Aug. 19, 1952 H. G. MUELLER 2,607,296

FLUID PUMP UNIT

Filed May 16, 1946 9 Sheets-Sheet 1

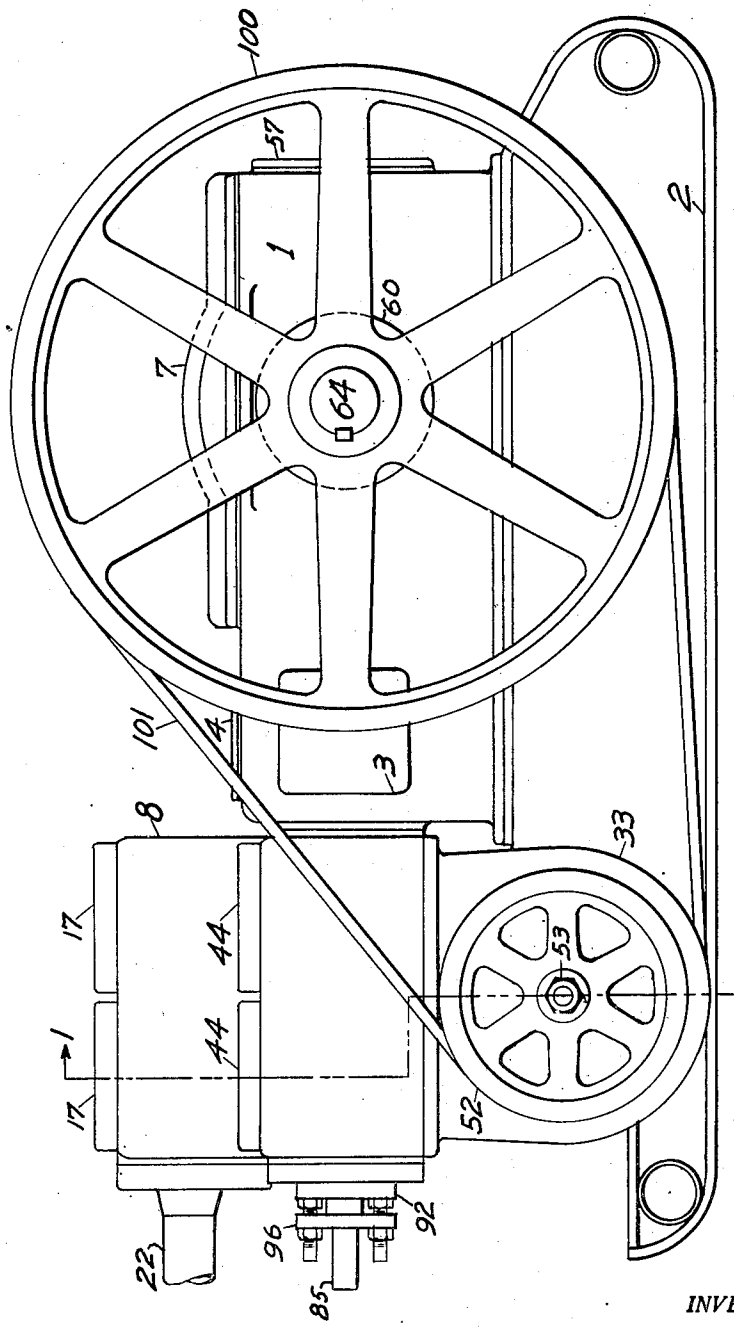


FIG. 1

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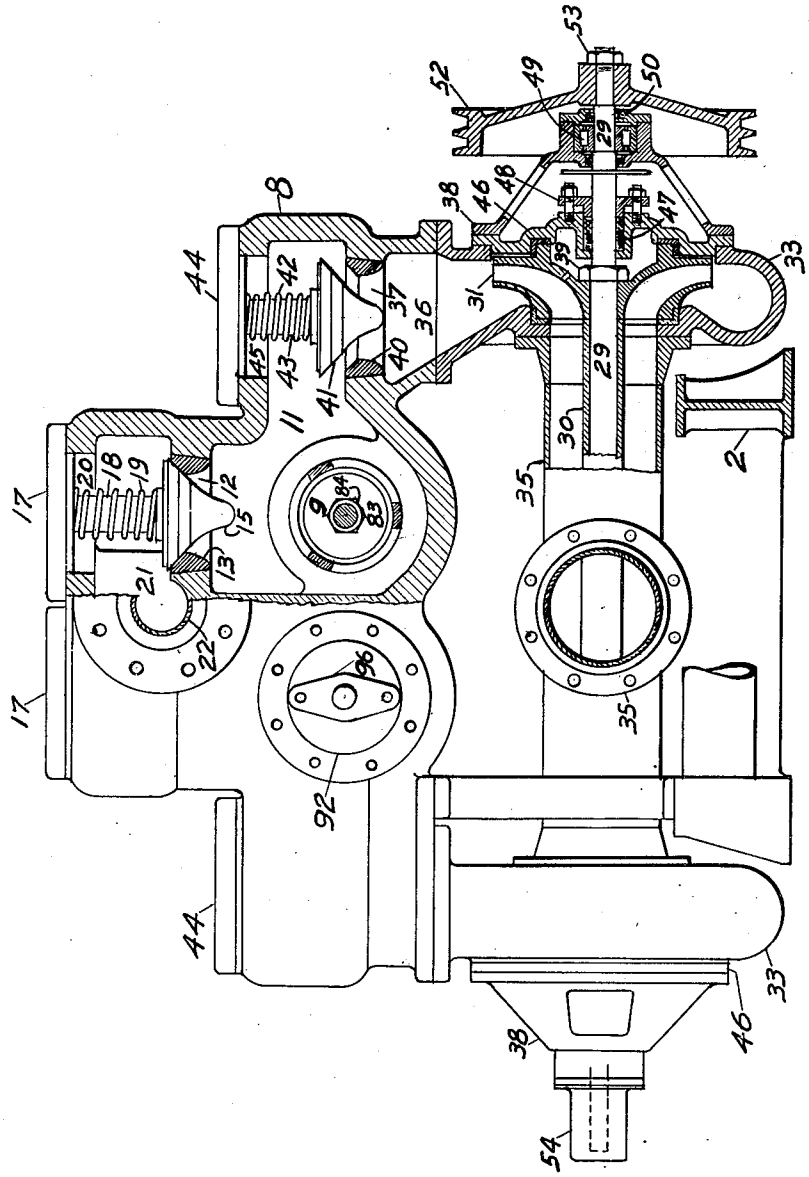


