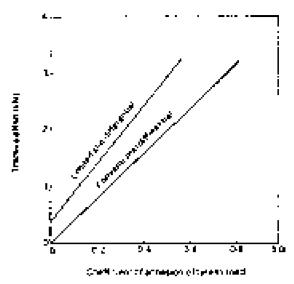

Fig. 7.12 Comparison of tractive effort and tyre to road adhesion for both conventional and limited slip differential

continuously, provided the friction of the multi-plate clutches can be overcome. When one wheel spins the traction of the other wheel is increased by an amount equal to the friction torque generated by the clutch plates until wheel traction is restored. A comparison of a conventional differential and a limited slip differential tractive effort response against varying tyre to road adhesion is shown in Fig. 7.12.

7.3. Torsen orm and heel differential

ifferential construction (Figs 7.13 and 7.14) The Torsen differential has a pair of worm gears, the left hand half shaft is splined to one of these worm gears while the right hand half shaft is splined to the other hand (Fig. 7.13). Meshing with each worm gear on each side is a pair of worm wheels (for large units triple worm wheels on each side). At both ends of each worm wheel are spur gears which mesh with ad acent spur gears so that both worm gear and half shafts are indirectly coupled together.

Normally with a worm gear and worm wheel combination the worm wheel is larger than the worm gear, but with the Torsen system the worm gear is made larger than the worm wheel. The important feature of any worm gear and worm wheel is that the teeth are cut at a helix angle such that the worm gear can turn the worm wheel but the worm wheel cannot rotate the worm gear. This is achieved with the Torsen differential by giving the
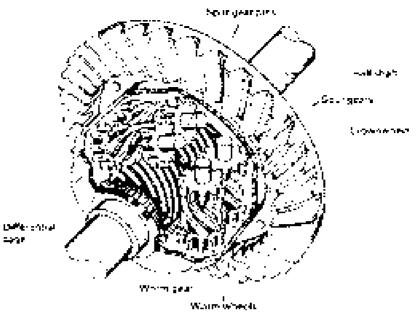
Fig. 7.13 ictorial view of Torsen worm and spur gear differential

worm gear teeth a fine pitch while the worm wheel has a coarse pitch.

Note that with the conventional meshing spur gear, be it straight or helical teeth, the input and output drivers can be applied to either gear. The reversibility and irreversibility of the conventional bevel gear differential and the worm and worm wheel differential is illustrated in Fig. 7.14 by the high and low mechanical efficiencies of the two types of differential.

ifferential action when moving straight ahead (Fig. 7.15) When the vehicle is moving straight ahead power is transferred from the propeller shaft to the bevel pinion and crownwheel. The crownwheel and differential cage therefore revolve as one unit (Fig. 7.15). Power is divided between the left and right hand worm wheel by way of the spur gear pins which are attached to the differential cage. It then flows to the pair of meshing worm gears, where it finally passes to each splined half shaft. Under these conditions, the drive in terms of speed and torque is proportioned equally to both half shafts and road wheels. Note that there is no relative rotary motion between the half shafts and the differential cage so that they all revolve as a single unit.

ifferential action when cornering (Fig. 7.15) When cornering, the outside wheel of the driven axle will
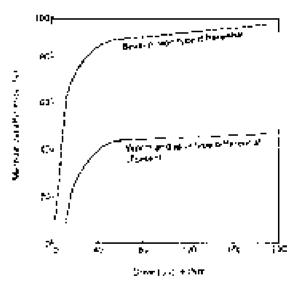
Fig. 7.14 Comparison of internal friction expressed in terms of mechanical efficiency of both bevel pinion type and worm and spur type differentials

tend to rotate faster than the inside wheel due to its turning circle being larger than that of the inside wheel. It follows that the outside wheel will have to rotate relatively faster than the differential cage, say by 20 rev/min, and conversely the inside wheel has to reduce its speed in the same proportion, of say 20 rev/min.
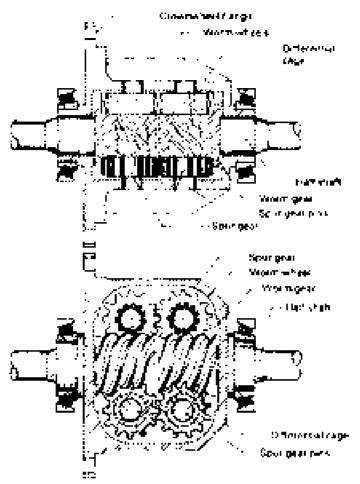
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Fig. 7.15 Sectioned views of Torsen worm and spur gear differential

When there is a difference in speed between the two half shafts, the faster turning half shaft via the splined worm gears drives its worm wheels about their axes (pins) in one direction of rotation. The corresponding slower turning half shaft on the other side drives its worm wheels about their axes (pins) in the opposite direction but at the same speed (Fig. 7.15).

Since the worm wheels on opposite sides will be revolving at the same speed but in the opposite sense while the vehicle is cornering they can be simply interlinked by pairs of meshing spur gears without interfering with the independent road speed requirements for both inner and outer driving road wheels.

ifferential tor ue distribution (Fig. 7.15) When one wheel loses traction and attempts to spin, it transmits drive from its set of worm gears to the worm wheels. The drive is then transferred from the worm wheels on the spinning side to the opposite (good traction wheel) side worm wheels by way of the bridging spur gears (Fig. 7.15). At this point the engaging teeth of the worm wheel with the corresponding worm gear teeth am. Thus the wheel which has lost its traction locks up the gear mechanism on the other side every time there is a tendency for it to spin. As a result of the low traction wheel being prevented from spinning, the transmission of torque from the engine will be concentrated on the wheel which has traction.

Another feature of this mechanism is that speed differentiation between both road wheels is maintained even when the wheel traction differs considerably between wheels.
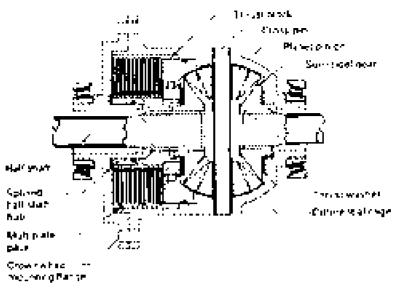
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Fig. 7.16 iscous coupling differential

7.3.3 iscous coupling differential

escription of differential and viscous coupling (Figs 7.16 and 7.17) The crownwheel is bolted to the differential bevel gearing and multiplate housing. Speed differentiation is achieved in the normal manner by a pair of bevel sun (side) gears, each splined to a half shaft. Bridging these two bevel sun gears are a pair of bevel planet pinions supported on a cross-pin mounted on the housing cage. A multiplate back assembly is situated around the left hand half shaft slightly outboard from the corresponding sun gear (Fig. 7.16).

The viscous coupling consists of a series of spaced interleaved multiplates which are alternatively splined to a half shaft hub and the outer differential cage. The cage plates have pierced holes but the hub plates have radial slots. Both sets of plates are separated from each other by a 0.25 mm gap. Thus the free gap between ad acent plates and the interruption of their surface areas with slots and holes ensures there is an adequate storage of fluid between plates after the sealed plate unit has been filled and that the necessary progressive viscous fluid torque characteristics will be obtained when relative movement of the plates takes place.

When one set of plates rotate relative to the other, the fluid will be sheared between each pair of ad acent plate faces and in so doing will generate an opposing torque. The magnitude of this resisting torque will be proportional to the fluid viscosity and the relative speed difference between the sets of plates. The dilatent silicon compound fluid which has been developed for this type of application has the ability to maintain a constant level of viscosity throughout the operating temperature range and life expectancy of the coupling (Fig. 7.17).
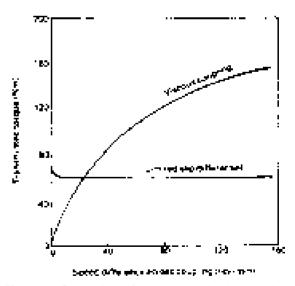

Fig. 7.17 Comparison of tor ue transmitted to wheel having the greater adhesion with respect to speed difference between half shafts for both limited slip and viscous coupling

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Speed differential action (Fig. 7.16) In the straight ahead driving mode the crownwheel and differential cage driven by the bevel pinion act as the input to the differential gearing and in so doing the power path transfers to the cross-pin and bevel planet gears. ne of the functions of these planet gears is to link (bridge) the two sun (side) gears so that the power flow is divided equally between the sun gears and correspondently both half shafts (Fig. 7.16).

When rounding a bend or turning a corner, the outer wheel will have a greater turning circle than the inner one. Therefore the outer wheel tends to increase its speed and the inner wheel decrease its speed relative to the differential cage rotational speed. This speed differential is made possible by the different torque reactions each sun gear conveys back from the road wheel to the bevel planet pinions. The planet gears float between the sun gears by rotating on their cross-pin, thus the speed lost relative to the cage speed by the inner road wheel and sun gear due to the speed retarding ground reaction will be that gained by the outer road wheel and sun gear.

Viscous coupling action (Figs 7.16 and 7.17) In the situation when one wheel loses traction caused by possibly loose soil, mud, ice or snow, the tyre–road tractive effort reaction is lost. Because of this lost traction there is nothing to prevent the planet pinions revolving on their axes, rolling around the opposite sun gear, which is connected to the road wheel sustaining its traction, with the result that the wheel which has lost its grip will ust spin (race) with no power being able to drive the good wheel (Fig. 7.16). Subsequently, a speed difference between the cage plates and half shaft hub plates will be established and in proportion to this relative speed, the two sets of coupling plates will shear the silicon fluid and thereby generate a viscous drag torque between ad acent plate faces (Fig. 7.17). As a result of this viscous drag torque the half shaft hub plates will proportionally resist the rate of fluid shear and so partially lock the differential gear mechanism. A degree of driving torque will be transmitted to the good traction wheel. Fig. 7.17 also compares the viscous coupling differential transmitted torque to the limited slip differential.

 ere it can be seen that the limited slip differential approximately provides a constant torque to the good traction wheel at all relative speeds, whereas the viscous coupling differential is dependent on speed differences between both half shafts so that

the torque transmitted to the wheel supplying tractive effort rises with increased relative speed between the half shaft and differential cage.

7.4   ouble reduction a les

7. .1 *The need for double reduction final drives*
The gearbox provides the means to ad ust and match the engine s speed and torque so that the vehicle s performance responds to the driver s expectations under the varying operating conditions. The gearbox gear reduction ratios are inadequate to supply the drive axle with sufficient torque multiplication and therefore a further permanent gear reduction stage is required at the drive axle to produce the necessary road wheel tractive effect. For light vehicles of 0.5–2.0 tonne, a final drive gear reduction between 3.5:1 and 4.5:1 is generally sufficient to meet all normal driving conditions, but with commercial vehicles carrying considerably heavier payloads a demand for a much larger final drive gear reduction of 4.5–9.0:1 is essential. This cannot be provided by a single stage final drive crownwheel and pinion without the crownwheel being abnormally large. ouble reduction axles partially fulfil the needs for heavy goods vehicles operating under normal conditions by providing two stages of gear reduction at the axle.

In all double reduction final drive arrangements the crownwheel and pinion are used to provide one stage of speed step down. At the same time the bevel gearing redirects the drive perpendicular to the input propellor shaft so that the drive then aligns with the axle half shafts.

7. .   *Double reduction a les ith first stage reduction before the cro n heel and pinion*

ouble reduction with spur gears ahead of bevel gears (Fig. 7.18) With a pair of helical gears providing the first gear reduction before the crownwheel and pinion, a high mounted and compact final drive arrangement is obtained. This layout has the disadvantage of the final gear reduction and thus torque multiplication is transmitted through the crownwheel and pinion bevel gears which therefore absorbs more end thrust and is generally considered to be less efficient in operation compared to helical spur type gears. The first stage of a double reduction axle is normally no more than 2:1 leaving the much larger reduction for the output stage.
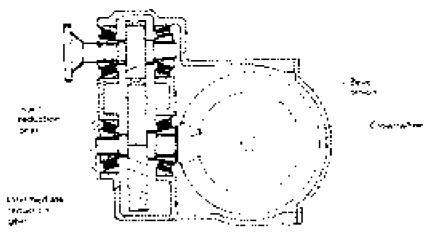
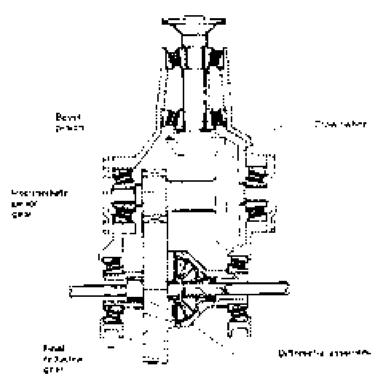**Fig. 7.18** Final drive spur double reduction ahead of bevel pinion

Fig. 7.19 Final drive spur double reduction between crownwheel and differential

ouble reduction with bevel gears ahead of spur gears (Fig. 7.19) A popular double reduction arrangement has the input from the propellor shaft going directly to the bevel pinion and crownwheel. The drive is redirected at right angles to that of the input so making it flow parallel to the half shafts, the first stage gear reduction being deter-mined by the relative sizes (number of teeth) of the bevel gears. A helical pinion gear mounted on the same shaft as the crownwheel meshes with a helical gear wheel bolted to the differential cage. The com-bination of these two gear sizes provides the second stage gear reduction. aving the bevel gears ahead of the helical gears ensures that only a proportion of

torque multiplication will be constrained by them, while the helical gears will absorb the full torque reaction of the final gear reduction.

7. .3 nboard and outboard double reduction a les

Where very heavy loads are to be carried by on-off highway vehicles, the load imposed on the crownwheel and pinion and differential unit can be reduced by locating a further gear reduction on either side of the differential exit. If the second gear reduction is arranged on both sides close to the differential cage, it is referred to as an inboard reduction. They can be situated at the wheel ends of the half shafts, where they are known as outboard second stage gear reduction. By having the reduction directly after the differential, the increased torque multiplication will only be transmitted to the half shafts leaving the crownwheel, pinion and differential with a torque load capacity proportional to their gear ratio. The torque at this point may be smaller than with the normal final drive gear ratio since less gear reduction will be needed at the crownwheel and pinion if a second reduction is to be provided. Alternatively, if the second reduction is in the axle hub, less torque will be transmitted by the half shafts and final drive differential and the dimensions of these components can be kept to a minimum. aving either an inboard or outboard second stage gear reduction enables lighter crownwheel and pinion combinations and differential assembly to be employed, but it does mean there are two gear reductions for each half shaft, as opposed to a single double reduction drive if the reduction takes place before the differential.

nboard epicyclic double reduction final drive a le (Scammell) (Fig. 7.20) With this type of double reduction axle, the first stage conforms to the conventional crownwheel and pinion whereas the second stage reduction occurs after passing through the differential. The divided drive has a step down gear reduction via twin epicyclic gear trains on either side of the differential cage (Fig. 7.20). Short shafts connect the differential bevel sun gears to the pinion sun gear of the epicyclic gear train. When drive is being transmitted, the rotation of the sun gears rotates the planet pinions so that they are forced to roll walk around the inside of the reaction annulus gear attached firmly to the axle casing. Support to the planet pinions and their pins is given by the planet carrier which is itself mounted on a ball race. Thus when the planet pinions are made to rotate on their own axes they also bodily rotate about the same axis of rotation as the sun gear, but at a reduced speed, and in turn convey power to the half shafts splined to the central hub portion of the planet carriers.

nboard epicyclic differential and double reduction a le (irkstall) (Fig. 7.21) This unique double reduction axle has a worm and worm wheel first stage gear reduction. The drive is transferred to an epicyclic gear train which has the dual function of providing the second stage gear reduction while at
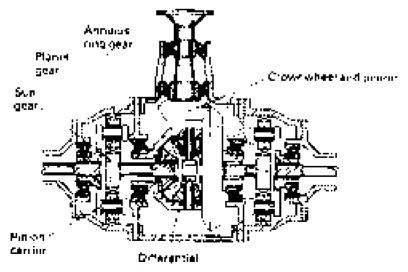

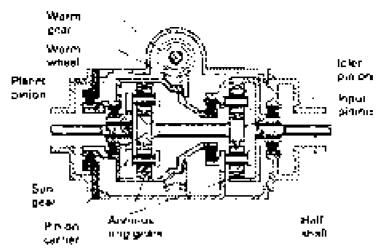**Fig. 7.20** nboard epicyclic double reduction final drive axle

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Fig. 7.21 nboard epicyclic double reduction axle

the same time performing as the final drive differential (Fig. 7.21).

Power is transmitted from the propellor shaft to the worm and worm wheel which produces a gear reduction and redirects the drive at right angles and below the worm axis of rotation (Fig. 7.21). The worm wheel is mounted on the annulus carrier so that they both rotate as one. Therefore the three evenly spaced planet pinions meshing with both the annulus and the sun gear are forced to revolve and move bodily on their pins in a forward direction. Since the sun gear is free to rotate (not held stationary) it will revolve in a backward direction so that the planet carrier and the attached left hand half shaft will turn at a reduced speed relative to the annulus gear.

Simultaneously, as the sun gear and shaft transfers motion to the right hand concentric gear train central pinion, it passes to the three idler pinions, compelling them to rotate on their fixed axes, and in so doing drives round the annulus ring gear and with it the right hand half shaft which is splined to it.

The right hand gear train with an outer internal ring gear (annulus) does not form an epicyclic gear train since the planet pins are fixed to the casing and do not bodily revolve with their pins (attached to a carrier) about some common centre of rotation. It is the purpose of the right hand gear train to produce an additional gear reduction to equalize the gear reduction caused by the planet carrier output on the left hand epicyclic gearing with the sun gear output on the right hand side.

The operation of the differential is quite straight forward if one imagines either the left or right hand half shaft to slow down as in the case when they are attached to the inner wheel of a cornering vehicle.

If when cornering the left hand half shaft slows down, the planet carrier will correspondingly reduce speed and force the planet pinions revolving on their pins to spin at an increased speed. This raises the speed of the sun gear which indirectly drives, in this case, the outer right hand half shaft at a slightly higher speed. Conversely, when cornering if the right hand half shaft should slow down, it indirectly reduces the speed of the central pinion and sun gear. ence the planet pinions will not revolve on their pins, but will increase their speed at which they also roll round the outside of the sun gear. Subsequently the planet pins will drive the planet carrier and the left hand half shaft at an increased speed.

7. .   utboard double reduction a les

utboard epicyclic spur gear double reduction a le (Fig. 7.22)

(Fig. 7.22) A gear reduction between the half shaft and road wheel hub may be obtained through an epicyclic gear train. A typical step down gear ratio would be 4:1. The sun gear may be formed integrally with or it may be splined to the half shaft (Fig. 7.22). It is made to engage with three planet gears carried on pins fixed to and rotating with the hub, thus driving
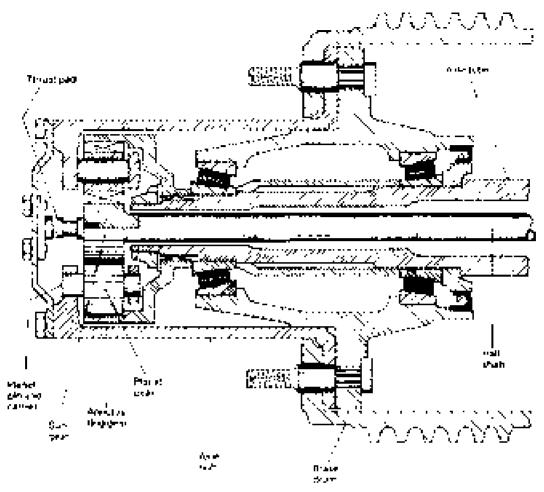
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Fig. 7.22 utboard epicyclic spur double reduction axle

the latter against the reaction of an outer annulus splined to the stationary axle tube. The sun wheel floats freely in a radial direction in mesh with the planet pinions so that driving forces are distributed equally on the three planet pinions and on their axes of rotation. A half shaft and sun gear end float is controlled and absorbed by a thrust pad mounted on the outside end cover which can be initially ad usted by altering the thickness of a shim pack.

(Fig. 7.22) In operation, power flows from the differential and half shaft to the sun gear where its rotary motion is distributed between the three planet pinions. The forced rotation of these planet pinions compels them to roll around the inside of the reaction annulus ring gear (held stationary) so that their

axes of rotation, and the planet pins, are forced to revolve about the sun gear axis. Since the planet pins are mounted on the axle hub, which is itself mounted via a fully floating taper bearing arrangement on the axle tube, the whole hub assembly will rotate at a much reduced speed relative to the half shaft s input speed.

utboard epicyclic bevel gear single and two speed double reduction a le (Fig. 7.23) This type of outboard double reduction road wheel hub employs bevel epicyclic gearing to provide an axle hub reduction. To achieve this gear reduction there are two bevel sun (side) gears. ne is splined to and mounted on the axle tube and is therefore fixed. The other one is splined via the sliding sleeve

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dog clutch to the half shaft and so is permitted to revolve (Fig. 7.23). Bridging both of these bevel sun gears are two planetary bevel gears which are supported on a cross-pin mounted on the axle hub.

The planetary bevel gear double reduction axle hub may be either two speed, as explained in the following text, or a single speed arrangement in which the half shaft is splined permanently and directly to the outer sun gear.

igh ratio (Fig. 7.23) igh ratio is selected and engaged by twisting the speed selector eccentric so that its offset peg pushes the sliding sleeve outwards (to the left) until the external teeth of the dog clutch move out of engagement from the sun gear and into engagement with the internal teeth formed inside the axle hub end plate. Power is transferred from the differential and half shaft via the sleeve dog clutch directly to the axle hub without producing any gear reduction.

ow ratio (Fig. 7.23) When low ratio is engaged, the sleeve dog clutch is pushed inwards (to the right) until the external teeth of the dog clutch are moved out of engagement from the internal teeth of the hub plate and into engagement with the internal teeth of the outer bevel sun gear. The input drive is now transmitted to the half shaft where it rotates the outer bevel sun gear so that the bevel planet gears are compelled to revolve on the cross-pin. In doing so they are forced to roll around the fixed inner bevel sun gear. Consequently, the cross-pin which is attached to the axle hub is made to revolve about the half shaft but at half its speed.

