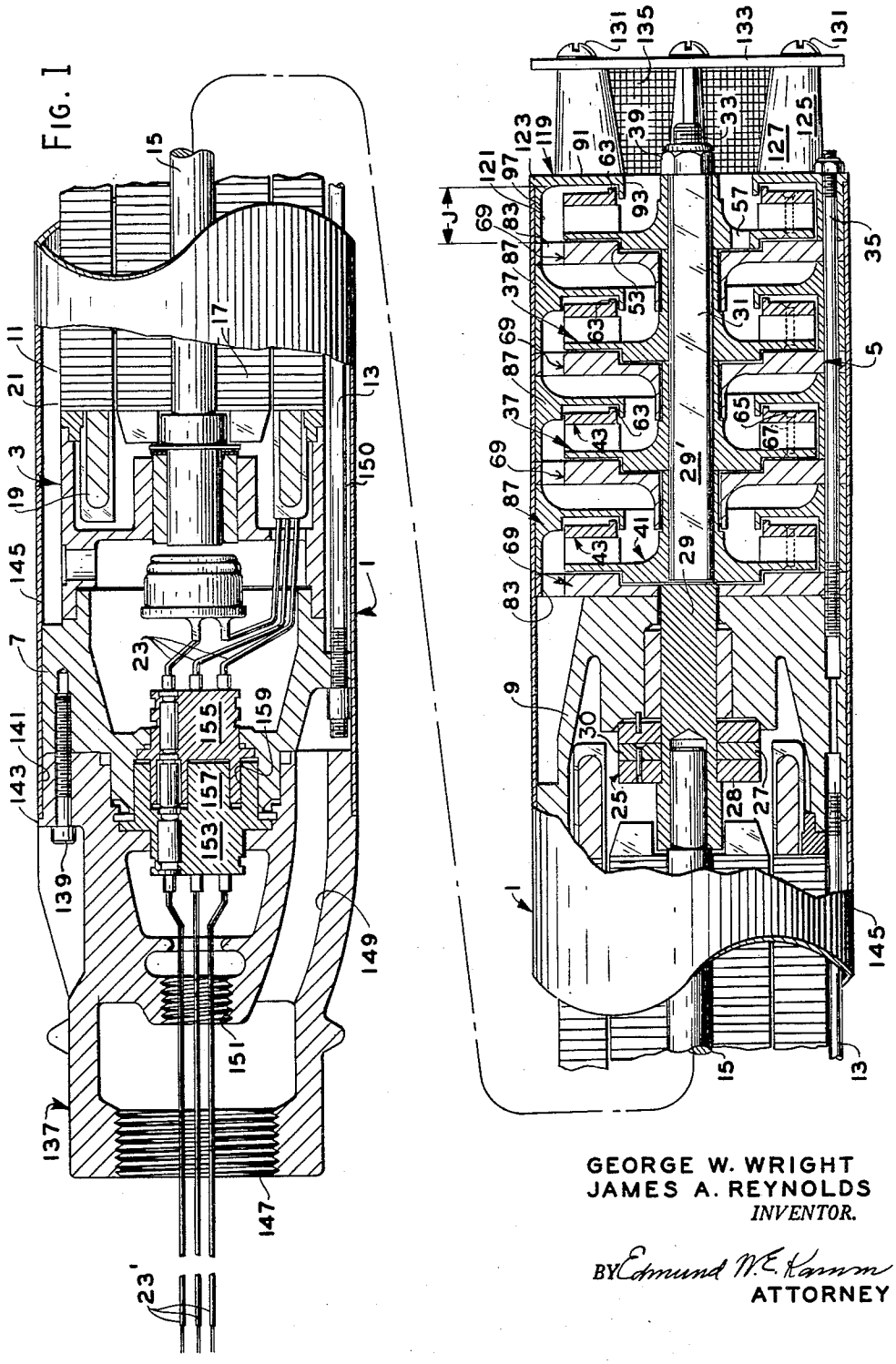
Nov. 22, 1960 G. W. WRIGHT ET AL 2,960,937

SUBMERSIBLE PUMP

Filed Dec. 17, 1957 3 Sheets-Sheet 1

FIG. 1



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Filed Dec. 17, 1957 3 Sheets-Sheet 2

