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(54) Title: METHOD FOR CONTROLLING A HYBRID DRIVELINE IN ORDER TO OPTIMIZE TORQUE FROM A COMBUSTION ENGINE ARRANGED AT THE DRIVELINE

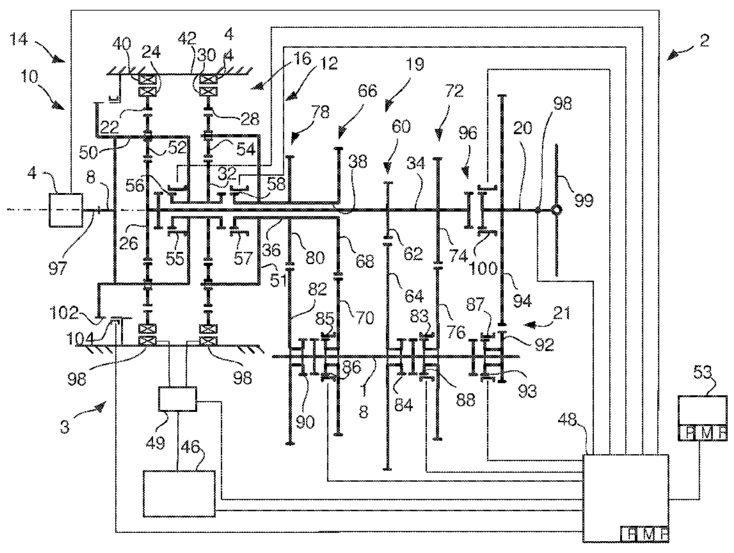

Fig.2

(57) Abstract: The present invention relates to a method to control a hybrid powertrain (3) to optimise the torque from a combustion engine 4 arranged in the hybrid driveline 3, wherein the hybrid powertrain (3) also comprises a gearbox (2) with an input shaft (8) and an output shaft (20); a first planetary gear (10), connected to the input shaft (8) and a first main shaft (34); a second planetary gear (12), connected to the first planetary gear (10) and a second main shaft (36); a first electrical machine (14), connected to the first planetary gear (10); a second electrical machine (16), connected to the second planetary gear (12); at least one gear pair (G1, 60, 72) connected with the first planetary gear (10) and the output shaft (20); and at least one gear pair (G2, 66, 78) connected with the second planetary gear (12) and the output shaft (20), wherein the combustion engine (4) is connected with a first planetary wheel carrier (50), arranged in the first planetary gear (10) via the input shaft (8) of the gearbox (2), and wherein the second main shaft (36) is connected with a planetary wheel carrier (51), arranged in the second planetary gear (12). The method comprises the steps: a) to disconnect a first planetary wheel carrier (50), arranged in the first planetary gear (10), and a first sun wheel (26) from each other, or to disconnect a second planetary wheel carrier (51), arranged in the second planetary gear (12), and a second sun wheel (32) from each other; b) to control the combustion engine (4) to a predetermined engine speed (nice); and c) to control the first and the second electrical machines (14; 16) in such a way that a desired torque (TDrv) is achieved in the output shaft (20), while simultaneously a requested total power consumption (PEM) of the first and the second electrical

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machines (14; 16), is achieved. The invention also relates to a computer program (P) to control a hybrid powertrain (3) and a computer program product comprising program code for an electronic control device (48) or another computer (53) to implement the method according to the invention.

Method for controlling a hybrid driveline in order to optimize torque from a combustion engine arranged at the driveline

BACKGROUND OF THE INVENTION AND PRIOR ART

The present invention relates to a method to control a hybrid powertrain according to the preamble to claim 1. The invention also relates to a vehicle, comprising such a hybrid powertrain according to the preamble to claim 7, a computer program to control such a hybrid powertrain according to the preamble to claim 8, and a computer program product comprising program code according to the preamble to claim 9.

Hybrid vehicles may be driven by a primary engine, which may be a combustion engine, and a secondary engine, which may be an electrical machine. The electrical machine is equipped with at least one energy storage device, such as an electro-chemical energy storage device, for storage of electric power and control equipment to control the flow of electric power between the energy storage device and the electrical machine. The electrical machine may thus alternately operate as a motor and as a generator, depending on the vehicle's operating mode. When the vehicle is braked, the electrical machine generates electric power, which is stored in the energy storage device. This is usually referred to as regenerative braking, which entails that the vehicle is decelerated with the help of the electrical machine and the combustion engine. The stored electric power is used later for operation of the vehicle.

A gearbox in a hybrid vehicle may comprise a planetary gear. The planetary gearbox usually comprises three components, which are rotatably arranged in relation to each other, namely a sun wheel, a planetary wheel carrier and an internal ring gear. With knowledge about the number of cogs in the sun wheel and the internal ring gear, the mutual speeds of the three components may be determined during operation. One of the components of the planetary gear may be connected with an output shaft in a combustion engine. This component of the planetary gear thus rotates with a rotational speed corresponding to the rotational speed of the output shaft in the combustion engine. A second component in the planetary gear may be connected with an input shaft to a transmission device. This component of the planetary gear thus rotates with the same rotational speed as the input shaft to the transmission device. A third component in the planetary gear is used to achieve hybrid operation, connected with a rotor in an electrical machine. This component in the planetary gear thus rotates with the same rotational speed as the rotor of the electrical machine, if they are directly connected with each other. Alternatively, the electrical machine may be connected with the third component of the planetary gear via a transmission that has a gearing. In this case, the electrical machine and the third component in the planetary gear may rotate with different rotational speeds. The engine speed and/or the torque of the electrical machine may be controlled steplessly. During operating times when the input shaft to the transmission device must be provided with a rotational engine speed and/or torque, a control device having knowledge about the engine speed of the combustion engine calculates the rotational speed with which the third component must be operated, in order for the input shaft to the transmission device to obtain the desired rotational speed. A control device activates the electrical machine, so that it provides the third component with the calculated engine speed and thus the input shaft to the transmission device with the desired rotational speed.

Depending on the design of the gearbox connected to the planetary gear, a torque interruption between the gear steps may be avoided. Often, however, separate and complex devices are required in the gearbox in order to eliminate or reduce the torque interruption, so that a perception of stepless gear shifts is obtained.

By connecting the combustion engine's output shaft, the electrical machine's rotor and the transmission device's input shaft with a planetary gear, the conventional clutch mechanism may be avoided. At acceleration of the vehicle, an increased torque must be delivered from the combustion engine and the electrical machine to the transmission device, and further to the vehicle's driving wheels. Since both the combustion engine and the electrical machine are connected with the planetary gear, the largest possible torque delivered by the combustion engine and the electrical machine will be limited by one of these drive units; i.e. the one whose maximum torque is lower than the second drive unit's maximum torque, having regard to the gearing between them. In case the electrical machine's highest torque is lower than the combustion engine's highest torque, having regard to the gearing between them, the electrical machine will not be able to generate a sufficiently large reaction torque to the planetary gear, entailing that the combustion engine may not transfer its highest torque to the transmission device and further to the vehicle's driving wheels. Thus, the highest torque that may be transferred to the transmission device is limited by the electrical machine's strength. This is also apparent from the so-called planet equation.

Using a conventional clutch, which disconnects the gearbox's input shaft from the combustion engine during shifting processes in the gearbox, entails disadvantages, such as heating of the clutch's discs, resulting in wear of the clutch discs and an increased fuel consumption. A conventional clutch mechanism is also relatively heavy and costly. It also occupies a relatively large space in the vehicle.

In a vehicle, the space available for the drive arrangement is often limited. If the drive arrangement comprises several components, such as a combustion engine, an electrical machine, a gearbox and a planetary gear, the construction must be compact. If there are additional components, such as a regenerative braking device, the requirements that the component parts must have a compact construction are even more stringent. At the same time, the component parts in the drive arrangement must be designed with dimensions that are able to absorb the required forces and torque.

For some types of vehicles, especially heavy goods vehicles and buses, a large number of gear steps is required. Thus, the number of component parts in the gearbox increases, which must also be dimensioned to be able to absorb large forces and torque arising in such heavy goods vehicles. This results in an increase of the size and weight of the gearbox.

There are also requirements for high reliability and high dependability of the components comprised in the drive device. In case the gearbox comprises multi-plate clutches, a wear arises, which impacts the reliability and life of the gearbox.

At regenerative braking, kinetic energy is converted into electric power, which is stored in an energy storage device, such as accumulators. One factor impacting on the life of the energy storage device is the number of cycles in which the energy storage device provides and extracts power to and from the electric machines. The more cycles, the shorter the life of the energy storage device.

Under certain operating conditions, a higher driving torque from the combustion engine may be requested than what is suitable with the combustion engine's current engine speed. One example of such an operating mode is at the start of a heavy goods vehicle. In cases where an unsuitably high torque is extracted from the combustion engine at low engine speeds, vibrations may arise in the combustion engine.

The document EP-B1-1126987 shows a gearbox with double planetary gears. Each sun wheel of the planetary gear is connected to an electrical machine, and the internal wheels of the planetary gears are connected with each other. The planetary wheel carrier in each planetary gear is connected to a number of gear pairs, so that an infinite number of gear steps is obtained. Another document, EP-B1-1280677, also shows how the planetary gears may be bridged with a gear step arranged on the combustion engine's output shaft.

The document US-A1-20050227803 shows a vehicle transmission with two electric machines, connected to the respective sun wheels in two planetary gears. The planetary gears have a common planetary wheel carrier, which is connected to the transmission's input shaft.

The document WO2008/046185-A1 shows a hybrid transmission with two planetary gears, wherein one electrical machine is connected to one of the planetary gears and a double clutch interacts with the second planetary gear. Both planetary gears also interact with each other via a cogwheel transmission.

SUMMARY OF THE INVENTION

Despite prior art solutions in the field, there is a need to further develop a method to control a hybrid powertrain, in order to achieve an optimal driving torque from the combustion engine.

The objective of this invention is to provide a novel and advantageous method to control a hybrid powertrain, which optimises the torque in a combustion engine arranged in the powertrain.

Another objective of the invention is to provide a novel and advantageous computer program to control a hybrid powertrain.

These objectives are achieved with the method specified at the beginning, which is characterised by the features specified in the characterising portion of claim 1.

These objectives are also achieved with the vehicle specified at the beginning, which is characterised by the features specified in the characterising portion of claim 7.

These objectives are also achieved with the computer program to control the hybrid powertrain, which is characterised by the features specified in the characterising portion of claim 8.

These objectives are also achieved with the computer program product to control the hybrid powertrain, which is characterised by the features specified in the characterising portion of claim 9.

