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(54) **ELECTRO-HYDRAULIC SYSTEM FOR THE ACTUATION OF MULTIPLE CLUTCHES AND GEAR SELECTORS WITH HIGH-PRECISION CONTROL OF SEVERAL SHIFT GEARBOX UNITS SIMULTANEOUSLY**

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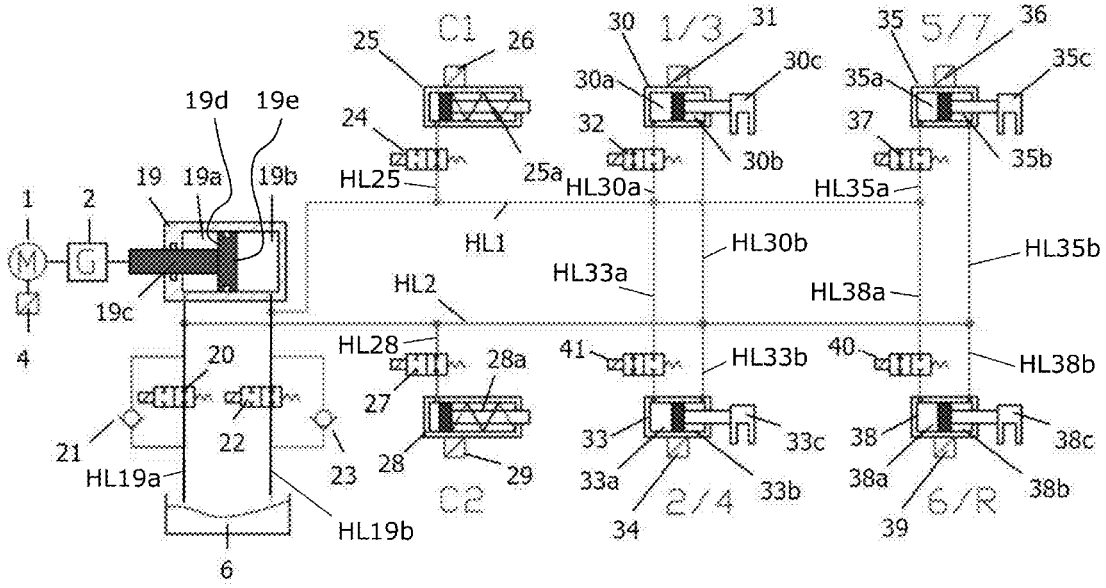
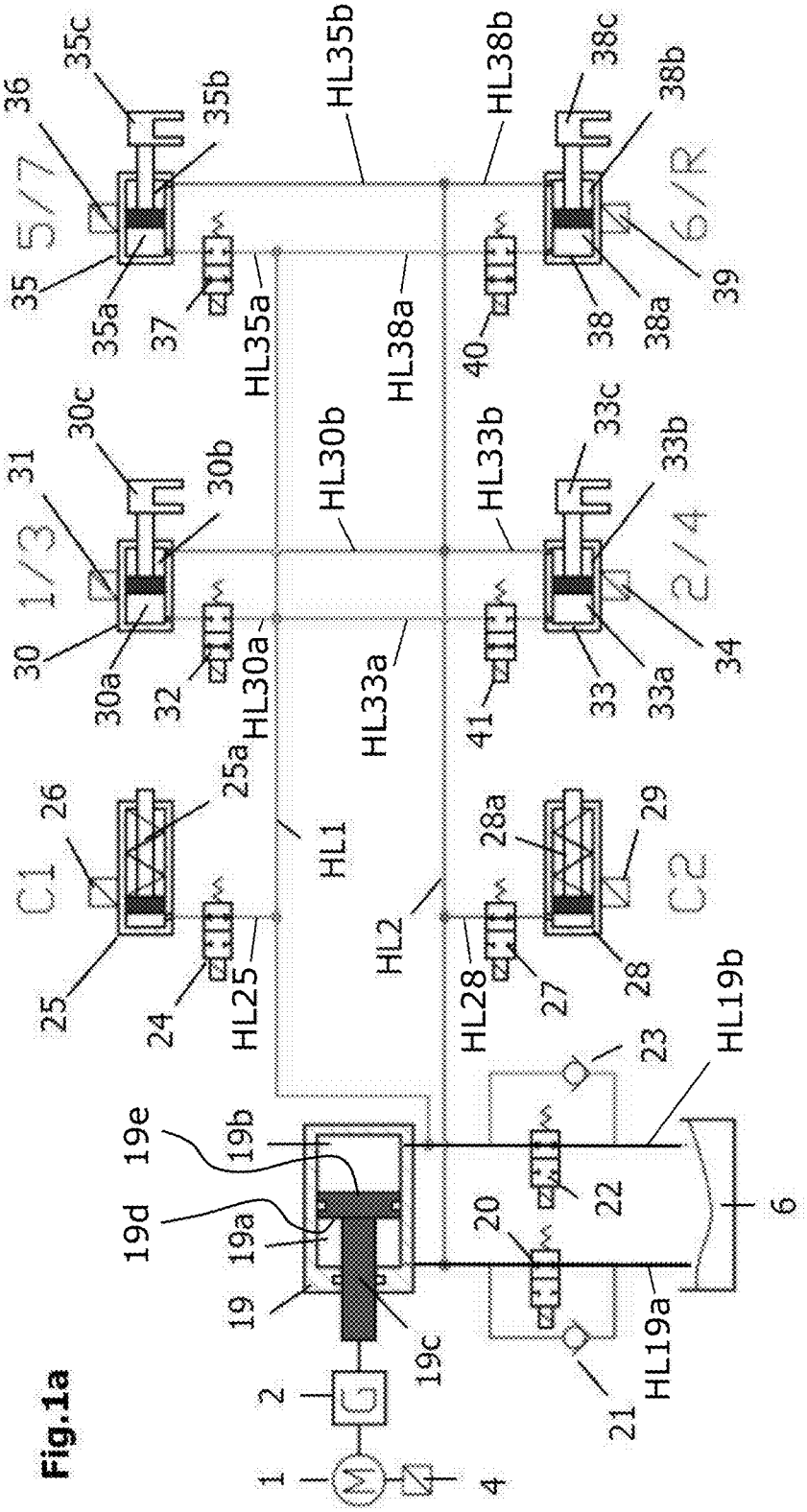
(57) **ABSTRACT**

A shift gearbox, a control unit and at least one electric-motor-driven piston-cylinder unit which has a piston and is connected via hydraulic lines to multiple shift gearbox units of the shift gearbox and shifts them, the shift gearbox units comprising at least two clutch units, characterised in that the piston of the piston-cylinder unit is in the form of a dual-action reciprocating piston, wherein the dual-action reciprocating piston sealingly separates two working chambers from each other, wherein each working chamber is connected via a main hydraulic line to one clutch each, and at least one working chamber of the dual-action reciprocating piston can be connected hydraulically via a switch valve to the reservoir.

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HL19a
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Fig.1a

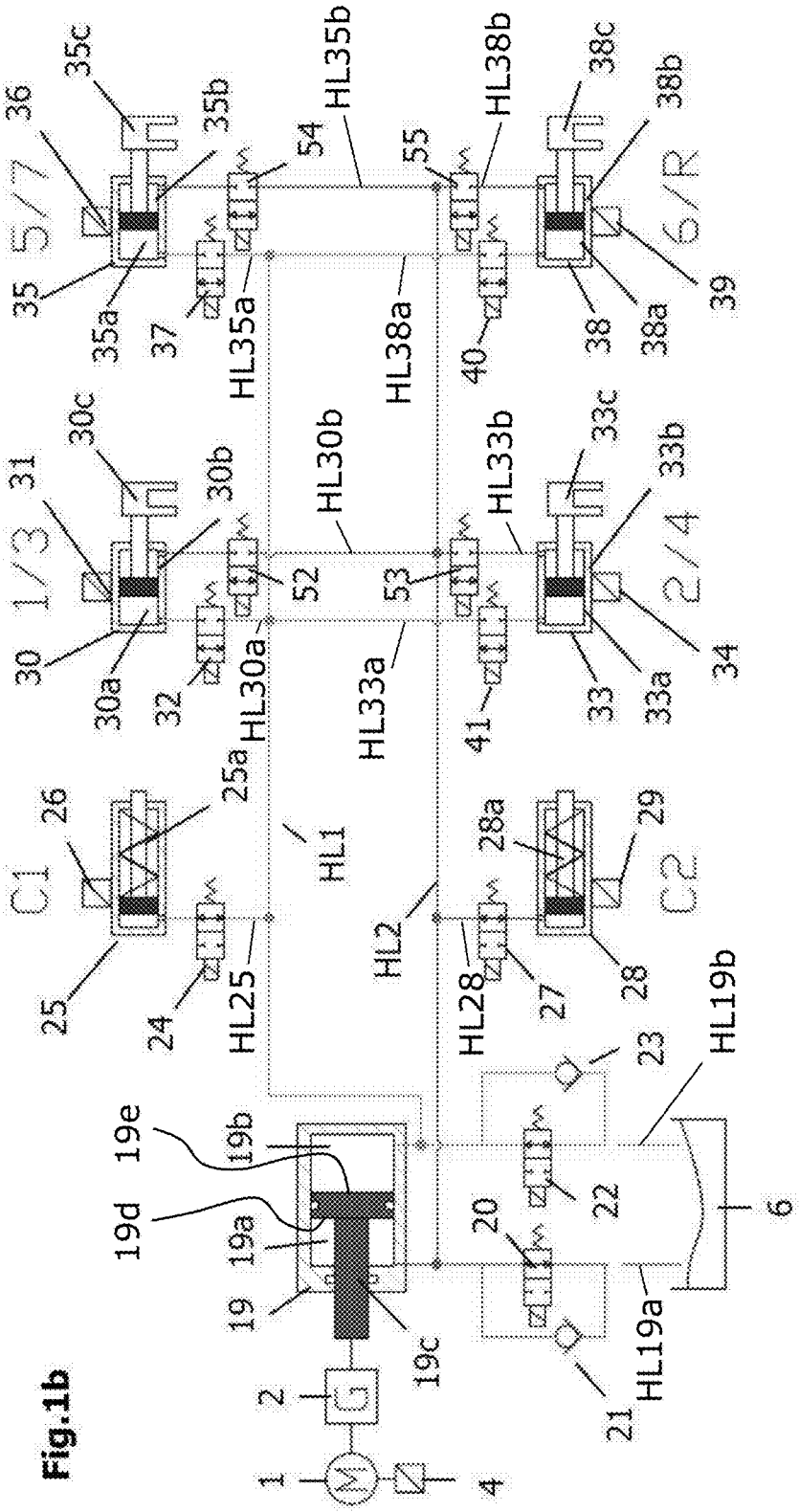
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Fig.1b

