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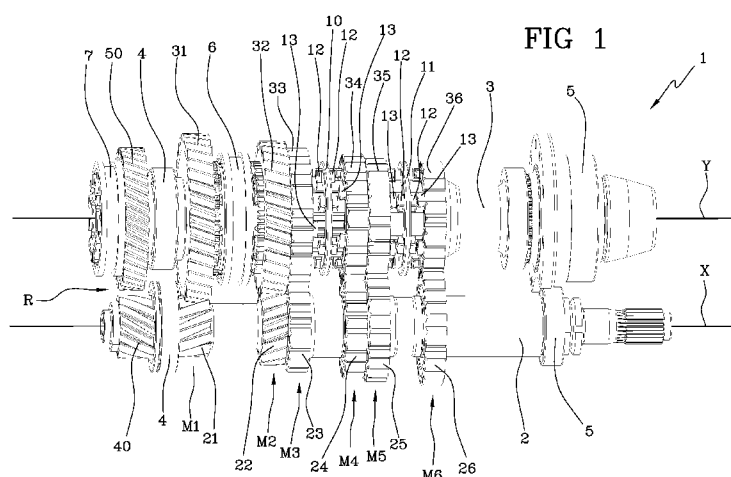
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(54) Title: A GEARBOX FOR MOTOR VEHICLES



(57) Abstract: A gearbox for a high performance road car comprises a primary shaft (2) with a set of first toothed profiles (21, 22, 23, 24, 25, 26); a secondary shaft (3) with a set of second toothed profiles (31, 32, 33, 34, 35, 36) simultaneously in constant mesh with the first toothed profiles (21, 22, 23, 24, 25, 26), the two sets combining to form distinct gear pairs (M1, M2, M3, M4, M5, M6) with different transmission ratios, ranging from a low ratio or first gear (M1) to a high ratio; a first synchronizer (6) operating on at least a first pair (M1) of the toothed profiles (21, 31) in such a way as to couple the first and second shafts (2, 3) in rotation by means of the first pair (M1) of toothed profiles (21, 31) and select the low ratio; and at least one collar (10, 11) with a dog type coupling and no synchronizer, operating on one or more of the remaining gear pairs (M3, M4, M5, M6) providing the higher ratios.

Description

A gearbox for motor vehicles

Technical Field

The present invention relates to a gearbox for motor vehicles, in particular, motor vehicles capable of high performance and preferably for road use.

Background Art

The prior art embraces mechanical transmissions fitted to motor vehicles
5 for road use, in which the gearbox is operated by means of a manual shift lever or, alternatively, by means of an automated manual shift mechanism.

In the case of the manual shift lever, this is operated directly by the driver, and connected to the gears by way of conventional rods and selectors forming part of an H-pattern or H-gate shift.

10 In the case of an automated manual transmission, or AMT, the gears are selected and engaged by means of one or more hydraulic or electric actuators linked to an input device operated by the driver.

In both of the solutions mentioned above, the gears in the gearbox are engaged with the aid of synchromesh devices serving to facilitate the
15 selection of the ratios, enabling a smooth and noiseless shift for optimum driving comfort.

Unfortunately, the speed of engagement obtainable with synchromesh devices (including those of the latest generation) is dependent on the time needed for energy to be dissipated, and for the revolutions of the primary
20 and secondary shafts to be equalized so that the gears are able to mesh without difficulty. Because of this conditioning factor, the fastest possible speed of engagement, approximately 160 ms, cannot be made quicker. In

practice, an attempt to engage the gears forcibly in a faster time can lead to failure of the synchromesh devices.

Accordingly, gearboxes designed with the type of structure outlined above are not able to deliver the performance expected of a truly high performance vehicle, even when optimized to the limit.

The prior art also embraces high performance vehicles with features of construction designed to favour high power transmission, especially where the gearbox is concerned.

The gearbox of a high performance motor vehicle, such as a racing or other competition car, is equipped with spur gears having frontally engaging dog type couplers with no synchromesh, resulting in a mechanical transmission characterized by fast response and great strength, that is to say swift gear changes, and the capacity to transmit high levels of power and torque. The omission of synchromesh components stems from the need to reduce gearshift times to the bare minimum, otherwise the lag would be excessive and detract from the performance of the vehicle.

This type of gearbox is designed especially for use in racing vehicles.

It has also been observed, however, that when fitted to high performance road cars, gearboxes of the type in question can give rise to annoying drawbacks.

These drawbacks are associated with an excessive roughness in operation of the transmission, especially when shifting the gears. In effect, the engagement of the dog clutches, unassisted by synchromesh, produces an extremely coarse and aggressive gear change that significantly reduces the driving comfort of the vehicle, rendering it unsuitable for road use.

Disclosure of the Invention

Accordingly, the object of the present invention is to provide a gearbox for motor vehicles in which the above noted drawbacks associated with the prior art can be overcome.

In particular, the object of the present invention is to design a gearbox for motor vehicles capable of delivering high performance (comparable to that of racing vehicles) while at the same time providing a level of driving comfort acceptable for road use.

- 5 The stated objects are substantially realized in a gearbox for motor vehicles according to the present invention, of which the characteristics are as recited in one or more of the appended claims.

Brief Description of the Drawings

The invention will now be described in detail, by way of example, with the aid of the accompanying drawings, in which:

- 10 - figure 1 shows a portion of gearbox according to the present invention, illustrated in perspective and with certain parts omitted better to reveal others;
- figure 2 shows the portion of the gearbox illustrated in figure 1, illustrated in section;
- 15 - figure 3 is a simplified diagrammatic view of a control unit used in the gearbox of figure 1;
- figure 4 is a perspective view illustrating a component of a control unit embodied differently to that of figure 3, used in the gearbox of figure 1;
- figure 5 is a diagrammatic illustration showing part of the lateral surface
- 20 presented by the component of figure 4, developed on a flat plane.

