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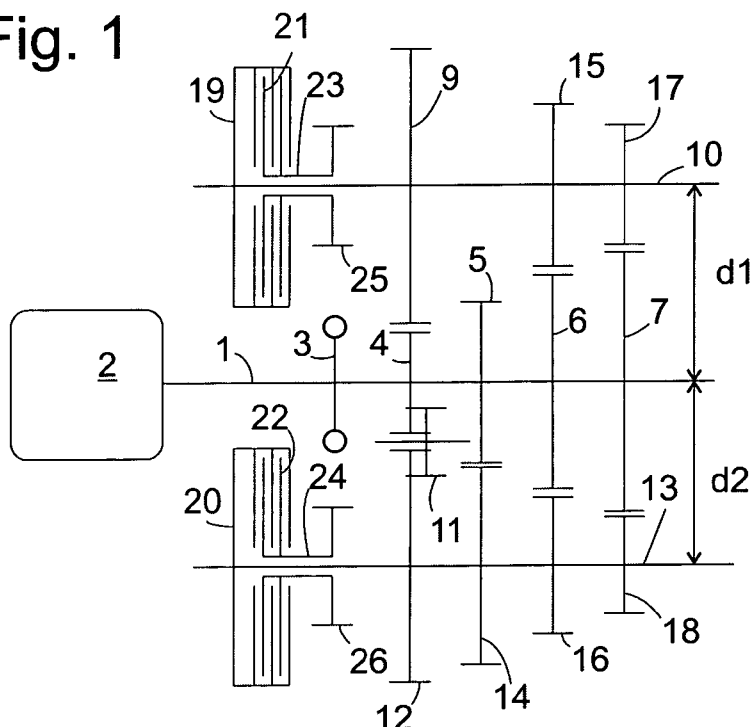
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(54) Title: DOUBLE CLUTCH TRANSMISSION

Fig. 1



(57) Abstract: A double-clutch transmission comprises an input shaft (1), an output shaft and at least two alternative paths for transmitting torque from the input shaft (1) to the output shaft. Each of the alternative paths comprises a layshaft (10; 13) and a friction clutch (19, 21; 20, 22) for selectively enabling or disabling torque flow along said path located on that layshaft (10; 13).



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Double Clutch Transmission

D e s c r i p t i o n

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Double clutch transmissions have been the subject of considerable public interest in recent times due to the capability to shift gears without interrupting torque flow. This allows for very smooth shifting operations which are hardly perceptible for the passengers of a motor vehicle.

Generally, conventional double clutch transmissions comprise an input shaft formed of at least two parts, a solid shaft member and a hollow shaft member extending along part of the solid shaft member and being rotatable with respect to the latter, both shaft members carrying a plurality of drive gearwheels that mesh with driven gearwheels on one or more layshafts, and each shaft member has a friction clutch associated to it for selectively enabling or disabling torque transmission from an engine to the shaft member.

In such a conventional double clutch transmission, the two friction clutches are arranged one behind the other in the axial direction of the input shaft, thus contributing significantly to the overall length of the transmission. In another prior art design, clutches are coaxially nested. Since in this design dimensions of the clutches are different, their load-bearing capacities tend to be different, too.

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The object of the present invention is to provide a double clutch transmission which is suitable for a particularly compact design.

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The object is achieved by a double clutch transmission comprising an input shaft, an output shaft and at least two alternative paths for transmitting torque from the input shaft to the output shaft, each of
10 the alternative paths comprising a friction clutch for selectively enabling or disabling torque flow along said path, in which each path comprises a layshaft and the friction clutch associated to a given path is located on the layshaft of said path. In such a transmission,
15 instead of placing the two clutches one behind the other in the axial direction or nesting them radially, as conventional, they can be placed one beside the other, so that the overall length of the transmission can be reduced. If the diameter of the two clutches is not
20 larger than that of the largest gearwheel on their respective layshaft, the lateral dimensions of a transmission are not increased by placing the friction clutches on the layshafts.