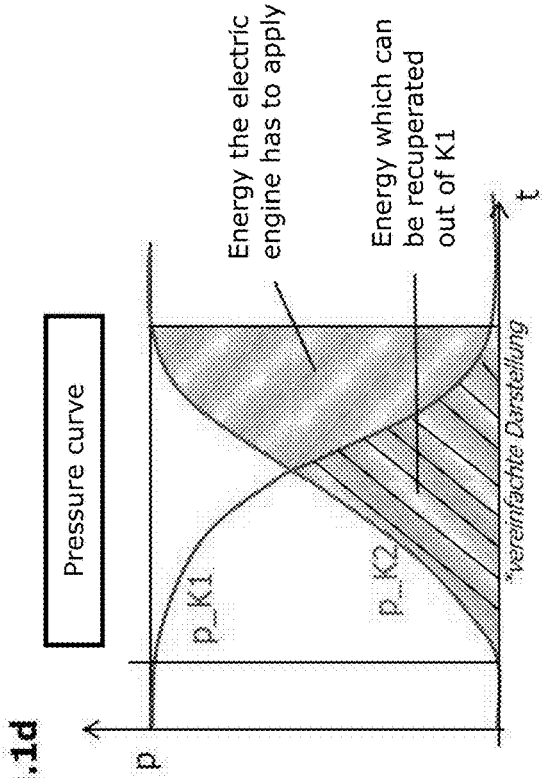


Fig.1d

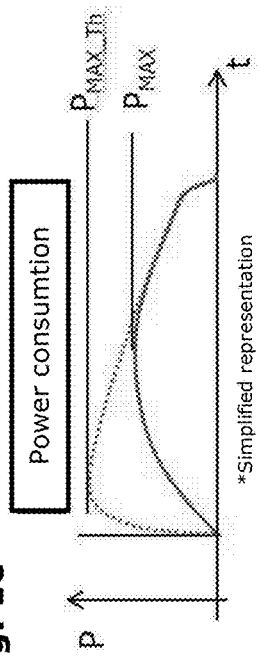


Fig. 1e

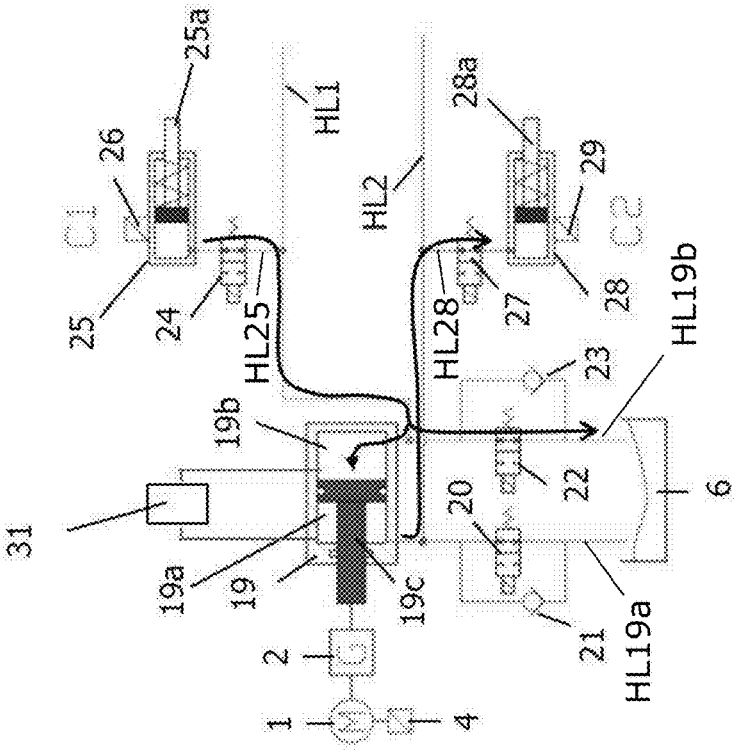
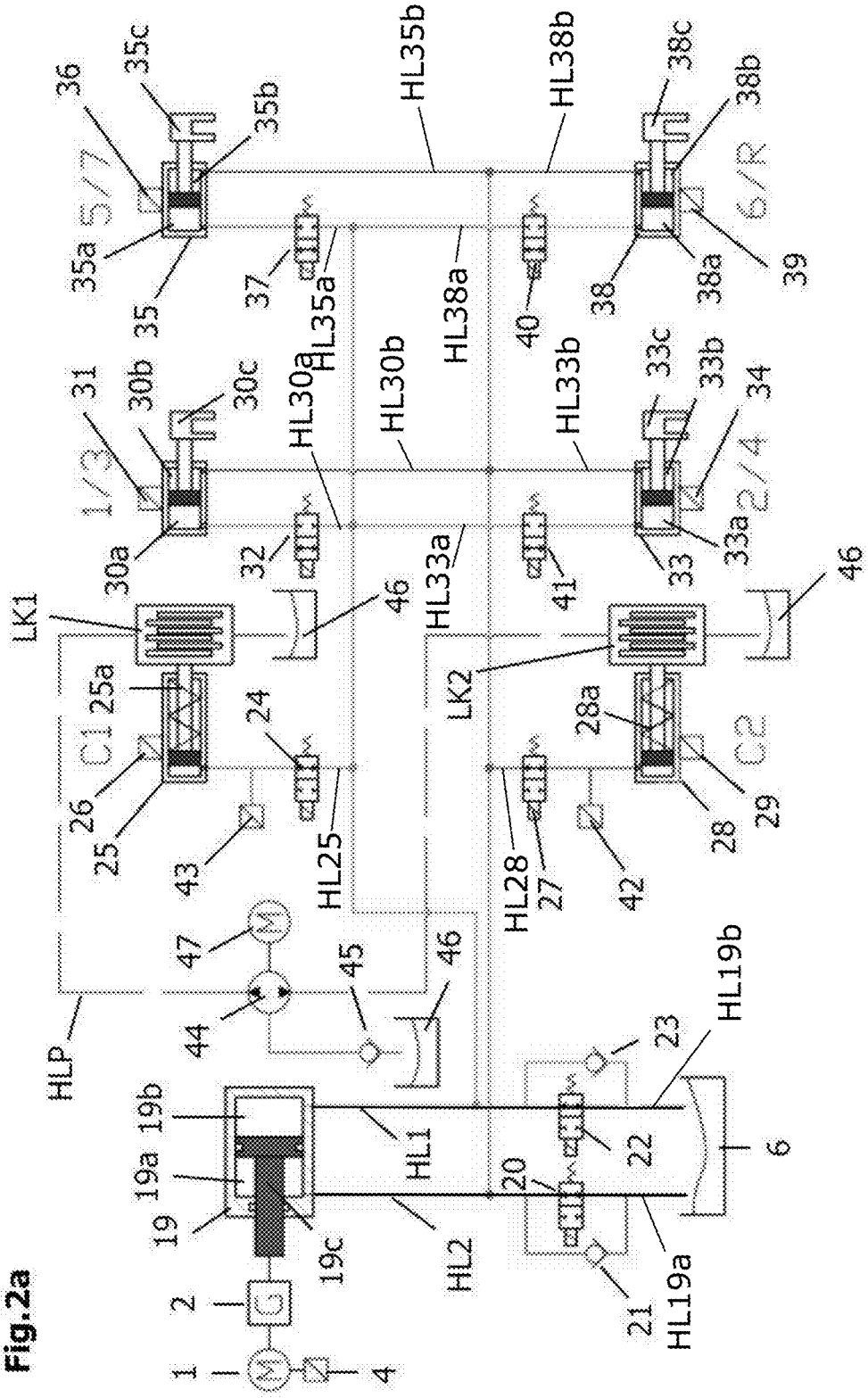