Detailed description of the preferred embodiments

With reference to the accompanying drawings, numeral 1 denotes a gearbox, in its entirety, according to the present invention.

- The gearbox 1 comprises a primary shaft 2 and a secondary shaft 3, connectable respectively to the crankshaft of an internal combustion engine
- 25 (by way of a disc clutch), and to a final drive, typically the drive axle of a motor vehicle (by way of a differential). The two shafts 2 and 3 are rotatable

respectively about mutually parallel axes of rotation X and Y.

The primary shaft 2 presents a plurality of first toothed profiles 21, 22, 23, 24, 25 and 26, and the secondary shaft 3 presents a plurality of second toothed profiles 31, 32, 33, 34, 35 and 36. The two pluralities of toothed profiles are matched and rotate in constant mesh, creating different speed ratios between the primary shaft 2 and the secondary shaft 3. In particular, the two pluralities of toothed profiles combine to establish a sequence of pairs of toothed profiles denoted M1, M2, M3, M4, M5 and M6 (one profile associated with the primary shaft 2 and the other profile with the secondary shaft 3), each pair having a specific ratio between the two respective pitch diameters that determines a specific transmission ratio between the two shafts 2 and 3 different from the transmission ratios determined by all the other pairs. Each pair of toothed profiles M1, M2, M3, M4, M5 and M6 thus corresponds to a specific gearbox ratio, ranging from the lowest, or first gear (M1), to the highest, or top gear (M6). Whilst reference is made in this specification to a gearbox with 6 ratios created with six distinct pairs of toothed profiles (M1...M6), the present invention is applicable similarly to gearboxes having a different number of ratios.

The tooth profiles 31, 32, 33, 34, 35 and 36 of the second plurality of are presented by gear wheels embodied separately one from another and mounted freely to the secondary shaft 3, whilst the toothed profiles 21, 22, 23, 24, 25 and 26 of the first plurality are associated rigidly with the primary shaft 2 (machined by removing metal directly from the shaft and/or consisting in gear wheels fastened permanently to the shaft).

The toothed profiles are configured in such a way as to create a train of gear pairs, denoted M1, M2; M3, M4; M5, M6, ordered in sequence from a lowest transmission ratio (first gear) to a highest transmission ratio (sixth gear).

Located alongside the toothed profiles of the first gear pair M1 is a reverse gear assembly denoted R. The reverse gear assembly R comprises a

respective first toothed profile 40 mounted to and associated rigidly with the primary shaft 2, and a respective second toothed profile 50 mounted freely to the secondary shaft 50. These two toothed profiles 40 and 50 do not mesh directly one with another, and accordingly, the reverse gear assembly R comprises a further gear, namely a reverse idler (not illustrated, being familiar in embodiment), meshing with the two toothed profiles 40 and 50, in such a way that the primary shaft 2 and the secondary shaft 3 can be driven in the same direction of rotation.

Numerals 4 denotes first bearings, located between the first pair of toothed profiles M1 (first gear) and those of the reverse gear assembly R, by which the shafts 2 and 3 are carried in the housing (not indicated) of the gearbox 1.

The two shafts 2 and 3 are carried similarly by second bearings 5, located at the opposite end of the housing, that is to say next to the endmost pair of toothed profiles M6 (sixth gear).

The gearbox 1 further comprises selection means operating between the secondary shaft 3 and the second toothed profiles 31, 32, 33, 34, 35 and 36, by which each of the selfsame second toothed profiles 31, 32, 33, 34, 35 and 36 can be caused to rotate as one with the secondary shaft 3.

More exactly, such selection means comprise a first synchromesh device, or synchronizer 6, serving to lock the second toothed profile 31 of the first gear pair M1 to the secondary shaft 3 so that the two rotate as one. The structure of the first synchronizer 6 is substantially familiar to a person skilled in the art. In effect, the first synchronizer 6 is mounted slidably to the secondary shaft 3 by way of splines, which allow the first synchronizer 6 to move along the axis of rotation X of the secondary shaft 3 while disallowing any angular movement of the synchronizer 6 and secondary shaft 3 one relative to the other. The first synchronizer 6 can be locked stably to the second toothed profile 31 of the first gear pair M1 by causing respective sets of teeth to engage frontally, thereby forcing the second toothed profile 31 of the first gear pair M1 to rotate as one with the secondary shaft 3 and

coupling the primary shaft 2 to the secondary shaft 3 by way of the first pair of profiles establishing the first gearbox ratio M1.

In one embodiment of the invention, the first synchronizer 6 can also be locked stably to the second pair M2 of toothed profiles establishing the second gearbox ratio, on the side opposite from the first pair of toothed profiles M1 establishing the first ratio. Thus, the first synchronizer 6 can also be locked stably to the second toothed profile 32 of the second gear pair M2 by causing respective sets of teeth to engage frontally, thereby coupling the primary shaft 2 to the secondary shaft 3 by way of the second pair of profiles M2 establishing the second gearbox ratio.