FIG. 2

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FLUID PUMP UNIT

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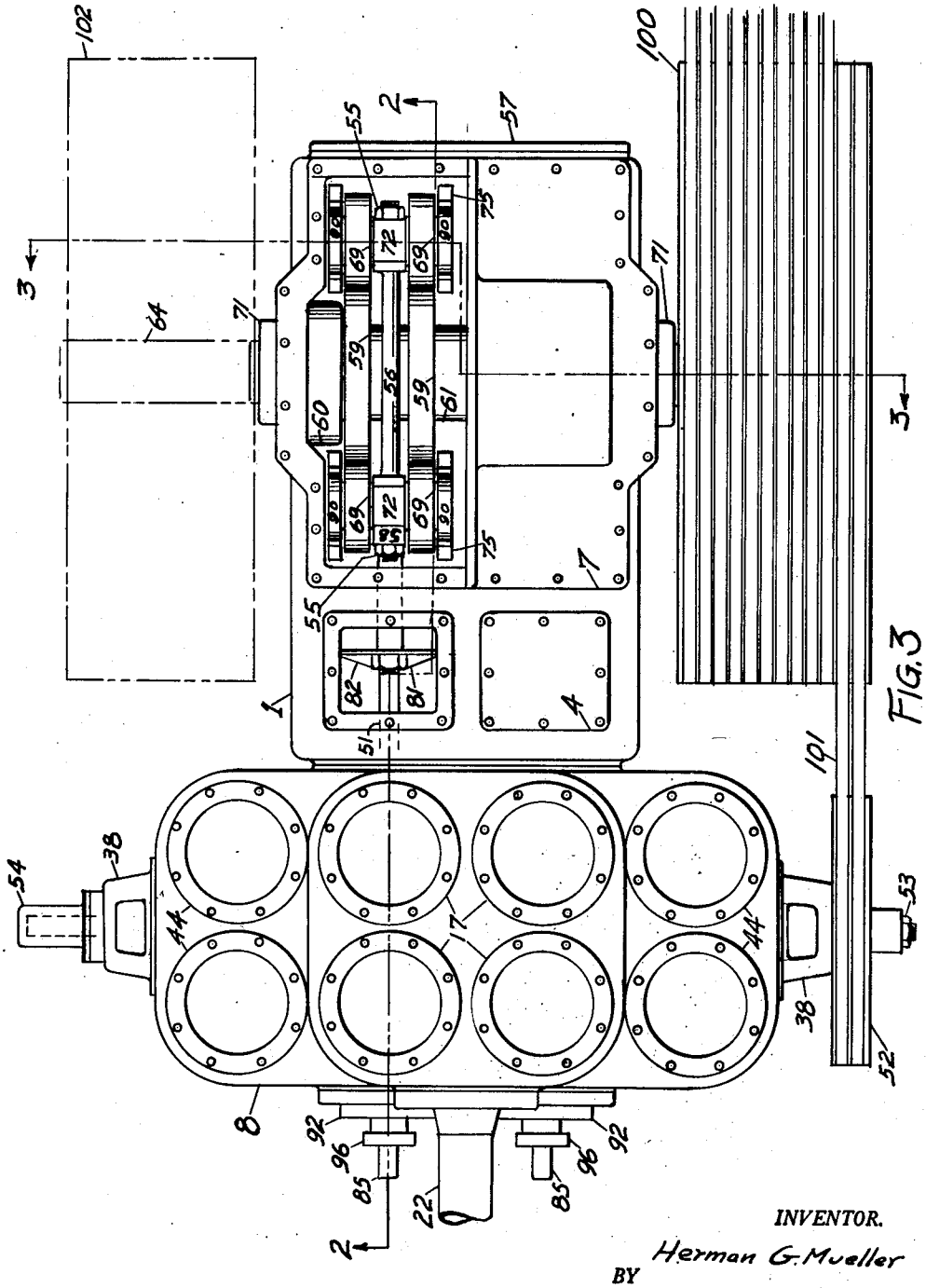


FIG. 3

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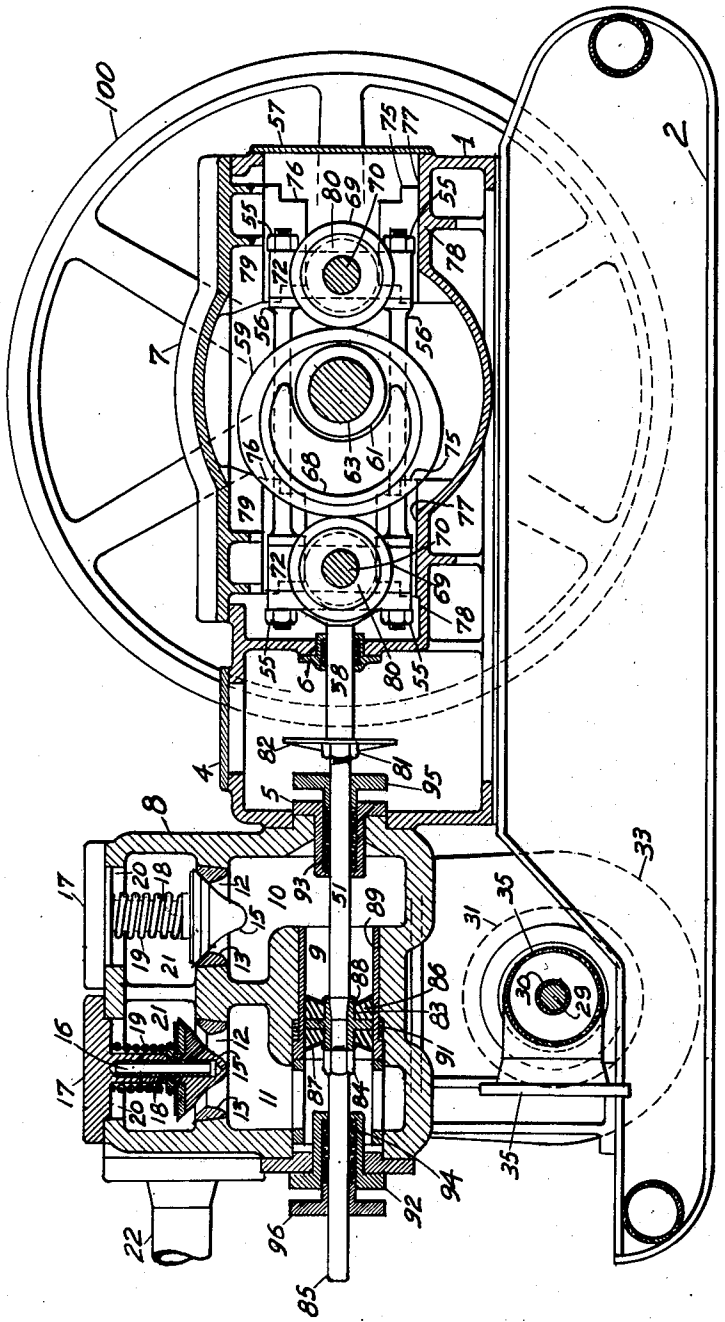


FIG. 4

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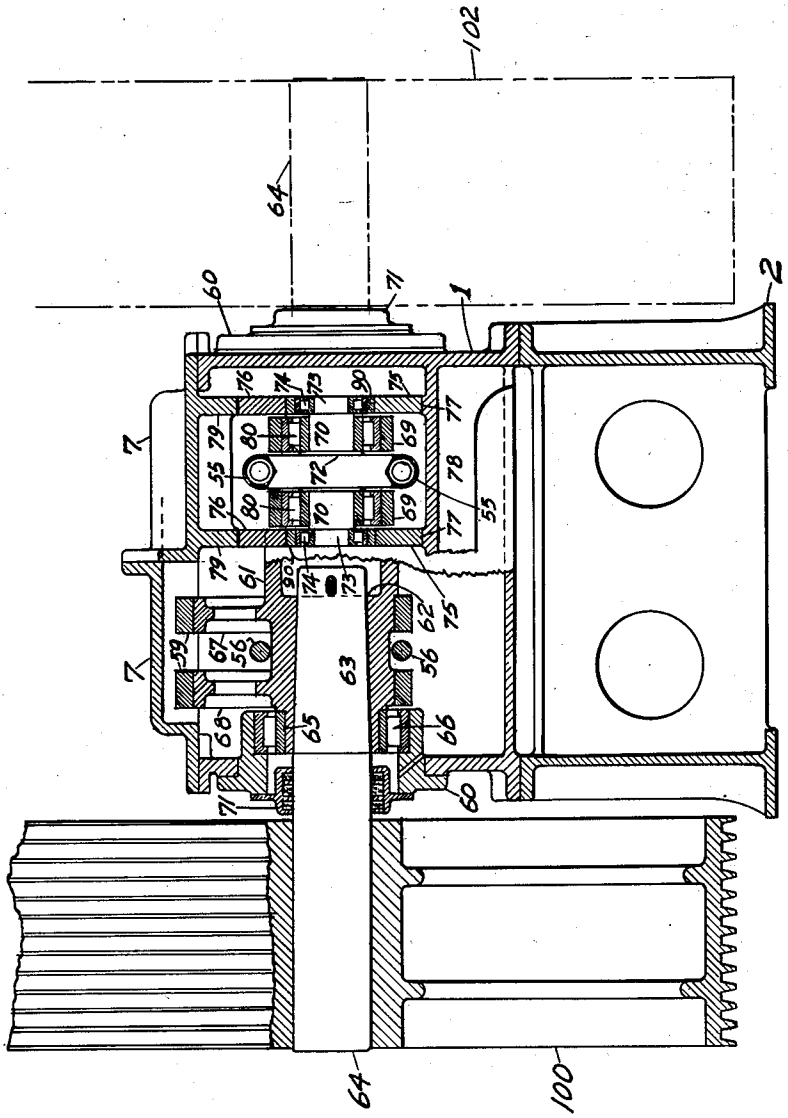