Fig.1c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19b
19c
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HL2
HLP
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HL19a
HL19b
6
C1
25
25a
26
43
24
HL25
LK1
30a
30b
30c
31
1/3
32
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HL30b
HL33a
HL33b
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33a
33b
33c
34
2/4
41
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35b
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Fig.2a

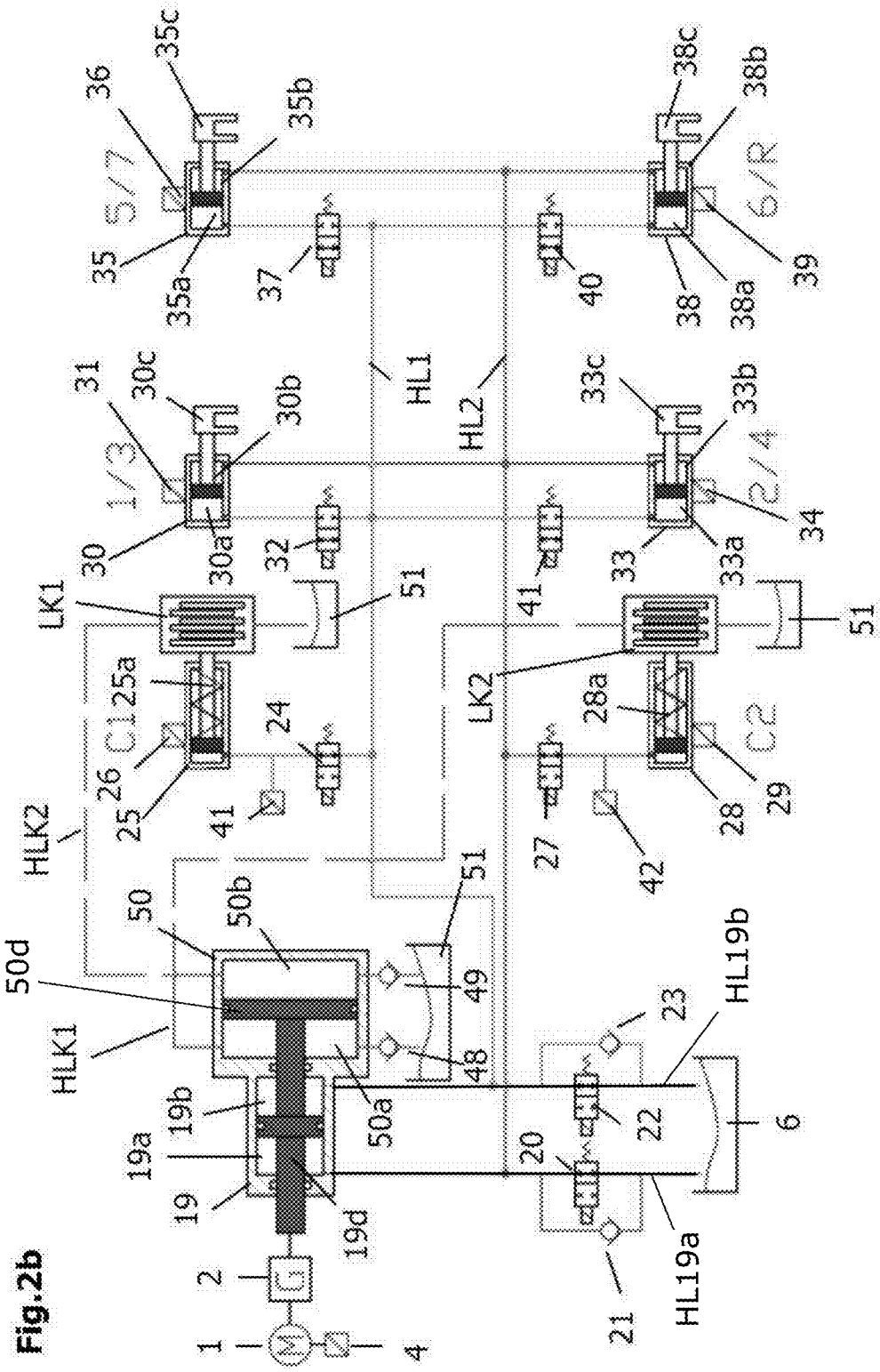
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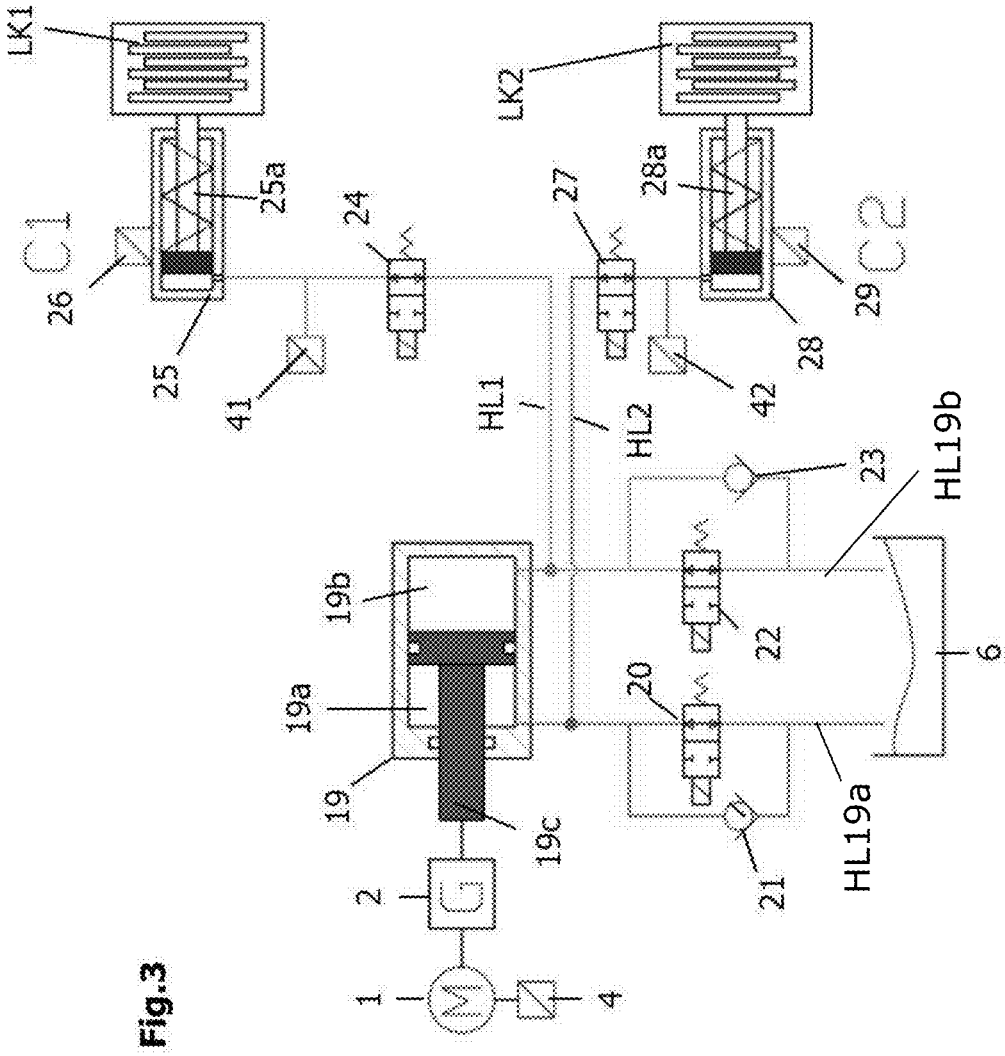
Fig.2b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HL19a
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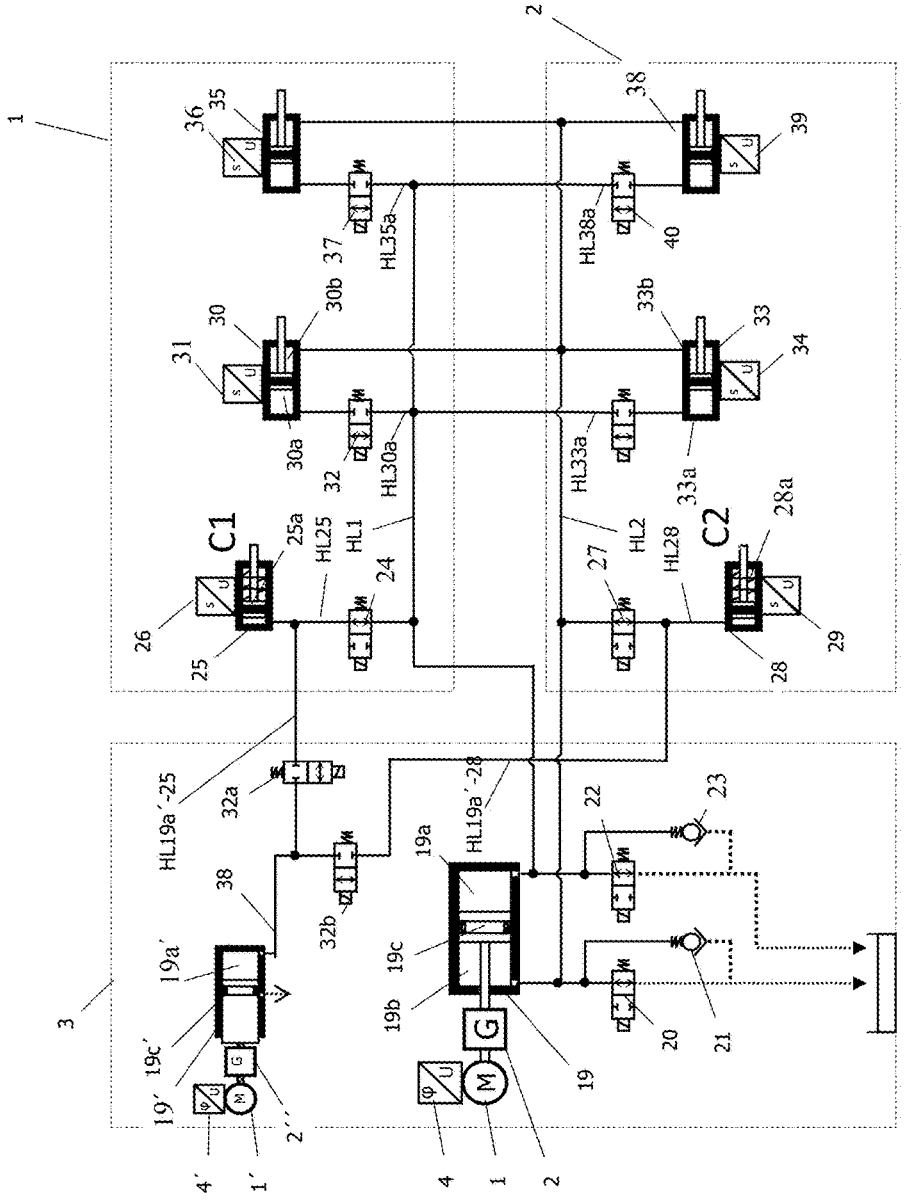
Fig.3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HL1
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HL19a´-25
HL19a´-28
19a
19b
19c
19a´
19c´
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Fig. 4

