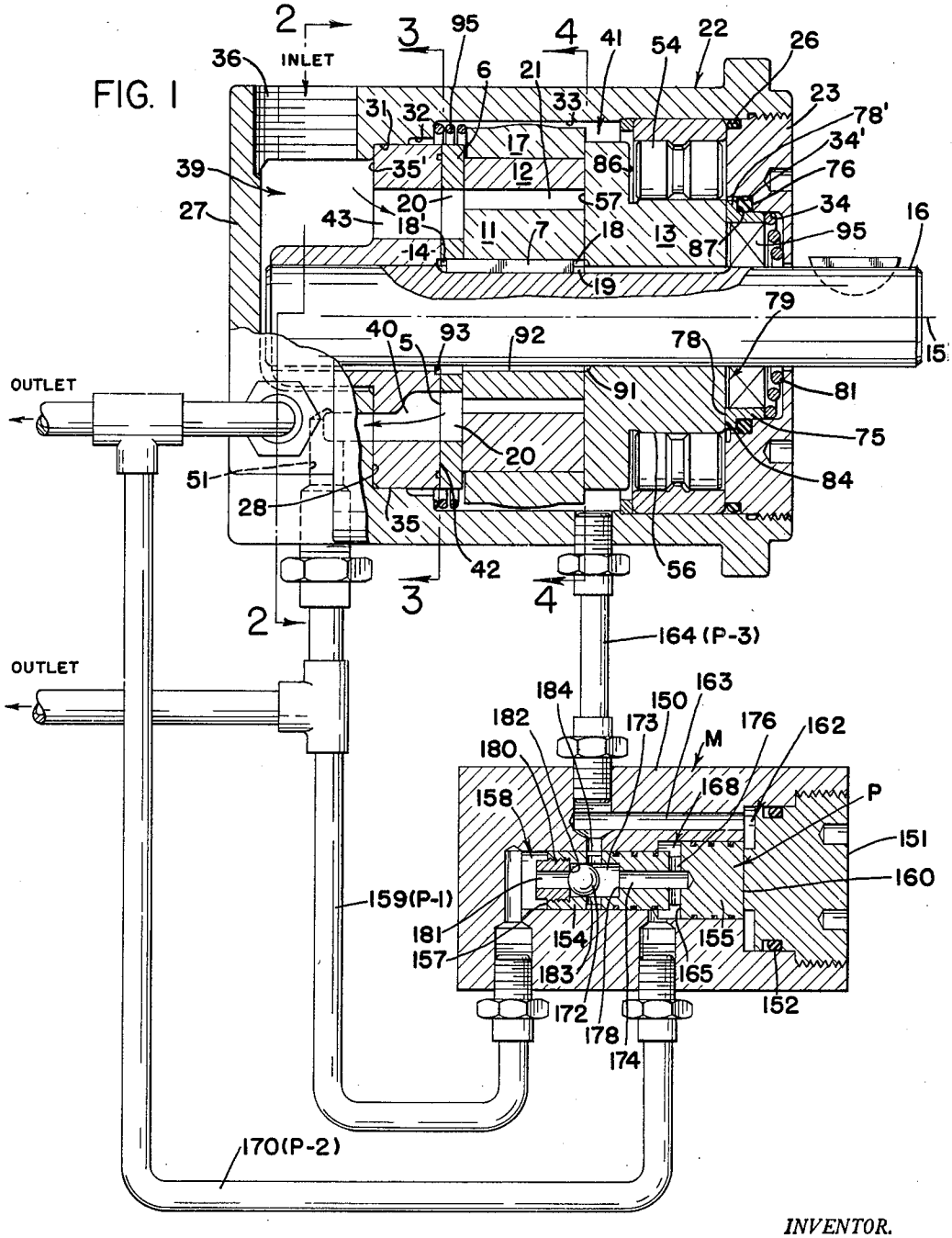
April 14, 1964 R. W. BRUNDAGE 3,128,707

MULTIPLE DISCHARGE HYDRAULIC PUMP

Filed March 11, 1960 3 Sheets-Sheet 1

FIG. 1



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MULTIPLE DISCHARGE HYDRAULIC PUMP

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FIG. 2

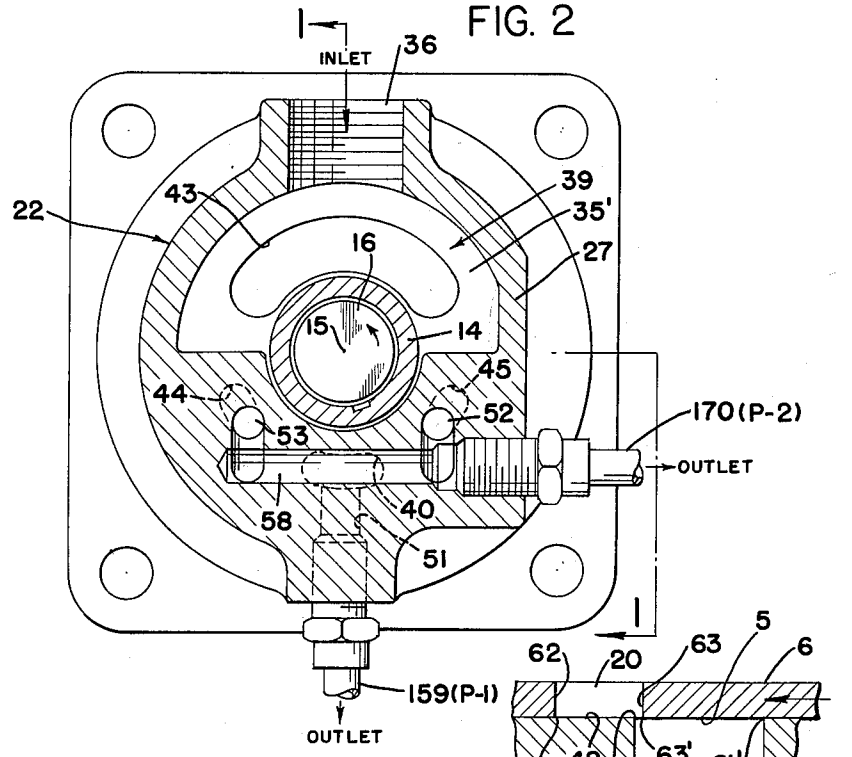
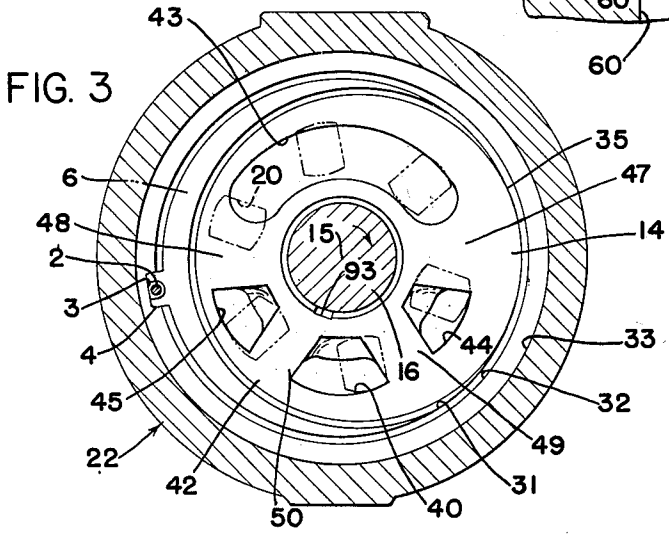