FIG. 5

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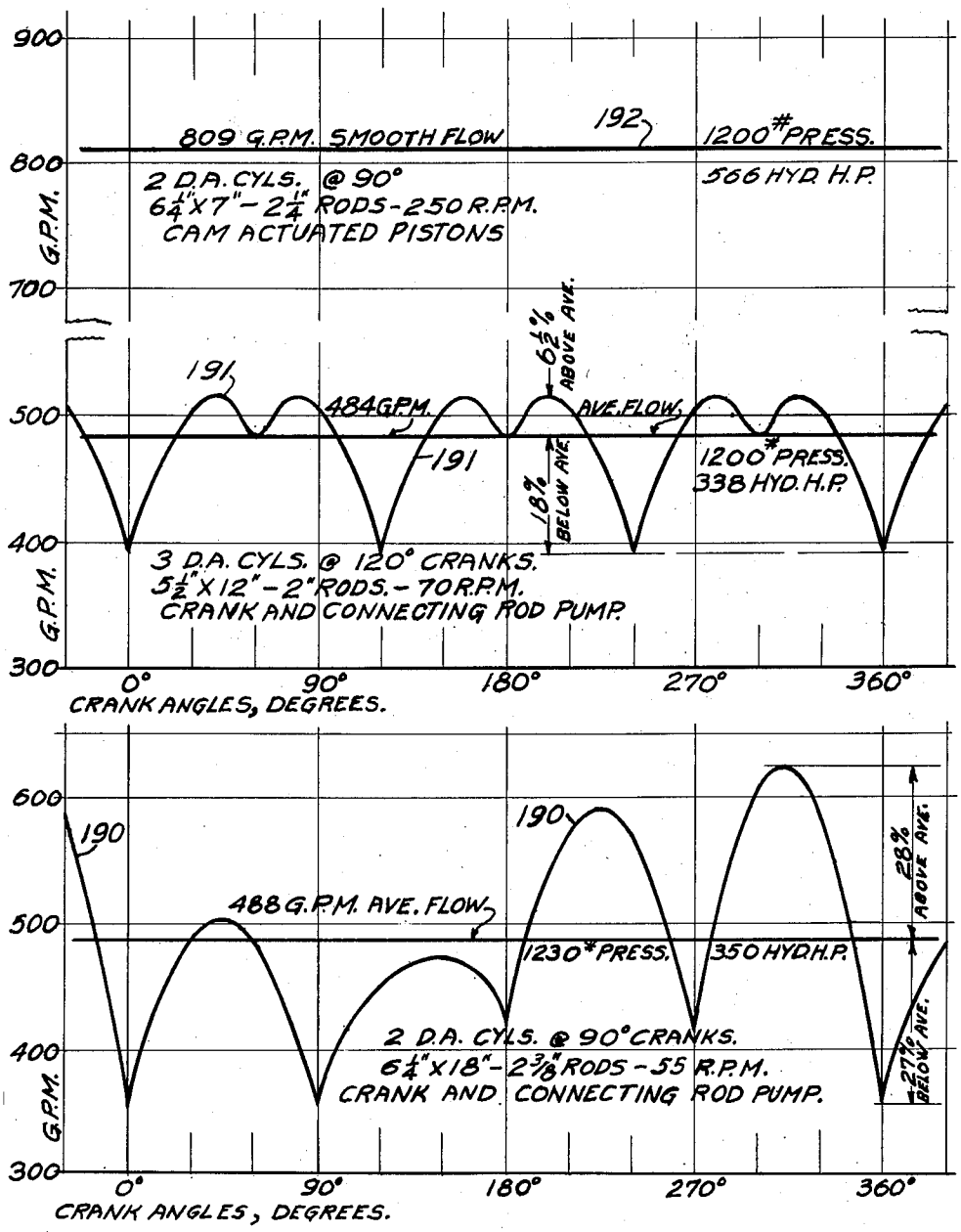


FIG. 6

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Aug. 19, 1952 H. G. MUELLER 2,607,296

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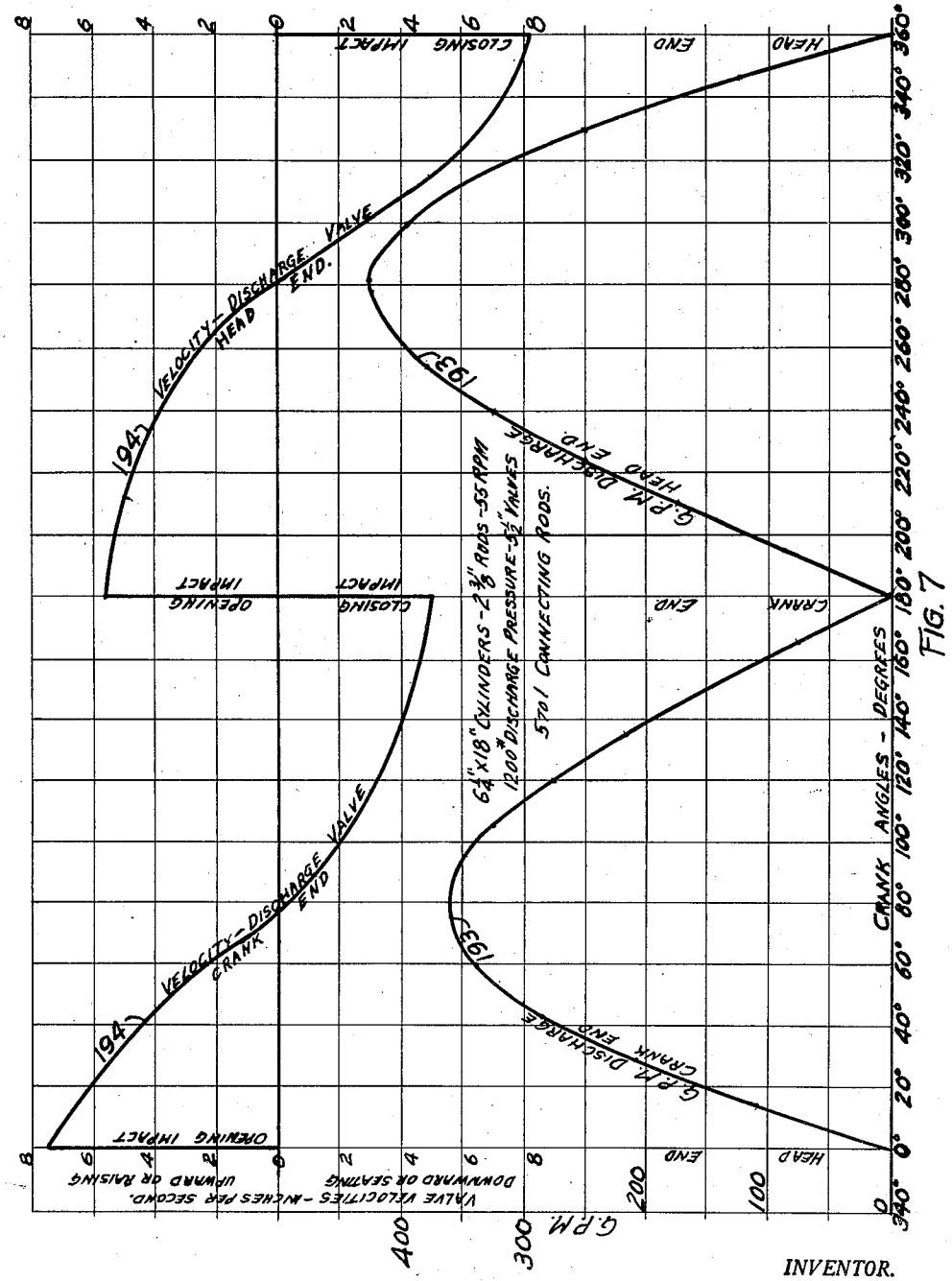


