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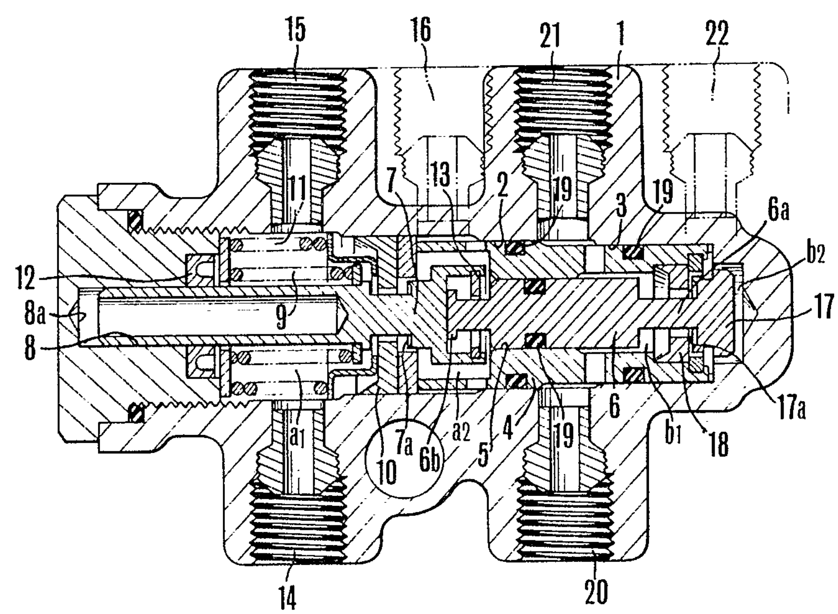
(54) **Dual-circuit Hydraulic Control Devices**

(57) In a dual-circuit hydraulic pressure control device adapted to perform proportioning control in two hydraulic channels, a control piston 7 defines a first proportioning valve in conjunction with a valve seat 10 and a spring 9, such that output pressure in chamber *a*2 depends upon the input pressure in chamber *a*1. The control piston 7 is connected to a balance piston 6 slidably mounted in a cylindrical member 4 which member 4 is also slidable within housing 1. The balance piston 6 defines in conjunction with a valve seat 18 a proportioning valve for the second channel, and the balance piston 6 is moved axially by pressure differences in the output chambers *a*2 and *b*2 of the first and second channels respectively, irrespective of the pressure in the input chamber *b*1 of the second channel. Should however one or the other channel fail, the proportioning valve of the still-operating channel is inhibited from operating.

FIG.1



The drawings originally filed were informal and the print here reproduced is taken from a later filed formal copy.