7. Two speed a les
The demands for a truck to operate under a varying range of operating conditions means that the overall transmission ratio spread needs to be extensive, which is not possible with a single or double reduction final
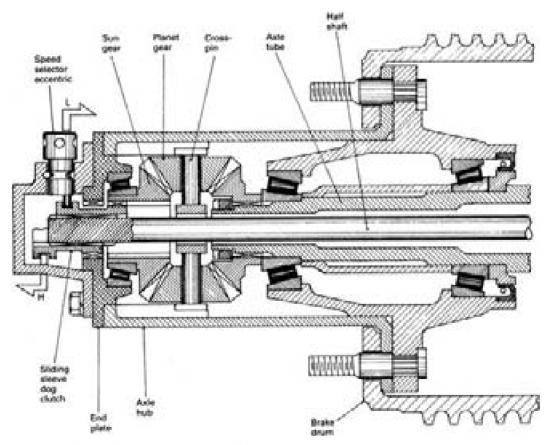

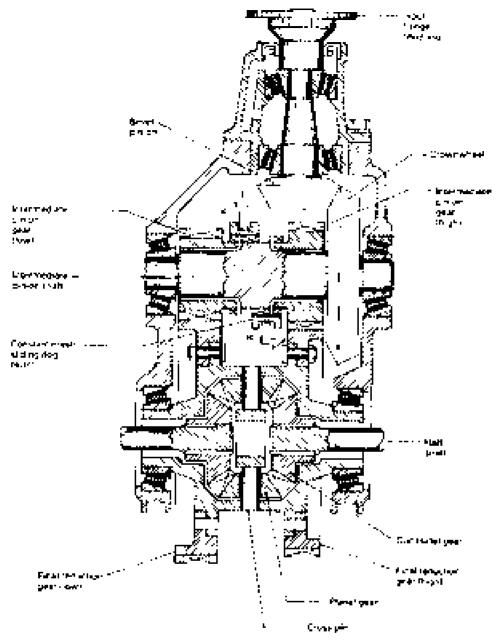**Fig. 7.23** utboard epicyclic bevel gear two speed double reduction axle

Fig. 7.24 Two speed double reduction helical gear axle

drive. For example, with a single reduction final drive the gear reduction can be so chosen as to provide a high cruising speed on good roads with a five speed gearbox. Conversely, if the truck is to be used on hilly country or for off-road use then a double reduction axle may provide the necessary gear reduction.

Therefore, to enable the vehicle to operate effectively under both motorway cruising and town stopping and accelerating conditions without overloading or overspeeding and without having to have an eight, ten or twelve speed gearbox, a dual purpose two speed gear reduction may be built into the final drive axle.

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Combining a high and low ratio in the same axle doubles the number of gears available from the standard gearbox. The low range of gears will then provide the maximum pulling power for heavy duty operations on rough roads, whereas the high range of gears allows maximum speed when conditions are favourable. From the wide range of gear ratios the driver can choose the exact combination to suit any conditions of load and road so that the engine will always operate at peak efficiency and near to its maximum torque speed band.

7.5.1 Two speed double reduction helical gear axle (Rockwell Standard) (Fig. 7.24)

This two speed double reduction helical gear axle has a conventional crownwheel and bevel pinion single speed first reduction with a second stage speed reduction consisting of two pairs of adjacent pinion and wheel helical cut gears. These pinions mounted on the crownwheel support shaft act as intermediate gears linking the crownwheel to the differential cage final reduction wheel gears (Fig. 7.24).

Low ratio (Fig. 7.24) Low ratio is engaged when the central sliding dog clutch splined to the crown-wheel shaft slides over the selected (left hand) low speed smaller pinion dog teeth. Power from the propellor shaft now flows to the bevel pinion where it is redirected at right angles to the crown-wheel and shaft. From here it passes from the locked pinion gear and crownwheel to the final reduction wheel gear bolted to the differential cage. The drive is then divided via the differential cross-pin and planet pinions between both sun gears where it is transmitted finally to the half shafts and road wheels.

High ratio (Fig. 7.24) High ratio is engaged in a similar way as for low ratio but the central sliding dog clutch slides in the opposite direction (right hand) over the larger pinion dog teeth. The slightly larger pinion meshing with a correspondently smaller differential wheel gear produces a more direct second stage reduction and hence a higher overall final drive axle gear ratio.

7.5.2 Two speed epicyclic gear train axle (Eaton) (Fig. 7.25)

With this arrangement an epicyclic gear train is incorporated between the crownwheel and differential cage (Fig. 7.25).

High ratio (Fig. 7.25) When a high ratio is required, the engagement sleeve is moved outwards

from the differential cage so that the dog teeth of both the sleeve and the fixed ring teeth disengage. At the same time the sun gear partially slides out of mesh with the planet pinions and into engagement with the outside pinion carrier internal dog teeth. Subsequently, the sun gear is free to rotate. In addition the planet pinions and carrier are locked to the sun gear, so that there can be no further relative motion within the epicyclic gear train (i.e. annulus, planet pinions, carrier, differential cage and sun gear). In other words, the crownwheel and differential cage are compelled to revolve as one so that the final drive second stage gear reduction is removed.

Low ratio (Fig. 7.25) When the engagement sleeve is moved inwards its dog clutch teeth engage with the stationary ring teeth and the sun gear is pushed fully in to mesh with the planet pinion low ratio that has been selected. Under these conditions, the input drive from the propellor shaft to the bevel pinion still rotates the crownwheel but now the sun gear is prevented from turning. Therefore the rotating crownwheel with its internal annulus ring gear revolving about the fixed sun gear makes the planet pinions rotate on their own axes (pins) and roll around the outside of the held sun gear. As a result of the planet pinions meshing with both the annulus and sun gear, and the crownwheel and annulus rotating while the sun gear is held stationary, the planetary pinions are forced to revolve on their pins which are mounted on one side of the differential cage. Thus the cage acts as a planet pinion carrier and in so doing is compelled to rotate at a slower rate relative to the annulus gear speed. Subsequently, the slower rotation of the differential cage relative to that of the crownwheel produces the second stage gear reduction of the final drive.

7.6 The third (central) differential

7.6.1 The necessity for a third differential

When four wheel drive cars or tandem drive axle bogie trucks are to be utilized, provision must be provided between drive axles to compensate for any difference in the mean speeds of each drive axle as opposed to speed differentiation between pairs of axle road wheels.

Speed difference between driving axles are influenced by the following factors:

1 Speed variation between axles when a vehicle moves on a curved track due to the slight differ-
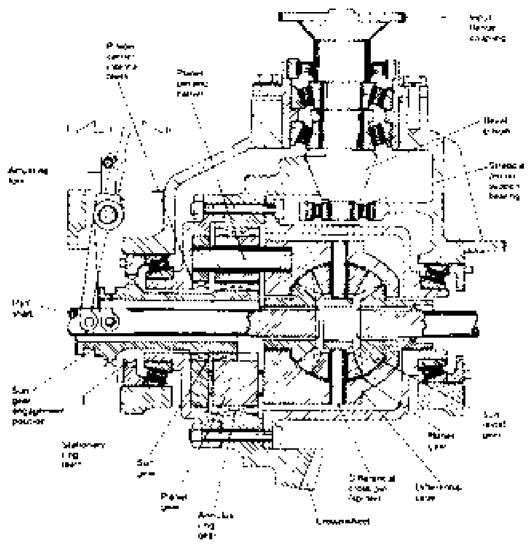
Fig. 7.25 Two speed epicyclic gear train axle

ence in rolling radius of both axles about some instantaneous centre of rotation.

2 Small road surface irregularities, causing pairs of driving wheels to locally roll into and over small dips and humps so that each pair of wheels are actually travelling at different speeds at any one moment.

3 Tyres which have different amounts of wear or different tread patterns and construction such as cross-ply and radials, high and low profile etc. and are mixed between axles so that their effective rolling radius of the wheel and tyre combination varies.

4 Uneven payload distribution will alter the effective rolling radius of a wheel and tyre so that

heavily laden axles will have smaller rolling radii wheels and therefore complete more revolutions over a given distance than lightly laden axles.

5 Unequal load distribution between axles when accelerating and braking will produce a variation of wheel effective rolling radius.

6 oss of grip between pairs of road wheels produces momentary wheel spin and hence speed differences between axles.

7.6. *Benefits of a third differential* (Fig. 7.26)

perating a third differential between front and rear wheel drive axles or rear tandem axles has certain advantages:

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1 The third differential equally divides driving torque and provides speed differentiation between both final drive axles so that the relative torque and speed per axle are better able to meet the individual road wheel requirements, thereby minimizing tyre distortion and scrub.
2 Transmission torsional wind-up between axles is minimized (Fig. 7.26) since driving and reaction torques within each axle are not opposing but are permitted to equalize themselves through the third differential.
3 dd tyres with different diameters are interchangeable without transmission wind-up.
4 Tractive effect and tyre grip is shared between four wheels so that wheel traction will be more evenly distributed. Therefore the amount of tractive effect per wheel necessary to propel a vehicle can be reduced.
5 Under slippery, snow or ice conditions, the third differential can generally be locked-out so that if one pair of wheels should lose traction, the other pair of wheels are still able to transmit traction.

7.6.3 *nter a le ith third differential*

escription of forward rear drive a le (Fig. 7.27) A third differential is generally incorporated in the forward rear axle of a tandem bogie axle drive layout because in this position it can be conveniently arranged to extend the drive to the rear axle (Fig. 7.27).
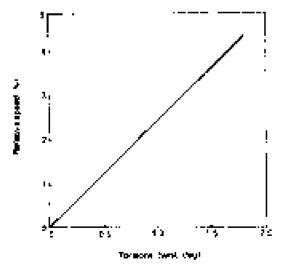
Fig. 7.26 Relationship of relative speeds of double drive axles and the amount of transmission twist

Power from the gearbox propellor shaft drives the axle input shaft. Support for this shaft is provided by a ball race mounted in the casing at the flanged end and by a spigot bearing built into the integral sun gear and output shaft at the other end. Bevel planet pinions supported on the cross-pin spider splined to the input shaft divide the drive between both of the bevel sun gears. The left hand sun gear is integral with the input helical gear and is free to rotate relative to the input shaft which it is mounted on, whereas the right hand bevel sun gear is integral with the output shaft. This output shaft is supported at the differential end by a large taper roller bearing and by a much smaller parallel roller bearing at the opposite flanged output end.

A tandem axle transmission arrangement is shown in Fig. 7.28(a) where $_1$, $_2$ and $_3$ represent the first axle, second axle and inter axle differential respectively.

When power is supplied to the inter axle (forward rear axle) through the input shaft and to the bevel planet pinion via the cross-pin spider, the power flow is then divided between both sun gears. The drive from the left hand sun gear then passes to the input helical gear to the final drive bevel pinion helical gear where it is redirected at right angles by the crownwheel and pinion to the axle differential and half shafts.

At the same time the power flowing to the right hand sun gear goes directly to the output shaft flange where it is then transmitted to the rear axle via a pair of universal oints and a short propellor shaft.

Third differential action (Fig. 7.27) When both drive axles rotate at the same speed, the bevel planet pinions bridging the opposing sun gears bodily move around with the spider but do not revolve on their own axes. If one axle should reduce its speed relative to the other one, the planet pinions will start to revolve on their cross-pins so that the speed lost by one sun gear relative to the spider s input speed will be gained by the other sun gear.

Therefore the third differential connecting the two axles permits each axle mean speed to automatically ad ust itself to suit the road operating conditions without causing any torsional wind-up between axle drives.

Third differential lock-out (Fig. 7.27) For providing maximum traction when road conditions are unfavourable such as driving over soft, slippery or steep ground, a differential lock-out clutch is incorporated. When engaged this device couples
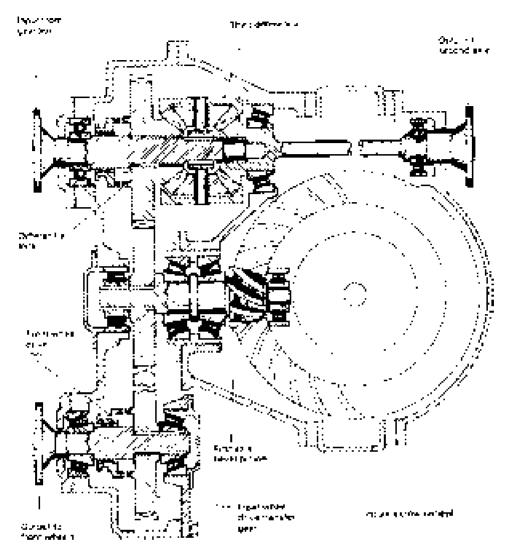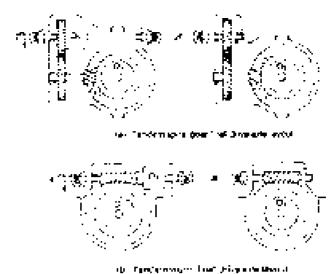
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Fig. 7.27 Final drive with third differential and lock and optional transfer gearing for front

the input shaft directly with the input helical gear and left hand bevel sun gear so that the differential planet pinions are prevented from equally dividing the input torque between the two axles at the expense of axle speed differentiation. Consequently, when the third differential is locked out each axle is able to deliver independently to the other axle tractive effect which is only limited by the grip between the road wheels and the quality of surface it is being driven over. It should be observed that when the third differential lock-out is engaged the vehicle should only be operated at slow road speeds, otherwise excessive transmission wind-up and tyre wear will result.

ront wheel drive transfer gear take-up (Fig. 7.28) An additional optional feature is the transfer gear take-up which is desirable for on-off highway applications where the ground can be rough and uneven. With the front wheel drive lock clutch engaged, 25% of the total input torque from the gearbox will be transmitted to the front steer drive axle, while the remainder of the input torque 75% will be converted into tractive effect by the tandem axles. Again it should be pointed out that this mode of torque delivery and distribution with the third differential locked-out must only be used at relatively low speeds.

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Fig. 7.28 (a and b) Tandem drive axle layout

7.6. Worm and Worm Wheel inter axle with third differential (Fig. 7.29)

Where large final drive gear reductions are required which may range from 5:1 to 9:1, either a double reduction axle must be used or alternatively a worm and worm wheel can provide a similar step down reduction. When compared with the conventional crownwheel and pinion final drive gear reduction the worm and worm wheel mechanical efficiency is lower but with the double reduction axle the worm and worm wheel efficiency is very similar to the latter.

Worm and worm wheel axles usually have the worm underslung when used on cars so that a very low floor pan can be used. For heavy trucks the worm is arranged to be overslung, enabling a large ground to axle clearance to be achieved.

When tandem axles are used, an inter axle third differential is necessary to prevent transmission wind-up. This unit is normally built onto the axle casing as an extension of the forward axle s worm (Fig. 7.29).

The worm is manufactured with a hollow axis and is mounted between a double taper bearing to absorb end thrust in both directions at one end and a parallel roller bearing at the other end which just sustains radial loads. The left hand sun gear is attached on splines to the worm but the right hand sun gear and output shaft are mounted on a pair of roller and ball bearings.

Power flow from the gearbox and propellor shaft is provided by the input spigot shaft passing through the hollow worm and coming out in the centre of the bevel gear cluster where it supports the internally
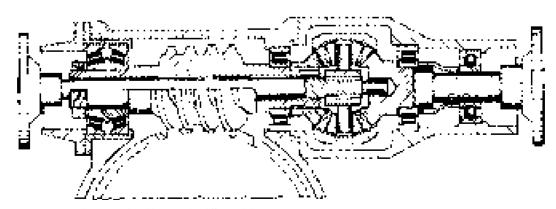

Fig. 7.29 Worm and worm wheel inter axle differential

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splined cross-pin spider and their corresponding planet pinions. Power is then split between the front axle (left hand) sun gear and worm and the rear axle (right hand) sun gear and output shaft, thus transmitting drive to the second axle.

Consequently if the two axle speeds should vary, as for example when cornering, the planet pinions will revolve on their axes so that the sun gears are able to rotate at speeds slightly above and below that of the input shaft and spider, but at the same time still equally divide the torque between both axles.

Fig. 7.28(b) shows the general layout of a tandem axle worm and worm wheel drive where $_1$, $_2$ and $_3$ represent the first axle, second axle and inter axle differentials respectively.

7.7   our wheel drive arrangements

7.7.1 Comparison of t o and four heel drives
The total force that a tyre can transmit to the road surface resulting from tractive force and cornering for straight and curved track driving is limited by the adhesive grip available per wheel.

When employing two wheel drive, the power thrust at the wheels will be shared between two wheels only and so may exceed the limiting traction for the tyre and condition of the road surface. With four wheel drive, the engine s power will be divided by four so that each wheel will only have to cope with a quarter of the power available, so that each individual wheel will be far below the point of transmitting its limiting traction force before breakaway (skid) is likely to occur.

uring cornering, body roll will cause a certain amount of weight transfer from the inner wheels to the outer ones. Instead of most of the tractive effort being concentrated on ust one driving wheel, both front and rear outer wheels will share the vertical load and driving thrust in proportion to the weight distribution between front and rear axles. Thus a four wheel drive (4W) when compared to a two wheel drive (2W) vehicle has a much greater margin of safety before tyre to ground traction is lost.

Transmission losses overall for front wheel drive (FW) are in the order of 10%, whereas rear wheel drive (RW) will vary from 10% in direct fourth gear to 13% in 1st, 2nd, 3rd, and 5th indirect gears. In general, overall transmission losses with four wheel drive (4W) will depend upon the transmission configuration and may range from 13% to 15%.

7.7. ndersteer and oversteer characteristics
(Figs 7.30 and 7.31)
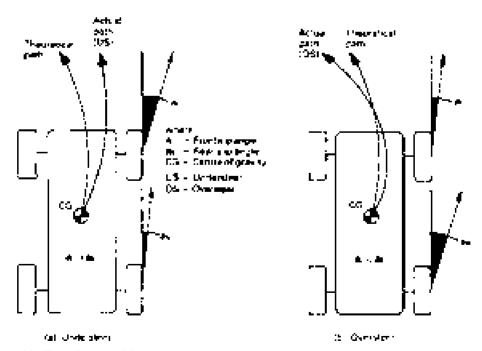In general, tractive or braking effort will reduce the cornering force (lateral force) that can be generated

Fig. 7.30 (a and b)   The influence of front and rear tyre slip angles on steering characteristics

for a given slip angle by the tyre. In other words the presence of tractive or braking effort requires larger slip angles to be produced for the same cornering force it reduces the cornering stiffness of the tyres. The ratio of the slip angle generated at the front and rear wheels largely determines the vehicle s tendency to oversteer or understeer (Fig. 7.30).

The ratio of the front to rear slip angles when greater than unity produces understeer,

i.e. Ratio $\dfrac{\Theta_F}{\Theta_R}$ 1.

When the ratios of the front to rear slip angles are less than unity oversteer is produced,

i.e. Ratio $\dfrac{\Theta_F}{\Theta_R}$ 1.

If the slip angle of the rear tyres is greater than the front tyres the vehicle will tend to oversteer, but if the front tyres generate a greater slip angle than the rear tyres the vehicle will have a bias to understeer.

Armed with the previous knowledge of tyre behaviour when tractive effort is present during cornering, it can readily be seen that with a rear wheel drive (RW) vehicle the tractive effort applied to propel the vehicle round a bend increases the slip angle of the rear tyres, thus introducing an oversteer effect. Conversely with a front wheel drive (FW) vehicle, the tractive effort input during a turn increases the slip angle of the front tyres so producing an understeering effect.

Experimental results (Fig. 7.31) have shown that rear wheel drive (RW) inherently tends to give oversteering by a small slightly increasing amount, but front and four wheel drives tend to understeer by amounts which increase progressively with speed, this tendency being slightly greater for the front wheel drive (FW) than for the four wheel drive (4W).

7.7.3 o er loss (Figs 7.32 and 7.33)
Tyre losses become greater with increasing tractive force caused partially by tyre to surface slippage. This means that if the total propulsion power is shared out with more driving wheels less tractive force will be generated per wheel and therefore less overall power will be consumed. The tractive force per wheel generated for a four wheel drive compared to a two wheel drive vehicle will only be half as great for each wheel, so that the overall tyre to road slippage will be far less. It has been found that the power consumed (Fig. 7.32) is least for the front wheel drive and greatest for the rear wheel drive, while the four wheel drive loss is somewhere in between the other two extremes.

The general relationship between the limiting tractive power delivered per wheel with either propulsion or retardation and the power loss at the wheels is shown to be a rapidly increasing loss as the power delivered to each wheel approaches the limiting adhesion condition of the road surface. Thus with a dry road the power loss is relatively small with
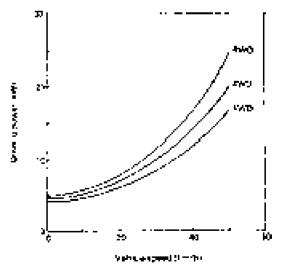

Fig. 7.31 Comparison of the over- and understeer tendency of R   F   and   cars on a curved track

Fig. 7.32 Comparison of the power re uired to drive R   F   and   cars on a curved track at various speeds
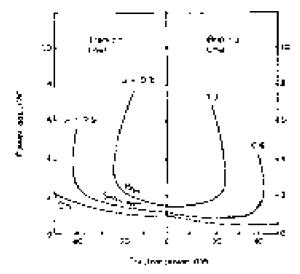
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Fig. 7.33 Relationship of tractive power and power loss for different road conditions
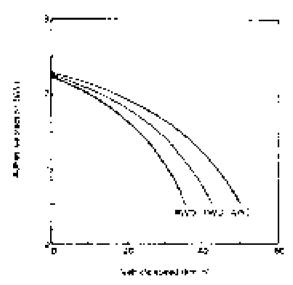

Fig. 7.34 Comparison of the adhesive traction available to rive R F and cars on a curved track at various speeds

increasing tractive power because the tyre grip on the road is nowhere near its limiting value. With semi-wet or wet road surface conditions the tyre s ability to maintain full grip deteriorates and therefore the power loss increases at a very fast rate (Fig. 7.33).

7.7. *a imum speed* (Fig. 7.34)

If friction between the tyre and road sets the limit to the maximum stable speed of a car on a bend, then the increasing centrifugal force will raise the cornering force (lateral force) and reduce the effective tractive effort which can be applied with rising speed (Fig. 7.34). The maximum stable speed a vehicle is capable of on a curved track is highest with four wheel drive followed in order by the front wheel drive and rear wheel drive.

7.7.5 *ermanent four heel drive transfer bo and and ange over* (Fig. 7.35)

Transfer gearboxes are used to transmit power from the gearbox via a step down gear train to a central differential, where it is equally divided between the front and rear output shafts (Fig. 7.35). Power then passes through the front and rear propellor shafts to their respective axles and road wheels. Both front and rear coaxial output shafts are offset from the gearbox input to output shafts centres by 230 mm.

The transfer box has a low ratio of 3.32:1 which has been found to suit all vehicle applications. The high ratio uses alternative 1.003:1 and 1.667:1 ratios to match the Range Rover and and Rover requirements respectively. This two stage reduction unit incorporates a three shaft six gear layout inside an aluminium housing. The first stage reduction from the input shaft to the central intermediate gear provides a 1.577:1 step down. The two outer intermediate cluster gears mesh with low and high range output gears mounted on an extension of the differential cage.

rive is engaged by sliding an internally splined sleeve to the left or right over dog teeth formed on both low and high range output gears respectively. Power is transferred from either the low or high range gears to the differential cage and the bevel planet pinions then divide the torque between the front and rear bevel sun gears and their respective output shafts. Any variation in relative speeds between front and rear axles is automatically compensated by permitting the planet pinions to revolve on their pins so that speed lost by one output shaft will be equal to that gained by the other output shaft relative to the differential cage input speed.