FIG. 7

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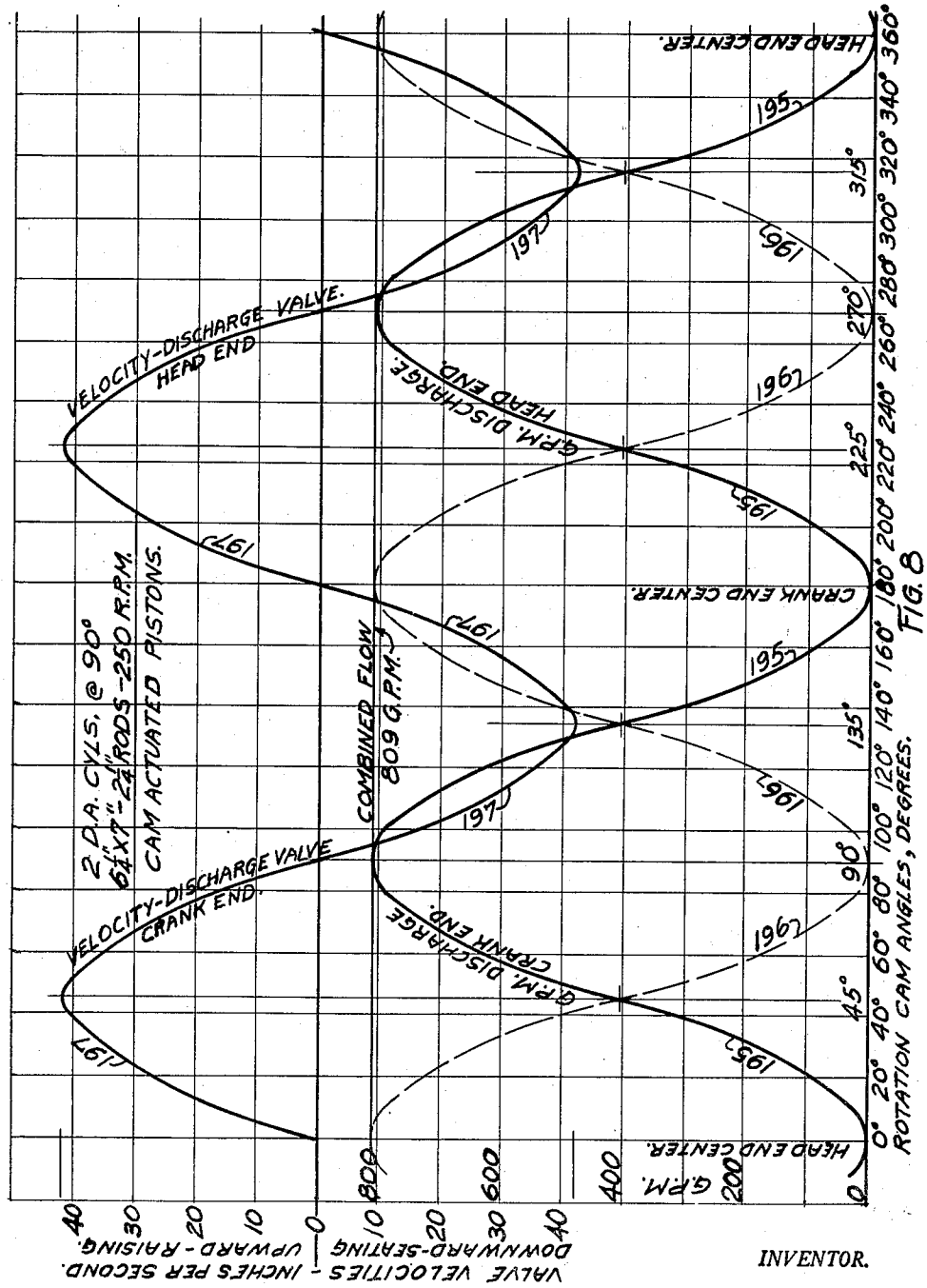


FIG. 8

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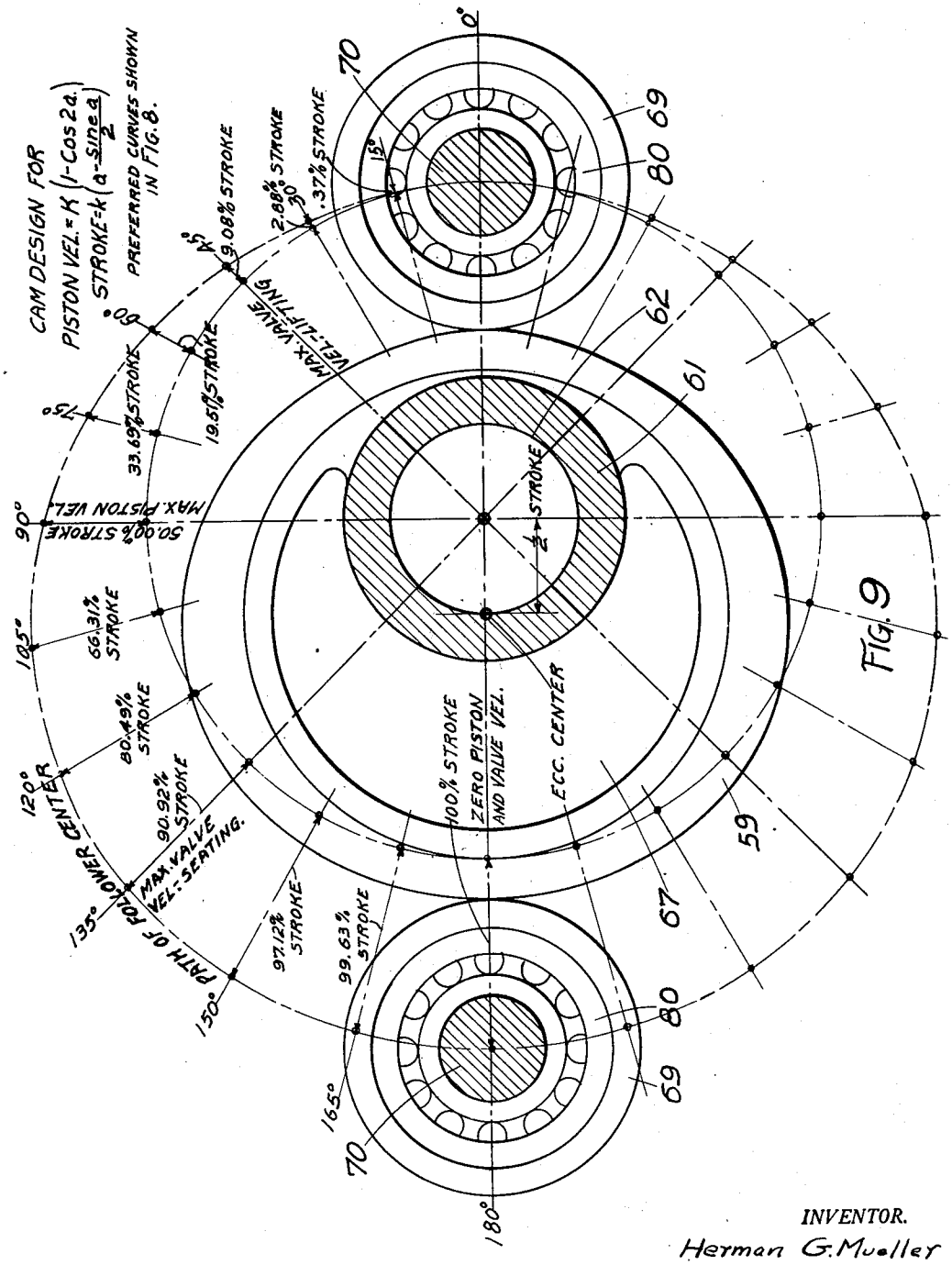