GB 2 056 003 A

FIG.1

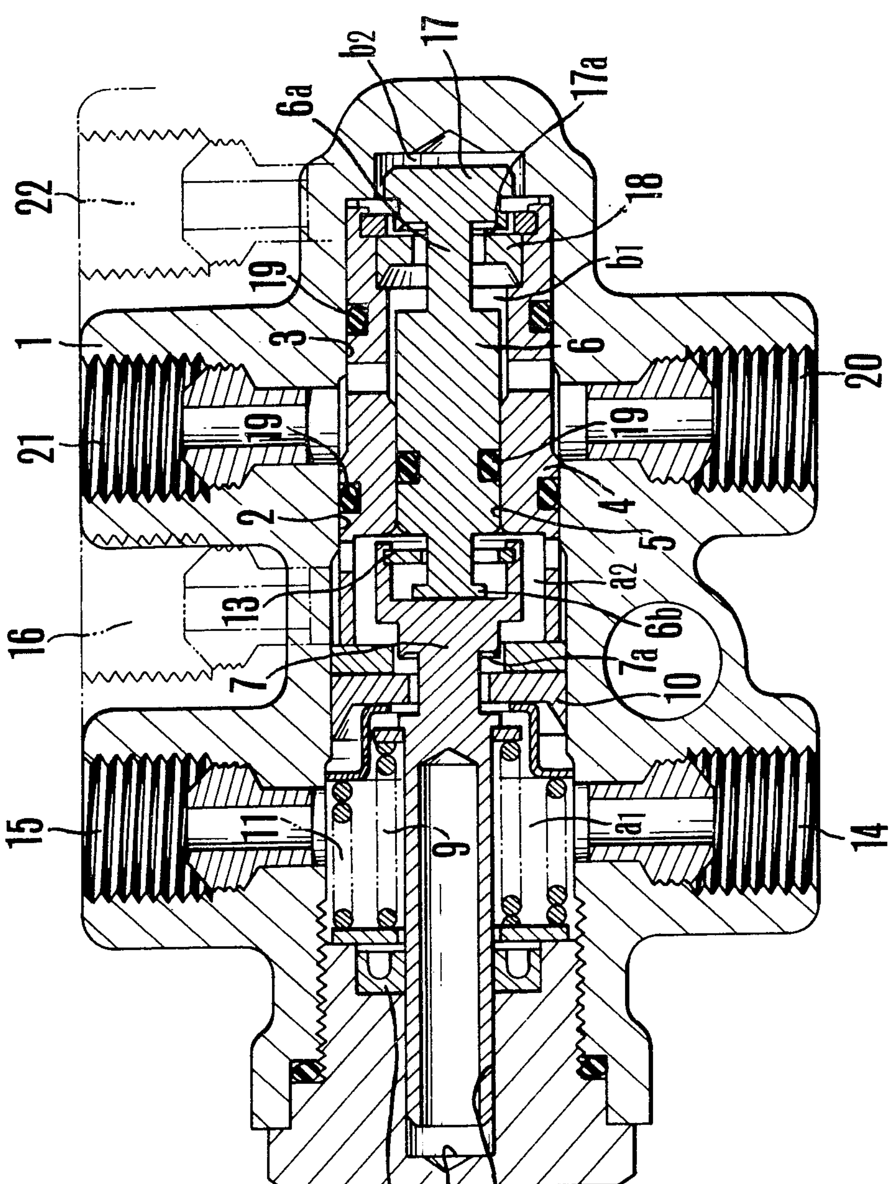
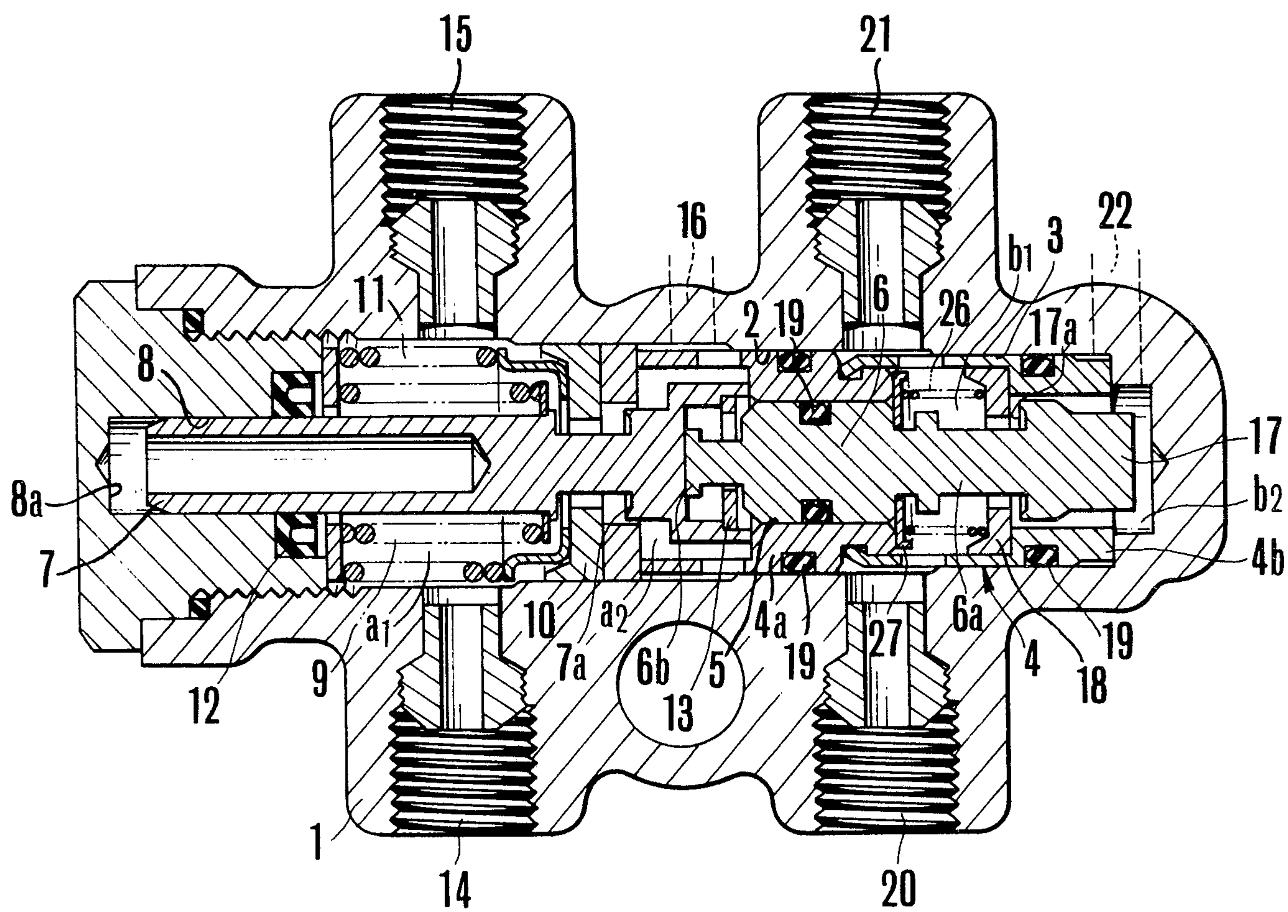

22
6a
b2
17
17a
18
b1
19
1
3
6
20
21
19
19
2
4
5
a2
13
6b
16
7a
10
7
15
11
9
a1
14
12
8a
8

FIG. 2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21
16
22
11
8
2
19
6
26
b1
3
17a
17
8a
b2
7
4b
12
9
a1
10
7a
a2
6b
13
5
4a
19
27
6a
4
18
19
1
14
20

