Sept. 29, 1953 L. F. MOTT 2,653,543

HYDRAULIC PUMP

Filed May 2, 1951 5 Sheets-Sheet 1

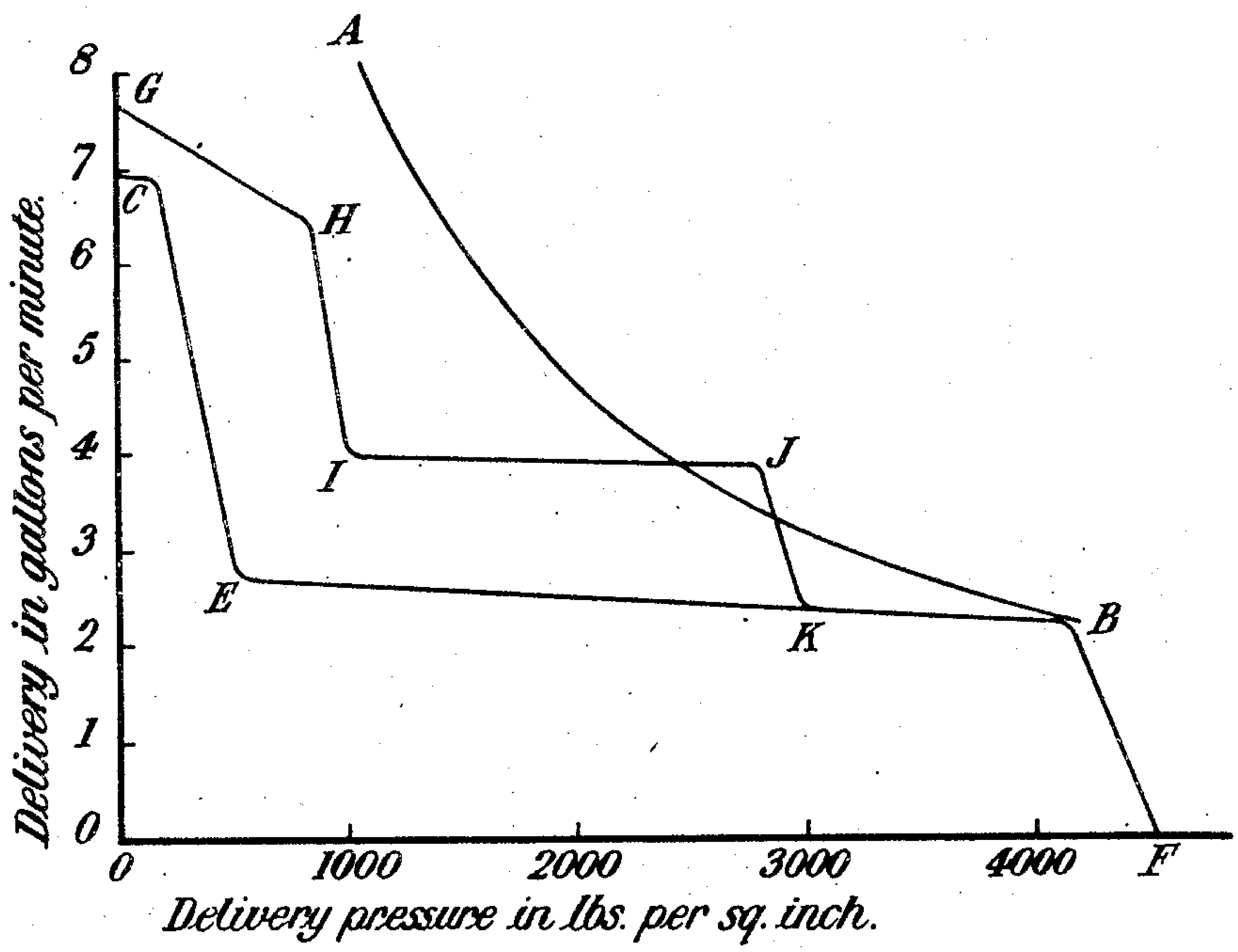

Fig. 1.

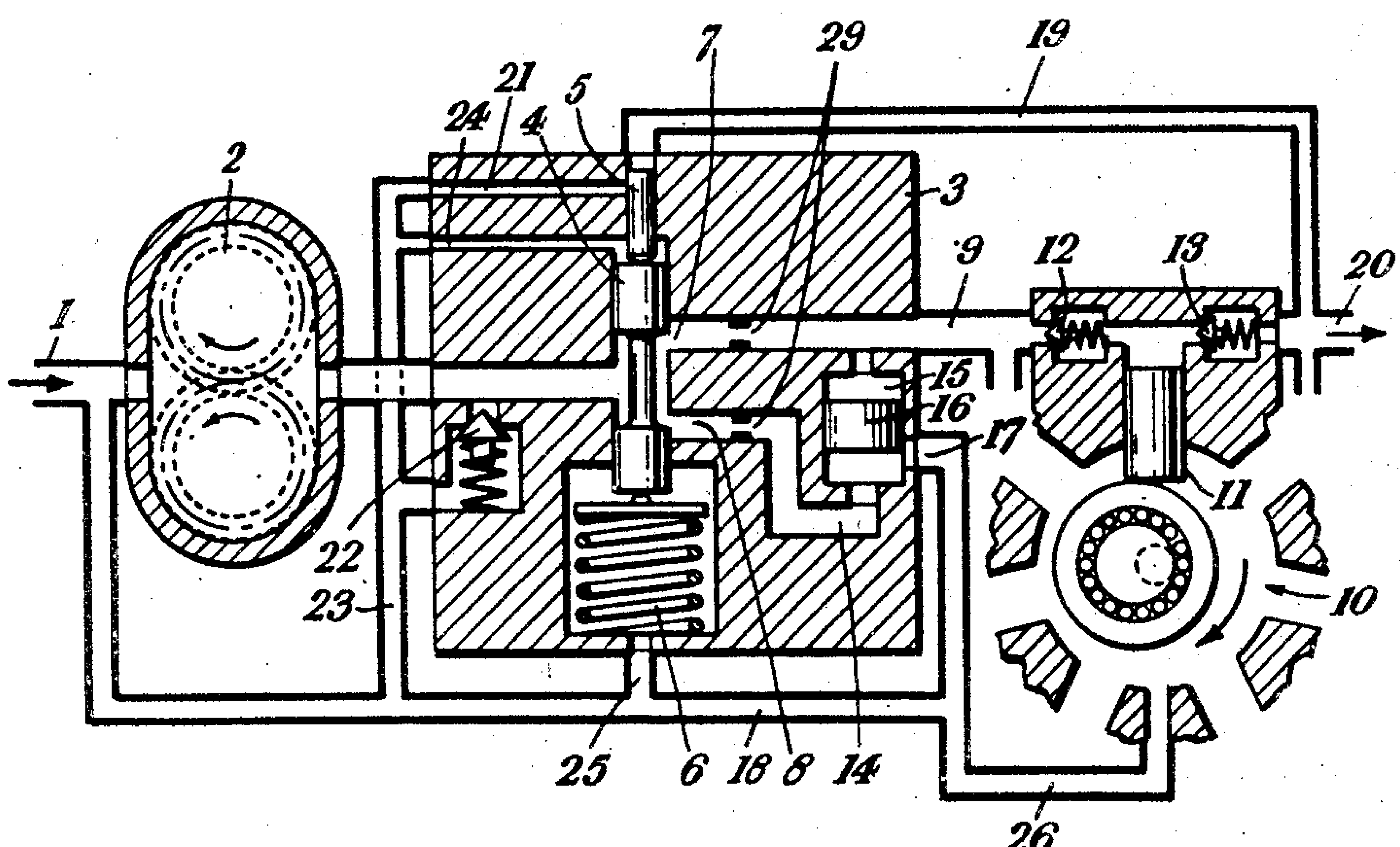

Fig. 2.