25 In most practical cases, there will be two or more driven gearwheels on each layshaft, meshing with drive gearwheels of the input shaft. In order for the friction clutch to be able to interrupt torque flow via its associated layshaft, no matter which gearwheels are
30 currently under load, a friction clutch of the given path should be arranged to enable or disable torque flow from the layshaft associated to that path to the output shaft.

To this effect, each friction clutch preferably
35 has a first member non rotatably mounted on its associated layshaft, and a sleeve carried by said

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layshaft has a second member of said friction clutch and an output pinion non-rotatably mounted on it.

The output pinions of the two layshafts
5 preferably mesh with a same gear element on the output shaft. The output shaft can be front or rear drive shaft or a propeller shaft, so that the gearbox of the invention can be used in front wheel drive, rear wheel drive or all wheel drive vehicles. The said gear element
10 can be a differential. In an all-wheel drive vehicle, the gear element preferably is a front-rear torque distribution unit. For rear wheel driven applications the output shaft can be arranged coaxially to the crankshaft as well with a given off-set.

15

In order to facilitate access to the friction clutches for maintenance or repair purposes, it is useful if the friction clutches are located at a side of the transmission, or, more precisely, on each layshaft the
20 output pinion is placed between the friction clutch and at least one of the driven gearwheels.

Preferably, the first path comprises odd-numbered forward gears, and the second path comprises even-
25 numbered forward gears. In this way, shifting between adjacent gears is possible by opening one of the friction clutches and simultaneously closing the other, so that the flow of torque is not interrupted during shifting.

30 A reverse gear is preferably incorporated into the second path. In this way, shifting between first forward and reverse gears becomes possible by simply opening one friction clutch and closing the other.

35 Since the amount of the transmission ratios of first forward and reverse gears usually is rather

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similar, it is convenient that a forward gear and the reverse gear should share a drive gearwheel on the input shaft.

5 For instance, one drive gearwheel may mesh directly with two other gearwheels, namely a forward driven gearwheel on one of the output shafts, preferably the driven gearwheel of the first gear, and a reversing gearwheel on an auxiliary shaft, which in turn meshes
10 with the driven gearwheel of the reverse gear on the other output shaft.

 Alternatively, there can be a driven gearwheel on one output shaft which meshes with an associated drive
15 gearwheel on the input shaft and with the reverse driven gearwheel on the other output shaft.

 Since the torque transmitted in the first forward gear can be higher than in higher forward gears, the
20 drive gearwheel shared by first forward and reverse gears is best placed closer to a torque receiving end of the input shaft than all other drive gearwheels.

 Similarly, the driven gearwheels on the two
25 layshafts associated to first forward and reverse gears are preferably closer to the friction clutches than any other driven gearwheels.

 In order to achieve a convenient spacing between
30 the gear ratios of the various forward gears, it is convenient for the second forward gear to have a drive gearwheel on the input shaft exclusively dedicated to it, in particular, if the total number of forward gears is even.

35 In order to achieve flexible gear ratios of the different gears, the two pinions on the layshaft may have

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different diameters. Also the distances between the input shaft and the two layshafts may be different.

5 A torsion damper may be provided in the input shaft in order to prevent clattering of meshing gearwheels due to speed variations of an engine driving the input shaft.

10 For preventing unwanted reverse movement of the vehicle, in particular when driving uphill, a shiftable one-way clutch may be associated to at least one of the layshafts.

15 The one-way clutch preferably has blocking elements which are rotatable around the layshaft and have an unblocking position radially outward of a blocking position, so that when the blocking elements rotate, they are urged towards the unblocking position by centrifugal force. In this way, the one-way clutch is automatically
20 disengaged if the vehicle speed is above a predetermined limit. Alternatively, other one-way clutch concepts without blocker bodies such as other friction based concepts, form- or force fit units may be used.

25 In a vehicle comprising the double clutch transmission as described above, the power train may be simplified by providing a rigid connection between a motor and the input shaft of the double clutch transmission. In such an arrangement the reflected
30 inertia of the driven gears to the input shaft shall be evenly distributed to reduce clatter sensitivity.

Further features and advantages of the invention will become apparent from the subsequent description of
35 embodiments thereof referring to the appended drawings.

