June 18, 1963 A. BRUEDER 3,094,078

SWITCHING DEVICES FOR HYDRAULIC PUMPS AND MOTORS

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Fig. 1

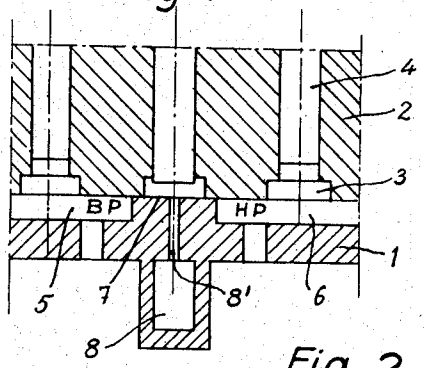

Fig. 3

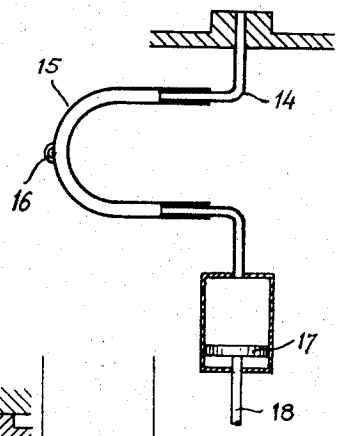

Fig. 2

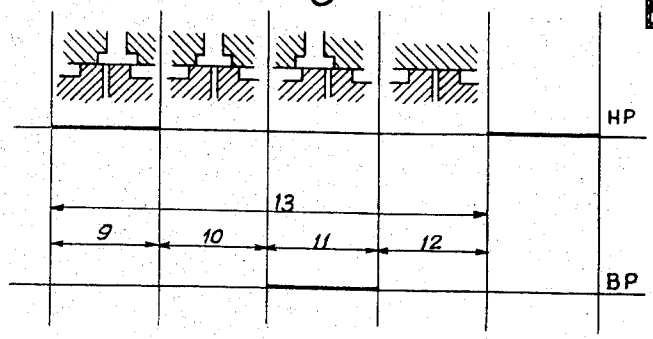

Fig. 4

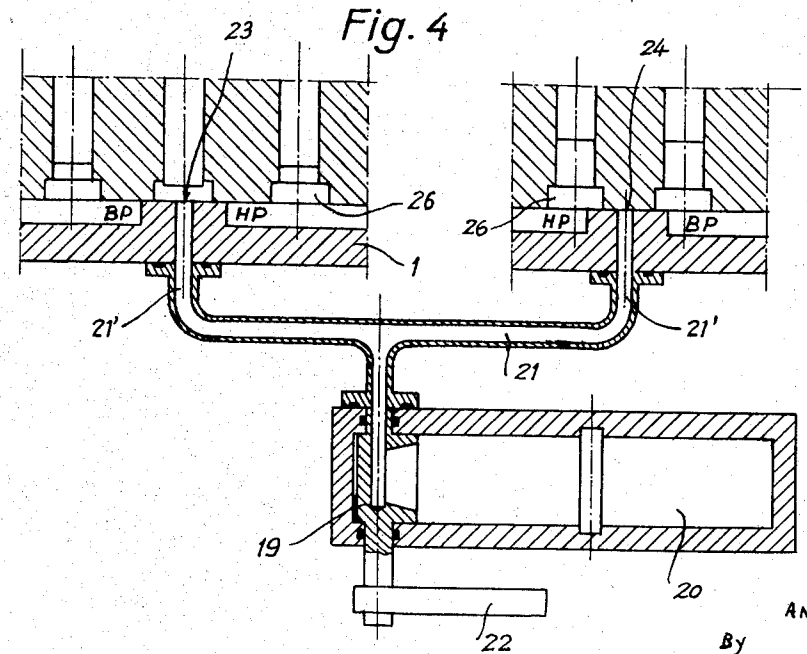

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Fig. 5

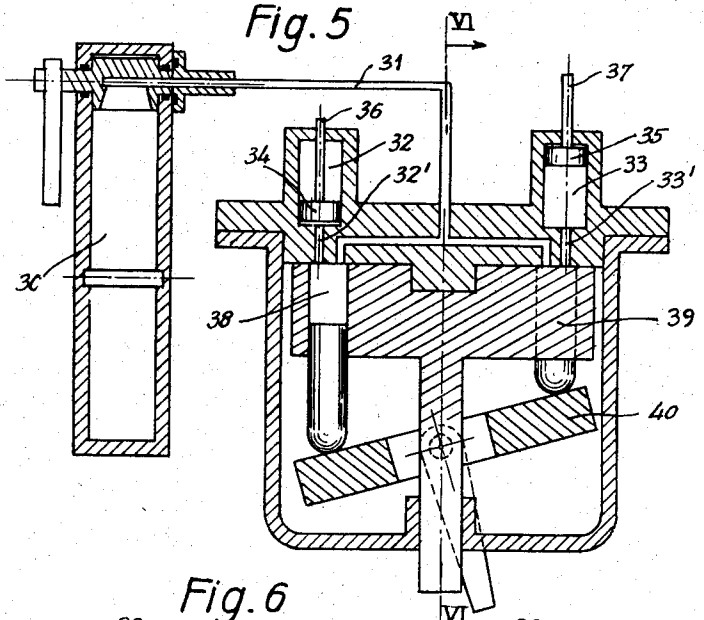


Fig. 6

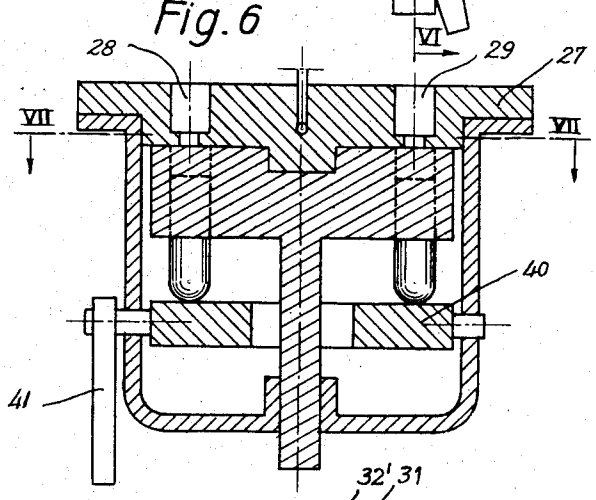
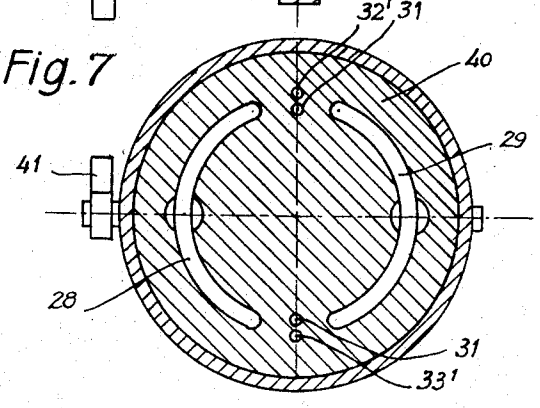


Fig. 7



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3,094,078
SWITCHING DEVICES FOR HYDRAULIC PUMPS AND MOTORS

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6 Claims. (Cl. 103—162)

The present invention relates to a device for switching hydraulic pumps and motors and more generally to the application of resonators to the switching of hydraulic pumps and motors.

The difficulty of switching hydraulic pumps and motors of the distributor type, notably for relatively high pressures, is well known to those conversant with the art. Many means have already been proposed with a view to obtain a noiseless operation and a higher efficiency. Thus, some known propositions are based on the shifting of the distributor, or the provision of valve feed systems with or without an auxiliary source of fluid.

Interesting results are provided by these methods under well-defined output, pressure and rating conditions.

But in certain application such as hydraulic transmissions, these parameters are particularly variable, and their rate of variation is sometimes relatively high; moreover, the change in the direction of rotation and the fact that the high-pressure side acts as a brake make this switching problem more complicated and the difficulty of applying the means set forth hereinabove is increased in proportion.

It is the essential object of this invention to provide a satisfactory solution to this problem with due consideration for the various parameters involved, by using a readily adjustable static system characterized by a short response time. It consists essentially in connecting each separating or overlapping surface of the distributor associated with a hydraulic motor or pump with a resonator-forming cavity, so that the latter be energized by the pump or motor pressure and becomes operative to permit a gradual pressure connection in the dead space or cylinder capacity during the overlapping.