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Sept. 29, 1953 L. F. MOTT **2,653,543**

HYDRAULIC PUMP

Filed May 2, 1951 5 Sheets-Sheet 2

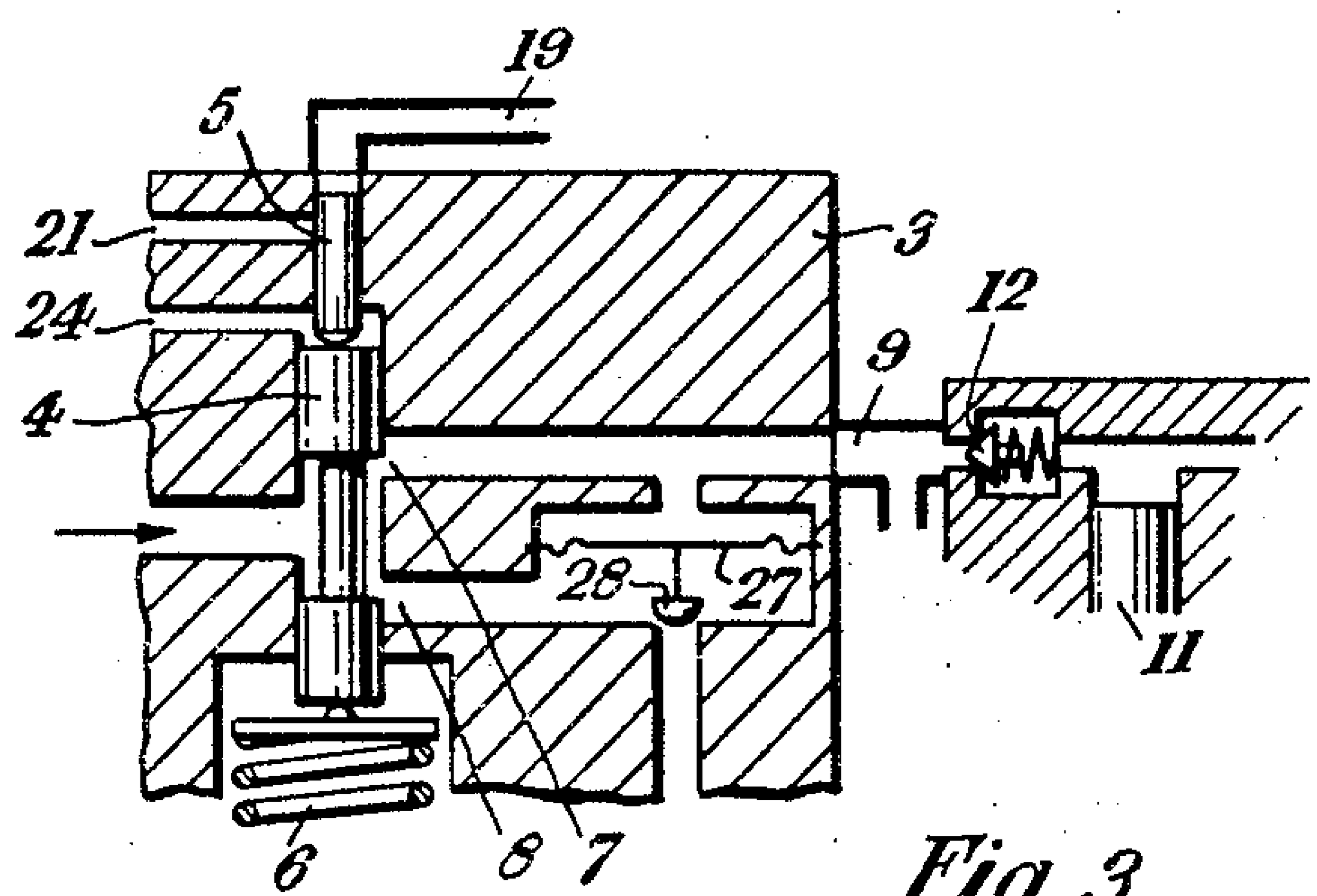

Fig. 3.

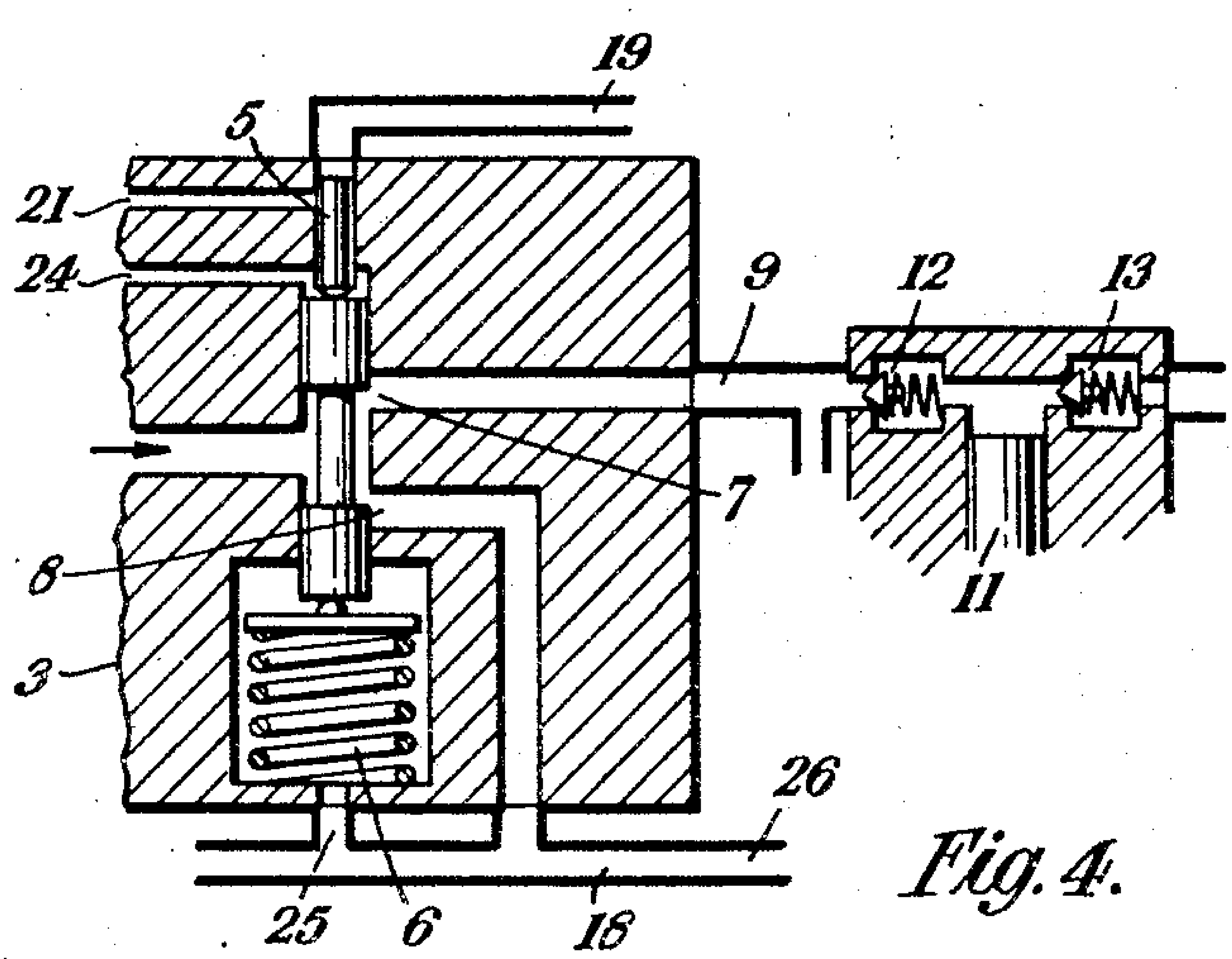

Fig. 4.

