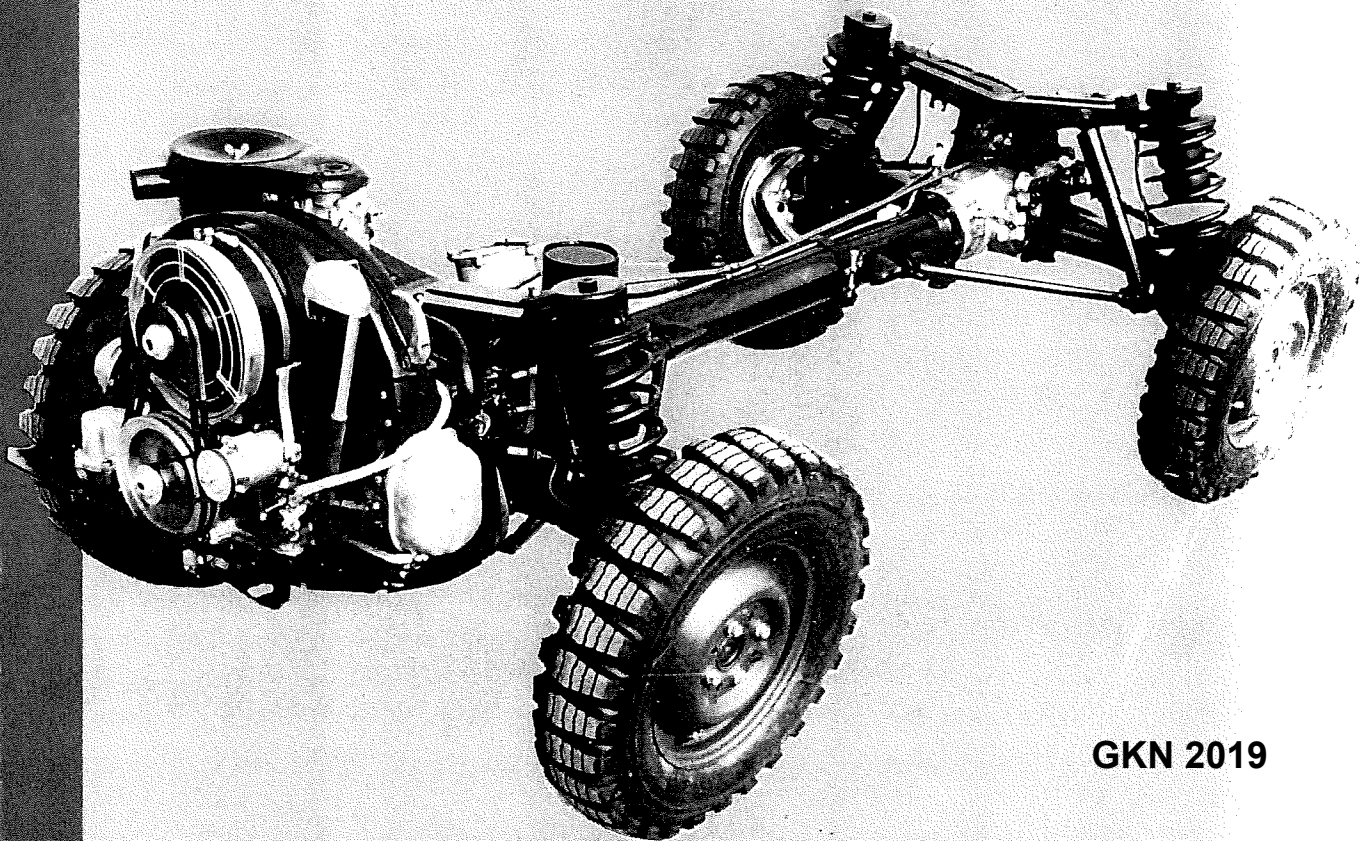


ELEVENTH EDITION

THE MOTOR VEHICLE

K NEWTON W STEEDS TK GARRETT



GKN 2019

The Motor Vehicle

Eleventh Edition

K. NEWTON

MC, BSc, ACGI, AMInstCE, MIMechE

*Late Assistant Professor, Mechanical and Electrical Engineering Department,
The Royal Military College of Science*

W. STEEDS

OBE, BSc, ACGI, FIMechE

*Late Professor of Mechanical Engineering,
The Royal Military College of Science*

T. K. GARRETT

CEng, FIMechE, MRAeS

Sometime Editor of Automobile Engineer

Butterworth-Heinemann Ltd
Linacre House, Jordan Hill, Oxford OX2 8DP



PART OF REED INTERNATIONAL BOOKS

OXFORD LONDON BOSTON
MUNICH NEW DELHI SINGAPORE SYDNEY
TOKYO TORONTO WELLINGTON

First published by Iliffe & Sons 1929
Eighth edition 1966
Ninth edition 1972
Tenth edition published by Butterworths 1983
Eleventh edition 1989
Reprinted 1991

© T. K. Garrett and Exors of K. Newton and W. Steeds 1989

All rights reserved. No part of this publication may be reproduced in any material form (including photocopying or storing in any medium by electronic means and whether or not transiently or incidentally to some other use of this publication) without the written permission of the copyright holder except in accordance with the provisions of the Copyright, Designs and Patents Act 1988 or under the terms of a licence issued by the Copyright Licensing Agency Ltd, 90 Tottenham Court Road, London, England W1P 9HE. Applications for the copyright holder's written permission to reproduce any part of this publication should be addressed to the publishers

British Library Cataloguing in Publication Data

Newton, K. (Kenneth)

The motor vehicle – 11th ed.

1. Motor vehicle engineering – for technicians

I. Title II. Steeds, W. (William)

III. Garret, T. K. (Thomas Kenneth)

629.2

ISBN 0 7506 0407 7

0 408 01082 7 Butterworth-Heinemann International Edition

Library of Congress Cataloguing in Publication Data

Newton, K. (Kenneth)

The motor vehicle/K. Newton, W. Steeds, T. K. Garrett – 11th ed.

p. cm.

Includes index

ISBN 0 7506 0407 7

1. Automobiles – Design and construction

I. Steeds, W. (William) II. Garrett, T. K. III. Title

TL205.N4 1989

629.2'3–dc19

GKN 2019

Printed and bound in Great Britain by Hartnolls Ltd, Bodmin, Cornwall

Contents

Preface to the eleventh edition	v
---------------------------------	---

Part 1 Fundamentals

1 Some fundamental principles of mechanics	3
--	---

Part 2 The Engine

2 General principles of heat engines	15
3 Engine balance	36
4 Constructional details of the engine	55
5 Six-, eight and twelve-cylinder engines	136
6 Sleeve-valve and special engines	164
7 The compression-ignition engine	173
8 The two-stroke engine	235
9 Carburation	258
10 Carburettors in practice	292
11 Emission control	354
12 Fuel supply and air cleaning	373
13 Supercharging and superchargers	381
14 Fuels for automobile engines	396
15 Lubrication	409
16 Engine cooling	422
17 Potential for future development	435
18 Bearings, gearing, chain and belt drives	464

Part 3 Transmission

19 Transmission requirements	475
20 Clutches	487
21 Why is a gearbox necessary?	517

22	Constructional arrangements of gearboxes	527
23	Epicyclic and pre-selector gearboxes	560
24	Torque converters and automatic gearboxes	573
25	Universal joints and driving steered wheels	620
26	The differential	631
27	The back axle	649
28	Axle constructions	657
29	The double-reduction axle	664

Part 4 The Carriage Unit

30	The basic structure	673
31	Brakes	682
32	Servo- and power-operated, and regenerative braking systems	708
33	Front axle and steering mechanism	750
34	Suspension principles	774
35	Suspension systems	809
36	Six-wheel vehicles	842
	Index	855

Chapter 26

The differential

The differential is the device that divides the torque input from the propeller shaft equally between the two output shafts to the wheels, regardless of the fact that they may be rotating at different speeds, for instance on rounding a corner. In principle, as can be seen from Fig. 26.1, it is a set of two bevel gears with a bevel pinion between them. The bevel pinion can be likened to a balance beam pivoted at its centre, its ends registering between pairs of teeth on the differential gears.