A differential lock-out dog clutch is provided which, when engaged, locks the differential cage directly to the front output shaft so that the bevel gears are unable to revolve within the differential cage. Consequently the front and rear output shafts are compelled to revolve under these conditions at the same speed.
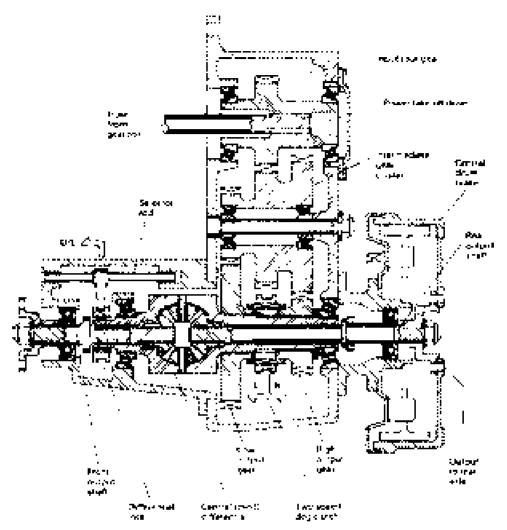
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Fig. 7.35 ermanent and and Range Rover type of transfer box

A power take-off coupling point can be taken from the rear of the integral input gear and shaft. There is also a central drum parking brake which locks both front and rear axles when applied.

It is interesting that the low range provides an overall ratio down to 40:1, which means that the gearbox, transfer box and crownwheel and pinion combined produce a gear reduction for gradient ability up to 45 .

7.7.6 Third central differential ith viscous coupling

escription of third differential and viscous coupling (Fig. 7.36) The gearbox mainshaft provides the input of power to the third differential (sometimes

referred to as the central differential). This shaft is splined to the planet pinion carrier (Fig. 7.36). The four planet pinions are supported on the carrier mesh on the outside with the internal teeth of the annulus ring gear, while on the inside the teeth of the planet pinions mesh with the sun gear teeth. A hollow shaft supports the sun gear. This gear transfers power to the front wheels via the offset input and output sprocket wheel chain drive. The power path is then completed by way of a pro-pellor shaft and two universal oints to the front crownwheel and pinion. Mounted on a partially tubular shaped carrier is the annulus ring gear which transfers power from the planet pinions directly to the output shaft of the transfer box unit. ere the power is conveyed to the rear axle
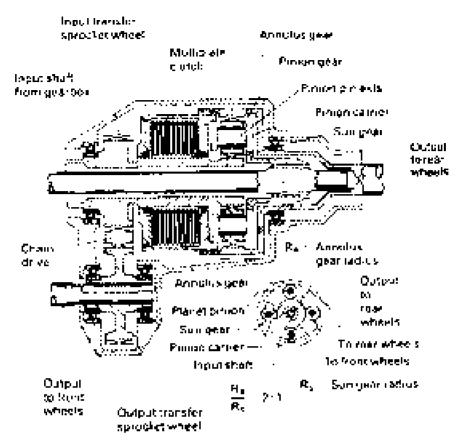
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Fig. 7.36 Third differential with viscous coupling

by a conventional propellor shaft and coupled at either end by a pair of universal oints.

Speed balance of third differential assembly with common front and rear wheel speed (Fig. 7.36) Power from the gearbox is split between the sun gear, taking the drive to the front final drive. The annulus gear conveys power to the rear axle. When the vehicle is moving in the straight ahead direction and all wheels are rotating at the same speed, the whole third differential assembly (the gearbox mainshaft attached to the planet carrier), planet pinions, sun gear and annulus ring gear will all revolve at the same speed.

Tor ue distribution with common front and rear wheel speed (Fig. 7.36) While rear and front pro- pellor shafts turn at the same speed, the torque split will be 66% to the rear and 34% to the front, determined by the 2:1 leverage ratio of this parti-

cular epicyclic gear train. This torque distribution is achieved by the ratio of the radii of the meshing teeth pitch point of both planet to annulus gear and planet to sun gear from the centre of shaft rotation. Since the distance from the planet to annulus teeth pitch point is twice that of the planet to sun teeth pitch point, the leverage applied to the rear wheel drive will be double that going to the front wheel drive.

Viscous coupling action (Fig. 7.36) Built in with the epicyclic differential is a viscous coupling resem- bling a multiplate clutch. It comprises two sets of mild steel disc plates one set of plates are splined to the hollow sun gear shafts while the other plates are splined to a drum which forms an extension to the annulus ring gear. The sun gear plates are disfigured by circular holes and the annulus drum plates have radial slots. The space between ad acent plates is filled with a silicon fluid. When the front and rear road wheels are moving at slightly different

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speeds, the sun and annulus gears are permitted to revolve at speeds relative to the input planet carrier speed and yet still transmit power without causing any transmission wind-up.

Conversely, if the front or rear road wheels should lose traction and spin, a relatively large speed difference will be established between the sets of plates attached to the front drive (sun gear) and those fixed to the rear drive (annulus gear). Immediately the fluid film between pairs of ad acent plate faces shears, a viscous resisting torque is generated which increases with the relative plate speed. This opposing torque between plates produces a semi-lock-up reaction effect so that tractive effort will still be maintained by the good traction road wheel tyres. A speed difference will always exist between both sets of plates when slip occurs between the road wheels either at the front or rear. It is this speed variation that is essential to establish the fluid reaction torque between plates, and thus prevent the two sets of plates and gears (sun and annulus) from racing around relative to each other. Therefore power will be delivered to the axle and road wheels retaining traction even when the other axle wheels lose their road adhesion.

7.7.7 *ongitudinal mounted engine ith integral front final drive four heel drive layout* (Fig. 7.37)

The power flow is transmitted via the engine to the five speed gearbox input primary shaft. It then transfers to the output secondary hollow shaft by way of pairs of gears, each pair combination having different number of teeth to provide the necessary range of gear ratios (Fig. 7.37). The hollow secondary shaft extends rearwards to the central differential cage. Power is then divided by the planet

pinions between the left and right hand bevel sun gears. alf the power flows to the front crownwheel via the long pinion shaft passing through the centre of the secondary hollow output shaft while the other half flows from the right hand sun gear to the rear axle via the universal oints and propellor shaft.

When the vehicle is moving forward in a straight line, both the front and rear axles rotate at one common speed so that the axle pinions will revolve at the same speed as the central differential cage. Therefore the bevel gears will rotate bodily with the cage but cannot revolve relative to each other.

Steering the vehicle or moving onto a bend or curved track will immediately produce unequal turning radii for both front and rear axles which meet at some common centre (instantaneous centre). Both axles will be compelled to rotate at slightly different speeds. ue to this speed variation between front and rear axles, one of the central differential sun gears will tend to rotate faster than its cage while the other one will move correspondently slower than its cage. As a result, the sun gears will force the planet pinions to revolve on their pins and at the same time revolve bodily with the cage. This speed difference on both sides of the differential is automatically absorbed by the revolving planet pinions now being permitted to move relative to the sun gears by rolling on their toothed faces. By these means, the bevel gears enable both axles to rotate at speeds demanded by their instantaneous rolling radii at any one moment without causing torsional wind-up. If travelling over very rough, soft, wet or steep terrain, better traction may be achieved with the central differential locked-out.
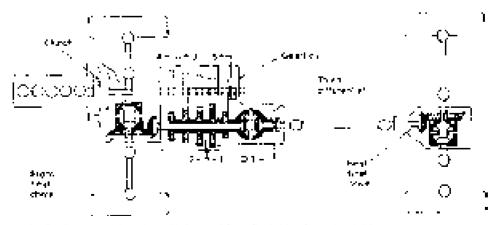

Fig. 7.37 ongitudinally mounted engine with integral front final drive four wheel drive system
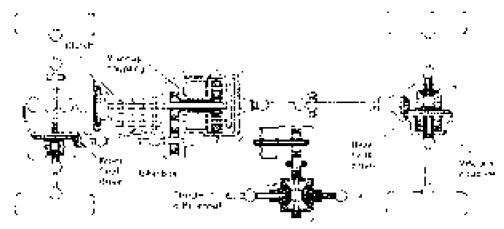
Fig. 7.38 ongitudinally mounted engine with independent front final drive four wheel drive system

7.7. ongitudinal mounted engine ith independent front a le four heel drive layout (Fig. 7.38)

Epicyclic gear central differential (Fig. 7.38) A popular four wheel drive arrangement for a front longitudinally mounted engine has a transfer box behind its five speed gearbox. This incorporates a viscous coupling and an epicyclic gear train to split the drive torque, 34% to the front and 66% to the rear (Fig. 7.38). A chain drives a forward facing drive shaft which provides power to the front differential mounted beside the engine sump. The input drive from the gearbox mainshaft directly drives the planet carrier and pinions. Power is diverted to the front axle through the sun gear and then flows to the hollow output shaft to the chain sprockets. utput to the rear wheels is taken from the annulus ring gear and carrier which transmits power directly to the rear axle. To minimize wheel spin between the rear road wheels a combined differential and viscous coupling is incorporated in the rear axle housing.

evel gear central differential (Fig. 7.38) In some cases vehicles may have a weight distribution or a cross-counting application which may find 50/50 torque split between front and rear wheel drives more suitable than the 34/66 front to rear torque split. To meet these requirements a conventional central (third) bevel gear differential may be preferred, see insert in Fig. 7.38. Again a transfer box is used behind the gearbox to house the offset central differential and transfer gears. The transfer gear train transmits the drive from the gearbox mainshaft to the central differential cage. Power

then passes to the spider cross-pins which support the bevel planet pinions. ere the torque is distributed equally between the front and rear bevel sun gears, these being connected indirectly through universal oints and propeller shafts to their respective axles. When the vehicle is moving along a straight path, the planet pinions do not rotate but ust revolve bodily with the cage assembly.

Immediately the vehicle is manoeuvred or is negotiating a bend, the planet pinions commence rotating on their own pins and thereby absorb speed differences between the two axles by permitting them not only to turn with the cage but also to roll round the bevel sun gear teeth at the same differential. owever, they are linked together by bevel gearing which permits them independently to vary their speeds without torsional wind-up and tyre scuffing.

7.7. Transversely mounted engine ith four heel drive layout (Fig. 7.39)

ne method of providing four wheel drive to a front transversely mounted engine is shown in Fig. 7.39. A 50/50 torque split is provided by an epicyclic twin planet pinion gear train using the annulus ring gear as the input. The drive to the front axle is taken from the central sun gears which is attached to the front differential cage, while the rear axle is driven by the twin planet pinions and the crownwheel, which forms the planet carrier. Twin planet pinions are used to make the sun gear rotate in the same direction of rotation as that of the annulus gear. A viscous coupling is incorporated in the front axle differential to provide a measure of wheel spin control.

Power from the gearbox is transferred to the annulus ring gear by a pinion and wheel, the ring
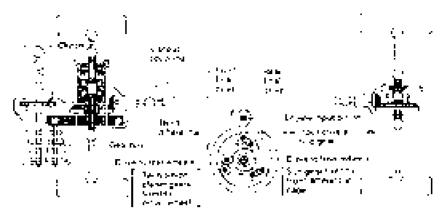
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Fig. 7.39 Transversely mounted engine four wheel drive system
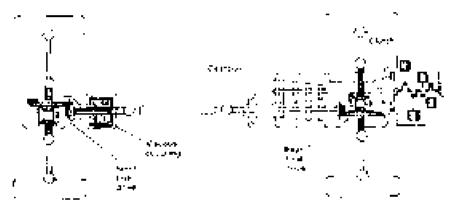

Fig. 7.40 Rear mounted engine four wheel drive system

gear having external teeth to mesh with the input pinion from the gearbox and internal teeth to drive the twin planet gears. Rotation of the annulus ring gear drives the outer and inner planet pinions and subsequently rotates the planet carrier (crownwheel in this case). The front crownwheel and pinion redirect the drive at right angles to impart motion to the propellor shaft. Simultaneously the inner planet pinion meshes with the central sun gear so that it also relays motion to the front differential cage.

7.7.10 Rear mounted engine four wheel drive layout (Fig. 7.40)

This arrangement has an integral rear engine and axle with the horizontal opposed four cylinder engine mounted longitudinally to the rear of the drive shafts and with the gearbox forward of the drive shafts (Fig. 7.40). Power to the rear axle is taken directly from the gearbox secondary output shaft to the crownwheel and pinion through 90 to the wheel hubs. Similarly power to the front axle is taken from the front end of the gearbox secondary output shaft to the front axle assembly comprised of the crownwheel and pinion differential and viscous coupling.

The viscous relative speed-sensitive fluid coupling has two independent perforated and slotted sets of steel discs. One set is attached via a splined shaft to a stub shaft driven by the propellor shaft from the gearbox, the other to the bevel pinion shank of the front final drive. The construction of the multi-interleaf discs is similar to a multiplate clutch but there is no engagement or release mechanism. Discs always remain equidistant from each other and power transmission is only by the silicon fluid which stiffens and produces a very positive fluid drag between plates. The sensitivity and effectiveness of the transference of torque is dependent upon the diameter and number of plates (in this case 59 plates), size of

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perforated holes and slots, surface roughness of the plates as well as temperature and generated pressure of fluid.

The drive to the front axle passes through the viscous coupling so that when both front and rear axle speeds are similar no power is transmitted to the front axle. Inevitably, in practice small differences in wheel speeds between front and back due to variations of effective wheel radii (caused by uneven load distribution, different tyre profiles, wear and cornering speeds) will provide a small degree of continuous drive to the front axle. The degree of speed sensitivity is such that it takes only one eighth of a turn in speed rotational difference between each end of the coupling for the fluid to commence to stiffen. nly when there is a loss of grip through the rear wheels so that they begin to slip does the mid-viscous coupling tend to lock-up to provide positive additional drive to the front wheels. A mechanical differential lock can be incorporated in the front or rear axles for travelling over really rough ground.

7. Electro-hydraulic limited slip differential

A final drive differential allows the driving wheels on each side of a vehicle to revolve at their true rolling speed without wheel slip when travelling along a straight uneven surface, a winding road or negotiating a sharp corner. If the surface should be soft, wet, muddy, or slippery for any other reason, then one or the other or even both drive wheels may lose their tyre to ground traction, the vehicle will then rely on its momentum to ride over these patchy slippery low traction surfaces. owever if the vehicle is travelling very slowly and the ground surface is particularly uneven, soft or slippery, then loss of traction of one of the wheels could easily be sufficient to cause the wheel to spin and therefore to transmit no drive. Unfortunately, due to the inherent design of the bevelled gear differential the traction delivered at the good gripping wheel will be no more than that of the tyre that has lost its grip. A conventional bevelled gear differential requires that each sun (side) gear provides equal driving torque to each wheel and at the same time opposite sun (side) gears provide reaction torque equal to the driving torque of the opposite wheel. Therefore as soon as one wheel loses ground traction its opposite wheel, even though it may have a firm tyre to ground contact, is only able to produce the same amount of effective traction as the wheel with limited ground grip.

7. .1 Description of multiplate clutch mechanism
(Fig. 7.41)

To overcome this deficiency a multiplate wet clutch is incorporated to one side of the differential cage, see Fig. 7.41. ne set of the clutch plates have internal spin teeth which mesh with splines formed externally on an extended sun (side) gear, whereas the other set of inter disc plates have external spline teeth which mesh with internal splines formed inside the differential cage. Thus, when there are signs of any of the wheels losing their grip the clutch plates are automatically clamped partially or fully together. The consequence of this is to partially or fully lock both left and right hand side output drive shafts together so that the loss of drive of one drive wheel will not affect the effectiveness of the other wheel. To activate the engagement and release of the multiplate clutch, a servo-piston mounted on the right hand side bearing support flange is used: the piston is stepped and has internal seals for each step so that hydraulic fluid is trapped between the internal and external stepped piston and bearing support flange respectively.

7. . perating conditions

ormal differential action (Fig. 7.41) With good road wheel grip the multiplate clutch is disengaged by closing the delivery solenoid valve and releasing fluid to the reservoir tank via the open return solenoid valve. Under these conditions when there is a difference in speed between the inner and outer road wheels, the bevel-planet pinions are free to revolve on their axes and hence permit each sun (side) gear to rotate at the same speed as its ad a-cent road wheel thereby eliminating any final drive transmission windup and tyre scrub.

ne wheel on the threshold of spinning (Fig. 7.41) If one wheel commences to spin due to loss of traction the wheel speed sensor instantly detects the wheel s acceleration and signals the ECU the computer then processes this information and taking into account that a small amount of slip improves the tyre to ground traction will then energize and de-energize the delivery and return solenoid valves respectively. Fluid will now be pumped from the power assistant steering systems pump to the servo-piston, the pressure build up against the piston will engage and clamp the multiplate clutch via the thrust-pins and plate so that the differential
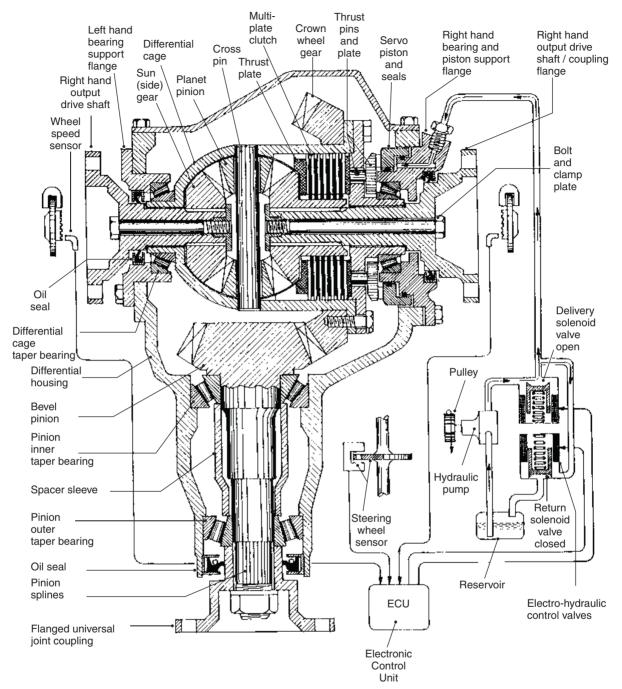
Fig. 7.41 Electro-hydraulic limited-slip (differential in locked position)

cage now is able to provide the reaction torque for the other wheel still delivering traction to the ground.

The ECU is able to take into account the speed of the vehicle and if the vehicle is turning gently or sharply which is monitored by the individual brake speed sensors and the steering wheel accelerator sensor. These two sensors therefore indirectly control the degree of lock-up which would be severe when pulling away from a standstill but would ease

up with increased vehicle speed and when negotiating a bend.

7.9 Tyre grip when braking and accelerating with good and poor road surfaces (Fig. 7.42)

The function of the tyre and tread is to transfer the accelerating and decelerating forces from the wheels to the road. The optimum tyre grip is achieved when there is about 15–25% slip between the tyre tread and road under both accelerating and decelerating driving conditions, see Fig. 7.42.

Tyre grip is a measure of the coefficient of friction (μ) generated between the tyre and road surface at any instant, this may be defined as $\mu = F/W$ where F is the frictional force and W is the perpendicular force between the tyre and road. If the frictional and perpendicular forces are equal ($\mu = 1.0$) the tyre tread is producing its maximum grip, whereas if $\mu = 0$ then the grip between the tyre and road is zero, that is, it is frictionless. Typical tyre to road coefficient for a good tarmac dry and wet surface would be 1.0 and 0.7 respectively, conversely for poor surfaces such as soft snow covered roads the coefficient of friction would be as low as 0.2.

Wheel slip for accelerating and decelerating is usually measured as the slip ratio or the percentage of slip and may be defined as follows:

$$\text{accelerating slip ratio} = \text{road speed/tyre speed}$$

$$\text{decelerating (braking) slip ratio} = \text{tyre speed/road speed}$$

where the tyre speed is the linear periphery speed.

Note the percentage of slip may be taken as the slip ratio $\times$ 100. There is no slippage or very little that takes place between the tyre and road surface when a vehicle is driven at a constant speed along a dry road, under these conditions the slip ratio is zero (slip ratio = 0). Conversely heavy acceleration or braking may make the wheels spin or lock respectively thus causing the slip ratio to approach unity (slip ratio = 1.0).

If the intensity of acceleration or deceleration is increased the slip ratio tends to increase since during acceleration the wheels tend to slip and in the
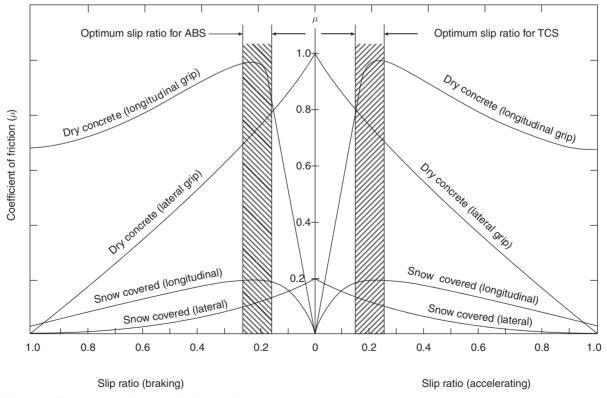

Fig. 7.42 Tyre grip as a function of slip ratio for various driving conditions

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extreme spin, whereas during deceleration (braking) the wheels tend to move slower than the vehicle speed and under very heavy braking will lock, that is stop rotating and ust slide along the surface. When considering the relationship between tyre slip and grip it should be observed that the tyre grip measurements are in two forms, longitudinal (lengthways) forces and lateral (sideways) forces, see Fig. 7.42. In both acceleration and deceleration in the longitudinal direction mode a tyre tends to produce its maximum grip (high μ) with a slip ratio of about 0.2 and as the slip ratio decreases towards zero, the tyre grip falls sharply however, if the slip ratio increases beyond the optimum slip ratio of 0.2 the tyre grip will tend to decrease but at a much slower rate. With lateral direction grip in terms of sideways force coefficient of friction, the value of μ (grip) is much lower than for the forward rolling frictional grip and the maximum grip (high μ) is now produced with zero tyre slip. Traction control systems (TCS) respond to wheel acceleration caused by a wheel spinning as its tyre loses its grip with the road surface, as opposed to antilock braking systems (ABS) which respond to wheel deceleration caused by a wheel braking and preventing the wheel from turning and in the limit making it completely lock.

7.1 Traction control system

With a conventional final drive differential the torque output from each driving wheel is always equal. Thus if one wheel is driven over a slippery patch, that wheel will tend to spin and its ad acent sun (side) gear will not now be able to provide the reaction torque for the other (opposite) sun (side) gear and driving road wheel. Accordingly, the output torque on the other wheel which still has a good tyre to surface grip will be no more than that of the slipping wheel and it doesn t matter how much the driver accelerates to attempt to regain traction, there still will be insufficient reaction torque on the spinning side of the differential for the good wheel to propel the vehicle forward.

ne method which may be used to overcome this loss of traction when one wheel loses its road grip, is to simply apply the wheel brake of the wheel showing signs of spinning so that a positive reaction torque is provided in the differential this counteracts the delivery to the good wheel of the half share of the driving torque being supplied by the combined engine and transmission system in terms of tractive effort between the tyre and ground.