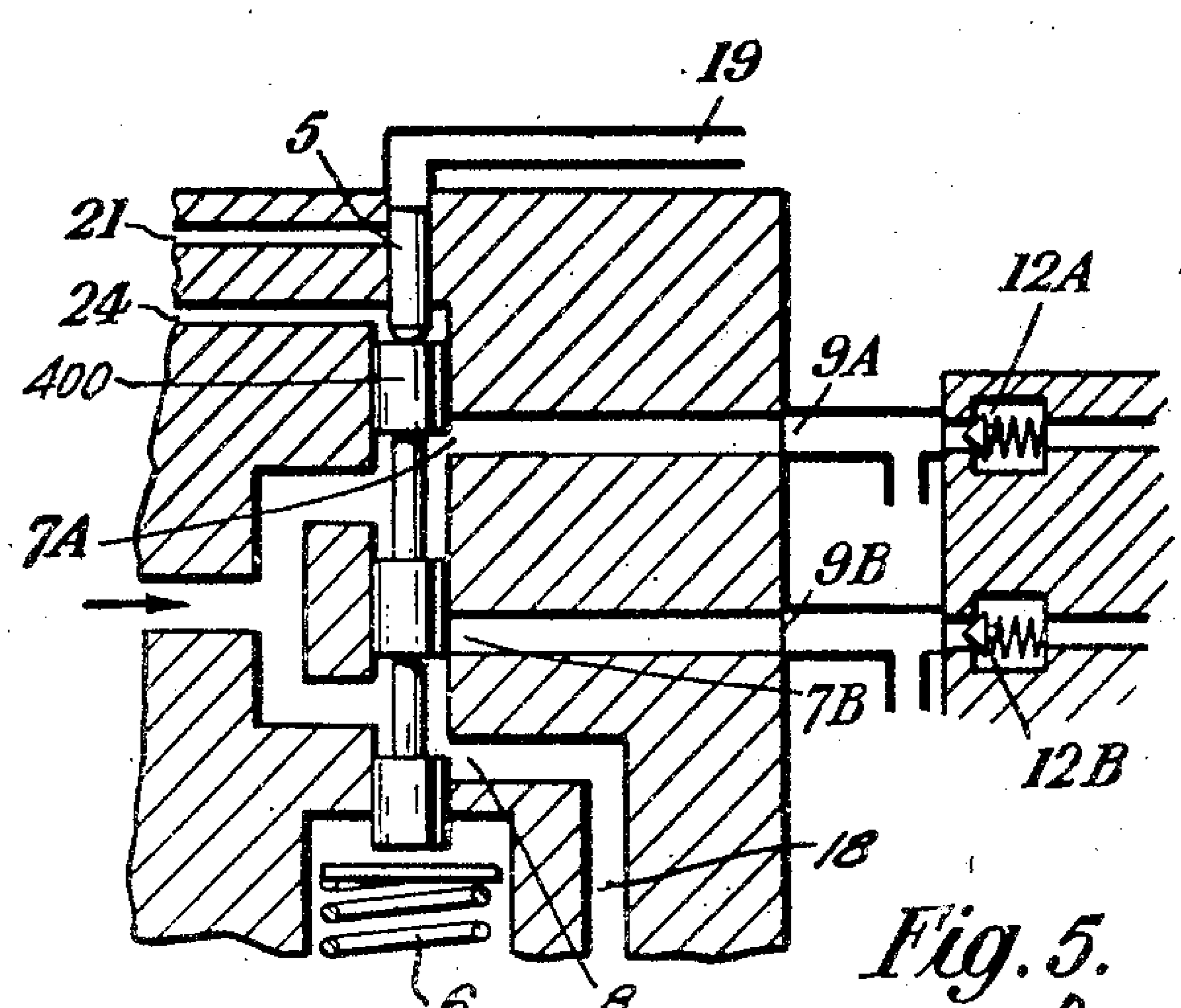

Fig. 5.

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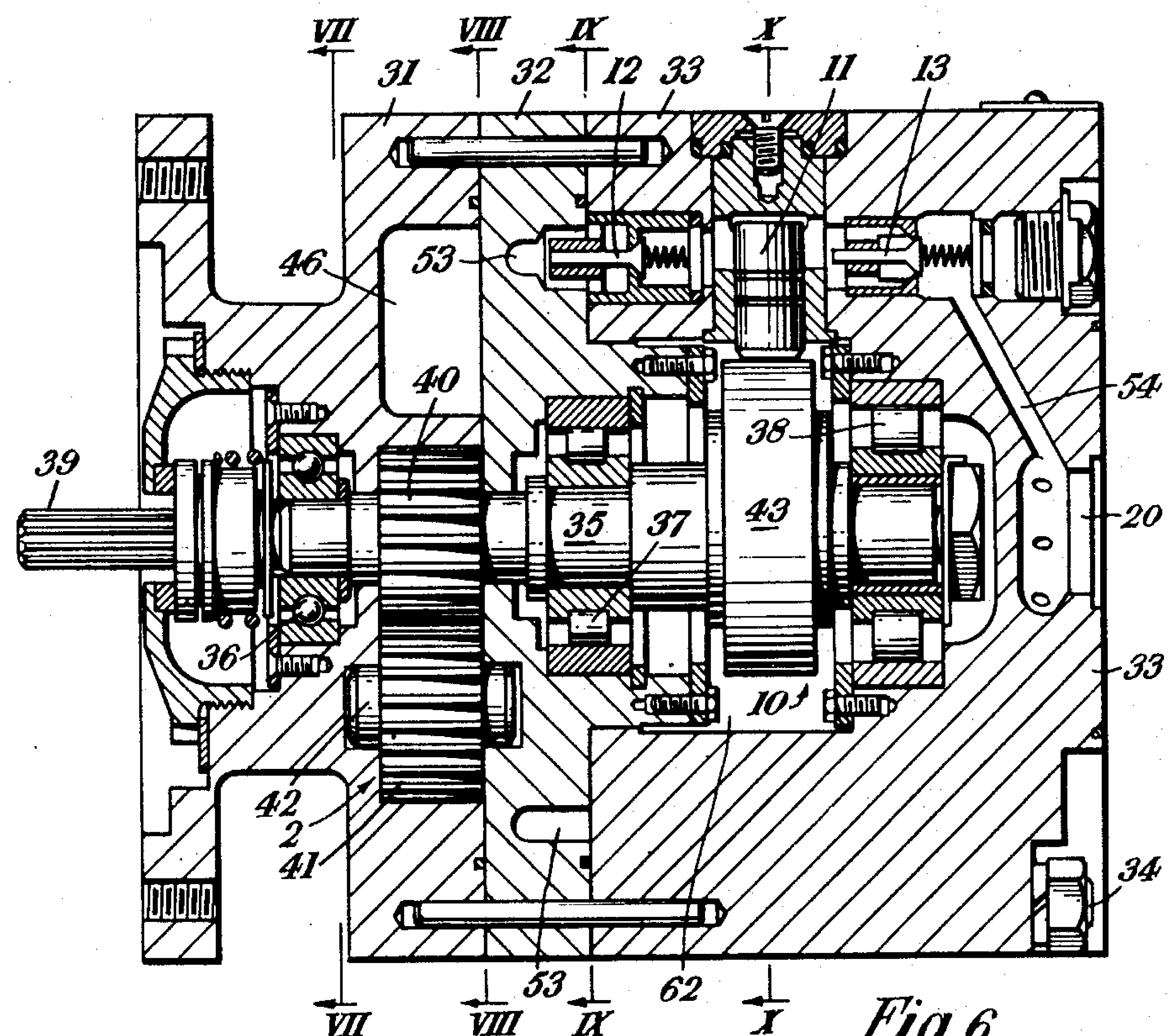

Fig. 6.