If a force P is applied at the central pivot, in a direction tangential to the two differential gears, and if the beam either does not swing, or swings at a uniform velocity, it follows that the forces at the ends of the beam will be equal to $P/2$. Hence, equal forces are applied at equal distances from the centres of the differential gears, and therefore the torques they transmit to the halfshafts are equal. Clearly, the force P represents the pressure between the differential pinion and its pin, while the forces $P/2$ represent the pressures between the teeth of the bevel pinion and those with which they mesh on the two differential gears.

In Fig. 26.2 is a typical differential unit in the back axle of a car. It consists of a drum-shaped cage A, which is carried in ball bearings BB in the axle casing and is therefore free to rotate about the axis XX of the road wheels. Fixed to the cage A is a crown wheel C driven by the bevel pinion D. The arrangement is similar to that of Fig. 27.1, except that the crown wheel is fixed to the cage, in which the bevel gears F_1F_2 and their pinions GG rotate on axes mutually at right angles. There are now two 'differential' or 'drive' shafts E_1E_2 , the outer ends of which are connected to the road wheels. Their inner ends pass through the bosses of the differential cage A in which they are quite free to turn. Inside the differential cage, their ends are splined into the bevel wheels F_1F_2 with which the bevel pinions GG mesh. The pinions GG are free to turn on the pin H fixed in the differential cage.

It should be clear that if the differential cage is held and the wheel F_1 is turned in the forwards direction at, say 2 rev/min then the wheel F_2 will be turned backwards at 2 rev/min, since it is equal in size to F_1 . Moreover, since these motions are relative to the differential cage, they will not be affected by any motion of that cage. If, therefore, the differential cage is rotating at, say, 200 rev/min in the forwards direction and the wheel F_1 is

still turning at 2 rev/min forwards relative to it, the wheel F_2 will still be turning at 2 rev/min backwards relative to it.

The actual speed of the wheel F_1 will then be 202 rev/min, because its forward motion of 2 rev/min relative to the differential cage is added to the forwards motion of 200 rev/min of that cage. The actual speed of the wheel F_2 will be 198 rev/min because its backwards motion of 2 rev/min relative to the differential cage is subtracted from the forwards motion of 200 rev/min of that cage.

This is the action that occurs when a car moves in a circle: the road wheels are constrained to move at different speeds and do so by virtue of one wheel going faster than the differential cage while the other goes an equal amount slower than the differential cage. Thus the speed of the differential cage is the mean of the road wheel speeds. When the car moves in a straight line, the road wheels turn at the same speed as the differential cage, and the differential pinions do not have to turn on their pins at all.

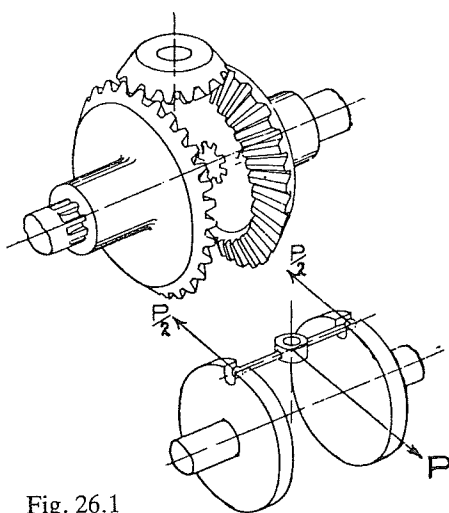


Fig. 26.1

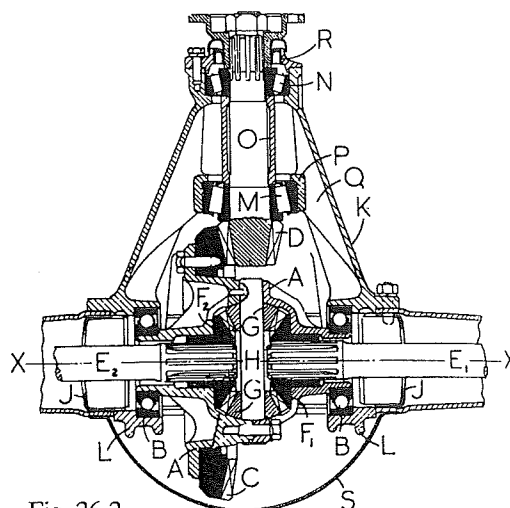


Fig. 26.2

The above description should make it clear how the road wheels can turn at different speeds; it remains to show that when so doing they are driven with equal torques. In Fig. 26.1 the bevel wheels are shown replaced by discs having notches in their peripheries. Lying with its ends in these notches is a beam. If a force P is applied to the centre of the beam in a direction tangential to the discs as shown, then if the beam does not turn about the vertical axis, or if it turns about that axis with uniform velocity, the forces at the ends of the beam must be equal and each will be equal to $P/2$. The reactions of the forces acting on the ends of the beam act on the discs, hence equal forces are applied to the discs at equal distances from their axes, and therefore the twisting moments or torques acting on the discs are equal.

It should readily be seen that the bevel pinion acts in a manner precisely similar to the beam. Hence the torques transmitted to the drive shafts are equal and each will be equal to half the torque applied to the differential case by the final drive. In the actual differential the force P appears as a pressure between the bevel pinion and its pin while the forces $P/2$ appear as pressures between the teeth of the bevel pinion and the bevel wheels.

26.1 Another arrangement of the bevel final drive

The bevel final drive is sometimes arranged in a different manner from that described previously, generally because some other difference in construction necessitates the change. The principle of this other method is shown in Fig. 26.3. The propeller shaft is coupled to the shaft A which passes right across the centre portion of the axle casing B in which it is supported. At the centre of the axle the shaft A is enlarged and formed into pins PP to carry the differential pinions CC. These mesh with the differential wheels Q_1 Q_2 , which are integral with the bevel pinions D_1 D_2 , of which D_1 meshes with the large bevel wheel E and D_2 with the smaller bevel wheel F. The bevel wheels E and F are supported in the axle casing and are splined on to the shafts which drive the road wheels.