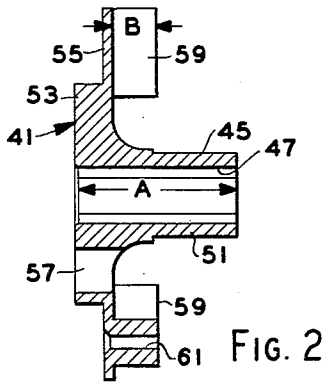


FIG. 2

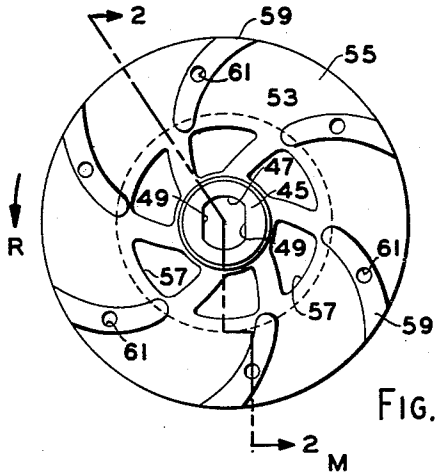


FIG. 3

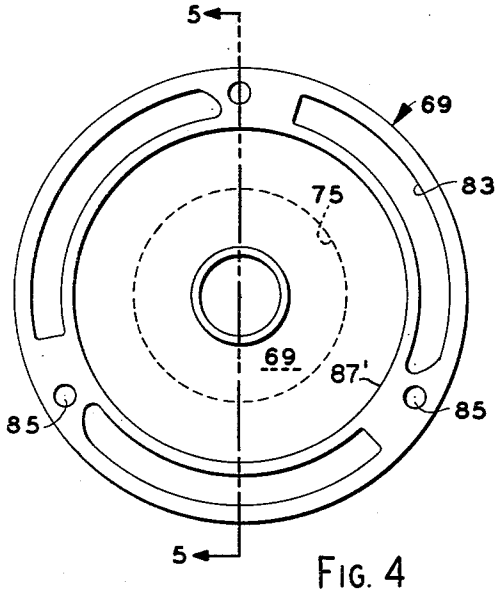


FIG. 4

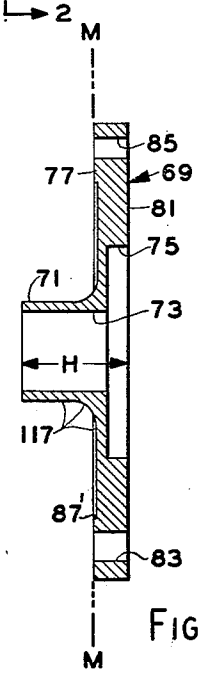


FIG. 5

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Filed Dec. 17, 1957 3 Sheets-Sheet 3