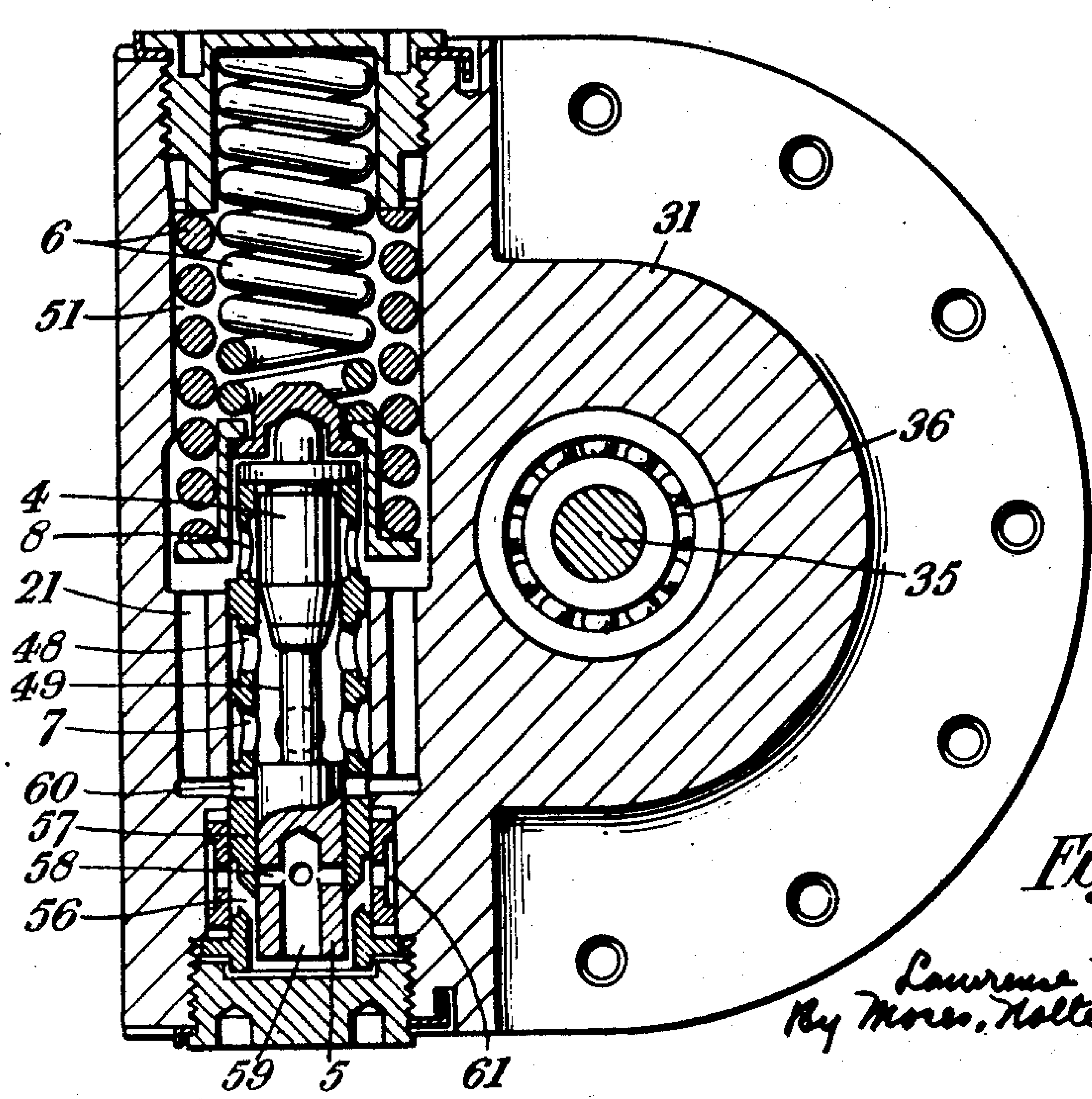

Fig. 7.

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HYDRAULIC PUMP

Filed May 2, 1951

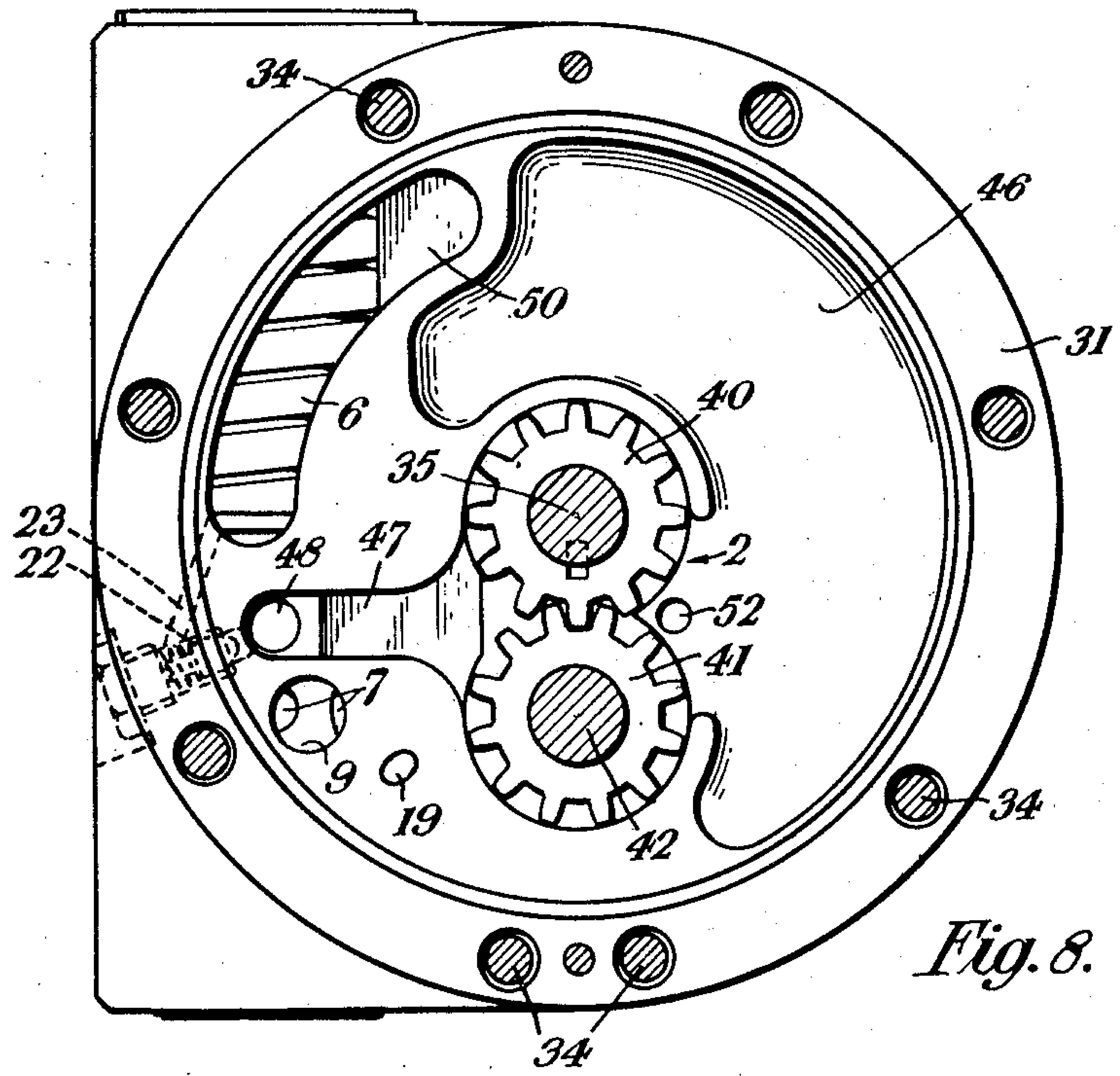

Fig. 8.

