March 22, 1960 J. PANHARD 2,929,334

VARIABLE-OUTPUT HYDRAULIC GENERATOR

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Fig. 1.

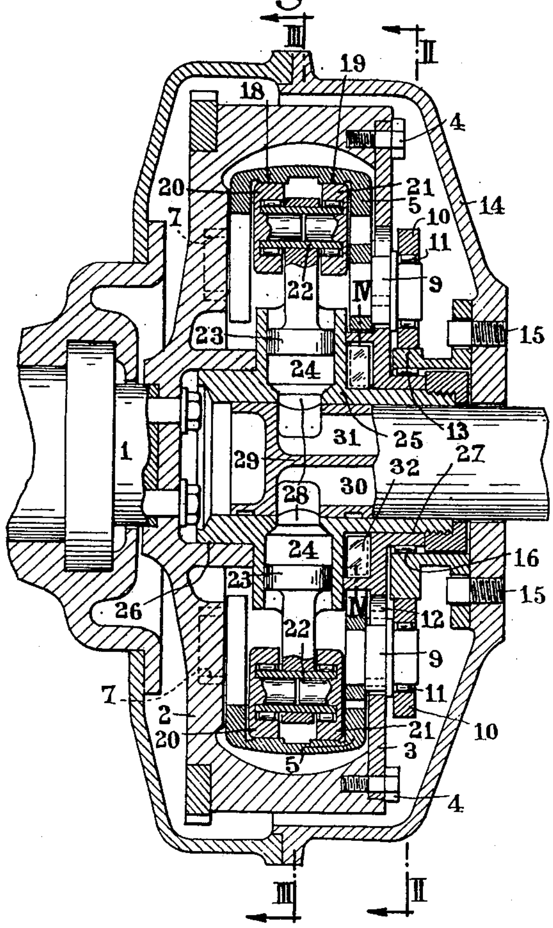

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Fig. 2.

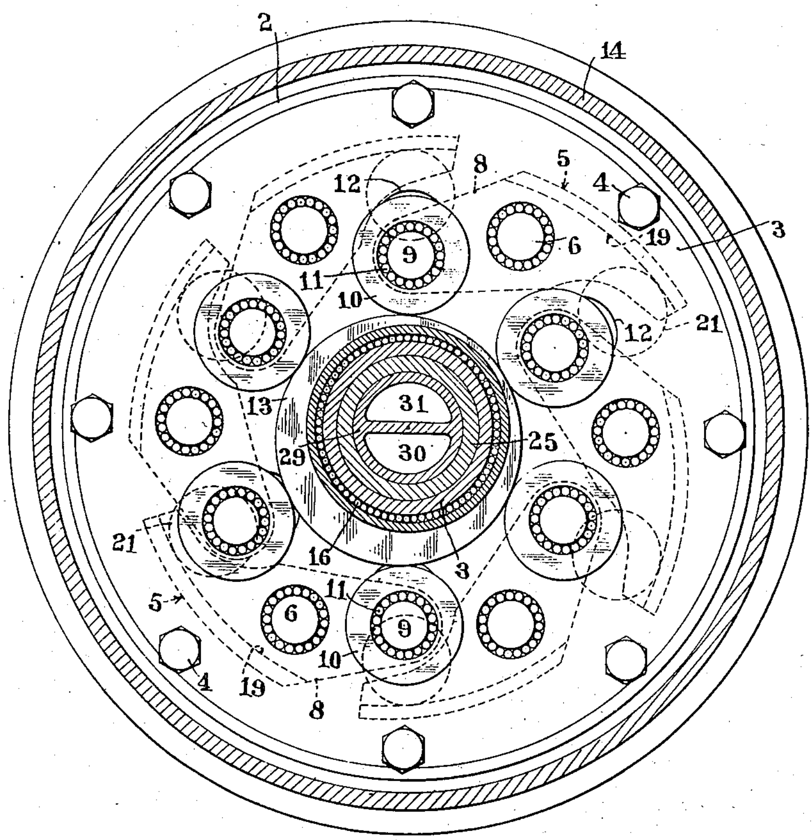

Fig. 4a.