Fig. 9

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UNITED STATES PATENT OFFICE

2,607,296

FLUID PUMP UNIT

Herman G. Mueller, Erie, Pa.

Application May 16, 1946, Serial No. 670,192

2 Claims. (Cl. 103—5)

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This invention relates generally to pumps and more particularly to hydraulic multi-cylinder liquid pumps driven from an external power source and customarily lifting the liquid to be pumped, by suction, from a sump and forcing it against hydraulic pressure into a discharge line.

Conventional multi-cylinder liquid pumps have 2, 3, or more hydraulic cylinders, either single or double acting, each with a reciprocating piston, actuated by a piston rod, a crosshead and a connecting rod, transmitting the rotative movement from a crank or eccentric to the reciprocating motion of a crosshead and a piston.

In some cases Scotch yokes are used with sliding crossheads in the yoke, which are mounted on the crankpin, the crosshead taking one component of the rotating crank motion and the yoke the other component, they being at right angles to each other. Poppet-type, spring-loaded suction and discharge valves are commonly used in hydraulic pumps, one or more of each generally being provided for each end of each cylinder.

The Scotch yoke construction results in a simple harmonic motion of the piston, and the crank with connecting rod construction, results in a similar piston motion somewhat modified, due to the angularity of the connecting rod. With either construction, the piston has its greatest acceleration at the ends of the stroke with the maximum velocity of the piston and maximum rate of suction or discharge of the liquid near the middle of the stroke. The lift of the spring-loaded valves is approximately proportional to the piston velocity, or the rate of flow through the valves, and it follows that the valve rate of lift or valve velocity is then approximately proportional to the piston acceleration which, as stated, is maximum at the end of the stroke when the liquid flow is zero and the valve is just beginning to open or is closing. In other words, the velocity is greatest at the beginning of its lift, requiring an impact force to accelerate it instantly from rest, to its maximum velocity. It is also at its maximum velocity, coming to an instant stop, as it seats. The result is a slamming of the valve in opening and seating and in order that this slamming does not become harmful, very slow crank speeds are necessary. Because of these very low speeds such pumps are necessarily very large and heavy and costly.

There are also other reasons necessitating slow rotative speeds with conventional pumps. Another of these is because of large variations in the rate of flow of the combined cylinders. With

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any single reciprocating piston the rate of flow, as stated above, is zero at each end of the stroke. To even out the combined flow, multi-cylinder arrangements are used and in general the more cylinders used the smoother the flow or the smaller the variations in flow.

With a conventional two-cylinder double-acting connecting rod or Scotch yoke pump and with two cranks in their best relative positions, namely 90° apart, these flow variations are on the order of twenty-seven per cent (27%) above and below the average with sharp accelerations resulting in enormous pressure pulsations in the discharge line where small pipes and high velocity are necessary due to the high pressure In the suction line, where only atmospheric pressure is used to force the liquid through the line, larger pipes are necessary thereby reducing the velocity and helping to minimize the pressure pulsations. On a discharge line, dampening air chambers are needed to make operation practical. With a three-cylinder conventional double-acting pump, these flow variations reduce to approximately six and one-half per cent (6½%) above, to eighteen and one-half percent (18½%) below the average flow, but still with sharp accelerations and still requiring air chambers. Due to these flow variations the lower the number of cylinders, the slower must be the speed of the pump to keep the resulting pressure variations within practical limits.

Another important speed limitation is caused by the necessity of keeping the piston accelerations down to values low enough to allow the liquid from the suction line to follow it on the suction or intake stroke. Where only atmospheric pressure is available in the suction line to provide acceleration of the liquid coming therefrom, which must work against the suction lift and friction losses through the line and the suction valve, very little pressure remains from atmospheric pressure on the suction line for accelerating the liquid to follow the piston. The result is that the liquid does not follow the piston and does not keep the enlarging cylinder entirely filled with liquid and cavitation results in the cylinder and causes very destructive impacts when the piston makes its return stroke. These impacts frequently wreck the pump and to avoid them slow speeds are very essential. Where the liquid is heavy and viscous very slow speeds become paramount.

It is accordingly, an object of my invention to overcome the above and other defects in liquid pumps and it is more particularly an object of

my invention to provide a liquid pump which is economical in cost, economical in operation, efficient in operation and economical in manufacture.

Another object of my invention is to provide a liquid pump which overcomes the speed limitations heretofore necessary in conventional liquid pumps and to provide a pump where very much increased speeds may be used.

Another object of my invention is to provide a liquid pump which is much smaller and lighter than conventional pumps designed for equal or greater liquid capacity and horsepower.

Another object of my invention is to provide a liquid pump which has perfectly even suction and discharge line flows by eliminating all pulsations and using only two double-acting or four single-acting cylinders.

Another object of my invention is to provide a liquid pump which has a minimum of moving parts.

Another object of my invention is to provide a liquid pump which has a smoother valve action without impacts or slamming of the valves and which also permits the use of higher valve speeds.

Another object of my invention is to provide ample pressures in the suction line when the piston is on the suction stroke using a pump for chiefly accelerating the liquid to follow the piston and keep the enlarging cylinder entirely full of liquid at higher speeds and to prevent cavitation in the liquid cylinder and destructive impacts.

Another object of my invention is to provide in a liquid pump with a minimum number of cylinders, a continuous smooth flow which will result in smooth and constant torque resistance on the driving shaft and thus prevent whipping of the driving V-belts, flat belts, chains or couplings.

Another object of my invention is to provide a hydraulic pump which makes unnecessary the use of heavy fly wheels.

Another object of my invention is to eliminate sliding crossheads, guide bearings, oscillating wrist pin bearings, by using roller bearings throughout, and to further eliminate Scotch yokes and connecting rods.

Another object of my invention is to eliminate the conventional large gear and pinion drive necessitated by the slow rotating speed of conventional pumps.

Another object of my invention is to provide a liquid pump capable of high enough speeds to eliminate gears for increasing or reducing the speeds.

Other objects of my invention will become evident from the following detailed description, taken in conjunction with the accompanying drawings, in which:

Fig. 1 is a side elevational view of an illustrative embodiment of my novel fluid pump unit, arranged for belt drive from an engine or a motor and disposed on a skid for portable use.