The invention will now be described with reference to the accompanying drawings illustrating diagrammatically by way of example a typical form of embodiment applicable to any similar or other types of slide-valve pumps and distributors. In the drawings:

FIGURE 1 is a fragmentary developed section showing a conventional axial-pump distributor provided with a resonator according to this invention;

FIGURE 2 is a diagram plotting the pressure as a function of time in the resonator of FIG. 1, the upper portion of this figure showing diagrammatically the relative position of a cylinder with respect to the distributor for each cycle of operation;

FIGURE 3 is a diagrammatic view showing a device according to this invention for adjusting the resonator;

FIGURE 4 is a developed fragmentary section showing the distributor of FIG. 1 with a resonator of the vortex type;

FIGURE 5 is a vertical section showing a modified form of embodiment of the invention;

FIGURE 6 is a section taken upon the line VI—VI of FIG. 5, and

FIGURE 7 is a cross-section taken upon the line VII—VII of FIG. 6.

In FIG. 1 there is shown in fragmentary section a developed view of a distributor 1 of a conventional axial pump 2 comprising cylinders 3 in which pistons 4 are slidably mounted. The pump rotation causes the cylinders to communicate by turns with the liquid input and output manifolds 5, 6 respectively, the distributor comprising so-called overlapping separating surfaces 7 between the manifolds, these surfaces being coincident with the passages of the pistons 4 at their top and bottom dead centres.

According to the present invention, there is provided a closed chamber 8 communicating through a duct 8^1 with the distributor, this duct opening on the separating surface.

Thus, during the overlapping period the cylinder capacity or the dead space, according as the piston is at its bottom dead centre or at its top dead centre, communicates only with the chamber 8 and the corresponding duct 8^1, acting as a resonator.

The dimensions of the overlapping surfaces and of the heads of the slide-forming cylinders 3 are so calculated that the resonator formed by the chamber 8 and duct 8^1 be energized by the pressure of the hydraulic pump or motor, and becomes the seat of a periodic pressure variation of which FIG. 2 represents the curve as a function of time; this function is periodic and its period is

$$\frac{2\pi}{n\omega}$$

wherein n is the number of cylinder of the pump or motor, and ω the angular speed. Considering more particularly the resonator corresponding to the passage of the piston shown in FIG. 1 at the bottom dead centre in the time interval 9 (FIG. 2), the resonator will communicate with the high-pressure manifold HP; in the time interval 10, with the cylinder 3 alone; in the time interval 11, with the low-pressure manifold BP, the resonator being isolated at 12.

This periodic energization produces in the resonator an alternating output which, by being adequately phase-shifted with respect to the pressure by a suitable selection of the resonator characteristics, permits of properly effecting the connection in the dead space of the cylinder or in the cylinder during the overlapping between the induction cycle and the exhaust cycle at the dead centre passage (according as it is the bottom or top dead centre).

The output amplitude and phase-displacement depend on the impedance Z of the resonator of which the inertance is

$$\rho \frac{l}{s}$$

and the capacitance

$$\frac{V}{K}$$

wherein:

ρ is the specific mass of the liquid,
l is the length of duct 8^1 (FIG. 1)
s is the cross-sectional area of this duct (FIG. 1)
V is the volume of chamber 8 (FIG. 1)
K is the module of compressibility of the liquid

To this a resistance term due to the loss of pressure in the duct 8^1 and to the loss of speed at the outlet end of this duct must be added.

The geometrical dimensions of the duct and chamber provide the desired impedance Z.

The rear phase displacement which had to be

$$\frac{\pi}{2}$$

will be only very close thereto due to the resistance term. The energy absorbed for this transfer will be very moderate for the output will be practically "reactive" in comparison with the wattless current of the electrical systems.

In FIG. 3 there is shown by way of example a device for adjusting the resonator.

The length of duct 14 is variable through the displacement of the U-shaped pipe element 15 controlled at 16

either from the regulation system or manually. The volume **1** of the chamber varies as the piston **17** is moved up or down in its cylinder, the piston rod **18** being also controlled either by the regulating system or manually. The regulation system will be subordinate to the rate of operation and possibly of the pressure and volume (cylinder capacity or dead space) to be subjected to the pressure connection.

