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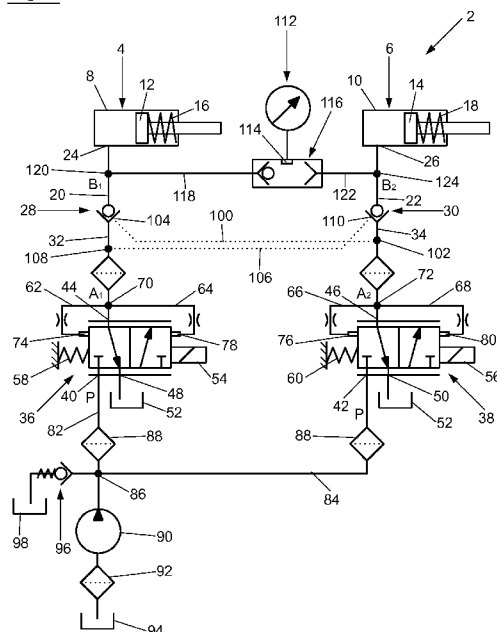
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(54) Title: HYDRAULIC CONTROL DEVICE FOR CONTROLLING A DUAL CLUTCH AND METHOD FOR HYDRAULIC CONTROL OF A DUAL CLUTCH

Fig. 1



(57) Abstract: The present invention relates to a hydraulic control device (2) for controlling a dual clutch with a hydraulically operated first actuator (4) for a first clutch of the dual clutch, a first proportional valve (36) comprising an input (40) to which a main hydraulic pressure (P) is applied, and comprising an output (44) from which a first hydraulic output pressure (A1) that can be adjusted by the first proportional valve (36) can be taken and which is connected to an input (24) of the first actuator (4) via a first shut-off valve (28) for blocking the flow towards the output (44) of the first proportional valve (36), a hydraulically operated second actuator (6) for a second clutch of the dual clutch and a second proportional valve (38) comprising an input (42) to which a main hydraulic pressure (P) is applied, and comprising an output (46) at which a second hydraulic output pressure (A2) that can be adjusted by the second proportional valve (38) can be taken and which is connected to an input (26) of the second actuator (6) via a second shut-off valve (30) for blocking the flow towards the output (46) of the second proportional valve (38), wherein the first shut-off valve (28) can be unblocked by the second hydraulic output pressure (A2) and the second shut-off valve (30) can be unblocked by the first hydraulic output pressure (A1). Moreover, the present invention relates to a method for the hydraulic control of a dual clutch.

HYDRAULIC CONTROL DEVICE FOR CONTROLLING A DUAL CLUTCH AND METHOD FOR HYDRAULIC CONTROL OF A DUAL CLUTCH

5 DESCRIPTION

The present invention relates to a hydraulic control device for controlling a dual clutch as well as a method for hydraulic control of a dual clutch.

From practice, control devices for controlling a dual clutch are known that
10 comprise a hydraulically operated first actuator for a first clutch of the dual clutch and a hydraulically operated second actuator for a second clutch of the dual clutch. The two actuators are each alternately subjected to an output pressure of a first proportional valve associated with the first actuator or of a second proportional valve associated with the second actuator. The proportional valves have the advantage here that the
15 respective output pressure can be slowly increased or reduced, so that the respective clutch is not closed or opened in an abrupt manner but rather can be opened or closed slowly and steadily. Nevertheless, there is the disadvantage that the individual proportional valve has to be operated permanently in order to keep the associated actuator permanently in the open or closed position, depending on whether it is a
20 normally engaged or normally disengaged clutch of the dual clutch. Moreover, relatively complex control of the proportional valves is necessary in order to change one actuator slowly into the closed position and the other actuator slowly into the open position at the same time. Moreover, the known control devices are associated with a relatively high leakage loss.

25 It is the object of the present invention to provide a hydraulic control device for controlling a dual clutch that guarantees low energy consumption, has a simple design and ensures substantially leakage-free operation of the dual clutch and the control device. It is also the object of the present invention to provide a method for the hydraulic control of a dual clutch that guarantees low energy consumption, that is
30 simple to implement and that ensures substantially leakage-free operation.

This object is achieved by the features specified in claims 1 and 9. Advantageous embodiments of the invention are the subject matter of the dependent claims.

The hydraulic control device according to the invention for controlling a dual clutch, especially in an automatic dual clutch transmission or in a hybrid drive
35 arrangement, comprises a hydraulically operated first actuator for a first clutch of the dual clutch and a hydraulically operated second actuator for a second clutch of the dual clutch. A first proportional valve associated with the first actuator comprises an input to which a main hydraulic pressure is applied and an output from which a first

hydraulic output pressure that is adjustable by the first proportional valve can be taken. The output of the first proportional valve is connected to an input of the first actuator by means of a first shut-off valve for blocking the flow towards the output of the first proportional valve. In a corresponding manner a second proportional valve associated with the second actuator is provided, which comprises an input to which a main hydraulic pressure is applied, which can correspond e.g. to the main hydraulic pressure at the input of the first proportional valve, and an output at which a second hydraulic output pressure that can be adjusted by the second proportional valve can be taken. The output of the second proportional valve is connected to an input of the second actuator by means of a second shut-off valve for blocking the flow towards the output of the second proportional valve. The main hydraulic pressure mentioned herein can e.g. be the system pressure of an existing hydraulic system. On the other hand, the shut-off valve can be a non-return valve. The first and second shut-off valves are designed such that the first shut-off valve between the output of the first proportional valve and the input of the first actuator can be unblocked by the second hydraulic output pressure that can be adjusted by the second proportional valve, whereas the second shut-off valve between the output of the second proportional valve and the input of the second actuator can be unblocked by the first hydraulic output pressure that can be adjusted by the first proportional valve.

Using the first proportional valve or by its operation, the first hydraulic output pressure can be steadily increased in order to steadily raise a first operating pressure on the first actuator until the desired first operating pressure on the first actuator has been achieved. Following this, the first hydraulic output pressure can be reduced, e.g. by ending the operation of the first proportional valve, wherein the first hydraulic operating pressure acting on the first actuator is maintained as a result of shutting off the first shut-off valve. Thus on the one hand further energy intake by the first proportional valve is no longer necessary in order to maintain the first hydraulic operating pressure acting on the first actuator, and on the other hand the first shut-off valve guarantees substantially leakage-free operation. This applies in a corresponding manner to the second actuator. Thus the second hydraulic output pressure can be slowly increased by the second proportional valve by operating the second proportional valve in order to correspondingly raise a second hydraulic operating pressure acting on the second actuator until the second hydraulic operating pressure has reached the desired level, in order to then reduce the second hydraulic output pressure, e.g. by ending the operation of the second proportional valve, wherein the second shut-off valve blocks the flow towards the output of the second proportional valve while maintaining the second hydraulic operating pressure acting on the second actuator.

Moreover, a simple change of the first actuator into its closed position or open position with a simultaneous change of the second actuator into its open position or closed position is possible, especially in that the first shut-off valve can be unblocked by the second hydraulic output pressure and the second shut-off valve can be unblocked by the first hydraulic output pressure. Thus not only is a control device provided whose proportional valves can guarantee low energy or current consumption and substantially leakage-free operation, but a particularly simple design of the hydraulic control device is also achieved.