With the method according to the invention, an efficient and reliable method to control a hybrid powertrain, in order to optimise the driving torque from a combustion engine arranged in the hybrid powertrain is obtained; wherein the hybrid powertrain also comprises a gearbox with an input shaft and an output shaft; a first planetary gear, connected to the input shaft and a first main shaft; a second planetary gear, connected to the first planetary gear and a second main shaft; a first electrical machine, connected to the first planetary gear; a second electrical machine, connected to the second planetary gear; at least one gear pair connected with the first planetary gear and the output shaft; and at least one gear pair connected with the second planetary gear and the output shaft, wherein the combustion engine is connected with a first planetary wheel carrier, arranged in the first planetary gear via the input shaft of the gearbox, and wherein the second main shaft is connected with a planetary wheel carrier, arranged in the second planetary gear. A gear pair connected with the first planetary gear is connected with a countershaft, which is connected to the output shaft, and a gear pair connected with the second planetary gear is also connected to the countershaft. Thus, torque may be transferred between the first and the second planetary gears via the countershaft to the output shaft. Depending on which gear is engaged, two of the first planetary gear’s moveable parts, such as a first planetary wheel carrier, a first sun wheel and a first internal ring gear are connected, or two of the second planetary wheel carrier’s moveable parts, such as a second planetary wheel carrier, a second sun wheel and a second internal ring gear are connected. By ensuring that the first planetary gear’s moveable parts and the second planetary gear’s moveable parts, respectively, are disconnected from each other, by controlling the combustion engine to a predetermined engine speed and by controlling the first and the second electrical machine in such a way that a desired torque is achieved in the output shaft, while simultaneously a requested total power consumption of the first and the second electrical machine is achieved, the torque on the combustion engine may be optimised, without having to change gears. That is to say, in operating modes where a driving torque is required, which is higher than the torque which the combustion engine is able to deliver with the current engine speed, the moveable parts of the planetary gears are disconnected and the combustion engine is allowed a higher engine speed, without having to change gears. Thus, vibrations in the combustion engine, which may arise when an unsuitably high torque is extracted from the combustion engine at a low engine speed, are avoided.

Preferably, the moveable parts arranged in the first planetary gear are disconnected from each other by way of controlling the first and/or the second electrical machine in such a way that torque balance is achieved in the first planetary gear, following which a first coupling device is shifted, so that the first planetary wheel carrier and the first sun wheel are disconnected from each other.

Preferably, the moveable parts arranged in the second planetary gear are disconnected from each other by way of controlling the first and/or the second electrical machine in such a way that torque balance is achieved in the second planetary gear, following which a second coupling device is shifted, so that the second planetary wheel carrier and the second sun wheel are disconnected from each other.

Torque balance relates to a state where a torque acts on an internal ring gear arranged in the planetary gear, representing the product of the torque acting on the planetary wheel carrier of the planetary gear and the gear ratio of the planetary gear, while simultaneously a torque acts on the planetary gear's sun wheel, representing the product of the torque acting on the planetary wheel carrier and (1- the planetary gear's gear ratio). In the event two of the planetary gear's component parts, i.e. the sun wheel, the internal ring gear or planetary wheel carriers, are connected with a coupling device, this coupling device does not transfer any torque between the planetary gear's parts when torque balance prevails. Accordingly, the coupling device may easily be shifted and the planetary gear's component parts be disconnected.

The torque on the output shaft of the hybrid powertrain may be obtained by way of a combination of torque from the first electrical machine and torque from the second electrical machine. The hybrid powertrain's driving power is the product of the torque on the output shaft and the rotational speed of the gearbox. The hybrid powertrain's driving power may also be obtained through the combustion engine's power plus the power from an energy storage device in the hybrid powertrain. The power in the energy storage device is the same as the power consumption of the first and the second electrical machines, when no other power consumer uses the energy storage device. The power of the combustion engine is obtained as the product of the combustion engine's torque and engine speed, and the power consumption of the first and the second electrical machines is obtained as the sum of the product of their respective torque and engine speed.

Suitably, the predetermined engine speed of the combustion engine is determined based on a desired driving power from the combustion engine. Since the power of the combustion engine is the product of its torque and engine speed, the engine speed may be obtained with a determined desired value for the power.

The electrical machines, which are connected to the planetary gears, may generate power and/or supply torque depending on the desired operating mode. The electrical machines may also, at certain operating times, supply each other with power.

Suitably, the first planetary wheel carrier in the first planetary gear is directly connected with the combustion engine via the input shaft. Alternatively, the first planetary wheel carrier is connected with the combustion engine via a coupling device. The second planetary wheel carrier in the second planetary gear is suitably directly connected with the second main shaft, and therefore with the transmission device. Thus, a hybrid powertrain is achieved, which may transfer a large torque to the output shaft, and the therewith connected driving wheels in all operating modes, without being dependent on electric power from thane energy storage device.

The first planetary wheel carrier in the first planetary gear is preferably connected with the second sun wheel of the second planetary gear. The first sun wheel in the first planetary gear is preferably connected with the first main shaft, and the second planetary wheel carrier in the second planetary gear is preferably connected with the second main shaft. Thus, a transmission is obtained, which shifts gears without torque interruption. Alternatively, the first planetary wheel carrier in the first planetary gear is connected with the second internal ring gear of the second planetary gear. Alternatively, the first main shaft is connected with an internal ring gear arranged in the first planetary gear.

Preferably, the first main shaft and the second main shaft are connected to a transmission device comprising a number of connectible and disconnectable gear pairs. The gear pairs comprise cogwheels, which are mechanically lockable with and disconnectable from the countershaft. Thus, a number of fixed gear steps is obtained, which may be shifted without torque interruption. The cogwheels that may be locked on the countershaft also result in a compact construction with a high reliability and high dependability. A gear pair may thus be disconnected, whereat the corresponding cogwheel is disconnected from the countershaft, and a gear pair may be connected, whereat the corresponding cogwheel is connected to the countershaft. Alternatively, pinion gears in the gear pairs may be arranged to be lockable with and disconnectable from the first or second main shaft.

Each of the gear pairs has a gearing, which is adapted to the vehicle's desired driving characteristics. The gear pair with the highest gearing, in relation to the other gear pairs, is suitably connected when the lowest gear is engaged.

With the gearbox according to the invention conventional slip clutches between the combustion engine and the gearbox may be avoided.

A locking mechanism is arranged to fixedly connect the combustion engine's output shaft with the gearbox housing. Thus, the first planetary wheel carrier will also be locked to the gearbox housing. By locking the combustion engine's output shaft with the locking mechanism and the first planetary wheel carrier with the gearbox's housing, the gearbox, and thus the vehicle, becomes adapted for electric operation by the electrical machines. The electrical machines thus emit a torque to the output shaft of the gearbox. Alternatively, a coupling device may be arranged between the output shaft of the combustion engine and the input shaft of the gearbox, wherein the combustion engine may be disconnected by opening the coupling device, and thus the gearbox, and therefore the vehicle may become adapted for electrical operation by the electric machines.

A first and second coupling device is arranged between the planetary wheel carrier and the sun wheel of the respective planetary gears. The task of the coupling devices is to lock the respective planetary wheel carriers with the sun wheel. When the planetary wheel carrier and the sun wheel are connected with each other, the power from the combustion engine will pass through the planetary wheel carrier, the coupling device, the sun wheel and further along to the gearbox, which entails that the planetary wheels do not absorb any torque. This entails that the dimension of the planetary wheels may be adapted only to the electrical machine's torque instead of the combustion engine's torque, which in turn means the planetary wheels may be designed with smaller dimensions. Thus, a drive arrangement according to the invention is obtained, which has a compact construction, a low weight and a low manufacturing cost.

The coupling devices and the locking mechanisms preferably comprise an annular sleeve, which is shifted axially between a connected and a disconnected state. The sleeve encloses, substantially concentrically, the gearbox's rotating components and is moved between the connected and disconnected state with a power element. Thus, a compact construction is obtained, with a low weight and a low manufacturing cost.

BRIEF DESCRIPTION OF THE DRAWINGS

Below is a description, as an example, of preferred embodiments of the invention with reference to the enclosed drawings, on which:

Fig. 1 schematically shows a vehicle in a side view with a hybrid powertrain, controlled according to the present invention,

Fig. 2 schematically shows a side view of a hybrid powertrain, controlled according to the present invention,

Fig. 3 shows a simplified schematic view of the hybrid powertrain in Fig. 2, and

Fig. 4 shows a flow chart of the method to control hybrid powertrain according to the present invention.

DETAILED DESCRIPTION OF PREFERRED EMBODIMENTS OF THE INVENTION

Fig. 1 shows a schematic side view of a vehicle 1, comprising a gearbox 2 and a combustion engine 4, which are comprised in a hybrid powertrain 3. The combustion engine 4 is connected to the gearbox 2, and the gearbox 2 is further connected to the driving wheels 6 of the vehicle 1 via a propeller shaft 9. The driving wheels 6 are equipped with brake devices 7 to brake the vehicle 1.

Fig. 2 shows a schematic side view of a hybrid powertrain 3 with a gearbox 2, comprising an input shaft 8, a first and a second planetary gear 10 and 12, respectively, a first and a second electrical machine 14 and 16, respectively, a countershaft 18 and an output shaft 20. The hybrid powertrain 3 comprises a combustion engine 4 connected to the gearbox 2. The combustion engine 4 is connected with the gearbox 2 via the input shaft 8 of the gearbox. The combustion engine has an output shaft 97. The output shaft 97 of the combustion engine 4 is connected to the input shaft of the gearbox 2. The first planetary gear 10 has a first internal ring gear 22, to which a first rotor 24 in the first electrical machine 14 is connected. The first planetary gear 10 also has a first sun wheel 26 and a first planetary wheel carrier 50. The first planetary wheel carrier 50 is connected with the combustion engine 4 via the input shaft 8 of the gearbox. The second planetary gear 12 has a second internal ring gear 28, to which a second rotor 30 of the second electrical machine 16 is connected. The second planetary gear 12 has a second sun wheel 32 and a second planetary wheel carrier 51. The first and the second sun wheels 26 and 32, respectively, are coaxially arranged, which, according to the embodiment displayed, entails that a first main shaft 34 arranged on the first sun wheel 26 extends inside a second main shaft 36, which is equipped with a central boring 38, arranged on the second planetary wheel carrier 51. It is also possible to arrange the first main shaft 34 in parallel with and next to the second main shaft 36. In this case, the countershaft 18 is suitably arranged between the first main shaft 34 and the second main shaft 36, and the torque may be extracted directly from the countershaft 18. The countershaft 18 thus constitutes, in this case, the output shaft 20.

The first electrical machine 14 is equipped with a first stator 40, which is connected to the vehicle 1, via a gear housing 42 surrounding the gearbox 2. The second electrical machine 16 is equipped with a second stator 44, which is connected to the vehicle 1, via a gear housing 42 surrounding the gearbox 2. The first and the second electrical machine 16 are connected to an energy storage device 46, such as a battery, which, depending on the vehicle's 1 operating mode, operates the electrical machines 14 and 16, respectively. At other operating modes, the electrical machines 14 and 16, respectively, may work as generators, wherein power is supplied to the energy storage device 46. An electronic control device 48 is connected to the energy storage device 46 and controls the supply of power to the electrical machines 14 and 16, respectively. Preferably the energy storage device 46 is connected to the electrical machines 14 and 16, respectively, via a switch 49, which is connected to the control device 48. In some operating modes, the electrical machines 14 and 16, respectively, may also operate each other. Electric power is then led from one of the electrical machines 14, 16 to the second electrical machine 14, 16 via the switch 49, connected to the electrical machines 14, 16. Thus, it is possible to achieve a power balance between the electrical machines 14, 16. Another computer 53 may also be connected to the control device 48 and the gearbox 2.