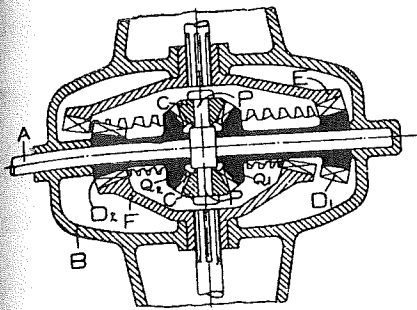


Fig. 26.3

Of course, the gear ratio between D_1 and E is the same as the ratio between D_2 and F. The action of the differential is just the same as in the conventional axle, but the reduction of speed now occurs between the differential and the road wheels instead of between the propeller shaft and the differential cage. The differential therefore runs at a higher speed than the road wheels, enabling smaller wheels to be used in it.

It should also be clear that it is quite possible to have the axes of the shafts inclined to each other in the end view, and advantage has been taken of this feature to arch the back axle casing and to tilt the rear wheels in order to reduce the overhang on the axle and to bring the road wheels perpendicular to the curved surface of a cambered road. Arched back axles are not now used, and the form of final drive described is uncommon.

26.2 Spur, or planetary type, differential

Most planetary type differentials have spur instead of bevel gears, as in Fig. 26.4, and therefore are often called the spur type. The wheels A, B are splined to the drive shafts that drive the road wheels. Meshing with the wheel A is a spur pinion E_1 , whose teeth extend nearly across the gap between the wheels A and B. The spur pinion F_1 meshes in a similar way with the wheel B, and at the centre the two pinions mesh together. The pinions are carried on pins which are supported by the ends of the differential cage, which in this design is formed of the worm wheel C of the final drive and two end cover plates DD. It should be clear that if the differential cage is held fixed and the wheel A is turned in, say, the clockwise direction, the pinion F_1 in a clockwise direction and the wheel B

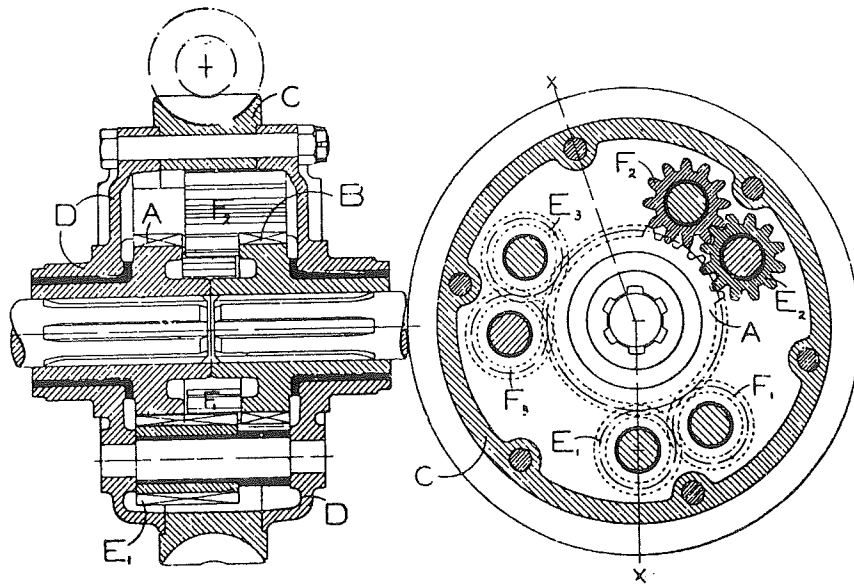


Fig. 26.4 Spur type differential

in a counter-clockwise direction. Hence, if one of the differential wheels goes faster than the differential cage, the other differential wheel will go an equal amount slower, just as in the bevel type.

As regards the equality of the torques, the torque on the wheel A is due to the pressure of the teeth of the pinion E_1 . This pressure tends to make the pinion E_1 revolve on its pin, and this tendency is opposed by a pressure between the teeth of the two pinions at the centre. This last pressure tends to make the pinion F_1 revolve on its pin, and this tendency is opposed by the pressure between the teeth of the pinion F_1 and the wheel B. If the pinion E_1 (and therefore the pinion F_1 also) is at rest relatively to, or if it is revolving uniformly on, its pin then the two pressures acting on it must be equal. Hence, all the pressures between the teeth of the wheels A and B and the pinions E_1 and F_1 are the same, and hence the torques acting on the wheels A and B are equal. Three pairs of pinions are provided as shown.

26.3 Traction control differentials

With the introduction of four-wheel-drive cars for use on the road, in countries where snow and ice are prevalent in winter, and for rallying, traction control devices, ranging from the complex electronic systems to simple differential locks, became necessary. The reason is that, if the grip of a driven wheel on one side is reduced, for example by ice, to zero, the total traction that can be transmitted by a conventional differential is also reduced to zero. This follows from the definition of a differential in the opening sentence of this chapter: a simple differential without any form of traction control reduces the torque transmitted to the wheel on the side that is gripping until it equals that to the wheel that is slipping. For similar reasons, traction control may be beneficial when applied to the differentials interposed in the transmission lines between the front and rear axles of four-wheel drive vehicles, see Section 19.4.

Table

Trade name

Sure-Drive

ZF

Lok-O-Matic

Powr-Lok

Trac-Aide

Trac-Loc

Traction Loc

Traction E

Sure-Grip

Super Trac

No-SPIN

Detroit Loc

True Trac

Knight

Torsen

Max-Trac

Super Ma

Viscodri

Mercede

Var
one w
was in
The s
be lo
differ
crow
slidin
vehic
tippe
by ro

Table 10—SOME TRACTION CONTROL DIFFERENTIAL SYSTEMS

Trade name	Type	Supplier
Sure-Drive	Freewheel	Formerly Borg-Warner, now Auburn Gear Inc.
ZF	Cam-and-pawl	Zahnradfabrik Friedrichshafen (ZF)
Lok-O-Matic	Multi-plate clutch, ramp-actuated	Zahnradfabrik Friedrichshafen (ZF)
Powr-Lok	Multi-plate clutch, ramp-actuated and rampless	GKN Axles Ltd., Light Division, ZF, Spicer Axle Division, Dana Corporation
Trac-Aide	Multi-plate clutch, ramp-actuated, but rampless	GKN Axles Ltd., Light Division, ZF, Spicer Axle Division, Dana Corporation
Trac-Loc	Multi-plate clutch, rampless	GKN Axles Ltd., Light Division, ZF, Spicer Axle Division, Dana Corporation
Traction Lok	Multiplate clutch, rampless	Ford
Traction Equaliser	Clutch type	Rockwell
Sure-Grip	Cone clutch	Formerly Borg-Warner, now Auburn Gear Inc.
Super Traction	Cone clutch	Formerly Borg-Warner, now Auburn Gear Inc.
No-SPIN	Face dog clutch	Tractech, Dyneer Corporation
Detroit Locker	Face dog clutch	Tractech, Dyneer Corporation
True Trac	Helical gear	Tractech, Dyneer Corporation
Knight	Helical and worm	Knight-Mechadyne Ltd.
Torsen	Worm and spur	Gleason, Power Systems Division and Quaife Power Systems Ltd.
Max-Trac	Variable leverage gear	Fairfield Manufacturing Co.
Super Max-Trac	Variable leverage gear, with friction	Fairfield Manufacturing Co.
Viscodrive	Viscous coupling	FF Developments, Viscodrive GmbH, Tochigi Fuji Sangyo KK
Mercedes ASD	Electronic	Daimler-Benz

Various forms of differential designed to prevent total loss of traction if one wheel spins freely have been in use since soon after the road vehicle was invented, and a list of those currently available is given in Table 10. The simplest is a mechanism by means of which the differential cage can be locked to the inner ends of the halfshafts, to cause the whole differential and halfshaft assembly to rotate in unison with the crownwheel. In most cases, this mechanism is either a dog-clutch or a sliding muff coupling. Such systems are commonly used on off-road vehicles, including light as well as the heavy commercial types such as tipper trucks. They may be actuated pneumatically, hydraulically, mechanically by rod or cable, or electrically.