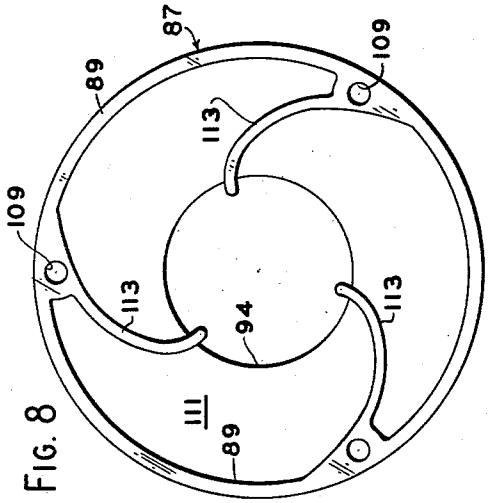


FIG. 8

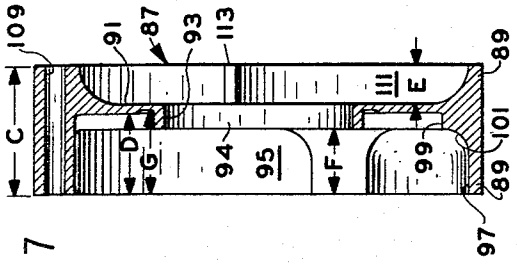


FIG. 7

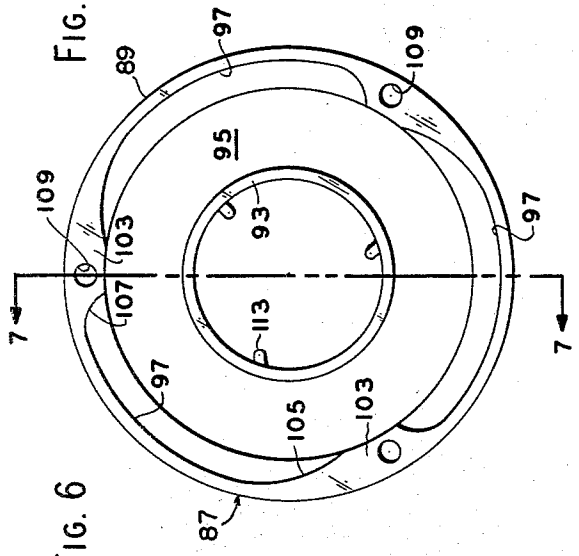


FIG. 6

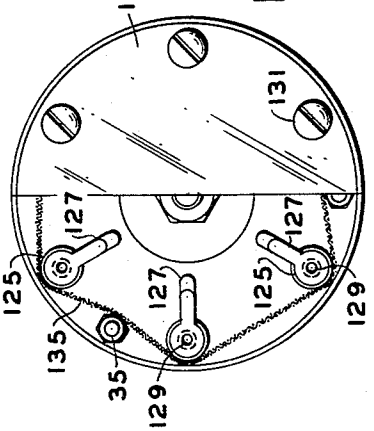


FIG. 9

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United States Patent Office 2,960,937 Patented Nov. 22, 1960

2,960,937

SUBMERSIBLE PUMP

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Filed Dec. 17, 1957, Ser. No. 703,358

2 Claims. (Cl. 103—87)

This invention relates to a submersible motor-pump unit such as that used for pumping gasoline from an underground tank to a number of dispensers in an automobile filling station.

The economics of filling station operation require pumps having different maximum rates of delivery but the maximum pressures at which liquid is delivered is substantially the same.

The storage tanks vary in size and capacity and the openings through which the motor pump is inserted vary from 3½ inch pipe diameter upward. Usually the larger tanks have greater than 3½ inch pipe openings. Consequently it has been the practice to design the larger capacity pumps with diameters which would be accepted by larger than 3½ inch pipe openings to secure the required flow rates and pressures.

Such practice encounters several obstacles. The first is that large tankage may be obtained by a number of small tanks interconnected by siphons, hence it may be necessary to insert a large capacity pump into a small tank which will have a 3½ inch opening.

Secondly, to supply all of the varying flow capacity demands, pumps of various diameters will be required. This necessitates patterns and tool equipment for each size and reduces the economy of parts procurement, manufacture, stock maintenance, etc.

Further, the driving motor diameter usually follows the pump diameter so that different dies are required for the rotor and stator laminations of the matching motors.

In motor-pump units of this general type, substantial difficulties are encountered in providing suitable bearings to support the thrust developed by the pumps.

This invention contemplates a construction in which the external diameter of both the motor and pump is held to that which will enter the 3½ inch pipe opening regardless of the discharge capacity of the motor pump unit and in which the same pump castings and motor laminations are used for all pump capacities and motor horsepowers. The capacity of the pump is varied by varying certain special dimensions of the finished parts and removing the amount of metal from the castings to obtain such dimensions. Other universal dimensions are constant for the corresponding parts for all of the pumps. The capacity of the motor is varied by varying the length of the rotor and stator lamination stacks of the motor and hence the length of the windings. Further, the thrust produced by the pump is minimized by hydraulic thrust balancing means so as to minimize the load on the thrust bearing which in turn materially reduces the cost thereof.

Further advantages of the invention will become apparent from a study of this specification and the drawings which are attached hereto, made a part hereof and in which:

Figure 1 is an elevation of the motor pump unit with parts broken away to show the structure thereof.