ELECTRO-HYDRAULIC SYSTEM FOR THE ACTUATION OF MULTIPLE CLUTCHES AND GEAR SELECTORS WITH HIGH-PRECISION CONTROL OF SEVERAL SHIFT GEARBOX UNITS SIMULTANEOUSLY

[0001] The invention relates to a shift gearbox, a control unit and at least one electric-motor-driven piston-cylinder unit which has a piston and is connected via hydraulic lines to multiple shift gearbox units of the shift gearbox and shifts them, the shift gearbox units comprising at least two clutch units.

PRIOR ART

[0002] DE 10 2006 038 446 A1 describes a shift gearbox having an electric-motor-driven piston-cylinder unit, in which one or two piston-cylinder units operate four gear selectors and two clutches. The piston-cylinder unit generates the pressure necessary for shifting the gear selectors and clutches, wherein a pressure sensor measures the pressure generated. DE 10 2006 038 446 A1 describes two possible embodiments for this. In the first embodiment clutches and gear selectors are shifted for actuation of what are known as multiplex valves by means of the piston-cylinder unit. Here, the pressure build-up and the pressure reduction can take place by means of the piston-cylinder unit. But it is also possible for additional exhaust valves to be provided for certain, or all, consumers, via which the pressure in the individual consumers can be lowered in a controlled manner.

OBJECT OF THE INVENTION

[0003] The object of the invention is to further improve the shift gearbox known from DE 10 2006 038 446 A1.

[0004] This object is achieved according to the invention by a shift gearbox in which the piston of the piston-cylinder unit is in the form of a dual-action reciprocating piston, wherein the dual-action reciprocating piston sealingly separates two working chambers from each other, and each working chamber is connected via a main hydraulic line to one clutch each, and at least one working chamber of the dual-action reciprocating piston can be connected hydraulically via a switch valve to the reservoir.

[0005] Advantageous configurations of this shift gearbox are indicated by the features of the subclaims.

[0006] By using a dual-action reciprocating piston, which via its two working chambers in both stroke directions of the dual-action reciprocating piston is able to deliver hydraulic medium to or from one of the shift gearbox units, inter alia, advantageously a short construction of the piston-cylinder unit can be achieved. Thus, both piston surfaces can have the same area, so that during the pre-stroke and the return stroke the same volume is delivered for the same travel of the piston. But it is also possible for the piston surface areas to be of different sizes, e.g. in a ratio of 1.5-2:1, so that during the pre-stroke between 1.5 and 2 times the volume is delivered compared to the return stroke, so that in the pre-stroke volume can be delivered more quickly for the purpose of a faster pressure build-up and thus a quicker operation of the clutch or a quicker gear operation. In this way, very short switching times of a dual-clutch gearbox can be achieved, in particular if at the same time in another clutch the pressure is reduced via a solenoid valve in the reservoir and optimum use can be made of the speed-torque characteristic of an electric motor at a given supplier voltage.

[0007] These different areas/two pressure chambers of the dual-action reciprocating piston can also be used for control as follows:

[0008] a) In gear selectors with 2 different areas, allowing volume control by a valve between gear selector and pressure supply unit (FIG. **1**).

[0009] b) Use of 2 areas of the pressure supply unit and gear selector (FIG. **2**).

[0010] c) Use for downsizing of the electric motor, power determined primarily at the time of clutch operation (pressure build-up with clutch with smaller working area). Use of small area during clutch operation, or changeover via 2 areas of the dual-action reciprocating piston by switch valve (->see extension in figures by a further switch valve).

[0011] d) Recuperation during dual-clutch operation or use of the stored hydraulic energy in a clutch during the process of switching between two clutches for downsizing of the electric motor (use of travel control and exhaust valve to reservoir) (FIG. **1***c*).

[0012] The volume ratio of 2:1 can also be expediently used, in that via a switch a volume compensation between the two working chambers of a dual-action reciprocating piston can be achieved and thus the axial force loading on the transmission is reduced, since in the pre-stroke and in the return stroke only half the area impinges on the transmission unit. This is particularly expedient at high pressures, since the axial force reduces the transmission loading thereby allowing an inexpensive plastic trapezoidal spindle drive to be used. The advantage of the dual-action reciprocating piston over a continuously operating pump is that the pressure generation unit only has to be operated during a shifting operation.

[0013] Controlling the travel of the piston, which amounts to volume control, results in a less expensive construction, in which the number of valves used can be advantageously reduced. Due to the travel or volume control, in a simple manner, without expensive pressure control, at least one shift gearbox unit can have more than two gearshift positions, since because of the incompressibility of the hydraulic medium via a predetermined volume delivered the respective shift gearbox unit can be purposefully shifted into one of the possible positions. Through the travel or volume control by pistons, the components of the shift gearbox units, in particular gear and clutch selectors can also be shifted accurately and more quickly than with proportional valves, since on the basis of prior knowledge of the displacement volume an additional control variable can be applied.

[0014] However, proportional valves can only make limited use of this advantage, since their control variable relates to valve flow and this in turn is dependent upon the hydraulic fluid condition and its viscosity. Furthermore, by virtue of the known volume management and the leak-free design there are few external leakages from the reservoir itself and valve leakages can be accurately diagnosed.

[0015] By using at least one pressure sensor or one position detector, for some shift gearbox units a pressure control or position control for the pressure build-up and also alternatively for the pressure reduction can advantageously be

provided, so that by means of the piston-cylinder unit both travel or volume control and also pressure regulation takes place.

[0016] The pressure control takes place through specific piston travel control or specific energisation of the electric motor. During pressure control, the non-linear relationship between pressure and piston travel is captured and stored in a map. This map is used during pressure control such that a particular travel is performed via the piston, corresponding to a particular pressure. If the map changes as a result of temperature or air inclusions, it is recalibrated or captured again. This can be done in various ways (compensation via pressure sensor, compensation via travel control and using the electric motor current).

[0017] Alternatively, via the current of the electric motor, a torque can be controlled. For an accurate torque determination, by way of example the torque constant kt (relationship between the torque of the electric motor and the phase current) of the electric motor can be used. The torque constant can be determined for electric motors during production or commissioning and is characterised in that kt changes minimally over time and essentially changes only linearly under the influence of temperature. Alternatively to the phase current, the supplier current of the electric motor can be used.

[0018] In the event that no pressure sensor is available, a pressure estimation can be achieved by modelling. According to the invention, then, a model can comprise a motor and gearbox which, for example, pushes or possibly pulls on a single-acting or double-acting hydraulic piston. For a sufficiently good pressure estimation for a transmission unit, the parameters in the subunits (motor torque constant kt, transmission efficiency and hydraulic piston cross-sectional area, friction from seals) must either be subjected to low influences or adapted at regular intervals of time to the fluctuations in the parameters.

[0019] An accurate model can be created in that the abovementioned parameter changes of the model which adversely affect the pressure estimation or pressure control are captured during operation. For example, pressure sensors can be used which are only active in partial operation or an indirect pressure calculation applied.

[0020] In a method for indirect measurements of the pressure via the current of the electric motor, a calculation can be performed via the position of the clutch piston in the slave cylinder and the effective cross-sectional area of the piston of the master cylinder, with the help of knowledge about the spring of the clutch pressure plate and the diameter of the clutch slave cylinder. In this way a system can completely dispense with a pressure sensor, leading to significant cost savings, since pressure sensors are the main cost drivers in hydraulic systems. In mass production applications, a pressure sensor is approximately 4 times as expensive as a switch valve and equally expensive as a proportional valve.