In a preferred embodiment of the control device according to the invention, the first shut-off valve can be unblocked by a second hydraulic output pressure that is greater than a predetermined second output pressure limit value, whereas the second shut-off valve can be unblocked by a first hydraulic output pressure that is greater than a predetermined first output pressure limit value. The predetermined first output pressure limit value and the predetermined second output pressure limit value can be of equal magnitude. With said embodiment it is preferred if the predetermined first output pressure limit value is less than a maximum second hydraulic operating pressure acting on the second actuator, while the predetermined second output pressure limit value is less than a maximum first hydraulic operating pressure acting on the first actuator. This can be achieved e.g. by suitable design of the first and second shut-off valves.

In an advantageous embodiment of the control device according to the invention, the first shut-off valve can be subjected to the second hydraulic output pressure via a first control line and the second shut-off valve can be subjected to the first hydraulic output pressure via a second control line.

In a particularly advantageous embodiment of the control device according to the invention, the first proportional valve and the second proportional valve are each biased into a closed position, in which a flow between the input and the output of the respective proportional valve is blocked. Thus no energy consumption or current consumption at the proportional valves is necessary in order to reduce the first or second hydraulic output pressure or to lower it to zero, so that the respective proportional valve can remain not operated while the respective associated shut-off valve may cause the first or second hydraulic operating pressure acting on the first or second actuator to be maintained. The bias into the closed position is preferably caused by means of a spring element in this case.

In another preferred embodiment of the control device according to the invention, the first proportional valve and the second proportional valve each comprise a second output that is associated with an open hydraulic fluid container. The second output of the respective proportional valve preferably has a flow connection to the output of the

respective proportional valve in the previously mentioned closed position. If e.g. the first shut-off valve is unblocked by increasing the second hydraulic output pressure on the second proportional valve, then the hydraulic medium can flow from the first actuator into the open hydraulic fluid container via the opened first shut-off valve and the first proportional valve in its closed position, so that a change of the position of the first actuator is possible. If on the other hand the second shut-off valve is unblocked by increasing the first hydraulic output pressure on the first proportional valve, then the hydraulic medium can flow from the second actuator into the open hydraulic fluid container via the opened second shut-off valve and the second proportional valve in the closed position, so that the position of the second actuator can be changed. A common open hydraulic fluid container for the two second outputs of the proportional valves also comes into consideration here.

In another preferred embodiment of the control device according to the invention, the first proportional valve and the second proportional valve are each in the form of an electromagnetically operated solenoid valve with variable control force, i.e. of a so-called VFS (Variable Force Solenoid). Such solenoid valves are characterized by precise control of the output pressure and can be operated with corresponding energy consumption or current consumption.

In another advantageous embodiment of the control device according to the invention, the first and second shut-off valves are each in the form of a non-return valve. It has been found to be advantageous if the non-return valve is in the form of a spring-loaded non-return valve in order to maintain the highest possible hydraulic operating pressure acting on the respective actuator if the output pressure, from which the hydraulic operating pressure results, is reduced; in particular, particularly rapid closure or blocking by the shut-off valve is guaranteed. In this connection a spring-loaded non-return valve in the form of a ball valve has especially proved to be advantageous.

In another preferred embodiment of the control device according to the invention, the first actuator and the second actuator are open or arranged in an open position in the unpressurized state. It can also be said that the first clutch and the second clutch are each a normally disengaged clutch, with which no hydraulic pressurization is necessary in order to keep it in the disengaged position.

In another advantageous embodiment of the control device according to the invention, the first actuator and the second actuator are each in the form of a single action piston-cylinder-arrangement, possibly with a return stroke spring for changing the respective actuator into the open position.

As previously mentioned, essentially leakage-free operation of the actuators is possible thanks to the shut-off valves. Nevertheless, leakage losses can occur in the region depending on the design of the actuators, which can ultimately result in a reduction of the hydraulic operating pressure acting on the respective actuator. In order to enable the monitoring of such a pressure loss, in another preferred embodiment of the control device according to the invention a pressure measurement device is provided, by means which on the one hand a first hydraulic operating pressure acting on the first actuator on the side of the first shut-off valve facing away from the first proportional valve can be measured and on the other hand a second hydraulic operating pressure acting on the second actuator on the side of the second shut-off valve facing away from the second proportional valve can be measured.

In another particularly preferred embodiment of the control device according to the invention, the pressure measurement device is designed such that an unintended, possible leakage-induced, reduction of the first or second hydraulic operating pressure can be compensated by the first or second proportional valve. On the one hand the pressure measurement device thus interacts with the first proportional valve such that this is operated, possibly energized, by increasing the first hydraulic output pressure if the first hydraulic operating pressure falls below a predetermined first operating pressure limit value, and on the other hand interacts with the second proportional valve such that this is operated, possibly energized, by increasing the second hydraulic output pressure if the second hydraulic operating pressure falls below a predetermined second operating pressure limit value. The first operating pressure limit value can correspond to the second operating pressure limit value, wherein it is further preferred if the first and second operating pressure limit values are smaller than the maximum first and second hydraulic output pressures. With this embodiment it is further preferred if the pressure measurement device interacts with the first proportional valve such that the first hydraulic output pressure is reduced, possibly by ending the operation or the energizing of the first proportional valve if the first hydraulic operating pressure exceeds the predetermined first operating pressure limit value again, and on the other hand interacts with the second proportional valve such that the second hydraulic output pressure is reduced, possibly by ending the operation or the energizing of the second proportional valve, if the second hydraulic operating pressure exceeds the predetermined second operating pressure limit value again.

In principle, the previously mentioned pressure measurement device could comprise a pressure sensor for measurement of the first hydraulic operating pressure and another pressure sensor for measurement of the second hydraulic operating pressure. However, in order to simplify the design of the control device according to

the invention, the pressure measurement device in another advantageous embodiment comprises a common pressure sensor, by means of which the first hydraulic operating pressure or the second hydraulic operating pressure can be optionally measured. This reflects the fact that as a rule only one actuator is kept continuously unpressurized, whereas the other actuator is subjected to pressure and both the first actuator and also the second actuator are arranged in the closed position or the open position as required in a transition period.

In order to achieve the use of a common pressure sensor in a particularly simple manner, the control device according to the invention in another preferred embodiment of the same comprises a changeover device, by means of which a pressure difference between the first hydraulic operating pressure and the second hydraulic operating pressure can be recorded and which interacts with the common pressure sensor such that the same measures the higher hydraulic operating pressure of the first and second hydraulic operating pressures. The changeover device is preferably a changeover valve or a double non-return valve. The changeover device can e.g. be connected to the side of the first shut-off valve remote from the first proportional valve on the one hand and to the side of the second shut-off valve remote from the second proportional valve on the other hand in order to record said pressure difference between the first hydraulic operating pressure and the second hydraulic operating pressure and to subject the common pressure sensor to the first or second hydraulic operating pressure, depending on which of the two hydraulic operating pressures is higher or greater.

The method according to the invention for the hydraulic control of a dual clutch comprises the steps of the method mentioned below. Initially a first hydraulic output pressure of a first proportional valve is increased by operation of the first proportional valve. A first actuator for a clutch of the dual clutch is subjected to said first hydraulic output pressure by increasing a first hydraulic operating pressure acting on the first actuator. In this phase the first hydraulic output pressure essentially corresponds to the first hydraulic operating pressure acting on the first actuator. If the desired first hydraulic operating pressure is achieved, then a return flow of the hydraulic medium from the first actuator to the first proportional valve is prevented by means of a first shut-off valve while maintaining the achieved first hydraulic operating pressure and the first hydraulic output pressure is reduced. The reduction of the first hydraulic output pressure is preferably carried out here by ending the operation, i.e. the energy supply or current supply as applicable, of the first proportional valve. Moreover, it is particularly preferred if the return flow of the hydraulic medium from the first actuator to the first proportional valve is automatically prevented by the reduction of the first hydraulic output pressure. Thus with said particularly preferred design variants, active control of

the first shut-off valve is not necessary, which simplifies the design of the control device or the implementation of the method for the hydraulic control of the dual clutch.