FIG. 3a

FIG. 3



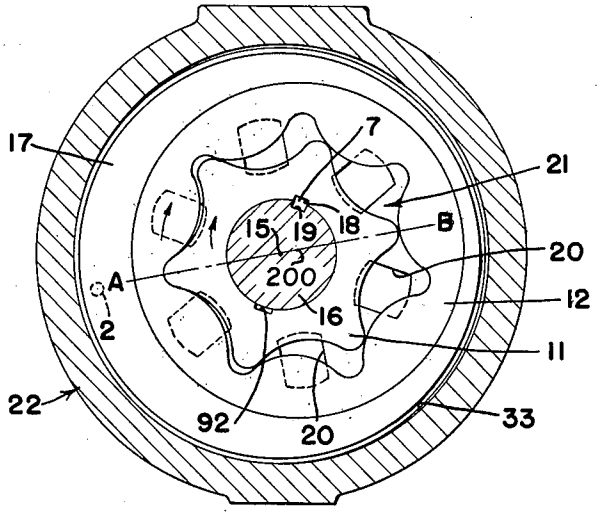
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MULTIPLE DISCHARGE HYDRAULIC PUMP

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FIG. 4



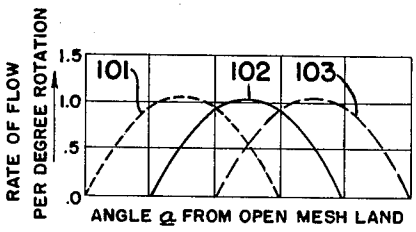


FIG. 5

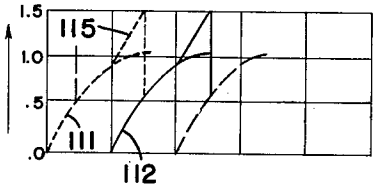


FIG. 6

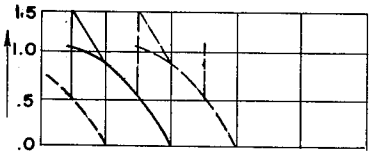


FIG. 7

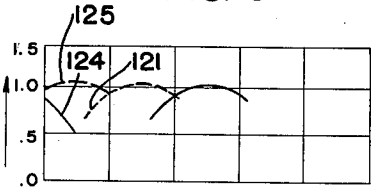


FIG. 8

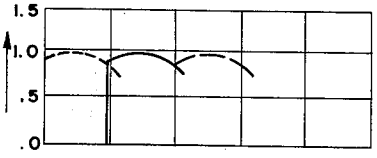


FIG. 9

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3,128,707

MULTIPLE DISCHARGE HYDRAULIC PUMP

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Filed Mar. 11, 1960, Ser. No. 14,258
11 Claims. (Cl. 103—2)

This invention pertains to the art of hydraulic pumps of the positive displacement type and more particularly to a hydraulic pump capable of simultaneously delivering hydraulic fluid at two different output pressures and volumes.

The invention is particularly applicable to what is generally known as an internal gear type pump and will be described with particular reference thereto, although it will be appreciated that the invention may be equally applied to vane or rotating cylinder type hydraulic pumps or the like.

Furthermore, the present invention is particularly applicable to hydraulic pumps operable at what may be termed very high hydraulic pressures; that is to say, above 1,000 pounds per square inch and oftentimes approaching or exceeding 4,000 pounds per square inch. At such pressures, extremely high internal forces are developed and constructions and expedients usable at the lower pressures, are often unsatisfactorily and inapplicable to the problems where the higher pressures are encountered.

Internal gear type hydraulic pumps are normally comprised of an internally toothed and an externally toothed gear member rotatable on spaced axes in a housing with the teeth of the gears in sliding sealing engagement. These teeth define axially open ended chambers which revolve and progressively increase in volume to a point of maximum volume which corresponds to the point of open mesh of the gears during the intake stage of the pump and then, during the discharge stage of the pump, the volume of each of the chambers progressively decreases to a point of minimum volume which corresponds to the point of closed mesh of the gears. Each chamber has an opening leading therefrom which at open mesh is normally momentarily closed by a land and then moves into communication with a discharge port and continues in communication with such discharge port until it reaches closed mesh when it is again momentarily closed by a land and then moves into communication with an inlet port.

Where it is desired to have a hydraulic pump simultaneously deliver hydraulic fluid at two different pressures, it has been conventional in the past to provide a single auxiliary land which sealingly separates the discharge port into first and second discharge ports (counting from the open mesh land in the direction of rotation). Each chamber thus discharges a portion of its fluid into each of the discharge ports in proportionate amounts depending upon the circumferential or line of movement width of the individual ports. Each of these discharge ports is then connected to separate outlet ports on the pump. Such an arrangement presents problems with which the present invention deals.

The principal problem is the large and almost instantaneous change in the rate of flow of fluid when the communication of each chamber is either cut off from the first port or opened to the second port. Thus, as is known, in an internal gear type pump, the rate of fluid flow from each decreasing volume chamber per degree of rotation varies by the sine of the angle of the chamber from the open mesh land. That is to say, the rate of fluid flow from each chamber varies sinusoidally from zero as its opening moves from the open mesh land to a maximum when the chamber is midway between the open and closed mesh lands. Thereafter, the rate of flow decreases sinusoidally to zero when the opening from the chamber is again closed by the closed mesh land.

If the auxiliary land dividing the discharge port into first and second ports is, as heretofore, located 90° from the open and closed mesh lands, then it will be seen that the rate of flow into the first port will decrease almost instantaneously from maximum to zero as the opening is closed by the auxiliary land. Immediately thereafter the rate of flow into the second discharge port will increase almost instantaneously from zero to maximum. These instantaneous changes in the rate of flow of an incompressible fluid into the two discharge ports produced hammering on the members of the pump as well as undesirable pulsations in the rate of fluid being discharged from the pump through each discharge port.

Thus it is to be noted that with a pump having two discharge ports with equal line of movement widths, the instantaneous change in the rate of flow of fluid to the point of use is one times the maximum rate of flow from each chamber.