Their main disadvantages are the extra complexity and weight represented by the controls and the fact that, in most instances, the vehicle has to be stationary while the lock is engaged: if, when the vehicle is stationary, the driven wheels bed down into very soft ground, moving off again may be difficult. A serious disadvantage, too, is that if the driver forgets to disengage the lock as he takes the vehicle off soft on to hard ground, transmission wind-up will ensue and cause excessive tyre wear and, ultimately, a mechanical failure in the driveline.

In some electronically-controlled automatic systems a wheelspin sensor, similar to the wheel-lock sensor in anti-skid systems, as described in Section 32.15, is used in conjunction with a computer which simply applies a brake to stop the spinning wheel so that the torque can be reacted back through a conventional simple differential to drive the wheel on the other side. There are also systems which, instead of applying the brake, reduce the engine power by cutting the fuel injection to one or more cylinders, which has the advantage of not only reducing the power output but also of using the inactive cylinder or cylinders as compressors to produce engine braking. Alternatively, in diesel engines, the fuel supply can be reduced to all cylinders or cut completely to one or more. An advantage of automatic operation, electronic or otherwise, is the absence of any remote manual controls and of the risk of the driver's forgetting to disengage the lock.

26.4 Vehicle design implications of traction control

A prime requirement, especially for road-going vehicles, is that the traction control device should come both into and out of operation rapidly. It must do so, however, entirely unobtrusively, otherwise the traction can become jerky and the steering can be adversely affected. Other problems arising include dissipation of the heat developed by the slipping clutch and the cam devices, wear, juddering and noise generated, and a loss of overall efficiency.

Ideally, the slip should be limited to shedding only that proportion of torque that exceeds the tractive capacity of the driven wheel on the side on which the wheelspin is tending to occur. This reduces jerk and optimises acceleration potential. Generally, however, the best that can be achieved is either a total lock-up or a torque bias ratio of up to 5 or 6:1 for rear axles but, for front axles, the torque bias has to be limited to no more than about 2.5:1, for the avoidance of adverse effects on steering. The effect of both axles is usually understeer while, on a front axle, there is also an increase in effort required at the steering wheel; both effects increase with torque bias ratio.

On the back axle, the steering effect is limited to that due to the asymmetry of the drive, coupled with the influence that it has on the slip angles of the tyres when cornering. At the front, the loads in the track-rod go out of balance instantly when the differential locks. In fact, dependent on the steering geometry and the degree of slip in the differential, the steering may even lock, especially when the vehicle is being turned from a straight ahead course. This is easy to understand if one envisages a condition in which changing the direction of the steering can be effected

only if there is relative rotation between the two front wheels: with the differential locked, this rotation cannot occur.

It follows that only soft-action, low bias-ratio traction control differentials are practicable for front axles. Indeed, most vehicle manufacturers avoid the problem simply by fitting an inter-axle traction control differential or a viscous coupling either alone, as on the rear-engine VW Transporter Synchro 4×4 , or in conjunction with a similar one in the rear final drive gear.

With only an inter-axle traction control unit installed, one axle can take over the drive if traction is lost beneath either one or two wheels on the other. On the other hand, if rear wheel spin occurs where both inter-axle and rear traction control differentials are fitted, the lost traction is transferred to the other wheel on the same axle, the front wheels continuing to transmit their share of the drive. However, if either front wheel spins, all the traction is transferred to the rear axle. This layout is particularly appropriate for vehicles in which, to give characteristics similar to those of rear-wheel-drive, the transmission system is geared so that the proportion of torque transmitted to the rear is in any case higher than that to the front axle.

If a viscous coupling serving as an inter-axle differential has to transmit all the torque from the engine it must be fairly large. Its actual dimensions, however, normally depend on what proportion of the total torque has to be transmitted from one axle to the other, and this in turn depends on factors such as the handling characteristics desired and the effects of the torque transmission on tyre slip angles.

Renault, for their front-wheel-drive cars, advocate a 65:35 torque split in favour of the front end, to retain the typical front-wheel-drive handling characteristics. For the Ford front-engine rear-wheel-drive *Scorpio*, on the other hand, the gearing is such that 66% of the drive goes to the rear axle and only 34% to the front, so the rear wheels are the more likely to spin. If this happens all the torque is transmitted to the front wheels. On this model the viscous coupling is on the rear end of the gearbox, and the torque is transmitted through a multi-strand toothed chain drive to a shaft, extending forwards alongside the engine, to the front axle.

In the VW installation, the inter-axle viscous drive coupling, see Section 26.10, is housed in the front final drive casing so that its weight helps to offset what otherwise would be a preponderance on the rear, owing to the fact that the engine is there. The gearing ratio, front to rear, is 1:1 but, owing to slip in the coupling, all the drive is normally transmitted directly, through the rear wheels. However, if one of these wheels spins, and traction is therefore lost on the rear axle, the coupling takes over and transmits all the torque to the front axle.

In general, the proportions of a viscous coupling incorporated in a driven axle depend on whether it is interposed between the two halfshafts or between the differential cage on one halfshaft.

26.5 Multi-plate clutch type traction control device

For cars, the most commonly used automatic device is a limited-slip mechanism incorporating some sort of clutch. The aim is at introducing

into the differentially geared coupling between the two halfshafts a degree of friction which will not only prevent one wheel from spinning freely, but also, by reacting the input torque to that wheel, provide for the transmission to the other wheel a torque equal to that arising from the frictional resistance.

The multi-plate clutch type is the most popular for cars. Most are of the loading, torque-sensitive type: as the torque differential between the two halfshafts increases, so also does the pressure on the clutch plates, as can be seen from Fig. 26.9, ultimately causing it to lock completely. Some other types, of which the ZF cam-and-pawl type is a good example, are speed-sensitive. Only these two types will be described in detail, the former because of its widespread use and the latter because its principle of operation is perhaps the most difficult to understand. For full descriptions of the others listed, the reader is recommended to refer to an article in *Automotive Engineer*, February/March 1987, Vol. 12, No. 1.

The other types listed in Table 10 can be summarised in broad principle as follows. The simplest is a freewheel coupling, see Section 22.21, on each halfshaft so that, for example during a turn, the outer wheel and shaft overruns its coupling and all the drive is transmitted by the inner wheel. This has several disadvantages; first, during differential operation, the total tractive potential is limited to that capable of being transmitted through only one tyre; secondly, the transitions from symmetrical to asymmetrical drive can adversely affect the steering; thirdly, unless the freewheels can be locked, not only engine braking but also driving in reverse are impossible.