The dual clutch can then be operated in the position of the first actuator thus achieved over a longer time period without a supply of energy or current to the first proportional valve being necessary, while the first shut-off valve ensures substantially leakage-free operation without a leakage-induced reduction of the first hydraulic operating pressure acting on the first actuator. However, should a change of the clutch positions or a changeover of the clutch positions be carried out, then in a subsequent step of the method a second hydraulic output pressure of a second proportional valve is increased by the operation of the second proportional valve, i.e. by the supply of energy or current as applicable. A second actuator for a second clutch of the dual clutch is subjected to said second hydraulic output pressure by increasing a second hydraulic operating pressure acting on the second actuator. Moreover, the first shut-off valve is subjected to the second hydraulic output pressure via a first control line by opening the first shut-off valve, so that accordingly a clutch changeover takes place. Regarding the advantages of the method according to the invention as well as its advantageous embodiments described below, reference is made to the advantages of the hydraulic control device described above, which apply in a corresponding manner to the method for the hydraulic control of a dual clutch.

In one advantageous embodiment of the method according to the invention, in another step of the method after achieving the desired second hydraulic operating pressure, which acts on the second actuator, a return flow of the hydraulic medium from the second actuator to the second proportional valve is prevented by means of a shut-off valve while maintaining the achieved second hydraulic operating pressure and the second hydraulic output pressure of the second proportional valve is reduced, wherein the reduction of the second hydraulic output pressure is preferably carried out by ending the operation, i.e. the supply of energy or current as applicable, of the second proportional valve. It is also particularly preferred in this case if the reduction of the second hydraulic output pressure already automatically prevents a reverse flow of the hydraulic medium from the second actuator to the second proportional valve, in order to simplify the method. This can be achieved – as above with the previously described embodiment – by means of a simple shut-off valve that allows a flow in only one direction.

In a particularly preferred embodiment of the method according to the invention, when the first actuator is subjected to the first hydraulic output pressure by increasing the first hydraulic operating pressure, the second shut-off valve is also subjected to the first hydraulic output pressure via a second control line by opening the second shut-off

valve. In this way the clutch changeover can also be carried out simply if the first actuator is subjected to the first hydraulic output pressure by increasing the first hydraulic operating pressure.

In one advantageous embodiment of the method according to the invention, the first hydraulic operating pressure is measured and the first hydraulic output pressure is increased by the operation, i.e. by supplying energy or current as applicable, of the first proportional valve following the previously occurring reduction of the first hydraulic output pressure if the first hydraulic operating pressure falls below a predetermined first operating pressure limit value. In this way pressure losses at the first actuator are compensated. With this embodiment it is further preferred if the first hydraulic output pressure is reduced again if the first hydraulic operating pressure exceeds the predetermined first operating pressure limit value again.

In another advantageous embodiment of the method according to the invention, the second hydraulic operating pressure is measured and the second hydraulic output pressure is increased by operating the second proportional valve, i.e. by supplying energy or current as applicable, following the reduction of the second hydraulic output pressure if the second hydraulic operating pressure falls below a predetermined second operating pressure limit value, in order to compensate a possibly leakage-induced pressure loss in the region of the second actuator. With this embodiment it is also preferable if the second hydraulic output pressure is reduced if the second hydraulic operating pressure exceeds the predetermined second operating pressure limit value again.

In a particularly advantageous embodiment of the method according to the invention, a pressure difference between the first hydraulic operating pressure and the second hydraulic operating pressure is recorded and only the higher hydraulic operating pressure of the first and second hydraulic operating pressures is measured. The measurement both of the first and also of the second hydraulic operating pressure is preferably carried out here by means of a common pressure sensor in order to simplify the process and the design of the control device operated according to the method.

The invention is explained in detail below using an exemplary embodiment with reference to the accompanying figures. In the figures:

Fig. 1 shows a circuit diagram of an embodiment of the hydraulic control device according to the invention for controlling a dual clutch and

Fig. 2 shows a schematic illustration of pressure profiles against time for illustrating the operation of the hydraulic control device according to Fig. 1 and of the method underlying the control device according to Fig. 1.

Fig. 1 shows an embodiment of the hydraulic control device 2 for controlling a dual clutch that is not illustrated in detail. The control device 2 comprises a hydraulically operated first actuator 4 for a first clutch of the dual clutch and a hydraulically operated second actuator 6 for a second clutch of the dual clutch. The first actuator 4 and the second actuator 6 are each in the form of a single action piston-cylinder-arrangement of a cylinder 8, 10 and a piston 12, 14 displaceably disposed in the cylinder 8, 10. A return stroke spring 16, 18 is associated with each piston 12, 14, wherein the return stroke spring 16, 18 causes both the first actuator 4 and also the second actuator 6 to be opened or changed to an open position in the unpressurized state.

The piston 12 or 14 can be brought into a closed position against the force of the return stroke spring 16 or 18 by means of a first hydraulic operating pressure B_1 or second hydraulic operating pressure B_2 . For this purpose, the first actuator 4 or the second actuator 6 is connected to a flow line 20, 22, which on the one hand is connected to an input 24 or 26 of the first or second actuator 4, 6 and on the other hand to a first shut-off valve 28 or a second shut-off valve 30. The shut-off valves 28, 30 are each in the form of a non-return valve, wherein the non-return valve can also be in the form of a spring-loaded non-return valve, preferably a spring-loaded ball valve, in contrast to the illustration in Fig. 1. On its side remote from the flow line 20 or 22, the first or second shut-off valve 28, 30 is connected to a flow line 32 or 34 that leads to a first proportional valve 36 or a second proportional valve 38.

The two proportional valves 36, 38 each comprise an input 40 or 42, an output 44 or 46, to which the flow line 32 or 34 is connected, and a second output 48 or 50, wherein the second outputs 48, 50 are each associated with an open hydraulic fluid container 52 or lead to the same. The two proportional valves 36, 38 are each in the form of an electromagnetically operated solenoid valve with variable control force. The proportional valves 36, 38 thus each comprise an electromagnet 54 or 56, wherein the proportional valve 36 or 38 can be displaced by energizing the electromagnet 54 or 56. The control force of the electromagnet 54 or 56 acts against a restoring force of a spring element 58 or 60 of the respective proportional valve 36, 38, wherein the first proportional valve 36 and the second proportional valve 38 are biased by means of the associated spring element 58, 60 into a closed position, in which a flow between the input 40 or 42 and the output 44 or 46 of the respective proportional valve 36, 38 is blocked. Moreover, the second output 48 or 50 of the respective proportional valve 36, 38 in said closed position has a flow connection to the output 44 or 46 of the respective proportional valve 36, 38. Operation of the proportional valves 36, 38 can thus take place by energizing the associated electromagnet 54 or 56, by means of which the

proportional valve 36 or 38 is more or less strongly displaced into an open position. Moreover, two control lines 62, 64 or 66, 68 are associated with each of the two proportional valves 36, 38 and lead, starting from a line connection 70 or 72 to the flow line 32 or 34, to a first control input 74 or 76 and a second control input 78 or 80 of the first or second proportional valve 36 or 38. Thus the control lines 62, 64 interact with the left and right end faces of the first proportional valve 36, whereas the control lines 66, 68 interact with the left or right end face of the second proportional valve 38. Furthermore, chokes (no reference characters) are provided in the control lines 62, 64, 66, 68.