The present invention deals with this problem by providing first and second auxiliary lands dividing the discharge port into first, second and third discharge ports (counting from the open mesh land in the direction of rotation) and by intercommunicating the first and third discharge ports so that the fluid discharged thereinto all flows to the same pump outlet port. Thus, in effect, three discharge ports provide two simultaneous pump discharges which may be either at the same or different pressures and depending upon a relative line of movement width of the ports at different volumes.

The improvement is readily apparent. For example, with a pump having six chambers and with the first, second and third ports having equal line of movement width, the rate of flow into the first port will vary from zero to .866 of the maximum rate of flow when the flow is instantaneously cut off. Simultaneously however, the rate of flow of fluid into the third port increases instantaneously to .866 and then varies sinusoidally to zero. These two discharges, however, are added together in the outlet and the maximum resultant variation in the rate of flow of fluid to the point of use is from .866 to 1 times the rate of flow of fluid from any one chamber. In a like manner, the rate of flow of fluid from each chamber into the second discharge port increases instantaneously from zero to .866, then increases sinusoidally to 1 and decreases sinusoidally to .866 when the flow is again instantatneously cut off. However, simultaneously with the instantaneous increase or the instantaneous decrease, another chamber comes into communication with the second discharge port and the result is that the rate of flow of fluid from all the chambers into the second discharge port varies from .866 to one times the maximum rate of flow from any one chamber.

A further advantage of employing the three discharge ports with the outer two interconnected is that the forces on the gear teeth created by the high pressure fluid in the chambers are the same as though the pump had but a single discharge port. These forces are readily handled and can, in fact, be employed to create a desired closing force on the gear teeth at open mesh sufficient to prevent leakage, but not large enough to cause teeth wear, all as is explained in my co-pending application Serial No. 814,320, now Patent No. 3,034,448, filed May 19, 1959.

This may be distinguished from a single auxiliary land as heretofore. Thus, if the pressure in the first discharge port is greater than that in the second discharge port, forces are created tending to urge the gear teeth apart at open mesh. Excessive leakage results. Alternatively, if the second discharge port has the greater pressure, forces are created tending to bias the teeth of the gear

at open mesh into pressure engagement. High gear tooth wear and a noisy operation results.

It is to be noted that the principle of the invention may be applied where three or more independent pump discharges are required. Thus, an odd number of discharge ports must be employed with the first and last, the second and next to last, etc. being intercommunicated and connected to separate outlet ports.

In internal gear type pumps, it has been conventional in the past to provide pressure sealing of the axial ends of the gears against the sealing face of the manifold member. This has been done by providing a sealing member in sealing engagement with the face of the gear members opposite from the manifold member, and exposing the axial end or portions of the end of this sealing member remote from the gears to the discharge pressure of the pump so that a force is created urging the sealing member and manifold member into sealing engagement with the axial end of the gears proportional to the forces of the hydraulic pressures in the decreasing volume chambers tending to separate these members from the faces of the gears.

However, with a multiple pressure discharge pump, it has heretofore been difficult, if not impossible, to obtain the desired sealing force. If one pressure is employed for the purpose of providing the sealing force, and the other discharge pressure goes to zero, then an excessive sealing force will result. On the other hand, if the sealing force is adjusted to provide for one of the discharge ports being at zero or intermediate pressure and this discharge port has an increase in pressure, then the sealing force is insufficient.

The present invention deals with this problem by automatically integrating the two discharge pressures to determine the necessary sealing pressure and deriving and maintaining such sealing pressure from one or the other of the discharge pressures whereby a sealing force is created which is always proportional to the two discharge pressures.

Thus, in accordance with invention, a piston has a plurality of discharge pressure surfaces of predetermined area facing in one direction and each exposed to one of the discharge pressures and a sealing pressure surface facing in the other direction and exposed to the sealing pressure, the piston being movable to close valve means communicating the higher of the two discharge pressures with the sealing pressure when the sealing pressure is of the desired magnitude.

A further problem with pumps of the type to which this invention pertains, namely where the manifold parts have a short line of movement width, is to maintain the opening between the passages from the chambers and the discharge ports with he maximum possible area for as long as possible as the passages move relative to the ports and lands.

The invention deals with this problem by providing a ported plate between the axial end of the gear members and the manifold member having axially extending passages one for each pumping chamber to communicate the pumping chambers with the inlet and discharge ports. The circumferentially facing walls of the passages and of the lands are so arranged that as each passage moves from a land into communication with a port or from communication with a port to be closed by a land, the opening is defined by elongated parallel side walls.

The result is that for a given arc of movement of a passage relative to a port the maximum possible area of opening between the two is presented for the maximum length of time. Also, the parallel walls as they approach or depart each other present an elongated sharp edge orifice to the flow of fluid. Fluid flow through such an orifice is independent of fluid viscosity, making the pump perform better and uniformly even though the viscosity of the fluid varies due, for example, to temperature variation.

The principal object of the present invention is the provision of a new and improved hydraulic pump of the multiple pressure discharge type, which is relatively simple in construction, inexpensive to build, and which has highly improved performance characteristics.

Another object of the invention is the provision of a new and improved hydraulic pump of the multiple discharge type wherein the rate of flow of fluid to each point of use is more uniform than heretofore.

Another object of the invention is the provision of a new and improved hydraulic pump which is capable of supplying hydraulic fluid simultaneously at different pressures with a minimum variation in the rate of flow.

Still another object of the invention is the provision of a new and improved pressure sealed hydraulic pump capable of supplying hydraulic fluid simultaneously at two different pressures wherein the sealing force is proportional to the two pressures.

Still another object of the invention is the provision of a new and improved hydraulic pump wherein a ported plate is positioned between the pumping chambers and the inlet and outlet ports wherein the side wall of the passages through the ported plate and the sides of the lands separating the inlet and outlet ports are so relatively shaped that the sides present a sharp edged orifice to the flow of fluid at the instant that the passages in the ported plate move into or out of communication with a port.

Still another object of the invention is the provision of a new and improved device for integrating the two pressure outputs on a hydraulic pump whereby sealing forces in the pump may be proportional to the two output pressures.

The invention may take physical form in certain parts and arrangements of parts, a preferred embodiment of which will be described in detail in this specification and illustrated in the accompanying drawing which is a part hereof and wherein:

FIGURE 1 is side cross sectional view of a hydraulic pump and a pressure integrator illustrating a preferred embodiment of the invention, the section of the pump being taken approximately on the line **1—1** of FIGURE 2;

FIGURE 2 is a cross sectional view of FIGURE 1 taken approximately in the line **2—2** thereof and showing the interconnection of two of the discharge ports;

FIGURE 3 is a cross sectional view of FIGURE 1 taken approximately on the line 3—3 thereof with the ported plates superimposed in phantom lines to show the relationship of the side walls of the ported plate passages to the side walls of the lands separating the inlet and discharge ports;

FIGURE 3*a* is a fragmentary cross sectional view showing the knife edge relationship of the sides of the ported plate passages and the lands at the instant of opening or closing a passage.

