Rotary power transmission.

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Description

This invention relates to rotary transmissions of the dual clutch kind by which is meant a transmission having gear trains each being one of a set providing a series of increasing speed ratios, and two clutches independently operable and providing alternative drive paths through the gear trains between a common input and a common output, the gear trains of alternate ratios in the set being driven respectively through one and the other clutch.

The invention is particularly though not exclusively applicable to drive transmissions for motor vehicles.

Examples of transmissions of the dual clutch kind are shown in British Patent Specification Nos. 145827 (Bramley-Moore), 585716 (Kegresse), 795280 (David Brown).

Current motor vehicle design trends indicate that an increasing proportion of vehicle production will be front wheel drive. Front wheel drive vehicles usually have the engine, clutch, gearbox and final drive mechanisms assembled as a single power unit; this can reduce material and manufacturing costs.

Many of such front wheel drive vehicles have a transverse engine installation, this can reduce overall vehicle length and so reduce manufacturing costs still further.

Small transverse engined vehicles have little space between the front wheel suspension mechanisms to site the power unit. It has been proposed to place the gearbox beneath the engine but this leads to low ground clearance and/or a high bonnet line with consequent reduced driver visibility. Although small transverse engined vehicles having gearboxes placed alongside the engine are known it is uncommon to find such vehicles fitted with a fully automatic transmission.

Current motor vehicle design trends further indicate an increased number of speed ratios being provided in the transmission for reasons of fuel economy.

It has been proposed hitherto to provide a rotary transmission having between an input and an output a set of gear trains each providing one of a series of increasing speed ratios and two independently operable clutches, sequential gear trains in the series being arranged respectively for drive through one and the other clutch, the gear trains including gears journalled on the layshafts and connectable thereto by clutch assemblies, the gear trains being arranged only for drive from an input shaft to one of two parallel layshafts, the layshafts being for continuous direct drive to the output by output means mounted on the layshafts and engaging with a front drive wheel. Such a transmission is described in Automotive Engineering Volume 90 September 1981 pages 104 to 109 and is set out in the pre-characterised part of Claim 1.

An object of the present invention is to provide an improved constant mesh rotary transmission which is suitable for small transverse engined vehicles, and which is suitable for adaption to fully automatic control.

The present invention is characterised in that a transfer gear train is additionally provided for drive between the layshafts, means being provided to disconnect one of said layshafts from continuous direct drive to the output.

Such an arrangement can provide six well spaced forward speed ratios from a very compact gear set.

The transfer gear train may include an idler wheel co-axial with said input shaft. The transfer gear train may include a gear wheel journalled on the other of said layshafts.

Preferably clutches are arranged on opposite sides of the set of gear trains.

Drive gears co-axial with the input shaft are respectively provided for each clutch driven member. Each drive gear may be one of a three wheel gear train between the layshafts. Preferably a reverse speed ratio is provided by a gear train including said idler wheel and one other gear of said transfer gear train. In such a case said idler wheel may have three gear wheel profiles thereon one of which forms part of the reverse speed gear train.

Other features of the invention are included in the following description of a preferred embodiment shown by way of example only, in the accompanying drawings in which:

Fig. 1 is an axial section through the transmission but with one of the layshafts positioned out of true position for ease of understanding.

Fig. 2 is a somewhat diagrammatic end view of the transmission and showing certain gear wheels to illustrate their true position with respect to one another; and

Fig. 3 is a table showing speed ratio (SR), respective clutch engaged (CE) and respective synchronisers engaged (SE).

For ease of understanding the transmission is depicted in Fig. 1 as having the three main shafts all lying in the same plane. As will be apparent from the following text and as shown in Fig. 2 one of the shafts may advantageously be out of the plane of the remaining two.

The torque input to the transmission may be direct from an engine or may be through a fluid coupling, for example a hydrodynamic torque converter. The coupling may include a lock up clutch engageable to directly connect its input and output.

The transmission is intended for fully automatic operation as is described hereinafter.

With reference to Figs. 1 and 2 there is shown an input shaft 11 from an engine and connectable through either a clutch 12 or clutch 13 and trains of gear wheels with output shafts 14 and 15 having respective output gear wheels 16 and 17.

The gear wheels 16 and 17 are in mesh with a gear wheel 18 of an output shaft 19 which may part of a differential gear assembly for transmission of driving torque to vehicle wheels. The wheel 16 is journalled on the shaft 14 whilst the wheel 17 is fast for rotation with its shaft 15.
Clutch 12 is preferably a conventional single dry plate clutch having the usual flywheel 21, driven plate 22, pressure plate 23, clutch cover 24 and spring 25. A release bearing 26 is operable to relieve the spring load and thus release the driven plate 22.

