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(54) SERVO-ASSISTED HYDRAULIC MASTER CYLINDER

(71) We, AUTOMOTIVE PRODUCTS LTD, a British Company of Tachbrook Road, Leamington Spa, Warwickshire, CV31 3ER do hereby declare the invention for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:-

The invention relates to servo-assisted hydraulic master cylinders, particularly, but not exclusively, for vehicle braking systems.

The invention provides a servo-assisted hydraulic master cylinder comprising a variable ratio master cylinder of the kind in which a stepped piston is slidable in a stepped bore to provide a large effective piston area when output pressure is below a pre-determined magnitude and a small effective piston area when output pressures is above the pre-determined magnitude, and in which the stepped piston and the stepped bore define a control chamber in which pressure is progressively varied by a metering valve when output pressure increases from the pre-determined magnitude to a higher magnitude to provide a ratio of input effort to output pressure which is intermediate that which pertains when output pressure is below the pre-determined magnitude and that which pertains when output pressure is above the higher magnitude and a direct-acting servo of the kind having an output member connected to the master cylinder, an operator controlled servo valve for controlling a pressure difference across a movable wall and reaction means which connects the movable wall to the output member and operates when the load transmitted by the output member exceeds a pre-determined magnitude to provide a reaction force on the servo valve which is proportional to an increment in the load transmitted by the output member above the pre-determined magnitude and wherein the load in the output member required to produce the pre-determined output pressure at which the metering valve becomes operative substantially coincides with the pre-determined magnitude of load at which the reaction means becomes operative.

Other features of the invention will now be described with reference to the accompanying drawings, of which:–

Figure 1 is a cross-section of a direct-acting vacuum servo unit which forms one sub-assembly of one example of a servo-assisted master cylinder according to the invention; and

Figure 2 is a cross-section of a hydraulic master cylinder which is for fitting to the sub-assembly of Figure 1 and which forms another sub-assembly of the same example of a servo-assisted master cylinder according to the invention.

The servo unit 112 shown in Figure 1 is of a kind commonly used in vehicle braking systems. It comprises two cup-shaped metal pressings 113, 114 joined at their respective rims, 115, 116 where they sandwich the outer bead 117 of a diaphragm 118.

The diaphragm 118 is supported by a flange 119 which extends radially outwards as an integral part of a servo valve housing 121. The diaphragm 118 and the diaphragm support flange 119 form a movable wall, a pressure differential across this movable wall being controlled by a servo valve member 122 slidable in an axial stepped bore in the servo valve housing 121 and connected to a push rod 123 for connection to a driver's brake pedal. A chamber 124 within the pressing 113 and on one side of the diaphragm 118 is connected to a source of vacuum, e.g. an engine inlet manifold, through an elbow fitting 125 which includes a non-return valve. Another chamber 126 inside the pressing 114 and on the other side of the diaphragm 118 is connected to an annular chamber 127 encircling the servo valve member 112 by a passage 128 in the servo valve housing 121. The servo valve member 122 co-operates with a rubber valve seat member 129 contained in a tubular extension 138 of the servo valve housing 121 to connect the annular chamber 127 either to vacuum through a passage 131 in the servo valve housing 121 or to atmospheric air through an annular passage 132 between the valve seat member 129 and the push rod 123.

A coil spring 133 biasses the push rod 123 away from the diapragm support flange 119 and another coil spring 134 biasses the valve seat member 129 in the opposite direction. The initial position of the push rod 123 and the servo valve member 122, as shown in Figure 1, is determined by a key 135 fitted into a radial slot in the servo valve housing 121 to project into an annular groove 139 in the servo valve member 122. In this position the annular chamber 127 is connected to the vacuum in chamber 124 through passage 131 and hence the pressures in chambers 124 and 126 are equal. Furthermore, the servo valve member 122 is seated on the valve seat member 129 so that air is prevented from entering through passage 132.