To achieve this traction control an electronic control unit (ECU) is used which receives signals from individual wheel speed sensors, and as soon as one of the driving wheels tends to accelerate (spin due to loss of tyre to ground traction), the sensor s generated voltage change is processed by the ECU computer, and subsequently current is directed to the relevant traction solenoid valve unit so that hydraulic brake pressure is transmitted to the brake of the wheel about to lose its traction. As soon as the braked wheel s speed has been reduced to a desirable level, then the ECU signals the traction solenoid valve unit to release the relevant wheel brake.

7.10.1 Description of system (Fig. 7.43)

This traction control system consists of: an electric motor driven hydraulic pump which is able to generate brake pressure independently to the foot brake master cylinder and a pressure storage accumulator a traction boost unit which comprises a cylinder housing, piston and poppet valve, the purpose of which is to relay hydraulic pressure to the appropriate wheel brake caliper and at the same time maintain the traction boost unit circuit fluid separate from the foot brake master cylinder fluid system a pair of traction solenoid valve units each having an outlet and return valve regulates the cut-in and -out of the traction control an electronic control unit (ECU) is provided and individual wheel speed sensors which monitor the acceleration of both driven and non-driving wheels. Should the speed of either of the driven wheels exceed the mean wheel speed of the non-driven wheels by more than about 20%, then the ECU will automatically apply the appropriate wheel brake via the traction solenoid valves and traction boost device.

7.10. perating conditions

oot brake applied (Fig. 7.43(a)) With the foot pedal released brake fluid is able to flow freely between the master cylinder and both brake calipers via the open traction boost unit. The boost piston will be in its outer-most position thereby holding the poppet valve in its fully open position. When the foot pedal is pushed down, brake fluid pressure will be transmitted though the fluid via the traction boost poppet valve to both of the wheel calipers thus causing the brakes to be applied.

Traction control system applied (off-side slipping wheel braked) (Fig. 7.43(b)) When one of the driving wheels begins to spin (off-side wheel in this example) the ad acent speed sensor voltage change signals the ECU, immediately it computes
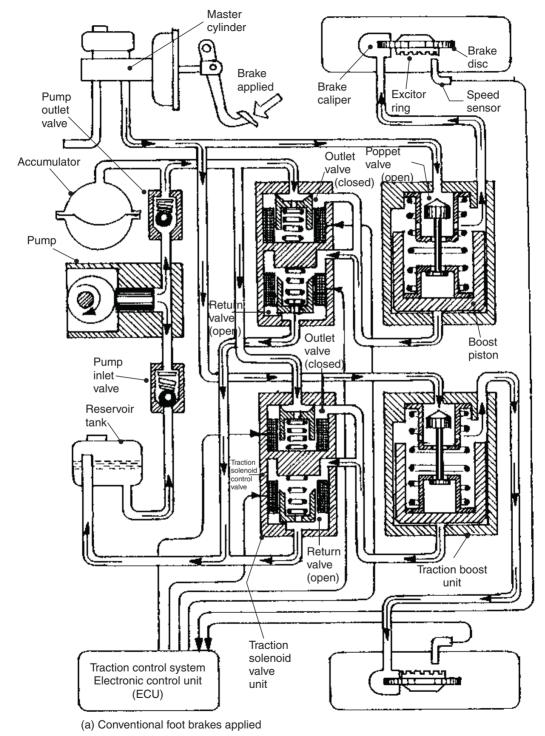
(a) Conventional foot brakes applied

Fig. 7.43 (a and b) Traction control system

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Wheel tending
to spin

Off-side wheel

MC

Foot
brake
off

BC   ER   SS

PIV

A

OVC

PVC

P

RVC

OVC

PIV

TBC

R

RVO

TBU

TSCV

TCS
ECU

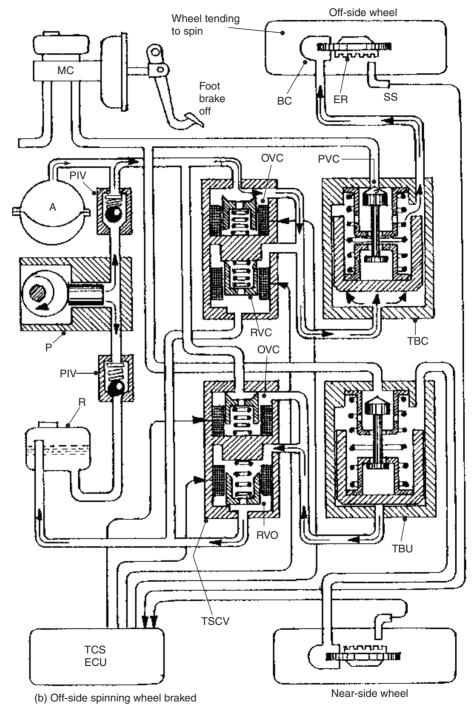(b) Off-side spinning wheel braked
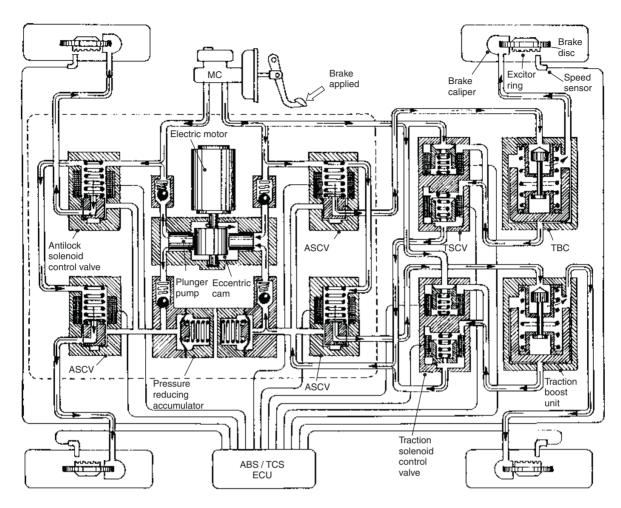
Near-side wheel

Fig. 7.43 *contd*

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Fig. 7.44 Combined antilock brake system/traction control system (brakes applied)

and relays current to the relevant traction solenoid valve unit to energize both valves, this closes the return valve and opens the outlet valve. Fluid pressure from the accumulator now flows through the open outlet valve and passes into the traction boost cylinder, the upward movement of the piston will instantly snap closed the poppet valve thereby trapping fluid between the upper side of the cylinder and piston chamber and the off-side caliper. The fluid pressure build up underneath the piston will pressurize the fluid above the piston so that the pressure increase is able to clamp the caliper pads against the brake disc.

As the wheel spin speed reduces to a predetermined value the monitoring speed sensor signals the ECU to release the wheel brake, immediately the solenoid outlet and return valves will be de-energized, thus causing them to close and open respectively.

Fluid pressure previously reaching the boost piston will now be blocked and the fluid underneath the piston will be able to return to the reservoir tank. The same cycle of events will take place for the near-side wheel if it happens to move over a slippery surface.

7.10.3 Combined ABS/TCS arrangement
(Fig. 7.44)

Normally a traction control system (TCS) is incorporated with the antilock braking system (ABS) so that it can share common components such as the electric motor, pump, accumulator, wheel brake sensors and high pressure piping. As can be seen in Fig. 7.44 a conventional ABS system described in section 11.7.2 has been added to. This illustration shows when the brakes are applied fluid pressure is transmitted indirectly through the antilock

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solenoid control valve to the front brake calipers however, assuming a rear wheel drive, fluid pressure also is transmitted to the rear brakes via the antilock solenoid control valve and then through the traction boost unit to the wheel brake calipers thus applying the brakes.

S operating (Fig. 7.44) When the wheel brake speed sensor signals that a particular wheel is tending towards wheel lock, the appropriate antilock solenoid control valve will be energized so that fluid pressure to that individual wheel brake is blocked and the entrapped fluid pressure is released to the pressure reducing accumulator (note Fig. 7.44 only shows the system in the foot brake applied position).

T S operating (Fig. 7.44) If one of the wheel speed sensors signals that a wheel is moving towards slip and spin the respective traction solenoid control valve closes its return valve and opens its outlet valve fluid pressure from the pump now provides the corresponding boost piston with an outward thrust thereby causing the poppet valve to close (note Fig. 7.44 only shows the system in the foot brake applied position). Further fluid pressure acting on the head of the piston now raises the pressure of the trapped fluid in the pipe line between the boost piston and the wheel caliper. Accordingly the relevant drive wheel is braked to a level that transmits a reaction torque to the opposite driving wheel which still retains traction.

ne limitation of a brake type traction control is that a continuous application of the TCS when driving over a prolonged slippery terrain will cause the brake pads and disc to become excessively hot it thus may lead to brake fade and a very high wear rate of the pads and disc.

Tyres

.1 Tractive and braking properties of tyres

.1.1 Tyre grip

Tyres are made to grip the road surface when the vehicle is being steered, accelerated, braked and/or negotiating a corner and so the ability to control the tyre to ground interaction is of fundamental importance. Road grip or friction is a property which resists the sliding of the tyre over the road surface due to a retardant force generated at the tyre to ground contact area. The grip of different tyres sliding over various road surface finishes may be compared by determining the coefficient of friction for each pair of rubbing surfaces.

The coefficient of friction may be defined as the ratio of the sliding force necessary to steadily move a solid body over a horizontal surface to the normal reaction supporting the weight of the body on the surface (Fig. 8.1).

i.e. $\quad$ Coefficient of friction $(\mu) = \dfrac{\text{Frictional force}}{\text{Normal reaction}}$

$$= \frac{F}{W}$$

where μ = coefficient of friction
$\quad\quad F$ = frictional force (N)
$\quad\quad W$ = normal reaction (weight of body) (N)

Strictly speaking, the coefficient of friction does not take into account the surface area tread pattern which maximizes the interlocking mechanism
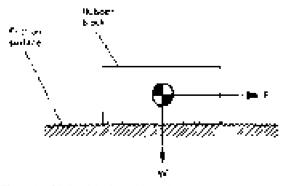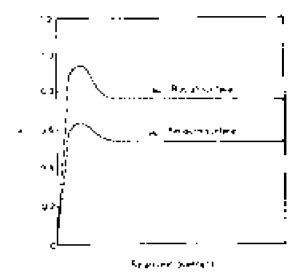

Fig. 8.1 Sliding block and board

Fig. 8.2 ariation of friction with relative movement

between the flexible tread elements and road. Therefore when dealing with tyres it is usual to refer to the $\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad$ The maximum coefficient of adhesive friction created between a sliding tread block and a solid surface occurs under conditions of slow movement or creep (Fig. 8.2). This critical stage is known as the $\quad\quad\quad\quad$, μ, and if the relative movement of the rubber on the surface is increased beyond this point the friction coefficient falls. It continues to fall until bodily sliding occurs, this stage being known as the $\quad\quad\quad\quad\quad\mu$. Sliding friction characteristics are consistent with the behaviour of rolling tyres.

A modern compound rubber tyre will develop a higher coefficient of friction than natural rubber. In both cases their values decrease as the road surface changes from dry to a wet condition. The rate of fall in the coefficient of friction is far greater with a worn tyre tread as opposed to a new tyre as the degree of road surface wetness increases (Fig. 8.3).

It has been found that the frictional grip of a bald tyre tread on a rough dry road surface is as good or even better than that achieved with a new tread (Fig. 8.3). The reason for this unexpected result is due to the greater amount of rubber interaction with the ground surface for a given size of contact patch. It therefore develops a larger reaction force which
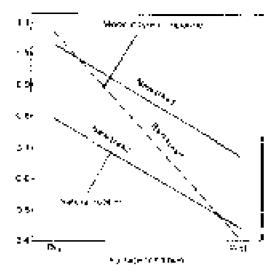
Fig. 8.3 Effect of surface condition on the coefficient of adhesive friction with natural and synthetic rubber using new and bald tyre treads

opposes the movement of the tyre. Under ideal road conditions and the amount of deformable rubber actually in contact with the road maximized for a given contact path area, the retarding force which can be generated between the tyre and ground can equal the vertical load the wheel supports. In other words, the coefficient of adhesive friction can reach a value of 1.0. owever, any deterioration in surface roughness due to surface ridges being worn, or chippings becoming submerged in asphalt, or the slightest amount of wetness completely changes the situation. A smooth bald tyre will not be able to grip the contour of the road, whereas the tyre with a good tread pattern will easily cope and maintain a relatively high value of retardant force.

When transmitting tractive or braking forces, the tyre is operating with slip or creep. It is believed that the maximum friction is developed when a maximum number of individual tread elements are creeping at or near an optimum speed relative to the ground. The distribution by each element of the tread is not equal nor is it uniform throughout the contact patch. The frictional forces developed depend upon the pressure distribution within the contact patch area and the creep effects. nce bodily slippage begins to occur in one region of the contact area, the progression to the fully sliding condition of the contact area as a whole is extremely rapid.

Under locked wheel conditions, the relative sliding speed between a tyre tread and the road is

the speed of the vehicle. If, however, the braking is such that the wheels are still rotating, the actual speed between the tyre tread and the road must be less than that of the vehicle. Even on surfaces giving good braking when wet, maximum coefficients occur at around 10–20% slip. This means that the actual speed between the tyre tread and the road is around one eighth of the vehicle speed or less. Under these conditions, it is possible to visualize that the high initial peak value occurs because the actual tyre ground relative speed relates to a locked wheel condition at a very low vehicle speed (Fig. 8.3).

The ability to utilize initial peak retardation under controlled conditions is a real practical asset to vehicle retardation and, because the tyre is still rolling, to vehicle directional control.

Braking effectiveness can therefore be controlled and improved if the wheels are prevented from completely locking in contrast to the wheels actually being locked when the brakes are applied. Thus when braking from different speeds (Fig. 8.4) it can be seen that the unlocked wheels produce a higher peak coefficient of adhesive friction as opposed to the locked condition which generates only a sliding coefficient of adhesive friction. In both situations the coefficient of adhesive friction decreases as the speed from which braking first commences increases.
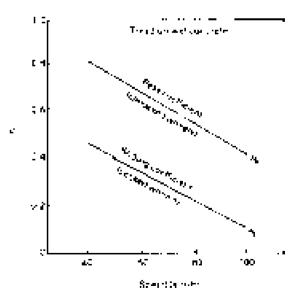

Fig. 8.4 Effect of speed on both peak and sliding coefficient of adhesive friction

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.1.   rip control

Factors influencing the ability of a tyre to grip the road when being braked are:

a) the vehicle speed,
b) the amount of tyre wear,
c) the nature of the road surface,
d) the degree of surface wetness.

(Fig. 8.4)   enerally as the speed of the vehicle rises, the time permitted for tread to ground retardation is reduced so that the grip or coefficient of adhesive friction declines (Fig. 8.4).

(Fig. 8.5)   As the tyre depth is reduced, the ability for the tread to drain off water being swept in front of the tread is reduced. Therefore with increased vehicle speed inadequate drainage will reduce the tyre grip when braking (Fig. 8.5).

(Fig. 8.6)   The reduction in tyre grip when braking from increased vehicle speed drops off at a much greater rate as the rainfall changes from light rain, producing a surface water depth of 1 mm, to a heavy rainstorm flooding the road to a water depth of about 2.5 mm (Fig. 8.6).

(Fig. 8.7)   A new tyre braked from various speeds will generate a higher peak coefficient of adhesive friction with a smaller fall off at the higher speeds on wet rough surfaces compared to braking on wet smooth surfaces (Fig. 8.7).
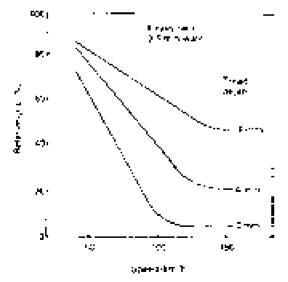

Fig. 8.5   Effect of speed on relative tyre grip with various tread depth when braking on a wet road
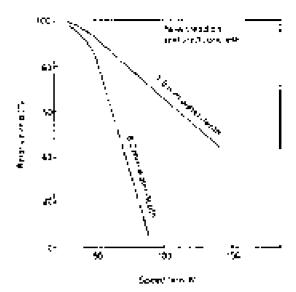

Fig. 8.6   Effect of speed on relative tyre grip with various road surface water depths
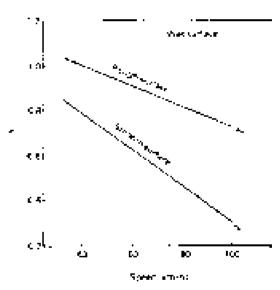

Fig. 8.7   Effect of speed on the coefficient of adhesive friction with both wet rough and smooth surfaces

The reduction in the coefficient of adhesive friction when braking with worn tyres on both rough and particularly smooth wet surfaces will be considerably greater.

.1.3   oad surface te ture (Fig. 8.8)

A road surface finish may be classified by its texture which may be broadly divided in
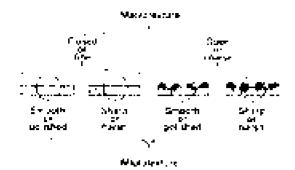
Fig. 8.8 Terminology and road surface texture

which represents the surface section peak to valley ripple or roughness, and which is a measure of the smoothness of the ripple contour (Fig. 8.8). Further subdivisions may be made macrotexture may range from closed or fine going onto open or coarse whereas microtexture may range from smooth or polished extending to sharp or harsh.

For good tyre grip under dry and wet conditions the road must fulfil two requirements. Firstly, it must have an open macrotexture to permit water drainage. Secondly, it should have a microtexture which is harsh the asperities of the texture ripples should consist of many sharp points that can pene-

trate any remaining film of water and so interact with the tread elements. If these conditions are fulfilled, a well designed tyre tread will provide grip not only under dry conditions but also in wet weather. A worn road surface may be caused by the hard chippings becoming embedded below the soft asphalt matrix or the microtexture of these chippings may become polished. In the case of concrete roads, the roughness of the brushed or mechanically ridged surface may become blunted and over smooth. To obtain high frictional grip over a wide speed range and during dry and wet conditions, it is essential that the microtexture is harsh so that pure rubber to road interaction takes place.

.1. Braking characteristics on wet roads (Fig. 8.9)

Maximum friction is developed between a rubber tyre tread and the road surface under conditions of slow movement or creep. A tyre s braking response on a smooth wet road with the vehicle travelling at a speed, say 100 km/h, will show the following characteristics (Fig. 8.9).

When the brakes are in the first instance steadily applied, the retardation rate measured as a fraction of the gravitational acceleration ($m s^2$) will rise rapidly in a short time interval up to about 0.5 g. This phase of braking is the normal mode of braking when driving on motorways. In traffic, it enables the
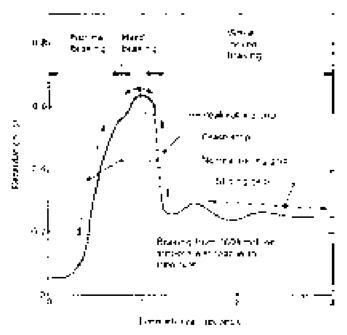

Fig. 8.9 ossible retardation braking cycle on a wet road

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driver to reduce the vehicle speed fairly rapidly with good directional stability and no wheel lock taking place. If an emergency braking application becomes necessary, the driver can raise the foot brake effort slightly to bring the vehicle retardation to its peak value of ust over 0.6 g, but then should immediately release the brake, pause and repeat this on-off sequence until the road situation is under control. Failing to release the brake will lock the wheels so that the tyre road grip changes from one of rolling to sliding. As the wheels are prevented from rotating, the braking grip generated between the contact patches of the tyres drops drastically as shown in the crash stop phase. If the wheels then remain locked, the retardation rate will steady at a much lower value of ust over 0.2 g. The tyres will now be in an entirely sliding mode, with no directional stability and with a retardation at about one third of the attainable peak value. With worn tyre treads the braking characteristics of the tyres will be similar but the braking retardation capacity is considerably reduced.

.1.5   olling resistance (Figs 8.10 and 8.11)

When a loaded wheel and tyre is compelled to roll in a given direction, the tyre carcass at the ground interface will be deflected due to a combination of the vertical load and the forward rolling effect on the tyre carcass (Fig. 8.10). The vertical load tends to flatten the tyre s circular profile at ground level, whereas the forward rolling movement of the wheel will compress and spread the leading contact edge and wall in the region of the tread. At the same time, the trailing edge will tend to reduce its contact pressure and expand as it is progressively freed from the ground reaction. The consequences of the continuous distortion and recovery of the tyre carcass at ground level means that energy is being used in rolling the tyre over the ground and it is not
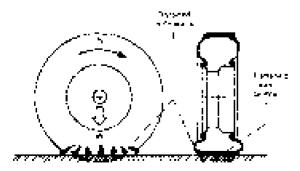
all returned as strain energy as the tyre takes up its original shape. (Note that this has nothing to do with a tractive force being applied to the wheel to propel it forward.) Unfortunately when the carcass is stressed, the strain produced is a function of the stress. n releasing the stress, because the tyre material is not perfectly elastic, the strain lags behind so that the strain for a given value of stress is greater when the stress is decreasing than when it is increasing. Therefore, on removing the stress completely, a residual strain remains. This is known as hysteresis and it is the primary cause of the rolling resistance of the tyre.

The secondary causes of rolling resistance are air circulation inside the tyre, fan effect of the rotating tyre by the air on the outside and the friction between the tyre and road caused by tread slippage. A typical analysis of tyre rolling resistance losses at high speed can be taken as 90–95% due to internal hysteresis, 2–10% due to friction between the tread and ground, and 1.5–3.5% due to air resistance.

Rolling resistance is influenced by a number of factors as follows:

a) cross-ply tyres have higher rolling resistance than radial ply (Fig. 8.11),
b) the number of carcass plies and tread thickness increase the rolling resistance due to increased hysteresis,
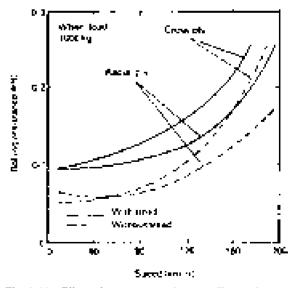c) natural rubber tyres tend to have lower rolling resistance than those made from synthetic rubber,

Fig. 8.10   Ilustration of side wall distortion at ground level

Fig. 8.11   Effect of tyre construction on rolling resistance

d) hard smooth dry surfaces have lower rolling resistances than rough or worn out surfaces,

e) the inflation pressure decreases the rolling resistance on hard surfaces,

f) higher driving speed increases the rolling resistance due to the increase in work being done in deforming the tyre over a given time (Fig. 8.11),

g) increasing the wheel and tyre diameter reduces the rolling resistance only slightly on hard surfaces but it has a pronounced effect on soft ground,

h) increasing the tractive effort also raises the rolling resistance due to the increased deformation of the tyre carcass and the extra work needed to be done.