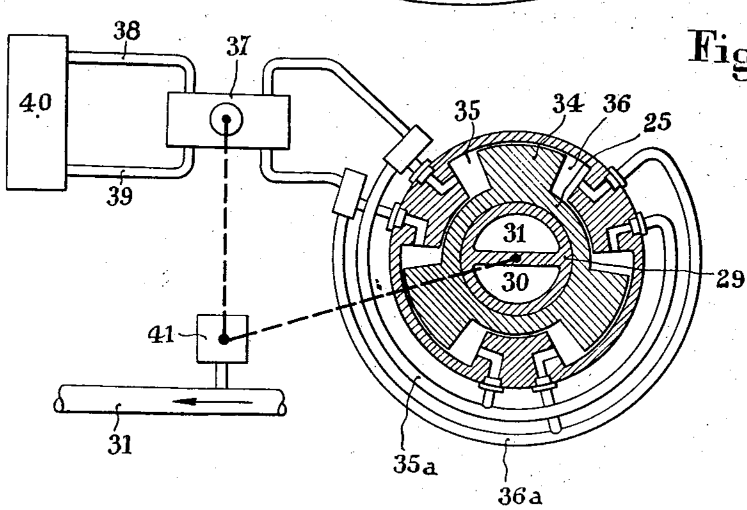

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Fig. 3.

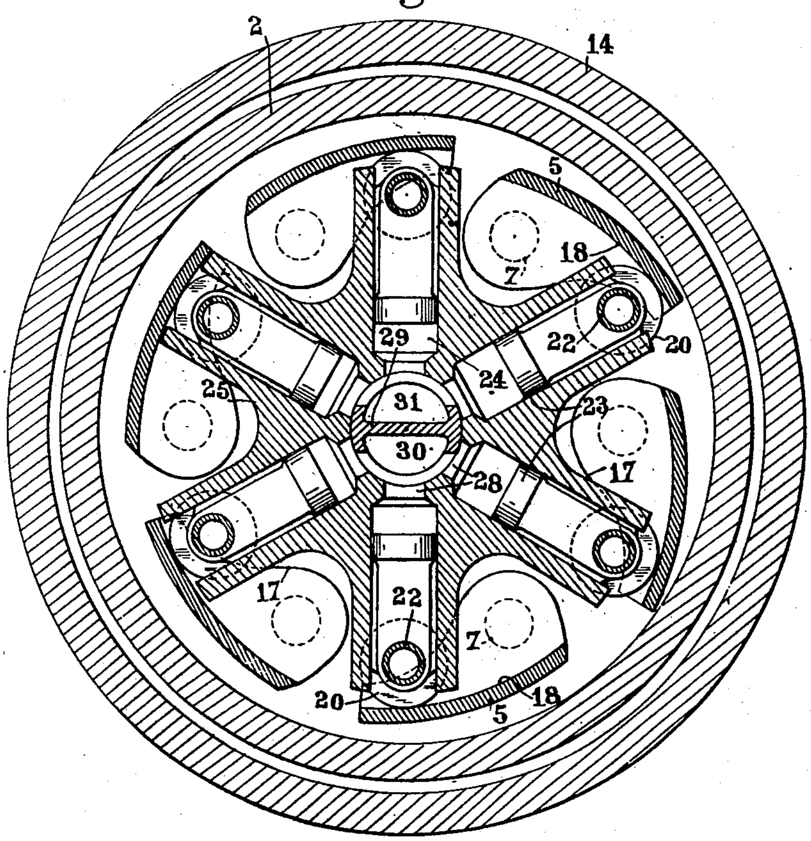


Fig. 4.

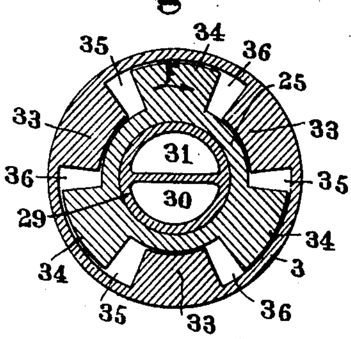
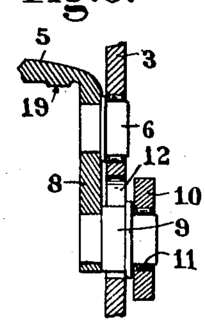


Fig. 5.



United States Patent Office

2,929,334

Patented Mar. 22, 1960

2,929,334

VARIABLE-OUTPUT HYDRAULIC GENERATOR

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Application June 10, 1957, Serial No. 664,743

Claims priority, application France June 11, 1956

5 Claims. (Cl. 103—161)

The invention deals with a generator or pump with variable capacity, more specifically, intended to control hydraulic receptor units directly located in the wheels of an automobile vehicle.