Movement of the push rod 123 in the brake-applying direction towards the pressing 113 initially moves the servo valve member 122 into a lap position where the valve seat member 129 is still abutting the servo valve member 122 but where it also abuts an annular seat 136 on the servo valve housing 121 to close off the annular chamber 127 to passage 131 as well as to annular passage 132. Further movement of the push rod 123 towards the pressing 113 then causes the servo valve member 122 to move away from the valve seat member 129, the valve seat member 129 remaining in abutment with the annular seat 136. Air can now enter chamber 126 through passage 128, annular chamber 127 and annular passage 132, the incoming air being filtered by an annular filter 137 provided between the push rod 123 and the tubular extension 138.

The master cylinder 111 shown in Figure 2 is adapted to be secured by studs 101 on the pressing 113 of the servo unit 112 such that the rear face 102 of the master cylinder body 158, which has two diametrically opposed mounting flange lobes in common with the majority of brake master cylinders, abuts the front face 103 of the pressing 113.

A stepped piston assembly is slidable in a stepped bore of the master cylinder body 158. It comprises an inner piston 141 having a stepped cylindrical surface which is slidable in an annular cup seal 164 and in the stepped axial bore of an outer piston 146.

The stepped piston assembly is adapted to be moved by a plunger 186 which forms the output member of the servo unit 112 and is received in the end of the outer piston 146 when the master cylinder 111 is assembled onto the servo unit 112. Plunger 186 abuts a rubber reaction disc 187, the function of which will be described later.

A primary pressurising chamber 155 in the master cylinder 111 is formed between the inner piston 141 and a secondary piston 156 and an annular control chamber 157 is formed between the inner piston 141 and the master cylinder body 158. A recuperation port 159 connects the control chamber 157 to a supply port 161 for connection to a tank of hydraulic fluid. The control chamber 157 is also connected through to the primary chamber 155 by a port 163, otherwise the control chamber 157 is sealed off from the primary chamber 155 by the annular cup seal 164. This seal 164 is supported by a backing rings 165A and 165B and positioned in the bore of the master cylinder 111 by a circlip 166. An outlet port 167 opening into the primary chamber 155 is for connection to a group of brake actuators on the vehicle.

The inner piston 141 has a blind axial bore which contains a metering valve assembly comprising a plunger 191 slidable in the bore of piston 141, a ball 188 which is normally seated on an annular seat formed by the open end of an axial stepped bore in the plunger 191 and a pin 189 which is slidable in the other, smaller diameter, portion of the bore of the plunger 191 so as to abut the end of the blind bore of the first piston 141. A coil spring 149 urges the plunger 191 away from the blind end of the bore in the outer piston 141 and into abutment with a thrust member 192 which in turn abuts a rubber cup seal 193. Seal 193 is held in abutment with the end face of the thrust member 192 by a cup spreader 194 retained by circlip 195. A backing collar 196 is a sliding fit over the reduced diameter end of the thrust member 192 which abuts cup seal 193, the resilience of seal 193 being such that pressure in the first chamber is transmitted as a force onto the end face of thrust member 192 as if the thrust member 192 were a piston working in the bore of the backing collar 196.

Cup seal 193 acts as a non-return valve to allow communication from the control chamber 157 to the primary chamber 155 through a radial passage 172 in the inner piston 141, axial grooves in the outer periphery of thrust member 192 and grooves in the outer periphery and rear face of collar 196. Another annular cup seal 178 seals between the inner piston 141 and the bore of the master cylinder and abuts the forward face of the outer piston 146. This seal 178 is capable of acting as non-return valve means to allow fluid from the tank to the control chamber 157 but it is supplemented by the ball 188 which is capable of being lifted from its seat on the metering valve plunger 191 against a coil spring 198.