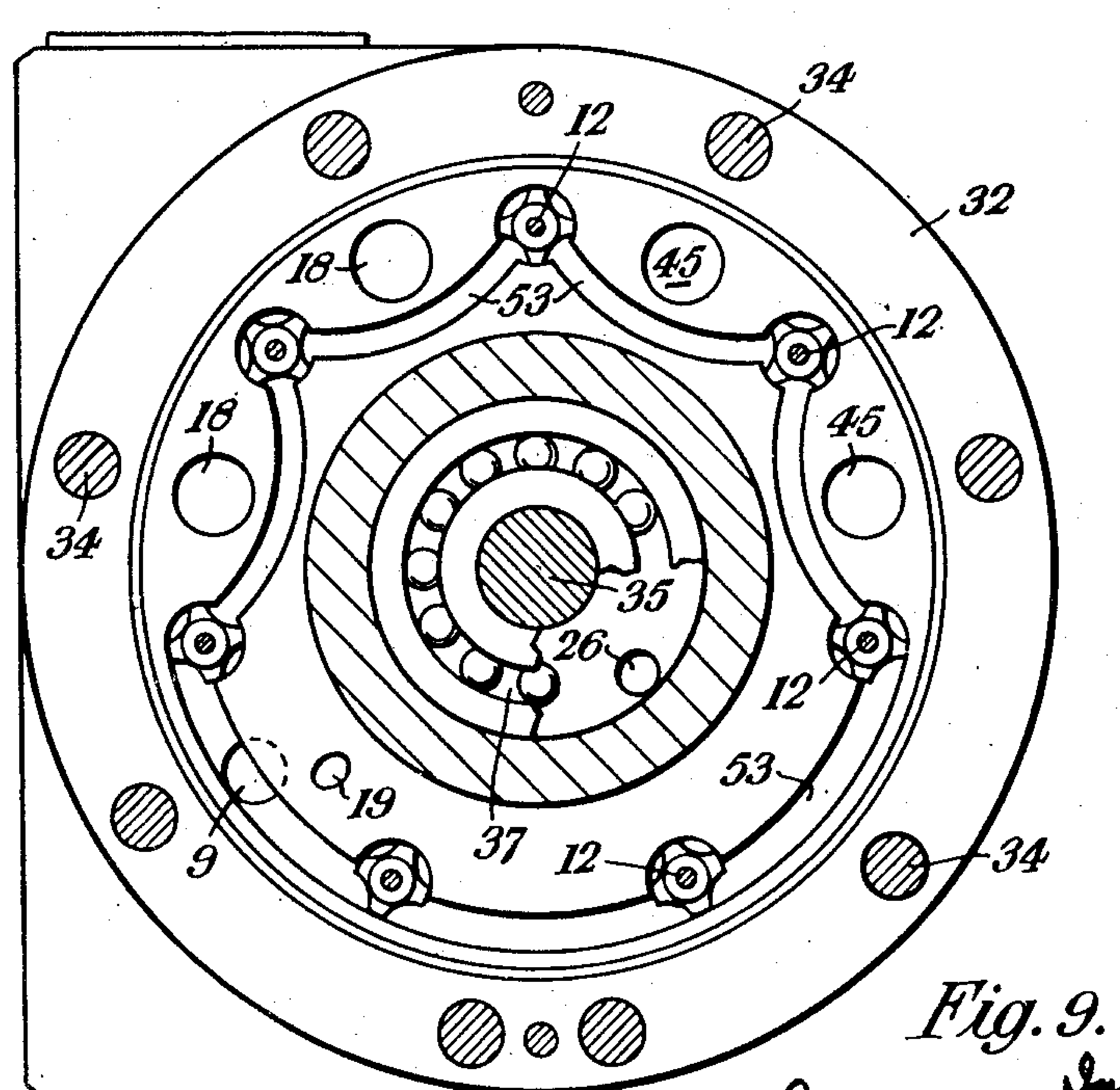

Fig. 9.

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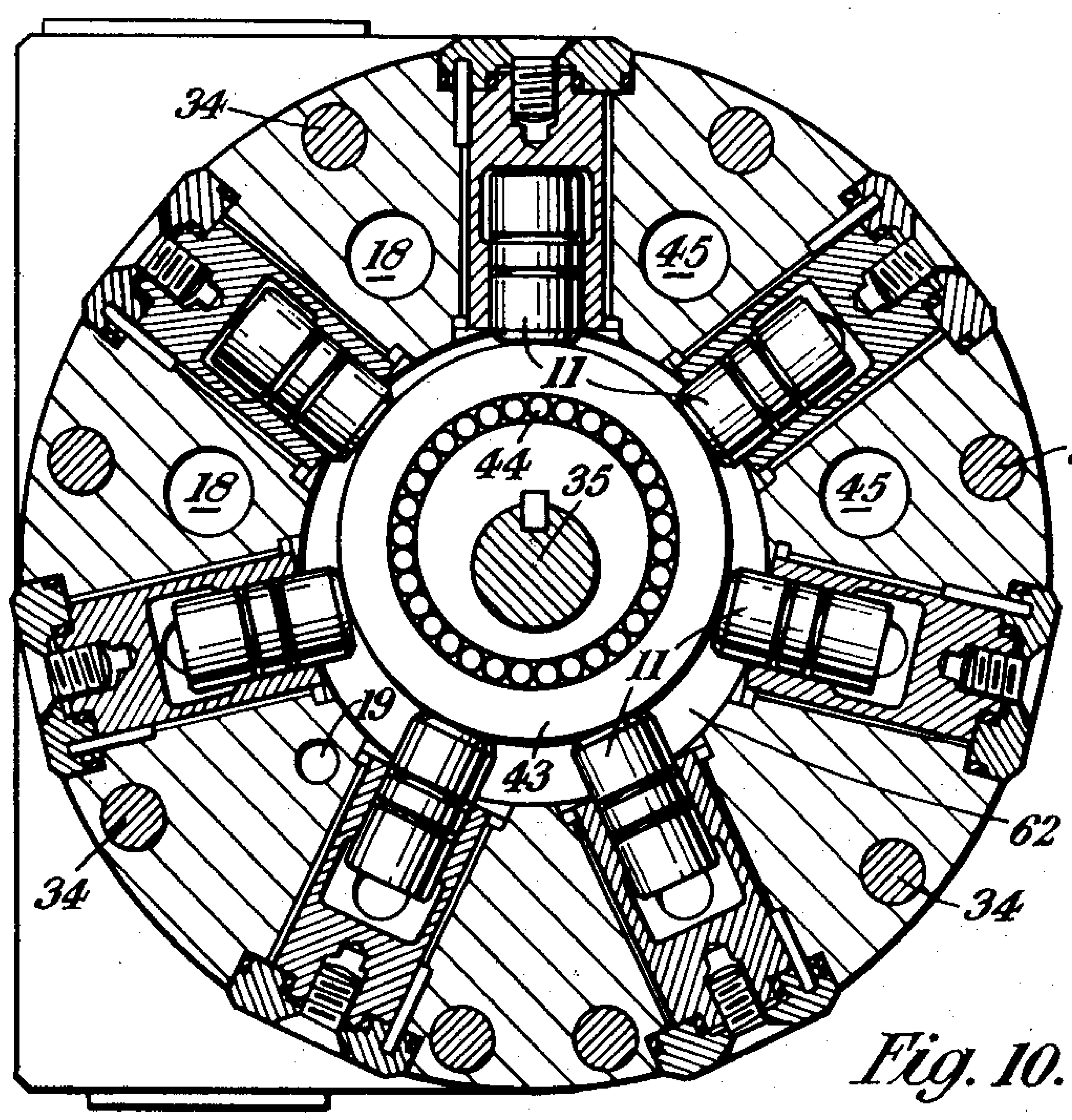

Fig. 10.