FIG. 2A

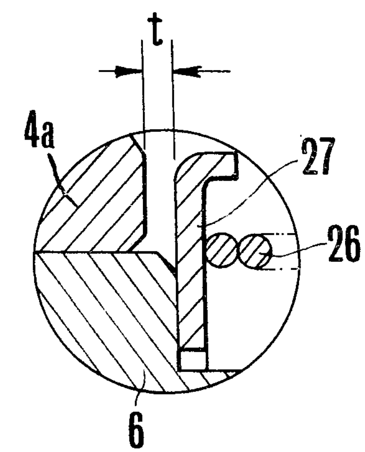

t
4a
27
26
6

FIG.3

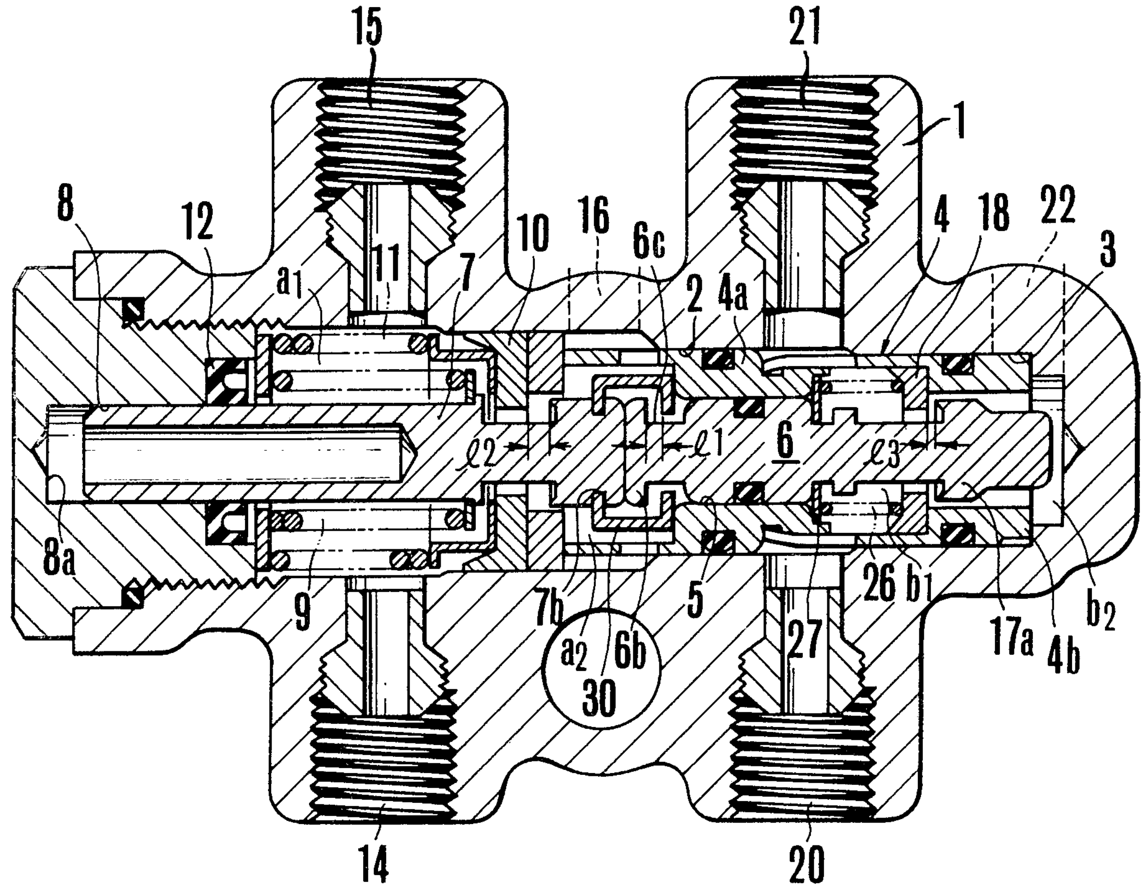


FIG.4

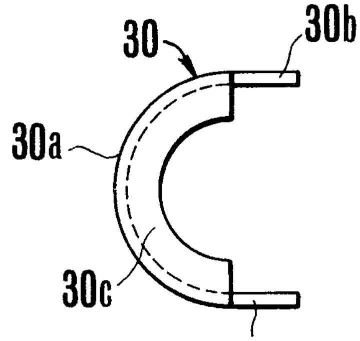


FIG.5

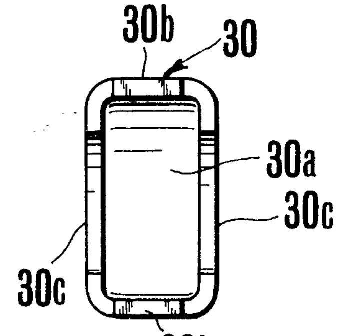
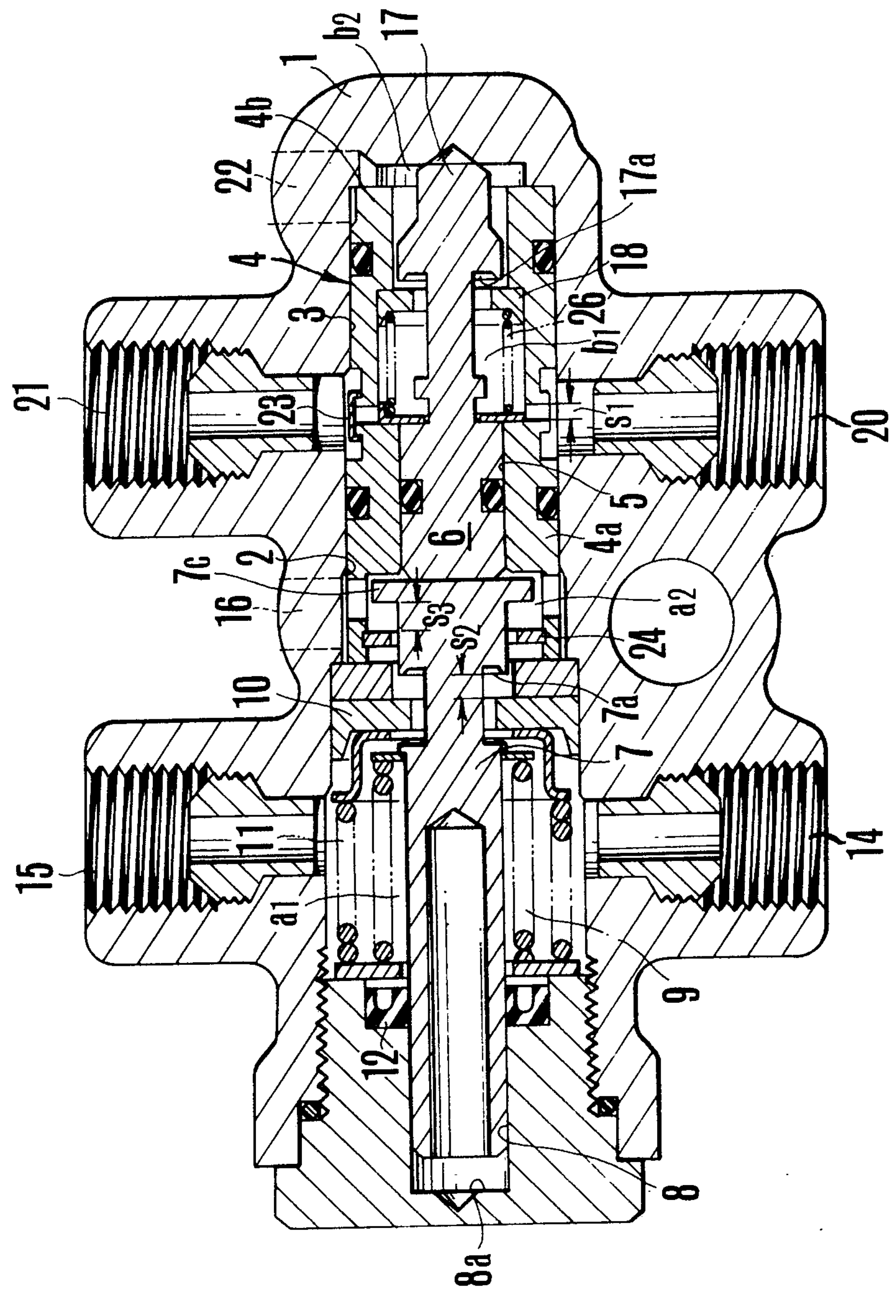


FIG.6


1
b2
17
4b
22
17a
4
18
3
26
b1
21
23
s1
20
5
4a
6
7c
2
16
s3
s2
a2
24
10
7a
7
15
11
a1
14
9
12
8
8a

SPECIFICATION

Dual Circuit Hydraulic Control Devices

This invention relates to hydraulic pressure control devices of the type intended for use with dual circuit hydraulic systems such as are found for instance on advanced motor vehicles.

Heretofore, there have been provided various kinds of dual circuit hydraulic pressure control devices for the hydraulic brake system of a vehicle when that system is divided into two channels arranged in an X-shape configuration relative to the front and rear wheels of the vehicle. The device itself functions to limit the pressure applied to the rear wheel brakes, so that the braking forces at the front and rear wheels are made suitable for the friction between the road surface and these wheels—that is, smaller braking forces are exerted at the rear wheels than at the front wheels.

A known control device of this type has a proportioning valve mechanism adapted proportionally to reduce the rate of increase of hydraulic pressure applied to one rear wheel brake with respect to the front wheel brake of the same hydraulic channel, and that proportioning is transmitted to the other channel through a balance piston which is arranged to effect hydraulic pressure control over the other channel performed in association with the hydraulic pressure control performed in the one channel. This invention aims at improving such a known device to obtain a better state of balance between the two channels. In the known device, imbalance can occur because proportioning in the second channel does not occur until there is a differential pressure between the proportioned output hydraulic pressure of the first channel (corresponding to the pressure applied to the rear wheel brake of that channel) and the front and rear wheel output pressures of the second channel, so that the differential pressure can cause the balance piston to move axially and thus to close, for example, a ball valve employed as a proportioning valve mechanism for the second channel. After that, the pressure applied to the rear wheel brake of the second channel is proportioned with respect to the pressure applied to the front wheel brake of the second channel in correspondence with a rise in the rear wheel output pressure of the first channel. The ball valve, which opens or closes a passage allowing the input and output chambers of second channel to communicate with each other, is arranged to receive, at a part which corresponds to the sectional area of the communication passage, a net force that corresponds to the difference between the input hydraulic pressure and the proportioned output hydraulic pressure of the second channel. This force is proportional to an increase in the differential pressure between the input hydraulic pressure and the output hydraulic pressure. Therefore, this arrangement tends to cause imbalance between the output hydraulic pressures of the two channels.