Fig. 2 is an end elevational view, with parts broken away on line 1—1 of Fig. 1 taken from the fluid cylinder end of my novel fluid pump unit.

Fig. 3 is a plan view, with one of the covers on top of the casing removed.

Fig. 4 is a view taken on the line 2—2 of Fig. 3.

Fig. 5 is a view taken on the line 3—3 of Fig. 3.

Fig. 6 is a diagrammatic view showing three curves, two of these curves representing the combined flow rates throughout one revolution of conventional pumps, and the third curve showing the combined flow rate through one revolution of my novel pump.

Fig. 7 is a diagrammatic view showing in a lower curve the flow rate throughout one crank revolution of a double acting cylinder of a conventional connecting rod and crank type unit for a fluid pump and the upper curve showing the valve velocities corresponding to the flow rates represented by the lower curves.

Fig. 8 is a diagrammatic view, similar to and to be compared with Fig. 7, showing the characteristics of my novel pump unit.

Fig. 9 is a diagrammatic view of the novel cams used in my novel pump unit.

Referring now to the drawings, Figures 1 to 5 inclusive, show a crank case 1 and a liquid casing 8 mounted on skids 2. Side covers 3 and top covers 4 are provided on the crank case 1 for ready access to piston rod packing cases 5 and 6.

A main cover 7 is disposed on the top of the crank case 1. The fluid casing 8 is disposed longitudinally of the crank case 1 and has formed therein liquid pumping cylindrical portions 9 connecting closed chambers 10 and 11. Apertures 12 in the casing 8 have seats 13 for seating inverted conical-shaped valve heads 15, guided by stems 16. Covers 17 have depending cylindrical portions 18 with coil springs 19 therearound to urge the valve heads 15 into sealing relationship with seats 13. Covers 17 close apertures 20 in the discharge chest 21, in the upper part of casing 8. Discharge outlet 22 leads outwardly from chest 21.

A shaft 29 is journaled in a sleeve 30 and is transversely mounted below the casing 8 and has fixedly mounted thereon two impellers 31, one with right and the other with left hand vanes disposed in annular casings 33 on opposite ends of shaft 29 and to increase the pressure of fluid entering the inlet line 35 and passing through suction chambers 36 and through suction apertures 37 to the chambers 10 and 11. Only the right hand impeller 31 is shown in Fig. 2 for purposes of illustration. Since one impeller has right hand vanes and the other has left hand vanes, they may be exchanged for rotation in an opposite direction. Seats 40 are disposed in the suction apertures 37 to seat inverted conical-shaped suction valve heads 41 urged against the seats 40 by coil springs 42 surrounding depending cylindrical portions 43 depending from closing covers 44 disposed over apertures 45 in the casing 8.

The suction and discharge valves are of the same general construction. One impeller 31 and one booster pump casing 33 is provided for each pair of suction valves in chambers 10 and 11. The booster pump casings 33 are closed by covers 46 carrying packing boxes 47 with glands 48. Bearing brackets 38 are bolted to the covers 46 and carry roller bearings 49 and sealing rings 50. Shaft 29 is journaled in the bearings 49 and extends outwardly therefrom on both ends thereof. I have shown mounted on one end of the shaft 29 a fixedly mounted V-belt pulley 52 secured by nut 53 and I provide a cap 54 to cover the extending shaft 29 on the opposite end of the shaft 29. The impellers 31 are secured by nuts 39 against the sleeve 30. It will be evident that pulley 52 may be used and mounted on either end of the shaft 29.

Referring now particularly to Figs. 3, 4, and 5, crank case 1 has disposed in the sides thereof, flanged members 60 for rotatably mounting a transversely disposed hub member 61 having

longitudinally extending internal apertures 62 for receiving the tapered ends 63 of shafts 64. The shafts 64 are oil sealed by glands 71. Reduced ends 65 of the hub member 61 engage roller bearings 66 carried by the flange members 60. Two pair of eccentric hubs 67 and 68 are cast integral with hub member 61, those in each pair being concentric with each other and one pair being spaced 90° from the other. The throw of each eccentric is preferably made equal to one half the piston stroke. Around each eccentric is shrunk a hardened and ground cam ring 59.

Engaging each cam ring 59 are two follower roller tires 69, shrunk over and encircling roller bearings 80 mounted on transversely extended shafts 70 integral with crossheads 72.

Shafts 70 have reduced end portions 73 for engaging roller bearings 74 movable with outer tires 90 between longitudinally extending guide-ways 75 and 76, the former mounted on pads 77 in the base block 78 of the casing 1 and depending ribs 79 forming part of the cover 7 of the casing 1.

There are two transversely extending cross-head shafts 70 on each end of each cross-head 72 and there are also cross-heads about each pair of cams 59. Each pair of cross-heads 72 are spaced 180 degrees apart around the periphery of the cams 59 with their centers riding in the horizontal plane through the longitudinal center line of the shafts 64. Nuts 55 at either end of each tie-rod 56 are adjustable to maintain all four cam followers 69 in contact with their respective cams 59 with an initial load and tension thereon. Suitable shims (not shown) or any other suitable means may be provided for adjustment. Guideways 75 and 76 can be adjusted vertically with shims in order to adjust the clearances between the upper guide-way 76 and the lower guide-way 75 and the tires 69, so that tires 69 can roll on either the upper or the lower guide-ways 76 or 75 and clear the opposed guide-way. End covers 57 are provided on the crank case 1 for access to the nuts 55 securing the tie-rods 56. After the nuts 55 are removed, the cross-heads 72 can be lifted through the top of the crank case 1 by removing top cover 7.

The inner ends of the tie-rods 56 extend through apertures in T-shaped cross-head extensions 58 and the latter are secured to the tie-rods 56 by nuts 55. The other ends of the extensions 58 are threaded internally for threadably engaging the threaded ends of piston rods 51 with nuts 81. Splash plates 82 are secured between nuts 81 and cross-head extensions 58. The extensions 58 are oil sealed by packing boxes 6. The piston rods 51 have secured thereto in the cylindrical portions 9 of the casing 8 conventional piston heads 83 by the internally threaded end 84 of extensions 85 of the rods 51 serving as a tail rod of the same diameter as the piston rods 51 in order to give equal displacements of fluid from either side of the piston heads 83. On either side of the piston heads 83 are mounted conventional rubber packings 86 secured with follower plates 87 and spring rings 88.

The rubber packings 86 ride on hard liners 89 sealed with a conventional rubber gasket 91. The piston rods 51 and the tail rods 85 pass through duplicate stuffing boxes 5 and 92 with conventional duplicate rubber packings 93 and 94 and follower glands 95 and 96. The liners 89 are pressed into the cylindrical portions 9 of the casing 8. It will be evident that the two chambers 10 and 11 are duplicated in casing 8 and have duplicate liners, pistons, rods and packing boxes for each cylinder whose parallel disposition is shown in Figs. 2 and 3.

A pulley 100 is mounted on the shaft 64 and rotative force is transferred to the pulley 52, mounted on the shaft 29 by V-belts 101. The shafts 64 are driven from a main power source (not shown) such as a gas engine, diesel engine, steam engine, or electric motor. If the drive requires two engines, another duplicate pulley 102 shown in dot and dash lines may be mounted on the opposite side of the crank case 1 and may be driven by additional V-belts. It will be evident that any form of drive may be used between shaft 64 mounted in the crank case 1 and shaft 29 mounted in the casing 33.