26.6 Some other clutch types

Among the cone-clutch types that were first introduced by Borg-Warner, and are now produced by Warner Gear Inc., are both loading and unloading types. The clutch of the unloading type, Fig. 26.5, in contrast to that in the loading type, is engaged by Belleville springs and therefore its arrangement is such that, as the transmitted torque increases, the

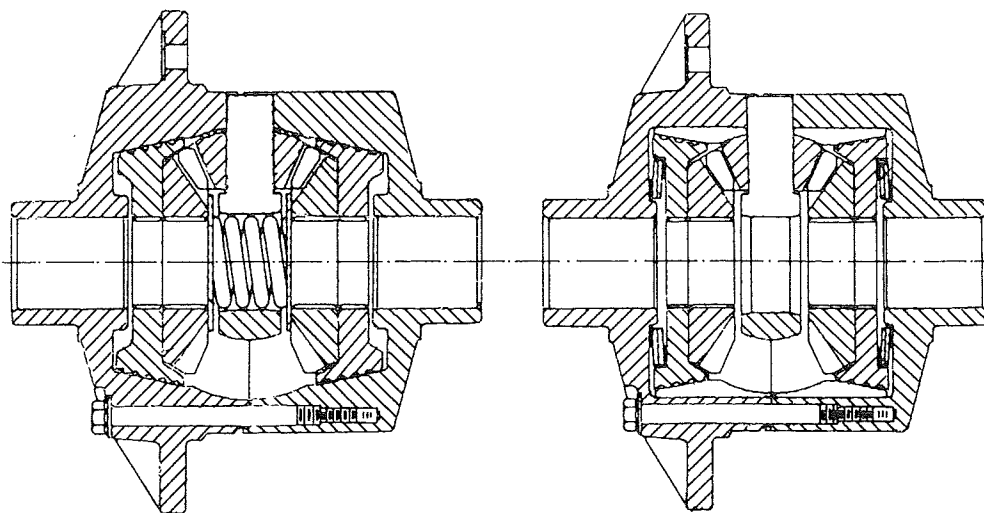


Fig. 26.5 Warner Gear loading and unloading type traction control differentials (left and right respectively)

engagement force progressively unloads it until it disengages altogether. Consequently, the loading type is best for vehicles for operation at high speed and in which, as the speed is reduced, the torque is increased, for example by using lower gears. In contrast, the unloading type is principally for vehicles, such as those employed for earthmoving, operated mainly at low speed and high torque.

Traction control differentials incorporating dog clutches are harsh in action and are suitable mainly for applications in which total torque transfer is required. Consequently they are mainly for off-highway vehicles. Most of the gear type units, on the other hand, depend for their operation on the friction generated in either the irreversible or semi-irreversible gears. The action of the semi-irreversible type is relatively gentle and therefore has minimal effect on steering, so the majority are suitable for incorporation in front axles.

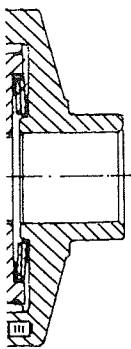
26.7 Gear type traction control devices

A gear type in which friction is not the factor relied upon is the Max-Trac, produced by Fairfield Manufacturing Company. This has gears with wide teeth, the diametral pitch of which varies along their width. As the gears rotate, the contact points traverse to and fro along the teeth, thus varying the effective ratio and pulsating the output. This pulsation of torque, it is claimed, is a measure a driver might take deliberately to get out of difficulty if his vehicle were stuck in the mud. The Super Max-Trac is more effective because friction is introduced in the gearing system, to provide a differential torque in addition to the pulsation effect.

Of the other gear types listed, all are of the planetary type. Both the Dyneer and Quaife units have pairs of helical differential pinions, similar to those in Fig 26.4 but floating in pockets in the housing that contains the differential gear assembly. The gears of each pair intermesh, one meshing also with the differential gear on one side and the second with that on the other side. Torque bias, up to about 2:5 to 3:1, is generated by the friction due to both the axial and radial thrusts of these gears in their housings. If the torque reactions from the wheels on each side are equal, the axial thrusts arising from the helical teeth of the pinions balance, so these pinions float in their housing pockets; if the torque reactions become unequal, the axial thrusts on the pinions go out of balance, forcing them against the ends of their individual housing pockets and generating friction.

An advantage of this type is that its maximum torque bias (the highest ratio of the torque differential between the wheels) and its load capacity can be adjusted at the design stage by two methods. First, varying the helix angle of the gear teeth affects not only its capacity but also both the gear-pinion separation forces and axial thrust, and thus the friction due to radial pressure between the pinions and their housings. Secondly, both the friction and torque capacity can be increased by varying the number of pinions.

Of similar layout is the Knight Mechadyne unit, but instead of intermeshing, its planetary pinions are interconnected by wormwheel idlers, the axes of which are radially disposed, as in Fig. 26.6, instead of being parallel to those of the gears and pinions. The advantages of this