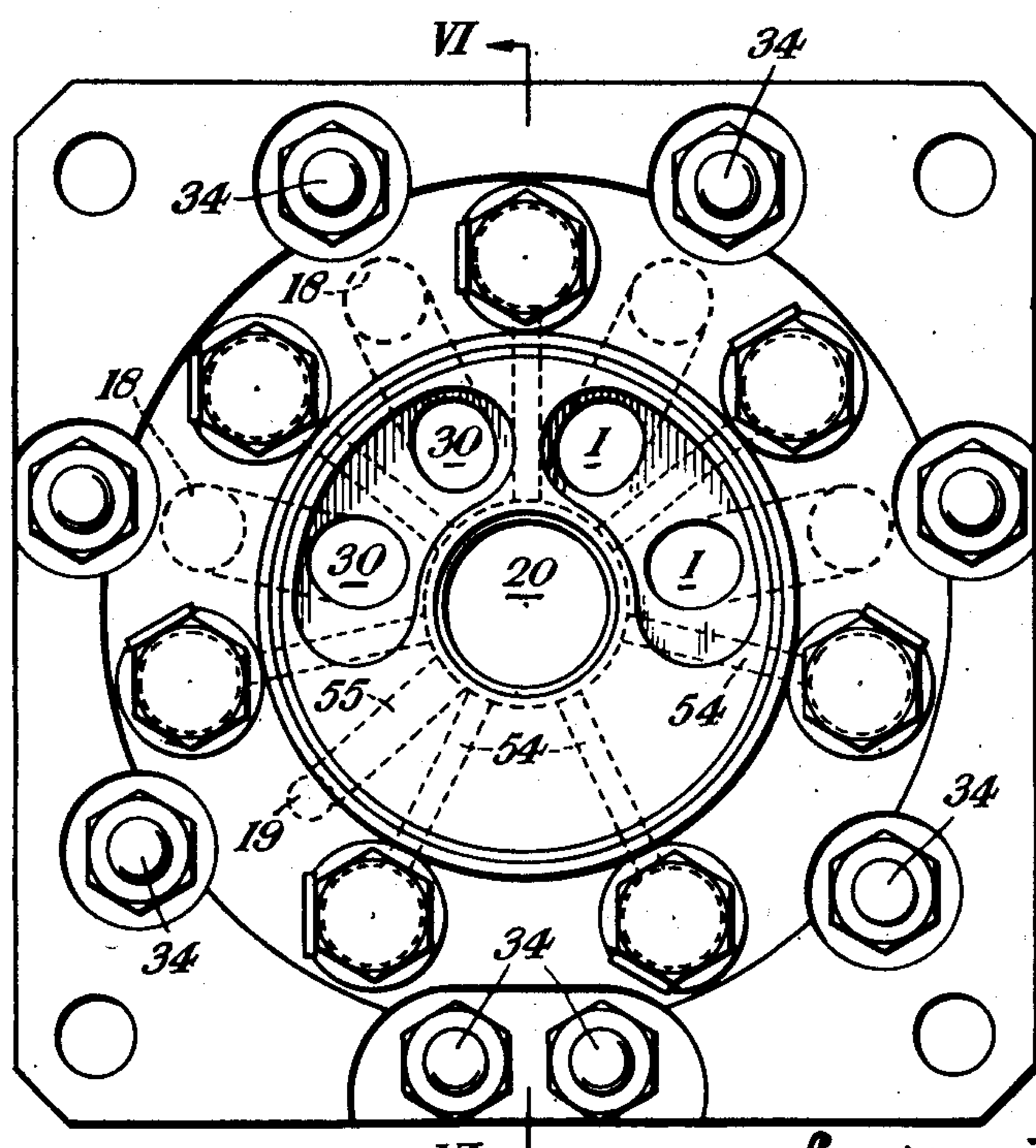

Fig. 11.

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UNITED STATES PATENT OFFICE

2,653,543

HYDRAULIC PUMP

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Application May 2, 1951, Serial No. 224,222
In Great Britain May 3, 1950

6 Claims. (Cl. 103—5)

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This invention relates to two stage hydraulic pumps of the kind comprising two pumping units arranged in tandem, and an off-loading valve, subject to the delivery pressure of the second stage, for automatically controlling the delivery of the pump by reducing, as the delivery pressure rises, the pressure at which liquid is supplied to the second stage pumping unit from the first stage pumping unit.

The invention relates especially to pumps of the above kind in which the second stage pumping unit is a piston pump having a number of radially arranged cylinders, whose pistons are operated on their working stroke by a cam or eccentric, with which they are held in contact only by the liquid supplied to them by the first stage pumping unit. It is, however, also applicable to pumps in which the second stage pumping unit comprises but a single piston, or a number of pistons arranged in cylinders disposed in line, in one or more rows, and in which the pistons are reciprocated by any desired type of actuating member, e. g. a cam, eccentric, crank shaft or swash-plate. Furthermore, the piston or pistons of the second stage pumping unit may be held in contact with the actuating member by fluid pressure, by springs, or may be mechanically attached thereto.

The relative performances of certain hydraulic pumps of this kind are shown in Fig. 1 of the accompanying drawings in comparison with the theoretical ideal represented by the curve AB. This ideal characteristic is that according to which the product of the delivery, in gallons per minute, and delivery pressure remains constant over a given range of delivery pressure.

As fully explained in U. S. application Serial No. 165,276, now Patent No. 2,643,613, the curve CEBF in Fig. 1 represents the characteristic of a pump in which the inlet valves to all the pistons of the second stage are simultaneously rendered inoperative, while curve GHIJKBF represents the charactertistic of a similar type of pump including an off-loading valve of the kind described in U. S. application Serial No. 165,276, now Patent No. 2,643,613. In this pump, liquid from the first stage is initially passed by the off-loading valve direct to the pump outlet as well as to the piston stage of the pump, and the delivery pressure is the same as the pressure developed by the first stage over the range GH. At the point H, the increased delivery pressure actuates the off-loading valve to effect a sudden reduction in the pressure of the liquid supplied from the first stage to the piston stage and at the same time cuts off the direct flow of liquid from the first stage to the outlet. The piston stage then operates at substantially constant delivery and with all its pistons effective, as indicated by IJ. At the point J the off-loading valve operates again, in response to the increased delivery pressure, to effect a further reduction in the pressure supplied from the first stage to the piston stage. The piston stage then operates, with some only of its pistons effective, at substantially constant delivery as indicated by KB. Finally, when the delivery pressure represented by B is reached, the off-loading valve moves to cut off communication between the first stage and the piston stage and the pump is completely off-loaded. It will be observed that, even with the improved form of off-loading valve described in U. S. application Serial No. 165,276, now Patent No. 2,643,613, the delivery remains substantially constant over comparatively wide ranges of varying delivery pressure represented by IJ and KB.

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The object of the present invention is to effect a further improvement by giving a characteristic curve which approaches more closely to the ideal curve AB.

The invention accordingly provides a hydraulic pump of the above described kind, in which liquid cannot flow directly from the first stage pumping unit to the outlet of the pump and in which the off-loading valve operates automatically, in accordance with variations in the delivery pressure of the pump over a given range, to regulate the rate of flow of liquid from the first stage pumping unit to the second stage pumping unit in such manner that the delivery of the pump decreases progressively and more or less continuously, without sudden changes, as the delivery pressure increases.

By appropriate design of the off-loading valve, a characteristic approximating to that shown by AB in Fig. 1 can be obtained.

When the second stage pumping unit is a piston pump, the total delivery from the cylinders of the piston stage will be less than the amount entering by only a small amount occasioned by leakage past the valves and pistons of this stage, and the desired relation between delivery pressure and delivery rate can be substantially achieved by regulating, by appropriate design of the off-loading valve, the relation between delivery pressure and rate of entry of the liquid into the cylinders. The maximum delivery of the first pumping stage, which is preferably a gear pump, may be equal to or greater than the maximum delivery of the piston stage.

Certain embodiments of the invention will now be described in detail, by way of example, with reference to the accompanying drawings, in which:

Fig. 1 shows the graphs above-referred to, representing the theoretical ideal AB, the performance CEBF of the known pump above referred to and the performance GHIJKBF of the pump according to U. S. application Serial No. 165,276,

Fig. 2 is a diagram showing one form of pump according to the present invention,

Figs. 3 and 4 show diagrammatically, modifications in the connections between the off-loading valve and the piston stage of the pump shown in Fig. 2,

Fig. 5 is a diagram showing an alternative form of off-loading valve,

Fig. 6 is a sectional elevation, taken on the line VI—VI in Fig. 11, through a pump according to the invention, operating in general accordance with the diagram of Fig. 4,

Figs. 7–10 are sections taken respectively on the lines VII—VII, VIII—VIII, IX—IX and X—X in Fig. 6, and

Fig. 11 is an end elevation of the pump, looking from the right-hand side of Fig. 6.

Like reference numerals designate like parts throughout the figures.

In the construction shown in Fig. 2, liquid is admitted via a passage 1 to the gear stage 2 of the pump and thence to a housing 3. Within the housing 3 is a piston type off-loading valve 4 which is slidable within a cylindrical hole formed in the housing 3. Coaxial with the piston 4 is another smaller piston 5 also slidable in a coaxial hole formed in the housing 3. A compression spring 6 is located coaxially with the valve 4 and piston 5 and abuts against the bottom of the valve 4, normally holding it and the piston 5 in the raised position shown. Lands on the valve 4 control the degree of opening of ports 7 and 8 which are so disposed that, when the effective area of one is increasing, that of the other is decreasing.

The port 7 communicates by a passage 9 with the piston stage 10 of the pump, one piston of which is shown at 11. Liquid enters the cylinder of the piston 11 by a non-return valve 12 and is expelled via a further non-return valve 13. The port 8 communicates by a passage 14 with one end of a cylinder 15, within which is a slidable piston 16. The piston 16 regulates the degree of uncovering of a port 17 formed in the wall of the cylinder 15 at the end which is in communication with the port 8. The port 17 communicates with the gear stage inlet via passage 18. The other end of the cylinder 15 is in communication with the passage 9.

The hole containing the piston 5 is connected via a passage 19 with the delivery line 20. A port 21 is formed in the wall of the bore containing piston 5 and is so positioned that it is only uncovered by the piston 5 when the piston has been displaced against the reaction of spring 6 by a predetermined delivery pressure applied to the piston 5 through the passage 19. Moreover the port 21 will not be exposed until the port 7 has been fully closed and the port 8 fully opened by displacement of the valve 4.