FIGURE 4 is cross sectional view of FIGURE 1 taken approximately on the line **4—4** thereof and showing the relationship of the openings in the ported plate to the gear members;

FIGURE 5 is a graph showing the rate of discharge of fluid from each of three chambers at each angular position of the chamber from the open mesh land.

FIGURES 6 and 7 are similar graphs showing the rate of discharge of fluid to the two outlets of pumps of the prior art.

FIGURES 8 and 9 are similar graphs but showing the rate of discharge of fluid to the two outlets of a pump constructed in accordance with the present invention.

Referring now to the drawings wherein the showings are for the purposes of illustrating a preferred embodiment of the invention only, and not for the purposes of limiting same, the figures show a hydraulic pump comprised of a housing having an internal pumping cavity in which are mounted a plurality of pumping members defining a plurality of closed chambers which progressively increase and decrease in volume as the members move relative to

each other. While such members may take a number of different conventional forms, such as rotating cylinders with axially reciprocating pistons, or rotating vanes or the like, in the embodiment of the invention shown, and preferably, they comprise generally: an externally toothed gear member 11; and internally toothed gear member 12; a ported plate 6 engaging the left hand end of the gears 11, 12; a manifold member 14 engaging the left hand axial face of the ported plate 6; and, a sealing member 13 engaging the right hand axial end of the gears 11, 12.

Pumping Members

The gear member 11 and ported plate 6 are supported for rotation on the axis 15 of a shaft 16 and are keyed thereto by a key 7 fitting into keyways 18, 18' on the gear 11 and ported plate 6 respectively and keyway 19 on the shaft 16. The plate 6 is axially movable relative to the shaft 16. The internally-toothed gear member 12 is in turn supported for rotation about an axis 200 spaced from the axis 15 in a bearing member 17 which is loosely mounted and radially and axially movable in the housing cavity for reasons as are taught in my said co-pending application Serial No. 814,320.

In this co-pending application, fluid film type lubrication was maintained between the outer surface of the ring gear 12 and the inner surface of the eccentric ring member 17 by biasing the eccentric ring member 17 axially into engagement with the sealing member 13 such that the bearing was axially closed at one end. The bias force was produced by means of the reaction force of the hydraulic fluid being discharged against one axial end of the eccentric member 17. In the embodiment of the invention shown, this reaction force is not available and in place a helical spring 95 is employed bearing at one end against the housing and at the other end against the left hand axial end of the eccentric ring members 17.

The gear member 12 has one tooth more than that of the gear member 11 and these teeth are in sliding sealing engagement so that as the gear members 11, 12, they, along with the sealing and manifold members 13, 14, define a plurality of closed pumping chambers 21 which revolve on a closed path of movement and progressively increase in volume from a point A of minimum volume to a point B of maximum volume and then decrease in volume to the point of minimum volume A. The points A and B are in what may be termed the neutral plane through the two axes of rotation and it will be further noted that the gear teeth at the point A are in what may be termed "closed mesh" and at the point B at "open mesh."

The ported plate 6 is in the form of a flat disc of a diameter at least greater than the diameter of the roots of the teeth of the gear 12 and has a plurality of axially extending passages 20 therethrough one for each chamber 21. In the preferred embodiment the inner gear member 11 has 6 teeth, and there are six passages 20, one located at the root of each tooth. Obviously, more or less teeth and passages may be provided. The right hand face of the plate 6 sealingly engages the left hand face of the gears 11, 12 and all fluid to and from the chambers 21 must pass through the passages 20.

It is to be noted that this ported plate may in some instances be dispensed with, but for use in connection with a pump having multiple discharge pressures, its use is advantageous for reasons which will appear.

Housing

The housing forms no part of the present invention, and as shown, includes a main part 22 generally in the shape of a cup, and a closure part 23 removably positioned in the open end of the cup 22 by any suitable means and sealed by means of an O-ring 26.

The main part 22 includes a base 27 having a circumferentially continuous surface 28 facing axially toward the open end of the cup. The main part 22 also includes a

plurality of radially inwardly facing cylindrical surfaces 31, 32, 33 to the right of the surface 28 and which are progressively larger in diameter from left to right. In a like manner, the closure part 23 has cylindrical surfaces 34, 34' surrounding a housing opening through which shaft 16 passes.

Manifold Member

The shaft 16 extends into and is rotatably supported in the manifold member 14 which in turn is fixedly mounted in the housing in any suitable manner, but in the embodiment shown, has an axially facing surface 35' bearing against the surface 28 and a cylindrical surface 35 fitting within the cylindrical surface 32, such surfaces all being in sealing engagement. A radially extending lug 4 engaging in a corresponding groove of the housing locates the member circumferentially in the housing.

The housing base 27, in its upper half, has a generally semi-circular cavity 39 communicating with an inlet opening 36 and sealed from the remainder of the cavity of the main part 22 by means of this sealing engagement.

The manifold member 14 has on its right hand axial end, a surface 42 in sealing engagement with the left hand axial surface 5 of the ported plate 6. An inlet manifold port 43 extending in an arcuate direction in the path of movement of the pumping chambers and of the passages 20 is formed in the sealing face 42 and extends axially through the manifold member 14 to communicate with the cavity 39. Additionally, first, second and third discharge manifold ports 44, 40, 45 are formed in the sealing surface 42 diametrically opposite from the inlet manifold port 43 which ports are in aligned relationships in an arcuate direction in the line of movement of the pumping chambers.

The circumferential ends of the various manifold ports are in circumferentially spaced relationship such that that portion of the sealing surface 42 between the arcuate ends of the ports forms the lands of the manifold member. Thus, between the ports 43, 44, there is a land 47 which may be termed the "open mesh" land. Between the opposite end of the port 43 and the adjacent end of the outlet port 45 is land 48 which may be termed the "closed mesh" land. Between ports 44, 40 is first auxiliary land 49, while between ports 40 and 45 is a second auxiliary land 50, first and second and third being used herein to indicate the position of the auxiliary lands or ports as the case may be from the open mesh land in the direction of rotation.

The second discharge port 40 extends axially through the manifold member 14 and communicates with a discharge passage 51 formed in the base portion 27 of the housing. This passage 55 as shown in the drawing extends horizontally from the discharge port 40 and then turns at a right angle and extends downwardly to the exterior of the housing.

In a like manner, the first and third discharge ports 44, 45 extend axially through the manifold member 14 and each communicate respectively with passages 52, 53 formed in the base 27 of the housing which passages 52, 53 are intercommunicated by means of a passage 58 extending through the housing and communicating with the exterior thereof. It will be appreciated that the first and third discharge ports 44, 45 can be intercommunicated in any other desired manner, the one shown being simple. It is to be noted that it is important in accordance with the invention that the first and third discharge ports be interconnected so that the fluid flowing therefrom all goes to the same point of use.