There are known in the art pumps with variable capacity containing a star-arranged group of radial cylinders the pistons of which are controlled by a cam set in motion by a relative movement of rotation (rotary motion) with respect to the afore-mentioned group of cylinders.

In a first group of familiar pumps of that type, the cam is a peripheral fixed cam, with variable eccentricity, surrounding a rotatory group of cylinders arranged in a star-formation, the chambers of the cylinders being located on the side of the axis of rotation of the group of cylinders, in which case the yield, acceptable under normal operational conditions (of the order of 85 percent), becomes unacceptable when the delivery pressure reaches a high value, at the time of starting, for instance. At such a time, the delivery falls to values that may reach as low as 40 percent as a result of the amount of the friction reached.

In a second group of known pumps of the type described above, the group of cylinders arranged in a star pattern is fixed, the chambers of the cylinders are located towards their end furthest away from the axis of the pump, and a central cam that rotates about that axis sets in motion the pistons by means of oscillating mounted levers around the movable or shiftable axes with relation to a group of cylinders, which permits the course of the pistons to be varied. The levers in question are such that for each one of them the point where the cam acts on the lever and the point where the lever acts on the piston are located on the same side of the axis of oscillation of the lever under consideration. However, these pumps are heavy and bulky, the pull-back of their pistons in a centripedal direction requires springs, and zero capacity or delivery can be obtained only by such a set-up of the levers interposed between the cam and the pistons, which will permit the lateral action of the aforesaid pistons in their respective cylinders to always operate in the same direction during the course of delivery, which causes excessive wear and tear.

The present invention has to do with pumps with variable capacity of the type referred to above, and one which consists of a rotative group, arranged in star-shape, of radially placed cylinders the chambers of which are located on the side of the axis of rotation and the pistons of which are controlled by a fixed cam acting on the outer ends of the afore-mentioned pistons (those that are turned on the side opposite to the aforesaid axis of rotation); and the pump, in line with the invention, is characterized by the fact that the fixed cam is a central cam which, in order to control each piston, acts on one of the two arms of a trip-lever pivoting about an axis held in place on a frame participating in the rotatory movement of a group of cylinders, the second arm of the trip-lever acting on the outer end of the piston under consideration, with means being provided for the regulating of the angular relation between the aforesaid turning frame and the aforesaid group of cylinders, in other words, to regulate the effective or active length of the aforesaid second arm, in order to modify the course of the piston, i.e., the delivery per turn of the pump.

In the drawings:

Figure 1 is a longitudinal axial section of the device.

Figure 2 is a cross-section taken upon the line II—II of Fig. 1.

Figure 3 is a cross-section taken upon the line III—III of Fig. 1.

Figure 4 is a cross-section taken upon the line IV—IV of Fig. 1 which shows more in detail the device controlling the variation in the piston stroke.

Figure 4*a* is a diagram illustrating the operation of the device shown in Fig. 4; and

Figure 5 is a detailed sectional view illustrating specific features of the rocker construction.

The hydraulic generator illustrated in the drawings comprises six radially disposed cylinders, but it will be readily understood that this specific arrangement is not compulsory since the number and relative arrangement of the cylinders may differ from those illustrated without departing from the principles of the invention as set forth hereinabove and in the appended claims.