Preferably, the selection means further comprise a second synchromesh device or synchronizer 7, likewise of substantially familiar embodiment, serving to engage reverse gear. The second synchronizer 7 is mounted slidably to the secondary shaft 3 by way of splines allowing the second synchronizer 7 to move along the axis of rotation X of the secondary shaft 3 toward and away from the corresponding second toothed profile 50 of the reverse gear assembly. The second synchronizer 7 can be locked stably to the second toothed profile 50 of the reverse gear by causing respective sets of teeth to engage frontally, in such a way as to couple the primary shaft 2 to the secondary shaft 3 by way of the reverse gear assembly R.

The aforementioned synchronizers 6 and 7, substantially conventional in embodiment as aforementioned, engage with the respective second toothed profiles 31, 32 and 50 by way of cone surfaces; more exactly, the two synchromesh devices engage by degrees with the respective second toothed profiles 31, 32 and 50, thus gradually equalizing the revolutions of the primary shaft 2 and the secondary shaft 3.

The synchronizers 6 and 7 are capable of axial movement along the axis of rotation of the secondary shaft 3, induced by selector forks of conventional embodiment (not illustrated).

Advantageously, the selection means further comprise two dog clutch

type coupling collars denoted 10 and 11, mounted slidably to the secondary shaft 3, the first of which is positioned between the third pair of toothed profiles 33 constituting the third gearbox ratio M3 and the fourth pair of toothed profiles 34 constituting the fourth gearbox ratio M4, and the second of which is positioned between the fifth pair of toothed profiles 35 constituting the fifth gearbox ratio M5 and the sixth pair of toothed profiles 36 constituting the sixth gearbox ratio M6. In other words, the first sliding collar 10 serves to engage and disengage third and fourth gear, whilst the second sliding collar 11 serves to engage and disengage fifth and sixth gear.

The two collars 10 and 11 function in substantially familiar fashion, interlocking frontally by way of respective dogs 12 with corresponding dogs 13 presented by the toothed profiles 33, 34, 35 and 36 of the secondary shaft 3. The engagement brought about with the two sliding collars 10 and 11 is not assisted by synchromesh devices. Each collar 10 and 11 presents dogs 12 on both faces, in such a manner as to allow the engagement of two different gear pairs M3, M4; M5 and M6.

In all of the embodiments mentioned, the pairs of toothed profiles M1 and M2 coinciding with the two lowest ratios have helical gear teeth, whereas the remaining pairs of toothed profiles M3, M4, M5 and M6 have spur gear teeth.

Preferably, the components of the reverse gear assembly R will also have helical teeth.

More generally, all of the pairs of toothed profiles associated with a synchromesh device have helical gear teeth.

In a further embodiment of the gearbox, nonetheless, all of the toothed profiles might have spur gear teeth.

Figure 3 illustrates an actuator device 14, shown in a first embodiment, by which the gearbox described above is operated.

The actuator device is an automated manual transmission, or AMT type, comprising a pair of hydraulic or electromechanical linear actuators,

denoted 15 and 16, operating in two respective directions that are mutually transverse, and preferably mutually perpendicular. A first actuator 15 produces a selection movement, that is to say a movement (transmitted by way of selector forks, not illustrated) serving to select and shift one of the following selectors:

- first synchronizer 6, first/second gear;
- second synchronizer 7, reverse gear;
- first sliding collar 10;
- second sliding collar 11;

In the example illustrated, accordingly, the selection movement covers four operating positions (R; M1-M2; M3-M4; M5-M6).

The second linear actuator 16 produces an engaging movement, in each of the aforementioned operating positions, by which one of the available gears is engaged (there are two gears available for each of the operating positions assumed by the first linear actuator 15, except in the case of reverse, which is the only gear engageable when selecting the second synchronizer 7).

In the embodiment illustrated, accordingly, the engaging movement includes two operating positions when selecting the forward gears (M1, M2, M3, M4, M5 and M6), and one operating position when selecting reverse gear (R).

In the example of figure 3, the gearbox 1 according to the present invention comprises an electronic control unit 17 governing the stroke of the two linear actuators 15 and 16. The electronic control unit 17 determines the stroke length of each actuator 15 and 16 in response to commands supplied by the driver through the agency of an input device (shift levers, for example, or buttons, paddles or other selection media).

To advantage, the stroke of the second linear actuator 16 that produces an engaging movement is varied in length by the electronic control unit 17, according to the current position of the first linear actuator 15, or in effect,

according to which of the gears can be selected at a given moment. In other words, the electronic control unit 17 varies the stroke length of the second linear actuator 16 to suit the selector on which the selfsame actuator 16 happens to be operating, and in particular:

5 - the electronic control unit 17 causes the second linear actuator 16 to complete a longer stroke when inducing motion in a synchronizer 6 or 7 (hence, when the first synchronizer 6 or the second synchronizer 7 is selected by the first linear actuator 15);

10 - the electronic control unit 17 causes the second linear actuator 16 to complete a shorter stroke when inducing motion in one of the two sliding dog clutch collars 10 or 11 (hence, when either of the two sliding collars 10 or 11 is selected by the first linear actuator 15).

15 In practice, the stroke required for the synchronizers 6 and 7 to engage fully is longer than the stroke required for the engagement of the sliding collars 10 and 11, and to allow for this difference, the electronic control unit 17 is designed to vary the stroke of the second actuator 16 when operating on one of the synchronizers 6 or 7 or on one of the collars 10 or 11, as the need dictates.

20 In a different embodiment of the gearbox, the actuator device is sequential in operation and comprises a selector drum, or barrel, of the type illustrated in figure 4 and denoted 18.

The selector barrel 18 presents a first peripheral groove 19 and a second peripheral groove 20, positioned side by side and centred on an axis Z of rotation of the barrel 18.