A spring loaded relief valve 22 is in communication with the gear stage delivery passage, the outlet from this valve being connected by a passage 23 to the passage 18 and thence to the gear stage inlet. The port 21 also communicates via passages 23 and 18 with the gear stage inlet, as do passages 24 and 25 and 26 which are connected respectively to the end space above the valve 4, to the spring chamber and to the piston stage eccentric chamber.

When there is no restriction to flow of liquid in the delivery line 20 the delivery pressure is low and the valve 4 and piston 5 are fully displaced upwards so that the port 7 is fully open and the port 8 is fully closed. The liquid delivered by the gear stage 2 is constrained to pass along the passage 9 and hence to the piston stage 10, apart from any liquid which may pass through the relief valve 22. This valve is adjusted so that it maintains a sufficient pressure at the inlet valves 12 completely to charge the cylinders of the piston stage 10 at the particular speed of revolution required. The rate of discharge of liquid through the valve 22 will therefore be the amount by which the delivery rate of the gear stage exceeds the rate of flow into the piston stage plus the rate of seepage into the passages communicating with the gear stage inlet.

If a degree of restriction is applied to flow in the delivery line 20, the delivery pressure will rise until the force exerted by this pressure upon the piston 5 will commence to overcome the spring 6, causing the valve 4 to commence to close the port 7 and at the same time to commence to open the port 8. This pressure corresponds to the point A in Fig. 1.

Further restriction to flow in the delivery line 20 will cause a further rise in delivery pressure, causing further downward displacement of the piston 5 and valve 4 against the spring 6 by an amount proportional to the pressure acting on the piston 5, the displacement of the valve 4 causing a reduction of area of the port 7 coincidentally with an increase of area of the port 8.

The liquid entering the piston stage via the passage 9 will exert a certain pressure upon the upper surface of the piston 16. Liquid flowing through the port 8 and thence via the passage 14 into the lower end of the cylinder 15, and thence through the port 17 and passage 18 to the gear stage inlet, will exert a certain pressure upon the underside of the piston 16. If the flow of liquid through the port 17 is sufficiently restricted, by reason of the port 17 being partly covered by the piston 16, the pressure on the underside of the piston 16 will move the piston upwards against the pressure upon the upper side, increasing the area of the port 17, reducing the restriction exerted thereby and therefore equalising the pressures acting on the two surfaces of the piston 16. Conversely if the pressure on the upper surface of the piston 16 exceeds that on its lower surface, the piston 16 will move downwards, reducing the area and increasing the restriction of the port 17 and causing the pressure below the piston to rise until it again equals the pressure above it.

While liquid is flowing through the port 17, and provided the port is not completely uncovered by the piston 16, the pressures on the two sides of the piston will be equal. If no restrictions exist between the ports 7 and 8 and the surfaces of the piston 16 the pressures in both ports 7 and 8 will be equal, and since the pressure around the waisted portion of the valve 4 will be uniform, the pressure drops due to the restrictions afforded by the lands of the valve 4 will be equal and therefore, if the coefficient of discharge is similar for the ports 7 and 8, the rates of flow of liquid through the ports will bear the same proportion to each other as the areas of the ports at any particular position of the valve.

Accordingly since the displacement of the valve 4 due to the delivery pressure acting upon the piston 5 against the spring 6 may be designed to have a particular relation to delivery pressure, and since the rate of flow of liquid through the port 7 and hence to the piston stage 10 may be designed to have a particular relation to the displacement of the valve 4, it follows that the rate of flow of liquid into the piston stage, and hence the delivery from same, may be designed to bear a particular desired relationship to the delivery pressure.

If the flow of liquid in the delivery line 20 is completely obstructed, the piston 5 will displace the valve 4 to such an extent that port 7 will be nearly completely obstructed, while the port 8 will be fully exposed enabling almost the entire flow of liquid from the gear stage to be returned to the inlet 1 via the passage 14, port 17 and passage 18. If the flow into the piston stage were completely checked, the delivery pressure would fall due to seepage past the pistons 11. The piston stage will therefore be at zero delivery and completely off-loaded when there is a certain small flow through the port 7 sufficient to maintain the loss due to seepage.

If the flow of liquid in the delivery line is checked abruptly, as for example by the rapid closure of a servomotor control valve, the flow through the port 7 will be interrupted by the displacement of valve 4 due to the pressure rise in the delivery line. A quantity of liquid would however remain in the cylinders of the piston stage 10, which would be discharged into the delivery line upon the next outward stroke of each piston, causing the delivery pressure to rise above that necessary to nearly close the port 7 and off-load the piston stage. When this pressure is exceeded by a certain amount the piston 5 will have moved downwards to an extent sufficient to open the port 21 and enable the remaining quantity of liquid in the cylinders to be discharged through the passage 19, the port 21 and thence via passages 23 and 18 to the gear stage inlet.

The form of the pressure/delivery curve of the pump will be determined by the shape of the ports 7 and 8 and of the cooperating lands of the valve 4. These will be shaped so that the pressure/delivery curve has the desired more or less continuously falling characteristic and will preferably be so shaped that the curve approximates to the curve AB in Fig. 1. A multiplicity of ports of similar or dissimilar shape may be arranged in the same or different radial planes adjacent to the lands of the valve 4 for the purpose of obtaining the desired characteristic curve.

The purpose of the cylinder 15, piston 16 and ports 17 is to maintain the pressures on the downstream sides of the ports 7 and 8 equal to each other in the absence of restriction to the flow of liquid in the passages between the ports and the surfaces of the piston 16, as explained earlier. The effect of restriction in these passages will be to cause the pressures immediately downstream of the ports to vary in relation to each other at different relative rates of flow of liquid through the ports. A degree of restriction between one or both of the ports and the surfaces of the piston 16 may be employed with advantage to modify the shape of the characteristic curve, and such restrictors having similar or different values, may be fitted when necessary, as at 29.

In the arrangement shown in Fig. 3, the cylinder 15, piston 16 and port 17 are replaced by a diaphragm 27 controlling a poppet valve 28, attached to the side of the diaphragm subjected to the pressure of the liquid flowing through the port 8. Increase in the pressure beneath the diaphragm relatively to the pressure on the upper side of the diaphragm, namely the pressure of the liquid flowing through the port 7 to the inlet valves 12 of the piston pump, will cause the valve 28 to increase its degree of opening to balance these pressures. Conversely, when the pressure on the under side of the diaphragm 27 becomes lower relative to the pressure on the other side, the valve 28 will close to re-establish balance of the pressures. The apparatus shown in Fig. 3 is otherwise similar to that shown in Fig. 2.

The apparatus shown in Fig. 4 is the same as that in Fig. 2, except that the cylinder 15, piston 16 and port 17 are omitted.

If desired, the port 7 of Fig. 4 may be replaced by a multiplicity of ports each communicating by a corresponding passage with individual piston stage inlet valves or groups of inlet valves, the ports being so disposed in relation to lands on the valve 4 that, as the valve 4 moves in response to increase in the delivery pressure, the ports are successively covered or uncovered, causing the deliveries of the corresponding pistons of the piston stage to vary in such a way that the total delivery rate varies in desired relationship with the delivery pressure.

Thus, in Fig. 5 the delivery from the gear stage is taken to two waists upon the valve 400, which has three lands, each controlling the degree of opening or closing of one of the ports 7*a*, 7*b* and 8. The port 7*a* is connected by a passage 9*a* to one group of piston pump inlet valves 12*a*, while the port 7*b* is connected to a second group of inlet valves 12*b*. The port 8 is connected via the passage 18 to the gear stage inlet 1.

At low delivery pressures the port 8 is fully closed and the ports 7*a* and 7*b* are both fully open thereby causing both groups of cylinders to be fully charged with liquid to give the maximum total delivery rate. As the delivery pressure increases beyond that at which maximum delivery is desired, the port 8 commences to open and the port 7*b* to close, causing the rate of flow of liquid to the inlet valves 12*b* to be reduced. The rate of flow through the port 7*a* meanwhile remains unchanged, the degree of restriction being designed to be insignificant until the valve 4 has moved sufficiently completely to close the port 7*b*, this occurring at a pressure approximately midway between the off-loading pressure and the delivery pressure for maximum delivery. Further increase of the delivery pressure causes a degree of closure of the port 7*a* with a corresponding further increase of the degree of uncovering of the port 8, giving a reduction in the rate of flow of liquid to the inlet valves 12*a*. When the delivery pressure reaches the maximum, the port 7*a* will be nearly closed, permitting a small amount of liquid to be pumped to compensate for leakage past the pistons, the rate of delivery being zero.