A first hydraulic output pressure A_1 that can be adjusted by means of the first proportional valve 36 can be taken via the flow line 32 at the output 44 of the first proportional valve 36, while a second hydraulic output pressure A_2 that can be adjusted by means of the second proportional valve 38 can be taken via the flow line 34. The first shut-off valve 28 is disposed between the flow line 20 and the flow line 32, such that the same blocks a flow from the flow line 20 into the flow line 32 or prevents a flow from the input 24 of the first actuator 4 to the output 44 of the first proportional valve 36. In a corresponding manner, the second shut-off valve 30 is disposed between the flow line 22 and the flow line 34 such that the same blocks the flow from the flow line 22 towards the flow line 34 or prevents the flow from the input 26 of the second actuator 6 to the output 46 of the second proportional valve 38. For the sake of completeness it should be mentioned that a filter is disposed within the flow lines 32, 34 in each case.

A main hydraulic pressure P , which can also be referred to as the system pressure, is applied both at the input 40 of the first proportional valve 36 and also at the input 42 of the second proportional valve 38. For this purpose, the inputs 40, 42 are each connected to a flow line 82, 84, the flow lines being combined at a line junction 86 on their side remote from the respective proportional valve 36 or 38, wherein a filter 88 is disposed within each of the flow lines 82, 84. The main hydraulic pressure P at the inputs 40, 42 is generated by means of a hydraulic pump 90 associated with the line junction 86, which is supplied with hydraulic fluid via a filter 92 from a hydraulic fluid container 94. In order to prevent inadmissibly high pressures at the inputs 40, 42 or to limit the main hydraulic pressure P , furthermore a pressure relief valve 96 is provided, which is in the form of a spring-loaded non-return valve here and is associated with the line junction 86. On its side remote from the line junction 86, the pressure relief valve 96 is connected to a hydraulic fluid container 98.

The first shut-off valve 28 can be unblocked by the second hydraulic output pressure A_2 , while the second shut-off valve 30 can be unblocked by the first hydraulic

output pressure A_1 . For this purpose, a first control line 100 is provided that opens at a line junction 102 in the flow line 34 and leads to a control input 104 at the first shut-off valve 28, while furthermore a second control line 106 is provided that opens at a line junction 108 in the flow line 32 and leads to a control input 110 at the second shut-off valve 30. Thus the first shut-off valve 28 can be subjected to the second hydraulic output pressure A_2 via the first control line 100, while the second shut-off valve 30 can be subjected to the first hydraulic output pressure A_1 via the second control line 106. It should be noted that in principle chokes or similar can also be provided in the control lines 100, 106 in order to reduce the second or first hydraulic output pressure A_2 or A_1 used to open the first or second shut-off valve 28 or 30.

A pressure measurement device 112 is further provided, which is indicated in Fig. 1 using a manometer. On the one hand the first hydraulic operating pressure B_1 acting on the side of the first shut-off valve 28 remote from the first proportional valve 36 and on the other hand the second hydraulic operating pressure B_2 acting on the second actuator 6 on the side of the second shut-off valve 30 remote from the second proportional valve 38 can be measured by means of the pressure measurement device 112.

In principle, the pressure measurement device 112 could comprise two pressure sensors, wherein one pressure sensor is used to record the first hydraulic operating pressure B_1 , while the second pressure sensor is used to record the second hydraulic operating pressure B_2 . In the embodiment according to Fig. 1, however, the pressure measurement device 112 comprises a common pressure sensor 114, by means of which the first hydraulic operating pressure B_1 or the second hydraulic operating pressure B_2 can be selectively measured. In order to enable a changeover between the measurement of the first hydraulic operating pressure B_1 and the measurement of the second hydraulic operating pressure B_2 by the common pressure sensor 114, in the illustrated embodiment a changeover device 116 is provided, wherein the changeover device 116 is in the form of a changeover valve or double non-return valve here. The changeover device 116 in the form of a changeover valve or double non-return valve is on the one hand connected by means of a first measurement line 118 to a line junction 120 in the flow line 20 and on the other hand by means of a second measurement line 122 to a line junction 124 in the flow line 22. Consequently, the first hydraulic operating pressure B_1 can thus be recorded via the first measurement line 118, while the second hydraulic operating pressure B_2 can be recorded via the second measurement line 122. The changeover device 116 in the form of the changeover valve or dual non-return valve thus enables the recording of a pressure difference between the first hydraulic operating pressure B_1 and the second hydraulic operating pressure B_2 , wherein the

common pressure sensor 114 interacts with the changeover device 116 such that this only measures the higher hydraulic operating pressure of the two hydraulic operating pressures B_1 and B_2 . If e.g. the second hydraulic operating pressure B_2 is higher than the first hydraulic operating pressure B_1 , then the changeover device 116 in the form of the changeover or dual non-return valve adopts the position shown in Fig. 1, in which the first measurement line 118 is closed by the changeover device 116, while the second hydraulic operating pressure B_2 can be measured by the common pressure sensor 114 of the pressure measurement device 112 via the second measurement line 122. If by contrast the first hydraulic operating pressure B_1 is greater than the second hydraulic operating pressure B_2 , then the switch position of the changeover device 116 in the form of a changeover valve or double non-return valve changes such that the second measurement line 122 is blocked by the changeover device 116, while the first hydraulic operating pressure B_1 can be measured by the common pressure sensor 114 of the pressure measurement device 112 via the first measurement line 118.

Irrespective of whether the pressure measurement device 112 comprises a pressure sensor for recording the first hydraulic operating pressure B_1 and another pressure sensor for recording the second hydraulic operating pressure B_2 or the common pressure sensor 114 with changeover device 116 shown in Fig. 1, the pressure measurement device 112 interacts with the first proportional valve 36 such that the same is operated with an increase of the first hydraulic output pressure A_1 by energizing the electromagnet 54 if the first hydraulic operating pressure B_1 falls below a predetermined first operating pressure limit value bg_1 . Accordingly, the pressure measurement device 112 interacts with the second proportional valve 38 such that the second proportional valve 38 is operated with an increase of the second hydraulic output pressure A_2 by energizing the electromagnet 54 if the second hydraulic operating pressure B_2 falls below a predetermined second operating pressure limit value bg_2 .

Further features of the control device 2 as well as its operation are explained in detail below using a description of an embodiment of the method underlying the control device 2 for the hydraulic control of a dual clutch with reference to Figs. 1 and 2, wherein Fig. 2 schematically illustrates the pressure profiles against time of the first hydraulic output pressure A_1 , of the second hydraulic output pressure A_2 , of the first hydraulic operating pressure B_1 and of the second hydraulic operating pressure B_2 .

It is assumed that at the point in time t_0 the first actuator 4 is changed into the open position without pressure and thus by means of the return stroke spring 16, while the second actuator 6 is subjected to an increased second hydraulic operating pressure B_2 , so that the same is changed into the closed position against the restoring force of

the return stroke spring 18. Maintenance of the second hydraulic operating pressure B_2 , which is acting on the second actuator 6, is effected by the second shut-off valve 30, which prevents a return flow of the hydraulic medium from the input 26 of the second actuator 6 to the output 46 of the second proportional valve 38 in the shut-off position.