In operation, the shaft 64 is rotated by a suitable power source, thereby rotating the pulley 100 mounted thereon, to drive the pulley 52 on the shaft 29, carrying the impellers 31. The rotation of the shafts 64 causes the followers 69 engaging the cam members 59 to reciprocate thereby causing the reciprocation of the piston rods 51, and the pistons 83 attached thereto. The suction and discharge valves 41 and 15 in the chambers 10 and 11 on each end of the cylindrical portions 9 of the casing 8 open and close in accordance with the pressure variations caused by movement of the pistons 83. There is preferably disposed one suction valve 41 and one discharge valve 15 in each chamber 10 and 11 on the end of each cylindrical portion 9 of the casing 8. The fluid entering the suction valves 41 is placed under pressure by the impellers 31, to follow the pistons 83 and to prevent cavitation when they move through the enlarging cylindrical portion 9 of the casing 8.

Suction valves 41 are opened by reason of reduced pressure in the chambers 10 and 11 and the pressure exerted by the pump impellers 31 receiving fluid from the inlet line 35. On the discharge stroke of the pistons 83, the discharge valves 15 are opened by pressure on their underside from the pistons 83, which closes the suction valves 41 and the discharge valves then communicate chambers 10 or 11 to the discharge chest 21 which is open to all four discharge valves 15 and leads to the discharge line 22.

The theory and advantages in the construction of my novel pump will become apparent from a study of the curves in Figs. 6, 7, and 8.

Referring now to Fig. 6, the lowermost curve 190 is the combined rate of discharge of two double-acting cylinders with 90° cranks of a conventional pump with the pistons driven by a conventional crank and connecting rods and without a tail-rod. The base of this curve is scaled with crank angles for one revolution of 360 degrees, and the vertical scale is the discharge rate in gallons per minute of twin cylinders 6¼" x 18" with 2⅜" piston rods and at 55 revolutions per minute, which is the conventional maximum speed for such a pump handling heavy liquids. The violent fluctuations in the flow, reach peaks of 26% above to 27% below the average. These exist in both the suction and discharge lines. This, as stated, is one reason why the speed of the pump is very limited, since such flow variations cause violent pressure variations, their values depending on the flow velocities in the suction and discharge lines, the length of these lines and the column of liquid to be accelerated and decelerated and the capacity of dampening air chambers, which are essential to dampen these pulsations. The variations are often quite severe and cause much vibration of the lines and

connections, and frequently severe liquid hammers are set up, often causing ruptures.

To help smooth out these pulsations, it is customary to increase the number of cylinders. The result is illustrated by the middle curve 191 in Fig. 6, which shows the combined flow rate of three double-acting cylinders of approximately a total capacity equal to that of the lower curve. These three cylinders would be 5½" x 12" stroke with 2" rods and with cranks spaced 120° apart and running at 70 revolutions per minute. The pulsations are somewhat reduced and more equal, but still rise 6½% above to 18% below the average flow with sharp accelerations.

With the use of my novel straight line 192 shown in the uppermost line in Fig. 6 is obtained. This line 192 indicates a constant steady uniform flow without pulsations. This is obtained with only two double-acting cylinders, which in this instance are 6¼" x 7", with 2¼" piston rods and tailrods. These cylinders, it will be noted, are smaller than either of the other two pumps, and the pistons can be run at much higher speeds (in this instance 250 R. P. M.), giving a much greater flow with a much smaller pump.

The reasons for this great advantage will become apparent by examination of the curves 193, 194, 195, 196, and 197, in Figs. 7 and 8. In each of these curves 193, 194, 195, 196, and 197, in Figs. 7 and 8, is shown the flow characteristics and the valve action for one double-acting cylinder throughout one revolution. In Fig. 7 is shown the characteristics of a conventional cylinder of the crank and connecting rod type. The lower curves 193, show the flow rate in gallons per minute on the vertical scale for one crank revolution of 360° indicated on the horizontal scale at the base. The flow is zero at either end of the stroke when the crank is on dead center, and reaches maximum at near mid-stroke. The maximum flow in the left lobe is smaller than the maximum flow in the right lobe, due to the piston rod reducing the displacement of the former.

It should also be noted that these lobes are not symmetrical about their vertical axes, which is due to the angularity of the connecting rod. Two of these cylinders at 90° give the lower curve 190 in Fig. 6, since they are of the same size and at the same speed. It should also be noted that the maximum flow on the left or smaller lobe is equal to the flow at the lowest points in the bottom of the deepest valleys in the lower curve 190 of Fig. 6. This is because the maxima on these lobes are lower than the average combined flow, and at these points the other cylinder at 90° on the crank is at dead center with zero flow.

With a conventional Scotch yoke and crank, instead of a connecting rod and crank, and with tail rod extensions equal in diameter to the piston rods, the two lobes would be equal and symmetrical sine curves, which would bring all the valleys to the same level on the combined curve, and at a flow rate of about 400 gallons per minute. This, however, would still be 18% below the combined average flow, and the peaks on a curve similar to curve 190 in Fig. 6 would be 16% above the average. The general characteristics would remain, although the variations would be somewhat less.

Three curves similar to curves 193 shown in Fig. 7, combined for three similar but smaller cylinders 5½" x 12" with 2" piston rods at 70 R. P. M., and with 120° crank spacings, result in the middle curve 191 of Fig. 6, and the same considerations apply, although the variations are further reduced. Thus, in general, the greater the number of cylinders, the smoother the combined flow curve. A smooth curve, however, is only obtained at the expense of a multiplicity of cylinders, and an absolutely straight line is never obtained. The inherent characteristic of each single cylinder still remains, regardless of the number of cylinders used.

We will now consider the valve action on a single conventional cylinder. The lower curves 193 in Fig. 7 not only represent the flow in gallons per minute shown by the G. P. M. vertical scale, but also the piston velocity to another suitable scale and also the valve lift to still another suitable scale. These valves are similar to those shown in Figs. 2 and 4 and are held against flow by a spring, which means that the lift or area of discharge through the valve will be approximately proportional to the flow rate of fluid discharge from or to the cylinder. Thus the lower curves 193 in Fig. 7 represent valve lift. The upper curve 194 in Fig. 7 represents the velocity of valve movement to the vertical scale shown which, when the valve is rising, is shown above the base line, and, when the valve is lowering, is shown below the base line. The valve velocity is the increment in lift divided by the time taken for this increment, and these to suitable scales are derived from the lower curves 193, representing lift vs. crank angles, of which the latter at constant speed represents time elements. A lift increment divided by the time increment gives the velocity of the valve as shown by the upper curve 194 in Fig. 7.

The important characteristic of the valve velocity curve, or upper curve 194, so derived in Fig. 7, shows a maximum upward velocity at the beginning of the lift, when the piston is on dead center, and a maximum downward velocity at seating on the opposite dead center. In the case of starting to lift, the valve is accelerated from a position of rest to its peak upward velocity, instantly, in nearly zero time, and, conversely, it is decelerated from its peak downward velocity, instantly, to rest in nearly zero time. These represent very heavy impacts, slamming the valve open and slamming it shut. The value of these impacts is in proportion to the square of the crank revolutions per minute, and thus again the speed is limited to avoid severe and destructive valve impacts. This is true of each cylinder and its valves, regardless of how many cylinders are used in the interest of smoothing out the combined flow curve. If a Scotch yoke is used, giving a symmetrical sine flow curve, the same applies, since the slope of the sine curve represents valve velocity and the slope of the sine curve is proportional to the cosine, which is maximum when the sine is zero. The only difference will be that the starting impacts will be equal to the seating impacts, and will be the average of the two from a connecting rod pump where they are unequal, as shown by the curve 194, in Fig. 7, the greater impact occurring when the piston is on head end dead center.