Figure 2 is a sectional view of the impeller body taken substantially on the line 2—2 of Fig. 3.

Figure 3 is an end elevation of the impeller body viewed from the right of Figure 2.

Figure 4 is an elevation of a port plate.

Figure 5 is a sectional view of the port plate taken substantially on the line 5—5 of Figure 4.

Figure 6 is an elevation of the pump case showing the impeller chamber as viewed from the left of Figure 7.

Figure 7 is a sectional view of the pump case taken substantially on the line 7—7 of Figure 6.

Figure 8 is an elevation of the diffusion chamber as viewed from the right of Figure 7.

Figure 9 is an end elevation of the motor-pump unit viewed from the right of Figure 1.

Referring to Figure 1, numeral 1 represents the motor-pump unit generally, which comprises the motor 3 and pump 5.

The motor comprises an upper end shield 7, a lower end shield or member 9 and stator laminations 11 which are held together by tie rods 13. It includes a rotor shaft 15 mounted in suitable bearings in the end shields. Rotor laminations 17 are mounted on the shaft. Power supply leads 23 are connected with the stator windings 19. A pump shaft 29 is shrink fitted to the lower end of motor shaft 15.

A thrust bearing 25, which may be of any suitable construction, preferably includes a metal disc 27 having a conical end mounted in a conical seat on the lower end shield. The disc is held against rotation but may nutate slightly. A metal disc 28 and a carbon disc 30 are fixed together and are mounted on the pump shaft and are held against movement along the axis thereof away from disc 27 by a shoulder on the shaft. Discs 28, 30 rotate with the shaft but due to the mounting of disc 27, the carbon disc will always lie in parallel contact with the metal disc 27.

The motor shaft has a portion 29' of reduced diameter which extends beyond the lower end shield, in preferably flattened on opposite sides as at 31 and terminates in a threaded projection 33.

The pump is made up of a set of pump parts comprising a number of axially stacked stationary elements which are mounted on the lower end shield by means of the studs 35, and of a number of impellers 37 which are mounted on the shaft portion 29 by means of the lock nut 39 mounted on the threaded projection 33.

If we first examine the impellers (Figures 1, 2 and 3) it will be seen that each comprises a body 41 and a cover 43. The body has an axially directed hub 45 provided with an axial opening 47 having flats on diametrically opposite sides at 49 so as to be driven thereby from the pump shaft. The right hand portion 51 of the hub is slightly reduced in diameter while the other end of the hub is formed as a shroud which comprises a relatively thick radial flange 53 from which extends a further, relatively thin, radial flange 55.

The thick flange 53 is provided with a number of axially directed pressure balancing ports 57, preferably the same in number as the spiral vanes or blades 59 which extend axially and radially outwardly from the right side of the thin vane from a point slightly inwardly of the shoulder formed at the juncture of the flanges 53, 55 and extend outwardly therefrom to the edge of the thin flange 55. The shoulder forms a sealing ring.

As shown in Figure 3, the vanes are arranged so that their outer ends trail the inner ends as the rotor is operated in the direction of rotation which is indicated by the arrow R. Also the inner end of each vane leads slightly, the advancing edge of the immediately adjacent port 57.

Each vane defines a rivet passage 61 which is parallel to the impeller axis and which is countersunk where it opens through flange 55.

As is shown in Figure 1, an impeller inlet cover or shroud 43 is a flat, annular disc having substantially the same outside diameter as the flange 55 and an inside diameter which is slightly less than outside diameter of flange 53 and is provided with a short, axially extending flange 63 to serve as a sealing ring. The cover is provided with countersunk, axially extending rivet holes 65 and is fastened against the axial ends of the blades 59 by the rivets 67.

The rate of flow capacity of the pump varies with the axial dimension of the blades, other conditions being held constant, and is greater for blades of greater length.

To secure the required pressure of 60 cycle motors of similar speeds, four of the impellers described above are arranged in endwise abutting relation on the projection 29. For other pressure requirements, more or fewer impellers may, of course, be used.