When the push rod 123 is moved in a brake-applying direction towards the master cylinder 111 to allow air into chamber 126 the stepped piston assembly is moved in the same direction so that seal 178 wipes over the recuperation port 159 to prevent communication from the control chamber 157 back to the supply port 161. With a small further movement of push rod 123 port 163 wipes over seal 164 to prevent communication from the primary chamber 155 to the control chamber 157. Thus a brake-applying pressure is built up in the primary chamber 155, fluid also being displaced into primary chamber 155 from the control chamber 157 past the cup seal 193 acting as a non-return valve. Fluid is displaced out of outlet port 167 by a piston area equivalent to that defined by the outer diamter of seal 178.

There is a substantial gap between the rear face of the reaction disc 187 and the forward face of the servo valve member 122 which is reduced, but not eliminated, when the servo valve member 122 is moved into the lap position. By virtue of this gap and the elasticity of the rubber of the reaction disc 187, small servo efforts are transmitted directly from the servo valve housing onto the plunger 186 without the disc 187 distorting enough for a load to be transmitted back onto the servo valve member 122. However, a load on plunger 186 above a pre-determined magnitude causes the disc 187 to distort so that a load is reacted back onto the servo valve member 122 and thus back to the driver. The rubber of disc 187 tends to act as a liquid under these conditions so that an increment in the load transmitted by plunger 186 is accompanied by a proportional increase in the driver effort applied through the servo valve member, the loads being apportioned according to the ratios of the areas of the flat faces which are in contact with the disc 187. This type of reaction mechanism is known, and can be replaced by equivalent reaction means, for example that described in our British Patent Specification No. 1, 363, 243. However, the present invention relates to the matching of the servo unit 112 and the master cylinder 111 so that the load on plunger 186 needed to react a load onto the servo valve member 122 substantially coincides with the load needed to produce the pressure in the primary chamber 155 when the meterin valve becomes operative. The point at which the metering valve becomes operative will be apparant from the following description of the functioning of the metering valve.

The rising pressure in the control chamber 157 following initial operation of the master cylinder as described above acts on the full diameter of the metering valve plunger 191 to generate a force which tends to act against the pre-load of spring 149, this force being supplemented by a small force from spring 198, used to ensure that the vall 188 seats on the plunger. When an appreciable pressure in the control chamber 157 has developed, plunger 191 moves against the load of spring 149 until the ball 188 rests on the pin 189, thrust member 192 remaining with its shoulder in abutment with collar 196 by virtue of spring 198. The force tending to move plunger 191 against spring 149 is now that developed by the annular area of plunger 191 outside the seating diameter with ball 188, and is without the help of spring 198 which is reacted through the ball 188 onto the pin 189. Hence the pressure in the control chamber 157 continues to rise at the same rate as that in the primary chamber 155 until the pressure is enough to move plunger 191 a further amount against the load of spring 149 and away from ball 188. This allows a quantity of fluid to flow out of the control chamber 157, through radial passage 172, through radial holes in thrust member 192, past the ball 188, into the bore of plunger 191 and out to tank through a radial hole 199 in the plunger 191, the bore space occupied by spring 149, radial holes 173 in the piston 141 corresponding holes in the piston 146, annular chamber 168, passage 169 and port 161. The effective piston area operative to expel fluid from the primary chamber 155 through outlet port 167 is now that of the inner diameter of seal 164.

With a small further increase in pressure in the primary chamber 155 caused by further brake-applying movement of the push rod 123, the metering valve plunger 191 acts as a relief valve, keeping the pressure in the control chamber 157 constant until the primary chamber pressure acting on cup seal 193 is sufficient to move the thrust member 192 against the load of spring 198 and into contact with the plunger 191. With both the pressure in the primary chamber 155 and in the control chamber 157 acting to move the plunger 191 against the spring 149, the metering valve becomes operative to progressively reduce the pressure in the control chamber 157 as pressure in the primary chamber 155 increases, eventually reaching tank pressure when valve plunger 191 moves further against spring 149 to rest against the base of the blind bore in the inner piston 141.