The circumferential or line of movement width of the three discharge ports is so adjusted at the time of manufacture as to provide the desired proportion of total discharge volume of the pump in either the first and third ports combined or in the second port, although for reasons which will appear, the second port should have a width at least greater than the width of the lands between the

passages 20 so that one passage is always in communication therewith. The second discharge port 40 in the embodiment shown is symmetrical about a line perpendicular to the diametrical line through the land 47, 48 and the first and third discharge ports 44, 45 are each of the same line of movement width and located symmetrically relative to the port 40. In this respect, it is to be noted that all of the lands have a line of movement width just slightly greater than the line of movement width of the passages 20.

In this respect it is to be noted that the ported plate 6 is a necessity in a multiple discharge pump. Thus the plate with its passages 20 serves to narrow the effective line of movement or circumferential width of the chambers 21 so that the line of movement width of the lands may be correspondingly reduced from that required without a ported plate. Absent the ported plate, the chambers 21 and a 6-tooth gear would have a line of movement width of approximately 60°. It would be impossible to get two auxiliary lands plus one half of two neutral axis lands in the manifold plate and still have discharge ports.

The diametrical line through the lands 47, 48 does not correspond exactly to the gear neutral axis, but is in fact, rotated in the direction of rotation of the gears through a small angle of from 4–10 degrees, but preferably 5 degrees whereby the force of the fluid in the high pressure chambers will create a radial force on the internally-toothed gear in a direction to exert a light closing force on the gear teeth at open mesh as is described in my co-pending application, Serial No. 814,320. This angle in the embodiment shown is maintained by an appropriately located groove 3 in the manifold member lug 4 and a pin 2 extending axially from the eccentric ring 17 on its neutral axis engages in this groove and relatively locates the two members.

Where the loose bearing ring of application Serial No. 814,320 is not employed, then it is desirable to have the diametrical center line of the lands 47, 48 correspond with and be parallel to the neutral axis of the gears. In such event, it is preferred that the third discharge port 44 have a slightly greater circumferential width than the first discharge port 45 and the second discharge port 40 is then displaced slightly from its symmetry about the line perpendicular to the land diametrical line and toward the open mesh land.

With either arrangement, a desirable force action results. Thus, if the second discharge port is at zero pressure, the force in the chambers communicating with the first and third ports is in a direction to urge the gear teeth at open mesh into closed engagement with a light pressure. If the first and third ports are at zero pressure, then the resultant force of the hydraulic pressures in the chambers communicating with the second port tends to urge the gear teeth at open mesh apart. However, this is not undesirable because the tendency for leakage from the chambers communicating the first discharge port to the inlet port is reduced or of no consequence.

The advantages of providing first, second and third discharge ports with the first and third port interconnected for a pump having two separate outputs, is apparent when considered in relation to a similar pump having but two discharge ports of equal line of movement width, each connected to a different pump output. Assuming that there are six passages 20, each having a line of movement width of 30° and with the lands there-between having a line of movement width of 30°, each of the two discharge ports on the prior art pumps would have a line of movement width of 60°. With such an arrangement, it will be apparent that two passages 20 are always in communication with each of the two discharge ports. The rate of fluid being discharged from each chamber is proportional to the sine of the angle *a* of the chamber from the open mesh land and may be illustrated graphically as is shown in FIGURE 5 by the curves 101, 102 and 103 which are plots of the volume of fluid discharged from each of three adjacent chambers per degree of rotation in relation to the angle *a* of each chamber from the open mesh land.

FIGURE 6 shows the rate of fluid being discharged into the first of the two discharge ports of such prior art pumps, curve 111 corresponding to the curve 101 of FIGURE 5 and curve 112 corresponding to the curve 102 of FIGURE 5 and showing how two chambers are always in communication with and discharging into one of the two discharge ports. Curve 115 illustrates the total rate of fluid being discharged into such first discharge port, it being noted that the curve 115 rises as a sine function of the angle *a* until the rate equals 1.5 times the volume being discharged from any one chamber and then drops precipitously and instantaneously to .5 times this volume. It will be apparent from FIGURE 6 that there is an abrupt and instantaneous variation in the rate of flow into the first discharge port of a two-discharge port type pump of from .5 to 1.5 times the rate of volume being discharged from any one chamber.

FIGURE 7 is similar to FIGURE 6 but shows that the same thing happens but in reverse for the second of the two discharge ports.

FIGURE 8 shows how the present invention completely eliminates the undesirable effects of cutting off the fluid from a chamber when the flow of fluid is at or close to its maximum. Thus, using the present invention, during the discharge stage of each chamber the chamber will be in communication with one of the three discharge ports for 60° of its rotation rather than 90° as heretofore. Curve 121 corresponds to curve 101 of FIGURE 5 and illustrates the rate of volume flow into the first discharge port, it being noted that this flow of fluid is cut off abruptly at 60°. However, and in accordance with the invention, the third discharge port is intercommunicated to the first discharge port and thus the flow of fluids into two ports are combined. Curve 124 shows the change in the rate of flow of fluid into the third discharge port. Curve 125 shows the total of the flow of fluid into the two ports, it being noted that the flow varies from .866 to one times the maximum rate of flow of fluid from any one chamber and this variation is a sinusoidal variation rather than an abrupt change.

FIGURE 9 shows the variations in the rate of flow of fluid into the second discharge port of a pump constructed in accordance with the present invention. It is to be noted that while each chamber is only in communication with the second discharge port for 60 degrees of its rotation and that while its flow of discharging fluid is cut off when the rate of flow is at or close to its maximum, by virtue of the fact that at the same instant that one chamber is cut off, another chamber comes into communication with the second discharge port, a rate of flow of fluid into the second discharge port can be obtained which varies sinusoidally from .866 to one times the maximum rate of flow from any one chamber, and there are no abrupt changes in the rate of flow as with the prior art discharge pumps. In this respect, it is to be noted that the second discharge port should have a sufficient line of movement width that one chamber is always in communication therewith.

It is to be noted that these desirable results may be obtained for pumps having two or more independent discharges at different pressures or different volumes. The principle of the invention may be applied to such pumps so long as an odd number of discharge ports are employed and the first and last discharge ports, the second and next to the last, the third and next to last, are connected in pairs with the odd or middle port being independent of any other port.

It will also be appreciated that this porting or manifolding arrangement may be applied with equal value to vane type pumps or rotating cylinder axially reciprocal type pumps. Also, if a hydraulic motor of this type or

internal gear type must operate with inputs at two different pressures, the manifolding and porting arrangement may also be employed there with equal advantage.

Further in accordance with the invention, the circumferentially facing side walls of the passages **20** and the circumferentially facing side walls of the discharge ports **40**, **44**, **45** are parallel over the entire radial width of the passages and ports at the instant a passage **20** comes into communication with or goes out of communication with one of the discharge ports so that at this instant there is a maximum rate of change per degree of relative movement in the area of the opening between the ports and the passages as well as presenting what is, in effect, a sharp edge orifice.