5 Both proportional valves 36, 38 are not operated, i.e. their electromagnets 54, 56 are not energized or supplied with energy, so that the two proportional valves 36, 38 are each in their previously described closed position. Because the second hydraulic operating pressure B_2 is greater than the first hydraulic operating pressure B_1 , the pressure measurement device 112 measures the second hydraulic operating pressure B_2
10 using the common pressure sensor 114 via the changeover device 116 and the second measurement line 122.

Should a clutch changeover take place, i.e. the first actuator 4 is changed into the closed position and the second actuator 6 is changed into the open position, then the first hydraulic output pressure A_1 at the first proportional valve 36 is increased by
15 operation of the first proportional valve 36 at the point in time t_1 . More accurately speaking, the electromagnet 54 of the previously not operated first proportional valve 36 is energized in order to change the first proportional valve 36 from its closed position into the subsequently different open position. This causes the first shut-off valve 28 to open, so that the first hydraulic operating pressure B_1 , which is acting on
20 the first actuator 4, is increased together with the first hydraulic output pressure A_1 . Thus the first actuator 4 is subjected to the first hydraulic output pressure A_1 by increasing the first hydraulic operating pressure B_1 acting on the first actuator 4, wherein the first hydraulic output pressure A_1 and the first hydraulic operating pressure B_1 are essentially of identical magnitude.

25 Moreover, the second shut-off valve 30 is also subjected to the first hydraulic output pressure A_1 via the second control line 106. If the first hydraulic output pressure A_1 set by the first proportional valve 36 exceeds a first output pressure limit value ag_1 at the point in time t_2 , then the first hydraulic output pressure A_1 causes opening or unblocking of the second shut-off valve 30 via the second control line 106, although a
30 relatively high second hydraulic operating pressure B_2 is still acting on the valve body of the second shut-off valve 30. Suitable control of the second shut-off valve 30 can be effected by suitable design of the same. By means of the unblocking or opening of the second shut-off valve 30, hydraulic medium can pass from the input 26 of the second actuator 6 via the second shut-off valve 30 and the second proportional valve 38, which
35 is in the closed position, to the second output 50 of the second proportional valve 38, from where it can flow out into the hydraulic fluid container 52. Thus the second hydraulic operating pressure B_2 consequently reduces, while the first hydraulic

operating pressure B_1 rises further because of the further increasing first hydraulic output pressure A_1 . The result of this is that the first actuator 4 is moved towards the closed position against the restoring force of the return stroke spring 16, whereas the second actuator 6 is moved into the open position because of the restoring force of the return stroke spring 18.

If the first hydraulic operating pressure B_1 exceeds the second hydraulic operating pressure B_2 at the point in time t_3 , then the changeover device 116 is changed over and from then on only the first hydraulic operating pressure B_1 is measured by means of the common pressure sensor 114 of the pressure measurement device 112.

If the desired first hydraulic operating pressure B_1 is reached at the point in time t_4 and the first actuator 4 thus changes into the closed position, then the first hydraulic output pressure A_1 is reduced again by ending the operation of the first proportional valve 36 or by ending the energization of the electromagnet 54 of the first proportional valve 36. Despite the reduction of the first hydraulic output pressure A_1 by resetting the first proportional valve 36 into its closed position by means of the spring element 58, a return flow of the hydraulic medium from the input 24 of the first actuator 4 via the first proportional valve 36 into the hydraulic fluid container 52 is inhibited by means of the first shut-off valve 28, so that the achieved first hydraulic operating pressure B_1 acting on the first actuator 4 is maintained. In this case the return flow of the hydraulic medium from the first actuator 4 to the first proportional valve 36 and moreover into the hydraulic fluid container 52 is automatically prevented by means of the previously mentioned reduction of the first hydraulic output pressure A_1 . Thus the first actuator 4 is held in the closed position reliably and - in relation to the first shut-off valve 28 - substantially leakage-free, without further energization of the electromagnet 54 of the first proportional valve 36 being necessary, while the second, in the meantime unpressurized actuator 6 is changed into the open position by the return stroke spring 18. The second proportional valve 38 also remains not operated or de-energized.

From this point in time, the control device 2 can be permanently operated without energy consumption or current consumption at the proportional valves 36, 38, wherein the first hydraulic operating pressure B_1 is measured and monitored by the pressure measurement device 112. Should the first hydraulic operating pressure B_1 , which keeps the first actuator 4 in its closed position, fall below the previously mentioned first operating pressure limit value bg_1 as a result of leakage losses or other causes at a point in time t_5 , then this is recorded by the pressure measurement device 112, which thereupon operates the first proportional valve 36 by renewed energization of the electromagnet 54 in order to increase the first hydraulic output pressure A_1 again. The first hydraulic output pressure A_1 is thereby increased such that the first shut-off valve

28 opens and the first hydraulic operating pressure B_1 rises again because of the increase of the first hydraulic output pressure A_1 until the first hydraulic operating pressure B_1 reaches the desired value above the first operating pressure limit value bg_1 at the point in time t_6 . This is also recorded by the pressure measurement device 112, which again terminates the operation of the first proportional valve 36 on reaching the desired first hydraulic operating pressure B_1 necessary for keeping the first actuator 4 in its closed position, so that the first proportional valve 36 is again de-energized. If the first hydraulic output pressure A_1 reduces as a result, then the first shut-off valve 28 again causes the now reproduced or achieved first hydraulic operating pressure B_1 to be maintained by returning to its shut-off position. The control device 2 can now be operated permanently again without supplying energy to the two proportional valves 36, 38.

Should a renewed clutch changeover be carried out or should the first actuator 4 be changed into the open position and should the second actuator 6 be changed into the closed position, then the second hydraulic output pressure A_2 of the second proportional valve 38 is increased by the operation of the second proportional valve 38 at the point in time t_7 . This takes place by energizing the electromagnet 56 of the second proportional valve 38, which is thereupon moved from the previously mentioned closed position and against the restoring force of the spring element 60. As a result of the increasing second hydraulic output pressure A_2 , the second shut-off valve 30 is unblocked or opened, so that the second actuator 6 is subjected to the second hydraulic output pressure A_2 by increasing the second hydraulic operating pressure B_2 acting on the second actuator 6. Moreover, the first shut-off valve 28 is subjected to the second hydraulic output pressure A_2 via the first control line 100. If the second hydraulic output pressure A_2 thereby exceeds a second output pressure limit value ag_2 at the point in time t_8 , then the second hydraulic output pressure A_2 causes unblocking or opening of the first shut-off valve 28 via the first control line 100, wherein the control pressure via the control line 100 can be significantly lower than the still prevailing first hydraulic operating pressure B_1 , which can be achieved by suitable design of the first shut-off valve 28. As a result hydraulic medium flows from the first actuator 4 via the opened first shut-off valve 28 and the first proportional valve 36 further arranged in the closed position into the hydraulic fluid container 52, so that the first hydraulic operating pressure B_1 reduces and the first actuator 4 is moved into the open position as a result of the restoring force of the return stroke spring 16. In contrast, the second actuator 6 is moved into the open position against the restoring force of the return stroke spring 18 by the increasing second hydraulic operating pressure B_2 . If the second hydraulic operating pressure B_2 exceeds the first hydraulic

operating pressure B_1 as a result, then the changeover device 116 causes the common pressure sensor 114 of the pressure measurement device 112 to now only measure and monitor the second hydraulic operating pressure B_2 via the second measurement line 122. If as a result the second hydraulic operating pressure B_2 reaches the desired value above the second operating pressure limit value bg_2 at the point in time t_{10} , then the same is recorded by the pressure measurement device 112, whereupon the second hydraulic output pressure A_2 is reduced again by ending the operation of the second proportional valve 38 or by ending the energization of the electromagnet 56, wherein the proportional valve is returned to its closed position because of the restoring force of the spring element 60. Despite the now reducing second hydraulic output pressure A_2 , the second hydraulic operating pressure B_2 acting on the second actuator 6 is now essentially maintained; in particular a return flow of the hydraulic medium from the second actuator 6 to the second proportional valve 38 and moreover into the hydraulic fluid container 52 is prevented by the second shut-off valve 30, which automatically returns to its shut-off position as a result of the reduction of the second hydraulic output pressure A_2 .