25 The first groove 19 is designed to accommodate a pair of pins inducing movement in respective forks (not illustrated, being of substantially familiar design) of which the function is to shift the synchronizers 6 and 7, and consequently to engage first gear, second gear, and reverse gear. The geometry of the first groove 19 is indicated in figure 5, developed in a plane
30 to illustrate the angular position of its various constituent parts relative to

the axis Z of rotation of the barrel.

The second groove 20 is designed to accommodate a pair of pins inducing movement in respective forks (not illustrated, being of substantially familiar design) of which the function is to shift the sliding collars 10 and 11, and consequently to engage third, fourth, fifth and sixth gears.

Each groove 19 and 20 presents a circular portion 19a and 20a, that is to say a segment of annular shape encircling the axis Z of the barrel 18, and a zigzag portion 19b and 20b functioning as a cam, designed to interact with the aforementioned pins (functioning as cam followers) operating the forks, in such a way that an angular movement of the barrel 18 about its axis Z will induce a translational movement of the forks, preferably parallel with the selfsame axis Z.

To advantage, the zigzag portion 19b of the first groove 19 (bold lines in figure 5) extends away from the annular portion 19a along the axis Z of the barrel 18 by a distance greater than the zigzag portion 20b of the second groove 20 (phantom lines in figure 5). This greater distance is instrumental in inducing a longer translational movement of the pins operating the respective selector forks, in such a way as to shift the synchronizers 6 and 7 through a distance greater than that of the sliding collars 10 and 11. Thus, the actuator is able to produce both the shorter stroke required to engage the sliding collars 10 and 11, and the longer stroke required for full engagement of the synchronizers 6 and 7.

The two zigzag portions 19b and 20b can be spaced apart angularly about the axis X of rotation of the barrel sufficiently to allow correct engagement of all the gears (six forward plus one reverse) one at a time. Alternatively, the two zigzag portions 19b, 20b can occupy the same angular position relative to the axis X of rotation, in which case the correct engagement of strictly one gear at a time will be assured by strategic angular spacing of the selector fork pins internally of the grooves 19 and 20.

The selector barrel 18 also presents a peripheral safety groove 21 extending partially around the developable circumference of the selfsame barrel 18 (figure 4). The safety groove 21 is occupied permanently by a fixed bolt (not illustrated, being of substantially familiar embodiment) such as will allow the rotation of the selector barrel 18 about the axis Z of rotation only through an angular distance determined by the length of the groove 21. With this arrangement, the limit positions assumable by the selector barrel 18 when rotating about its axis Z are determined automatically.

In addition, the selector barrel 18 presents an auxiliary groove 22, extending likewise partially around the developable circumference of the barrel 18. The auxiliary groove 22 can interact with an auxiliary bolt (not illustrated, being of substantially familiar embodiment), of which the tip is retained normally in the groove 22 by a spring bias. The auxiliary bolt is releasable by pulling upwards, thereby allowing the barrel 18 to assume angular positions that would otherwise be inhibited by the bolt, and can function as a lock and release mechanism for reverse gear, by way of example.

Alongside the aforementioned grooves 19, 20, 21 and 22, the selector barrel 18 presents a plurality of depressions 23, arranged along a peripheral circumference around the axis Z of rotation, equispaced angularly and engageable by a spring-loaded plunger (not illustrated, being of substantially familiar embodiment), so that the barrel 18 can be held in angular positions established by the depressions 23. In the example illustrated, the depressions 23 are nine in number and equispaced angularly at 40° one from the next.

In an alternative embodiment of the actuator (not illustrated), the selector barrel 18 might present four distinct grooves, one for the engagement of reverse gear, each of the other three for the engagement of two consecutive forward gears; accordingly, each groove in this instance will be occupied by just one selector fork pin. In this further embodiment, more generally, the

selector barrel presents at least one first groove for the engagement of reverse gear, a second groove for the engagement of first and second gear, and a third groove for the engagement of third and fourth gear (with possibly a fourth groove for the engagement of fifth and sixth gear), of which the first and second grooves control the movement of a synchronizer, whilst the third groove controls the movement of a sliding dog clutch collar. Accordingly, the first and second grooves will be configured in such a way as to induce an axial displacement greater than the displacement induced by the third groove.

The drawbacks associated with the prior art are overcome by the present invention, and the stated objects duly realized.

Using synchromesh devices associated with the lower ratios (first gear and reverse, and possibly second gear), in combination with sliding dog clutch collars for the higher ratios, the gearbox is able to ensure a level of driving comfort suitable for road use, while also delivering the kind of performance typical of competition or racing cars.

Moreover, the use of dog clutches (operating in combination with gears having spur teeth) brings the further advantage of reducing gear shift times to the bare minimum (20 milliseconds, or thereabouts), typical of high performance cars in general, and competition cars in particular.

At the same time, using synchromesh devices for first gear and reverse gear shifts (possibly in conjunction with helical gear teeth for first gear and/or reverse gear), the engagement of these lower gears from neutral is rendered notably smooth, compared with that provided by a sliding dog clutch collar and no synchromesh. Given that with the clutch disengaged, the primary shaft is driven in rotation at particularly low torque to facilitate the engagement of first gear or reverse gear (with the primary and secondary shafts at a complete standstill, it might not be possible for the gears to mesh in certain angular configurations of the shafts), the use of synchromesh allows a smooth and gradual engagement of these lower gears, with no

annoyance to the driver.

Moreover, the engagement of first gear from neutral has no impact on the performance of the vehicle, as this particular gear shift is performed generally with the vehicle at standstill, and therefore at a time when high
5 performance is not a prime consideration.