By parallel, it is not meant to infer that the side walls must be straight or flat; to the contrary, "parallel" means that the leading wall, or edge, **62** of the passage **20** matches the trailing wall, or edge, **61** of the discharge ports **40**, **44**, **45** as the passage **30** initially registers with one of the ports **40**, **44** or **45** and the trailing wall, or edge, **63** of the passage matches the leading wall, or edge, **60** of the ports as the passage goes out of register with the ports. Accordingly, the walls, or edges, can be appropriately curved and still be parallel at the desired instant in their relative movement. Thus this parallelism may be accomplished in a number of different ways, but preferably each discharge port has flat leading and trailing circumferentially facing side walls **60**, **61** which are each parallel to the radial line through the center of the adjacent land. These side walls are also perpendicular to the sealing surface **42** so as to form leading and trailing right-angled corners **60'**, **61'**. By leading and trailing is meant the side remote from and closer to the open mesh land respectively.

In a like manner, each passage **20** has flat leading and trailing circumferentially facing side walls **62**, **63** which are each parallel to the radial line through the middle of the passage and the circumferential spacing of the walls **62**, **63** equals the spacing of the lands **47**, **48**. These surfaces are also perpendicular to the surface **5** and form leading and trailing right-angled corners **62'**, **63'**.

As will be apparent from the geometry, as the corners **60'**, **63'** approach each other to close a passage **20** from communication with one of the discharge ports, the corners will be parallel to each other to form an opening which is generally elongated and rectangular in shape, as distinguished from the bi-convex shaped opening of the prior art formed by arcuate leading and trailing side walls of the port and passages. By this construction, a maximum change in the area of the opening communicating the passages and ports may be obtained for each degree of relative movement as the passages come into communication with or go out of communication with a port. Furthermore, at this instant the right-angled corners present what may be termed a sharp edged orifice to the movement of fluid through the opening. As is known, flow of fluid through such an orifice is independent of the fluid viscosity and only dependent on the pressure differential. Variations in the fluid viscosity do not affect the operation and efficiencies of the pump to the same degree that they did in prior art pumps wherein the side walls of the passages and ports were not parallel at the moment of closing the opening.

Sealing Member

The sealing member **13** is mounted on the shaft **16** and desirably forms an integral part thereof. It may be welded thereto, but in the preferred embodiment, has an interference fit with the shaft.

The sealing member **13** and shaft **16** are mounted for limited axial movement and for rotation within the housing by any suitable means such as a roller bearing **54** consisting of an outer race press-fitted into the housing and a plurality of circumferentially spaced cylindrical rollers engaging an outer cylindrical surface **56** on the sealing member **13**.

The sealing member **13** has a surface **57** which extends radially outwardly beyond the outer surface of the ring gear **12** and is in sealing engagement with the right hand axial faces of the gears **11**, **12** to close the right axial end of the chambers **21**. The high pressure fluids in the high pressure chambers exert a radially offset axial force to the right on the member **13** which force as will appear, is opposed by hydraulic pressures in the pump cavity exerting an axial force on the surfaces **84**, **86** of the sealing member **13** facing in an axial direction opposite to that of the surface **57**. The size of the force is equal to the product of the hydraulic pressure and the sum of the area of the surfaces **84**, **86** exposed to such pressures. Preferably the size of this force is approximately 14% greater than the size of the force in the opposite direction and as will appear, the pressures in the housing cavity are so proportioned in relation to the area of these two surfaces to produce the desired axial force. The area of the surface **84** exposed to such housing pressures is limited or held to zero (as is shown in the drawings) by a sealing ring **75**.

Sealing Ring

The sealing ring **75** is generally in the shape of a sleeve and is axially slidable in a sealed relationship with the housing cavity defined by the cylindrical surface **34'** by means of an O-ring **76** mounted on the outer surface of the ring. The ring **75** surrounds the shaft **16** and has a left hand axially facing surface **78** formed on a radially outwardly extending flange **78'** in pressure sealing relationship with the right hand axially facing surface **84** of the sealing member **13**. The ring **75** in conjunction with a packing member **95** thus defines an internal cavity **79** which is at inlet pump pressure, it being noted that the cavity **79** is communicated with the inlet through the keyway **19**, a small counterbore **91** in the surface **57**, an opposite keyway **92** in the gear **11**, and a groove or passage **93** in the manifold member **14**.

A compression spring **81** between the base of the housing part **23** and the right end of the sealing ring **75** biases the surfaces **78**, **84** into a limited pressure engagement. It is to be noted that this spring also presses the sealing member **13** into engagement with the axial faces of the gear **11**, **12** and presses the gear **11**, **12** into pressure engagement with the sealing surface of the manifold member **14**. The spring **81** is relatively weak and simply provides an initial force to maintain the various surfaces in pressure engagement when the pump is not operating or when it is started into operation. The principal sealing force is the hydraulically produced force above-referred to.

The surface **78** engaging the surface **84** seals the pressures in the pump cavity **41** from the cavity **79** which is at low pressure. The sealing pressure between these surfaces is provided by the pressures in the cavity **41** exerting a force on a right hand axially facing surface **87** on the flange **78'**, the area of which surface is so proportioned that the force produced urging the surfaces together is just equal to the force of the pressures in the cavity **41** tending to separate these surfaces.

Pressure Integrating

As is known, the hydraulic pressures of the fluids in the high pressure chambers and in the discharge ports, create forces tending to separate the sealing member **13** from its sealing engagement with the gears **11**, **12** and the sealing surface **5** of the ported plate **6** from the sealing surface **42** of the manifold member **14**.

My co-pending applications Serial Nos. 656,657, now Patent No. 2,956,512, 656,117, now Patent No. 3,007,418, and 814,320, now Patent No. 3,034,448, together with my co-pending application Serial No. 16,765 filed concurrently herewith, all describe an arrangement for

creating an axial sealing force on the member to oppose the forces created by the hydraulic fluids in the chambers and discharge ports. In the apparatus described in all of these applications, the interior cavity of the pump has the discharge pressure communicated thereto, and this pressure bears on the side of the sealing member **13** axially remote from the gears and creates such a sealing force. Such an arrangement is satisfactory for a pump having a single discharge pressure. However, with a pump having multiple discharge pressures, this is not possible because if one of the discharge pressures is communicated to the housing cavity and the other discharge pressure goes to zero, then too high a sealing force results. On the other hand, if the discharge port which is communicated with the housing cavity drops or even goes to zero, then insufficient or no sealing force results.

In accordance with the present invention, both discharge pressures are combined to provide a pressure on the inside of the housing cavity and against the axial side of the sealing member **13** remote from the gears **11, 12** which is proportional to both of the discharge pressures.