In the position of the control device 2 now achieved, in which the first actuator 4 is held in the open position by the return stroke spring 16, while the second actuator 6 is held in the closed position by the second hydraulic operating pressure B_2 , the control device 2 can be operated permanently and without energy supply to the proportional valves 36, 38 without leakage losses occurring in the region of the shut-off valves 28, 30. Should a leak or similar occur elsewhere, which results in a reduction of the second hydraulic operating pressure B_2 to below the previously mentioned second operating pressure limit value bg_2 , this is recorded by the pressure measurement device 112, which thereupon increases the second hydraulic output pressure A_2 by operating the second proportional valve 38 following the previous reduction of the second hydraulic output pressure A_2 . The pressure measurement device 112 thereby interacts with the second proportional valve 38, as has already been described in connection with the first proportional valve 36 at the points in time t_5 and t_6 , so as to eliminate the mentioned pressure loss again. Reference is made to the above description in this respect.

REFERENCE CHARACTER LIST

	2	hydraulic control device
	4	first actuator
5	6	second actuator
	8	cylinder
	10	cylinder
	12	piston
	14	piston
10	16	return stroke spring
	18	return stroke spring
	20	flow line
	22	flow line
	24	input
15	26	input
	28	first shut-off valve
	30	second shut-off valve
	32	flow line
	34	flow line
20	36	first proportional valve
	38	second proportional valve
	40	input
	42	input
	44	output
25	46	output
	48	second output
	50	second output
	52	hydraulic fluid container
	54	electromagnet
30	56	electromagnet
	58	spring element
	60	spring element

	62	control line
	64	control line
	66	control line
	68	control line
5	70	line junction
	72	line junction
	74	first control input
	76	first control input
	78	second control input
10	80	second control input
	82	flow line
	84	flow line
	86	line junction
	88	Filter
15	90	hydraulic pump
	92	filter
	94	hydraulic fluid container
	96	pressure relief valve
	98	hydraulic fluid container
20	100	first control line
	102	line junction
	104	control input
	106	second control line
	108	line junction
25	110	control input
	112	pressure measurement device
	114	common pressure sensor
	116	changeover device
	118	first measurement line
30	120	line junction
	122	second measurement line
	124	line junction

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	A ₁	first hydraulic output pressure
	A ₂	second hydraulic output pressure
	B ₁	first hydraulic operating pressure
	B ₂	second hydraulic operating pressure
5	P	main hydraulic pressure
	ag ₁	first output pressure limit value
	ag ₂	second output pressure limit value
	bg ₁	first operating pressure limit value
10	bg ₂	second operating pressure limit value
	t ₀ to t ₁₀	successive points in time

CLAIMS

1. Hydraulic control device (2) for controlling a dual clutch with
a hydraulically operated first actuator (4) for a first clutch of the dual clutch, a first
5 proportional valve (36), which comprises an input (40) to which a main hydraulic
pressure (P) is applied, and comprises an output (44), at which a first hydraulic output
pressure (A_1) that can be adjusted by the first proportional valve (36) can be taken and
which is connected to an input (24) of the first actuator (4) via a first shut-off valve
10 (28) for shutting off the flow towards the output (44) of the first proportional valve
(36),
a hydraulically operated second actuator (6) for a second clutch of the dual clutch and a
second proportional valve (38), which comprises an input (42) to which a main
hydraulic pressure (P) is applied, and comprises an output (46) at which a second
hydraulic output pressure (A_2) that can be adjusted by the second proportional valve
15 (38) can be taken and which is connected to an input (26) of the second actuator (6) via
a second shut-off valve (30) for shutting off the flow towards the output (46) of the
second proportional valve (38),
wherein the first shut-off valve (28) can be unblocked by the second hydraulic output
pressure (A_2) and the second shut-off valve (30) can be unblocked by the first
20 hydraulic output pressure (A_1).
2. Hydraulic control device (2) as claimed in claim 1, **wherein** the first
shut-off valve (28) can be subjected to the second hydraulic output pressure (A_2) via a
first control line (100) and the second shut-off valve (30) can be subjected to the first
25 hydraulic output pressure (A_1) via a second control line (106).
3. Hydraulic control device (2) as claimed in any one of claims 1 or 2,
wherein the first proportional valve (36) and the second proportional valve (38) are
each biased, preferably by means of a spring element (58; 60), into a closed position in
30 which a flow between the input (40; 42) and the output (44; 46) of the respective
proportional valve (36; 38) is blocked.
4. Hydraulic control device (2) as claimed in any one of the preceding
claims, **wherein** the first proportional valve (36) and the second proportional valve
35 (38) each have a second output (48; 50), which is associated with an open hydraulic
fluid container (52), wherein in the closed position the second output (48; 50) of the
respective proportional valve (36; 38) preferably has a flow connection to the output
(44; 46) of the respective proportional valve (36; 38).

5. Hydraulic control device (2) as claimed in any one of the preceding claims, **wherein** the first proportional valve (36) and the second proportional valve (38) are each in the form of an electromagnetically operated solenoid valve with variable control force or/and the first and second shut-off valves (28, 30) are each in the form of a non-return valve.

6. Hydraulic control device (2) as claimed in any one of the preceding claims, **wherein** the first actuator (4) and the second actuator (6) are open in the unpressurized state or/and each is in the form of a single action piston-cylinder arrangement, possibly with a return stroke spring (16; 18).

7. Hydraulic control device (2) as claimed in any one of the preceding claims, **wherein** a pressure measurement device (112) is provided, by means of which on the one hand a first hydraulic operating pressure (B_1) acting on the first actuator (4) on the side of the first shut-off valve (28) remote from the first proportional valve (36) can be measured and on the other hand a second hydraulic operating pressure (B_2) acting on the second actuator (6) on the side of the second shut-off valve (30) remote from second proportional valve (38) can be measured, wherein the pressure measurement device (112) preferably interacts on the one hand with the first proportional valve (36) such that the same is operated while increasing the first hydraulic output pressure (A_1) if the first hydraulic operating pressure (B_1) falls below a predetermined first operating pressure limit value (bg_1), and on the other hand interacts with the second proportional valve (38) such that the same is operated while increasing the second hydraulic output pressure (A_2) if the second hydraulic operating pressure (B_2) falls below a predetermined second operating pressure limit value (bg_2).