Referring now to the stationary pump parts shown in Figure 1 and starting adjacent the end shield 9, a port plate 69 similar in all respects to that shown in Figures 4 and 5, except that it has no axially extending hub 71, is mounted in direct contact with the end shield and with its central, circular cavity 75 facing away from the end shield. The plate is made from a casting which has a hub 71 which is cut off and machined along the dot-dash line M—M of Figure 5.

As shown in Figures 4 and 5, the plate has a central opening 73 which has a running fit with the motor shaft 15.

A number of segmental, axially directed ports 83 are formed in the plate 69 adjacent its outer periphery. These ports are preferably three in number, are equally spaced end to end and the lands between the ports are perforated axially at 85 to receive the studs 35.

The other face of the plate is provided with a shallow, annular recess 87' which extends from a line spaced inwardly from the inner edges of the ports 83 to an inner diameter represented by the cut off hub 71. The axially facing surfaces which bound the ports are parallel and smooth so as to establish sealing contact with the adjacent parts with which they are in endwise abutment.

It should be noted that the cavity 75 of the port plate and the flange or sealing ring 53 of the impeller body are so proportioned that the latter fits within the cavity with small radial clearance so that, in effect, the flange is a piston acting in the cavity as a cylinder. Liquid pressure, applied to this piston through the ports 57, will balance that applied to the corresponding area on the opposite side of the impeller body so as to balance the thrust of the impeller on the shaft.

The flange 53 and cavity 75 also serve as sealing means to impede the flow of liquid from the discharge ends of the vanes toward the cavity 75 to minimize the circulation of high pressure fluid. The flange 55 is, however, subjected to pressures which are balanced by corresponding pressures on the impeller cover 43.

As shown in Figure 1, the stationary element which abuts the orifice plate just described is a pump casing 87. This element (Figs. 6, 7 and 8) has a cylindrical outer wall 89 and a radially inwardly extending flange 91 which terminates in a cylindrical seal ring 93 which extends axially toward and enters telescopically within the seal ring 63 on the impeller cover with which it has a running fit, to impede the circulation of high pressure liquid from the discharge ends of the vanes to the inlet. The flange 91 is located between the central radial plane and the end of the casing which is remote from the orifice plate described above. The casing thus defines an axially open impeller chamber 95 at one end and an axially open diffusion chamber 111 at the other end.

As shown in Figure 6, the inner portion of wall 89 of the impeller chamber is cut away in three places to form

three equally spaced, similar flow channels 97 for the fluid discharged by the impeller. These channels do not extend to the full depth of the chamber but terminate in a cylindrical recess 99, which is slightly deeper than the length of the seal ring 63 on the impeller cover. The recess is adapted to receive at least a part of the impeller cover so that fluid issuing from the impeller will strike the curved surface 101 of the channel and be deflected upwardly toward the ports 83 of the associated port plate which closes the chamber.

The remaining portions 103 of the wall 89 serve as cut offs or stops with which the impeller has running clearance, and which minimize the flow of liquid from one channel to the next.

The channels 97 are preferably formed with curved ends 105, 107 in order to minimize turbulence in the fluid flowing into and out of the channels.

The portions of the wall forming the stops 103 are perforated axially at 109 to receive the studs 35.

The cavity 111 is defined by the wall 89 on the opposite side of the flange 91 from the pump chamber and a number of curved vanes 113 extend from the wall, inwardly, to the opening 94 defined by the seal ring 93. The ends of the vanes preferably extend substantially radially out over the opening 94 defined by ring 93 a slight distance. The vanes are preferably three in number and serve to reduce swirl and to direct liquid flowing up through the ports 83 of the subjacent port plate to the eye or inlet of the impeller just described, by way of the opening 94 defined by the seal ring 93.

The next fixed element which abuts the end of the diffusion chamber end of the pump casing 87 just described is another port plate 69 which is formed as shown in Figures 4 and 5 and which includes the hub 71. The hub has a smooth curved contour at 117 to aid in minimizing turbulence in the fluid flowing to the eye of the impeller. The impeller hub has a similar contour as shown in Figures 1 and 2 for reducing turbulence in the fluid which is moving to the impeller vanes. The hub 71 has a running fit with the reduced portion 51 of the impeller hub to minimize the flow of liquid from one stage to an adjacent one.