While such means may take a number of different forms, in the embodiment of the invention shown in FIGURE 1, a pressure integrator is provided. Such pressure integrator will preferably be formed as a component part of the pump housing or manifold plate. For the purpose of clarity, it is shown as separate, and in the embodiment shown, includes a housing M having an interior cylindrical bore with a piston P reciprocally mounted therein.

The housing M may take any desired form, but in the embodiment shown includes a main body portion **150** and a closure member **151** threadably inserted therein and sealed by an O-ring **152**.

The piston P includes a first portion **154** of a predetermined diameter and a second portion **155** of a predetermined and larger diameter. The portion **154** has a pressure surface **157** on its left hand axial end defining with the housing body **150** a pressure cavity **158** which is communicated with the second discharge port **40** through a conduit **159**. This port may be considered as having a pressure P–**1**.

In a like manner, the left hand end of the larger piston portion **155** forms a pressure surface **165** defining with the bore and the outer surface of the piston portion **154** a pressure cavity **168** which communicates with the first and third discharge ports through a conduit **170**. The ports may be considered as having a pressure P–**2**.

Also the larger piston portion **155** has a pressure surface **160** on its right hand axial end defining with the housing M a pressure cavity **162** which communicates through an internal passage **163** in the housing M and a conduit **164** with the interior of that portion of the pump cavity in communication with the right hand axial end of the sealing member **13**. The cavity may be considered as having a pressure P–**3**.

It will be appreciated that the pressure in each cavity **158, 168** and **162** on its respective pressure surface, will exert an axial force on the piston P tending to move it axially either to the right or left depending upon the vector sum of the forces produced. This movement of the piston is employed to actuate a valve controlling flow of fluid from either of the discharge ports to the housing cavity such that a hydraulic force of the desired amount is produced on the sealing member **13**.

Such valve may take a number of different forms, but in accordance with the present invention, a preferred form of valve includes a valve member **172** in the form of a spherical ball axially movable in a valve chamber **173** formed on the interior of the piston portion **154**. An axial passage **174** of a diameter smaller than that of the valve chamber **173** extends to the right therefrom and communicates with a radial passage **176** which in turn communicates with the cavity **168**. The intersection of the walls of the passage **174** with the end wall of the valve chamber **173** forms a valve seat **178** against which the valve member **172** can move into sealing engagement.

A sleeve **180** threaded in the left end of the piston portion **154** has an axial passage **181** of a diameter smaller than the valve chamber **173** which communicates with the pressure cavity **158**. The walls of the bore **181** where they intersect with the left end of the valve chamber **173** form a valve seat **182** which the valve member **172** can sealingly engage. Passages **183** in the piston and **184** in the housing, when aligned communicate the valve chamber with the passage **163** which, as previously pointed out, communicates with the interior of the pump cavity.

In operation, let it be assumed that the second discharge port is operating at a pressure P–1 and the first and third discharge ports are operating at a pressure P–2 and that the interior of the cavity 41 is at a pressure P–3, such pressure P–3 being selected in relation to the areas on the sealing member **13** above referred to to provide the desired axial sealing force on the gears **11, 12**. The forces tending to move the piston P to the right are equal to P–1 times the area of the pressure surface **157** plus P–2 times the area of the surface **165** while the forces tending to move the piston P to the left are equal to P–3 times the area of pressure surface **160**. If it be assumed that the pressure P–1 is zero, then hydraulic fluid will flow from the conduit **170** to the pressure cavity **168** to the passages **176, 174** to the valve chamber **173**, thence through the passages **183, 184** to the conduit **164** and pressure P–3 will rise. The valve member **172** is held against the valve seat **182** and the fluids will not flow to the conduit **159**. Pressure P–3 therefore rises until the axial force of the pressures in the chamber **162** move the piston P to the left until the passage **183** moves out of communication with the passage **184** at which point further increase of pressure P–3 ceases.

If pressure P–3 drops due to leakage in the pump, the left hand axial force on the piston P also drops and the piston P moves to the right to bring two passages **183, 184** again into communication until the pressure P–3 again rises to the desired amount and the piston P again moves to the left so that the passages **183, 184** are no longer in communication.

Assuming that pressure P–1 rises, this means that the axial force to the right on the member **13** will also rise. A force is exerted to the right on the piston P and moves it to the right to bring the passages **183, 184** again into communication and fluid again flows from the conduit **170** until pressure P–3 again rises to where the piston P can move to the left and cut off the flow of fluid from P–2 to P–3. This action will continue as pressure P–1 increases until it equals pressure P–2.

If it now be assumed that pressure P–1 exceeds pressure P–2, the valve member **172** will move to the right and engage valve seat **178** to close pressure P–2 off from communication with the valve chamber **173**. At the same time, however, pressure P–1 is now communicated with the valve chamber **173** and can flow pressure P–3 and the action will continue as before.

By properly interrelating the various areas, a proper sealing force on the sealing member **13** can always be obtained for any combination of output pressures.

It will thus be apparent that embodiments of the invention have been described in detail which accomplish all of the objectives heretofore set forth and others and provide a hydraulic pump of a multiple pressure discharge type which is improved in performance, a much more uniform flow of fluid to each discharge, and which always has a sealing force proportional to the two discharge pressures.

The invention has been described with reference to a preferred embodiment. Obviously modifications and alterations will occur to others upon a reading and under-

standing of this specification, and it is my intention to include all such modifications and alterations insofar as they come within the scope of the appended claims.

Having thus described my invention, I claim:

1. In a hydraulic pump of the type capable of simultaneously discharging hydraulic fluid at two different pressures, said pump including a plurality of members defining a plurality of chambers revolving on a fixed line of movement and which progressively increase in volume to a maximum and then decrease in volume to a minimum and then again increase in volume to a maximum, an inlet port arranged to be in communication with said chambers as they increase in volume, the improvement which comprises three separate discharge ports each arranged on the line of movement of said chambers to be in communication at any one instant with different decreasing volume chambers as they revolve to receive hydraulic fluid therefrom, and means interconnecting the first and third of said ports.

2. In a multiple discharge hydraulic pump comprised of a plurality of members movable relative to each other and defining a plurality of pumping chambers revolving in a fixed closed path of movement, said chambers gradually increasing in volume after they pass a fixed point of minimum volume on said path of movement until they reach a fixed point of maximum volume on said path of movement and then gradually decreasing in volume until they reach said fixed point of minimum volume, means defining an inlet port having circumferentially spaced ends and outlet ports including lands separating the adjacent ends of said ports, means defining a passage from each chamber and revolving therewith, each of said passages moving past said lands to sequentially communicate its associated chamber with said inlet and said outlet ports, the improvement which comprises there being first, second and third outlet ports and means intercommunicating the first and third ports.