In Fig. 1 the driving shaft **1** rotates the flywheel **2** mounted in the conventional manner. A plate **3** is bolted at **4** on this flywheel and forms therewith a case revolving with the aforesaid shaft **1**. A set of rockers **5** (six in the example illustrated and described herein) are spaced concentrically about the axis of rotation of the flywheel and pivoted on the latter, on the one hand, and on the plate **3**, on the other hand, by means of pivot pins **6**, **7** fitted in adequate bearings received in corresponding recesses. One arm **8** of each rocker **5** carries at its end a journal member **9** (see Fig. 5) on which a roller **10** is mounted, adequate bearing means **11** being interposed therebetween. This journal member **9** is displaceable in an elongated notch **12** formed in the plate **3**. The roller **10** engages a cam member **13** having a suitable contour which is secured by screws **15** on the outer case **14** of the generator. This cam member **13** is eccentric relative to the axis of rotation of the rockers **5** and carried by the plate **3** through the intermediary of a bearing **16**. With this arrangement it is possible to avoid the only external reaction to a torque by avoiding the radial stress which would act in overhanging relationship on the flywheel and crankshaft assembly. The other arm **17** of these trip-levers bears a longitudinal bearing **18—19** for two wheels **20** and **21** trunnioned on the axis **22** of a piston **23**, this bearing-path taking the form of an arc turning its concave surface towards the axis of rotation of the group of cylinders. The radius of the rollers is equal to the distance separating the pivoting axis of each trip-lever and the zone of the path of the bearing in an arc-like direction, which is in contact with the roller under consideration when the axis of the aforementioned roller is in the radial field or plane passing the axis of the corresponding trip-lever. These pistons **23** are slidably fitted in cylinders **24** bored in a spider-forming hub **25**. This hub **25** is in direct contact with the flywheel **2** by being centered by the surface **26** thereof, and also with the plate **3** by engaging its surface **27**. Thus, the hub **25** may be shifted angularly through a certain angle relative to the movable assembly comprising the flywheel **2**, plate **3** and rocker **5**. The cylinders **24** communicate through ducts **28** with a common cylindrical chamber divided into two compartments **30**, **31** by a stationary distributor **29**. The compartment **30** is used for drawing in the hydraulic fluid, and the other compartment **31** delivers the fluid to the members or apparatus to be actuated thereby. In the circular cylindrical space **32** formed between the spider-forming hub **25** and the plate **3** a number of segments are disposed (six in the case contemplated herein). One half of the segments (see Fig. 4) is solid with the plate **3** and designated by the reference numeral **33**, the other half being solid with the hub **25** and designated by the reference numeral **34**. The fluid-filled gaps **35** and **36** constitute a hydraulic coupling between the hub **25** and the movable plate **3**. Means are provided for example in the form of a reversing valve **37** (see Fig. 4*a*) for connecting the gaps **35** (connected to a common line **35*a***) with a high-pressure line **38**, whereas the gaps **36** (connected to a common line **36*a***) communicate at the same time with a low-pressure line **39**, both lines **38** and **39** being connected to a separate source of hydraulic power **40**. As a result, a rotation of the spider-forming hub relative to the flywheel and plate assembly in the direction of the arrow F is obtained. If the reverse conditions prevail, that is, if the gaps are connected to the lines **39** and **38** respectively, the hub is shifted angularly in the opposite direction.

The operation of the hydraulic generator according to this invention is as follows:

Under normal operating conditions, which correspond substantially to the piston positions illustrated in Figs. 2 and 3, the spider-forming hub is so set relative to the movable rocker assembly that the rollers **20, 21** carried by the pins **22** of pistons **23** contact the extremities of cam faces **18** and **19**. Due to the rotational movement accomplished by the rocker assembly **5**, the rollers **10** engaging the stationary cam member **13** impart a slight oscillating motion to the rockers **5** about their pivot pins **6**. This motion is transmitted through the arms **17** and rollers **20, 21** to the piston pins **22** so that a reciprocating movement is imparted to these pistons, the maximum amplitude of this movement being maximum in the case illustrated in Figs. 2 and 3. Therefore, in case of a relatively low delivery pressure the pistons **23** will suck in the maximum quantity of fluid when they communicate with the induction line **30** to force it into the line **31**.

If the reaction torque, that is, the pressure obtaining in the delivery line **31**, increases, a servo-control system **41** (see Fig. 4*a*) operates the reversing valve **37** and brings about an angular shift between the spider-forming hub **25** and the rocker assembly **5** by moving the segments **33, 34** (Fig. 4). As a result, the rollers **20, 21** move along the cam faces **18, 19** and the lever arm is reduced accordingly as well as the amplitude of oscillation of the pistons **23**. This amplitude may even be cancelled completely when the axis of the piston-pin **22** and the axis of the pivot pins **6, 7** of rockers **5** are coincident.

The maximum angular shift between the hub **25** and the series of pivoted rockers **5** is about 30° in the example illustrated, but it may obviously be smaller or greater than this value. It will be noted that the cam faces **18, 19** constitute as many segments of a circle having its centre coincident with the axis of rotation of the flywheel **2** when the rocker **5** is in its intermediate position. As a result, the angle formed between the normal to the cam faces **18, 19** and the force exerted by these cam faces on the rollers **20, 21** is zero when the piston **23** is in its intermediate position. Consequently, any side reactions are reduced accordingly.