Claims

1) A gearbox for motor vehicles, comprising:

- a first shaft (2) presenting a plurality of first tooth profiles (21, 22, 23, 24, 25, 26);

5 - a second shaft (3) presenting a plurality of second toothed profiles (31, 32, 33, 34, 35, 36) simultaneously in constant mesh with the first toothed profiles (21, 22, 23, 24, 25, 26) of the first shaft (2), the two pluralities combining thus to form distinct pairs of toothed profiles (M1, M2, M3, M4, M5, M6) with different transmission ratios, ranging from a lowest or first
10 gear (M1) to a highest or top gear (M6);

- selection means (6, 7, 10, 11), acting on certain of the first toothed profiles (21, 22, 23, 24, 25, 26) and/or the second toothed profiles (31, 32, 33, 34, 35, 36) in such a way as to couple the first and second shafts (2, 3) in rotation by means of one of the pairs of toothed profiles (M1, M2, M3, M4, M5, M6) and thus select a desired gearbox ratio;

15 characterized in that the selection means (6, 7, 10, 11) comprise at least one first synchromesh device (6), associated with the pair of toothed profiles (M1) constituting the first gear, and at least one collar (10; 11) presenting a dog type coupling and no synchromesh device, associated with at least one
20 of the remaining pairs of toothed profiles (M3, M4, M5, M6) constituting the higher gears.

2) A gearbox as in claim 1, further comprising at least one reverse gear assembly (R), wherein selection means (6, 7, 10, 11) comprise at least one second synchromesh device (7) associated with the selfsame reverse gear
25 assembly (R).

3) A gearbox as in claim 2, wherein the first synchromesh device (6) operates also on a second pair (M2) of toothed profiles constituting the

second gear, thereby coupling the first and second shafts (2, 3) one to another in rotation by means of the second pair of toothed profiles (M2).

4) A gearbox as in any one of the preceding claims, wherein at least one of the toothed profiles (21, 22, 31, 32) associated with a synchromesh device
5 (6, 7) presents helical gear teeth, whilst the remaining toothed profiles (23, 24, 25, 26, 33, 34, 35, 36) associated with a collar (10; 11) having a dog type coupling and no synchromesh device, present spur gear teeth.

5) A gearbox as in any one of claims 1 to 3, wherein all of the first toothed
10 profiles (21, 22, 23, 24, 25, 26) and the second toothed profiles (31, 32, 33, 34, 35, 36) present spur gear teeth.

6) A gearbox as in any one of the preceding claims, further comprising an actuator device (14) acting on the selection means (6, 7, 10, 11) and operable by a user to engage a selected gearbox ratio, wherein the gearbox (1) is equipped with an electronic unit (17) controlling the actuator device
15 (14) and designed to induce an engage/disengage stroke for the first and/or second synchromesh device (6; 7) that is different to and preferably longer than the engage/disengage stroke induced for the collar (10; 11) having a dog type coupling.

7) A gearbox as in claim 6, wherein the actuator device (14) comprises two
20 linear actuators (15, 16), the first (15) designed to select one of the selection means (6, 7, 10, 11) by inducing a selection movement, and the second (16) designed to control an engaging movement of the selection means (6, 7, 10, 11), preferably in a direction perpendicular to the selection movement; the electronic control unit (17) being designed to induce a stroke of the second
25 linear actuator (16) that is variable in length depending on the selected gearbox ratio, and designed preferably to induce a stroke of the second linear actuator (16) that is longer for the engagement of the first gear (M1),

associated with the synchromesh device (6), than for the engagement of a higher gear (M3, M4, M5, M6) associated with the collar (10; 11) having a dog type coupling.

8) A gearbox as in claim 6 or 7, wherein the actuator device (14) is an automated actuator of H-pattern type.

9) A gearbox as in any one of claims 1 to 5, further comprising a sequential actuator device acting on the selection means (6, 7, 10, 11) and operable by a user to engage a gearbox ratio selected by the selfsame user, wherein the sequential actuator comprises a selector barrel (18) acting on the selection means (6, 7, 10, 11) and configured geometrically in such a way as to induce an engage/disengage stroke for the first and/or second synchromesh device (6; 7) that is different to and preferably longer than the engage/disengage stroke induced for the collar (10; 11) having a dog type coupling.

10) A gearbox as in claim 9, wherein the selector barrel (18) comprises at least one first peripheral groove (19) controlling the first and/or second synchromesh device (6, 7) by means of a first cam and follower mechanism, and a second peripheral groove (20) controlling the collar (10, 11) having a dog type coupling by means of a second cam and follower mechanism; the first peripheral groove (19) and the second peripheral groove (20) being configured geometrically in such a way that a rotation of the selector barrel (18) about its axis (Z) of rotation induces a movement in the first and/or second synchromesh device (6, 7) along the axis (Z) that is different and preferably longer than a movement induced in the collar (10, 11) having a dog type coupling along the selfsame axis (Z).