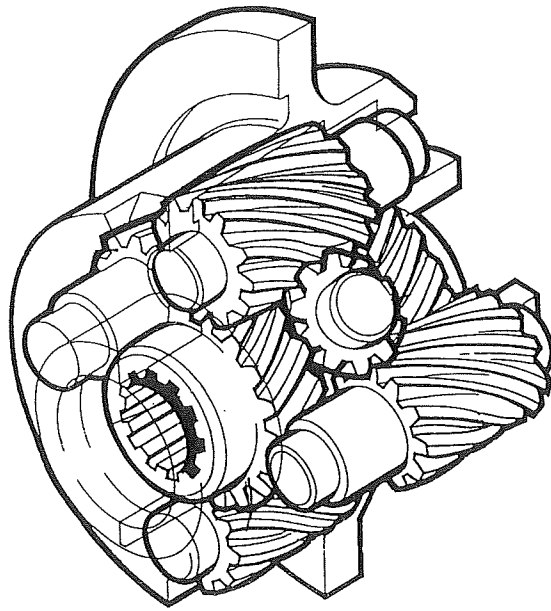


Fig. 26.6

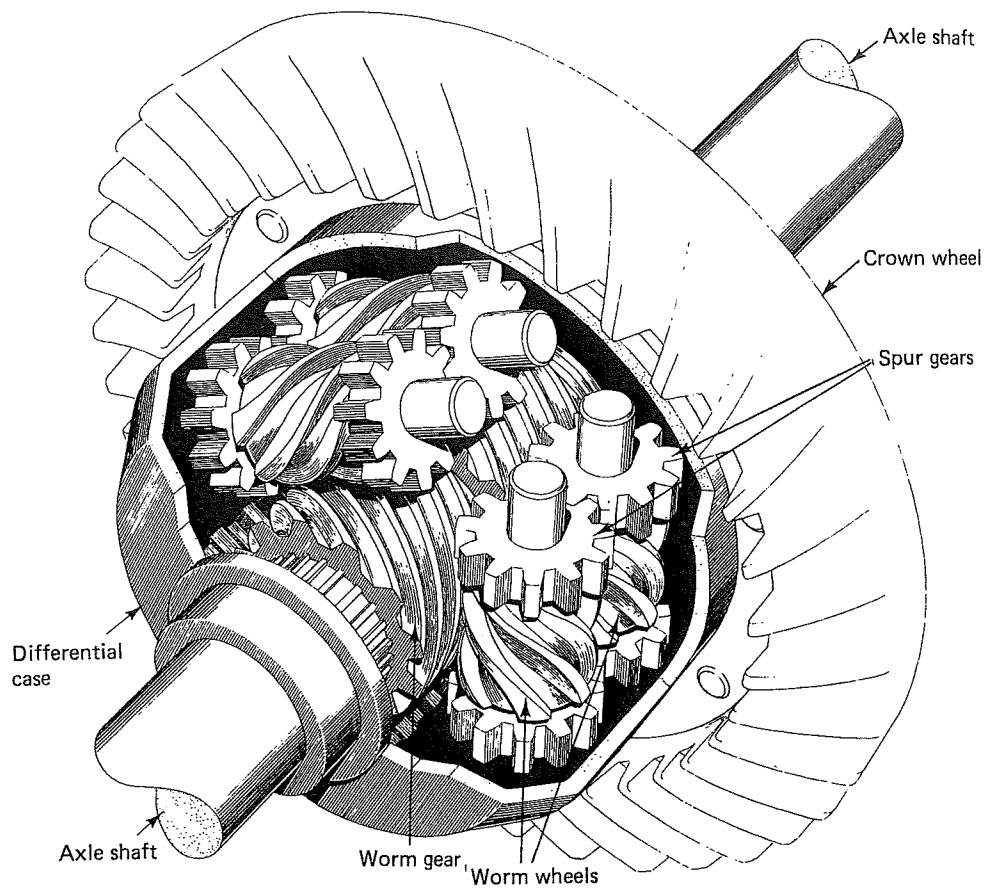


Fig. 26.7

arrange
substitu
paralle

The
Vernor
manuf:
Gleasc
gears a
the tw
ends.
have t
other
differ
wheel
the u
divide
rotati

26.8

The
simpl
comp
hard
wear
few
the
disel

Fig

in
th
op
cc
pa
sl
al
d

arrangement are first its compactness and secondly, by virtue of the substitution of radially and axially orientated intermeshing gears for parallel ones, the obviation of some high precision machining operations.

The Gleason Torsen type, Fig. 26.7, was originally invented, in 1958, by Vernon Gleasman, who called it the Dual Drive. Subsequently, it was manufactured by Tripple-D, before the production was taken over by Gleason, in 1982. It differs from those already described in that its output gears are wormwheels meshing with pairs of worm type planetary pinions, the two in each pair being interconnected by straight spur gears on their ends. These pinions could not themselves intermesh because they must have teeth the profiles of which are designed for meshing, not with each other but specifically with those of their wormwheels. If the torque differential between the two halfshafts suddenly rises, perhaps due to wheelspin, the irreversibility of its worm and wormwheel elements causes the unit to lock. On the other hand, if the torque is relatively equally divided, the worm and wormwheels are in floating mesh, so differential rotation can occur, for example when the vehicle is steered round a curve.

26.8 ZF limited slip differential

The ZF cam-and-pawl type unit, Fig. 26.8, has the advantages of simplicity, and therefore inherently low cost, and light weight and compactness. On the other hand, it is liable to wear fairly rapidly if worked hard. It is particularly suitable for application to racing cars, where rapid wear is immaterial because it is normally not required to last longer than a few races. Moreover, racing drivers are all highly skilled and can cope with the steering and other effects associated with its jerky engagement and disengagement.

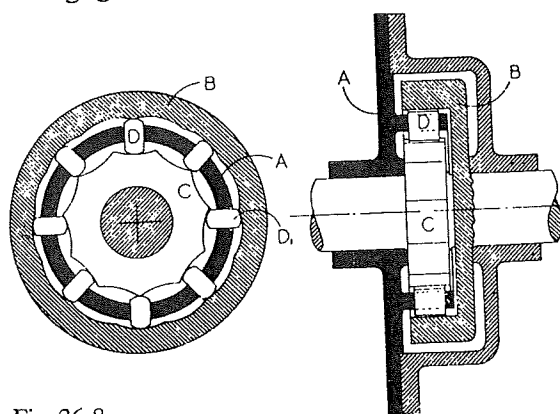


Fig. 26.8

It comprises three main parts: a driving member A which is, in effect, integral with the crownwheel and which therefore corresponds to and takes the place of the differential cage of a conventional final drive gear; two opposed cam-rings, B and C, splined one to each halfshaft, and therefore corresponding to the differential gears, which they replace; and a set of pawls, or cam-followers, free to slide, radially between the cam rings, in slots in the previously mentioned driving member. These cam rings can alternatively be in the form of face-cam rings, to reduce the overall diameter, though at the expense of increased width.

In general, the performance of the unit depends to a major extent on the numbers of lobes on its cam rings. For most applications, about thirteen are ground in the outer ring and only eleven around the periphery of the inner one so that, when the rings rotate relative to one another, the radial motions of the plungers are phased, giving a slight gearing effect: when ring B rotates once relative to A, ring C makes 13/11 revs.

Torque from the crownwheel is transmitted through the flanks of the slots in the driving member A to the pawls D. So, if the car were jacked up clear of the ground, both halfshafts would rotate at equal speeds. Imagine now that the driving member A is rotating anti-clockwise and a brake is applied to only the wheel driven by the halfshaft to which ring C is splined: in these circumstances, the cam at C would force the plunger between the letters B and C in the left-hand diagram outwards but, to be able to move out, it would have to force the outer ring B, and its associated road wheel, to rotate anti-clockwise faster than A.

Obviously, there would be a considerable frictional resistance between the plungers and cams, so the anti-clockwise motion of the unbraked wheel, despite its being clear of the ground and therefore otherwise free to rotate, would absorb a great deal of torque. This torque would, of course, be reacted by the brake on the opposite wheel until it was released, when both wheels would again rotate at equal speeds.

Now imagine the brake released and the car's being lowered, one wheel coming down on to wet ice, and therefore still free to spin, and the other descending on to dry road. The circumstances, so far as the drive is concerned, will ultimately be similar to those in the previous paragraph except in that while the wheel on ice will be rotating slowly and laboriously, owing to friction in the differential, the torque reaction on the other wheel, instead of being applied to the brake, will drive the car.

26.9 Multi-plate clutch type

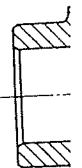
The most widely used clutch type has been developed from the Powr-Lok unit produced originally by the Thornton Axle Co. but now manufactured by the Spicer Axle Division, Dana Corporation, who also make a variant that they call the Trac-Aide. Both have multi-plate clutches interposed between the differential gears and the outer ends of their cages. The difference between the two is as follows. The multi-plate clutches in the Powr-Lok are engaged primarily by axial pressure exerted as a result of movement of the differential spindles up the flanks, or ramps, in V-shape slots, though about a third of the engagement thrust is attributable to the axial components of the meshing forces, tending to move the differential gears outwards relative to their pinions; the similar clutches in the Trac-Aide and the later Trac-Loc devices, on the other hand, are engaged solely by the axial components.

Units of these types are produced also in the UK by GKN Axles Ltd., Light Division, and in Germany under the name Lok-o-Matic by ZF, while others have been produced under names such as Anti-Slip, Traction Lok, Posi-Traction, and Super Traction by various companies including Borg Warner, Ford, GM and at least one company in Japan.