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Figs. 1 to 5 are schematic diagrams of double clutch transmissions according to first to fifth embodiments of the invention.

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The transmission shown in Fig. 1 has an input shaft 1 which is permanently connected to the crankshaft of a motor 2. The input shaft 1 carries a torsion damper, e.g. a double-mass flywheel 3. Drive gearwheels 4 to 7
10 are splined to input shaft 1. The diameter of the drive gearwheels increases with their distance from motor 2. The smallest drive gearwheel 4, closest to motor 2, meshes with a driven gearwheel 9 on a layshaft 10, and with a reversing gearwheel 11, which, in turn, meshes
15 with a driven gearwheel 12 on a second layshaft 13.

Second smallest drive gearwheel 5 meshes only with a driven gearwheel 14 on second layshaft 13. The two largest drive gearwheels 6, 7 mesh with driven gearwheels
20 15, 16 and 17, 18, respectively, on first and second layshafts 10, 13.

The transmission comprises two friction clutches, one on each layshaft 10, 13. The friction clutches may be
25 of the wet or the dry type, as load requires and available package size permits. Each friction clutch has an upstream member 19, 20 comprising a plurality of discs splined to its respective layshaft 10, 13, and a downstream member 21, 22 formed of a plurality of discs
30 connected to a sleeve 23, 24 rotatably mounted on shaft 10, 13. In a space between each friction clutch and a facing driven gearwheel 9, 12, an output pinion 25, 26 is mounted on sleeve 23, 24. The two output pinions 25, 26 mesh with a same differential, not shown.

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A dog clutch, not shown, is associated to each drive gearwheel 4, 5, 6 and 7, in order to selectively lock one of them at a time to input shaft 1.

5 The transmission of Fig. 1 has six forward gears and a reverse gear. Since the distance d_1 between input shaft 1 and layshaft 10 is greater than the distance d_2 between input shaft 1 and layshaft 13, among the two gears formed e.g. by gearwheels 7, 17, 18, the one formed
10 by gearwheels 7 and 18 is the higher one. Given the size relations of the gearwheels shown in Fig. 1, it is readily apparent that gearwheels 7, 18 form the sixth gear, gearwheels 7, 17 form the fifth gear, gearwheels 6, 16 form the fourth gear, and gearwheels 6, 15 form the
15 third gear. Among the gear ratios of these four gears, three can be set arbitrarily by choosing the distances d_1 , d_2 between the shafts 1, 10, 13 and the diameters of e.g. gearwheels 6, 7. The diameters of gearwheels 15-18 are determined by these choices, so that a fourth gear
20 ratio is predetermined by the geometrical constraints. If first and second gear shared a drive gearwheel, it would not be possible to choose their gear ratios independently from one another. By instead using dedicated drive
25 gearwheel 5 only for the second gear, the transmission ratio of this gear can be set arbitrarily. Since d_2 is smaller than d_1 , gearwheel 12 must be smaller than gearwheel 9, and the amount of the gear ratio of the reverse gear can only be smaller than that of the first gear.

30

Geometrical constraints concerning the amount of the transmission ratio of the reverse gear can be relieved by providing not a single reversing gearwheel 11 as shown in Fig. 1, but stepped reversing gearwheels 27,
35 28, as shown in Fig. 2. By having the larger one, 27, of the two reversing gearwheels 27 28 mesh with drive

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gearwheel 4 and the smaller one, 28, with driven gearwheel 9, the amount of the transmission ratio of the reverse gear can be increased beyond what the geometrical constraints would allow in the embodiment of Fig. 1.

5

Fig. 3 is a diagram of a seven-gear transmission according to the invention. Components common to this embodiment and the previous ones have identical reference numerals and are not described again. An additional
10 driven gearwheel 29 is provided on layshaft 10, meshing with the second smallest drive gearwheel 5. Here the distance d_2 between layshaft 13 and input shaft 1 is larger than the distance d_1 between layshaft 10 and input shaft 1. Therefore, gearwheel pairs 5 + 27, 6 + 15 and 7
15 + 17 give third, fifth and seventh gears, respectively, whereas gearwheel pairs 5 + 14, 6 + 16 and 7 + 18 combine to form gears 2, 4 and 6, as in the previous embodiments. Due to the increased distance d_2 between shafts 1 and 13, gearwheel 12 need not be smaller than gearwheel 9, so
20 that the amount of the transmission ratio of the reverse gear is not necessarily smaller than that of the first gear if a single reversing gearwheel 11 meshing with gearwheels 4, 12 is used.