According to this invention, there is provided a dual circuit hydraulic control device for first and second hydraulic pressure channels and having input and output chambers for the two channels, which valve comprises: a control piston arranged to receive hydraulic pressure of the first channel; a first valve mechanism which performs pressure reducing control over the output hydraulic pressure of the first channel by allowing the control piston to move to abut upon a first valve seat against the force of a control spring; a balance piston arranged to receive the output pressures of the first and second channels respectively at the two ends thereof; and a second valve mechanism which, in association with the movement of the balance piston under excess output pressure in the second channel with respect to the output pressure in the first channel, closes communication between the input and output chambers of the second channel, the second valve mechanism having a second valve seat arranged between the input and output chambers of the second channel and a valve body part unified with the balance piston and disposed within the output chamber of the second channel, which valve body part is arranged to block communication between the input and output chambers of the second channel when engaged with the second valve seat.

It will be appreciated that in the device of this invention, the proportioning control effected in the first channel is transmitted to the second channel in such a way that the hydraulic pressures exerted on the balance piston for proportioning in the second channel are irrespective of the input hydraulic pressures to the two channels. However, the device may be configured so that should either channel fail, the device will no longer effect proportioning control in the still-operating channel.

Preferably, the balance and control piston are arranged co-axially for sliding movement within a bore in a common housing, to allow a compact device to be constructed which is easy to install in a vehicle dual circuit braking system.

By way of example only, four specific embodiments of this invention will now be described in detail, reference being made to the accompanying drawings, in which:—

Figure 1 is a longitudinal sectional view showing a first embodiment of hydraulic pressure control device of the invention;

Figure 2 is a longitudinal sectional view of a second embodiment of the invention;

Figure 2A is an enlarged view of an essential part of a modification of the second embodiment;

Figure 3 is a longitudinal sectional view of a third embodiment of the invention;

Figures 4 and 5 are enlarged views showing a clip used in the third embodiment; and

Figure 6 is a longitudinal sectional view showing a fourth embodiment of the invention.

The first embodiment of the invention shown in Figure 1 comprises a valve body 1 defining a stepped cylinder 2 and 3 in which is slidably mounted a tubular cylindrical member 4. A balance piston 6 is slidably received within a cylinder 5 formed in an inner cylindrical part of the cylindrical member 4, which balance piston 6 is arranged to divide the interior of the valve body 1 into two hydraulic pressure channels A and B, as well as to transfer the proportioning control performed in channel A (on the left hand side as viewed in Figure 1) to channel B (on the right hand side as viewed in Figure 1), as will be described below.

A proportioning valve mechanism disposed in channel A comprises a control piston 7 which has a smaller diameter end slidably received in a blind bore 8 so that the larger diameter end of the piston 7 confronts the balance piston 6, a control spring 9 pushing the control piston 7 towards the balance piston 6. A valve seat 10 is loosely carried on a shaft-part of the control piston 7 to divide the interior of a chamber of the valve mechanism into an input hydraulic liquid chamber a1 and an output hydraulic liquid chamber a2 and which seat 10 is arranged to be engageable with a valve body part 7a of the control piston 7 in such a manner as to permit or block communication between the input and output liquid chambers a1 and a2. The embodiment also includes a spring 11 which is arranged to maintain the valve seat 10 in position; a lip seal 12; and a stop member 13 which prevents the control piston 7 and the balance piston 6 from separating more than a predetermined extent, so as to inhibit the hydraulic pressure proportioning control in channel A when channel B becomes defective, with no hydraulic pressure.

Also shown in Figure 1 are an input port 14 for connecting the input liquid chamber a1 of channel A to a master cylinder (not shown); an output port 15 for connecting the input liquid chamber a1 to a front wheel brake device (not shown); and an output port 16 for connecting the output liquid chamber a2 to a rear wheel brake (not shown).

A valve mechanism is provided in the channel B and is arranged as follows. A large diameter head part 17 is formed at one end of a small diameter part 6a of the balance piston 6, so as to be disposed within an output liquid chamber b2. The head part 17 is provided with a valve body part 17a which has the same sealing sectional area as the sectional area of the inner cylinder 5 of the cylinder member 4. A valve seat 18 is secured to the cylinder member 4 and has the smaller diameter part 6a of the balance piston 6 loosely passing therethrough, to divide the inside of a chamber formed there into an input hydraulic liquid chamber b1 and an output hydraulic liquid chamber b2. The valve seat 18 is thus arranged to allow or block communication between the input and output liquid chambers b1 and b2 by engaging the above stated valve body part 17a.

In channel B, there are also provided a seal member 19; an input port 20 for connecting the input liquid chamber b1 to channel B of a master cylinder; an output port 21 for connecting the input liquid chamber b1 to a front wheel brake device; and an output port 22 for connecting the output liquid chamber b2 to a rear wheel brake device.