The contact between the rollers **10** and cam member **13**, on the one hand, and between the rollers **20, 21** and cam faces **18, 19**, on the other hand, is ensured by the centrifugal force alone. The weight of the rockers and pistons, the distribution of this weight and the cam lift are calculated accordingly.

On the other hand, by adequately setting the cam member **13** the circulation of the fluid by gravity is facilitated by positioning the suction compartment **30** beneath the delivery compartment **31**.

It is advantageous to select angular values approximating those given hereinafter:

Dead centre / Beginning of piston stroke	7° 30′
Acceleration period (conjugated with the deceleration period of another piston located 120° ahead of the former)	45°
Piston moving at constant speed	75°
Deceleration period (conjugated with the acceleration period of another piston located 120° behind the former)	45°
Idle period between the end of the piston movement and the dead centre	7° 30′
Total	180°

As the same sequence takes place during the next 180° but to effect the fluid suction instead of the fluid delivery, it will be seen that with this cam arrangement providing one lift per revolution a three-cylinder disposition is perfectly suitable for a strictly regular output without pulsations.

In the arrangement illustrated by way of example the other three cylinders permit of doubling the power output for same over-all dimensions while reducing the unitary stress exerted on the parts, on the one hand, and on the bearings, on the other hand.

Consequently, there are constantly at least two cylinders in their fluid-delivering phase, this number increasing to three during the junction periods, but the output is always uniform and without pulsation.

The servo-control system **41** (Fig. 4*a*) may also be used for driving the rotary distributor **29** which should generally have the same angular shift as the spider-forming hub if the fluid-distributing ports **28** do not ensure the proper distribution (as it generally occurs) with a 30-degree shift of the spider forming hub **25**.

It will also be noted that though the invention is described herein as being applied to a hydraulic generator having a constant output per revolution, it is also applicable to a generator having a variable output per revolution. This advantageous feature results from the fact that the variation in the output obtains without changing the control device, as the cam movement is constantly the same and the variation in the piston stroke occurs subsequently by using a proportional system independent of the control hub.

It would not constitute a departure from the principle of this invention if the number of cylinders of the generator were changed, or if a cam member having a double symmetry giving two piston strokes per revolution were used, however these modifications may be contemplated only in those cases where the velocity of rotation is consistent therewith.

What I claim is:

1. A variable output pump having a driving shaft and comprising a group of cylinders radially mounted to rotate with said shaft about a common axis of rotation, a piston slidably mounted on each said cylinder, said piston having a roller mounted on its outer end, a stationary annular cam mounted with its center in eccentric relationship with the axis of rotation of said shaft and cylinders, a frame structure mounted on said shaft and around said cylinder group to rotate with said shaft and said cylinder group, a rocker for each of said pistons, each said rocker pivoted on said frame structure adjacent each said piston, and each said rocker having two rocker arms oppositely disposed about said pivot, a roller mounted on the end of one of said rocker arms of each said rocker for engaging said stationary annular cam to oscillate said rocker about its pivot; a longitudinal bearing path comprising an inner surface of the periphery of said other rocker arm, said bearing path for engaging the roller mounted on the outer end of said piston to oscillate said piston in said cylinder, means for

successively connecting the interior axial end of each of said cylinders as they rotate to a hydraulic induction line as their associated pistons move outward in oscillation and to a hydraulic delivery line as said pistons move inward in oscillation, and means for angularly adjusting said frame structure with respect to said cylinders and pistons and thus for moving the engagement of said piston rollers with said longitudinal bearing paths toward or away from their respective said rocker pivots thereby providing for adjusting the effective length of said other rocker arm and extent of the oscillation of said pistons in said cylinders and therefore of the pump output.

2. A pump as described in claim 1 characterized in that said piston rollers are held in engagement with the longitudinal bearing paths, and the said rocker arm rollers are held in engagement with the stationary annular cam by centrifugal force generated in said movable pistons and rocker arms by their rotation with said drive shaft.

3. A variable output pump as set forth in claim 1, wherein each said bearing path has substantially the

shape of an arc having its concave side facing toward the common axis of rotation of said group of cylinders.

4. A variable output pump as set forth in claim 3, wherein each said bearing path is substantially the arc of a circle which is concentric with said common axis of rotation when said rocker is at a midpoint of its oscillation.

5. A variable output pump as set forth in claim 3, wherein the radius of each of said piston rollers is substantially equal to the shortest distance between the axis of said rocker pivot and said bearing path.

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