Two additional sets of casings 87 and port plates 69 are stacked end to end and the stack terminates in the pump cover 119 which defines an impeller chamber 121 (Fig. 1) which is in all respects similar to the chamber 95 in the other pump casings 87 and includes a seal ring similar to 93. The cover has a radially outwardly extending flange 123 disposed substantially coplanar with the flange 91 and has no diffusion chamber.

Instead, a number of substantially cylindrical posts 125 (Figs. 1 and 9) extend axially from the flange away from chamber 121, are disposed in circular spaced relation and have radially inwardly directed fluid guides 127 which minimize the swirling tendency of fluid entering the adjacent impeller.

The posts are axially drilled and tapped at 129 to receive screws 131 which hold a bottom plate 133 in place on the ends of the posts. A strainer 135 which is generally hexagonal in cross-section and is preferably of wire cloth, is held in place between the pump cover and bottom plate and is supported against radial collapse by the posts.

A discharge fitting 137 (Fig. 1) is attached to the motor end shield 7 by means of screws 139. The fitting has an axial flange 141 and a radial flange 143. A tubular sleeve 145 fits closely around the flange 141, the end shields 7 and 9, the port plates, the pump casings and a portion of the pump cover and abuts the flange 123 of the latter. Thus all of these parts are held in close alignment with each other by the sleeve. The studs 35 project through the pump cover and the nuts thereon serve to draw all of the parts together in axially abutting relation and hold them clamped together as a rigid unit.

The fitting 137 has a threaded outlet 147 which re-

ceives the fluid discharge pipe (not shown) and which communicates through fluid channels **149** with the space **150** between the sleeve **145** and the motor laminations **11**. The ports **83** of the first described port plate **69** communicate with the same space.

The fitting is also internally threaded at **151** to receive a conduit (not shown) for housing the leads **23'** which are connected with an electric connector socket element **153**. A connector plug **155** is mounted in the end shield **7** and is connected with the field windings by leads **23**. The plug and socket are preferably keyed by an interfitting projection **157** and recess **159** so that they can be assembled only when they are in the proper relative positions with respect to each other.

ASSEMBLY

Assuming that the finished parts for a particular model of motor-pump unit are available, the usual procedure is to assemble the various parts of the motor. The stators have the windings encapsulated. Thereafter the sleeve **145** and studs **35** are installed and the proper stationary pump parts and impellers are assembled in the sleeve on studs **35** and on the pump end of the shaft in the proper order. The fasteners are applied to the studs. The strainer **135** and plate **133** are placed and screws **131** are drawn down and the pump is conpleted.

OPERATION

When the motor **3** is energized, all of the impellers **41—43** will be rotated in a counterclockwise direction when viewed from the pump end of the unit. Fluid will flow inwardly through the strainer **135**, and then upwardly through the openings defined by seal rings **93** and **63** into the first or lowermost impeller which will throw the fluid substantially radially outwardly into the various channels or passages **97** which are formed in the impeller chamber of the pump cover **119**.

These passages direct the fluid flow in a substantially axial direction to and through the ports **83** of the adjacent port plate and into the diffusion cavity **111** of the next pump casing **87** which again directs the flow in substantially radial inward direction. The vanes **113** of the diffuser minimize swirl and turbulence in the streams issuing from the ports **83** so that they may properly enter the inlet of the second stage impeller.

It will be seen from Figure 1 that the contours of the port plate hub and the impeller hub are such that the stream of fluid as again turned axially and thereafter radially outwardly in a manner to minimize turbulence while at the same time avoiding restriction of the flow passage.

The fluid moves through the remaining stages in the same manner described above and is finally discharged through the ports **83** of the last port plate **69** into the space **150** between the motor **3** and sleeve **145**, to the channels **149** and the outlet **147** of the discharge fitting.

In all of the stages, the hydraulic axially acting forces on the impeller flange **53**, which are unbalanced in the usual designs of such pumps, are substantially balanced because the pressure is equalized through the ports **57**. Since the impeller cover **43** and the flange **55** of the impeller are of substantially equal radial areas and since both are subjected to equal and opposing pressures, the axially acting hydraulic forces on these portions of each impeller are also substantially equalized.