The first planetary gear 10 is equipped with a first planetary wheel carrier 50, on which a first set of planetary wheels 52 is mounted. The second planetary gear 12 is equipped with a second planetary wheel carrier 51, on which a second set of planetary wheels 54 is mounted. The first set of planetary wheels 52 interacts with the first internal ring gear 22 and the first sun wheel 26. The second set of planetary wheels 54 interacts with the second internal ring gear 28 and the second sun wheel 32. The input shaft 8 of the gearbox 2 is connected with the first planetary wheel carrier 50.

A first coupling device 56 is arranged between the first sun wheel 26 and the first planetary wheel carrier 50. By arranging the first coupling device 56 in such a way that the first sun wheel 26 and the first planetary wheel carrier 50 are connected with each other, and may therefore not rotate in relation to each other, the first planetary wheel carrier 50 and the first sun wheel 26 will rotate with equal rotational speeds.

A second coupling device 58 is arranged between the second sun wheel 32 and the second planetary wheel carrier 51. By arranging the second coupling device 58 in such a way that the second sun wheel 32 and the second planetary wheel carrier 51 are connected with each other, and may therefore not rotate in relation to each other, the second planetary wheel carrier 51 and the first sun wheel 32 will rotate with equal rotational speeds.

Preferably, the first and second coupling devices 56, 58 comprise a first and a second splines-equipped coupling sleeve 55 and 57, respectively, which is axially shiftable on a splines-equipped section on the first and second, respectively, planetary wheel carrier 50 and 51, and on a splines-equipped section on the respective sun wheels 26 and 32. By shifting the respective coupling sleeve 55, 57 so that the splines-equipped sections are connected via the respective coupling sleeves 55, 57, the first planetary wheel carrier 50 and the first sun wheel 26, as well as the second planetary wheel carrier 51 and the second sun wheel 32, respectively, become mutually interlocked with each other and may not rotate in relation to each other.

The first and second coupling device 56, 58 according to the embodiment displayed in Fig. 2 are arranged between the first sun wheel 26 and the first planetary wheel carrier 50, and between the second sun wheel 28 and the second planetary wheel carrier 51, respectively. However, it is possible to arrange an additional or alternative coupling device (not displayed) between the first internal ring gear 22 and the first planetary wheel carrier 50, and also to arrange an additional or alternative coupling device (not displayed) between the second internal ring gear 28 and the second planetary wheel carrier 51.

The first planetary wheel carrier 50 in the first planetary gear 10 is, in this embodiment, fixedly connected with the second sun wheel 32 of the second planetary gear 12.

A transmission device 19, which comprises a first gear pair 60, arranged between the first planetary gear 10 and the output shaft 20 is connected to the first and the second main shaft 34, 36. The first gear pair 60 comprises a first pinion gear 62 and a first cogwheel 64, which are in engagement with each other. A second gear pair 66 is arranged between the second planetary gear 12 and the output shaft 20. The second gear pair 66 comprises a second pinion gear 68 and a second cogwheel 70, which are in engagement with each other. A third gear pair 72 is arranged between the first planetary gear 10 and the output shaft 20. The third gear pair 72 comprises a third pinion gear 74 and a third cogwheel 76, which are in engagement with each other. A fourth gear pair 78 is arranged between the second planetary gear 12 and the output shaft 20. The fourth gear pair 78 comprises a fourth pinion gear 80 and a fourth cogwheel 82, which are in engagement with each other.

On the first main shaft 34, the first and the third pinion gears 62 and 74, respectively, are arranged. The first and the third pinion gears 62 and 74, respectively, are fixedly connected with the first main shaft 34, so that they may not rotate in relation to the first main shaft 34. On the second main shaft 36, the second and the fourth pinion gears 68 and 80, respectively, are arranged. The second and the fourth pinion gears 68 and 80, respectively, are fixedly connected with the second main shaft 36, so that they may not rotate in relation to the second main shaft 36.

The countershaft 18 extends substantially in parallel with the first and the second main shaft 34 and 36, respectively. On the countershaft 18, the first, second, third and fourth cogwheels 64, 70, 76 and 82, respectively, are mounted. The first pinion gear 62 engages with the first cogwheel 64, the second pinion gear 68 engages with the second cogwheel 70, the third pinion gear 74 engages with the third cogwheel 76 and the fourth pinion gear 80 engages with the fourth cogwheel 82.

The first, second, third and fourth cogwheels 64, 70, 76 and 82, respectively, may be individually locked with and released from the countershaft 18 with the assistance of the first, second, third and fourth coupling elements 84, 86, 88 and 90, respectively. The coupling elements 84, 86, 88 and 90, respectively, preferably consist of splines-equipped sections on the cogwheels 64, 70, 76 and 82, respectively, and on the countershaft 18, which interact with fifth and sixth coupling sleeves 83, 85 which engage mechanically with the splines-equipped sections of the respective first to fourth cogwheel 64, 70, 76 and 82 and of the countershaft 18. The first and third coupling elements 84, 88 are preferably equipped with a common coupling sleeve 83, and the second and fourth coupling elements 86, 90 are preferably equipped with a common coupling sleeve 85. In the released state, a relative rotation may occur between the cogwheels 64, 70, 76 and 82 and of the countershaft 18. The coupling elements 84, 86, 88 and 90, respectively, may also consist of friction clutches. On the countershaft 18 a fifth cogwheel 92 is also arranged, which engages with a sixth cogwheel 94, which is arranged on the output shaft 20 of the gearbox 2.

The countershaft 18 is arranged between the respective first and second planetary gears 10, 12 and the output shaft 20, so that the countershaft 18 is connected with the output shaft 20 via a fifth gear pair 21, which comprises the fifth and the sixth cogwheel 92, 94. The fifth cogwheel 92 is arranged so it may be connected with and disconnected from the countershaft 18 with a fifth coupling element 93.

By disconnecting the fifth cogwheel 92, which is arranged to be disconnectable from the countershaft 18, it is possible to transfer torque from the second planetary gear 12 to the countershaft 18 via, for example, the second gear pair 66, and to further transfer torque from the countershaft 18 to the output shaft 20 via, for example, the first gear pair 60. Thus, a number of gear steps is obtained, wherein torque from one of the planetary gears 10, 12 may be transferred to the countershaft 18, and further along from the countershaft 18 to the main shaft 34, 36 connected with the second planetary gear 10, 12, finally to transfer torque to the output shaft 20 of the gearbox 2. This presumes, however, that a coupling mechanism 96 arranged between the first main shaft 34 and the output shaft 20 is connected, which is described in more detail below.

The fifth cogwheel 92 may be locked to and released from the countershaft 18 with the assistance of a fifth coupling element 93. The coupling element 93 preferably consists of splines-equipped sections adapted on the fifth cogwheel 92 and the countershaft 18, which sections interact with a ninth coupling sleeve 87, which engages mechanically with the splines-equipped sections of the fifth cogwheel 92 and the countershaft 18. In the released state, a relative rotation may occur between the fifth cogwheel 92 and the countershaft 18. The fifth coupling element 93 may also consist of friction clutches.

Torque transfer from the input shaft 8 of the gearbox 2 to the output shaft 20 of the gearbox 2 may occur via the first or the second planetary gear 10 and 12, respectively, and the countershaft 18. The torque transfer may also occur directly via the first planetary gear 10, whose first sun wheel 26 is connected, via the first main shaft 34, to the output shaft 20 of the gearbox 2 via a coupling mechanism 96. The coupling mechanism 96 preferably comprises a splines-equipped seventh coupling sleeve 100, which is axially shiftable on the first main shaft 34 and on the splines-equipped sections of the output shaft 20. By shifting the seventh coupling sleeve 100, so that the splines-equipped sections are connected via the seventh coupling sleeve 100, the first main shaft 34 becomes locked with the output shaft 20, which, when rotating, will therefore have the same rotational speed. By disconnecting the fifth cogwheel 92 of the fifth gear pair 21 from the countershaft 18, torque from the second planetary gear 12 may be transferred to the countershaft 18, and further along from the countershaft 18 to the first main shaft 34, connected with the first planetary gear 10, in order finally to transfer torque via the coupling mechanism 96 to the output shaft 20 of the gearbox 2.

During operation, the gearbox 2 may in some operating modes operate so that one of the sun wheels 26 and 32, respectively, are locked with the first and the second planetary wheel carrier 50 and 51, respectively, with the help of the first and the second coupling device 56 and

58, respectively. The first and the second main shaft 34 and 36, respectively, then obtain the same rotational speed as the input shaft 8 of the gearbox 2, depending on which sun wheel 26 and 32, respectively, is locked with the respective planetary wheel carriers 50 and 51. One or both of the electrical machines 14 and 16, respectively, may operate as a generator to generate electric power to the energy storage device 46. Alternatively, the electrical machine 14 and 16, respectively, may provide a torque injection, in order to thus increase the torque in the output shaft 20. At some operating times, the electrical machines 14 and 16, respectively, will supply each other with electric power, independently of the energy storage device 46.

It is also possible that both the first and the second electrical machine 14 and 16, respectively, generate power to the energy storage device 46. At engine braking the driver releases the accelerator pedal (not displayed) of the vehicle 1. The output shaft 20 of the gearbox 2 then operates one or both electrical machines 14 and 16, respectively, while the combustion engine 4 and the electrical machines 14 and 16, respectively, engine brake. The electrical machines 14 and 16, respectively, in this case generate electric power, which is stored in the energy storage device 46 in the vehicle 1. This operating state is referred to as regenerative braking. In order to further reinforce the effect of deceleration, the output shaft 97 of the combustion engine 4 may be locked and therefore prevented from rotating. Thus, only one of or both the electrical machines 14 and 16, respectively, will function as brakes and 16 generate electric power, which is stored in the energy storage device 46. The locking of the output shaft 97 of the combustion engine 4 may also be carried out when the vehicle must accelerate by only one or both the electrical machines 14 and 16, respectively. If the torque of one or both of the respective electrical machines 14 and 16 overcomes the torque off the combustion engine 4, and having regard to the gearing between them, the combustion engine 4 will not be able to resist the large torque which the respective electrical machines 14 and 16 generate, so that it becomes necessary to lock the output shaft 97 of the combustion engine’s 4. The locking of the output shaft 97 of the combustion engine 4 is preferably carried out with a locking device 102, which is arranged between the first planetary wheel carrier 50 and the gear hosing 42. By locking the first planetary wheel carrier 50 and the gear housing 42, the output shaft 97 of the combustion engine 4 will also be locked, since the output shaft 97 of the combustion engines 4 is connected with the first planetary wheel carrier 50 via the input shaft 8 of the gearbox. The locking device 102 preferably comprises a splines-equipped eighth coupling sleeve 104, which is axially shiftable on a splines-equipped section of the first planetary wheel carrier 50, and on a splines-equipped section of the gear housing. By shifting the eight coupling sleeve

104 so that the splines-equipped sections are connected via the coupling sleeve 104, the first planetary wheel carrier 50, and therefore the output shaft 97 of the combustion engine 4 is prevented from rotating.