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Fig. 4 illustrates a six-gear transmission which is similar to that of Fig. 1 except for the presence of a one-way clutch 30 in sleeve 24. In a way familiar to the man of the art, one way clutch 30 has concentric inner and outer races and a plurality of blocking elements 31
30 in a space between the two races, which are movably connected to one of the races to engage steps formed at the other race, whereby a rotation of the two races with respect to each other in a prohibited direction is blocked. In the one way clutch 30 of Fig. 4, the outer
35 race has the blocking elements 30 connected to it and rotates with sleeve 24, whereas the inner race is solidly

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connected to a casing of the transmission. While a forward gear is engaged in the transmission, the blocking elements 30 are freely movable, so that they come to engage the steps of the inner race if the differential rotates in a reverse direction, preventing the vehicle from rolling backwards if no gear is engaged. The blocking elements 31 are connected to a gearshift lever, not shown, in order to be blocked in a position adjacent to the outer race if the reverse gear is selected, in order to prevent them from blocking in that case. If the vehicle is rolling forward above a predetermined speed, the blocking elements 31 rotate with sleeve 24 and are urged into that same position adjacent to the outer race by centrifugal force. Thus there is no contact between them and the inner race, and, hence, no wear, when the vehicle is being driven at a speed of more than a few kilometres per hour.

The transmission of Fig. 5 differs from that of Fig. 1 by the fact that there is no reversing gearwheel 11 on an auxiliary shaft, so that drive gearwheel 4 meshes only with driven gearwheel 9 on layshaft 10. Gearwheel 9, in turn, meshes with gearwheel 12 on layshaft 13, causing the latter to run in the reverse direction.

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L i s t o f r e f e r e n c e s i g n s

	1	input shaft
	2	motor
5	3	double-mass flywheel
	4	drive gearwheel
	5	drive gearwheel
	6	drive gearwheel
	7	drive gearwheel
10	8	
	9	driven gearwheel
	10	layshaft
	11	reversing gearwheel
	12	driven gearwheel
15	13	second layshaft
	14	driven gearwheel
	15	driven gearwheel
	16	driven gearwheel
	17	driven gearwheel
20	18	driven gearwheel
	19	friction clutch/upstream member
	20	friction clutch/upstream member
	21	friction clutch/downstream member
	22	friction clutch/downstream member
25	23	sleeve
	24	sleeve
	25	output pinion
	26	output pinion
	27	driven gearwheel
30	28	reversing gearwheel
	29	reversing gearwheel
	30	one-way clutch
	31	blocking elements

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C l a i m s

1. A double-clutch transmission comprising an input shaft (1), an output shaft and at least two
5 alternative paths for transmitting torque from the input shaft (1) to the output shaft, each of the alternative paths comprising a friction clutch (19, 21; 20, 22) for selectively enabling or disabling torque flow along said path, characterized in that
10 each path comprises a layshaft (10; 13) and that the friction clutch (19, 21; 20, 22) associated to a given path is located on the layshaft (10; 13) of said path.
- 15 2. The double-clutch transmission of claim 1, wherein the friction clutch (19, 21; 20, 22) of a given path is arranged to enable or disable torque flow from the layshaft (10; 13) associated to that path to the
20 output shaft.
3. The double-clutch transmission of claim 2, wherein each friction clutch (19, 21; 20, 22) has a first member (19; 20) non-rotatably mounted on its associated layshaft (10; 13) and that a sleeve (23; 24) carried by said layshaft (10; 13) has a second
25 member (20; 22) of said friction clutch and an output pinion (25; 26) non-rotatably mounted on it.
4. The double-clutch transmission of claim 3, wherein
30 the output pinions (25; 26) of said two layshafts (10; 13) mesh with a differential of the output shaft.
5. The double-clutch transmission of claim 3 or 4,
35 wherein on each layshaft (10; 13) the output pinion (25; 26) is placed between the friction clutch (19,

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21; 20, 22) and at least one gearwheel (9; 12; 14-18) which meshes with a gearwheel (4-7) of the input shaft (1).