8. Hydraulic control device (2) as claimed in claim 7, **wherein** the pressure measurement device (112) comprises a common pressure sensor (114), by means of which the first hydraulic operating pressure (B_1) or the second hydraulic operating pressure (B_2) can be selectively measured, wherein preferably a changeover device (116) is provided, by means which a pressure difference between the first hydraulic operating pressure (B_1) and the second hydraulic operating pressure (B_2) can be recorded and which interacts with the pressure sensor (114) such that the pressure sensor (114) measures the higher hydraulic operating pressure of the first and second hydraulic operating pressures (B_1 , B_2), and the changeover device (116) is preferably in the form of a changeover valve or double non-return valve.

9. A method for the hydraulic control of a dual clutch with the steps of the method of

increasing a first hydraulic output pressure (A_1) of a first proportional valve (36) by
5 operating the first proportional valve (36),

subjecting a first actuator (4) for a first clutch of the dual clutch to the first hydraulic
output pressure (A_1) by increasing a first hydraulic operating pressure (B_1) acting on
the first actuator (4),

10 preventing a return flow of the hydraulic medium from the first actuator (4) to the first
proportional valve (36) by means of a first shut-off valve (28) while maintaining the
achieved first hydraulic operating pressure (B_1),

reducing the first hydraulic output pressure (A_1), preferably by ending the operation of
the first proportional valve (36), wherein prevention of the return flow of the hydraulic
medium from the first actuator (4) to the first proportional valve (36) is particularly
15 preferably caused by the reduction of the first hydraulic output pressure (A_1),

increasing a second hydraulic output pressure (A_2) of a second proportional valve (38)
by operating the second proportional valve (38),

subjecting a second actuator (6) for a second clutch of the dual clutch to the second
hydraulic output pressure (A_2) by increasing a second hydraulic operating pressure (B_2)
20 acting on the second actuator (6) and

subjecting the first shut-off valve (28) to the second hydraulic output pressure (A_2) via
a first control line (100) by opening the first shut-off valve (28).

10. The method as claimed in claim 9 with the further steps of the method
25 of

preventing a return flow of the hydraulic medium from the second actuator (6) to the
second proportional valve (38) by means of a second shut-off valve (30) while
maintaining the achieved second hydraulic operating pressure (B_2) and

30 reducing the second hydraulic output pressure (A_2), preferably by ending the operation
of the second proportional valve (38), wherein prevention of the return flow of the
hydraulic medium from the second actuator (6) to the second proportional valve (38) is
particularly preferably caused by the reduction of the second hydraulic output pressure
(A_2).

35 11. The method as claimed in claim 10, **wherein** when subjecting the first
actuator (4) to the first hydraulic output pressure (A_1) by increasing the first hydraulic
operating pressure (B_1), the second shut-off valve (30) is also subjected to the first

hydraulic output pressure (A_1) via a second control line (106) by opening the second shut-off valve (30).

12. The method as claimed in claim 9 with the further step of the method of
5 measuring the first hydraulic operating pressure (B_1) and increasing the first hydraulic output pressure (A_1) by operation of the first proportional valve (36) following the reduction of the first hydraulic output pressure (A_1) if the first hydraulic operating pressure (B_1) falls below a predetermined first operating pressure limit value (bg_1).

10 13. The method as claimed in claim 10 with the further step of the method of
measuring the second hydraulic operating pressure (B_2) and increasing the second hydraulic output pressure (A_2) by operating the second proportional valve (38) following the reduction of the second hydraulic output pressure (A_2) if the second
15 hydraulic operating pressure (B_2) falls below a predetermined second operating pressure limit value (bg_2).

14. The method as claimed in any one of claims 9 to 13 with the further steps of the method
20 recording a pressure difference between the first hydraulic operating pressure (B_1) the second hydraulic operating pressure (B_2) and measuring only the higher hydraulic operating pressure of the first and second hydraulic operating pressures (B_1 , B_2), wherein the measurement both of the first and also of the second hydraulic operating pressure (B_1 , B_2) is preferably carried out by means of a common pressure sensor
25 (114).