.1.6 Tractive and bra ing effort (Figs 8.12, 8.13, 8.14, 8.15, 8.16 and 8.17)

A tractive effort at the tyre to ground interface is produced when a driving torque is transmitted to the wheel and tyre. The twisting of the tyre carcass in the direction of the leading edge of the tread contact patch is continuously opposed by the tyre contact patch reaction on the ground. Before it enters the contact patch region a portion of the tread and casing will be deformed and compressed.

ence the distance that the tyre tread travels when sub ected to a driving torque will be less than that in free rolling (Fig. 8.12).

If a braking torque is now applied to the wheel and tyre, the inertia on the vehicle will tend to pull the wheel forward while the interaction between the tyre contact patch and ground will oppose this motion. Because of this action, the casing and tread elements on the leading side of the tyre become stretched ust before they enter the contact patch region in contrast with the compressive effect for driving tyres (Fig. 8.13). As a result, when braking torque is applied the distance the tyre moves will be greater than when the tyre is sub ected to free rolling only. The loss or gain in the distance the tread
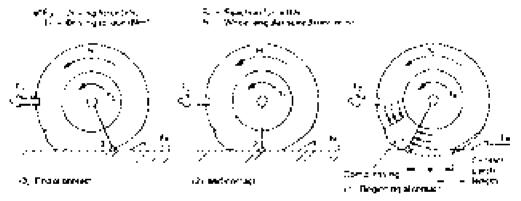

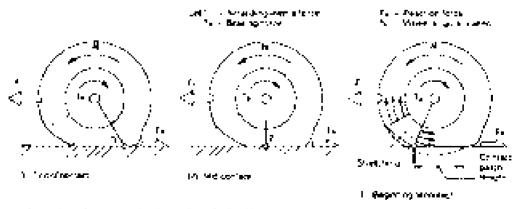**Fig. 8.12** eformation of a tyre under the action of a driving tor ue

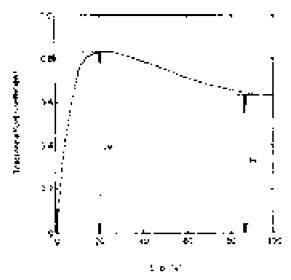**Fig. 8.13** eformation of a tyre under the action of a braking tor ue

Fig. 8.14 Effect of tyre slip on tractive effort

Fig. 8.15 Effect of ground surface on braking effort

travels under tractive or braking conditions relative to that in free rolling is known as
and it can be said that under steady state conditions slip is a function of tractive or braking effort.

When a driving torque is applied to a wheel and tyre there will be a steep initial rise in tractive force matched proportionally with a degree of tyre slip, due to the elastic deformation of the tyre tread. Eventually, when the tread elements have reached their distortion limit, parts of the tread elements will begin to slip so that a further rise in tractive force will produce a much larger increase in tyre slip until the peak or limiting tractive effort is developed. This normally corresponds to on a hard road surface to roughly 15–20% slip (Fig. 8.14). Beyond the peak tractive effort a further increase in slip produces an unstable condition with a considerable reduction in tractive effort until pure wheel spin results (the tyre ust slides over the road surface). A tyre sub ected to a braking torque produces a very similar braking effort response with respect to wheel slip, which is now referred to as   It will be seen that the maximum braking effort developed is largely dependent upon the nature of the road surface (Fig. 8.15) and the normal wheel loads (Fig. 8.16), whereas wheel speed has more influences on the unstable skid region of a braking sequence (Fig. 8.17).

.1.7 Tyre reaction due to concurrent longitudinal and lateral forces (Fig. 8.18)
A loaded wheel and tyre rolling can generate only a limited amount of tread to ground reaction to
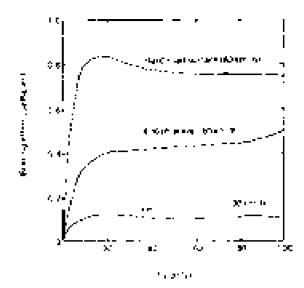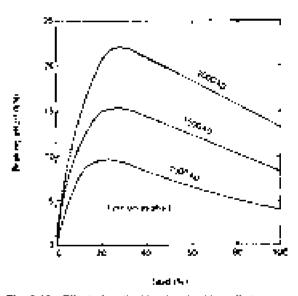

Fig. 8.16 Effect of vertical load on braking effort

resist the tyre slipping over the surface when the tyre is sub ected to longitudinal (tractive or braking) forces and lateral (side) (cornering or crosswind) forces simultaneously. Therefore the resultant components of the longitudinal and lateral forces must not exceed the tread to ground resultant reaction force generated by all of the tread elements within the contact area biting into the ground.

The relative relationship of the longitudinal and lateral forces acting on the tyre can be shown by
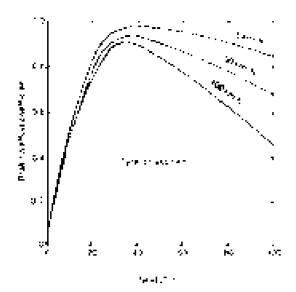
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Fig. 8.17 Effect of vehicle speed on braking effort

resolving both forces perpendicularly to each other within the boundary of limiting reaction force circle (Fig. 8.18(a and b)). This circle with its vector forces shows that when longitudinal forces due to traction or braking forces is large (Fig. 8.18(c and d)), the tyre can only sustain a much smaller side

force. If the side force caused either by cornering or a crosswind is large, the traction or braking effort must be much reduced.

.2 Tyre materials

. .1 The structure and properties of rubber
(Figs 8.19, 8.20 and 8.21)

The outside carcass and tread of a tyre is made from a rubber compound that is a mix of several substances to produce a combination of properties necessary for the tyre to function effectively. Most metallic materials are derived from simple molecules held together by electrostatic bonds which sustain only a limited amount of stretch when subjected to tension (Fig. 8.19). Because of this, the material s elasticity may be restricted to something like 2% of its original length. Rubber itself may be either natural or synthetic in origin. In both cases the material consists of many thousands of long chain molecules all entangled together. When stretched, the giant rubber molecules begin to untangle themselves from their normal coiled state and in the process of straightening out, provide a considerable amount of extension which may be of the order of 300% of the material s original length. Thus it is not the electrostatic bonds being stretched
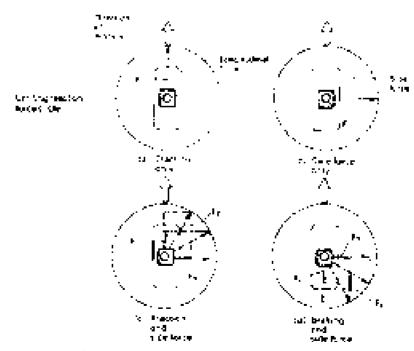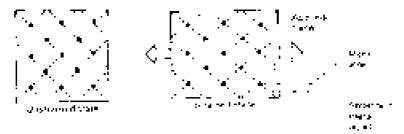

Fig. 8.18 (a–d) imiting reaction force circle

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Fig. 8.19 etal atomic lattice network
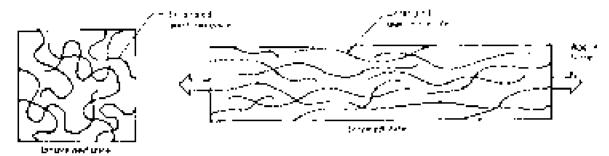

Fig. 8.20 Raw rubber network of long chain molecules

Fig. 8.21 ulcani ed rubber cross-linked network of long chain molecules

but the uncoiling and aligning of the molecules in the direction of the forces pulling the material apart (Fig. 8.20). Consequently, when the tensile force is removed the molecules revert to their free state and thereby draw themselves into an entangled network again. ence it is not the bonds being stretched but the uncoiling and aligning of the molecules in the direction of the force pulling the material apart.

Vulcani ation To reduce the elasticity and to increase the strength of the rubber, that is to restrict the molecules sliding past each other when the substance is stretched, the rubber is mixed with a small amount of sulphur and then heated, usually under pressure. The chemical reaction produced is known either as curing or more commonly as vulcanization (named after Vulcan, the Roman god of fire). As a result, the sulphur molecules form a network of cross-links between some of the giant rubber molecules (Fig. 8.21). The outcome of the cross-linking between the entangled long chain molecules is that it makes it more difficult for these molecules to slip over each other so that the rubber becomes stronger with a considerable reduction in flexibility.

nitiators and accelerator To start off and speed up the vulcanization process, activators such as a metallic zinc oxide are used to initiate the reaction and an organic accelerator reduces the reaction

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time and temperature needed for the sulphur to produce a cross-link network.

arbon black Vulcanized rubber does not have sufficient abrasive resistance and therefore its rate of wear as a tyre tread material would be very high. To improve the rubber s resistance against wear and tear about a quarter of a rubber compound content is made up of a very fine carbon powder known as carbon black. When it is heated to a molten state the carbon combines chemically with the rubber to produce a much harder and tougher wear resistant material.

il e tension To assist in producing an even dispersion of the rubber compound ingredients and to make processing of the tyre shape easier, an emulsion of hydrocarbon oil is added (up to 8%) to the rubber latex to dilute or extend the rubber. This makes the rubber more plastic as opposed to elastic with the result that it becomes tougher, offers greater wear resistance and increases the rubber s hysteresis characteristics thereby improving its wet grip properties.

nti-o idants and -o onates ther ingredients such as an anti-oxidant and anti-ozonate are added to preserve the desirable properties of the rubber compound over its service life. The addition of anti-oxidants and -ozonates (1 or 2 parts per 100 parts of rubber) prevents heat, light and particularly oxygen ageing the rubber and making it hard and brittle.

. . *echanical properties*
To help the reader understand some of the terms used to define the mechanical properties of rubber the following brief definitions are given:

Material resilience This is the ability for a solid substance to rebound or spring back to its original dimensions after being distorted by a force. A material which has a high resilience generally has poor road grip as it tends to spring away from the ground contact area as the wheel rolls forward.

Material plasticity This is the ability for a solid material to deform without returning to its original shape when the applied force is removed. A material which has a large amount of plasticity promotes good road grip as each layer of material tends to cling to the road surface as the wheel rolls.

Material hysteresis This is the sluggish response of a distorted material taking up its original form so that some of the energy put into deforming the carcass, side walls and tread of a tyre at the contact patch region will still not be released when the tyre has completed one revolution and the next distortion period commences. As the cycle of events continues, more and more energy will be absorbed by the tyre, causing its temperature to rise. If this heat is not dissipated by the surrounding air, the inner tyre fabric will eventually become fatigued and therefore break away from the rubber encasing it, thus destroying the tyre. For effective tyre grip a high hysteresis material is necessary so that the distorted rubber in contact with the ground does not immediately spring away from the surface but is inclined to mould and cling to the contour of the road surface.

Material fatigue This is the ability of the tyre structure to resist the effects of repeated flexing without fracture, particularly with operating temperatures which may reach something of the order of 100 C for a heavy duty tyre although temperatures of 80–85 C are more common.

. .3 *atural and synthetic rubbers*
Synthetic materials which have been developed as substitutes for natural rubber and have been utilized for tyre construction are listed with natural rubber as follows:

a) Natural rubber (NR)
b) Chloroprene (Neoprene) rubber (CR)
c) Styrene–butadiene rubber (SBR)
d) Polyisoprene rubber (IR)
e) Ethylene propylene rubber (EPR)
f) Polybutadiene rubber (BR)
g) Isobutene–isoprene (Butyl) rubber (IIR)

atural rubber (R) Natural rubber has good wear resistance and excellent tear resistance. It offers good road holding on dry roads but retains only a moderately good grip on wet surfaces. ne further merit is its low heat build-up, but this is contrasted by high gas permeability and its resistance to ageing and ozone deterioration is only fair. The side walls and treads have been made from natural rubber but nowadays it is usually blended with other synthetic rubbers to exploit their desirable properties and to minimize their shortcomings.

hloroprene (eoprene) rubber (R) This synthetic rubber is made from acetylene and hydro-

chloric acid. Wear and tear resistance for this rubber compound, which was one of the earliest to compete with natural rubber, is good with a reasonable road surface grip. A ma or limitation is its inability to bond with the carcass fabric so a natural rubber film has to be interposed between the cords and the Neoprene covering. Neoprene rubber has a moderately low gas permeability and does not show signs of weathering or ageing throughout a tyre s working life. When blended with natural rubber it is particularly suitable for side wall covering.

Styrene butadiene rubber (S R) Compounds of this material are made from styrene (a liquid) and butadiene (a gas). It is probably the most widely used synthetic rubber within the tyre industry. Styrene–butadiene rubber (SBR) forms a very strong bond to fabrics and it has a very good resistance to wear, but suffers from poor tear resistance compared to natural rubber. ne outstanding feature of this rubber is its high degree of energy absorption or high hysteresis and low resilience. It is these properties which give it exceptional grip, especially on wet surfaces. ue to the high heat build up, SBR is restricted to the tyre tread while the side walls are normally made from low hysteresis compounds which provide greater rebound response and run cooler. Blending SBR with NR enables the best properties of both synthetic and natural rubber to be utilized so that only one rubber compound is necessary for some types of car tyres. The high hysteresis obtained with SBR is partially achieved by using an extra high styrene content and by adding a large proportion of oil to extend the compound, the effects being to increase the rubber plastic properties and to lower its resilience (i.e. reduce its rebound response).

Polyisoprene rubber (R) This compound has very similar characteristics to natural rubber but has improved wear and particularly tear resistance with a further advantage of an extremely low heat build up with normal tyre flexing. These properties make this material attractive when blended with natural rubber and styrene–butadiene rubber to produce tyre treads with very high abrasion resistance. For heavy duty application such as track tyres where high temperatures and driving on rough terrains are a problem, this material has proved to be successful.

Ethylene propylene rubber (EPR) The ma or advantage of this rubber compound is its ability to be mixed with large amounts of cheap carbon black and oil without destroying its rubbery properties. It has excellent abrasive ageing and ozone resistance with varying road holding qualities in wet weather depending upon the compound composition. Skid resistance on ice has also been varied from good to poor. A great disadvantage, however, is that the rubber compound bonds poorly to cord fabric. enerally, the higher the ethylene content the higher the abrasive resistance, but at the expense of a reduction in skid resistance on ice. Rubber compounds containing EPR have not proved to be successful up to the present time.

Polybutadiene rubber (R) This rubbery material has outstanding wear resistance properties and is exceptionally stable with temperature changes. It has a high resilience that is a low hysteresis level.

When blended with SBR in the correct proportions, it reduces the wet road holding slightly and considerably improves its ability to resist wear. Because of its high resilience (large rebound response), if mixed in large proportions, the road holding in wet weather can be relatively poor. It is expensive to produce. When it is used for tyres it is normally mixed with SBR in the proportion of 15 to 50%.

sobutene isoprene (utyl) rubber (R) Rubber of this kind has exceptionally low permeability to gas. In fact it retains air ten times longer than tubes made from natural rubber, with the result that it has been used extensively for tyre inner tubes and for linings of tubeless tyres. Unfortunately it will not blend with SBR and NR unless it is chlorinated, but in this way it can be utilized as an inner tube lining material for tubeless tyres. The resistance to wear is good and it has a high hysteresis so that it responds more like plastic than rubber to distortion at ground level. Road grip is good for both dry and wet conditions. When mixed with carbon black its desirable properties are generally improved. ue to its high hysteresis tyre treads made from this material do not generate noise in the form of squeal since it does not readily give out energy to the surroundings.

. . Summary of the merits and limitations of natural and synthetic rubber compounds

Some cross-ply tyres are made from one compound from bead to bead, but the severity of the carcass flexure with radial ply tyres encourages the manufacturers of tyres to use different rubber

composition for various parts of the tyre structure so that their properties match the duty requirements of each functional part of the tyre (i.e. tread, side wall, inner lining, bead etc.).

Side walls are usually made from natural rubber blended with polybutadiene rubber (BR) or styrene–butadiene rubber (SBR) or to a lesser extent Neoprene or Butyl rubber or even natural rubber alone. The properties needed for side wall material are a resistance against ozone and oxygen attack, a high fatigue resistance to prevent flex cracking and good compatibility with fabrics and other rubber compounds when moulded together.

Tread wear fatigue life and road grip depends to a great extent upon the surrounding temperatures, weather conditions, be they dry, wet, snow or ice bound, and the type of rubber compound being used. A comparison will now be made with natural rubber and possibly the most important synthetic rubber, styrene–butadiene (SBR). At low temperatures styrene–butadiene (SBR) tends to wear more than natural rubber but at higher temperatures the situation reverses and styrene–butadiene rubber (SBR) shows less wear than natural rubber. As the severity of the operating condition of the tyre increases SBR tends to wear less relative to NR. The fatigue life of all rubber compounds is reduced as the degree of cyclic distortion increases. For small tyre deflection SBR has a better fatigue life but when deflections are large NR provides a longer service life. Experience on ice and snow shows that NR offers better skid resistance, but as temperatures rise above freezing, SBR provides an improved resistance to skidding. This cannot be clearly defined since it depends to some extent on the amount of oil extension (plasticizer) provided in the blending in both NR and SBR compounds.

il extension when included in SBR and NR provides similarly improved skid resistance and in both cases becomes inferior to compounds which do not have oil extension.

Two examples of typical rubber compositions suitable for tyre treads are:

a)   igh styrene butadiene rubber   31%
      il extended butadiene rubber   31%
      Carbon black   30%
        il   6%
      Sulphur   2%
b) Styrene butadiene rubber   45%
      Natural rubber   15%
      Carbon black   30%
        il   8%
      Sulphur   2%

.3 Tyre tread design

.3.1 Tyre construction
The construction of the tyre consists basically of a carcass, inner beads, side walls, crown belt (radials) and tread.

The carcass is made from layers of textile core plies. Cross-ply tyres tend to still use nylon whereas radial-ply tyres use either raylon or polyester.

The inside diameter of both tyre walls support the carcass and seat on the wheel rim. The edges of the tyre contacting the wheel are known as beads and moulded inside each bead is a strengthening endless steel wire cord.

The outside of the tyre carcass, known as the side walls, is covered with rubber compound. Side walls need to be very flexible and capable of protecting the carcass from external damage such as cuts which can occur when the tyre is made to climb up a kerb.

Between the carcass and tyre tread is a crown reinforcement belt made from either synthetic fabric cord such as raylon or for greater strength steel cores. This circumferential endless cord belt provides the rigidity to the tread rubber.

The outside circumferential crown portion of the tyre is known as the tread. It is made from a hard wearing rubber compound whose function is to grip the contour of the road.

.3.   Tyre tread considerations
The purpose of a pneumatic tyre is to support the wheel load by a cushion of air trapped between the well of the wheel rim and the toroid-shaped casing known as the carcass. Wrapped around the outside of the tyre carcass is a thick layer of rubber compound known as the tread whose purpose is to protect the carcass from road damage due to tyre impact with the irregular contour of the ground and the abrasive wear which occurs as the tyre rolls along the road. While the wheel is rotating the tread provides driving, braking, cornering and steering grip between the tyre and ground. Tyre grip must be available under a variety of road conditions such as smooth or rough hard roads, dry or wet surfaces, muddly tracks, fresh snow or hard packed snow and ice and sandy or soft soil terrain. Tread grip may be defined as the ability of a rolling tyre to continuously

develop an interaction between the individual tread elements and the ground so that any longitudinal (driving) or lateral (side) forces imposed on the wheel will not be able to make the tread in contact with the ground slide.

A tyre tread pattern has two main functions:

1 to provide a path for drainage of water which might become trapped between the tyre contact patch and the road,
2 to provide tread to ground bite when the wheel is sub ected to both longitudinal and lateral forces under driving conditions.

.3.3 Tread bite

Bite is obtained by selecting a pattern which divides the tread into many separate elements and providing each element with a reasonably sharp well defined edge. Thus as the wheel rotates these tread edges engage with the ground to provide a degree of mechanical tyre to ground interlock in addition to the frictional forces generated when transmitting tractive or braking forces.

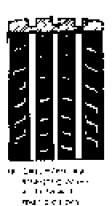The ma or features controlling the effectiveness of the tread pattern in wet weather are:

1 drainage grooves or channels,
2 load carrying ribs,
3 load bearing blocks,
4 multiple microslits or sipes.

Tread drainage grooves (Fig. 8.22(a, b, c and d)) The removal of water films from the tyre to ground interface is greatly facilitated by having a number of circumferential grooves spaced out across the tread width (Fig. 8.22(a)). These grooves enable the leading elements of the tread to push water through the enclosed channels made by the road sealing the underside of the grooves. Water therefore emerges from the trailing side of the contact patch in the form of ets. If these grooves are to be effective, their total cross-sectional area should be adequate to channel all the water immediately ahead of the leading edge of the contact patch away. If it cannot cope the water will become trapped between the tread ribs or blocks so that these elements lift and become separated from the ground, thus reducing the effective area of the contact patch and the tyre s ability to grip the ground.

To speed up the water removal process under the contact patch, lateral grooves may be used to oin together the individual circumferential grooves and to provide a direct side exit for the outer circumferential grooves. Normally many grooves are preferred to a few large ones as this provides a better drainage distribution across the tread.

Tread ribs (Fig. 8.22(a and b)) Circumferential ribs not only provide a supportive wearing surface for the tyre but also become the walls for the drainage grooves (Fig. 8.22(a and b)). ateral (transverse) ribs or bars provide the optimum bite for tractive and braking forces but circumferential ribs are most effective in controlling cornering and steering stability. To satisfy both longitudinal and lateral directional requirements which may be acting concurrently on the tyre, ribs may be arranged diagonally or in the form of zig-zag circumferential ribs to improve the wiping effect across the tread surface under wet conditions. It is generally better to break the tread pattern into many narrow ribs than a few wide ones, as this prevents the formation of hydrodynamic water wedges which may otherwise tend to develop

Fig. 8.22 (a–d) Basic tyre tread patterns

with the consequent separation of the tread elements from the road.

Tread blocks (Figs 8.22(c and d) and 8.23(a and b)) If longitudinal circumferential grooves in the middle of the tread are complemented by lateral (transverse) grooves channelled to the tread shoulders, then with some tread designs the drainage of water can be more effective at speed. The consequences of both longitudinal and lateral drainage channels is that the grooves encircle portions of the tread so that they become isolated island blocks (Fig. 8.22 (c and d)). These blocks can be put to good use as they provide a sharp wiping and biting edge where the interface of the tread and ground meet. To improve their biting effectiveness for tractive and braking forces as well as steering and cornering forces, these forces may be resolved into diagonal resultants so that the blocks are sometimes arranged in an oblique formation. A limitation to the block pattern concept is caused by inadequate support around the blocks so that under severe operating conditions, the bulky rubber blocks tend to bend and distort. This can be partially overcome by incorporating miniature buttresses between the drainage grooves which lean between blocks so that ad acent blocks support each other. At the same time, drainage channels which burrow below the high mounted buttresses are prevented from closing. Tread blocks in the form of bars, if arranged in a herringbone fashion, have proved to be effective on rugged ground. Square or rhombus-shaped blocks provide a tank track unrolling action greatly reducing movement in the tread contact area. This pattern helps to avoid the break-up on the top layer of sand or soil and thus prevents the tyre from digging into the ground. Because of the inherent tendency of the individual blocks to bend somewhat when they are sub ected to ground reaction forces, they suffer from toe to heel rolling action which causes blunting of the leading edge and trailing edge feathering. enerally tyres which develop this type of wear provide a very good water sweeping action when new, which permits the tread elements to bite effectively into the ground, but after the tyre has been on the road for a while, the blunted leading edge allows water to enter underneath the tread elements. Consequently the slightest amount of water interaction between the block elements and ground reduces the ability for the tread to bite and in the extreme cases under locked wheel braking conditions a hydrodynamic water

wedge action may result, causing a mild form of aquaplaning to take place.