[0021] If the system architecture is now one of a transmission actuator, operated with a motor with hydraulic pistons, this does not necessarily have to be provided with a pressure sensor. Differing pressures in the system can, as described above, be adequately estimated by modelling. In particular, the information on pressure for a shift position can be advantageous. If a gear selector is operated, the force on its shifter fork can be calculated. This means that the position in the gear selector where the synchronisation starts is known and so there is no need for separate algorithms which teach the synchronisation points in all gear selectors. Previously-known systems, such as for example the transmission actuator described in DE 101 34 115 B4, do not have pressure sensors, only position sensors in the gear selectors. The synchronisation point is then evaluated if the speed in the gear train or in the sub-gear train changes. Due to the high mass inertia of the gear trains the speed changes significantly more slowly than the pressure in the gear selector and must therefore, to keep the dynamics high, rely on empirical values from earlier shiftings or learning processes.

[0022] Wet clutches can also be advantageously used, wherein the fluid for cooling the wet clutches can be used either by means of the drives for the dual-action reciprocating piston or separate drives. Thus, for example, an additional dual-action reciprocating piston can be coupled with or rigidly connected to the first dual-action reciprocating piston, which is used to displace the coolant. When the first dual-action reciprocating piston is shifted, then the coolant is also simultaneously delivered. If no clutch or gear selector has to be shifted, the first dual-action reciprocating piston can deliver the fluid by means of suitable valves solely from the reservoir and directly back into this. But it is also possible for a separate pump and additional drive to be used for the coolant.

[0023] Similarly possible is a microslip control of the clutch and simultaneous shifting via multiplexing as shown and described in FIG. **1***b*.

[0024] The shift gearbox according to the invention can also be configured with just two clutch selectors, e.g. without a gear selector, as is the case for two-speed gearboxes for E-vehicles with two clutches and shown and described in FIG. **3**.

[0025] The following advantages can be obtained with the shift gearbox according to the invention:

- **[0026]** a) Weight due to a reduction in the number of components.
- **[0027]** b) Improved reliability through the introduction of diagnostic processes for leak testing and calibration processes for establishing the change in flow resistances.
- **[0028]** c) Reduction in system costs
  - **[0029]** Through a reduction in the number of components, in particular by dispensing with the pump, storage, pressure sensor, filter and non-return valves. This is simply replaced by a motor-transmission-piston-unit.
  - **[0030]** Through a reduction in the hydraulic fluid required.
  - **[0031]** Replacing cost-intensive proportional valves by simple switch valves.
- **[0032]** d) Functional improvement
  - **[0033]** Use of a position-controlled dual-action reciprocating piston as the pressure supply with pressure reduction via the pressure supply unit for closed systems.
  - **[0034]** Optimum utilisation of the torque-speed characteristic of an electric motor for the purpose of a faster operation of one or both clutches.
  - **[0035]** Intelligent pressure control procedures with the potential for a reduction in motor size (described in point 2c).

[0036] e) Improved reliability

[0037] Diagnostic methods for testing components (valves, tightness of pistons of the gear and clutch selectors and of the pressure supply unit) for leaks via piston control.

[0038] Improvement of the hydraulic system by measuring the hydraulic resistances in the system and detection of changes in operation.

[0039] Measurement methods for checking flow resistances of the hydraulic system and its components (e.g. valves, lines) and determination of shifting forces of the pistons of gear selectors and clutch selectors.

[0040] f) Platform concept for automated gear shifting and dual clutches with the fewest possible changes to components in the system.

[0041] Advantageous possible embodiments of the shift gearbox according to the invention are explained in more detail below using the drawings.

[0042] These show as follows:

[0043] FIG. **1***a*: A shift gearbox with a piston-cylinder unit with dual-action reciprocating piston with eight valves and two dry-running clutch selectors and four gear selectors in the closed hydraulic circuit;

[0044] FIG. **1***b*: Shift gearbox with a piston-cylinder unit with dual-action reciprocating piston with twelve valves and two dry-running clutches and four gear selectors in the closed hydraulic circuit;

[0045] FIG. **1***c*: Shift gearbox with a piston-cylinder unit with dual-action reciprocating piston with intelligent control for clutch operation with potential for downsizing of the motor-transmission-piston-unit by virtue of the use of the stored energy in a clutch;

[0046] FIG. **1***d*: Use of the stored energy in a clutch during the process of switching over between two clutches;

[0047] FIG. **1***e* Performance diagram for a shift gearbox, in which an intelligent pressure control via piston control and exhaust valves takes place to reduce the power consumption;

[0048] FIG. **2***a*: Shift gearbox with a piston-cylinder unit with dual-action reciprocating piston with two wet-running clutches and four gear selectors in the closed hydraulic circuit with additional pump;

[0049] FIG. **2***b*: Shift gearbox with a piston-cylinder unit with dual-action reciprocating piston with two wet-running clutches and four gear selectors in the closed hydraulic circuit with dual-action reciprocating piston (dual-action reciprocating piston-pump) driven by the drive of the piston-cylinder unit;

[0050] FIG. **3** Piston-cylinder unit with dual-action reciprocating piston for two-speed system with closed hydraulic circuit;

[0051] FIG. **4** Expanded shift gearbox with additional piston-cylinder unit.

[0052] FIG. **2***a* shows a first possible embodiment of the shift gearbox according to the invention in the form of a dual-clutch gearbox with a piston-cylinder unit **19** with dual-action reciprocating piston **19***c* for displacing the hydraulic medium in the clutch selector **25**/C**1**, **28**/C**2**.

[0053] The piston-cylinder unit **19** is driven by the drive **1** via the transmission **2**. The dual-action reciprocating piston **19***c* separates the two working chambers **19***a* and **19***b* from each other, wherein the piston area **19***e*, which delimits the working chamber **19***b*, is larger than the effective piston area **19***d*, which delimits the working chamber **19***a*. The working chamber **19***a* is connected via the main hydraulic line HL**2**. The working chamber **19***b* is connected with the main hydraulic line HL**1**. The hydraulic feed lines HL**25**, HL**28**, HL**30***a*, HL**30***b*, HL**33***a*, HL**33***b*, HL**35***a*, HL**35***b*, HL**38***a* and HL**38**, which connect the main hydraulic lines HL**1**, HL**2** with the clutches **25**/C**1**, **28**/C**2** and the speed selectors **30**, **33**, **35** and **38**, branch off from the main hydraulic lines HL**1**, HL**2**. In each of the hydraulic feed lines HL**25**, HL**28**, HL**30***a*, HL**30***b*, HL**33***a*, HL**33***b*, HL**35***a*, HL**35***b*, HL**38***a* and HL**38**, switchable valves **24**, **27**, **32**, **33**, **37**, **40** and **41** are arranged for optional shutting off or opening of the feed lines. The two working chambers **19***a* and **19***b* are in each case connected via hydraulic lines HL**19***a* and HL**19***b* with a reservoir **6**, wherein in the hydraulic lines HL**19***a* and HL**19***b* switchable 2/2-way valves **20**, **22** are arranged. In parallel with each 2/2-way valve **20**, **22** a non-return valve is arranged.

[0054] The shift gearbox according to FIG. **1***a* with two clutch selectors and four gear selectors requires only eight switchable 2/2-way valves.

[0055] The gear selectors **30**, each have two working chambers **30***a*, **30***b*, **33***a*, **33***b*, **35***a*, **35***b* and **38***a*, **38***b* which create seals and are separated by pistons from each other. With this arrangement it is important that the first working chambers **30***a*, **33***a*, **35***a*, and **38***a* are connected with the first main hydraulic line HL**1** and thus with the working chamber **19***b*, and that the second working chambers **30***b*, **33***b*, **35***b*, and **38***b* are connected via the second main hydraulic line HL**2** with the working chamber **19***a* of the piston-cylinder unit **19**.

[0056] Through this separate arrangement of the connecting lines HL**1** and HL**2** a gear shift can be implemented as follows: For a gear shift from first to second gear, it is first necessary to select second gear, wherein the clutch C**1**(**25**) is depressed into this initial state and thereby also closed. However, so that the volume or the pressure from the clutch C**1** does not escape, the clutch selector valve **24** must be closed. To initiate the gear shift, the gear selector valve **1** (**35**) is opened and the exhaust valve **1** and the clutch selector valve **2** closed. Thereafter, the dual-action reciprocating piston **19***c* with the motor and transmission units **1** and **2** can be moved to the left, as a result of which volume is displaced into the gear selector **2**/**4** (**33**) especially into the chamber **33***b*. Were the valve **35** not opened during this process, to allow the shifting of the gear selector **33**, the system would be hydraulically isolated. Once second gear in the gear selector **2**/**4** (**33**) is synchronised in the sub-transmission with, for example, the crankshaft, the gear can finally be selected. Gear selector valve **35** is closed again, clutch selector valve **27** is opened and exhaust valve **20** continues to be closed and clutch operation can be commenced in the clutch C**2** (**28**). To allow shifting without interruptions in traction, a continuous change of load of the two clutches C**1** (**25**) and C**2** (**28**) must take place. The closing of the clutch C**2** is undertaken with the help of the pressure build-up in the dual-action reciprocating piston **19**, which in turn moves to the left. The simultaneous opening of the clutch C**1**(**25**) is achieved by a stepped or also smooth control of the clutch selector valve **24**, such that the fluid leaves in a controlled manner via the corresponding exhaust valve **22**. Once the load change is complete, the gear selector **1**/**3** (**30**) can either be shifted to neutral (mid-position of the shifter fork **30***c*) or the next gear pre-selected. In doing so,

the clutch selector valves **24**, **27**, and the exhaust valve **22** are closed and the gear selector valve **32** opened. The dual-action reciprocating piston **19** displaces the volume from the chamber **19***b* and thus moves the gear selector **30** to the right, according to the volume displaced. Selection of first or second gear is finally completed.