5 6. The double-clutch transmission of any of the preceding claims, wherein the first path comprises odd-numbered forward gear ratios, and the second path comprises even-numbered gear ratios.

10 7. The double-clutch transmission of claim 6, wherein the second path further comprises a reverse gear.

8. The double-clutch transmission of claim 6 or 7, wherein first forward gear and reverse gear share a
15 drive gearwheel (4) on the input shaft (1).

9. The double-clutch transmission of claim 8, wherein among all drive gearwheels (4-7) of the input shaft (1), the drive gearwheel (4) shared by first forward
20 and reverse gears is closest to a torque-receiving end of the input shaft (1).

10. The double-clutch transmission of claim 8 or 9, wherein driven gearwheels (9; 12) on said two
25 layshafts (10; 13) associated to said first forward and reverse gears are closer to the friction clutches (19, 21; 20, 22) than any other driven gearwheels.

30 11. The double-clutch transmission of any of claims 6 to 10, wherein the total number of forward gears is even, and the second forward gear has a drive gearwheel (5) on the input shaft (1) exclusively dedicated to it.

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12. The double-clutch transmission of any of the preceding claims, wherein a torsion damper (3) is provided in the input shaft (1).
- 5 13. The double-clutch transmission of any of the preceding claims, wherein at least one of the layshafts (13) has a one-way clutch (30) associated to it.
- 10 14. The double-clutch transmission of claim 13, wherein blocking elements (31) of the one-way clutch (30) are rotatable around the layshaft (13), and an unblocking position of the blocking elements (31) is radially outward of a blocking position.
- 15 15. A motor vehicle in which a motor (2) is permanently connected to the input shaft (1) of a double-clutch transmission according to any of the preceding claims.

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Fig. 1

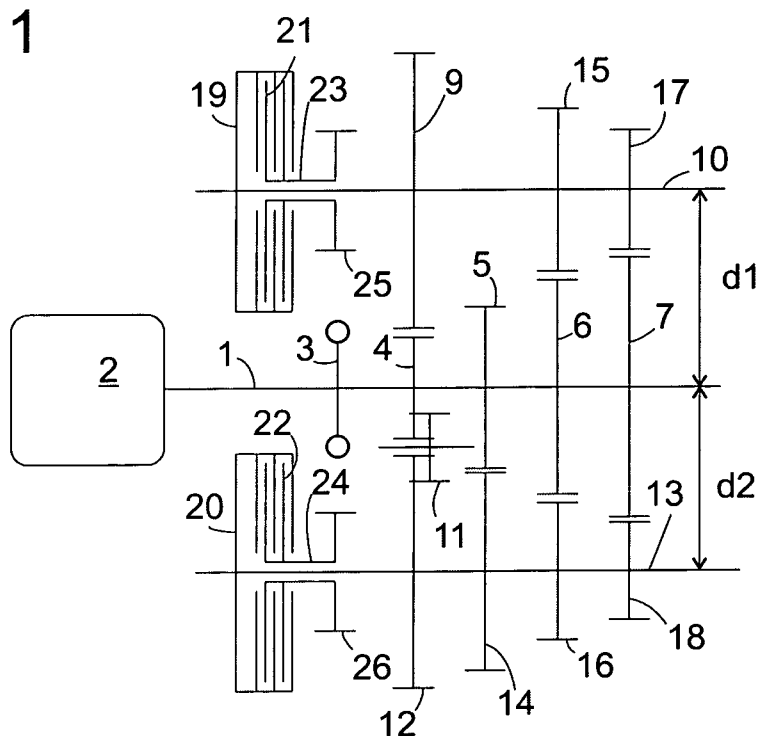
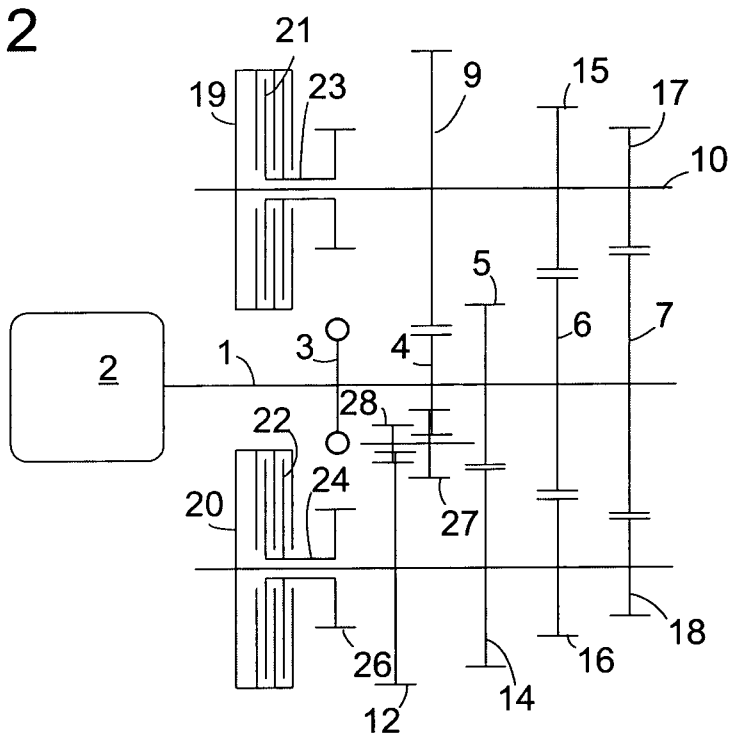