3. In a multiple discharge hydraulic pump comprised of a plurality of members movable relative to each other and defining a plurality of pumping chambers revolving on a fixed closed path of movement, said chambers gradually increasing in volume after they pass a fixed point of minimum volume on said path of movement until they reach a fixed point of maximum volume on said path of movement, and then gradually decreasing in volume until they reach said fixed point of minimum volume, a plurality of ports separated at their adjacent ends by lands which ports supply and receive fluid from said chambers, the improvement which comprises: said ports including an inlet port and first, second and third discharge ports all arranged in circumferentially spaced relationship on said path of movement to be sequentially in communication with said chambers and means interconnecting said first and third discharge ports.

4. In a multiple discharge hydraulic pump comprised of a plurality of members movable relative to each other and defining a plurality of pumping chambers revolving in a fixed closed path of movement, said chambers gradually increasing in volume after they pass a fixed point of minimum volume on said path of movement until they reach a fixed point of maximum volume on said path of movement, and then gradually decreasing in volume until they reach said fixed point of minimum volume, a ported plate movable with said members and having a plurality of passages therethrough one communicating with each chamber, the improvement which comprises means defining circumferentially arranged inlet and first, second and third discharge ports including lands between the adjacent ends of said inlet port and said first, second and third discharge ports respectively and means interconnecting said first and third discharge ports.

5. In a multiple discharge hydraulic pump comprised of a plurality of members movable relative to each other and defining a plurality of pumping chambers revolving in a fixed closed path of movement, said chambers gradually increasing in volume after they pass a fixed point of minimum volume on said path of movement until they reach a fixed point of maximum volume on said path of movement and then gradually decreasing until they reach said fixed point of minimum volume, a ported plate movable with said members and having a plurality of passages therethrough, one communicating with each chamber which passages move with their associated chambers, means defining a circumferentially arranged inlet and at least one discharge port including a land between adjacent ends of said ports and sealingly separating said ports one from the other, said passages of said ported plate moving past said lands to communicate its associated chamber with one of said ports, the improvement which comprises the adjacent side walls of said passages and said ports being parallel immediately as a passage comes into communication with a port or goes out of communication therefrom.

6. The improvement of claim 5 wherein both circumferentially facing end walls of said passages are parallel to the radial line through the center of said passages and the end wall of said discharge ports facing opposite to the direction of rotation is parallel to the radial line through the center of the adjacent land.

7. In a hydraulic pump comprised of in combination a housing having an inwardly facing surface defining a pumping cavity, a shaft extending into said housing, an externally toothed gear supported on said shaft for rotation therewith, an internally toothed gear having teeth in sliding sealing engagement with said externally toothed gear and rotatable about an axis spaced from the shaft axis, a plane defined by said axes defining the gear neutral plane, bearing means supporting said internally toothed gear for rotation, said gear teeth moving from open to closed mesh on the gear neutral plane as the gears rotate and defining a plurality of revolving chambers, a sealing member in sealing engagement with one axial end of said gear, a ported plate in sealing engagement with the other axial end of said gears and having a plurality of axially extending passages, one for each of said chambers, a manifold member in sealing engagement with the other side of said ported plate and having circumferentially arranged inlet and outlet ports therein opening towards said ported plate passages, said manifold member also having lands separating the adjacent ends of said ports, the lands at the ends of said inlet port being located at said closed and said open mesh points of said gears, each passage moving past said lands and communicating its respective chamber alternately with said ports, the improvement which comprises there being first, second and third outlet ports, counting from the open mesh land, and means intercommunicating said first and third ports.

8. In a hydraulic pump comprised of in combination a housing having an inwardly facing surface defining a pumping cavity, a shaft extending into said housing, an externally toothed gear supported on said shaft for rotation therewith, an internally toothed gear having teeth in sliding sealing engagement with said externally toothed gear and rotatable about an axis spaced from said shaft axis, a plane defined by said axes defining the gear neutral plane, bearing means supporting said internally toothed gear for rotation, said gear teeth moving from open to closed mesh on the gear neutral plane as the gears rotate and defining a plurality of revolving chambers, a sealing member in sealing engagement with one axial end of said gear, a ported plate in sealing engagement with the other axial end of said gears, and having a plurality of axially extending passages, one for each of said chambers, a manifold member in sealing engagement with the other side of said ported plate and having circumferentially arranged inlet and outlet ports therein opening towards said ported plate passages, means forming a land between adjacent ends of said ports, the improvement which comprises the circumferentially spac-

ing ends of said outlet port and of said ported plate passages being generally flat and parallel at the moment that a ported plate passage comes into communication with the outlet port or leaves communication with such outlet ports.

9. A hydraulic pump comprising a plurality of members defining a plurality of pumping chambers revolving along a fixed line of movement, said chambers alternating between an intake stage wherein the volume of said chambers progressively increases to a maximum and a discharge stage wherein the volume of said chambers progressively decreases to a minimum, said chambers having a gradually increasing rate of discharge to a maximum rate and then a gradually decreasing rate of discharge to a minimum as the chambers move through said discharge stage, the improvement which comprises: a separate passage communicating with each of said chambers, a fluid inlet adapted to register with said passages of said chambers during said intake stage to direct fluid into said chambers, a first, second and third outlet port each adapted to register with a separate passage of said pumping chambers during said discharge stage, said ports arranged in sequence along said line of movement, said first port communicating with a chamber having an increasing rate of discharge until the rate is at a predetermined level below the maximum rate of discharge, said second port communicating with a chamber having an increasing rate of discharge above said predetermined level and then a decreasing rate of discharge until said rate decreases to said predetermined level, said third port communicating with a chamber having a decreasing rate of discharge from said predetermined level to a minimum rate of discharge, said first and third ports being interconnected so that the rate of discharged into said ports are added.

10. The improvement as defined in claim 9 wherein said passages and ports have leading and trailing edges in the direction of movement of said chambers, said leading edges of said passages matching said trailing edges of said ports as said leading edges of said passages pass over said trailing edges of said ports and said trailing edges of said passages matching said leading edges of said ports when said trailing edges of said passages pass over said leading edges of said ports.

11. The improvement as defined in claim 10 wherein the leading edge of one of said passages simultaneously passes over the trailing edge of said first port as the leading edge of another passage is passing over the trailing edge of said third port and the trailing edge of said one passage simultaneously passing over the leading edge of said first port as the trailing edge of said other passage is passing over the leading edge of said third port.

References Cited in the file of this patent

UNITED STATES PATENTS

2,335,814	Stevenson	Nov. 30, 1943
2,368,883	Roth	Feb. 6, 1945
2,437,791	Roth et al.	Mar. 16, 1948
2,452,470	Johnson	Oct. 26, 1948
2,487,732	Schanzlin	Nov. 8, 1949
2,521,592	McManus	Sept. 5, 1950
2,732,802	Eames	Jan. 31, 1956
2,811,979	Presnell	Nov. 5, 1957
2,956,512	Brundage	Oct. 18, 1960

FOREIGN PATENTS

487,779	Canada	Nov. 4, 1952
529,664	Great Britain	Nov. 26, 1940