Alternatively, the inlet valves of individual piston stage cylinders or groups of cylinders may be so designed, by variation of their spring load or their valve area or both, that the degree of charging of individual cylinders or groups of cylinders will be different at similar pressures within the passage 9 of Figs. 1–4, and the passages 9*a* and 9*b* of Fig. 5.

The pump shown in Figs. 6–11 operates in substantial accordance with the diagram of Fig. 4, but the piston 5 is integral with the valve 4 as will be clear from Fig. 7, thereby rendering it unnecessary to provide a passage corresponding with the passage 24 of Fig. 4. Also, the return flow from the relief valve 22 and the port 8 does not go direct to the inlet of the gear stage 2 of the pump, but to outlets 30 (Fig. 11) which are connected to a reservoir from which the gear draws liquid through inlets 1.

The pump has a casing consisting of three parts 31, 32, 33 (Fig. 6) joined together by bolts 34. The pump shaft 35 is mounted in bearings 36, 37 and 38 and has a splined portion 39 external to the pump casing, by which it is driven. On the shaft 35 is fixed the driving gear wheel 40 of the gear stage, which meshes with a companion gear wheel 41 mounted on a pin 42, and an eccentric 43 is supported on the shaft 35 by a bearing 44 (Fig. 10). The eccentric 43 actuates the pistons 11 of the piston stage 10.

Oil from the inlets 1 passes through ducts 45 (Figs. 10 and 9) to a cavity 46 (Fig. 8) in the casing member 31, whence it is pumped by the gear wheels 40, 41 to a passage 47 communicating, via a hole 48 (see also Fig. 7) with the waisted portion 49 of the off-loading valve 4. Surplus oil passes from the passage 47 through the relief valve 22 (Fig. 8) to a passage 23 leading to an arcuate chamber 50 in the casing member 31 which is open, as shown in Fig. 8, to the chamber 51 (Fig. 7) containing the loading springs 6 for the off-loading valve 4. Liquid passing through the ports 8, when the off-loading valve 4 is open, liquid leaking past the off-loading valve, and liquid passed by the relief valve 22, all flow to the chamber 50 and thence, through passages 18 (Figs. 9, 10 and 11) to the outlets 30.

Liquid, at inlet pressure, flows from the chamber 46 (Fig. 8) through a passage 26 (Fig. 9) to the eccentric chamber 62 (Fig. 6) and through a passage 52 (Fig. 8) to the bearing 36 (Fig. 6).

Liquid, at gear stage pressure, flows through the ports 7 (Fig. 7) to a passage 9 (Fig. 8) which communicates with a distributing gallery 53 (Fig. 9) which, in turn, communicates with the inlet valves 12 of all the cylinders of the piston stage. Liquid at delivery pressure passes from each outlet valve 13 through a passage 54 (Figs. 6 and 11) to the outlet 20. Liquid at delivery pressure also passes, through a passage 55 (Fig. 11) communicating with the outlet 20, to a passage 19 (Figs. 11, 10, 9 and 8) leading to an annular groove 61 (Fig. 7) and thence, through ports 56 in a sleeve 57 surrounding the off-loading valve 4, to the undersurface of the off-loading valve. Accordingly, as the delivery pressure rises the off-loading valve will be lifted, against its springs 6, to gradually close the ports 7 and open the ports 8. After the pump has been completely off-loaded by complete closure of the ports 7, the off-loading valve can rise still further to bring ports 58 in the head 5 of the valve, which communicate with its hollow interior 59, into register with ports 60 communicating, via passages 21, with the spring chamber 51, so allowing the remaining liquid in the cylinders of the piston stage to be discharged to the chamber 50 (Fig. 8) and thence to the outlets 30 (Fig. 11).

What I claim as my invention and desire to secure by Letters Patent is:

1. A two stage hydraulic pump, comprising a first pumping unit, a second pumping unit, a delivery conduit for receiving the discharge from the first pumping unit, a supply conduit to the second pumping unit, a pressure relief conduit, a piston-type off-loading valve for controlling the flow of liquid from said delivery conduit to said supply conduit and to said relief conduit, said valve having lands controlling respectively admission ports to said supply conduit and to said relief conduit, an outlet to receive the discharge from the second pumping unit, a conduit communicating with said outlet for applying to one end of said valve the delivery pressure in said outlet and a spring for balancing said valve against said delivery pressure, said valve being movable against said spring in response to increase in said delivery pressure to reduce progressively the effective area of the admission port to the supply conduit and to increase progressively the effective area of the admission port to the relief conduit and the relative configuration of said lands and ports being such that, over a given range of delivery pressure, the delivery of the pump decreases progressively and without sudden changes in response to increase in the delivery pressure.

2. A two stage hydraulic pump, comprising a first pumping unit, a second pumping unit, said second pumping unit comprising a number of radially arranged cylinders, pistons in said cylinders, inlet and outlet valves associated with said cylinders, and a cam for imparting delivery strokes to said pistons, a delivery conduit for receiving the discharge from the first pumping unit, a valve chamber communicating with said delivery conduit, a supply conduit communicating with said chamber for supplying liquid to the inlet valves of the second pumping unit, a pressure relief conduit also communicating with said chamber, a piston-type off-loading valve mounted to slide in said chamber for controlling the flow of liquid from said delivery conduit to said supply conduit and to said relief conduit, said valve having lands controlling respectively admission ports from said chamber to said supply conduit and to said relief conduit, an outlet to receive the discharge from the second pumping unit, a conduit communicating with said outlet for applying to one end of said valve the delivery pressure in said outlet and a spring for balancing said valve against said delivery pressure, said valve being movable against said spring in response to increase in said delivery pressure to reduce progressively the effective area of the admission port to the supply conduit and to increase progressively the effective area of the admission port to the relief conduit and the relative configuration of said lands and ports being such that, over a given range of delivery pressure, the delivery of the pump decreases progressively and without sudden changes in response to increase in the delivery pressure.

3. A two stage hydraulic pump, comprising a first pumping unit, a second pumping unit, said second pumping unit comprising a number of cylinders, pistons in said cylinders, inlet and outlet valves associated with said cylinders, and means coacting with said pistons for imparting delivery strokes to said pistons in cyclical order, a delivery conduit for receiving the discharge from the first pumping unit, a plurality of supply conduits for respectively conveying liquid to the inlet valves of different groups of cylinders of the second pumping unit, a pressure relief conduit, a piston-type off-loading valve for controlling the flow of liquid from said delivery conduit to said supply conduits and to said relief conduit, said valves having lands controlling respectively admission ports to said supply conduits and to said relief conduit, an outlet to receive the discharge from the second pump-

ing unit, a conduit communicating with said outlet for applying to one end of said valve the delivery pressure in said outlet and a spring for balancing said valve against said delivery pressure, said valve being movable against said spring in response to increase in said delivery pressure to close in succession the admission ports leading to said supply conduits and simultaneously to increase progressively the effective area of the admission port to the relief conduit and the relative configuration of said lands and ports being such that, over a given range of delivery pressure, the delivery of the pump decreases progressively and without sudden changes in response to increase in the delivery pressure.

4. A pump as claimed in claim 1, comprising a connecting conduit connecting the supply conduit and the relief conduit and a pressure regulating piston in said connecting conduit, said piston being subject at one end to the pressure in said supply conduit and at the other end to the pressure in said relief conduit.

5. A pump as claimed in claim 4, comprising a restriction in at least one of said conduits between the admission port to said conduit and the connecting conduit.

6. A pump as claimed in claim 1, comprising a pressure regulating valve in the relief conduit for controlling the flow of liquid through the relief conduit, and a pressure sensitive member coupled to and serving to control the position of said pressure regulating valve, said pressure sensitive member being subject at one side to the pressure in the supply conduit and on its other side to the pressure in the relief conduit.

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References Cited in the file of this patent

UNITED STATES PATENTS

Number	Name	Date
1,770,297	Bussmann	July 8, 1930
2,295,833	Deschamps	Sept. 15, 1942
2,482,956	Wirth	Sept. 27, 1949

FOREIGN PATENTS

Number	Country	Date
582,182	Great Britain	Nov. 7, 1946