The control device 48 is connected to the electrical machines 14 and 16, respectively, to control the respective electrical machines 14 and 16, so that they, during certain operating times, use stored electric power to supply driving power to the output shaft 20 of the gearbox 2, and during other operating times use the kinetic energy of the output shaft 20 of the gearbox 2 to extract and store electric power. The control device 48 thus detects the rotational speed and/or the torque of the output shaft 97 of the combustion engine 4 via sensors 98 arranged at the electrical machines 14 and 16, respectively, and in the output shaft 20 of the gearbox 2, in order thus to gather information and to control the electrical machines 14 and 16, respectively, to operate as electric motors or generators. The control device 48 may be a computer with software suitable for this purpose. The control device 48 also controls the flow of power between the energy storage device 46 and the respective stators 40 and 44 of the electrical machines 14 and 16, respectively. At times when the electrical machines 14 and 16, respectively, operate as engines, stored electric power is supplied from the energy storage device 46 to the respective stators 40 and 44. At times when the electrical machines 14 and 16 operate as generators electric power is supplied from the respective stators 40 and 44 to the energy storage device 46. However, as stated above, the electrical machines 14 and 16, respectively, may, during certain operating times, supply each other with electric power, independently of the energy storage device 46.

The first and the second coupling devices 56 and 58, respectively, the first, second, third, fourth and fifth coupling elements 84, 86, 88, 90 and 93, respectively, the coupling mechanism 96 between the first main shaft 34 and the output shaft 20, and the locking device 102 between the first planetary wheel carrier 50 and the gear housing 42, are connected to the control device 48 via their respective coupling sleeves. These components are preferably activated and deactivated by electric signals from the control device 48. The coupling sleeves are preferably shifted by non-displayed power elements, such as hydraulically or pneumatically operated cylinders. It is also possible to shift the coupling sleeves with electrically powered power elements.

The example embodiment in Fig. 2 shows four pinion gears 62, 68, 74 and 80, respectively, and four cogwheels 64, 70, 76 and 82, respectively, and two respective planetary gears 10 and 12, , with associated electrical machines 14 and 16, respectively. However, it is possible to adapt the gearbox 2 with more or fewer pinion gears and cogwheels, and with more planetary gears with associated electrical machines.

Below, an up-shift from a first to a seventh gear will be described, wherein the gearbox 2 is arranged in a vehicle 1 and the vehicle is propelled by the combustion engine 4.

The input shaft 8 of the gearbox 2 is connected to the output shaft 97 of the vehicle's 1 combustion engine 4. The output shaft 20 of the gearbox 2 is connected to a driving shaft 99 in the vehicle 1. At idling of the combustion engine 4 and when the vehicle 1 is at a standstill, the input shaft 8 of the gearbox 2 rotates at the same time as the output shaft 20 of the gearbox 2 is at a standstill. The locking device 102 is deactivated, so that the output shaft 97 of the combustion engine 4 may rotate freely. Since the input shaft 8 of the gearbox 2 rotates, the first planetary wheel carrier 50 will also rotate, which entails that the first set of planetary wheels 52 will rotate. Since the first planetary wheel carrier 50 is connected to the second sun wheel 32, the second sun wheel 32, and thus also the second set of planetary wheels 54, will rotate. By not supplying power to the first and the second electrical machines 14 and 16, respectively, the first and the second internal rings 22 and 28, respectively, which are connected with the respective first and second rotor 24 and 30 of the electrical machines 14 and 16, respectively, will rotate freely, so that no torque is absorbed by the respective internal rings 22 and 28. The first and the second coupling devices 56 and 58, respectively, are disconnected and thus not actuated. Thus, no torque will be transferred from the combustion engine 4 to the sun wheel 26 of the first planetary gear 10 or to the planetary wheel carrier 51 of the second planetary gear 12. The coupling mechanism 96 between the first main shaft 34 and the output shaft 20 is disconnected, so that the first main shaft 34 and the output shaft 20 may rotate freely in relation to each other. Since the first planetary gear's sun wheel 26, the planetary wheel carrier 51 of the second planetary gear 12 and the output shaft 20 of the gearbox 2 are, at this stage, at a standstill, the countershaft 18 is also at a standstill. In a first step the fourth cogwheel 82 and the third cogwheel 76 are connected with the countershaft 18 with the assistance of the fourth and third coupling elements 90 and 88, respectively. The first cogwheel 64 and the second cogwheel 70 are disconnected from the countershaft 18. Thus, the first cogwheel 64 and the second cogwheel 70 are allowed to rotate freely in relation to the countershaft 18. The fifth cogwheel 92 of the fifth gear pair 21 is locked on the countershaft 18 with the assistance of the fifth coupling element 93.

In order to start the rotation of the output shaft 20 of the gearbox 2, with the objective of driving the vehicle 1, the fourth pinion gear 80 and the fourth cogwheel 82 on the countershaft 18 must be brought to rotate. This is achieved by making the second planetary wheel carrier 51 rotate. When the second planetary wheel carrier 51 rotates, the second main shaft 36 will also rotate and thus the fourth pinion gear 80, which is arranged on the second main shaft 36, also rotates. The second planetary wheel carrier 51 is made to rotate by controlling the second internal ring gear 28 with the second electrical machine 16. By activating the second electrical machine 16 and controlling the combustion engine 4 to a suitable engine speed, the vehicle 1 begins to move as the second main shaft 36 begins to rotate. When the second planetary wheel carrier 51 and the second sun wheel 32 achieve the same rotational speed, the second sun wheel 32 is locked with the second planetary wheel carrier 51 with the assistance of the second coupling device 58. As mentioned above, the second coupling device 58 is preferably adapted in such a way that the second sun wheel 32 and the second planetary wheel carrier 51 engage mechanically with each other. Alternatively, the second coupling device 58 may be adapted as a slip brake or a multi-plate clutch which connects, in a smooth way, the second sun wheel 32 with the second planetary wheel carrier 51. When the second sun wheel 32 is connected with the second planetary wheel carrier 51, the second planetary wheel carrier 51 will rotate with the same rotational speed as the output shaft 97 of the combustion engine 4. Thus, the torque generated by the combustion engine 4 is transferred to the output shaft 20 of the gearbox 2 via the fourth pinion gear 80, the fourth cogwheel 82 on the countershaft 18, the fifth cogwheel 92 on the countershaft 18, and the sixth cogwheel 94 on the output shaft 20 of the gearbox 2. The vehicle 1 will thus begin to move off and be propelled by a first gear.

Each of the first, second, third and fourth gear pairs 60, 66, 72, 78 has a gearing, which is adapted to the vehicle’s 1 desired driving characteristics. According to the example embodiment displayed in Fig. 2, the fourth gear pair 78 has the highest gearing compared to the first, second and third gear pairs 60, 66, 72, which results in the fourth gear pair 78 being connected when the lowest gear is engaged. The second gear pair 66 transfers, as does the fourth gear pair 78, torque between the second main shaft 36 and the countershaft 18, and could instead be fitted out with the highest gearing, compared with other gear pairs 60, 72, 78, which is why in such an embodiment the second gear pair 66 could be connected when the lowest gear is engaged.

When the countershaft 18 is made to rotate by the fourth cogwheel 82 on the countershaft 18, the third cogwheel 76 on the countershaft 18 will also rotate. Thus, the countershaft 18 operates the third cogwheel 76, which in turn operates the third pinion gear 74 on the first main shaft 34. When the first main shaft 34 rotates, the first sun wheel 26 will also rotate, and thus, depending on the rotational speed of the output shaft 97 of the combustion engine 4 and thus the rotational speed of the first planetary wheel carrier 50, it will cause the first internal ring gear 22 and the first rotor 24 of the first electrical machine 14 to rotate. It is thus possible to allow the first electrical machine 14 to operate as a generator to supply power to the energy storage device 46, and/or to supply power to the second electrical machine 16. It is also possible for the second electrical machine 16 to be operated as a generator. Alternatively, the first electrical machine 14 may emit a torque injection, by way of the control device 48 controlling the first electrical machine 14 to provide a driving torque.

In order to shift gears from the first to the second gear, the locking between the second sun wheel 32 and the second planetary wheel carrier 51 must cease, which is achieved by way of the first and/or the second electrical machine 14, 16 being controlled in such a way that torque balance prevails in the second planetary gear 12. Subsequently, the second coupling device 58 is controlled, so that it disconnects the second sun wheel 32 and the second planetary wheel carrier 51 from each other. The second gear is connected, by way of the control device 48 controlling the combustion engine 4, so that a synchronous rotational speed arises between the first planetary wheel carrier 50 and the first sun wheel 26, in order to achieve a locking between the first planetary wheel carrier 50 and the first sun wheel 26. This is achieved by way of controlling the first coupling device 56 in such a way that the first planetary wheel carrier 50 and the first sun wheel 26 are mechanically connected with each other. Alternatively, the first coupling device 56 may be adapted as a slip brake or a multi-plate clutch which connects, in a smooth way, the first sun wheel 26 with the first planetary wheel carrier 50. By synchronising the control of the combustion engine 4 and the second and first electrical machine 14 and 16, respectively, a soft and disruption-free transition from a first to a second gear may be carried out.

The first main shaft 34 now rotates, operated by the output shaft 97 of the combustion engine 4, and the first main shaft 34 now operates the third pinion gear 74. The first planetary wheel carrier 50 thus operates the third pinion gear 74 via the first sun wheel 26 and the first main shaft 34. Since the third cogwheel 76 is in engagement with the third pinion gear 74 and is connected with the countershaft 18, the third cogwheel 76 will operate the countershaft 18, which in turn operates the fifth cogwheel 92 on the countershaft 18. The fifth cogwheel 92 in turn operates the output shaft 20 of the gearbox 2 via the sixth cogwheel 94, which is arranged on the output shaft 20 of the gearbox 2. The vehicle 1 is now operated with a second gear.

When the countershaft 18 is made to rotate by the third cogwheel 76, the fourth cogwheel 82 will also rotate. Thus, the countershaft 18 operates the fourth cogwheel 82, which in turn operates the fourth pinion gear 80 on the second main shaft 36. When the second main shaft 36 rotates, the second planetary wheel carrier 51 will also rotate, and thus, depending on the rotational speed of the output shaft 97 of the combustion engine 4, and thus the rotational speed in the first planetary wheel carrier 50, it will cause the second internal ring gear 28 and the second rotor 30 of the second electrical machine 16 to rotate. It is thus possible to allow the second electrical machine 16 to operate as a generator to supply power to the energy storage device 46, and/or to supply power to the first electrical machine 14. The second electrical machine 16 may also emit a torque injection, by way of the control device 48 controlling the second electrical machine 16 to provide a propulsion torque.