Since the motor pump unit is normally mounted in an upright position with the pump at the bottom, at least some of the weight of the motor rotor, impellers and shaft is unbalanced by hydraulic forces and there remains a downward thrust which is supported by the thrust bearing **25**. However, a major reduction in this thrust has been achieved by the substantial balancing of the hydraulic pressures as explained above. A reduction of from 115 pounds at 25 p.s.i. in a pump not having the balancing arrangement to 53 pounds at 30 p.s.i. in a pump having this feature has been found by actual comparative tests.

FLEXIBILITY

As noted at the outset, it is possible and commercially practical to manufacture at least three pumps having different discharge capacities and substantially the same cutoff pressures and six motors from the same castings and laminations.

The pump specifications are as follows:

Model	H.P.	Rated Capacity	Cutoff Pressure, p.s.i.
55E	¾	60 g.p.m. at 6 p.s.i.	31
72E	½	40 g.p.m. at 7 p.s.i.	27.5
70E	⅓	30 g.p.m. at 6.5 p.s.i.	25.5

Motor specifications:

	⅓ H.P.	½ H.P.	¾ H.P.
Cycles	50, 60	50, 60	50, 60.
Voltages	230, 208 or 230	230, 208 or 230	230, 208 or 230.

It is of course possible to build pumps of different capacities and cut off pressures with the same castings and motors for additional current specifications and horsepowers using the same castings and laminations. Those set forth in the tabulations above however, represent a range of motor-pump units which satisfy the requirements for the usual filling station installations.

Additional delivery and cut off characteristics can be attained in the manner indicated below and by adding or subtracting stages. For example, the pumps operated by 50 cycle motors as indicated above require five stages instead of four because of the inherent lower speed of the 50 cycle motor.

To illustrate the ease with which any of the castings used to make up the pump may be machined to make any of the pump models mentioned above.

The following are tabulations of the "special" dimensions to which the castings of the corresponding pump parts must be finished to produce sets of parts for the three groups of pumps described above. All of the other dimensions of the corresponding parts for all groups are constant and are referred to as "universal" dimensions.

Impeller body (Fig. 2)

Model	Impeller	Dim. A	Dim. B
55E	Lower	.838	.347
	3 additional	1.243	.347
72E	Lower	.562	.148
	3 additional	.970	.148
70E	Lower	.562	.101
	3 additional	.865	.101

Port plate (Fig. 5)

Model	Plate	Dim. H (Hub Length)
All	Upper	.243
55E	3 additional	.774
72E	do	.501
70E	do	.396

Pump casing (Fig. 7)

Model	Dimensions				
	C	D	E	F	G
55E	.999	.625	.313	.502	.630
72E	.726	.425	.239	.303	.401
70E	.621	.379	.181	.256	.384

Pump cover (Fig. 1)

Model	Dimensions			
	D	F	G	J
55E	Same as above	Same as above		.592
72E	do	do		.393
70E	do	do		.346

Obviously the shafts **15** and **29** for the motor and pump units will be made in six different lengths and the reduced portions **29'** thereof will be of six different lengths to accommodate the various stacks of rotor laminations and to accommodate the total impeller hub lengths for each of the six groups of pumps. All other dimensions of these shafts will remain the same. Similarly the sleeves **145** will have to be cut to the proper lengths for the different groups or models of pumps. The thrust and other bearings are the same for all groups.

The invention makes it possible to effect important economies. Only one set of pattern and die equipment, tools, jigs, fixtures etc. are required to manufacture the pump and motor parts for all of the pumps.

Large runs of castings, bearings and stampings may be ordered because the corresponding parts of all of the pumps and motors are alike in this state and the orders covering all of the models can be combined. However, the total number of castings, bearings and stampings required to be stocked is materially reduced over that which would be required if the corresponding parts were different for the different models because the parts can be used for any of the models.