A drive shaft 27, fast for rotation with the flywheel 21, extends through the set of gear trains to the opposite side of the transmissions to drive the other clutch.

Clutch 13 is a wet plate clutch having a drive member 31, pressure plate 32 and driven plate 33. A piston 34, housed in the drive member 31 is supplied with fluid under pressure via a duct 35 in a transmission end cover 36.

A pressure relief valve 37, whose function is explained subsequently, is housed in the piston 34. For each clutch driven plate 22, 33 an input gear wheel respectively 41 and 42 is journaled on the drive shaft 27. Each gear wheel 41, 42 drives two gear wheels respectively 43, 44 and 45, 46 journaled on the output shafts 14, 15.

A transfer gear train between the output shafts comprises a gear wheel 47 journaled on shaft 15, an idler wheel 48 journaled on the drive shaft 27 intermediate the input gears 41, 42, and a wheel 49 fast for rotation with the shaft 14.

The idler wheel 48 has a respective gear profile for engagement with each of the wheels 47, 49 in order to provide an even spread of transmission ratios.

A synchroniser and dog clutch assembly S1 is provided to connect output wheel 16 for rotation with shaft 14. Further assemblies S2—S6 are operable to connect the gear wheels 43—47 with the shafts on which they are journaled. Such synchroniser assemblies are of a type found in manual change transmissions and are well understood in the rotary change speed transmission art.

A reverse speed ratio is provided between a gear profile of the idler gear wheel 48, a disengageable idler wheel 51 and a gear profile 52 formed on the double synchroniser assembly S2, S4.

A drive gear wheel 53, for a hydraulic pump 54, is fast for rotation with the drive shaft 27 adjacent the idler wheel 48. The pump provides fluid under pressure for transmission operation, for example engagement of the clutch 13.

A parklock wheel 55 is splined to the output shaft 15 and is engageable with a pawl (not shown) to provide a mechanical lock for the transmission. Such a lock is a usual provision for vehicles fitted with fully automatic transmissions.

Operation of the transmission, with additional reference to Fig. 3 is as follows:-

Both clutches 12, 13 and synchroniser assemblies S1—S6 are disengaged.

To engage first speed ratio synchroniser assemblies S5 and S2 are engaged. Clutch 12 is gradually engaged to transmit drive from the input shaft 11 through gear wheels 41, 44, 49, 48, 47 to output shaft 15 and hence through gear 17 to the differential gear 18.

Gear wheel 16 is driven by differential gear 18 at all times but is free to turn relative to output shaft 14 by virtue of being journaled thereon.

Second speed may be preselected by engagement of synchroniser assembly S6, thus driven plate 33 of clutch 13 is driven idly by output shaft 14 through gear train 46, 42.

To effect a speed ratio change from 1st to 2nd clutch 13 is engaged simultaneously with disengagement of clutch 12, drive is now through the gear train 42, 46, 49, 48, 47 with the driven plate 22 of clutch 12 driven idly through train 44, 41.

Synchroniser assembly S5 may remain engaged in anticipation of a ratio change back into 1st speed or may be disengaged in preparation for a change to third ratio.

Third speed ratio may be preselected by engagement of synchroniser assembly S3 to idly drive driven plate 22 through gear train 43, 41.

A speed ratio change from 2nd to 3rd is effected in the manner of the previous ratio change, by simultaneously disengaging clutch 13 and engaging clutch 12, synchroniser assemblies S6 and S2 then being disengaged in preparation for a further upchange if desired.

To preselect 4th speed ratio synchroniser assembly S4 is engaged, the ratio change being effected by change of drive from clutch 12 to clutch 13, synchroniser assembly S3 being subsequently disengaged.

For 5th speed ratio synchronisers S1 and S5 are engaged to idly drive driven plate 22 from output gear 16. Drive is transferred from clutch 13 to clutch 12 to effect the ratio change.

The ratio change to 6th speed is effected as previously described after preselection engagement of synchroniser assembly S6.

Downchanges through the transmission are effected in the same manner by preselection of the next sequential gear train in the set followed by change of drive from the hitherto engaged clutch to the hitherto disengaged clutch.

The relief valve 37 is to prevent a build up of fluid pressure behind the piston 34 due to centrifugal forces. The dimensions of the ball and valve bore are carefully chosen to ensure that supply of pressure through duct 35 will force the ball into its seat to close the valve. Release of clutch engaging pressure allows the ball to roll up its seat under centrifugal load to open the valve.