For tread block elements to maintain their wiping action on wet surfaces, wear should be from toe to heel (Fig. 8.23(a)). If, however, wear occurs in the reverse order, that is from heel to toe (Fig. 8.23(b)), the effectiveness of the tread pattern will be severely reduced since the tread blocks then become the platform for a hydrodynamic water wedge which at speed tries to lift the tread blocks off the ground.

Tread slits or sipes (Figs 8.22(a, b, c and d) and 8.24(a, b and c)) Microslits, or   as they are commonly called, are incisions made at the surface of the tyre tread, going down to the full depth of the tread grooves. They resemble a knife cut, except that instead of being straight they are mostly of a zig-zag nature (Fig. 8.22(a, b, c and d)). Normally these sipes terminate within the tread elements, but sometimes one end is permitted to intersect the side wall of a drainage groove. In some tread patterns the sipes are all set at a similar angle to each other, the zig-zag shape providing a large number of edges which point in various directions.   ther designs have sets of sipes formed at different angles to each other so that these sipes are effective whichever way the wheel points and whatever the direction the ground reaction forces operate.

Sipes or slits in their free state are almost closed, but as they move into the contact patch zone the ribs or blocks distort and open up (Fig. 8.24(a)). Because of this, the sipe lips scoop up small quantities of water which still exist underneath the tread. This wiping action enables some biting edge reaction with the ground.   enerally, the smaller the sipes are and more numerous they are the greater will be their effective contribution to road grip. The
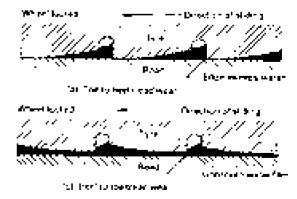

Fig. 8.23 (a and b) Effect of irregular tread block wear
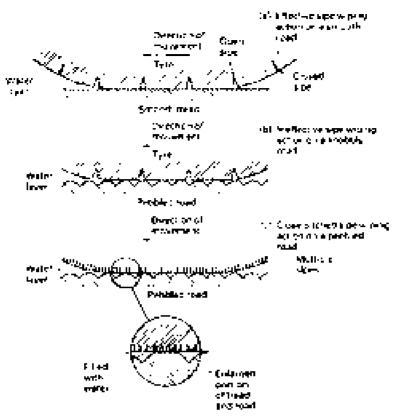
Fig. 8.24 (a–c) Effectiveness of microslits on wet road surfaces

normal spacing of sipes (microslits) on a tyre tread makes them ineffective on a pebbled road surface because there will be several pebbles between the pitch of the sipes (Fig. 8.24(b)), and water will lie between these rounded stones, therefore only a few of the stones will be sub ected to the wiping edge action of the opened lips. An alternative method to improve the wiping process would be to have many more wiping slits (Fig. 8.24(c)), but this is very difficult to implement with the present manufacturing techniques. The advantages to be gained by multislits are greatest under conditions of low friction associated with thin water films on smooth and polished road surfaces. This is because the road surface asperities are not large and sharp enough to penetrate the thin water film trapped under plain ribs and blocks.

Selection of tread patterns (Fig. 8.25(a–1))

(Fig. 8.25(a, b and c)) eneral duty car tyres which are capable of operating effect-ively at all speeds tend to have tread blocks situated in an oblique fashion with a network of surrounding drainage grooves which provide both circumferential and lateral water release.

W (Fig. 8.25(d, e and f)) Winter car tyres are normally very similar to the general duty car tyre but the tread grooves are usually wider to permit easier water dispersion and to provide better exposure of the tread blocks to snow and soft ice without sacrificing too much tread as this would severely reduce the tyre s life.

(Fig. 8.25(g and h)) Truck tyres designed for steered axles usually have circumferential zig-zag ribs and grooves since they provide very good lateral reaction when being steered on curved tracks. rive axle tyres, on the other hand, are designed with tread blocks with adequate grooving so that optimum traction grip is obtained under both dry and wet conditions. Some of these tyres also have provision for metal studs to be inserted for severe winter hard packed snow and ice conditions.
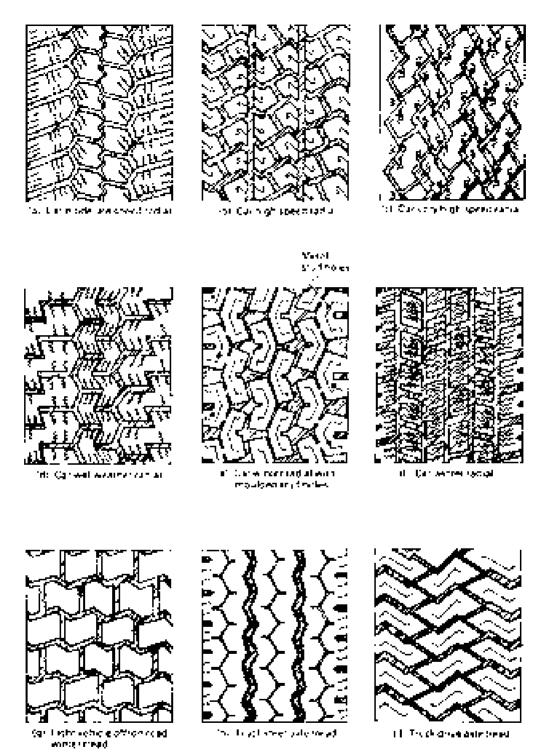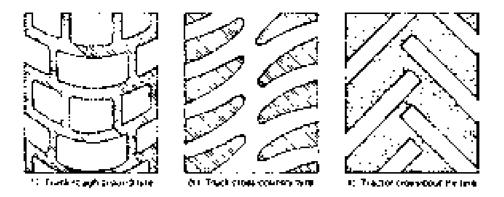
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(a) Laminated tread radial   (b) Car high speed radial   (c) Car on high speed radial

(d) Car wet weather radial   (e) Car normal radial with moulded wing tread   (f) Car winter radial

(g) Light vehicle off-road and winter tread   (h) Truck heavy duty tread   (i) Truck drive axle tread

Fig. 8.25 (a–)   Survey of tyre tread patterns

Fig. 8.25 *contd*

(Fig. 8.25(i)) ff/on road vehicle tyres usually have a much simpler bold block tread with a relatively large surrounding groove. This enables each individual block to react independently with the ground and in this manner bite and exert traction on soil which may be hard on the surface but soft underneath without break-up of the top layer, thus preventing the tyre digging in. The tread pattern blocks are also designed to be small enough to operate on hard surfaced roads at moderate speeds without excessive ride harshness.

(Fig. 8.25(, k and l)) Truck or tractor tyres designed for building sites or quarries generally have slightly curved rectangular blocks separated with wide grooves to provide a strong flexible casing and at the same time present a deliberately penetrating grip. Cross-country tyres which tend to operate on soft soil tend to prefer diagonal bars either merging into a common central rib or arranged with separate overlapping diagonal bars, as this configuration tends to provide exceptionally good traction on muddy soil, snow and soft ice.

.3. *The three one concept of tyre to ground contact on a et surface* (Fig. 8.26)

The interaction of a tyre with the ground when rolling on a wet surface may be considered in three phases (Fig. 8.26):

( ) The leading zone of the tread contacts the stagnant water film covering the road surface and displaces the ma ority of the water into the grooves between the ribs and blocks of the tread pattern.

( ) The middle zone of the tread traps and reduces the thickness of the remaining water
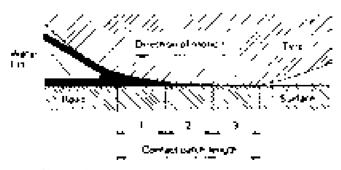

Fig. 8.26 Tyre to ground ones of interaction

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between the faces of the ribs or blocks and ground so that some of the road surface asperities now penetrate through the film of water and may actually touch the tread. It is this region which is responsible for the final removal of water and is greatly assisted by multiple sipes and grooved drainage channels. If the ribs and blocks are insufficiently relieved with sipes and grooves it is possible that under certain conditions aquaplaning may occur in this region.

The effectiveness of this phase is determined to some extent by the texture of the road surface, as this considerably influences the dryness and potency of the third road grip phase.

( ) The water film has more or less been completely squeezed out at the beginning of this region so that the faces of the ribs and blocks bearing down on the ground are able to generate the bite which produces the tractive, braking and cornering reaction forces.

.3.5 A uaplaning hydroplaning (Fig. 8.27)

The performance of a tyre rolling on wet or semi-flooded surface will depend to some degree upon the tyre profile tread pattern and wear. If a smooth tread is braked over a very wet surface, the forward rotation of the tyre will drag in the water immediately in front between the tread face and ground and squeeze it so that a hydrodynamic pressure is created. This hydrodynamic pressure acts between the tyre and ground, its magnitude being proportional to the square of the wheel speed. With the wheel in motion, the water will form a converging wedge between the tread face and ground and so exert an upthrust on the underside of the tread. As a result of the pressure generated, the tyre tread will tend to separate itself from the ground. This condition is known as aquaplaning or hydroplaning. If the wheel speed is low only the front region of the tread rides on the wedge of water, but if the speed is rising the water wedge will progressively extend backward well into the contact patch area (Fig. 8.27). Eventually the upthrust created by the product of the hydrodynamic pressure and contact area equals the vertical wheel load. At this point the tyre is completely supported by a cushion of water and therefore provides no traction or directional control. If the tread has circumferential (longitudinal) and transverse (lateral) grooves of adequate depth then the water will drain through these passages at ground level so that aquaplaning is minimized even at high speeds. As the tyre tread wears the critical speed at which aquaplaning occurs becomes much lower. n very wet roads a bald tyre is certain to be sub ected to aquaplaning at speeds above 60 km/h and therefore the vehicle when driven has no directional stability. ow aspect ratio tyres may find it difficult to channel the water away from the centre of the tread at a sufficiently high
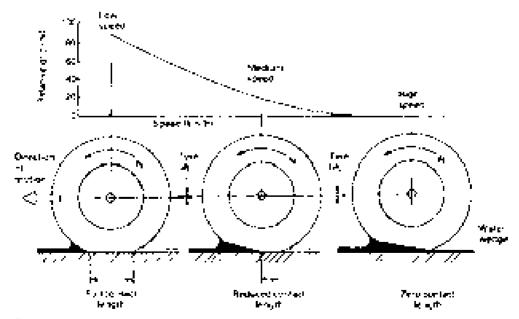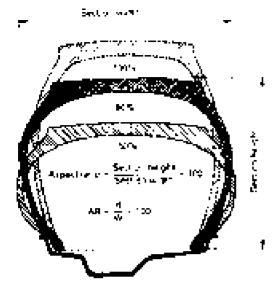

Fig. 8.27 Tyre a uaplaning

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Fig. 8.28 Tyre profiles with different aspect ratios

rate and therefore must rely more on the circumferential grooves than on transverse grooving.

.3.6 Tyre profile and aspect ratio (Fig. 8.28)

The profile of a tyre carcass considerably influences its rolling and handling behaviour. Because of the importance of the tyre s cross-sectional configuration in predicting its suitability and performance for various applications, the aspect ratio was introduced. This constant for a particular tyre may be defined as the ratio of the tyre cross-sectional height (the distance between the tip of the tread to the bead seat) to that of the section width (the outermost distance between the tyre walls) (Fig. 8.28).

i.e. $\quad \text{Aspect ratio} = \dfrac{\text{Section height}}{\text{Section width}} \times 100$

A tyre with a large aspect ratio is referred to as a   and a tyre with a small aspect is known as a   Until about 1934 aspect ratios of 100% were used, but with the better understanding of pneumatic tyre properties and improvement in tyre construction lower aspect ratio tyres became available. The availability of lower aspect ratio tyres over the years was as follows 1950s–95%, 1962–88% (this was the standard for many years), 1965–80% and about 1968–70%. Since then for special applications even lower aspect ratios of 65%, 60%, 55% and even 50% have become available.

owering the aspect ratio has the following effects:

1 The tyre side wall height is reduced which increases the vertical and lateral stiffness of the tyre.
2 A shorter and wider contact patch is established. The overall effect is to raise the load carrying capacity of the tyre.
3 The wider contact patch enables larger cornering forces to be generated so that vehicles are able to travel faster on bends.
4 The shorter and wider contact patch decreases the pneumatic trail which correspondingly reduces and makes more consistent the self-aligning torque.
5 The shorter and broader contact patch will, under certain driving conditions, reduce the slip angles generated by the tyre when sub ected to side forces. Accordingly this reduces the tread distortion and as a result scuffing and wear will decrease.
6 With an increase in vertical stiffness and a reduction in tyre deflection with lower aspect ratio tyres, less energy will be dissipated by the tyre casing so that rolling resistance will be reduced. This also results in the tyre being able to run continuously at high speeds at lower temperatures which tends to prolong the tyre s life.
7 The increased lateral stiffness of a low profile tyre will increase the sensitivity to camber variations and quicken the response to steering changes.
8 Wider tyre contact patches make it more difficult for water drainage at speed particularly in the mid tread region.   ence the tread pattern design with low profile tyres becomes more critical on wet roads, if their holding is to match that of higher aspect ratio tyres.
9 The increased vertical stiffness of the tyre reduces static deflection of the tyre under load, so that more road vibrations are transmitted through the tyre. This makes it a harsher ride so that ride comfort is reduced unless the suspension design has been able to provide more isolation for the body.

.4   ornering properties of tyres

. .1 Static load and standard heel height
(Figs 8.29 and 8.30)

A vertical load acting on a wheel will radially distort the tyre casing from a circular profile to a short flat one in the region of the tread to ground interface (Fig. 8.29). The area of the tyre contact with the ground is known as the
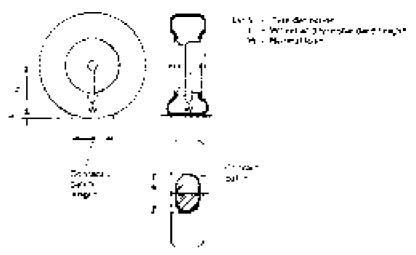its plan view shape is roughly elliptical. The consequence of this tyre deflection is to reduce the

Fig. 8.29 llustration of static tyre deflection

standard height of the wheel, that is the distance between the wheel axis and the ground. enerally tyre deflection will be proportional to the radial load imposed on the wheel increasing the tyre inflation pressure reduces the tyre deflection for a given vertical load (Fig. 8.30). Note that there is an initial deflection (Fig. 8.30) due to the weight of the wheel and tyre alone. The steepness of the load deflection curve is useful in estimating the static stiffness of the tyres which can be interpreted as a measure of its vibration and ride qualities.

. . Tyre contact patch (Figs 8.29 and 8.31)
The downward radial load imposed on a road wheel causes the circular profile of the tyre in contact with the ground to flatten and spread towards the front and rear of its natural rolling plane. When the wheel is stationary, the interface area between the tyre and ground known as the contact patch will take up an elliptical shape (Fig. 8.29), but if the wheel is now sub ected to a side thrust the grip between the tread and ground will distort the patch into a semibanana configuration (Fig. 8.31 (a)). It is the ability of the tyre contact patch casing, and elements of the tread to comply and change shape due to the imposed reaction forces, which gives tyres their steering properties. enerally, radial ply tyres form longer and broader contact patches than their counterpart cross-ply tyres, hence their superior road holding.

. .3 Cornering force (Fig. 8.31(a and b))
Tyres are sub ected not only to vertical forces but also to side (lateral) forces when the wheels are in
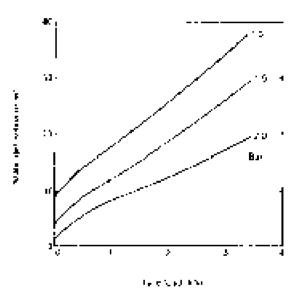

Fig. 8.30 Effect of tyre vertical load on static deflection

motion due to road camber, side winds, weight transfer and centrifugal force caused by travelling round bends and steering the vehicle on turns. When a side force, sometimes referred to as lateral force, is imposed on a road wheel and tyre, a reaction between the tyre tread contact patch and road surface will oppose any sideway motion. This resisting force generated at the tyre to road interface is known as the cornering force (Fig. 8.31(a and b)), its magnitude being equal to the lateral force but it acts in the opposite sense. The amount of
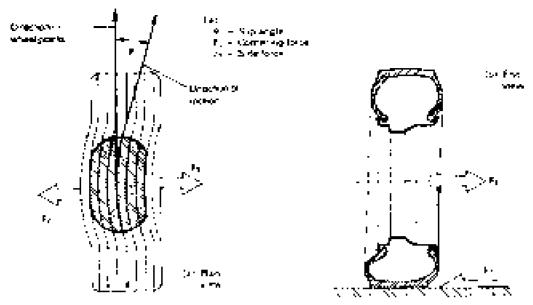
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Fig. 8.31 Tyre tread contact patch distortion when subjected to a side force

cornering force developed increases roughly in proportion with the rise in lateral force until the grip between the tyre tread and ground diminishes. Beyond this point the cornering force cannot match further increases in lateral force with the result that tyre breakaway is likely to occur. Note that the greater the cornering force generated between tyre and ground, the greater the tyre s grip on the road.

The influencing factors which determine the amount of cornering force developed between the tyre and road are as follows:

Initially the cornering force increases linearly with increased slip angle, but beyond about four degrees slip angle the rise in cornering force is non-linear and increases at a much reduced rate (Fig. 8.32), depending to a greater extent on tyre design.

As the vertical or radial load on the tyre is increased for a given slip angle, the cornering force rises very modestly for small slip angles but at a far greater rate with larger slip angles (Fig. 8.33).

Raising the tyre inflation pressure linearly increases the cornering force for a given slip angle (Fig. 8.34). These graphs also show that increasing the tyre slip angle considerably raises the cornering forces generated.

Slip angle (Fig. 8.31(a))
Any lateral force applied to a road wheel will tend to push the supple tyre walls sideways, but the opposing tyre to ground reaction causes the tyre contact patch to take up a curved distorted shape. As a result, the rigid wheel will point and roll in the direction it is steered, whereas the tyre region in contact with the ground will follow the track continuously laid down by the deformed tread path of the contact patch (Fig. 8.31(a)). The angle made between the direction of the wheel plane and that which it travels is known as the slip angle. Provided the slip angle is small, the compliance of the tyre construction will allow each element of tread to remain in contact with the ground without slippage.

.5 Cornering stiffness cornering po er
(Fig. 8.32)
When a vehicle travels on a curved path, the centrifugal force (lateral force) tends to push sideways each wheel against the opposing tyre contact patch to ground reaction. As a result, the tyre casing and tread in the region of the contact patch very slightly deform into a semicircle so that the path followed by the tyre at ground level will not be quite the same as the direction of the wheel points. The resistance offered by the tyre crown or belted tread region by the casing preventing it from deforming and generating a slip angle is a measure of the tyre s cornering power. The cornering power, nowadays more

usually termed _____ may be defined as the cornering force required to be developed for every degree of slip angle generated.

i.e.

$$\text{Cornering stiffness} = \frac{\text{Cornering force}}{\text{Slip angle}}(\text{kN deg})$$

In other words, the cornering stiffness of a tyre is the steepness of the cornering force to slip angle curve normally along its linear region (Fig. 8.32). The larger the cornering force needed to be generated for one degree of slip angle the greater the cornering stiffness of the tyre will be and the smaller the steering angle correction will be to sustain the intended path the vehicle is to follow. Note that the supple flexing of a radial ply side wall should not be confused with the actual stiffness of the tread portion of the tyre casing.

. .6 Centre of pressure (Fig. 8.35)

When a wheel standing stationary is loaded, the contact patch will be distributed about the geometric centre of the tyre at ground level, but as the wheel rolls forward the casing supporting the tread is deformed and pushed slightly to the rear (Fig. 8.35). Thus in effect the ma ority of the cornering force generated between the ground and each element of tread moves from the static centre of pressure to some dynamic centre of pressure behind the vertical centre of the tyre, the amount of displacement corresponding to the wheel construction, load, speed and traction. The larger area of
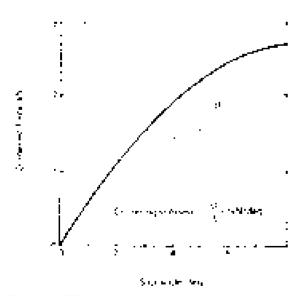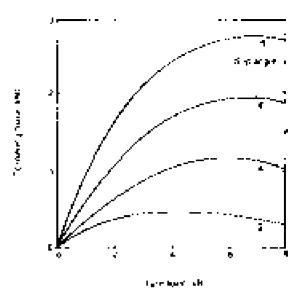

Fig. 8.33 Effect of tyre vertical load on cornering force

tread to ground reaction will be concentrated behind the static centre of the wheel and the actual distribution of cornering force from front to rear of the contact patch is shown by the shaded area between the centre line of the tyre and the cornering force plotted line. The total cornering force is therefore roughly proportional to this shaded area and its resultant dynamic position is known as the centre of pressure (Fig. 8.35).

. .7 neumatic trail (Fig. 8.35)

The cornering force generated at any one time will be approximately proportional to the shaded area between the tyre centre line and the cornering force plotted line so that the resultant cornering force (centre of pressure) will act behind the static centre of contact. The distance between the static and dynamic centres of pressure is known as the _____ (Fig. 8.35), its magnitude being dependent upon the degree of creep between tyre and ground, the vertical wheel load, inflation pressure, speed and tyre constriction. enerally with the longer contact patch, radial ply tyres have a greater pneumatic trail than those of the cross-ply construction.