In the event of a hydraulic line failure to the brake actuator served by port 167, a disc 195A contacts the secondary piston 156, servo effort and manual effort being applied in proportions fixed by the reaction ring 187. The secondary piston 156 serves to transmit pressure from the primary chamber 155 to a secondary pressurising chamber 174 to actuate another group of brake actuators through an outlet port 175 in the usual manner of a tandem master cylinder. The secondary piston 156 is slotted for clearance around a roll pin 176 which limits rearward movement of piston 156 and unseats a recuperation valve 177. The roll pin 176 also provides a passage to the tank of hydraulic fluid.

In a modification, the thrust member 192 is firmly attached to the metering valve plunger 191 or is integral therewith. The metering valve works substantially in the manner described above, except that there is no intermediate operating phase during which the pressure in chamber 157 remains constant for increasing pressure in chamber 155 as fluid is relieved to tank past the ball 198. Instead, the increasing pressure in chamber 155 is effective to decrease pressure in chamber 157 as soon as the valve starts to open.

The stepped piston assembly is manufactured in two parts 141 and 146 for convenience. A ring 200 having tags which extend through radial holes in the outer piston 146 into a groove in the inner piston 141 encircling holes 173 keeps the pistons 141 and 146 together during assembly of the master cylinder.

Reference is hereby directed to our co-pending Application No. 11754/77 (Serial No. 1 589 082) which claims certain aspects of the master cylinder shown in Figure 2.

WHAT WE CLAIM IS:-

1. A servo-assisted hydraulic master cylinder comprising a variable ratio master cylinder of the kind in which a stepped piston is slidable in a stepped bore to provide a large effective piston area when output pressure is below a pre-determined magnitude and a small effective piston area when output pressure is above the pre-determined magnitude and in which the stepped piston and the stepped bore define a control chamber in which pressure is progressively varied by a metering valve when output pressure increases from the pre-determined magnitude to a higher magnitude to provide a ratio of input effort to output pressure which is intermediate that which pertains when output pressure is below the pre-determined magnitude and that which pertains when output pressure is above the higher magnitude, and a direct-acting servo of the kind having an output connected to the master cylinder, an operator controlled servo valve for controlling a pressure difference across a movable wall and reaction means which connects the movable wall to the output member and operates when the load transmitted by the output member exceeds a pre-determined magnitude to provide a reaction force on the servo valve which is proportional to an increment in the load transmitted by the output member above the pre-determined output pressure at which the metering valve becomes operative substantially coincides with the pre-determined magnitude of load at which the reaction means becomes operative.

2. A servo-assisted hydraulic master cylinder substantially as described herein with reference to the accompanying drawings.

For the Applicants
L. H. STREET.
Solicitor

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2 SHEETS *This drawing is a reproduction of the Original on a reduced scale*