[0057] Preferably, prior to a journey, the piston **19***c* is in a mid-position, since it cannot be anticipated if when the vehicle starts first gear or reverse gear will be selected. Thus, corresponding volume for operating a gear selector and a clutch is present for both manoeuvres. Alternatively, when the valves **20** or **22** are open, the piston would have to be moved into the correct position.

[0058] During the load change from one sub-transmission to the other sub-transmission, if a clutch **25** is depressed by means of the motor-transmission-piston unit **1**, **2** and from the other clutch **28** via the corresponding clutch selector valve **27** fluid is discharged, the clutches can be controlled via possible position sensors **26**, **29** or pressure sensors. Depending on the embodiment of the transmission in modern transmissions a pressure or position sensor is used. Dry clutches are generally built with position sensors and wet clutches with pressure sensors. The controlled discharge of the clutches can take place either with the valves **24** and **27** or with the valves **20** and **22** either in a stepped or a smooth manner, depending on the type of valve used. In the embodiment shown simple switch valves (stepped) or a valve with flying similarly-controlled plunger (smooth) are used.

[0059] For safety reasons, in each embodiment of the double clutch actuator with eight valves a position sensor **31**, **34**, **36**, **39** is provided in each gear selector **30**, **33**, **35**, **38**, so that any leaks in the valves **32**, **37**, **40**, **41** cannot lead to mechanical destruction. The valves **20**, **22**, **24** and **27** must have a normally open design, so that in the event of a system failure both clutches **25**, **28** are immediately opened without requiring further supply.

[0060] FIG. **1***b* shows a design, in which the pressure can be shut off by switch valves **32** and **52**, **41** and **53**, **37** and **54**, **40** and **55** in the gear selectors **30**, **33**, **35**, **38**. In the case of a dual-clutch gearbox the clutch C**1** or C**2** can be operated, which furthermore is operated with what is known as microslip and with which the dual-action reciprocating piston **19** is controlled. Microslip is used in order to attenuate to a certain extent undesired speed fluctuations at the crankshaft and to be able to better estimate the open position of the clutches. The effect of the attenuation is dependent upon the amount of the slip applied to the corresponding clutch. If it is intended to complete a gear shift, then this often lasts for a few hundred milliseconds, since the synchronisation of the unloaded sub-transmission accounts for a majority of the total switching time. With the help of a dual-action reciprocating piston **19** operated by a trapezoidal spindle or a ball screw drive **2**, a momentary gear shift can be initiated. Here, the clutch **25** or **28** most recently under load is shut off in the corresponding sub-transmission with the clutch selector-valve **24** or **27** and fluid can now be discharged with valves **24** or **22** and also **27** or **21**. In this short time, microslip control is not possible or only to a limited extent, but the clutch nevertheless continues to operate with slip. Then the desired gear selector is operated and moved only as far as the synchronisation point, wherein the pressure in the gear selector can be calculated from the motor current. If the synchronisation is initiated, in the corresponding gear selector the hydraulic pressure can be shut off with switch valves and after a brief time interrupt the dual-action reciprocating piston **19** can restart the microslip control on the loaded clutch **25** or **28**. However, to do this the pressure level in the dual-action reciprocating piston **19** must encounter that of the loaded clutch and then open the clutch selector valve **27** or **24** again without pressure differential. Once synchronisation in the unloaded sub-transmission is complete, the final gear shift can be initiated and the load change is completed.

[0061] FIG. **1***c* shows a variant for controlling the two clutches **25**/C**1** and **28**/C**2**. This is an intelligent modification to allow a reduction in size of the motor **1** for the drive of the hydraulic piston **19**, which is driven by means of a spindle **2**, and thus save power, weight and space. For example, if it is intended to perform a gear shift from sub-transmission **1** with the clutch C**1**/**25** to sub-transmission **2** with the clutch C**2**/**28**, the stored potential energy of clutch C**1**/**25** can be used for the pressure build-up in clutch C**2**/**28**. A schematic representation of the process is shown in FIGS. **1***d* and **1***e*. FIG. **1***d* shows possible pressure gradients in the clutches for this modification and FIG. **1***e* a simplified representation for reduced power consumption of the electric motor **1**.

[0062] FIG. **1***c* shows, by means of the arrows, how the fluid flows when there is a load change. Thus, the stored and pressurised fluid in the clutch C**1**/**25** is directed via the lines HL**25** and HL**1** into the working chamber **19***b* and exerts a force on the piston **19***c* towards the left. This force supports the motor **1** when moving the piston **19***c* to the left to shrink the working chamber **19***a* in order to build up a pressure in the clutch C**2**/**28**. The hatched area shown in FIG. **1***d* corresponds to the energy which can be saved by the supporting power of the pressurised fluid in the clutch C**1**/**25** when switching the clutch C**2**/**28**. If it is intended to open the clutch C**2**/**28** and close the clutch C**1**/**25**, then the pressure stored in the clutch C**2**/**28** can similarly be used to support movement of the piston **19***c*. In this way, the maximum power needed by the motor can be reduced from P_{max_Th} to P_{max}, as shown in FIG. **1***e*. Thus, the motor **1** can have smaller dimensions.

[0063] Due to hysteresis and friction losses in the closed hydraulic transmission actuator, during this process there may be too much volume in the system for a proper load change. The exhaust valves **20** and **22** can simultaneously ensure appropriate volume management and discharge any excess fluid via the lines HL**19***a*, HL**19***b* into the reservoir **6**. Depending on the design of the motor-transmission-piston-unit **1**, **2**, **19**, in this embodiment when there is a load change between the clutches the maximum output of the motor **1** is required. This means that the motor **1** with overarching intelligent control (motor **1** and valves **20**, **22**, **24**, **27**) can generally be built with smaller dimensions. During the start-up phase in particular, until the pressure of the two clutches **25** and **28** are the same, apart from when there are reductions in efficiency (ball screw drive or trapezoidal spindle, hydraulic losses, etc) the motor can generally be dispensed with. Only if the clutch pressure in clutch C**2**/**28** is greater than in clutch C**1**/**25** must the motor, with the support of the residual pressure in clutch C**1**/**25**, fully develop the pressure in clutch C**2**.

[0064] FIG. **2***a* describes the construction of a dual-clutch gearbox with wet-running clutches C**1** and C**2** and a separate cooling circuit HLP with independent pump **44** with drive motor **43**. The functioning and execution of a gear shift are

the same as described in FIG. **1***a*, but the clutches C**1** and C**2** are controlled via the pressure sensors **41**, **42** and not position sensors **26**, **29**. The position sensors can therefore be dispensed with. Due to high torque transfers and the possible use of multi-plate clutches, the pump **44** is cooled by a separate cooling circuit HLP with its own medium supplied from the tank **46**.

[0065] FIG. **2***b* describes a system architecture of a dual-action reciprocating piston with wet-running clutches and separate cooling circuits HLK**1** and HLK**2** with simultaneously running dual-action reciprocating piston-pump **50** which is connected to the piston provision of the motor-transmission-piston unit **1**, **2**, **19**. With the operation of the independent transmission actuator, by means of a separate dual-action reciprocating piston **50**, a pumping function can be undertaken. Thus, an additional pump with motor can be saved. The cooling circuits HLK**1**, HLK**2** operate with separate media as a result of which contamination cannot enter the actual dual-action actuator **19**. In this embodiment, the additional dual-action reciprocating piston **50***d* must be significantly larger than the actual actuator with dual-action reciprocating piston **19***c*, since several litres per minute of fluid for cooling must be delivered. Since in some circumstances the actuator does not have to perform any gear or clutch selection, coolant can continue to be delivered from the reservoir **47** and via the non-return valves **48** and **49**, while valves **20** and **22** are open. The piston **50***d* can be moved back and forth by means of the drive **1** as a function of the required flow rate—high frequency with intense cooling, low frequency with low cooling—without the clutches C**1** and C**2** and gear selector being shifted in the process. This is achieved in that the associated valves **24**, **27**, **32**, **37**, **40** and **41** are closed and the valves **20** and **22** opened. Optionally, in all the embodiments shown and described in the figures the valve **31** shown in FIG. **1***c* can be arranged, which in the open state hydraulically connect with each other or short-circuit the two working chambers **19***a*, **19***b*. Consequently, cooling can take place in power-on-demand mode. Where the clutch and gear selectors have to be operated, the required flow rate may not be achieved, but this is not critical since the operation is usually over very quickly.

[0066] FIG. **3** shows a dual-clutch concept with two gears, which can advantageously be used for electric drives. A modular utilisation of the dual-action reciprocating piston assembly kit is possible, wherein the components for the gear selector are not needed. Thus, a two-gear system without interruptions in traction is possible for electric motor drives. Clutch control is the same as described in FIG. **1***a* and can take place with pressure sensors **41**, **42** or also position sensors **26**, **29**.