Fig. 1

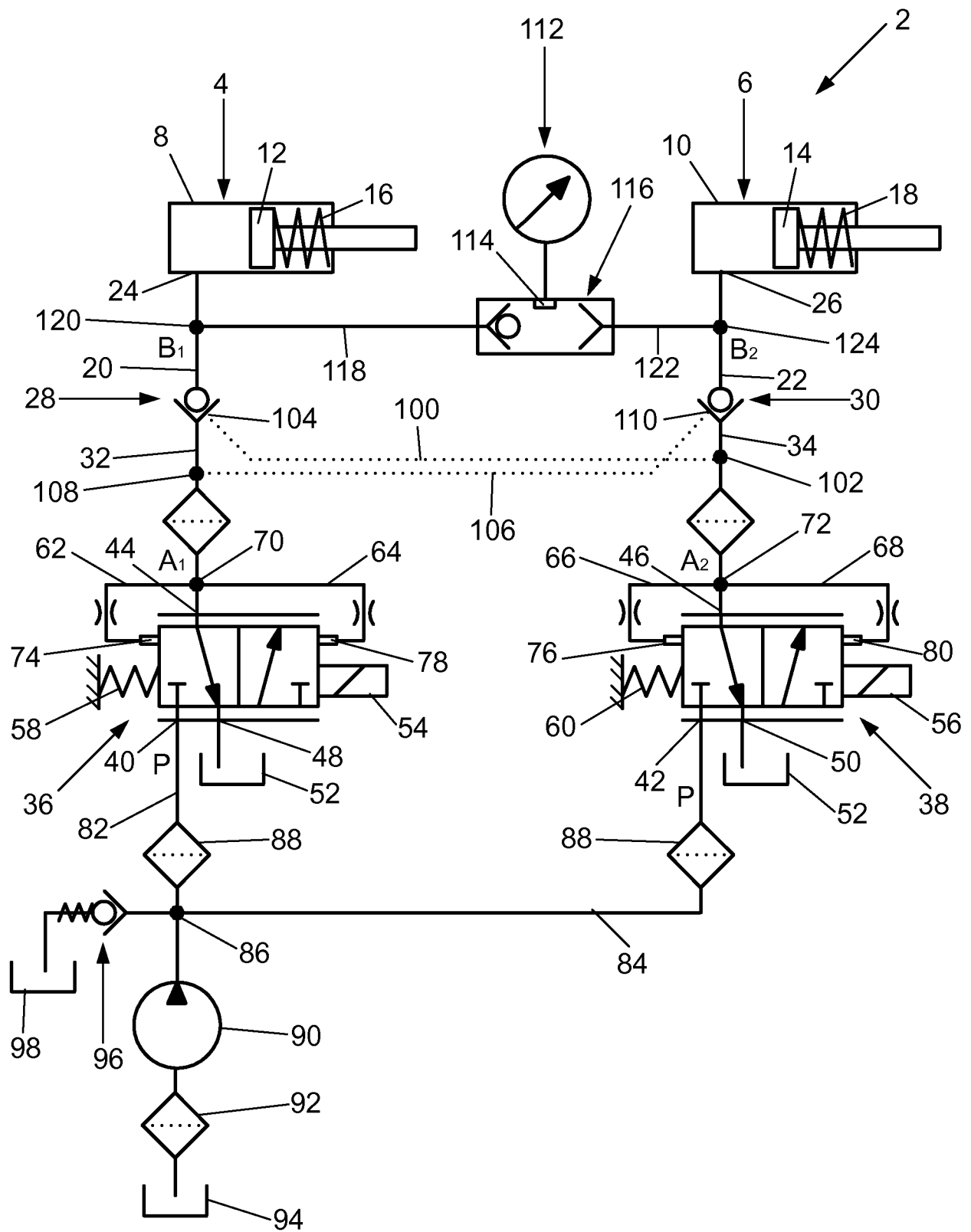
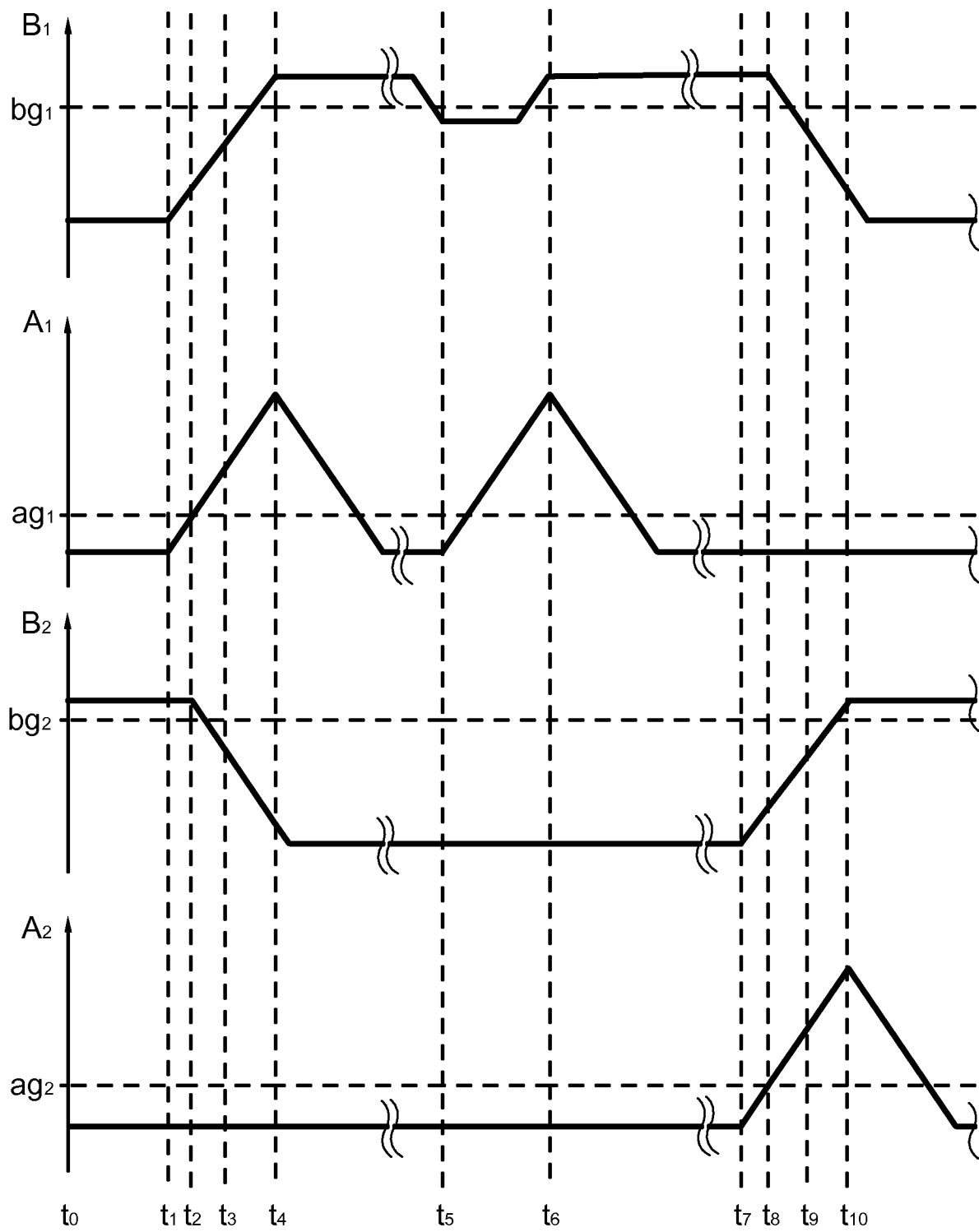


Fig. 2

INTERNATIONAL SEARCH REPORT

International application No.
PCT/US2014/047573**A. CLASSIFICATION OF SUBJECT MATTER****F16H 61/688(2006.01)i**

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

F16H 61/688; B62D 5/08; F16H 37/00; B62D 5/09; F16H 61/28; B60T 13/66; F16H 3/00; F16H 61/02

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

Korean utility models and applications for utility models

Japanese utility models and applications for utility models

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

eKOMPASS(KIPO internal) & keywords: dual clutch, proportional valve, shut-off valve, actuator, spring and electromagnet

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	EP 2410212 A2 (HOFER MECHATRONIK GMBH.) 25 January 2012 See abstract, paragraphs [0071]-[0082], claims 1-3 and figures 1-4.	1-3, 9-14
A	US 7517026 B1 (HERBST et al.) 14 April 2009 See abstract, claims 1-10 and figures 1-3.	1-3, 9-14
A	EP 1400733 A2 (VOLKSWAGEN AKTIENGESELLSCHAFT) 24 March 2004 See abstract, paragraphs [0017]-[0034] and figures 1-3.	1-3, 9-14
A	US 2011-0284311 A1 (CADMAN et al.) 24 November 2011 See abstract, paragraphs [0019]-[0035] and figures 1-5.	1-3, 9-14
A	US 2007-0151376 A1 (MATSUFUJI, MIZUYA) 05 July 2007 See abstract, paragraphs [0090]-[0093] and figures 11, 12.	1-3, 9-14



Further documents are listed in the continuation of Box C.



See patent family annex.

* Special categories of cited documents:

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"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

"&" document member of the same patent family

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INTERNATIONAL SEARCH REPORT

Information on patent family members

International application No.

PCT/US2014/047573

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
EP 2410212 A2	25/01/2012	CN 102345733 A	08/02/2012
		CN 102345734 A	08/02/2012
		EP 2410212 A3	09/05/2012
		EP 2410212 B1	01/01/2014
		EP 2410213 A2	25/01/2012
		EP 2410213 A3	16/05/2012
		EP 2557336 A2	13/02/2013
		EP 2557336 A3	17/07/2013
		EP 2685136 A2	15/01/2014
		EP 2685136 A3	02/04/2014
US 7517026 B1	14/04/2009	EP 2067675 A2	10/06/2009
		EP 2067675 A3	01/12/2010
		EP 2067675 B1	11/12/2013
		EP 2380792 A1	26/10/2011
		EP 2380792 B1	11/12/2013
EP 1400733 A2	24/03/2004	AT 404809 T	15/08/2008
		DE 10243282 A1	01/04/2004
		DE 50310309 D1	25/09/2008
		EP 1400733 A3	23/11/2005
		EP 1400733 B1	13/08/2008
US 2011-0284311 A1	24/11/2011	US 2009-0218161 A1	03/09/2009
		US 7984785 B2	26/07/2011
		US 8651225 B2	18/02/2014
		WO 2010-076614 A1	08/07/2010
US 2007-0151376 A1	05/07/2007	EP 1757834 A1	28/02/2007
		EP 1757834 A4	16/07/2008
		EP 2273146 A2	12/01/2011
		EP 2273146 A3	02/02/2011
		JP 04643160 B2	02/03/2011
		JP 2005-263117 A	29/09/2005
		JP 2005-264981 A	29/09/2005
		US 2008-0035102 A1	14/02/2008
		US 8033298 B2	11/10/2011
		WO 2005-088150 A1	22/09/2005