11) A gearbox as in claim 10, wherein each of the first and second peripheral grooves (19, 20) of the selector barrel (18) presents a

- predominating portion (19a, 20a) of substantially annular shape encircling the axis (Z) of rotation of the selector barrel (18), and at least one zigzag portion (19b, 20b) designed to generate the movement of the synchromesh device (6, 7) or of the collar (10, 11) having a dog type coupling back and forth along the axis (Z) of rotation of the selector barrel (18); the zigzag portion (19b) of the first peripheral groove (19) being of length, measured along the axis (Z) of rotation of the selector barrel (18), longer than the length of the zigzag portion (20b) of the second peripheral groove (20) measured likewise along the selfsame axis (Z).
- 12) A gearbox as in any of the preceding claims, wherein the first shaft (2) is a primary shaft connectable to the crankshaft of an internal combustion engine, and the second shaft (3) is a secondary shaft connectable to a final drive, preferably a differential connected to the drive wheels of a vehicle; the first toothed profiles (21, 22, 23, 24, 25, 26) being integral with the primary shaft, the second toothed profiles (31, 32, 33, 34, 35, 36) mounted freely to the secondary shaft and caused alternately to rotate as one with the secondary shaft through the agency of the selection means (6, 7, 10, 11).
- 13) A road vehicle, in particular a high performance vehicle, comprising a gearbox (1) as in any one of the preceding claims.

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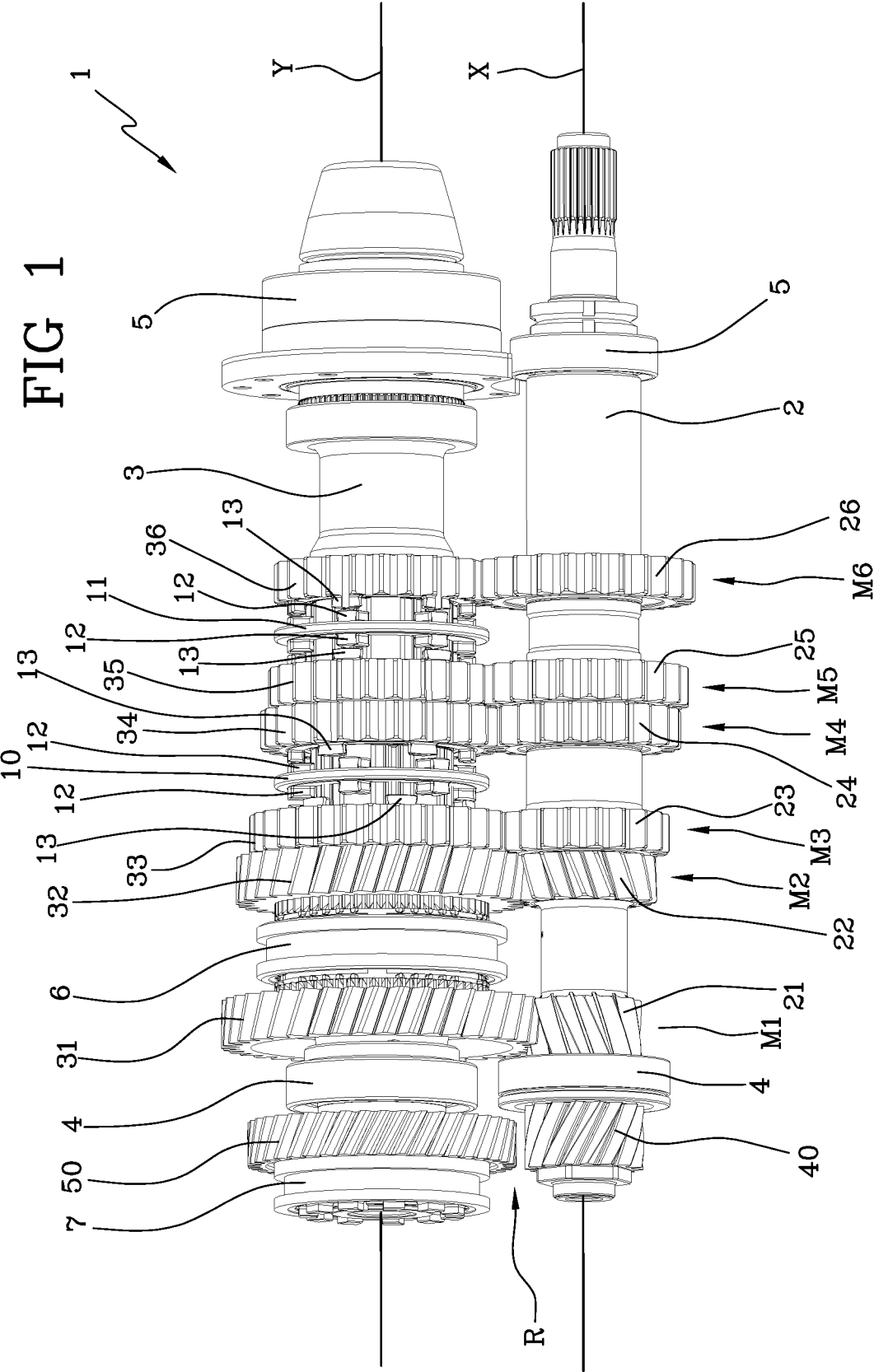