[0067] FIG. **4** shows an extension to a system described above. The original system comprises the valve circuits in the sub-transmission **1** and sub-transmission **2** with the respective valves **24**, **27**, **32**, **37**, **40** and **41** for operation of the clutches **25**, **28** and the gear selector **30**, **33**, **35**, **38**. The hydraulic actuator **19**, which is driven by the motor **1** via the transmission **2** and has a dual-action reciprocating piston **19***c*, is connectable with its working chambers **19***a*, **19***b* via the two valves **20**, **22** with the tank or reservoir **6**.

[0068] The extension to the shift gearbox consists of the possibility of using the pressure modulator **19**, which is driven by the motor **1'** via the transmission **2'**, for operating the clutches C**1** and C**2**. To this end, the working chamber **19***a*' can be connected via the hydraulic lines HL**19***a*'-**25** and HL**19***a*'-**28** with the clutch selectors **25**, **29**, wherein a switch valve **32***a*, **32***b* is arranged in each of the hydraulic lines HL**19***a*'-**25** and HL**19***a*'-**28** for shutting off or opening these. This allows continuous microslip of the respective clutch in traction. Here, the valves **32***a*, **32***b*, connecting the pressure modulator **19***a*' with the clutch selectors **25**, **28**, can have a normally open or normally closed design.

[0069] The functional features of the switching are explained in more detail in the following.

[0070] Situation 1: Microslip Control of Clutch Selector **25** with Simultaneous Gear Shifting in the Sub-Transmission **2**.

[0071] In the situation described, the pressure modulator **19***a*' provides continuous microslip control of the selector **25**, while the pressure modulation valve **32***b* for the clutch selector is open and the pressure modulation valve **32***b* for the other clutch selector **28** and the clutch valve **24** is closed. Here, the pressure modulator **19***a*' controls the microslip of clutch **25** as a function of the clutch travel sensor **26**. If now in parallel a gear shift in sub-transmission **2** is necessary, this can be performed by the hydraulic actuator **19**. If, by way of example, at the gear selector **33** a switching from the neutral position to the right is necessary, then the valves **20**, **22** and **27** are closed and the gear selector admission valve **41** is opened and through a movement of the dual-action reciprocating piston **19***c* to the right, the dual-action reciprocating piston of the gear selector **33** is shifted to the right and into fourth gear. A movement of the dual-action reciprocating piston to the left similarly offers the possibility of moving the gear selector **33** to the left and thus selecting the corresponding gear. The same applies, of course, to all further gear selectors in sub-transmission **2**. There is also a purely theoretical possibility that in parallel with the microslip of clutch **25**, via the dual-action reciprocating piston, volume can be shifted to or from the clutch selector **28**, and gear selectors **30** and **35**.

[0072] Situation 2: Deactivation of Clutch a and Simultaneous Activation of Clutch b

[0073] Here, the position of clutch a is not controlled via a similarly controlled valve **24** or **30***a*, but via the pressure modulator **19***a*'. This allows the valves **24**, **27**, **30***a*, **30***b* to be pure and simple digital switch valves.

[0074] On the basis of situation 1 described above, valves **24** and **27** are now open. Valves **30***a*, **30***b*, **32**, **37**, **40** and **41** and pressure modulation valve **32***b* between clutch selector **28** and the pressure modulator **19***a*' are closed where this is not already the case. Via the dual-action reciprocating piston **19***c* the pressure build-up or the position of the clutch selector **19***c* are now controlled. The dual-action reciprocating piston **19***c* also moves to the left. The right-hand chamber of the dual-action reciprocating piston thereby simultaneously draws volume via **24** from the clutch selector **25**. In this case, the pressure modulator **19***a*' controls the pressure or the position of clutch selector **25**. In this situation, the main volume flow is displaced by the dual-action reciprocating piston **19***c*. The pressure modulator **19***a*' only corrects the volume for the clutch selector **28** as required. Once sub-transmission **2** has been activated and sub-transmission **1** deactivated, clutch valves **24** and **27** are closed and the pressure modulator **19***a*' is separated by the pressure modulation valve **32***a* from the clutch selector **25** and connected by the other pressure modulation valve **32***b* with

the clutch selector **28**. The pressure modulator **19***a*' now controls the microslip on clutch selector **28**.

[0075] The advantage of this circuit is that the pressure modulator **19***a*' has a significantly lower volume management than the dual-action reciprocating piston **19***c*. The volume flow demands on the pressure modulator **19***a*' are also significantly lower than for the volume flow of the dual-action reciprocating piston **19***c*. Added to this is fact that the system is completely free of analogue valves and works purely with less expensive digitally-switching valves.

[0076] For system efficiency diagnosis, this system offers the possibility, by way of example by opening the valves **32***b* and **27**, of connecting the two pressure chambers to each other and thereby balancing the transmission efficiency of pressure modulator **19**, **19***a*' and hydraulic actuator **19**. This balancing can be very helpful both in predicting failures and also matching the pressure settings more accurately and thereby increasing the comfort. The diagnostic option mentioned exists for virtually all systems that have two hydraulic actuators or pressure modulators and offer the possibility of hydraulically connecting the systems quickly.

[0077] In emergency mode, in the event of failure of one of the motors **1**, **1**" of the pressure modulator **19**, **19**' or the hydraulic actuator **19** there is possibility of the respective other pressure supply taking over the gear selection and clutch selection. If in emergency mode, the pressure modulator has to take over the clutch selection and gear selection then so muss by virtue of the lower volume management via the exhaust valves **30***a* and/or **30***b* in the meantime volume in the pressure modulator **33** must be called upon. But if the pressure modulator **33** fails, then via the hydraulic actuator **19**, apart from brief interruptions in microslip control, function can be maintained. Basically, the extension to the original circuit is only necessary if brief interruptions in microslip control are unacceptable during the gear shifting processes.

LIST OF REFERENCE SIGNS

[0078] **1** EC motor
[0079] **2** Transmission
[0080] **3** Piston-cylinder unit
[0081] **4** Rotary sensor for motor commutation
[0082] **5** Position sensor for clutch selector in automatic transmission
[0083] **6** Reservoir
[0084] **7** Clutch unit **1**
[0085] **8** Pressure sensor for clutch selector in automatic transmission
[0086] **9** 2/2-way valve
[0087] **10** Gear selector unit **1** (rotational movement)
[0088] **10***a*, **10***b* Piston-cylinder units of gear selector **10**
[0089] **11** Gear selector unit **2** (linear movement)
[0090] **12** Piston of gear selector mechanism **1**, rotation (**3** positions)
[0091] **13** Gear selector mechanism **2**, translation (**3** positions)
[0092] **14** 2/2-way valve
[0093] **15** Return spring of the gear selector mechanism **2**
[0094] **16** 2/2-way valve
[0095] **17** Rotation body of gear selector-mechanism (**3** positions)
[0096] **18** 2/2-way valve
[0097] **19** Dual-action reciprocating piston
[0098] **19***a* Hydraulic chamber of the dual-action reciprocating piston for hydraulic circuit HL**2**
[0099] **19***b* Hydraulic chamber of the dual-action reciprocating pistons for hydraulic circuit HL**1**
[0100] **19***b* Hydraulic operation piston
[0101] **20** 2/2-way admission and exhaust valve for HL**2**
[0102] **21** Non-return valve for HL**2**
[0103] **22** 2/2-way admission and exhaust valve for HL**1**
[0104] **23** Non-return valve for HL**1**
[0105] **24** 2/2-way admission and exhaust valve for clutch C**1**
[0106] **25** Clutch selector C**1**
[0107] **25***a* Hydraulic piston of clutch selector C**1**
[0108] **26** Position sensor for clutch selector C**1**
[0109] **27** 2/2-way admission and exhaust valve for clutch C**2**
[0110] **28** Clutch selector C**228***a* Hydraulic piston of clutch selector C**2**
[0111] **29** Position sensor for clutch selector C**230** Gear selector **1**/**3**
[0112] **30***a* Hydraulic chamber **1** of gear selector **1**/**3**
[0113] **30***b* Hydraulic chamber **2** of gear selector **1**/**3**
[0114] **30***c* Piston with shifter fork of gear selector **1**/**3**
[0115] **31** Position sensor of gear selector **1**/**3**
[0116] **32** 2/2-way admission and exhaust valve **1** for gear selector **1**/**3**
[0117] **33** Gear selector **2**/**4**
[0118] **33***a* Hydraulic chamber **1** of gear selector **2**/**4**
[0119] **33***b* Hydraulic chamber **2** of gear selector **2**/**4**
[0120] **33***c* Piston with shifter fork of gear selector **2**/**4**
[0121] **34** Position sensor of gear selector **2**/**4**
[0122] **35** Gear selector **5**/**7**
[0123] **35***a* Hydraulic chamber **1** of gear selector **5**/**7**
[0124] **35***b* Hydraulic chamber **2** of gear selector **5**/**7**
[0125] **35***c* Piston with shifter fork of gear selector **5**/**7**
[0126] **26** Position sensor of gear selector **5**/**7**
[0127] **37** 2/2-way admission and exhaust valve **1** for gear selector **5**/**7**
[0128] **38** Gear selector **6**/R
[0129] **38***a* Hydraulic chamber **1** of gear selector **6**/R
[0130] **38***b* Hydraulic chamber **2** of gear selector **6**/R
[0131] **38***c* Piston with shifter fork of gear selector **6**/R
[0132] **39** Position sensor of gear selector **6**/R
[0133] **40** 2/2-way admission and exhaust valve **1** for gear selector **6**/R
[0134] **41** 2/2-way admission and exhaust valve **1** for gear selector **2**/**4**
[0135] **42** Pressure sensor for clutch selector **2**
[0136] **43** Pressure sensor for clutch selector **1**
[0137] **44** Pump of cooling circuit HLP
[0138] **45** Non-return valve of cooling circuit HLP
[0139] **46** Reservoir of cooling circuit HLP
[0140] **47** Motor for pump of cooling circuit HLP
[0141] **48** Non-return valve of dual-action reciprocating piston-pump hydraulic chamber **1**
[0142] **49** Non-return valve of dual-action reciprocating piston-pump hydraulic chamber **2**
[0143] **50** Dual-action reciprocating piston-pump hydraulics
[0144] **51** Reservoir of dual-action reciprocating piston-pump hydraulics
[0145] **52** 2/2-way admission and exhaust valve **2** for gear selector **1**/**3**