In order to shift from a second gear to a third gear, the fourth cogwheel 82 on the countershaft 18 must be disconnected from the countershaft 18 with the fourth coupling element 90, so that the fourth cogwheel 82 may rotate freely in relation to the countershaft 18. Subsequently, the countershaft 18 is connected with the second cogwheel 70 on the countershaft 18 via the second coupling element 86. In order to achieve a connection of the countershaft 18 and the second cogwheel 70 on the countershaft 18, preferably the second electrical machine 16 is controlled in such a way that a synchronous rotational speed arises between the countershaft 18 and the second cogwheel 70 on the countershaft 18. A synchronous rotational speed may be determined by way of measuring the rotational speed of the second rotor 30 in the second electrical machine 16, and by measuring the rotational speed of the output shaft 20. Thus, the rotational speed in the second main shaft 36 and the rotational speed in the countershaft 18 may be determined by way of given gear ratios. The rotational speed of the respective shafts 18, 36 is controlled, and when a synchronous rotational speed has arisen between the countershaft 18 and the second cogwheel 70, the countershaft 18 and the second cogwheel 70 are connected with the assistance of the second coupling element 86.

In order to complete the shift from a second gear to a third gear, the locking between the first sun wheel 26 and the first planetary wheel carrier 50 must cease, which is achieved by way of the first and/or the second electrical machine 14, 16 being controlled in such a way that torque balance is achieved in the first planetary gear 10, following which the first coupling device 56 is controlled, so that it releases the first sun wheel 26 and the first planetary wheel carrier 50 from each other. Subsequently, the combustion engine 4 is controlled in such a way that a synchronous rotational speed arises between the second sun wheel 32 and the second planetary wheel carrier 51, so that the second coupling device 58 may be engaged in order thus to connect the second sun wheel 32 with the second planetary wheel carrier 51, via the coupling sleeve 57. By synchronising the control of the combustion engine 4 and the second and first electrical machine 14 and 16, respectively, a soft and disruption-free transition from a second to a third gear may be carried out.

The third cogwheel 76 is disconnected by controlling the first electrical machine 14 in such a way that a substantially zero torque state arises between the countershaft 18 and the third cogwheel 76. When a substantially zero torque state arises, the third cogwheel 76 is disconnected from the countershaft 18 by controlling the third coupling element 88, so that it releases the third cogwheel 76 from the countershaft 18. Subsequently, the first electrical machine 14 is controlled in such a way that a synchronous rotational speed arises between the countershaft 18 and the first cogwheel 64. When a synchronous rotational speed arises, the first cogwheel 64 is connected to the countershaft 18 by way of controlling the first coupling element 84, so that it connects the first cogwheel 64 on the countershaft 18. A synchronous rotational speed may be determined, since the rotational speed of the first rotor 24 in the first electrical machine 14 is measured and the rotational speed of the output shaft 20 is measured, following which the rotational speeds of the shafts 18, 34 are controlled in such a way that a synchronous engine speed arises. Thus, the rotational speed of the first main shaft 34 and the rotational speed of the countershaft 18 may be determined by way of given gear ratios.

The second main shaft 36 now rotates with the same rotational speed as the output shaft 97 of the combustion engine 4, and the second main shaft 36 now operates the second pinion gear 68 via the second main shaft 36. Since the second cogwheel 70 is in engagement with the second pinion gear 68 and is connected with the countershaft 18, the second cogwheel 70 will operate the countershaft 18, which in turn operates the fifth cogwheel 92 on the countershaft 18. The fifth cogwheel 92 in turn operates the output shaft 20 of the gearbox 2 via the sixth cogwheel 94, which is arranged on the output shaft 20 of the gearbox 2. The vehicle 1 is now driven in a third gear.

When the countershaft 18 is made to rotate by the second cogwheel 70 on the countershaft 18, the first cogwheel 64 on the countershaft 18 will also rotate. Thus, the countershaft 18 operates the first cogwheel 64, which in turn operates the first pinion gear 62 on the first main shaft 34. When the first main shaft 34 rotates, the first sun wheel 26 will also rotate, and thus, depending on the rotational speed of the output shaft 97 of the combustion engine 4, and thus the rotational speed of the first planetary wheel carrier 50, it will cause the first internal ring gear 22 and the first rotor 24 of the second electrical machine 16 to rotate. It is thus possible to allow the first electrical machine 14 operate as a generator to supply power to the energy storage device 46, and/or to supply power to the second electrical machine 16. Alternatively, the first electrical machine 14 may emit a torque injection, by way of the control device 48 controlling the first electrical machine 14 to provide a driving torque.

In order to complete a shift of gears from the third to the fourth gear, the locking between the second sun wheel 32 and the second planetary wheel carrier 51 must cease, which is achieved by way of the first and/or the second electrical machine 14, 16 being controlled in such a way that torque balance prevails in the second planetary gear 12, following which the second coupling device 58 is controlled so that it releases the second sun wheel 32 and the second planetary wheel carrier 51 from each other. A fourth gear is subsequently connected by way of the control device 48 controlling the combustion engine 4, in such a way that a synchronous rotational speed arises between the first planetary wheel carrier 50 and the first sun wheel 26, in order to achieve a locking between the first planetary wheel carrier 50 and the first sun wheel 26. This is achieved by way of controlling the first coupling device 56 in such a way that the first planetary wheel carrier 50 and the first sun wheel 26 are mechanically connected with each other. By synchronising the control of the combustion engine 4 and the second and first electrical machine 14 and 16 a soft and disruption-free transition from a third to a fourth gear may be carried out.

The first main shaft 34 now rotates and is operated by the output shaft 97 of the combustion engine 4 and the first main shaft 34 now operates the first pinion gear 62. The first planetary wheel carrier 50 thus operates the first pinion gear 62 via the first sun wheel 26 and the first main shaft 34. Since the first cogwheel 64 is in engagement with the first pinion gear 62 and is connected with the countershaft 18, the first cogwheel 64 will operate the countershaft 18, which in turn operates the fifth cogwheel 92 on the countershaft 18. The fifth cogwheel 92 in turn operates the output shaft 20 of the gearbox 2 via the sixth cogwheel 94, which is arranged on the output shaft 20 of the gearbox 2. The vehicle 1 is now driven in a fourth gear.

When the countershaft 18 is made to rotate by the first cogwheel 64, the second cogwheel 70 will also rotate. Thus, the countershaft 18 operates the second cogwheel 70, which in turn operates the second pinion gear 68 on the second main shaft 36. When the second main shaft 36 rotates, the second planetary wheel carrier 51 will also rotate, and thus, depending on the rotational speed of the output shaft 97 of the combustion engine 4, and thus the rotational speed in the first planetary wheel carrier 50, it will cause the second internal ring gear 28 and the second rotor 30 of the second electrical machine 16 to rotate. It is thus possible to allow the second electrical machine 16 to operate as a generator to supply power to the energy storage device 46, and/or to supply power to the first electrical machine 14. The second electrical machine 16 may also emit a torque injection, by way of the control device 48 controlling the second electrical machine 16 to provide a propulsion torque.

In order to shift gears from a fourth gear to a fifth gear, the first cogwheel 64 must be disengaged from the countershaft 18, so that the fourth gear is disengaged. This is achieved by way of controlling the combustion engine 4 and the first electrical machine 14, in such a way that the first cogwheel 64 is brought to a substantially zero torque state in relation to the countershaft 18. When a substantially zero torque state has arisen, the first coupling element 84 is disengaged, so that the first cogwheel 64 is disconnected from the countershaft 18.

Subsequently, the rotational speed of the first main shaft 34 is synchronised with the rotational speed of the output shaft 20, following which the coupling mechanism 96 is controlled in such a way that it connects the first main shaft 34 with the output shaft 20.

Subsequently, the combustion engine 4 and the first electrical machine 14 are controlled in such a way that the propulsion torque occurs via the first main shaft 34 and via the coupling mechanism 96, and further along to the output shaft 20. By reducing the torque from the second electrical machine 16, the fifth coupling element 93 may be brought to a substantially zero torque state in relation to the countershaft 18. When a substantially zero torque state has arisen, the fifth coupling element 93 is disengaged, so that the fifth cogwheel 92 of the fifth gear pair 21 is disconnected from the countershaft 18.

Subsequently, with the help of the second electrical machine 16, the rotational speed of the countershaft 18 is synchronised with the rotational speed of the third cogwheel 76, following which the third coupling element 88 is controlled in such a way that it connects the third cogwheel 76 with the countershaft 18. When this connection has been completed, the propulsion torque may be shared between the combustion engine 4, the first electrical machine 14 and the second electrical machine 16. Subsequently, torque balance is created in the first planetary gear 10, following which the first coupling device 56 disconnects the first planetary wheel carrier 50 and the first sun wheel 26 from each other. Finally, the second planetary wheel carrier 51 is rotational speed synchronised with the second sun wheel 32, following which the second coupling device 58 connects the second planetary wheel carrier 51 and the second sun wheel 32 with each other.

The second main shaft 36 now rotates, operated by the output shaft 97 of the combustion engine 4, and the second main shaft 36 operates the second pinion gear 68. Since the second cogwheel 70 is in engagement with the second pinion gear 68 and is connected with the countershaft 18 via the second coupling element 86, the second cogwheel 70 will operate the countershaft 18, which in turn operates the third cogwheel 76 on the countershaft 18. The third cogwheel 76 in turn operates the first main shaft 34 via the third pinion gear 74, and the output shaft 20 of the gearbox 2 is thus operated via the coupling mechanism 96, which connects the first main shaft 34 and the output shaft 20 of the gearbox 2. The vehicle 1 is now driven in a fifth gear.

In order to shift gears from the fifth to the sixth gear, the locking between the second sun wheel 32 and the second planetary wheel carrier 51 must cease, which is achieved by way of the first and/or the second electrical machine 14, 16 being controlled in such a way that torque balance is achieved in the second planetary gear 12, following which the second coupling device 58 is controlled so that it releases the second sun wheel 32 and the second planetary wheel carrier 51 from each other. A sixth gear is subsequently connected by way of the control device 48 controlling the combustion engine 4, in such a way that a synchronous engine speed arises between the first planetary wheel carrier 50 and the first sun wheel 26, in order to achieve a locking between the first planetary wheel carrier 50 and the first sun wheel 26. This is achieved by way of controlling the first coupling device 56 in such a way that the first planetary wheel carrier 50 and the first sun wheel 26 are mechanically connected with each other. By synchronising the control of the combustion engine 4 and the second and first electrical machine 14 and 16, respectively, a soft and disruption-free transition from a fifth to a sixth gear may be carried out.

The first main shaft 34 now rotates operated by the output shaft 97 of the combustion engine 4, whereat the first main shaft 34 operates the output shaft 20 of the gearbox 2 via the coupling mechanism 96, which connects the first main shaft 34 and the output shaft 20 of the gearbox 2. The vehicle 1 is now driven in a sixth gear.

In order to shift from a sixth to a seventh gear, the third cogwheel 76 on the countershaft 18 must first be disconnected from the countershaft 18 with the third coupling element 88, so that the third cogwheel 76 may rotate freely in relation to the countershaft 18. Subsequently, the countershaft 18 is connected with the first cogwheel 64 on the countershaft 18 via the first coupling element 84. When the countershaft 18 and the first cogwheel 64 on the countershaft 18 have a synchronous rotational speed, the first coupling element 84 is controlled in such a way that the first cogwheel 64 and the countershaft 18 are connected.