A disadvantage of most of the multi-plate clutch actuated devices is that,

being served by very little preload, a slight dimensional change in application time, however, the clutch

The pinion Fig. 26.9 limited speed is bolted casing. parallel split into the differential radially



being servo actuated by the transmitted torque, they are useless if there is very little or no tractive resistance. Most, however, have clutches with preload springs, usually of the Belleville type, to ensure that there is always a slight degree of resistance to relative rotation and to cater for dimensional variations arising from manufacturing tolerances and wear. In applications where wheel spin occurs over a high proportion of operating time, however, this arrangement can tend to accelerate the rate of wear of the clutch plates,

The principle of the ZF Lok-o-Matic, ramp-actuated type is illustrated in Fig. 26.9. Around one end of the cylindrical housing for the combined limited slip and differential mechanism is a flange to which the crownwheel is bolted. This housing is carried in two bearings, one each side, in the axle casing. Machined in its bore are four slots equally pitched around, and parallel to, its axis of rotation. Within it is the differential cage, which is split into halves on the longitudinal vertical plane that contains the axes of the differential pinions; each half houses a differential gear. Projecting radially outwards from, and equally pitched around, the periphery of each

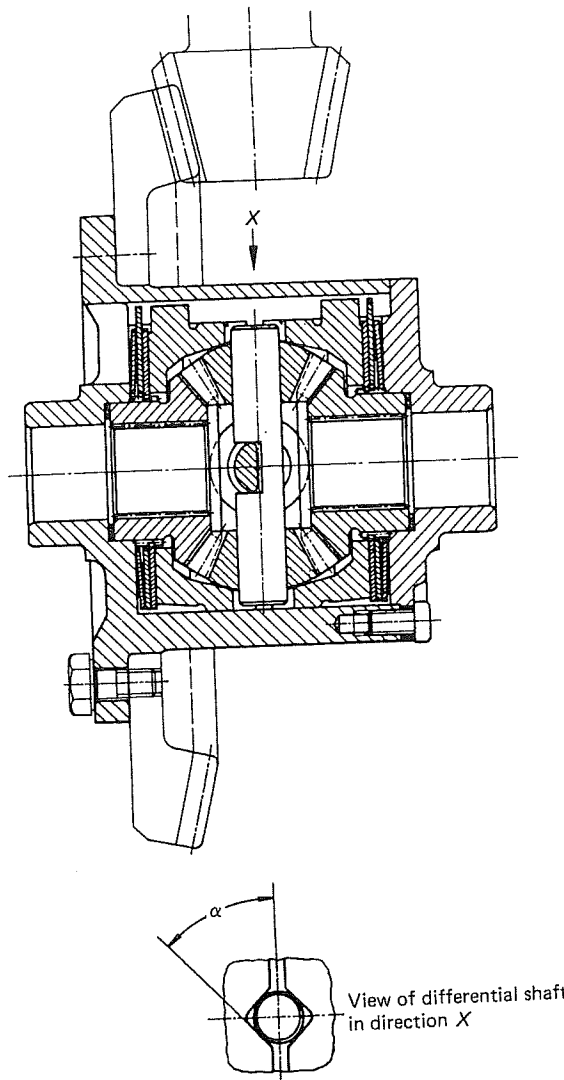


Fig. 26.9

half of the cage are four lugs, which are a clearance fit in the slots in the bore of the housing so that, although the cage is driven by the rotation of the housing, its halves are allowed a limited degree of differential rotation.

The differential gears are splined to the inner ends of the halfshafts and mesh, in the usual manner, with four differential pinions. However, these pinions, instead of being carried on a one-piece spider, are on the ends of two separate pins cranked at their centres, where they cross over to form a two-piece, four-arm spider. Furthermore, the ends of the pins are not carried in cylindrical bearings but seat in V-slots in the mating faces of the two-piece differential cage, as in the scrap view in Fig. 26.9. In some versions, however, the cylindrical form of these ends is retained.

With either arrangement since the transmission of torque to the halfshafts tends to rotate the spider relative to the differential cage it causes the ends of the pins to ride up the flanks of the V-shape slots in which they seat. This forces outwards the two parts of the cage and thus applies pressure axially to the clutch plates, which are splined alternately to the differential cage and the bosses of the differential gears. Consequently, the greater the applied torque, the tighter is the engagement of the clutches, and thus any tendency towards differential rotation of the gears and the halfshafts to which they are splined becomes increasingly limited.

26.10 The traction control by viscous coupling

Gaining ground rapidly is the viscous coupling, initially introduced as a practicable transmission component by FF Developments Ltd and now manufactured by Viscodrive GmbH, a joint GKN-ZF company. It is installed in, among others, BMW, Ford and VW cars and light commercial vehicles. Its major advantages are simplicity and relative freedom from wear and maintenance. A disadvantage of its being incapable of transmitting a differential torque without a differential speed is that there tends to be a significant time lag before it comes into operation as a limited slip device.

The performance of this type of coupling is to a major extent dependent on the properties of the fluid used, which is the main reason why development, from its pre-FF conception in the nineteen-twenties to its widespread acceptance in the early nineteen-eighties, took such a long time. Although viscous couplings have been used for competition cars they are more suitable for other types and, because they can be designed for soft action, they can be installed in front axles. Since their torque transmission capacity increases with rotational speed, they tend to be less suitable for drive-axle than for inter-axle installation, that is, before the speed has been reduced by the crownwheel and pinion, though they are used for both.

In its simplest form the coupling consists of two sets of plates alternately splined to a housing and a shaft, with the viscous fluid between them. the housing is driven and the torque is transmitted from it by viscous friction between the driving and driven plates to the shaft. The plates are perforated to increase viscous drag by optimising the distribution of fluid between them. These perforations also optimise the hump mode of operation, a phenomenon that will be explained later.

A silicon fluid is employed because its viscosity falls linearly, but only very little, with increase of temperature. The housing is not initially full of fluid, otherwise the expansion due to generation of heat internally as a result of work done on the fluid would burst either it or the seals. Consequently, thermal expansion of the fluid progressively increases the wetted (effective) area of the plates, thus tending to offset the effect of the simultaneous reduction in viscosity owing to the rise in temperature.

As temperature increases further, the housing eventually becomes full of fluid, at which point input of energy causes a rapid rise in temperature, and therefore pressure, locally between the plates. These instantaneous local pressure rises cause the plates to move axially and the fluid between some of them to be squeezed out through the perforations. The outcome is metal-to-metal contact and a rapid rise in transmitted torque: this is called the hump mode of operation. Then, as the speed differential drops precipitately because of the suddenly increased friction, the unit rapidly reverts to the viscous mode.

The hump mode must not, however, be regarded as normal. It is an advantage only as a self-protection effect, considerably increasing the traction at the one wheel, and thus rapidly freeing a vehicle stuck because the other wheel is spinning. Alternatively, if the traction potential of the previously effective wheel is exceeded, the load on the coupling is again relieved because it, too, will spin. If, on the other hand, neither of these reliefs comes into operation, the engine stalls and the load on the coupling is once more effectively shed.

There are two ways, of installing this device in a driveline, Fig. 26.10: one is in series, as a viscous transmission (VT), and the other in parallel, as a viscous coupling (VC). Moreover, in a differential there are, again, two ways of installing it, and these are: in the shaft-to-carrier and shaft-to-shaft layouts, Figs. 26.11 and 26.12 respectively. Possible four-wheel-drive layouts are illustrated in Fig. 26.13.