FIG 2

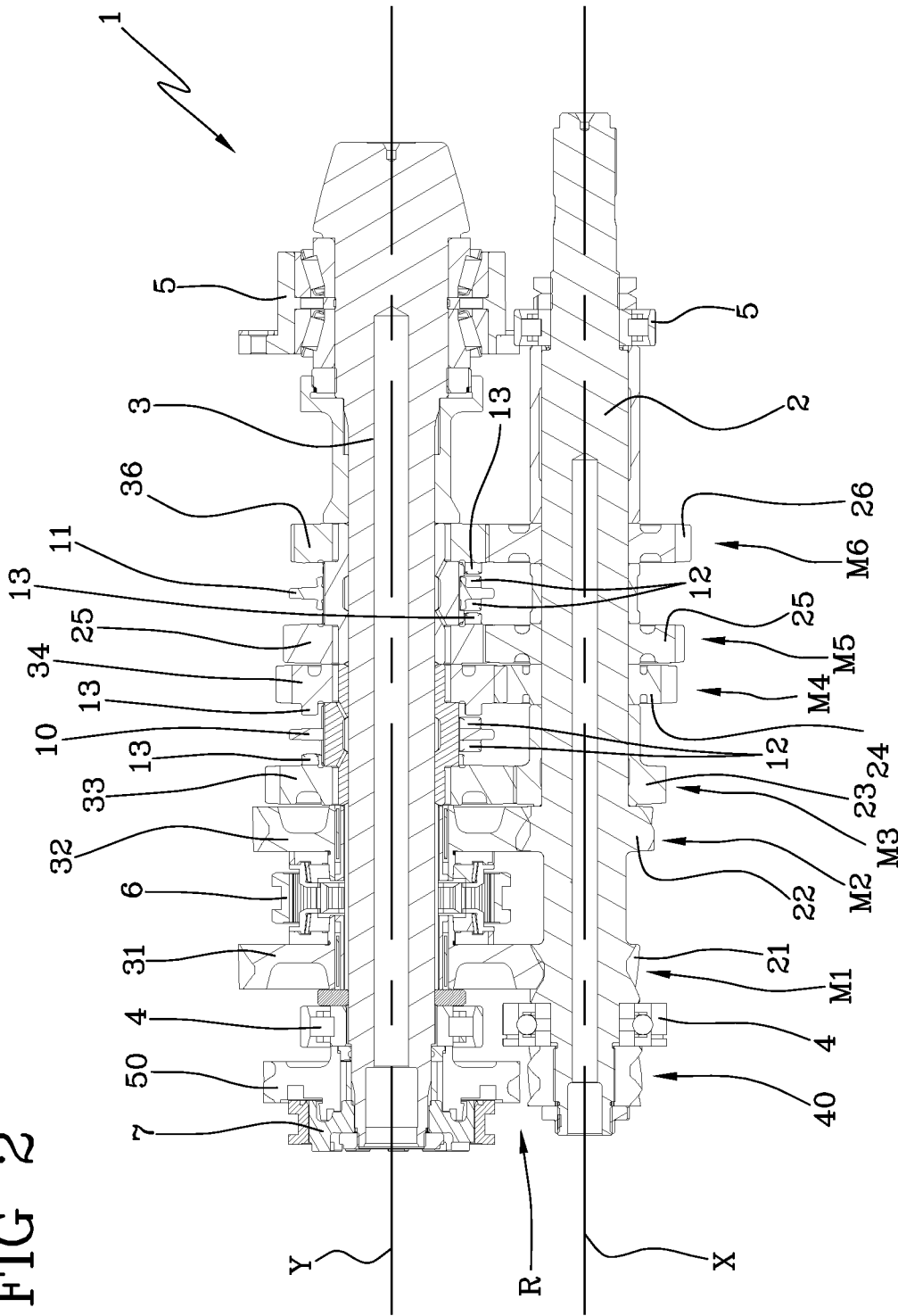


FIG 3

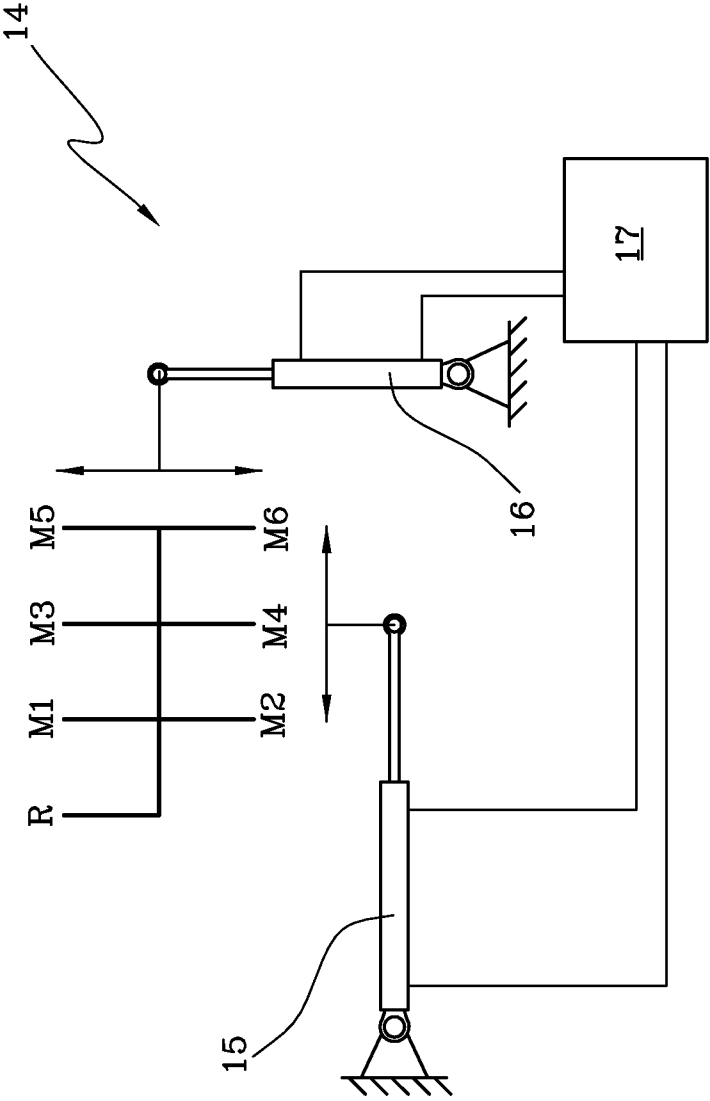


FIG 5

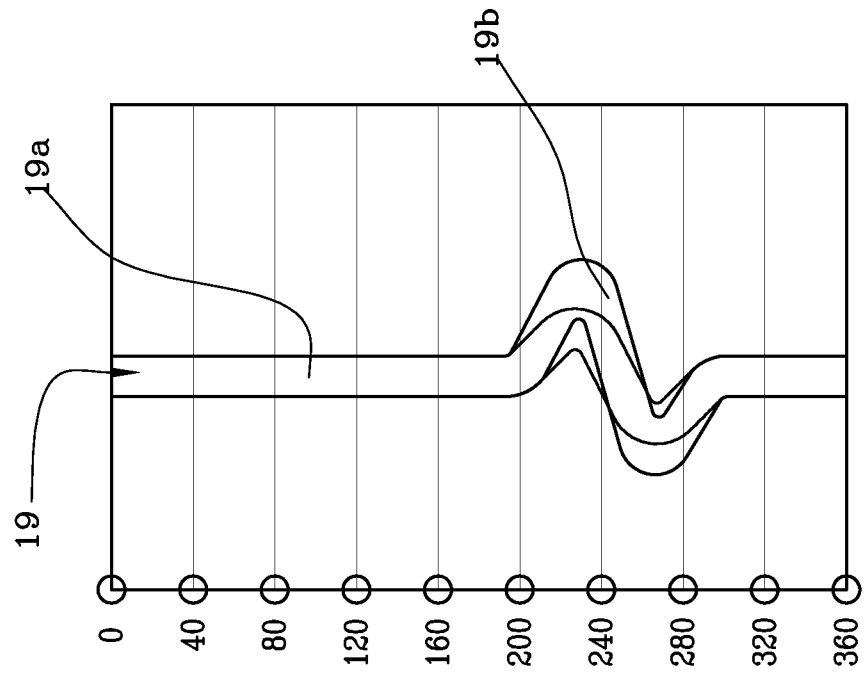
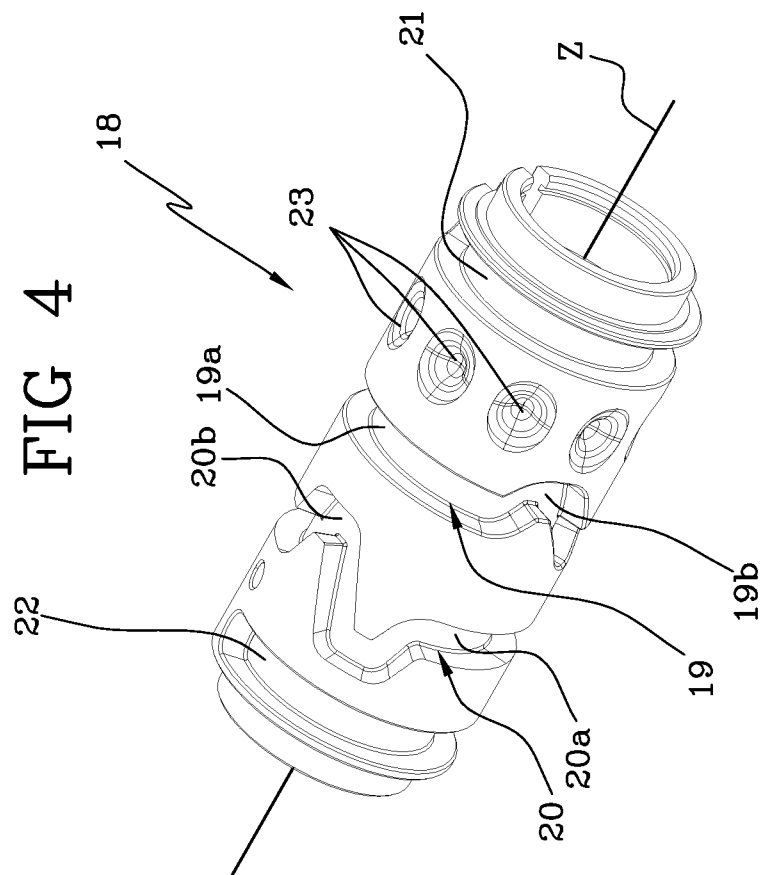


FIG 4



INTERNATIONAL SEARCH REPORT

International application No

PCT/IB2010/053356

A. CLASSIFICATION OF SUBJECT MATTER

INV. F16H3/089

ADD.

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

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

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 2005/204842 A1 (BALDWIN REID [US] ET AL) 22 September 2005 (2005-09-22) paragraphs [0034] - [0036]; figure 1	1-13
A	EP 2 042 783 A1 (HONDA MOTOR CO LTD [JP]) 1 April 2009 (2009-04-01) the whole document	1-13
A	US 5 479 835 A (ESPARSA RODOLFO [MX] ET AL) 2 January 1996 (1996-01-02) * abstract; figures	2
A	EP 1 233 212 A2 (EATON CORP [US]) 21 August 2002 (2002-08-21) figure 1 figure 3 figure 5 paragraph [0045]	4-8
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☒ Further documents are listed in the continuation of Box C.☒ See patent family annex.

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Date of the actual completion of the international search

15 September 2010

Date of mailing of the international search report

04/10/2010

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C(Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
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INTERNATIONAL SEARCH REPORT

Information on patent family members

International application No

PCT/IB2010/053356

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