The device which has been described in the foregoing operates as follows. When a brake hydraulic pressure is transmitted from the master cylinder equally to the input liquid chambers a1 and b1 of the two channels A and B, initially the pressure in the output liquid chambers a2 and b2 rises uniformly with the input pressure until the point at which proportioning is to take place. Initially, the hydraulic pressure acting on the control piston 7, which is arranged to be balanced by the spring force of the control spring 9, causes the control piston 7 to move from the illustrated condition to the left against the force of the control spring 9 until the valve body part 7a comes to engage the valve seat 10. Accordingly, the input and output hydraulic pressure values Pa1 and Pa2 of the input and output liquid chambers a1 and a2 increase in unison up to that point but, after that, the output hydraulic pressure value Pa2 increases more slowly than the input hydraulic pressure value Pa1, at a rate of:

$$\tan\Theta=(A1-A2)/A1\ (>1)$$

which is determined by the ratio of the sealing sectional area A1 of the valve body part 7a to the sectional area A2 of the blind bore 8 (A2>A1).

The hydraulic pressure control occurring in channel A is transmitted to the channel B by means of the balance piston 6. Initially in channel B, the hydraulic pressure exerted on the balance piston 6 to the left (as viewed in Figure 1) is governed by the input hydraulic pressure Pb1 in the input liquid chamber b1, because in this initial stage the input and output liquid chambers b1 and b2 of channel B are in communication with each other. Moreover, the input hydraulic pressure Pb1 is equal to the input hydraulic pressure Pa1 of channel A, and the hydraulic pressure on the balance piston 6 exerted from channel A in the rightward direction as viewed on the drawing is governed by the output hydraulic pressure Pa2. Therefore, up to the point at which proportioning control takes place in channel A to give a lower rate of increase in the hydraulic pressure Pa2 as compared with increase in pressure Pa1, the hydraulic forces acting in the balance piston 6 are equal. As soon as part 7a closes the communication between chambers a1 and a2, the resultant imbalance of pressures on the piston 6 causes the piston 6 to move to the left, as viewed in the drawing, until the valve body part 17a comes into engagement with the valve seat 18. Then, the hydraulic pressure is exerted on the balance piston in channel B becomes Pb2=Pa2, because the sectional area A3 of the cylinder 5 is the same as the sealing sectional area of the valve body part 17a. Following this, the balance piston 6 opens and closes the communication between the chambers b1 and b2 so as to balance the output hydraulic pressure Pa2 of channel A with that of Pb2 of channel B, serving to keep these output hydraulic pressure values equal. In any event, since the hydraulic pressures Pa2 and Pb2 are arranged to act in opposition in the same pressure receiving areas A3 of the balance piston 6, the input hydraulic pressure Pb1 does not participate controlling the communication between the input and output liquid chambers b1 and b2 of channel B. Therefore, the output hydraulic pressure Pb2 of channel B is stably kept equal to the output hydraulic pressure Pa2 of channel A, irrespective of the differential pressure value between the input hydraulic pressure Pb1 and the output hydraulic pressure Pb2.

Should the hydraulic system fail on channel A, the hydraulic pressure in channel B causes the control piston 7, the balance piston 6 and the cylindrical member 4 to move together to the left, as viewed in Figure 1, until the control piston 7 abuts the end 8a of the blind bore 8. At this end position, there is produced a clearance between the valve body part 17a and the valve seat 18, so that the output hydraulic pressure Pb2 cannot be lower than the input pressure Pb1. Conversely, should the hydraulic system of channel B fail, the balance piston 6 is moved to the right as soon as there is a small hydraulic pressure in channel A. Though the control piston 7 tries to move against the force of the spring 9 on a pressure increase in channel A, no movement is possible because the sectional area A3 of the cylinder 5 is larger than the sectional area A2 of the bore 8 and the stop member 13 secured to the piston 7 engages a head part 6b of the balance piston 6, restraining the piston 7 against movement. In this way, the output pressure Pa2 will be the same as the input pressure Pa1.

The described embodiment of dual circuit hydraulic pressure control device of the present invention has the great advantage that the hydraulic pressure break point control can be stably performed over the interlocked channel (or the channel B) by a relatively simple improvement on the structural arrangement of the conventional device.

Figure 2 shows a second embodiment of the invention, wherein parts similar to those of the first embodiment are given the same reference characters, and will not be described in detail again. In the second embodiment, the cylindrical member 4, which is slidably received in the stepped cylinders 2 and 3, is somewhat differently arranged and comprises a first part 4a having an inner cylinder 5 formed therein and a second part 4b connected to one end of the first part 4a. It is a feature of this embodiment that, in channel B of the hydraulic pressure control device, there is provided a movement restricting mechanism which comprises a spring 26 and a spring seat 27 which resist the movement of the balance piston 6 towards the right under normal operating conditions.

Strictly speaking, for many tandem master cylinders the hydraulic pressure inputs Pa1 and Pb1 to the two channels A and B are not equal to each other. During brake application (when the hydraulic pressure increases), one hydraulic pressure input is higher than the other input. Conversely, during brake release (when the hydraulic pressure decreases), the other hydraulic pressure input becomes higher than the one input, and this difference in pressure is greater during brake release. In this particular embodiment, the one hydraulic pressure input coming from the master cylinder is connected to the channel B while the other is connected to the channel A. Then, the above-stated pressure difference presents the problem that, during brake release, the hydraulic pressure acting on the balance piston 6 urging the piston to the right (as viewed in Figure 2) is greater than that urging the piston to the left.