By using a vortex resonator (FIG. 4) the desired impedance Z may be obtained without varying the capacitance, but in this case only the inertance is acted upon.

This adjustable resonator consists of a chamber **20** (in this case a circular chamber) communicating with a pivoting choke **19**.

With this arrangement, the inertance of the duct **21** is completed by the cariable inertance of the vortex created by the tangential component of the input or output speed of choke **19**. This tangential component is a function of the angle of setting of the choke. The angular displacement is controlled by means of a lever **22** responsive to the regulation system or to manual adjustment means.

This resonator may be used for the two dead centres mainly in pumps or motors having an odd number of cylinders.

FIGURES 4 and 5 illustrate this application and the alternate positions of ducts **21'** with respect to the cylinders may be clearly seen therein.

Thus, when at the bottom dead centre **23** the cylinder communicates with the resonator, the communication is shut off at the top dead centre **24**, and vice versa. If reference is made to the diagram of FIG. 2 to explain the energization of the resonator, it will be seen that in position **9** the resonator communicates with the high-pressure manifold **26** through the two dead centres **23** and **24**; at **10**, the cylinder in the bottom dead centre position communicates with the resonator of which the top dead centre orifice is closed; at **11**, the resonator communicates with the low-pressure manifold through the two dead centres **23** and **24**, at **12** the cylinder passing through the dead centre **24** communicates with the resonator of which the orifice of dead centre **23** is closed. Therefore, the function of the resonator is to transfer with a very good efficiency the expansion energy from one cylinder in the form of recompression energy into the other cylinder.

However, the excitation would not be symmetrical, notably for the maximum volumetric adjustment of the pump or motor, for the cylinders passing through the dead centres **23** and **24** have a different cubic capacity. To restore the symmetry, a complementary volume corresponding to the cylinder capacity defined by the volumetric adjustment must be added to the cylinder in the dead centre position.

In FIG. 5 there is shown by way of example in the specific case of an axial pump or receiver, provided with a distributor **27** receiving the manifolds BP (low-pressure) **28** and HP (high-pressure) **29**, a device meeting this requirement. The assembly comprises the vortex resonator **30** communicating through a duct **31** with the overlapping surfaces and at each dead centre there is an adjustable volume consisting of a cylinder **32** or **33**, provided with a piston **34** or **35**, actuatable by means of its rod **36** or **37**.

The resonator **30** will communicate simultaneously with the cylinder **38** passing through a dead centre position and with the corresponding auxiliary volume as shown in FIG. 5. The piston **34** will reduce the volume of the cylinder to a minimum for it communicates with the cylinders passing through the bottom dead centre, whilst the piston **35** will give to the cylinder **39** the volume corresponding to the cubic capacity of a piston. The displacement of piston **35** will be adjusted as a function of the volumetric adjustment of the pump or receiver which is effected by inclining a plate or disc **40** by means of a lever **41** in this specific example. For negative inclinations of the plate or disc **40** the functions of the cylinders **32** and **33** will be reversed.

This device suitable for a practical application is efficient and reliable in operation. Moreover, it is characterized by the advantageous feature of being applicable to the switching of hydraulic pumps and motors operating under extremely variable speed conditions.

I claim:

1. In a hydraulic machine operating as a generator or a receiver of the type comprising a rotating barrel fixed on a shaft, spaced cylinders provided in said barrel, a piston in each cylinder, and a distributor having high and low pressure manifolds cooperating with the cylinders, and connecting surfaces between said high and low pressure manifolds, the combination with a resonator and a duct connecting said resonator with said connecting surfaces to permit a gradual connection of the pressure with said cylinders, and means for regulating the frequency at which the resonator is resonant.

2. A device as set forth in claim 1 wherein the length of the duct is adjustable.

3. A device as set forth in claim 1, wherein the resonator comprises a pivoting choke communicating with the duct and means for manual adjustment of this choke.

4. A device as set forth in claim 1, wherein a resonator is connected to each connecting surfaces.

5. A device as set forth in claim 1, wherein a complementary volume is connected by a duct with each of said connecting surfaces.

6. A device as set forth in claim 5, wherein means are provided to adjust the volume of the complementary volume.

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