In order to complete the shift from a sixth gear to a seventh gear, the locking between the first sun wheel 26 and the first planetary wheel carrier 50 must cease, which is achieved by way of the first and/or the second electrical machine 14, 16 being controlled in such a way that torque balance is achieved in the first planetary gear 10, following which the first coupling device 56 is controlled, so that it releases the first sun wheel 26 and the first planetary wheel carrier 50 from each other. Subsequently, the combustion engine 4 is controlled in such a way that a synchronous rotational speed arises between the second sun wheel 32 and the second planetary wheel carrier 51, so that the second coupling device 58 may be engaged in order thus to connect the second sun wheel 32 with the second planetary wheel carrier 51, via the coupling sleeve 57. By synchronising the control of the combustion engine 4 and the second and first electrical machine 14 and 16, respectively, a soft and disruption-free transition from a sixth to a seventh gear may be carried out.

The second main shaft 36 now rotates with the same rotational speed as the output shaft 97 of the combustion engine 4, and the second main shaft 36 operates the second pinion gear 68. Since the second cogwheel 70 is in engagement with the second pinion gear 68 and is connected with the countershaft 18, the second cogwheel 70 will operate the countershaft 18, which in turn operates the fifth cogwheel 64 on the countershaft 18. The first cogwheel 64 in turn operates the first main shaft 34 via the first pinion gear 62, and the output shaft 20 of the gearbox 2 is thus operated via the coupling mechanism 96, which connects the first main shaft 34 and the output shaft 20 of the gearbox 2. The vehicle 1 is now driven in a seventh gear.

According to the embodiment above, the gearbox 2 comprises pinion gears 62, 68, 74, 80 and cogwheels 64, 70, 76, 82 arranged on the main shafts 34, 36 and the countershaft 18, respectively, to transfer rotational speed and torque. However, it is possible to use another type of transmission, such as chain and belt drives, to transfer rotational speed and torque in the gearbox 2.

The transmission device 19 has four gear pairs 60, 66, 72, 78 according to the example embodiment. However, the transmission device 19 may comprise any number of gear pairs.

As described, torque is extracted from the gearbox 2 from the output shaft 20. It is also possible to extract torque directly from the first or second main shaft 34, 36, or directly from the countershaft 18. Torque may also be extracted in parallel from two or all of the three shafts 18, 34, 36 simultaneously.

Fig. 3 illustrates the hybrid powertrain 3 according to Fig. 2 in a simplified view, where some components have been excluded in the interest of clarity. G1 in Fig. 3 consists of at least one gear pair connected with the first main shaft 34 and therefore with the first planetary gear 10, and a gear pair G2 consists of at least one gear pair connected with the second main shaft 36 and therefore with the second planetary gear 12. These gear pairs G1, G2 are also connected to the output shaft 20. These gear pairs G1, G2 are suitably connected to the output shaft 20 via the countershaft 18. G1 and G2, respectively, may consist of one or several gear pairs. The gear pair G1, connected with the first planetary gear 10, may for example consist of the first gear pair 60 and/or the third gear pair 72, as described in Fig. 2. The gear pair G2, connected with the second planetary gear 12, may for example consist of the second gear pair 66 and/or the fourth gear pair 78, as described in Fig. 2. Further, at least one gear pair G3, connected with the output shaft 20 and the countershaft, 18 is displayed, which may consist of the fifth gear pair 21 described in Fig. 2. G3 may consist of one or several gear pairs. Alternatively, torque may be extracted directly from the countershaft 18, which thus constitutes the output shaft.

In the following, an embodiment is described to control the hybrid powertrain 3 to optimise the torque from a combustion engine 4 arranged in the hybrid powertrain 3. The hybrid powertrain 3 comprises a combustion engine 4; a gearbox 2 with an input shaft 8 and an output shaft 20; a first planetary gear 10, connected to the input shaft 8 and a first main shaft 34; a second planetary gear 12, connected to the first planetary gear 10 and a second main shaft 36; a first electrical machine 14, connected to the first planetary gear 10; a second electrical machine 16, connected to the second planetary gear 12; at least one gear pair G1 connected with the first planetary gear 10 and the output shaft 20; and at least one gear pair G2 connected with the second planetary gear 12 and the output shaft 20, wherein the combustion engine 4 is connected with a first planetary wheel carrier 50, arranged in the first planetary gear 10 via the input shaft 8 of the gearbox 2, and wherein the second main shaft 36 is connected with a second planetary wheel carrier 51, arranged in the second planetary gear 12.

A gear pair G1, connected with the first planetary gear 10, is connected with a countershaft 18, which is connected to the output shaft 20, and a gear pair G2, connected with the second planetary gear 12, is also connected to the countershaft. Thus, torque may be transferred between the first and the second planetary gears 10, 12, via the countershaft 18, to the output shaft 20. Depending on which gear is engaged, two moveable parts in the first planetary gear 10 are connected, for example a first planetary wheel carrier 50 and a first sun wheel 26, or two moveable parts in the second planetary gear 12 are connected, for example a second planetary wheel carrier 51 and a second sun wheel 32.

Where a higher torque is required from the combustion engine 4, than what is suitable for the current engine speed in the combustion engine 4, the moveable parts of the first and second planetary gears 10, 12, respectively, are disconnected from each other. The first planetary wheel carrier 50, arranged in the first planetary gear 10, and the first sun wheel 26 are thus disconnected from each other, or the second planetary wheel carrier 51 and the second sun wheel 32 are disconnected from each other, depending on which are connected. The disconnection relating to the first planetary wheel carrier and the first sun wheel is achieved by way of the first and/or the second electrical machine 14, 16 being controlled in such a way that torque balance is achieved in the first planetary gear 10, following which the first coupling device 56 is shifted, so that the first planetary wheel carrier 50 and the first sun wheel 26 are disconnected from each other. The disconnection relating to the second planetary wheel carrier 51 and the second sun wheel 32 is achieved by way of the first and/or the second electrical machine 14, 16 being controlled in such a way that torque balance is achieved in the second planetary gear 12, following which the second coupling device 58 is shifted so that the second planetary wheel carrier 51 and the second sun wheel 32 are disconnected from each other. At the disconnection, the first and the second electrical machines 14, 16 are controlled in such a way that a desired torque T_{Drv} is achieved in the output shaft 20. Subsequently, the combustion engine 4 is controlled to a predetermined engine speed n_{ice}, which is suitable to achieve a desired driving power from the combustion engine 4. Furthermore, the first and the second electrical machines 14, 16 are controlled in such a way that the desired torque T_{Drv} in the output shaft 20 is achieved, while simultaneously the requested total power consumption of the first and the second electrical machines 14, 16, P_{EM} is achieved. The desired torque T_{Drv} in the output shaft 20, which may consist of a torque requested by the vehicle's driver, is suitably determined based on the position of the accelerator pedal according to prevalent methods. Alternatively, the desired torque T_{Drv} is a torque requested by one of the vehicle's control systems. The desired torque T_{Drv} is determined in the control device 48. Below is a description of how, with known values for among others desired torque T_{Drv}, and a total power consumption requested by the first and second electrical machines 14, 16, P_{EM}, it is possible to derive how the first and the second electrical machines 14, 16 should be controlled. Further, it is described how the predetermined engine speed n_{ice} may be determined, based on a known desired driving power from the combustion engine. Control of the hybrid powertrain 3 and its component parts to optimise the torque from the combustion engine 4 arranged in the hybrid powertrain 3 is suitably carried out by the control device 48.

In cases where the gear pair G3, connected with the countershaft 18 and the output shaft 20, is connected and locked on the countershaft 18, and a coupling mechanism S6, 96, arranged between the first main shaft 34 and the output shaft 20, is open, the torque T_{drv} desired in the output shaft 20 of the gearbox, also referred to as the requested powertrain torque, may be obtained through a combination of torque from the first and the second electrical machines 14, 16, according to the equation E1 below:

$$T_{Drv} = -T_{EM1}\frac{S_1}{R_1}\frac{1}{G_1G_3} + T_{EM2}\frac{S_2+R_2}{R_2}\frac{1}{G_2G_3}$$ [E1]

where T_{EM1} is the torque emitted by the first electrical machine 14, and T_{EM2} is the torque emitted by the second electrical machine 16, S_1 is the number of cogs on the first sun wheel 26, R_1 is the number of cogs on the first internal ring gear 22, S_2 is the number of cogs on the second sun wheel, and R_2 is the number of cogs on the second internal ring gear 28. G_1 is the gear ratio between the first main shaft 34 and the countershaft 18, G_2 is the gear ratio between the second main shaft 36 and the countershaft 18, and G_3 is the gear ratio between the countershaft 18 and the output shaft 20, for the selected connected gear pairs.

In cases where the gear pair G3, connected with the countershaft 18 and the output shaft 20, is disconnected from the countershaft 18 and the coupling mechanism S6, 96 is locked, and thus connects the first main shaft 34 and the output shaft 20, the torque T_{drv} in the output shaft 20 of the gearbox may be obtained from the equation E1' below:

$$T_{Drv} = -T_{EM1}\frac{S_1}{R_1} + T_{EM2}\frac{S_2+R_2}{R_2}\frac{G_1}{G_2}$$ [E1']

The power consumption of the first and the second electrical machines 14, 16 P_{EM} is obtained from the equation E2 below:

$$P_{EM} = (T_{EM1}n_{EM1} + T_{EM2}n_{EM2})\frac{2\pi}{60}$$ [E2]

where n_{EM1} is the first electrical machine's 14 engine speed and n_{EM2} is the second electrical machine's 16 engine speed. The desired total power consumption P_{EM} is suitably a predetermined parameter. With a specified desired power consumption P_{EM}, and a known desired torque T_{Drv} in the output shaft 20, the two equations E1, E2 or alternatively E1', E2' may be solved to thus determine what torque the first and the second electrical machine 14, 16, respectively, must be controlled to achieve.

The power P_{ice} produced by the combustion engine 4 may be obtained from the third equation E3. Where P_{Drv} is the power of the hybrid powertrain, which is obtained as a product of the torque T_{Drv} of the output shaft 20 and the engine speed of the gearbox 2. The combustion engine's 4 produced power P_{ice} may also be obtained from the equation E4, where T_{ice} is the torque of the combustion engine 4, and n_{ice} is the engine speed of the combustion engine. The combustion engine's 4 torque T_{ice} may be obtained from the equation E5 below.

$$P_{ice} = P_{Drv} + P_{EM} \quad \text{[E3]}$$

$$P_{ice} = T_{ice} \times n_{ice} \times \frac{2\pi}{60} \quad \text{[E4]}$$

$$T_{ice} = -\frac{(S_1 + R_1)T_{EM1}}{R_1} + \frac{S_2 T_{EM2}}{R_2} \quad \text{[E5]}$$

In order to achieve a given power P_{ice} from the combustion engine 4, the torque and engine speed of the combustion engine 4, and from the first and second electrical machines 14, 16 may be varied in different combinations, as is found based on the equations E3 and E4 in combination with E1, E2 and E5. One combination is more favourable than the others, and the predetermined engine speed n_{ice}, towards which the combustion engine 4 is controlled, is the most favourable engine speed n_{ice}, given the power P_{ice}, which is desired to be produced, given the known desired torque T_{Drv} of the output shaft 20 and the determined desired total power consumption P_{EM}. The predetermined engine speed n_{ice} may thus be obtained from the equation E4.