Sheet 1

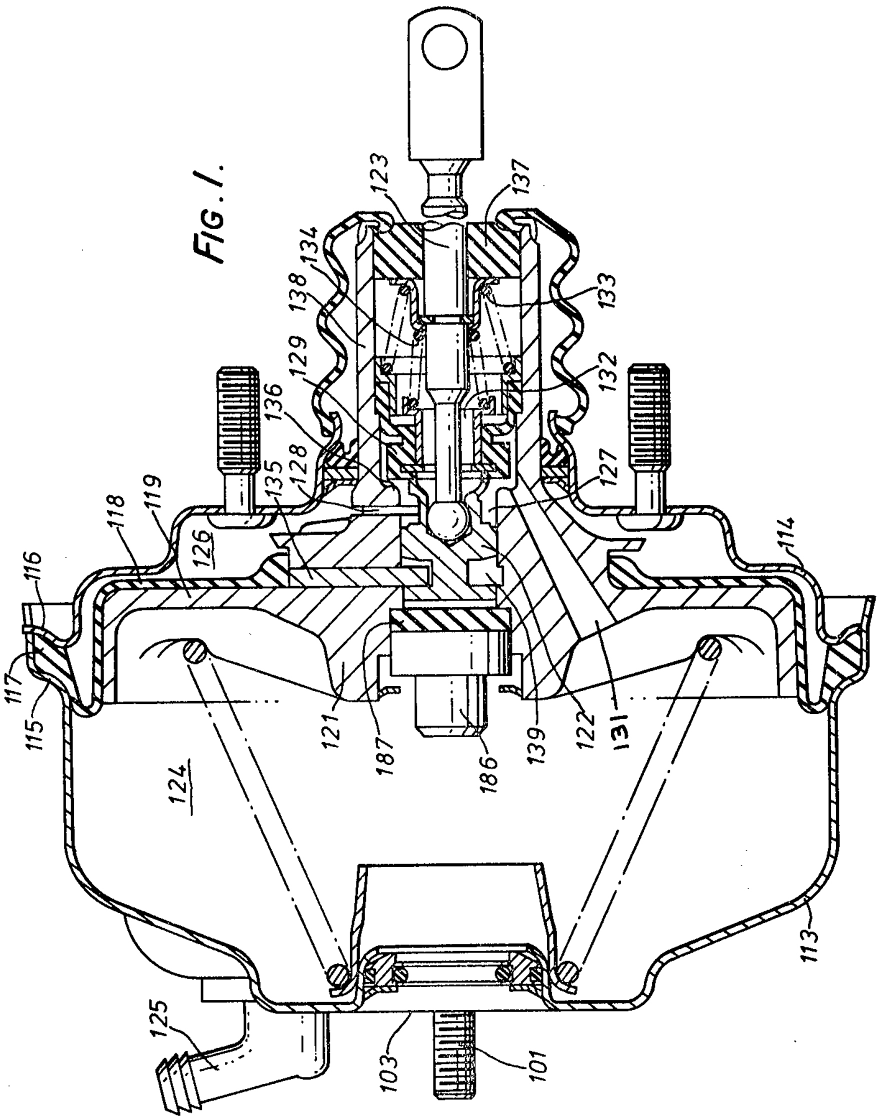
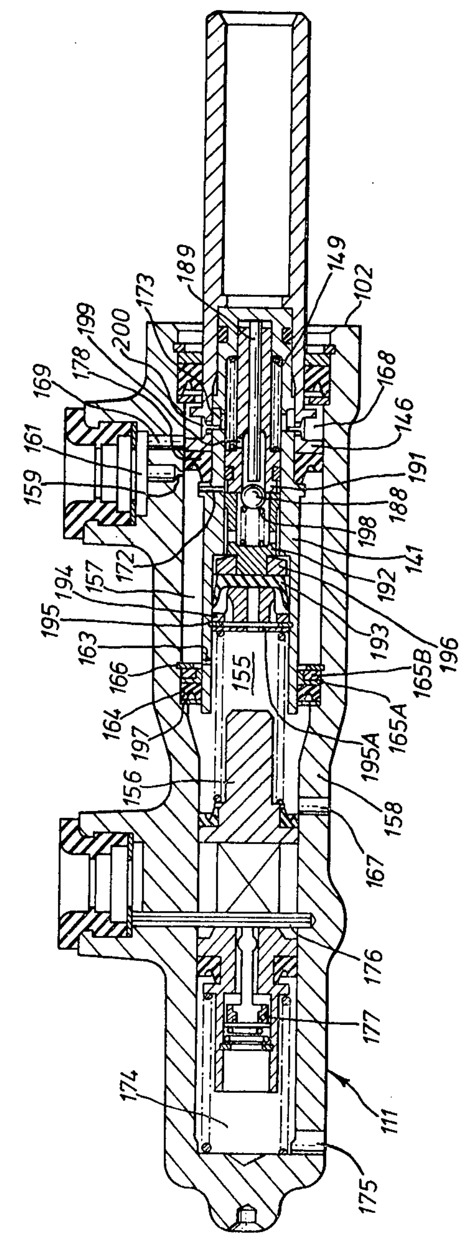


FIG. 1.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195A
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FIG. 2.