[0146] **53** 2/2-way admission and exhaust valve **2** for gear selector **2**/**4**

[0147] **54** 2/2-way admission and exhaust valve **2** for gear selector **5**/**7**

[0148] **55** 2/2-way admission and exhaust valve **2** for gear selector **6**/R

[0149] HL Hydraulic line of an automatic transmission

[0150] HL_R Recirculation and lag of the hydraulics of an automatic transmission

[0151] HL**1** Hydraulic line **1** of a dual-action reciprocating piston

[0152] HL**2** Hydraulic line **2** of a dual-action reciprocating piston

[0153] HLP Hydraulic line of a cooling circuit with pump

[0154] HLK**1** Hydraulic line **1** of a cooling circuit with dual-action reciprocating piston pump

[0155] HLK**2** Hydraulic line **2** of a cooling circuit with dual-action reciprocating piston pump

[0156] LK**1** Multi-plate clutch **1**

[0157] LK**2** Multi-plate clutch **2**

1. A shift gearbox, comprising:

a control unit;

at least one electric motor drive;

multiple shift gearbox units, including at least two clutch units; and

at least one piston-cylinder unit, driven by a respective one of the electric motor drives, and which has a piston and is connected via hydraulic lines to the multiple shift gearbox units, and which is configured to shift the multiple shift gearbox units, wherein the piston of the piston-cylinder unit is in the form of a dual-action reciprocating piston, wherein the dual-action reciprocating piston sealingly separates two working chambers of the dual-action reciprocating piston from each other, wherein each working chamber is connected via a respective main hydraulic line to one respective clutch, and at least one of the two working chambers of the dual-action reciprocating piston or at least one main hydraulic line is enabled to be connected to a reservoir hydraulically via a switch valve.

2. The shift gearbox according to claim **1**, wherein at least one of the working chambers of the at least one piston-cylinder unit is hydraulically connected via a hydraulic line with at least one gear selector, wherein the hydraulic connection thusly established between the at least one of the working chambers and the at least one gear selector is enabled to be optionally shut off by means of a valve associated with the at least one gear selector.

3. The shift gearbox according to claim **1**, further comprising at least two main hydraulic lines that are enabled to be hydraulically connected to each other via a switch valve, wherein the at least two main hydraulic lines are connected to hydraulic feed lines to the two working chambers of the piston-cylinder unit.

4. The shift gearbox according to claim **3**, wherein pressure stored in a shift gearbox unit a clutch is used to support driving of the piston of the piston-cylinder unit, wherein the pressure from the respective shift gearbox unit is fed via an open valve in a respective one of the hydraulic feed lines and an associated main hydraulic line into one of the working chambers of the piston-cylinder unit, wherein the piston is driven by means of the electric motor drive for shrinking the other working chamber of the piston-cylinder unit, whereby hydraulic volume is displaced into another main hydraulic line or pressure is built up in the another main hydraulic line.

5. The shift gearbox according to claim **1**, wherein the piston has two different-sized piston active surface areas, in a ratio of between 1.5:1 and 2.5:1, which delimit the two working chambers.

6. The shift gearbox according claim **5**, wherein the different surface areas of the piston are utilised to reduce torque, with slip control of one of the clutches, wherein a valve associated with the respective one of the clutches is open during pressure build-up and/or pressure reduction in the respective clutch.

7. The shift gearbox according to claim **1**, wherein the dual-action reciprocating piston has two differently-sized hydraulically active piston surface areas, and a faster pressure build-up or volume delivery takes place via the working chamber delimited by the larger piston active surface area.

8. The shift gearbox according to claim **1**, further comprising a switch valve, through which, in its open position, the two working chambers are hydraulically connected to each other.

9. The shift gearbox according to claim **1**, wherein a pressure reduction takes place via one working chamber of the piston-cylinder unit and simultaneously a pressure build-up takes place via the other working chamber of the piston-cylinder unit, or a volume of the working chamber is delivered via valves to the reservoir and thus only pressure reduction takes place.

10. The shift gearbox according to claim **1**, wherein the control unit is configured to control the electric motor drive for shifting at least one of the shift gearbox units, wherein a control variable for controlling the drive is a rotation angle of the drive, a motor current flowing through the drive, a position of the piston and/or a travel of the piston, wherein the piston thereby delivers a required hydraulic volume to or from the at least one of the shift gearbox units.

11. The shift gearbox according to claim **1**, wherein pressure reduction in a shift gearbox unit takes place by opening a valve associated with the respective shift gearbox unit and the switch valve that connects the main hydraulic line coupled to the respective shift gearbox unit with the reservoir.

12. The shift gearbox according to claim **11**, wherein pressure regulation during pressure build-up and/or reduction in a shift gearbox unit takes place using a signal from a sensor associated with the respective shift gearbox unit.

13. The shift gearbox according to claim **1**, wherein at least one clutch is cooled by means of a coolant, wherein the coolant is delivered by means of the electric motor drive or a separate drive configured to drive a pump.

14. The shift gearbox according to claim **1**, further comprising a further piston, which sealingly separates two further working chambers from each other, wherein the further working chambers are connected via hydraulic lines with wet clutches for their cooling, wherein fluid pumped from the further working chambers reaches a further reservoir via the wet clutches and is drawn from the further reservoir into the further working chambers via suction lines and non-return valves.

15. The shift gearbox according to claim **14**, wherein the electric motor drive is configured to drive the further piston, wherein the piston is rigidly connected with the further piston, via a piston rod.

16. The shift gearbox according to claim **2**, wherein for shifting a gear selector, a predetermined quantity of hydraulic medium is delivered via the valve associated with the gear selector to a respective first working chamber of the gear selector from a working chamber of the dual-action reciprocating piston and simultaneously hydraulic medium is taken from the other working chamber of the dual-action reciprocating piston and/or discharged via one of the switch valves into the reservoir.

17. The shift gearbox according to claim **1**, wherein control for the electric motor drive uses at least one map, in the form of a pressure-volume characteristic.

18. The shift gearbox according to claim **17**, wherein the control uses a model for pressure calculation, wherein the model for determining a control variable for the electric motor drive for a pressure to be controlled in a clutch unit takes account of at least a motor current, and a clutch spring stiffness.

19. The shift gearbox according to claim **1**, wherein the shift gearbox further includes at least one pressure sensor for control compensation or for pressure control of clutch pressure.

20. The shift gearbox according to claim **1**, wherein at least one hydraulic feed line branches from or extends a hydraulic line that connects a main hydraulic line with one first working chamber of a shift gearbox unit, wherein, for optional shutting off of the hydraulic feed line, a switchable valve is arranged therein.

21. The shift gearbox according to claim **1**, wherein a first working chamber of a shift gearbox unit is connected via a hydraulic feed line with a main hydraulic line, wherein a second working chamber of the respective shift gearbox unit is connected via a further hydraulic feed line with another main hydraulic line, wherein in one of or in both the feed lines a switchable valve is arranged for optional opening and shutting off of the feed line.

22. The shift gearbox according to claim **1**, wherein one more of the at least one shift gearbox units has or have a position sensor or positioning sensor.

23. The shift gearbox according to claim **22**, wherein signals from the position sensor or positioning sensor are used for controlling the electric motor drive and/or for calibrating a control and a simulation model.

24. The shift gearbox according to claim **22**, wherein pressure reduction in a shift gearbox unit takes place via the hydraulic feed line and the main hydraulic line coupled to the respective shift gearbox unit, wherein a switching valve arranged in the hydraulic feed line is controlled through evaluation of a signal from the position or positioning sensor for pressure reduction, wherein the switching valve is opened for a predetermined time or by means of pulse-width modulation (PWM).

25. The shift gearbox according to claim **22**, wherein the position or positioning sensor has a discrete configuration, which is used in control solely for checking positions of a gear or clutch selector or for leak diagnosis.

26. The shift gearbox according to claim **1**, wherein at least two shift gearbox units are simultaneously shifted in multiplex mode, wherein shifting of each of the at least two shift gearbox units takes place in small partial steps, carried out alternately for each shift gearbox unit.

27. The shift gearbox claim **1**, wherein the shift gearbox has only two clutch selectors, without a gear selector.

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