. . Self aligning tor ue (Fig. 8.35)

When a moving vehicle has its steering wheels turned to negotiate a bend in the road, the lateral (side) force generates an equal and opposite
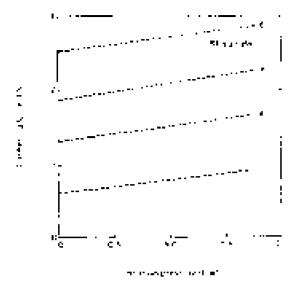

Fig. 8.32 Effect of slip angle on cornering force

Fig. 8.34 Effect of tyre inflation pressure on cornering force

reaction force at ground level known as the

As the cornering force centre of pressure is to the rear of the geometric centre of the wheel and the side force acts perpendicularly through the centre of the wheel hub, the offset between the these two forces, known as the pneumatic trail, causes a moment (couple) about the geometric wheel centre which endeavours to turn both steering wheels towards the straight ahead position. This self-generating torque attempts to restore the plane of the wheels with the direction of motion and it is known as the (Fig. 8.35). It is this inherent tyre property which helps steered tyres to return to the original position after negotiating a turn in the road. The self-aligning torque (SAT) may be defined as the product of the cornering force and the pneumatic trail.

i.e. $T_{SAT} = F_c \times t_p$ (Nm)

igher tyre loads increase deflection and accordingly enlarge the contact patch so that the pneumatic trail is extended. Correspondingly this causes a rise in self-aligning torque. n the other hand increasing the inflation pressure for a given tyre load will shorten the pneumatic trail and reduce the self-aligning torque. ther factors which influence self-aligning torque are load transfer during braking, accelerating and cornering which alter the contact patch area. As a general rule, anything which increases or decreases the contact patch length raises or reduces the self-aligning torque respectively. The self-aligning torque is little affected with small slip angles when braking or accelerating, but with larger slip angles braking decreases the aligning torque and acceleration increases it (Fig. 8.36).
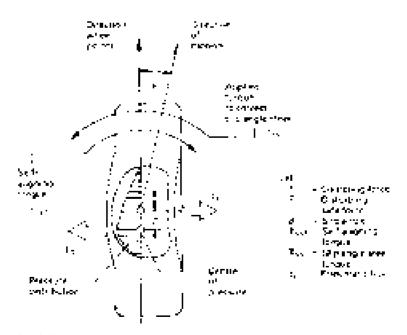

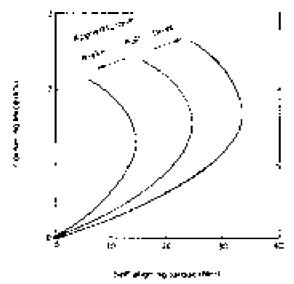**Fig. 8.35** llustration of self-aligning tor ue

Fig. 8.36 Variation of self-aligning torque with cornering force

Static steering torque, that is the torque needed to rotate the steering when the wheels are not rolling, has nothing to do with the generated self-aligning torque when the vehicle is moving. The heavy static steering torque experienced when the vehicle is stationary is due to the distortion of the tyre casing and the friction created between the tyre tread elements being dragged around the wheels' point of pivot at ground level. With radial ply tyres the more evenly distributed tyre to ground pressure over the contact patch makes manoeuvring the steering harder than with cross-ply tyres when the wheels are virtually stationary.

. . **Camber thrust** (Figs 8.37 and 8.38)

The tilt of the wheel from the vertical is known as the camber. When it leans inwards towards the turning centre it is considered to be negative and when the top of the wheel leans away from the turning centre it is positive (Fig. 8.37). A positive camber reduces the cornering force for a given slip angle relative to that achieved with zero camber but negative camber raises it.

Constructing a vector triangle of forces with the known vertical reaction force and the camber inclination angle, and projecting a horizontal component perpendicular to the reaction vector so that it intersects the camber inclination vector, enables the magnitude of the horizontal component, known as camber thrust, to be determined (Fig. 8.37). The camber thrust can also be calculated as the product of the reaction force and the tangent of the camber angle.

i.e.   Camber thrust = Wheel reaction $\times$ tan

The total lateral force reaction acting on the tyre is equal to the sum of the cornering force and camber thrust.

i.e. $F = F_c \quad F_t$

Where F = total lateral force

           F_c = cornering force

           F_t = camber thrust

When both forces are acting in the same direction, that is with the wheel tilting towards the centre of the turn, the positive sign should be used, if the wheel tilts outwards the negative sign applies (Fig. 8.38).

Thus negative camber increases the lateral reaction to side forces and positive camber reduces it.
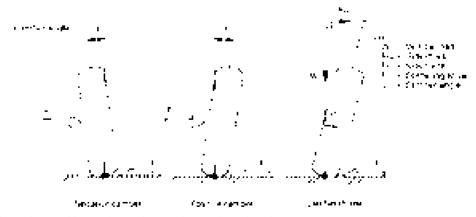

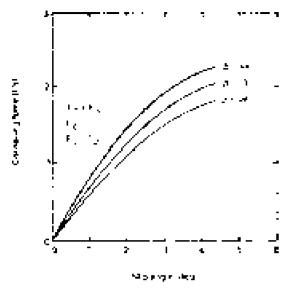**Fig. 8.37** Illustrating positive and negative camber and camber thrust

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Fig. 8.38 Effect of slip angle on cornering force with various camber angles

. .10 Camber scrub (Fig. 8.39)
When a wheel is inclined to the vertical it becomes cambered and a pro ection line drawn through the wheel axis will intersect the ground at some point. Thus if the wheel completes one revolution a cone will be generated about its axis with the wheel and tyre forming its base.

If a vehicle with cambered wheels is held on a straight course each wheel tread will advance along a straight path. The distance moved along the road will correspond to the effective rolling radius at the mid-point of tyre contact with the road (Fig. 8.39). The outer edge of the tread (near the apex) will have a smaller turning circumference than the inner edge (away from apex). Accordingly, the smaller outer
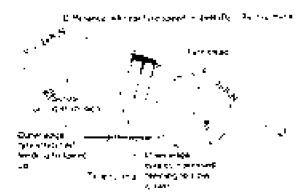

Fig. 8.39 Ilustration of camber scrub

edge will try to speed up while the larger inner edge will tend to slow down relative to the speed in the middle of the tread. As a result, the tread portion in the outer tread region will slip forward, the portion of tread near the inner edge will slip backwards and only in the centre of tread will true rolling be achieved.

To minimize tyre wear due to camber scrub modern suspensions usually keep the wheel camber below $1\frac{1}{2}$ degrees. Running wheels with a slight negative camber on bends reduces scrub and improves tyre grip whereas positive camber increases tread scrub and reduces tyre to road grip.

. .11 Camber steer (Fig. 8.40)
When a vehicle s wheels are inclined (cambered) to the vertical, the rolling radius is shorter on one side of the tread than on the other. The tyre then forms part of a cone and tries to rotate about its apex (Fig. 8.40(a and b)). ver a certain angular motion of the wheel, a point on the larger side of the tyre will move further than a point on the smaller side of the tyre and this causes the wheel to deviate from the straight ahead course to produce camber steer. Positive camber will make the wheels turn away from each other (Fig. 8.40(b)), i.e.   , whereas negative camber on each side will make the wheels turn towards each other, i.e.   . This is one of the reasons why the wheel track has to be set to match the design of suspension to counteract the inherent tendency of the wheels to either move away or towards each other.

Slightly inclining both wheels so that they lean towards the centre of turn reduces the angle of turn needed by the steered wheels to negotiate a curved path since the tyres want to follow the natural directional path of the generated cone (Fig. 8.41(a)). Conversely, if the wheels lean outwards from the centre of turn the tyres are compelled to follow a forced path which will result in a greater steering angle and consequently a degree of camber scrub (Fig. 8.41(b)).

. .1   ateral eight transfer
(Figs 8.42 and 8.43)
For a given slip angle the cornering force generally increases with the increase in vertical load. This increase in cornering force with respect to vertical load is relatively small with small slip angles, but as larger slip angles are developed between the tyre and ground increased vertical load enables much greater cornering forces to be generated (Fig. 8.42). Unfortunately the relationship between cornering force and vertical load is non-linear. This is because
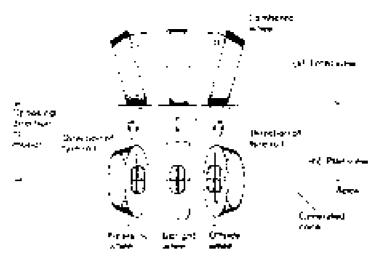
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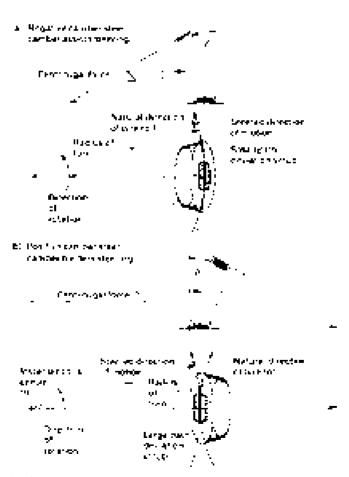
Fig. 8.40 Camber steer producing toe-out

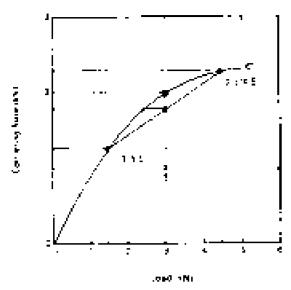**Fig. 8.41 (a and b)** rinciple of camber steer

Fig. 8.42 Effect of transverse load transfer on the cornering force developed by a pair of tyres attached to axle
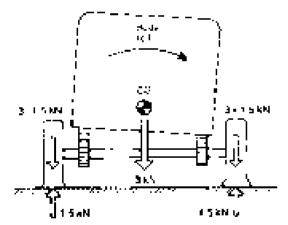

Fig. 8.43 oad transfer with body roll

an initial increase in vertical wheel load where the curve rise is steep produces a relatively large increase in cornering force, but as the imposed loading on the wheel becomes much larger a similar rise in vertical load does not produce a corresponding proportional increase in cornering force.

Consider a pair of tyres on a beam axle (Fig. 8.43), each with a normal vertical load of 3 kN. The cornering force per tyre with this load

will be 2 kN for a given slip angle of 6 . If the vehicle is sub ected to body roll under steady state movement on a curved track, then there will be certain amount of lateral weight transfer. Thus if the normal load on the inside wheel is reduced by 1.5 kN, the load on the outer wheel will be increased by the same amount.

As a result the total cornering force of the two tyres sub ected to body roll will be 1.3   2.3 = 3.6 kN (Fig. 8.42) which is less than the sum of both tyre cornering forces when they support their normal vertical load of 2 × 2 = 4 kN. The difference between the normal and body roll tyre loading thus reduces the cornering force capability for a given slip angle by 0.4 kN. This demonstrates that a pair of tyres on the front or rear axle to develop the required amount of cornering force to oppose a given centrifugal force and compensate for lateral weight transfer must increase the slip angles of both tyres. Thus minimizing body roll will reduce the slip angles necessary to sustain a vehicle at a given speed on a circular track.

.   Vehicle steady state directional stability

.5.1 Directional stability along a straight trac

eutral steer (Fig. 8.44)   Consider a vehicle moving forward along a straight path and a side force due possibly to a gust of wind which acts through the vehicle s centre of gravity which for simplicity is assumed to be mid-way between the front and rear axles. If the side force produces equal steady state slip angles on the front and rear tyres, the vehicle will move on a new straight line path at an angle to the original in proportion to the slip angles generated (Fig. 8.44). This motion is without a yaw velocity a rotation about a vertical axis passing through the centre of gravity, and therefore is known as

Note that if pro ection lines are drawn perpendicular to the tyre tread direction of motion when the front and rear tyres are generating equal amounts of slip angle, then these lines never meet and there cannot be any rotational turn of the vehicle.

versteer (Fig. 8.45)   If, due possibly to the suspension design, tyre construction and inflation pressure or weight distribution, the mean steady static slip angles of the rear wheels are greater than at the front when a disturbing side force acts through the vehicle centre of gravity, then the path
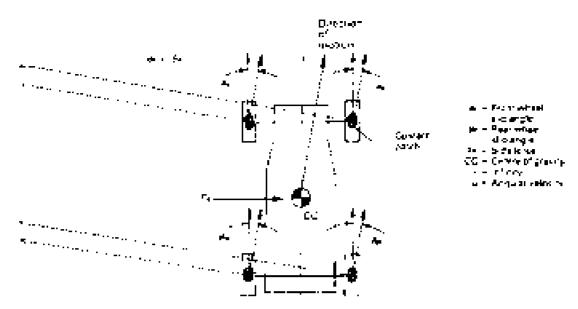
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Fig. 8.44 Neutral steer on straight track
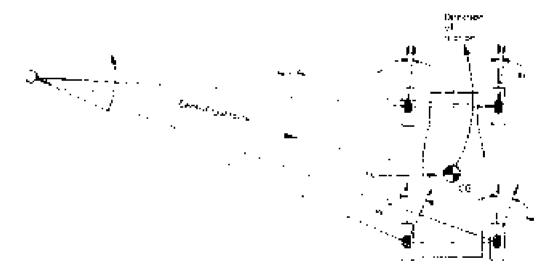
Fig. 8.45 versteer on straight track

of the vehicle is in a curve towards the direction of the applied side force (Fig. 8.45). The reason for this directional instability can be better understood if pro ection lines are drawn perpendicular to the direction the tyres roll with the generated slip angles. It can be seen that these pro ection lines roughly intersect each other at some common point known as the instantaneous centre, and

therefore a centrifugal force will be produced which acts in the same direction as the imposed side force. Thus the whole vehicle will tend to rotate about this centre so that it tends to swing towards the disturbing force. To correct this condition known as   the driver therefore has to turn the steering in the same direction as the side force away from the centre of rotation.

Fig. 8.46 Understeer on straight track

nderteer (Fig. 8.46)   Now consider the situation
of a vehicle initially moving along a straight path
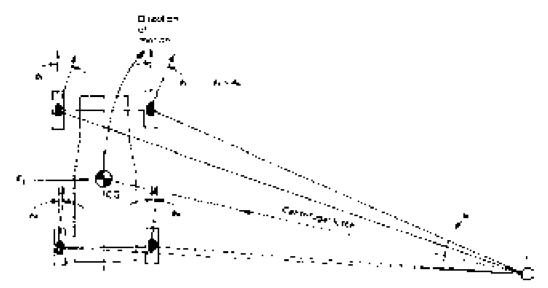when a disturbing side force is imposed through the
vehicle s centre of gravity. This time there is a
larger slip angle on the front tyres than at the rear
(Fig. 8.46). Again pro ect lines perpendicularly to
the tyre tread direction of motion when they are
generating their slip angles but observe that these
pro ections meet approximately at a common point
on the opposite side to that of the side force. The
vehicle s directional path is now a curve away from
the applied side force so that a centrifugal force will
be produced which acts in opposition to the dis-
turbing side force. Thus the vehicle will be encour-
aged to rotate about the instantaneous centre so
that it moves in the same direction as the disturbing
force. Correction for this steering condition which
is known as   is achieved by turning the
steering in the opposite direction to the disturbing
force away from the instantaneous centre of rota-
tion. It is generally agreed that an oversteer condi-
tion is dangerous and undesirable, and that the slip
angles generated on the front wheels should be
slightly larger than at the rear to produce a small
understeer tendency.

.5.   *Directional stability on a curved trac*
True rolling of all four wheels can take place when
pro ection lines drawn through the rear axle and
each of the front wheel stub axles all meet at a
common point somewhere along the rear axle pro-
 ected line. This steering layout with the front

wheels pivoted at the ends of an axle beam is
known as the   but strictly it
can only be applied when solid tyres are used and
when the vehicle travels at relatively slow speeds.
With the advent of pneumatic tyres, the instant-
aneous centre somewhere along the extended pro ec-
tion from the rear axle now moves forwards relative
to the rear axle. The reason for the positional change
of the instantaneous centre is due to the centrifugal
force produced by the vehicle negotiating a corner
generating an opposing cornering force and slip
angle under each tyre. Therefore pro ection lines
drawn perpendicular to the direction each wheel
tyre is moving due to the slip angles now converge
somewhere ahead of the rear axle. This is essential if
approximate true rolling conditions are to prevail
with the vehicle travelling at speed.

versteer (Fig. 8.47)   If the slip angles of the rear
wheel tyres are made greater than on the front tyres
when the vehicle is turning a corner (Fig. 8.47), the
pro ected lines drawn perpendicular to the direc-
tion of motion of each tyre corresponding to its slip
angle will all merge together at some common
point (dynamic instantaneous centre) forward of
the rear axle, further in and therefore at a shorter
radius of turn than that produced for the Acker-
mann instantaneous centre for a given steering
wheel angle of turn.

   Under these driving conditions the vehicle will
tend to steer towards the bend. Because the radius
of the turn is reduced, the magnitude of the

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Fig. 8.47 versteer on turns

centrifugal force acting through the vehicle centre of gravity will be larger it therefore raises the oversteer tendency of the vehicle. At higher vehicle speeds on a given circular path, the oversteer response will become more pronounced because the rise in centrifugal force will develop more tyre to ground reaction and correspondently increase the slip angles at each wheel. This is an unstable driving condition since the vehicle tends to turn more sharply into the bend as the speed rises unless the lock is reduced by the driver. For a rear wheel drive vehicle the application of tractive effort during a turn reduces the cornering stiffness and increases the slip angles of the rear wheels so that an oversteering effect is produced.

ndersteer (Figs 8.48 and 8.49) If the slip angles generated on the front wheel tyres are larger than those on the rear tyre when the vehicle is turning a corner (Fig. 8.48) then pro ection lines drawn perpendicular to the direction of motion of each tyre, allowing for its slip angle, will now all intersect approximately at one point also forward of the rear axle, but further out at a greater radius of turn than that achieved with the Ackermann instantaneous centre.

With the larger slip angles generated on the front wheels the vehicle will tend to steer away from the bend. Because the radius of turn is larger, the magnitude of the centrifugal force produced at the centre of gravity of the vehicle will be less than for the oversteer situation. Thus the understeer tendency generally is less severe and can be corrected by turning the steering wheels more towards the bend. If tractive effort is applied when negotiat-

Fig. 8.48 Understeer on turns

ing a circular path with a front wheel drive vehicle, the cornering stiffness of the front tyres is reduced. As a result, the slip angles are increased at the front, thereby introducing an understeer effect.

A comparison between the steered angle of the front wheels or driver s steering wheel angle and vehicle speed for various steering tendencies is shown in Fig. 8.49. It can be seen that neutral steer maintains a constant steering angle throughout the vehicle s speed range, whereas both under- and oversteer tendencies increase with speed. An important difference between over- and understeer is that understeer is relatively progressive as the speed rises but oversteer increases rapidly with speed and can become dangerous.

. **Tyre marking identification** (Tables 8.1 and 8.2)
To enable a manufacturer or customer to select the recommended original tyre or to match an equivalent tyre based on the vehicle s application
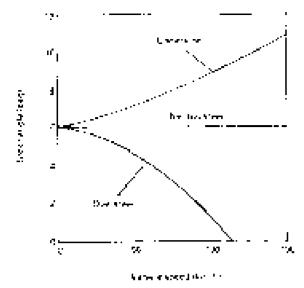
Fig. 8.49 Relationship of steer angle speed and vehicle speed of neutral steer understeer and oversteer

requirement, wheel and tyre dimensions, tyre profile, maximum speed and load carrying capacity, a standard marking code has been devised.

.6.1 Car tyres
Current tyres are marked in accordance with the standards agreed by the European Tyre and Rim Technical rganisation. Tyres with cross-ply construction and normal 82% aspect ratio do not indicate these features but radial construction and lower aspect ratios are indicated. Tyre section width, speed capacity, wheel rim diameter and tread pattern are always indicated.

a) 165 SR 13 Mx
b) 185/70 VR 15   W

165 or 185	=	nominal section width of tyre in millimetres
70	=	70% aspect ratio (Note no figures following 165 indicates 82% aspect ratio)
S or V	=	letter indicates speed capability (S = 180, V = 210 km/h)
R	=	radial construction
13 or 15	=	nominal wheel rim diameter in inches
M , W	=	manufacturer s tread pattern

In some instances section width is indicated in inches.

6.45   14

6.45	=	nominal section width of tyre in inches
	=	letter indicates speed capability (speed symbol = 160 km/h)
14	=	nominal wheel rim diameter in inches

No aspect ratio or construction indicated. Therefore assume 82% aspect ratio and cross-ply construction.

A revised form of marking has been introduced to include the maximum speed and load carrying capacity of the tyre under specified operating conditions.

A letter symbol indicates the maximum speed (Table 8.1) and a numerical code will identify the load carrying capacity (Table 8.2).

205/70 R 13 80 S M V

205	=	normal section width in millimetres
70	=	70% aspect ratio
R	=	radial construction
13	=	nominal wheel rim diameter in inches
80	=	load index (from Table 8.2: 80 = 450 kg)
S	=	speed symbol (from Table 8.1: S = 180 km/h)
M V	=	manufacturer s tread pattern code

ab 8.1 Speed symbols (SS)

Speed symbol (SS)	Speed (km/h)	SS	Speed (km/h)	SS	Speed (km/h)	SS	Speed (km/h)
A	2	E			12	R	1
A		F			1	S	1
A				N	1	T	1
B			1		1	U	2
C			11		1	H	21

(V = over 210)

ab 8.2   oad index ( )

	kg		kg		kg		kg
1				1		22	2
2				1		2	
	1	1		1		2	
1	11	1		1		2	
1	12	1		1	1	2	
2	1	1	2	1		2	1
	1	2	21	1		2	1

.6. ight medium and heavy truc tyres

Truck tyres sometimes include ply rating which indicates the load carrying capacity.

$$10 \; R \; 20.0 \; PR12 \qquad A$$

10 = nominal section width of tyre in inches
R = radial construction
20.0 = nominal wheel rim diameter in inches
PR12 = ply rating
A = manufacturer s tread pattern

The revised form of marking indicates the load carrying capacity and speed capability for both single and twin wheel operation. The ply rating has been superseded by a load index because with improved fabric materials such as rayon, nylon and polyester as opposed to the original cotton cord ply, fewer ply are required to obtain the same strength using cotton as the standard, and therefore the ply rating does not give an accurate indication of tyre load bearing capacity.

$$295/70 \; R \; 22.5 \; Tubeless \; 150/140 \qquad T$$

295 = nominal section width of tyre in millimetres
70 = 70% aspect ratio
R = radial construction
22.5 = nominal rim diameter in inches
150 = load index for singles (from Table 8.2: 150 = 3350 kg per tyre)
140 = load index for twins (from Table 8.2: 140 = 2500 kg per tyre)
= speed symbol (from Table 8.1: = 120 km/h)
T = manufacturer s tread pattern

.7 heel balancing

The wheel and tyre functions are the means to support, propel and steer the vehicle forward and backward when rolling over the road surface. In addition the tyre cushions the wheel and axle from all the shock impacts caused by the roughness of the road contour. For the wheel and tyre assembly to rotate smoothly and not to generate its own vibrations, the wheel assembly must be in a state of rotatory balance.

An imbalance of the mass distribution around the wheel may be caused by a number of factors as follows:

a) tyre moulding may not be fitted concentric on the wheel rim,
b) wheel lateral run out or buckled wheel rim,

c) tyre walls, crown tread thickness may not be uniform all the way round the carcass when manufactured,
d) wheel lock when braking may cause excessive tread wear over a relatively small region of the tyre,
e) side wall may scrape the curb causing excessive wear on one side of the tyre,
f) tyre over or under inflation may cause uneven wear across the tread,
g) tyre incorrectly assembled on wheel relative to valve.