Fig. 4 shows a flow chart relating to a method to control a hybrid powertrain 3 to optimise the torque from a combustion engine 4 arranged in the hybrid powertrain 3. The hybrid powertrain 3 also comprises a gearbox 2 with an input shaft 8 and an output shaft 20; a first planetary gear 10, connected to the input shaft 8 and a first main shaft 34; a second planetary gear 12, connected to the first planetary gear 10 and a second main shaft 36; a first electrical machine 14, connected to the first planetary gear 10; a second electrical machine 16, connected to the second planetary gear 12; at least one gear pair G1, 60, 72 connected with the first planetary gear 10 and the output shaft 20; and at least one gear pair G2, 66, 78 connected with the second planetary gear 12 and the output shaft 20, wherein the combustion engine 4 is connected with a first planetary wheel carrier 50, arranged in the first planetary gear 10 via the input shaft 8 of the gearbox 2, and wherein the second main shaft 36 is connected with a planetary wheel carrier 51, arranged in the second planetary gear 12. A gear pair G1, 60, 72, connected with the first planetary gear 10, is connected with a countershaft 18, which is connected to the output shaft 20, and a gear pair G2, 66, 78, connected with the second planetary gear 12, is also connected to the countershaft. Thus, torque may be transferred between the first and the second planetary gears 10, 12, via the countershaft 18, to the output shaft 20. Depending on which gear is engaged, two moveable parts in the first planetary gear 10 are connected, for example a first planetary wheel carrier 50 and a first sun wheel 26, or two moveable parts in the second planetary gear 12 are connected, for example a second planetary wheel carrier 51 and a second sun wheel 32.

The method comprises the steps:
a) ensuring that the moveable parts 22 of the first planetary gear 10, 26, 50 are disconnected from each other, and that the moveable parts 28, 32, 51 of the second planetary gear 12 are disconnected from each other;
b) to control the combustion engine 4 to a predetermined engine speed n_{ice}; and
c) to control the first and the second electrical machines 14; 16 in such a way that a desired torque T_{Drv} is achieved in the output shaft 20, while simultaneously a requested total power consumption P_{EM}, of the first and the second electrical machines 14, 16, is achieved.

Thus, the combustion engine 4 may be permitted to have a higher engine speed without having to change gears, and the torque from the combustion engine 4 is optimised, while simultaneously the desired torque T_{Drv} on the output shaft 20 is achieved.

The first and the second electrical machines 14, 16 are suitably controlled in step c) in such a way that they emit a torque T_{EM1} and T_{EM2}, which may be determined by way of calculating the equations [1] and [2] or [1'] and [2], as described in Fig. 3.

Suitably, the moveable parts 22, 26, 50 in the first planetary gear are disconnected from each other by way of controlling the first and/or the second electrical machine 14, 16, in such a way that torque balance is achieved in the first planetary gear 10, wherein a first coupling device 56 is shifted, so that a first planetary wheel carrier 50, arranged in the first planetary gear 10, and a first sun wheel 26 are disconnected from each other.

Suitably, the moveable parts 28, 32, 51 arranged in the second planetary gear 12 are disconnected from each other by way of controlling the first and/or the second electrical machine 14, 16, in such a way that torque balance is achieved in the second planetary gear 12, wherein a second coupling device 58 is shifted, so that a second planetary wheel carrier 51, arranged in the second planetary gear 12, and a second sun wheel 32 are disconnected from each other.

Suitably, the first and the second electrical machines 14, 16 are also controlled in step a) in such a way that the desired torque T_{Drv} is achieved in the output shaft 20.

Preferably the predetermined engine speed n_{ice} in the combustion engine 4 is determined based on a desired power P_{ice} from the combustion engine 4 (see Fig. 3). The predetermined engine speed n_{ice} to which the combustion engine 4 is controlled in step b) may preferably be determined through equation [4], as described in Fig. 3.
Preferably, steps b) and c) are carried out in parallel.

According to the invention, a computer program P is provided, stored in the control device 48 and/or the computer 53, which may comprise procedures to control the hybrid powertrain 3 according to the present invention.

The program P may be stored in an executable manner, or in a compressed manner, in a memory M and/or a read/write memory R.

The invention also relates to a computer program product, comprising program code stored in a medium readable by a computer, in order to perform the method steps specified above, when said program code is executed in the control device 48, or in another computer 53 connected to the control device 48. Said program code may be stored in a non-volatile manner on said medium readable by a computer 53.

The components and features specified above may, within the framework of the invention, be combined between different embodiments specified.

Claims

1. Method to control a hybrid powertrain (3) to optimise the torque from a combustion engine 4, arranged in the hybrid powertrain (3), wherein the hybrid powertrain (3) also comprises; a gearbox (2) with an input shaft (8) and an output shaft (20); a first planetary gear (10), connected to the input shaft (8) and a first main shaft (34); a second planetary gear (12), connected to the first planetary gear (10) and a second main shaft (36); a first electrical machine (14), connected to the first planetary gear (10); a second electrical machine (16), connected to the second planetary gear (12); at least one gear pair (G1, 60, 72) connected with the first planetary gear (10) and the output shaft (20); and at least one gear pair (G2, 66, 78) connected with the second planetary gear (12) and the output shaft (20), wherein the combustion engine (4) is connected with a first planetary wheel carrier (50), arranged in the first planetary gear (10) via the input shaft (8) of the gearbox (2), and wherein the second main shaft (36) is connected with a planetary wheel carrier (51), arranged in the second planetary gear (12), **characterised by** the steps:
a) ensuring that the moveable parts (22, 26, 50) of the first planetary gear (10) are disconnected from each other, and that the moveable parts (28, 32, 51) of the second planetary gear (12) are disconnected from each other;
b) to control the combustion engine (4) to a predetermined engine speed (n_{ice}); and
c) to control the first and the second electrical machines (14; 16) in such a way that a desired torque (T_{Drv}) is achieved in the output shaft (20), while simultaneously a requested total power consumption (P_{EM}) of the first and the second electrical machines (14; 16) is achieved.

2. Method according to claim 1, **characterised in that** in step a) the first and/or the second electrical machine (14; 16) is controlled in such a way that torque balance is achieved in the first planetary gear (10), wherein a first coupling device (56) is shifted, so that the first planetary wheel carrier (50), arranged in the first planetary gear (10), and a first sun wheel (26) are disconnected from each other.

3. Method according to claim 1, **characterised in that** in step a) the first and/or the second electrical machine (14; 16) is controlled in such a way that torque balance is achieved in the second planetary gear (12), following which a second coupling device (58) is shifted, so that the second planetary wheel carrier (51), arranged in the second planetary gear (12), and a second sun wheel (32) are disconnected from each other.

4. Method according to any of the previous claims, **characterised in that** in step a) the first and the second electrical machines (14; 16) are also controlled in such a way that the desired torque (T_{Drv}) is achieved in the output shaft (20).

5. Method according to any of the previous claims, **characterised in that** the predetermined engine speed (n_{ice}) of the combustion engine (4) is determined based on a desired output (P_{ice}) from the combustion engine (4).

6. Method according to any of the previous claims, **characterised by** the steps b) and c) being carried out in parallel.

7. Vehicle with a hybrid powertrain (3), comprising a combustion engine (4); a gearbox (2) with an input shaft (8) and an output shaft (20); a first planetary gear (10), connected to the input shaft (8) and a first main shaft (34); a second planetary gear (12), connected to the first planetary gear (10) and a second main shaft (36); a first electrical machine (14), connected to the first planetary gear (10); a second electrical machine (16), connected to the second planetary gear (12); at least one gear pair (G1, 60, 72) connected with the first planetary gear (10) and the output shaft (20); and at least one gear pair (G2, 66, 78) connected with the second planetary gear (12) and the output shaft (20), wherein the combustion engine (4) is connected with a first planetary wheel carrier (50), arranged in the first planetary gear (10) via the input shaft (8) of the gearbox (2), and wherein the second main shaft (36) is connected with a planetary wheel carrier (51), arranged in the second planetary gear (12), **characterised in that** the hybrid powertrain (3) is arranged to be controlled according to the method in one of claims 1-6.

8. Computer program (P) for controlling a hybrid powertrain (3), wherein said computer program (P) comprises program code to cause an electronic control device (48), or another computer (53) connected to the electronic control device (48), to perform the steps according to any of patent claims 1-6.

9. Computer program product, comprising a program code stored in a medium readable by a computer, in order to perform the method steps according to any of claims 1-6, when said program code is executed in an electronic control device (48), or in another computer (53)

connected to the electronic control device (48).