Further, all of the castings may be manufactured to the universal dimensions and stored in an intermediate stock. This makes it possible to keep loaded, machines which might otherwise be idle and assists in providing work so as to avoid laying off workmen. Upon the release of orders for the final manufacture of the various models of the motor pump units, the partly finished parts may be withdrawn from intermediate stock and groups of sets of parts for the various models may then be finished to the special dimensions for such groups. At the same time the required shafts and outer sleeves can be machined to match the various sets of parts. These are made from standard lengths of rod and tubing stock which too can be purchased in relatively large quantities because they are all the same diameter for all of the models. Again the total stock required to be stored is minimized.

Since only a few special dimensions need to be machined to completely finish the sets of pump parts and since the shafts and sleeves involve only a few, simple finishing operations to complete them, the numbers of the motor pump units for the various groups ordered can be quickly completed for shipment, thus minimizing the time required to complete the manufacturing process once the manufacturing order is released.

The manufacturing process is thus greatly facilitated and is highly flexible because shortages of parts for building a particular model, which often result from an unexpected demand for such model, are eliminated. The amount of capital tied up in inventory is reduced and the turnover of such capital is increased. Both machine loading and employment are greatly stabilized.

All of the above advantages and economies are in addition to the great advantage that, whether the tankage in a filling station be made up of large, small or mixed tanks, a pump of the desired capacity may be inserted in any of them to meet the capacity requirements of the dispensing system which it is to supply.

It is obvious that various changes may be made in the form, structure and arrangement of parts of the specific embodiments of the invention disclosed herein for purposes of illustration, without departing from the spirit of the invention. Accordingly, applicants do not desire to be limited to such specific embodiments but desire protection falling fairly within the scope of the appended claims.

We claim:

1. A centrifugal pump comprising a shaft having a longitudinal axis, an initial and additional pumping stages disposed in axially spaced relation therealong, each stage comprising a shrouded impeller, a case having a chamber for receiving said impeller, and comprising a transverse wall defining a central inlet and a peripheral wall defining a number of axially directed discharge channels, means including a port plate mounted to close said chamber, said plate defining ports communicating with said channels, said impeller and transverse wall having first axially interfitting sealing means surrounding said inlet, said impeller and port plate having second axially interfitting sealing means disposed at substantially the same radius as said first sealing means, said impeller shroud adjacent said port plate defining pressure balancing openings disposed within said second sealing means, the case of each additional stage having said peripheral wall extended axially beyond said transverse wall, away from said chamber to define with said wall and the port plate of the preceding stage, a diffusion chamber, a number of vanes disposed between the inlet and said peripheral wall, between said transverse wall and said last mentioned port plate and between the ends of adjacent ports thereof for guiding the discharge from each such port to said inlet, a motor on said shaft having an end shield, a tube for housing said motor and end shield and extending beyond said end shield, said case peripheral walls and said port plates having external diameters to fit snugly in the extending portion of said tube for axial alignment thereby, and means mounted on said end shield for clamping the end shield, port plates and cases in sequential end to end contact.

2. A centrifugal pump comprising a shaft having a longitudinal axis, an initial and additional pumping stages disposed in axially spaced relation therealong, each stage comprising a shrouded impeller, a case having a chamber for receiving said impeller, and comprising a transverse wall defining a central inlet and a peripheral wall defining a number of axially directed discharge channels, means including a port plate mounted to close said chamber, said plate defining ports communicating with said channels, said impeller and transverse wall having first axially interfitting sealing means surrounding said inlet, said impeller and port plate having second axially interfitting sealing means disposed at substantially the same radius as said first sealing means, said impeller shroud adjacent said port plate defining pressure balancing openings disposed within said second sealing means, the case of each additional stage having said peripheral wall extended axially beyond said transverse wall, away from said chamber to define with said wall and the port plate of the preceding stage, a diffusion chamber, a number of axial vanes disposed on said case between the inlet and said peripheral wall, between said transverse wall and said last mentioned port plate and between the ends of adjacent ports thereof for guiding the discharge from each such port to said inlet, the impeller of each additional stage including a hub having a portion of reduced diameter which extends through the inlet for said stage and through the port plate of the preceding stage, into abutment with the impeller of said stage, said impellers be-

ing clamped between a shoulder on said shaft at one end and a nut on the shaft at the opposite end of said rotor, such port plate including a central axial flange extending into said inlet and telescopically and sealingly receiving said hub portion.

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