In a shaft-to-carrier layout one set of discs is splined to the differential carrier, while the other set, the alternate discs, is splined to the differential gear on one side which, in turn, is of course splined to its shaft. On the other hand, with a shaft-to-shaft arrangement the discs are connected alternately, one set to each of differential gears. With the latter

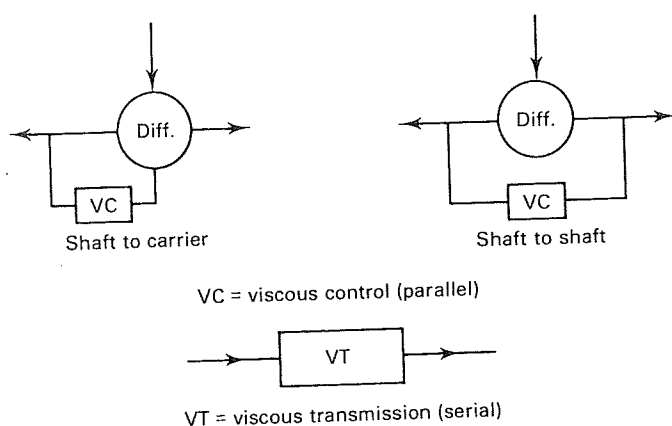


Fig. 26.10

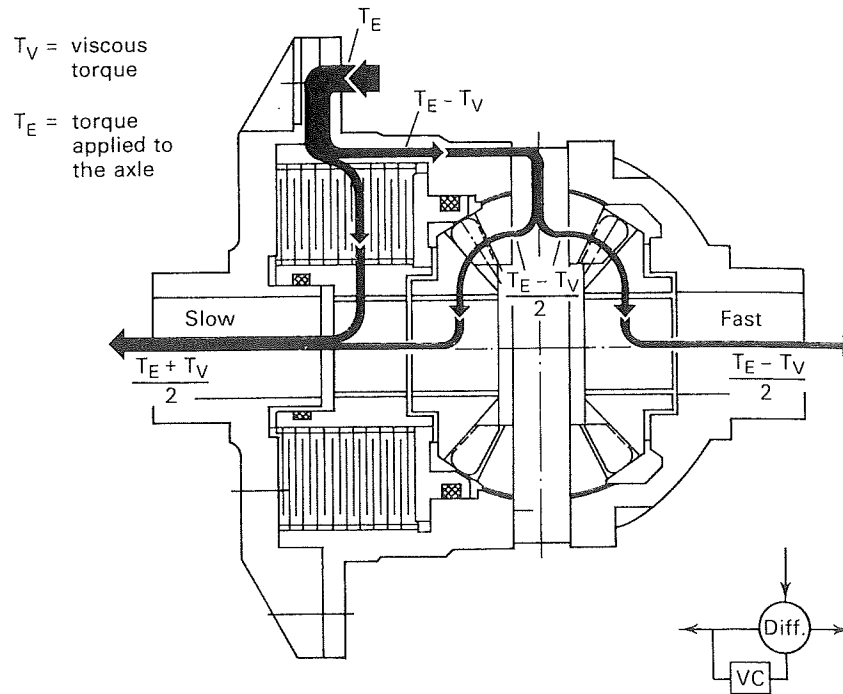


Fig. 26.11 Schematic torque distribution of a shaft-to-carrier axle differential

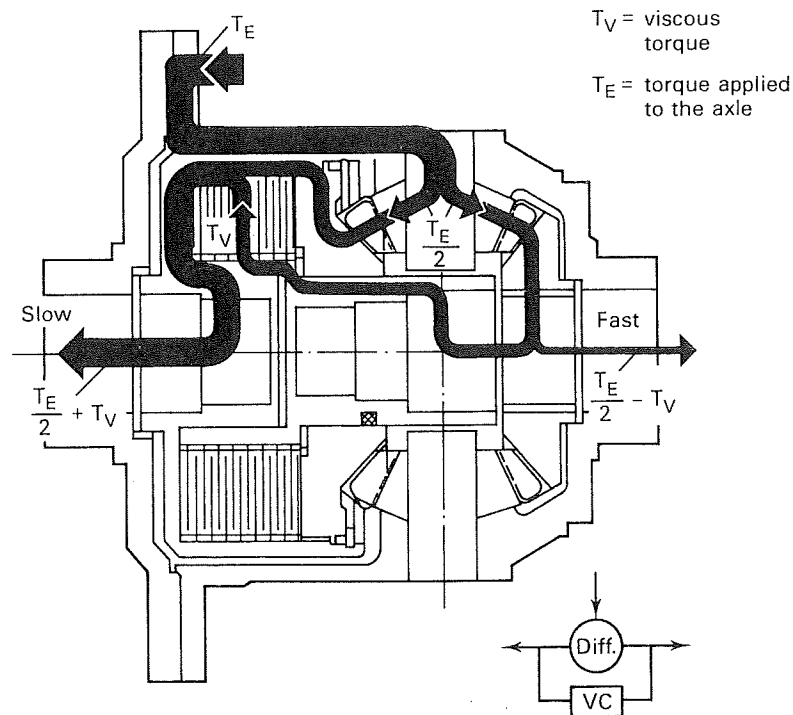


Fig. 26.12 Schematic torque distribution of a shaft-to-shaft axle differential

arrangement, although the viscous coupling is connected, in effect, in series between the ends of the two halfshafts, the differential gear is nevertheless still in parallel with it.

Either arrangement leaves the differential carrier and pinions to function normally except when there is a significant speed difference between the

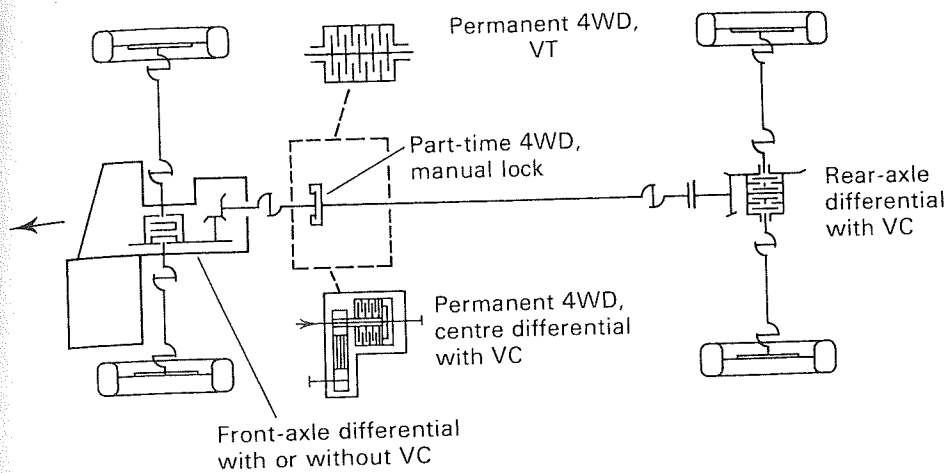


Fig. 26.13 Four-wheel-drive systems

halfshafts, in which case the viscous coupling comes automatically into operation at a limited rate of slip, thus greatly reducing the potential for relative rotation. For any given speed difference, however, the shaft-to-shaft layout has approximately three times the locking torque of the shaft to carrier design. It is therefore the preferred layout for applications in which the space available is restricted and high torques are to be transmitted.