In my improved pump, with the pistons operated by properly designed cams, these serious defects are entirely eliminated and thus permit much higher speeds, giving not only a constant flow without pulsations, but also valve impacts are entirely eliminated.

Referring now to Fig. 8, in the lower portion, curves 195 represent the flow for one double-

acting cylinder throughout one revolution in heavy lines, and curves **196** the flow from the other cylinder at 90° in dashed lines. The chief characteristics of these curves are that they are symmetrical about vertical centerlines at each 90 degrees of cam rotation, and, at the same time, are symmetrical about a horizontal centerline at one-half the peak flows. The other important characteristic is that the curves come in tangent to the base line at zero flow, or, in other words, with zero slope. Two such curves **195** and **196** give a combined flow for two double-acting cylinders at 90° apart on the cam hubs, of a constant value, or perfectly smooth flow passing through the peak values or peak flows of each single cylinder. In other words, one cylinder, when at its peak, maintains the constant average flow, while the other is on dead center with zero flow. In the particular curves shown, the flow for each cylinder is equal to K (1—cos.2a) where "K" is a constant function of the stroke and the revolution per minute and "a" the cam angle of rotation from dead center. However, any other curves having the above characteristics will serve. Thus, a perfectly smooth flow is obtained from two double-acting cylinders with equal displacements at both ends, whereas with Scotch yokes or connecting rods the equivalent is not obtained regardless of how great a number of cylinders are used.

The valve impacts are also eliminated, as illustrated by the upper curve in Fig. 8. Here again, the flow curve **195** for one cylinder represents, to another suitable scale, the valve lift, and the increment in valve lift divided by the time used for that increment, the velocity of valve movement. This, to a suitable scale, is, as before, the slope of the flow, or lower curve. This slope, as required above, is zero at either dead center or at zero flow, so that the valve velocity of movement, as shown by the upper curve **197** is also zero at the beginning of the upward lift and at the end of the downward movement, or at seating. Thus the valve motion lifts and seats with no impacts. The peak valve velocities are at 45° cam angle instead of at dead center. Thus, my novel pump can be operated at much higher speeds without valve impacts, and gives a valve action similar to a mechanically operated cam actuated valve. In my novel pump the cams actuate the pistons which in turn actuate the valves.

The cams which actuates the pistons, are mounted on an eccentric having a throw of one-half the stroke, so that the curvature of the cam is easy and provides only the difference in piston movement derived from the eccentric and that desired to obtain the above characteristics.

Where, as illustrated in Fig. 8, a piston velocity, or flow curve, equal to K (1—cos.2a) is used, the piston position at various cam rotation angles will be the integral of this curve, or

$$K\left(a-\frac{\text{sine } 2a}{2}\right)$$

where "a" is the cam rotation angle from dead center expressed in radians. From this the shape of the cam is readily derived by conventional methods.

Fig. 9 shows a cam developed by using the above formulae, namely, piston velocity equals formulae, namely, piston velocity equals K (1—cos.2a) and the stroke equals

$$K\left(a-\frac{\text{sine } 2a}{2}\right)$$

It will be noted that the peak valve velocities are at 45° cam angle. It will further be noted that the eccentric **61** has a throw equal to one-half of the stroke. The cam **59** is so designed to give flow curves which are symmetrical about vertical axes at each 90° of rotation and at the same time are symmetrical about a horizontal axis through one-half of the peak flows and the bottom of the curve is tangent to the base line. This cam is designed to eliminate valve impacts and to give smooth, constant, continued flow.

There is one other characteristic of liquid pumps which limit their speed, and that is to accelerate the liquid on the suction stroke so that it will follow the piston and keep the cylinder entirely full of liquid without cavitation. If this is not insured and cavitation results, the piston will receive terriffic impacts when it closes in on the liquid on the return stroke. The pressure in the suction line is all that is available to accelerate the liquid through the line and passages and through the suction valve, and must still have sufficient pressure left to keep accelerating the liquid to the speed of the piston. Particularly where heavy viscous liquids are pumped and where high suction lifts are used, the speed of the pump is very limited, with only atmospheric pressure on the pump to supply this acceleration. Even without these limitations, the pump speed is limited, due to the low pressure of atmosphere on the suction line, and this is further aggravated by the sharp variations in the flow of a conventional pump, as already illustrated, requiring sharp accelerations, for which atmospheric pressure is not sufficient. The cam **59** is preferably developed as shown in Fig. 9 by extending radii outwardly from the center of the shaft **62** at equal angles of 15 degrees, thereby making 24 divisions of the outer circle. The position of the piston at each 15 degree rotation of the shaft **62** is determined by utilizing the curves shown in Fig. 8 with the formula

$$K\left(a-\frac{\text{sine } 2a}{2}\right)$$

where a is the cam rotation angle from dead center expressed in radians. By measuring the particular piston position on each radii extending outwardly from the center of the shaft **62**, the pitch line of the cam is found which I have designated as "path of follower center." Arcs are described on each of the radii of the same radius as the cam followers **69** thereby establishing the necessary points to define the outer periphery of the cam **59**. The legends shown in Fig. 9 are self-explanatory.

These limitations are eliminated in my novel pump by supplying a booster in the suction line in the form of a centrifugal pump built integral with the main pump and driven by the main pump drive as illustrated in the Figs. 1, 2, 3, and 4. The speed of this pump and the impeller diameter are proportioned to give the necessary pressure at the suction valves to accelerate the liquid being pumped, so that it will more than cover the piston acceleration after deducting friction losses and thus keep the cylinder safely full of liquid at the higher speeds. Since the piston acceleration is proportional to the square of the pump revolutions per minute, and since the pressure developed by the centrifugal pump is also proportional to the square of its revolutions per minute, these will maintain the proper ratio when arranged with a drive as shown in Figs. 1, 2 and 3.

It will be evident from the foregoing that my novel pump unit eliminates speed limitations

heretofore necessary in conventional pumps, which prevents cavitation, and which eliminates all large gears and pinions and other moving parts in conventional pumps. It is further evident that I have provided a novel pump which has a smooth constant flow in the suction and discharge lines, which has a smooth and constant torque resistance in the cam hubs thereby avoiding the usual slipping and vibration occuring where V-belts, chains, and couplings, are used on conventional pumps and one in which speed may be greatly increased thereby greatly decreasing the size, weight and cost of the pump.

My novel fluid pump unit may take many different forms such as a unit with fluid cylinders on opposite sides of the crankcase, without departing from the basic design and principles of my invention.

Various changes may be made in the specific embodiment of my invention without departing from the spirit thereof or from the scope of the appended claims.

What I claim is:

1. A fluid pump unit comprising a casing having a reciprocating pump chamber with cylindrical portions in one section thereof, a shaft journalled in another section thereof, an inlet passageway leading to said reciprocating pump chamber and an outlet passageway leading therefrom; centrifugal pump chambers having a portion defining a passage leading to said inlet passageway in said casing, impellers of a centrifugal pump fixedly mounted on said shaft disposed in said centrifugal pump chambers, a pulley on said shaft disposed externally of said casing, a second shaft disposed substantially parallel to said first shaft and paced therefrom, offset cams movable with said second shaft, a pulley on said second shaft, means for rotatively engaging said pulleys, means for rotating said second mentioned shaft, rods reciprocated by said cams, pistons operable in the cylindrical portions of said recpirocating pump chamber in said casing reciprocated by said rods, and suction and discharge valves in the inlet passageway and outlet passageway of said casing, respectively, cooperating with said pistons, said impellers increasing the pressure of the fluid entered by said suction valves.

2. A fluid pump unit as set forth in claim 1 wherein said impellers are disposed in said centrifugal pump chambers on opposite ends of said shaft and they have oppositely disposed vanes.

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