Fig. 2



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Fig. 3

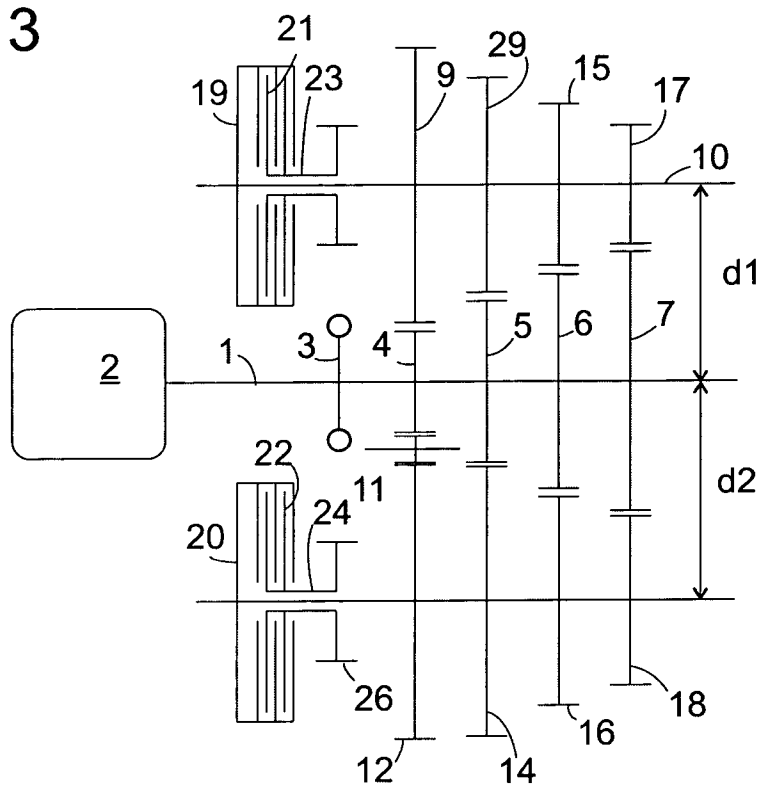
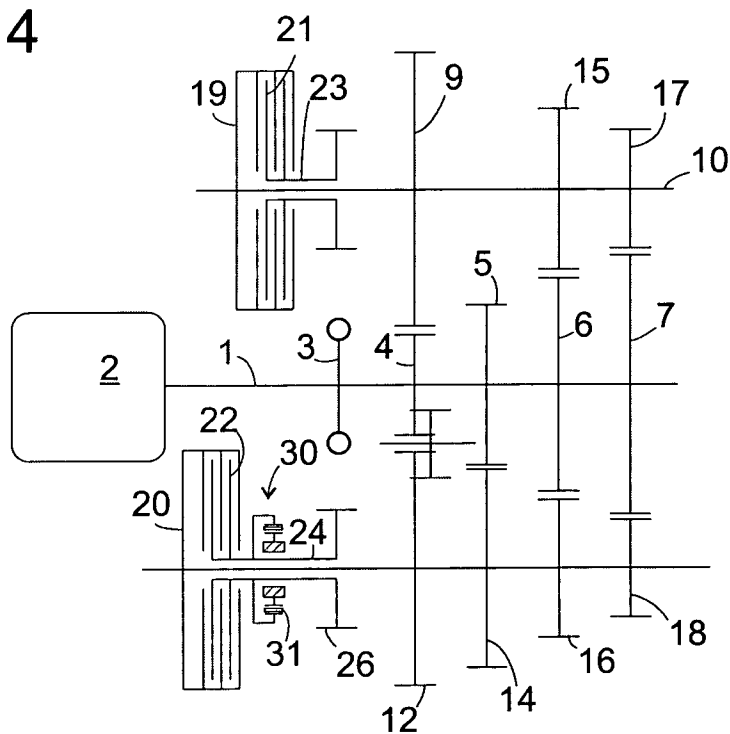
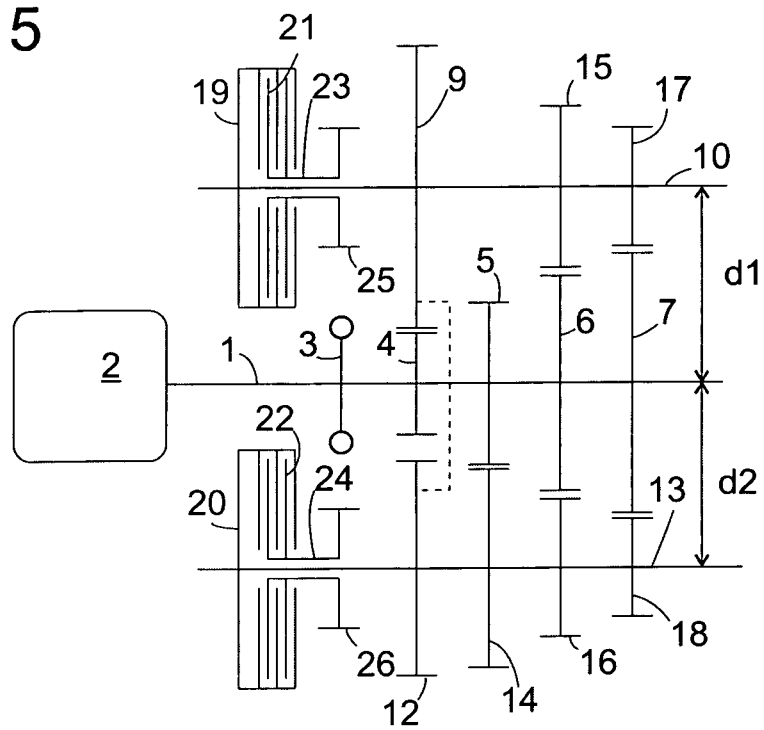


Fig. 4



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Fig. 5



INTERNATIONAL SEARCH REPORT

International application No

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A. CLASSIFICATION OF SUBJECT MATTER
 INV. F16H3/093 F16H3/00
 ADD. F16H3/10

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
 F16H B60K

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

EPO-Internal

C. DOCUMENTS CONSIDERED TO BE RELEVANT

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Y	page 6 - page 7; figures 4,5 page 4, lines 11-13	10-14
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A	pages 1-4; figures 3-5	1

☐ Further documents are listed in the continuation of Box C.

☒ See patent family annex.

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INTERNATIONAL SEARCH REPORT

Information on patent family members

International application No

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