This problem means that, during the next brake application, the leftward movement of the control piston 7 takes place together with the balance piston 6 because of engagement of the stop member 13 with head 6b, and the sliding movement of the control piston 7 is thus affected by the sliding resistance of the balance piston. Further, while the hydraulic proportioning control in channel B is arranged to be effected only when the balance piston 6 is moved to the left (as viewed in the drawing) to a predetermined extent when the valve body part 7a of the control piston 7 comes to abut the valve seat 10, the free movement of the balance piston 6 is restricted and the hydraulic pressure output Pb2 of channel B might consequently be increased to an unnecessarily great extent because, in this state, communication between the output liquid chamber a2 and the input liquid chamber a1 of channel A is already blocked.

In this embodiment, the abovestated problem is solved by resisting the rightward movement of the balance piston 6 (as viewed in the drawing) by means of the spring 26 and the spring seat 27, thus preventing the balance piston 6 from unnecessarily moving toward channel B during brake release. With the provision of this movement resisting mechanism, the extent of movement of the balance piston 6 under normal conditions is stabilised and set to be from the illustrated stationary state to a state wherein the valve body part 20a is in contact with the valve seat 21, so that the adverse effect of the sliding resistance and so on can be eliminated. Furthermore, when channel B fails, the large hydraulic force in channel A causes the balance piston 6 to compress the spring 26 and thus to restrict the movement of the control piston to the left, as viewed in the drawing.

Also, as shown in Figure 2A, it is also possible to provide a clearance t between the first part 4a of the cylinder member 4 and the spring seat 27 in such a way as to have the balance piston 6 moved to the left by the spring 26, in response to initial movement of the control piston 7, until the spring seat 27 comes into contact with the first part 4a. This arrangement allows a large clearance between the valve body part 17a and the valve seat 18 in the initial stage so that, should the master cylinder brake pedal be pushed down suddenly to cause a large hydraulic liquid flow, the large flow can readily be passed from the input liquid chamber b1 to the output liquid chamber b2.

A third embodiment of the invention is shown in Figures 3 to 5, and again the parts the same as those of the preceding embodiments are given the same reference characters and will not be described in detail once more. It is a feature of the third embodiment that the control piston 7 and the balance piston 6, which respectively form the proportioning valve mechanisms of channels A and B, are connected to each other by means of a fail-safe clip 30 which is fitted in a circumferential groove 7b in the control piston 7. By this arrangement, these pistons 7 and 6 are restrained from separating by more than a predetermined extent $l1$ and the rightward movement of the control piston 7 is limited by the fail-safe clip 30 abutting the cylindrical member 4. Referring to Figures 4 and 5, the fail-safe clip 30 comprises a semi-cylindrical arcuate part 30a; arm parts 30b which extend from both ends of the arcuate part 30a in the tangential directions; and semi-annular flange parts 30c. One of the flange parts 30c is fitted into the circumferential groove 7b formed in the right-hand end neck part of the control piston 7, whereas the other flange part 30c lies loosely in a circumferential groove 6c formed in the balance piston 6. The arm parts 30b are bent round the respective grooves in such a way as to prevent the clip 30 from coming out of position radially.

The dual circuit hydraulic pressure control device arranged as described above normally, when in the static condition as represented by the drawing, has the fail-safe clip 30 fixed to the control piston 7 abutting the cylindrical member 4 by virtue of the spring force of the control spring 9, and the balance piston 6 is biassed rightward by the control piston 7 so that there is communication between the input and output liquid chambers a1 and a2 and between the input and output liquid chambers b1 and b2 respectively of channels A and B.

When hydraulic pressure is applied in both channels, the central piston 7 moves against the force of the control spring 9 to begin the hydraulic pressure proportioning control of channel A, and then similar control of the other channel B also takes place in association therewith. In this instance, the small spring force applied by spring 11 allows the cylinder member 4 to remain in the stationary state. Furthermore, assuming that the extent to which the control piston of the valve mechanism in channel A moves until it abuts the valve seat 10 is $l2$ and the extent to which the balance piston 6 moves until it abuts the valve seat 18 is $l3$, it is necessary to arrange $l1 < (l2 - l3)$, $l1$ being the relative movement of the pistons 6 and 7 permitted by the fail-safe clip 30.

Should channel B fail, hydraulic pressure from channel A acts to move the balance piston 6 to the right as viewed in the drawing, and this restricts the movement of the control piston 7 leftward towards the associated left valve seat 10, through the fail-safe clip 30. Therefore, the hydraulic proportioning control performed by the valve mechanism of channel A in inhibited and the input and output hydraulic liquid chambers a1 and a2 are continuously in communication.

Should channel A fail, the balance piston 6 and the cylinder member 4 move leftward as viewed in the drawing, maintaining the illustrated relation to the control piston 7, until the control piston comes to a stop by abutting on end 8a of the blind bore 8. Accordingly, the valve body part 17a in the valve mechanism of channel B is not allowed to abut the valve seat 18 and the input and output hydraulic liquid chambers b1 and b2 are continuously in communication.

Figure 6 shows a fourth embodiment of the invention and similar parts to those of the preceding embodiments are given like reference characters; these parts will not be described again. In the fourth embodiment, the cylindrical member 4 comprises a first part 4a and a second part 4b which are restrained from separating axially by means of a clip 23 which is arranged around the outer circumferences of the adjacent ends of the first and second parts 4a and 4b. These first and second parts 4a and 4b are urged apart by means of the spring 26 (mentioned in the second and third preceding embodiments) to the extent permitted by the clip 23, the spring providing a relatively small spring force. In this arrangement, there is a maximum clearance of S1 between the first and second parts 4a and 4b of the cylindrical member 4.