In an alternative arrangement to that shown in Fig. 1 gear wheel 16 is fast for rotation with output shaft 14 whilst synchroniser assemblies S5 and S6 and gear wheels 44 and 49 are mounted on an annular shaft journaled on the output shaft 14. Synchroniser assembly S1 then connects output shaft 14 to the annular shaft. Operation of the transmission is unchanged but the alternative arrangement may be more suitable under different installation conditions.

Claims

1. A rotary transmission having between an input (11) and an output (19) a set of gear trains
für einen kontinuierlichen direkten Antrieb zu dem Ausgang durch eine Ausgabeanrichtung (16, 17) vorgesehen sind, die auf den Vorgelegewellen befestigt und mit einem Vorderantriebsrad (18) in Eingriff stehen, dadurch gekennzeichnet, daß ein Verteilergetriebe (47, 48, 49) zusätzlich für den Antrieb zwischen den Vorgelegewellen (14, 15) vorgesehen ist, wobei eine Einrichtung (51) vorgesehen ist, um eine der Vorgelegewellen (14) von dem kontinuierlichen direkten Antrieb zu dem Ausgang zu trennen.

2. Drehgetriebe nach Anspruch 1, dadurch gekennzeichnet, daß das Verteilergetriebe ein Zwillenrad (48) aufweist, das koaxial zu der Eingangsseite (11) ist.

3. Drehgetriebe nach Anspruch 2, dadurch gekennzeichnet, daß das Verteilergetriebe ein Zahnrad (47) aufweist, das auf der anderen der Vorgelegewellen dreht oder gelagert ist.


5. Drehgetriebe nach Anspruch 4, dadurch gekennzeichnet, daß Antriebszahnradkörper (41, 42), die koaxial zu der Eingangsseite sind, für jedes angeordnete Kupplungsteil jeweils vorgesehen sind.


Revidenctions

1. Transmission rotative présentant entre une entrée (11) et une sortie (19) un jeu de trains d'engrenages dont chacun fournit l'un d'une série de rapports de transmission croissants, et deux embrayages manœuvrables de manière indépendante (12, 13), des trains d'engrenage successifs de la série (41, 43, 44; 42, 45, 46) étant disposés de manière à être respectivement entraînés par l'un et par l'autre embrayage, ces trains d'engrenage comprenant des engrenages (43, 44, 45, 46, 47) qui sont montés à palier sur des arbres intermédiaires et peuvent être reliés à ceux-ci par des ensembles d'embrayage (51—56), ces trains d'engrenage étant disposés de façon à ne permettre qu'un entraînement sur l'un de deux arbres intermédiaires parallèles (14, 15) à partir d'un arbre d'entrée (11), ces arbres intermédiaires étant prévus pour entraîner directement et de manière continue la sortie à l'aide de moyens de sortie (16, 17) qui sont montés sur ces arbres intermédiaires et qui coopèrent avec une roue menante frontale (18), caractérisée en ce qu'il est...
en outre prévu un train d'engrenages de transfert (47, 48, 49) permettant un entraînement entre les arbres intermédiaires (14, 15), des moyens (51) étant prévus pour déconnecter l'un desdits arbres intermédiaires (14), de son entraînement direct continu sur la sortie.

2. Transmission selon la revendication 1, caractérisée en ce que le train d'engrenages de transfert comprend un pignon intermédiaire (48) coaxial audit arbre d'entrée (11).

3. Transmission selon la revendication 2, caractérisée en ce que le train d'engrenages de transfert comprend un pignon d'engrenage (47) monté à palier sur l'autre desdits arbres intermédiaires.

4. Transmission selon la revendication 3, caractérisée en ce que les embrayages (12, 13) sont disposés sur des côtés opposés du jeu de trains d'engrenages.

5. Transmission selon la revendication 4, caractérisée en ce qu'il est prévu pour chaque disque d'embrayage particulier des pignons menants (41, 42) coaxiaux à l'arbre d'entrée.

6. Transmission selon la revendication 5, caractérisée en ce que chaque pignon menant est un pignon d'un train d'engrenages à trois pignons disposé entre les arbres intermédiaires (14, 15).

7. Transmission selon la revendication 6, caractérisée en ce qu'un rapport de transmission de marche arrière est fourni par un train d'engrenages constitué dudit pignon intermédiaire (48) et d'un autre pignon dudit train d'engrenages de transfert.

8. Transmission selon la revendication 7, caractérisée en ce que ledit pignon intermédiaire (48) peut présenter sur lui trois profils d'engrenage dont l'un fait partie du train d'engrenages de marche arrière.