Whichever reason or combination of reasons has caused the uneven mass concentration (or lack of mass) about the wheel, one segment of the wheel and tyre will become lighter and therefore the tyre portion diametrically opposite will be heavier.

ence the heavy region of the tyre can be considered as a separate mass which has no diametrically opposing mass to counteract this inbalance.

Consequently the heavier regions of the wheel and tyre assembly when revolving about its axis (the axle or stub axle) will experience a centrifugal force. This force will exert an outward rotating pull on the support axis and bearings. The magnitude of this outward pull will be directly proportional to the out of balance mass, the square of the wheel rotational speed, and inversely proportional to the radius at which the mass is concentrated from its axis of rotation.

i.e. Centrifugal force $(F) = \dfrac{^2}{} \; (N)$

where F = centrifugal force (N)
= out of balance mass (kg)
= linear wheel speed (m/s)
= radius at which mass is concentrated from the axis of rotation (m)

If, due to excessive braking, 100 g of rubber tread has been removed from a portion of the tyre tread 250 mm from the centre of rotation, determine when the wheel has reached a speed of 160 km/h the following:

a) angular speed of wheel in revolutions per minute,
b) centrifugal force.

inear speed of wheel $= \dfrac{160 \times 10^3}{60}$

$= 2666.666$ m min

or $= \dfrac{2666.666}{60}$

$= 44.444$ m s

a) Angular speed of wheel $= \dfrac{}{}$

$$= \frac{2666.666}{0.5}$$

$$= 1697.65 \text{ rev min}$$

b) Centrifugal force $F = \dfrac{2}{}$

$$= \frac{0.1\,(44.444)^2}{0.25}$$

$$= 790.1 \text{ N}$$

From this calculation based on a vehicle travelling at a speed of one hundred miles per hour (160 km/h) and a typical wheel size for a car, the hundred gramme imbalance of the tyre produces a radial outward pull on the wheel axis of 790 Newtons. The magnitude of this 790 Newton force can be best appreciated by converting it to weight (mass) (79 kg) and then imagine lifting and carrying 79 one kilogramme bags of sugar for some distance.

.7.1 Cyclic movement of a heavy spot on a heel relative to the ground (Fig. 8.50)

When a road wheel rolls over a flat surface for one complete revolution, a point P on its circumference starting and finishing at ground level plots a curve known as a cycloid which represents the changing linear speed of the point P during each cycle of rotation (Fig. 8.50). For the short time point P is at ground level, its velocity remains at zero and at its highest position from the ground its forward velocity will be at a maximum. The average forward velocity of point P is at mid-height axle level,

this also being the vehicle s forward speed. Thus the top of the tyre moves at twice the speed of the vehicle and in the same direction.

If point P is a heavy spot on the tyre, it will accelerate from zero to a maximum velocity for half a revolution and then decelerate to zero velocity again to complete the second half revolution. Since this spot has mass and changes its velocity, it will be sub ected to a varying acceleration force which acts in a direction tangential to this curve. Consequently the direction of the inertia pull caused by this heavy spot constantly changes as the wheel moves forwards. The greatest reaction experienced on the wheel occurs within the short time the heavy spot decelerates downwards to ground level, momentarily stops, changes its direction and accelerates upwards. ence at the end of each cycle and the beginning of the next there will be a tendency to push down and then lift up the tyre from the ground. At very low speeds this effect may be insignificant but as the vehicle speed increases, the magnitude of the accelerating force acting on this out of balance mass rises and thereby produces the periodic bump and bounce or erking response of the tyre.

The balancing of rotating masses can be considered in two stages: firstly the static balance in one plane of revolution, this form of balance is known as   and secondly the balance in more than one plane of revolution, commonly referred to as

.7. Static balance (Fig. 8.51)

This form of imbalance is best illustrated when a wheel and tyre assembly has been mounted on the hub of a wheel balancing machine which is then spun around by hand and released. The momentum put
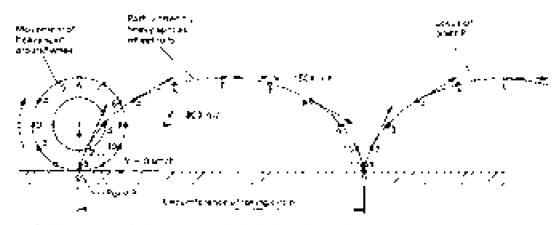

Fig. 8.50 Cyclic movement of a heavy spot on wheel relative to the road

into rotating the wheel tends to spin it a few times. It then stops momentarily and starts to oscillate to and fro with decreasing amplitude until eventually coming to rest. If a chalk mark is made on the tyre at its lowest point and the wheel is now turned say 90 and then released again, it will immediately commence to rotate on its own, one way and then the other way, until coming to rest with the chalk mark at the lowest point as before. This demonstrates that the heaviest part of the wheel assembly will always gravitate to the lowest position. If a small magnetic weight is placed on the wheel rim diametrically opposite the heavy side of the wheel and it has been chosen to be equivalent to the out of balance mass, then when the wheel is rotated to any other position, it remains in that position without any tendency to revolve on its own. If, however, there is still a slight movement of the wheel, or if the wheel wants to oscillate faster than before the magnetic weight was attached, then in the first case

the balancing weight is too small and in the second case too large. This process of elimination by either adding or reducing the amount of weight placed opposite the heavy side of the wheel and then moving round the wheel about a quarter of a turn to observe if the wheel tries to rotate on its own is a common technique used to check and correct any wheel imbalance on one plane. When the correct balancing weight has been determined replace the magnetic weight with a clip-in weight of similar mass. With a little experience this trial and error method of statically balancing the wheel can be quick, simple and effective.

The consequences of a statically unbalanced wheel and tyre is that the heavy side of the wheel will pull radially outwards as it orbits on a fixed circular path around its axis of rotation, due to the centrifugal force created by the heavier side of the wheel (Fig. 8.51). If the swivel pins and the centre of the unbalanced mass are offset to each other,
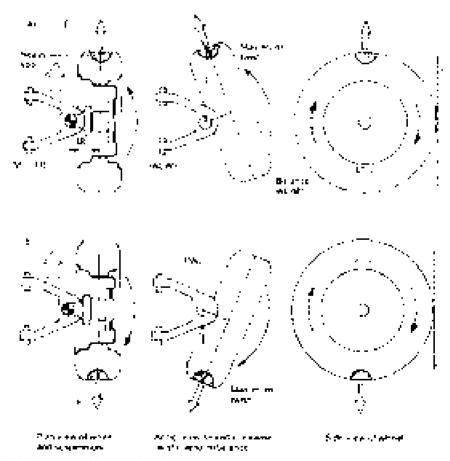

Fig. 8.51 (a and b) Illustration of static wheel imbalance

then when the heavy spot is in the horizontal plane pointing towards the front of the vehicle a moment of force is produced (= F) which will endeavour to twist the stub axle and wheel assembly anticlockwise about the swivel pins (Fig. 8.51(a)). As the wheel rolls forward a further half turn, the heavy spot will now face towards the rear so that the stub axle and wheel assembly will try to swivel in the opposite direction (clockwise) (Fig. 8.5(b)).

ence with a statically unbalanced tyre the stub axle will twist about its pivot every time the heavier side of the wheel completes half a revolution between the extremities in the horizontal plane. The oscillations generated will thus be transmitted to the driver s steering wheel in the form of tremors which increase in frequency and magnitude as the vehicle s speed rises. If there is a substantial amount of swivel pin or kingpin wear, the stub axle will be encouraged to move vertically up or down on its supporting oints. This might convey vibrations to the body via the suspension which could become critical if permitted to resonate with possibly the unsprung or sprung parts of the vehicle.

.7.3 Dynamic balance (Fig. 8.52)
If a driven drum is made to engage the tread of the tyre so that the wheel is spun through a speed range there is a likelihood that the wheel will develop a violent wobble which will peak at some point as the wheel speed rises and then decreases as the speed is further increased.

This generated vibration is caused by the balance weight having been placed correctly opposite the heavy spot of the tyre but on the wheel rim which may be in a different rotational plane to that of the original out of balance mass. As a result the tyre heavy spots pull outwards in one plane while the balance weight of the wheel rim, which is being used to neutralize the heavy region of the tyre, pulls radially outwards in a second plane. Consequently, due to the offset of the two masses, a rocking couple is produced, its magnitude being proportional to the product of centrifugal force acting through one of the masses and the distance between the opposing forces (= F). The higher the wheel speed and the greater the distance between the opposing forces, the greater the magnitude of the rocking couple will be which is causing the wheel to wobble.

The effects of the offset statically balanced masses can be seen in Fig. 8.52(a, b and c). When the heavy spot and balancing weight are horizontal (Fig. 8.52(a)), the mass on the outside of the wheel

points in the forward direction of the vehicle and the mass on the inside of the wheel points towards the rear so that the wheel will tend to twist in an anticlockwise direction about the swivel pins. With a further 180 rotation of the wheel, the weights will again be horizontal but this time the outer weight has moved to the rear and the inner weight towards the front of the vehicle. Thus the sense of the unbalanced rocking couple will have changed to a clockwise one. For every revolution of the wheel, the wheel will rock in both a clockwise and anticlockwise direction causing the driver s steering wheel to erk from side to side (Fig. 8.52(c)). Note that when the masses have moved into a vertical position relative to the ground, the swivel pins constrain the rocking couple so that no movement occurs unless the swivel ball oints or kingpins are excessively worn.

The characteristics of the resulting wheel wobble caused by both static and dynamic imbalance can be distinguished by the steady increase in the amplitude of wheel twitching about the swivel pins with rising wheel speed in the case of static unbalanced wheels, whereas with dynamic imbalance the magnitude of wheel twitching rises to a maximum and then declines with further wheel speed increase (Fig. 8.53). Thus with dynamic imbalance, a wheel can be driven at road speeds which are on either side of the critical period of oscillation (maximum amplitude) without sensing any undue instability. If the wheel is driven within the relative narrow critical speed range violent wheel wobble results.

Slackness in the swivel pins or steering linkage ball oints with unbalanced tyres will promote excessive wheel twitch or wobble, resulting not only in the steering wheel sensing these vibrations, but causing heavy tyre tread scrub and wear.

.7. ethods of balancing heels
Wheel balancing machines can be of the on- or off-vehicle type. The on-vehicle wheel balancer has the advantage that it balances the wheel whole rotating wheel assembly which includes the hub, brake disc or drum, wheel and tyre. owever, it is not really suitable for drive axles because the transmission drive line does not permit the wheel hub to spin freely (which is essential when measuring the imbalance of any rotating mass). ff-vehicle balancing machines require the wheel to be removed from the hub and to be mounted on a rotating spindle forming part of the balancing equipment.
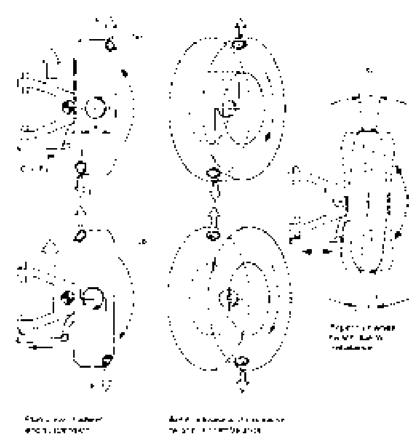
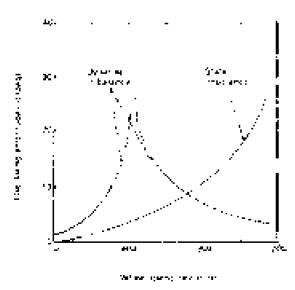**Fig. 8.52 (a–c)** llustration of dynamic wheel imbalance

Fig. 8.53 Relationship of wheel speed and oscillating amplitude for both static and dynamic imbalance

alancing machine which balances statically and dynamically in two separate planes (Fig. 8.54) The wheel being balanced is mounted on the spindle of the mainshaft which is supported by a pair of spaced out ball bearings. This machine incorporates a self-aligning ball bearing at the wheel end mounted rigidly to the balancing machine frame, whereas the rear bearing is supported between a pair of stiff opposing springs which are themselves attached to the balancing machine frame. An electric motor supplies the drive to the spindle by way of the engagement drum rubbing hard against the tyre tread of the wheel assembly being balanced.

When the wheel and tyre is spun and the assembly commences to wobble about the self-aligning bearing, the restraining springs attached to the second bearing absorb the out of balance forces and the deflection of the mainshaft and spindle.

An electro-magnetic moving coil vibration detector (   ) is installed vertically between the second bearing and the machine frame. When
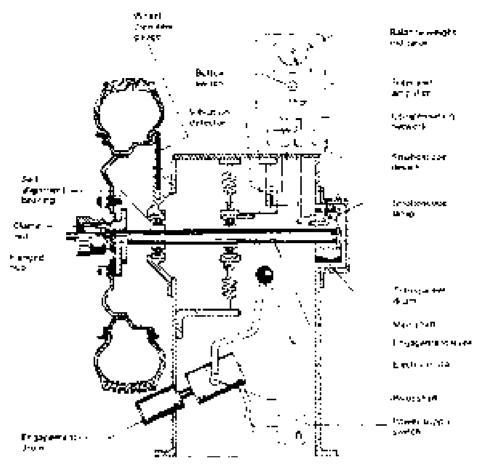
Fig. 8.54 heel balancing machine which balances statically and dynamically in two different planes

the wheel assembly wobbles, the armature (rod) in the centre of the transducer coil moves in and out of a strong magnetic field provided by the permanent magnet. This causes the armature coil to generate a voltage proportional to the relative movement of the rod. The output signal from the detector is a direct measure of the imbalance of the wheel assembly. It is therefore fed into a compensating network which converts the signals into the required balance weight to be attached to the outside of the wheel rim in the left hand plane. These modified, but still very weak, electrical signals are then passed through a filter which eliminates unwanted side interference. They are then amplified so that they can activate the stroboscope device and the weight indicator meter.

The weight indicator meter computes the voltage amplitude signal coming from the detector and, when calibrated, indicates the size of the weight to be added to the plane of balance, in this case the

outside of the wheel. Conversely, the stroboscope determines the angular phase of the balance weight on the wheel. This is achieved by the sinusoidal voltage being converted into a sharply defined bright flash of light in the stroboscope lamp. A rotating numbered transparent drum is illuminated by the stroboscope flash and the number which appears on the top of the drum relates to the phase position of the required balance weight.

(Fig. 8.54) Mount the wheel onto the flanged multi-hole steel plate. Align the wheel stud holes with corresponding threaded holes in the flange plate and screw on the wheel studs provided. Slide the wheel hub assembly along the spindle until the inside of the wheel rim ust touches the ad ustable distance rod and then lock the hub to the spindle via the sleeve nut. The positioning of the wheel assembly relative to the supporting self-aligning bearing

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is important since the inside wheel rim now is in the same rotating plane as the centre of the bearing. Any couple which might have been formed when the balance weights were attached to the inside of the wheel rim are eliminated as there is now no offset.

(Fig. 8.54) To achieve dynamic balance, switch on the power, pull the drive roller lever until the roller is in contact with the tyre and allow the wheel to attain full speed.

nce maximum speed has been reached, push the lever so that the roller is freed from the tyre. If the wheel assembly is unbalanced the wheel will pass through a violent period of wobble and then it will steady again as the speed falls. While the wheel is vibrating, the magnitude and position of the imbalance can be read from the meter and from the stroboscope disc aperture respectively. A correction factor is normally given for the different wheel diameters which must be multiplied by the meter reading to give the actual balance weight. Select the nearest size of balance weight to that calculated, then rotate the wheel by hand to the number constantly shown on the stroboscope disc when the wheel was spinning and finally attach the appropriate balance weight to the top of the wheel on the outside (away from the machine). Thus the outer half of the wheel is balanced.

(Fig. 8.54) Static balance is obtained by allowing the wheel to settle in its own position when it will naturally come to rest with the heaviest point at bottom dead centre. Select a magnetic weight of say 50 grammes and place this on the inside rim at top dead centre and with this in position turn the wheel a quarter of a revolution. If the magnetic weight is excessive, the weighted side will naturally gravitate towards the bottom but if it is insufficient, the weight will rise as the wheel slowly revolves. Alter the size of the magnetic weight and repeat the procedure until there is no tendency for the wheel to rotate on its own whatever its position. The wheel is now statically and dynamically balanced and a quick check can be made by repeating the spin test for dynamic balance. nce the correct static balance weight has been found, replace the magnetic weight by a clip-on type.

alancing machine which dynamically balances in two planes (Fig. 8.55) The machine is so constructed that the wheel being balanced is mounted on the spindle of the mainshaft which is supported by a pair of spaced out ball bearings housed in a cylindrical cradle, which itself is supported on four strain rods which are reduced in diameter in their mid region (Fig. 8.55). An electric motor supplies the drive to the mainshaft via a rubberized flat belt and pulleys.

Vibration detectors are used to sense the out of balance forces caused by the imbalance of the wheel assembly. They are normally of electro-mechanical moving coil type transducers. The tranducer consists of a small armature in the form of a stiff rod which contains a light weight coil. The rod is free to move in a strong magnetic field supplied by a permanent magnet. The armature rods are rigidly attached to the mainshaft and bearing cradle and the axes of the rods are so positioned as to coincide with the direction of vibration. The housing and permanent magnets of the detectors are mounted onto the supporting frame of the cradle. The relative vibratory motion of the armature rod to the casing causes the armature coil to generate a voltage proportional to the relative vibrational velocity.

The output signals from the two detectors are fed into a compensating network and then into the selector switch. The compensating network is so arranged that the output signals are proportional to the required balance weights in the left and right hand balancing planes respectively. The output voltages from the selector switch are very small signals and will include unwanted frequency components. These are eliminated by the filter. At the same time the signals are amplified so that they can operate the stroboscope device and both weight indicating meters. These weight indicating meters measure the amplitude of the voltage from the detectors and when calibrated indicate the actual weights to be added in each plane. The stroboscope device changes the sinusoidal voltage into a sharply defined pulse which occurs at the same predetermined point in every cycle. This pulse is converted into a very bright flash of light in the stroboscope lamp when focused on the rotating numbered transparent drum one number will appear on the top of the apparently stationary surface. The number is a measure of the relative phase position of the voltage and is arranged to indicate the position of the required balance weight.

(Fig. 8.55)
Mount the wheel over the spindle and slide the conical adaptor towards the wheel so that its taper enters the central hole of the wheel. Screw the sleeve nut so that the wheel is centralized and wedged against the flanged hub (Fig. 8.55). Any
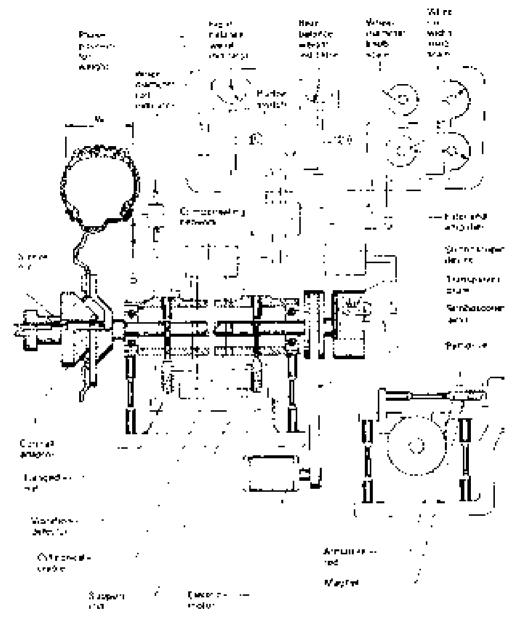
Fig. 8.55 heel balancing machine which dynamically balances in two planes

existing balance weights should now be removed and the wheel should also be brushed clean.

Before the wheel assembly is actually balanced on the machine, the two basic wheel dimensions must be programmed into the electronic network circuit. This is carried out by simply moving the wheel diameter indicator probe against the inside of the wheel rim and reading off the wheel diameter and also measuring with a caliper gauge normally provided the wheel width. The wheel diameter and wheel rim width measurements are then set by rotating the respective potentiometer knob on the display console to these dimensions so that the electronic network is altered to correct for changes in the centrifugal force and rocking couples which will vary with different wheel sizes.

Balancing machines of this type usually measure and provide correction for wheel imbalance for both static and dynamic balance with respect to both the outer and inner wheel rim rotating planes.

The start button is now pressed to energize the electric motor. As a result the drive belt will rotate the mainshaft and spin the wheel assembly under test.

First the state of wheel balance in the outer wheel rim plane of rotation is measured by pressing the outer rim weight indicator meter switch. The meter pointer will align on the scale with size of balance weight required in grammes the stroboscope indicator window will also show a number which corresponds to the wheel position when a balance weight is to be attached.

nce the balance weight size and angular position has been recorded, the wheel assembly is brought to a standstill by pressing the stop button and then releasing it when the wheel ust stops rotating.

The wheel should now be rotated by hand until the number previously observed through the stroboscope window again appears, then attach the selected size of balance weight to the top of the outer wheel rim.

The whole procedure of spinning the wheel assembly is then repeated but on the second time the inner rim weight indicator button is pressed so that the balance weight reading and the phase position of the wheel refer to the inner rim rotating plane. Again the wheel is braked, then turned by hand to the correct phase position given by the stroboscope number. Finally the required balance

weight is attached, this time to the inner wheel rim at the top of the wheel.

.7.5 heel and tyre run out

Before proceeding to balance the wheel asseembly the wheel should be checked for run-out in both lateral and radial directions relative to the axis of rotation. If the wheel or tyre run-out is excessive it should be corrected before the wheel assembly is balanced.

(Fig. 8.56) If the wheel being examined has been acked clear of the ground and when spun appears to wobble so that the wheel rim or tyre wall moves axially inward or outward in a wavy fashion lateral (sideway), run-out is taking place. This may be caused by either the wheel rim being buckled or the tyre being fitted unevenly around the rim of the wheel so that the wheel assembly will produce dynamic imbalance. eflating the tyre and repositioning the bead against the inside of the wheel rim will usually correct any tyre lateral run-out. ateral run-out of the wheel itself should be no greater than 2.0 mm.

(Fig. 8.57) If with the wheel acked clear of the ground, the wheel and tyre assembly appears to lift and fall every time the wheel completes one revolution, then the distance from the axis of rotation to the tyre tread instead of being constant around the periphery of the tyre is varying.
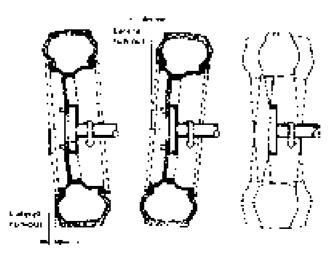

Fig. 8.56 llustration of lateral tyre run-out

Fig. 8.57 Illustration of radial tyre run-out

This error is usually caused by the tyre having been fitted eccentrically about the wheel rim so that when the wheel assembly is spun, radial run-out will be observed, and as a result, the wheel assembly will be in a state of static imbalance. Tyre eccentricity can usually be cured by repositioning the tyre on the wheel rim. The maximum wheel eccentricity should not exceed 2.0 mm.

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