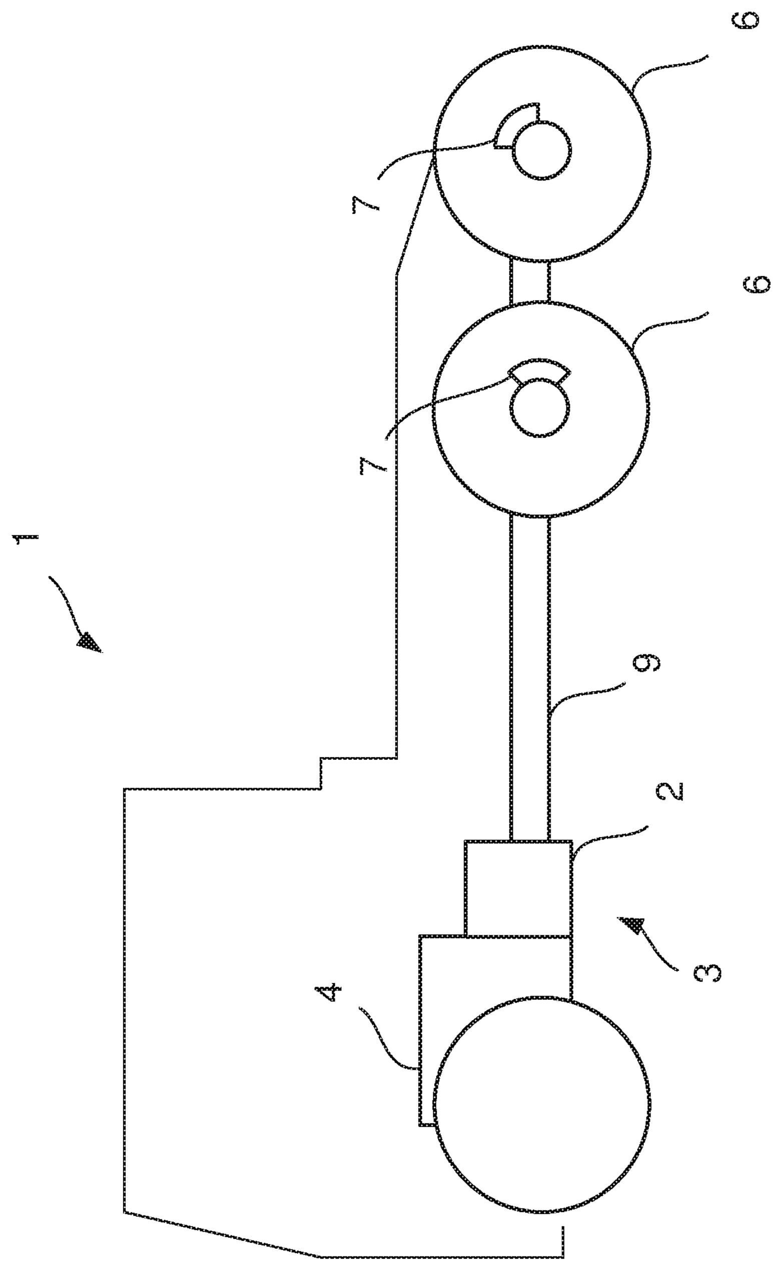

Fig. 1

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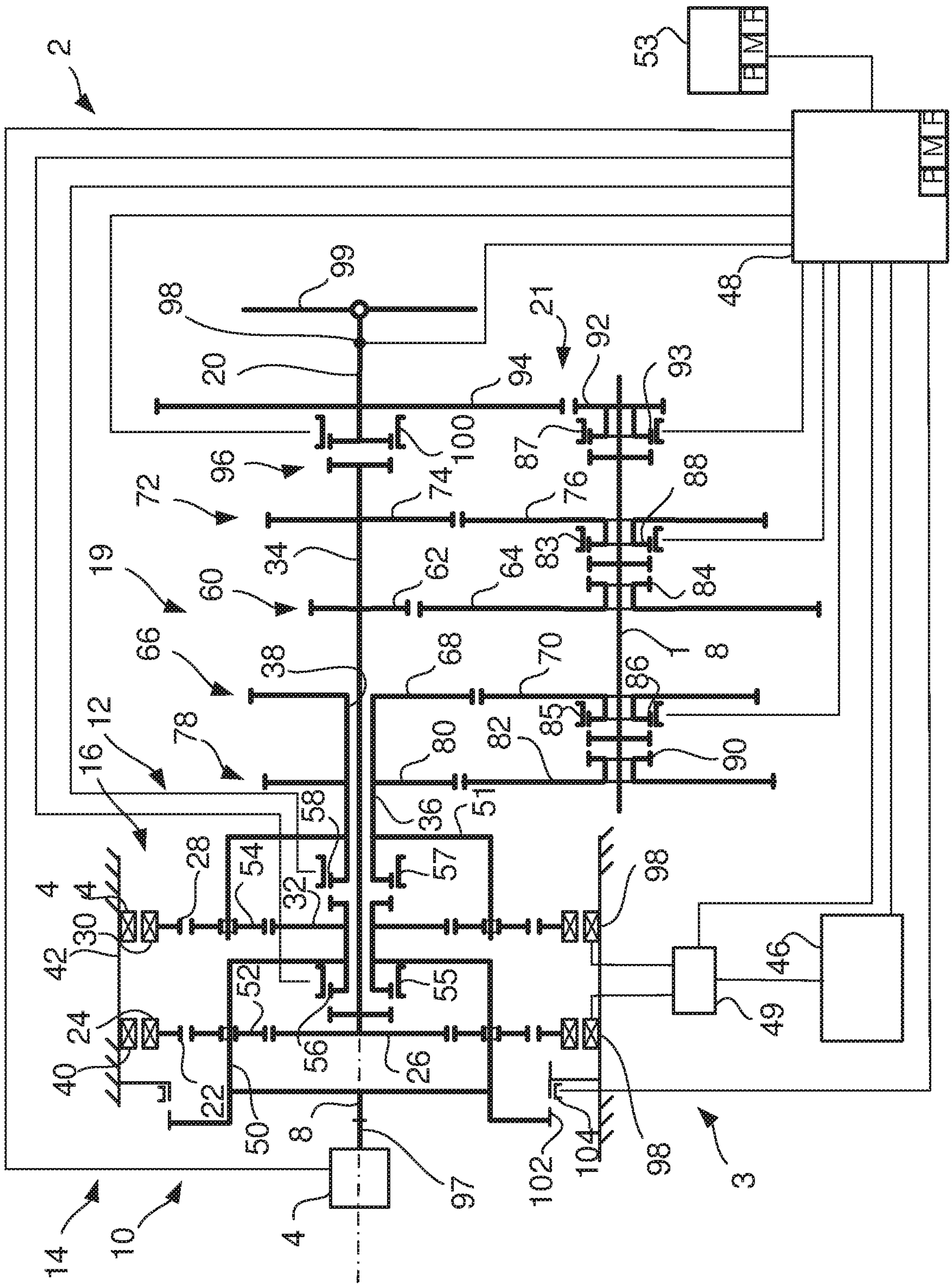

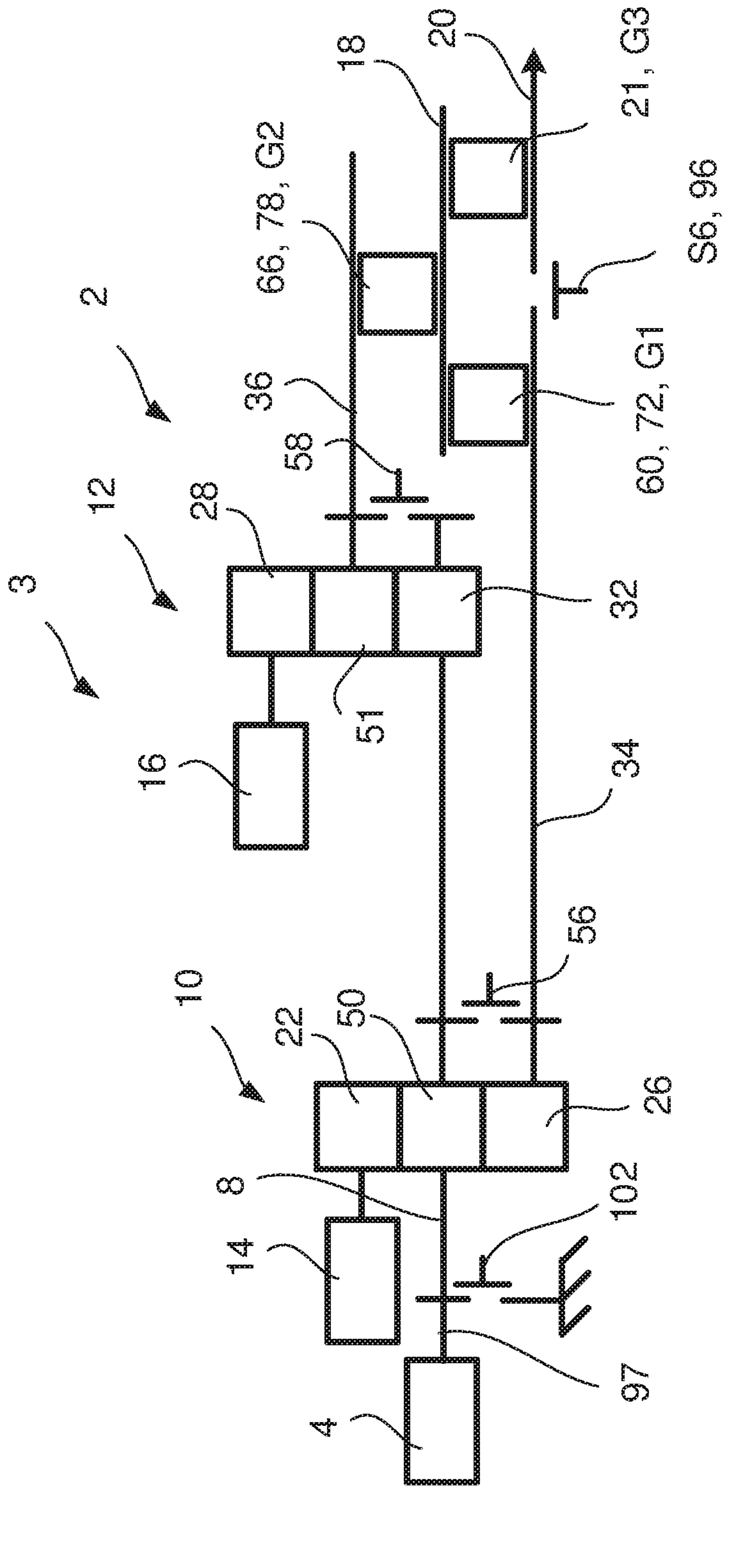
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21, G3
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60, 72, G1
66, 78, G2
97
102
S6, 96

Fig. 3

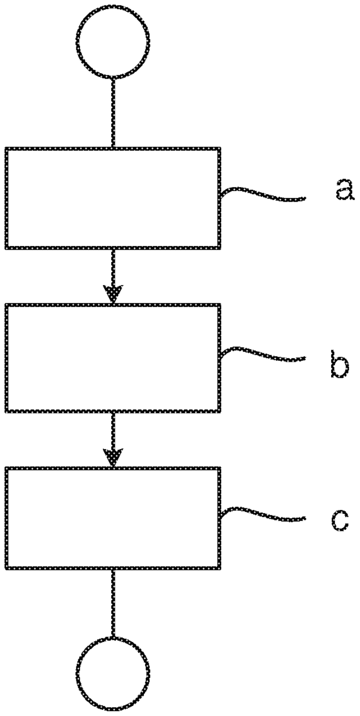

Fig. 4

INTERNATIONAL SEARCH REPORT

International application No.
PCT/SE2015/050294

A. CLASSIFICATION OF SUBJECT MATTER

IPC: see extra sheet

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)

IPC: B60K, B60W

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

SE, DK, FI, NO classes as above

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

EPO-Internal, PAJ, WPI data

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	US 20030166429 A1 (TUMBACK STEFAN), 4 September 2003 (2003-09-04); figure 4; paragraphs [0022]-[0025], [0048]-[0053] --	1-9
A	US 5730676 A1 (SCHMIDT MICHAEL ROLAND), 24 March 1998 (1998-03-24); figures 1-2; column 1, line 66 - column 4, line 48 --	1-9
A	DE 102006025525 A1 (ZAHNRADFABRIK FRIEDRICHSHAFEN), 6 December 2007 (2007-12-06); whole document --	1-9

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

☒ See patent family annex.

* Special categories of cited documents:

"A" document defining the general state of the art which is not considered to be of particular relevance

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"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

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Date of the actual completion of the international search	Date of mailing of the international search report
06-07-2015	06-07-2015

Name and mailing address of the ISA/SE	Authorized officer
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INTERNATIONAL SEARCH REPORT

International application No.
PCT/SE2015/050294

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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A	WO 2008081893 A1 (HONDA MOTOR CO LTD ET AL), 10 July 2008 (2008-07-10); whole document -- --------	1-9

Continuation of: second sheet

International Patent Classification (IPC)

B60K 6/365 (2007.10)
B60K 1/02 (2006.01)
B60K 6/40 (2007.10)
B60K 6/445 (2007.10)
B60K 6/547 (2007.10)
B60W 10/06 (2006.01)
B60W 10/08 (2006.01)
B60W 10/26 (2006.01)
B60W 10/30 (2006.01)
B60W 20/00 (2006.01)

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