The operation of this fourth embodiment under normal conditions is the same as the operation of the preceding embodiment and, therefore, will not be described here.

Should channel A fail, hydraulic pressure causes the balance piston 6 to move leftward, as viewed in the drawing, compressing the control spring 9 through the control piston 7 until the rear end of the control piston 7 contacts the end 8a of the blind bore 8. In this instance, the first part 4a of the cylindrical member 4 is also moved to the left as viewed in the drawing until that part contacts the piston 7, at the same time compressing the holding spring 11 through the valve seat 10. The second part 4b to follow the movement of the first part 4a, by virtue of the clip 23. Also, the balance piston 6 moves to the left until it abuts the piston 7. Therefore, the valve body part 17a of the balance piston 6 is unable to abut the valve seat 18, so that hydraulic pressure proportioning control in channel B is inhibited.

In the case of a failure in channel B, the valve mechanism in the channel A tries to operate in the same manner as under a normal condition. However, the first part 4a of the cylinder member 4 is caused by the hydraulic pressure in channel A to move to the right, as viewed in the drawing, against the force of the spring 26 until that part abuts the second part 4b. Assuming that the extent to which the first part 4a moves is S1, that is, the amount of clearance normally existing between the first and second parts 4a and 4b, the gap between the valve seat 10 and the valve body part 7a is S2, and the gap between a stop ring 24 and a flange 7c provided on the piston 7 is S3, these clearances should be arranged in the relation of S3—S1<S2. Provided this is so, the movement of the control piston 7 is restricted by the stop ring 24 attached to the first part 4a and the flange 7c of the piston 7, so that the valve seat part 7a cannot abut the valve seat 10 and, eventually, the hydraulic proportioning control of the valve mechanism of channel A is inhibited, so that the output hydraulic pressure Pa2 rises in unison with the input hydraulic pressure Pa1, because of the constant communication between the two chambers a1 and a2.

Claims

1. A dual circuit hydraulic control device for first and second hydraulic pressure channels and having input and output chambers for the two channels, which valve comprises: a control piston arranged to receive hydraulic pressure of the first channel; a first valve mechanism which performs pressure reducing control over the output hydraulic pressure of the first channel by allowing the control piston to move to abut upon a first valve seat against the force of a control spring; a balance piston arranged to receive the output pressures of the first and second channels respectively at the two ends thereof; and a second valve mechanism which, in association with the movement of the balance piston under excess output pressure in the second channel with respect to the output pressure in the first channel, closes communication between the input and output chambers of the second channel, the second valve mechanism having a second valve seat arranged between the input and output chambers of the second channel and a valve body part unified with the balance piston and disposed within the output chamber of the second channel, which valve body part is arranged to block communication between the input and output chambers of the second channel when engaged with the second valve seat.

2. A dual circuit hydraulic control device according to claim 1, wherein the balance piston has a pressure receiving area arranged to receive the pressure within the output chamber of the first channel equal to another pressure receiving area thereof which is arranged to receive the pressure of the output chamber of the second channel when the second valve seat and the valve body part are engaged.

3. A dual circuit hydraulic control device as claimed in claim 1 or claim 2, wherein the control piston and the balance piston are disposed co-axially within a bore formed in a common housing therefor.

4. A dual circuit hydraulic control device according to claim 3, wherein there is provided stop means for restricting the maximum axial separation of the balance piston and the control piston.

5. A dual circuit hydraulic control device according to claim 3 or claim 4, wherein the balance piston is arranged to slide within a cylindrical member biased to an end position within the bore of the housing.

6. A dual circuit hydraulic control device according to claim 5, wherein a spring is arranged to urge the balance piston in a direction opposing the movement of the balance piston caused by excess pressure in the output chamber of the first channel.

7. A dual circuit hydraulic control device according to claim 6, wherein the spring is arranged to urge the balance piston through a spring seat which spring seat also may engage the cylindrical member, there being provided a clearance between the cylindrical member and the spring seat in the normal rest state of the device.

8. A dual circuit hydraulic control device according to any of claims 3 to 7, wherein there is a first connection member secured to the control piston and loosely connected to the balance piston in the region of the adjacent ends of the control and balance pistons.

9. A dual circuit hydraulic control device according to claim 8, wherein the first connection member abuts the cylindrical member in the normal rest state of the device, to determine the position of the control piston by means of a control spring which is arranged to urge the control piston in a direction opposed to the direction of movement of the control piston caused by excess hydraulic pressure in the output chamber of the first channel.

10. A dual circuit hydraulic pressure control device according to claim 5 or any claim dependent thereon, wherein the cylindrical member comprises first and second axial parts, a spring being arranged to urge the two parts apart and a second connection member coupling the two parts together to restrict the maximum separation thereof.

11. A dual circuit hydraulic control device according to claim 10, wherein the balance piston is arranged to slide within the first part of the cylindrical member, and the extent to which the control piston and the first part may move away from each other is limited to a predetermined value.

12. A dual circuit hydraulic control device substantially as hereinbefore described, with reference to and as illustrated in Figure 1, or in Figures 2 and 2A, or in Figures 3 to 5, or in Figure 6 of the accompanying drawings.

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