Sept. 14, 1965 W. E. RUPP 3,205,828

HIGH EFFICIENCY LOW SPECIFIC SPEED CONTRIFUGAL PUMP

Filed Aug. 23, 1963 4 Sheets-Sheet 1

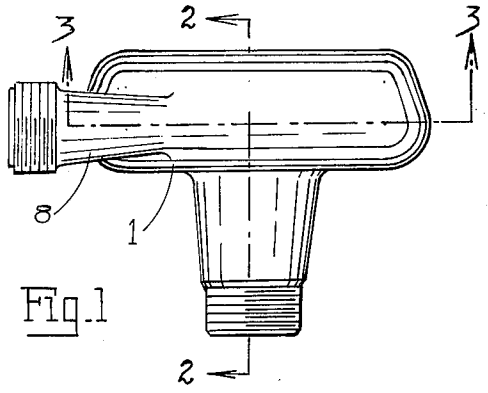


Fig. 1

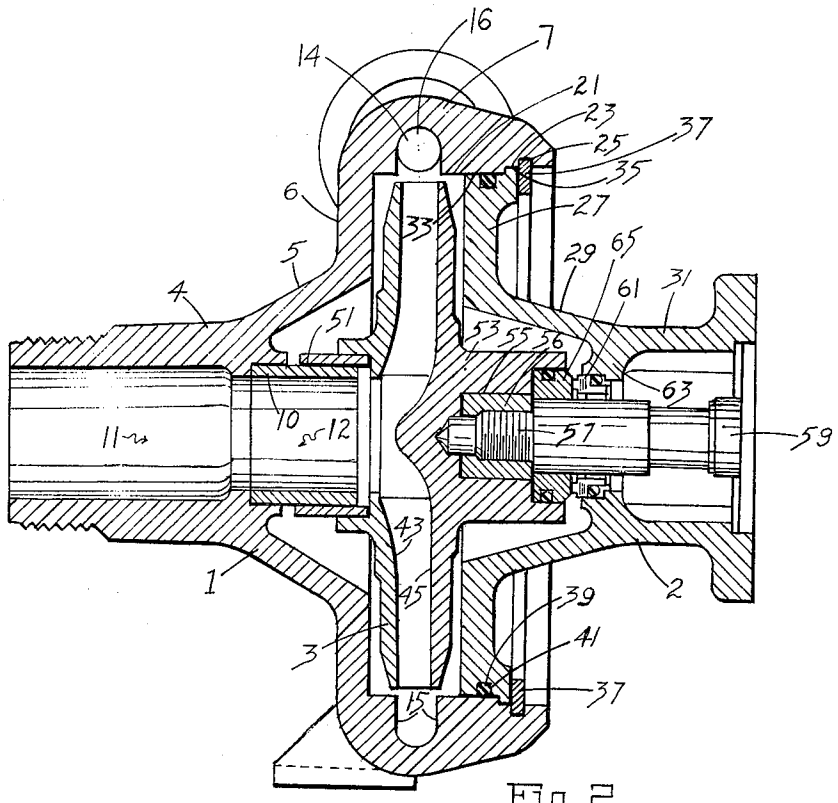


Fig. 2

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Filed Aug. 23, 1963 4 Sheets-Sheet 2

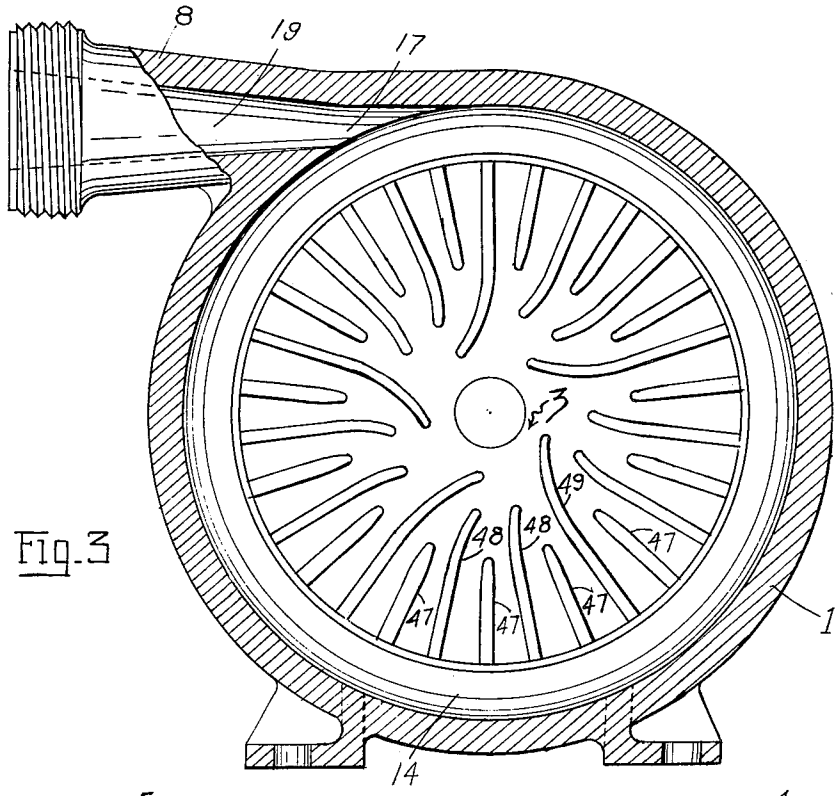


Fig. 3

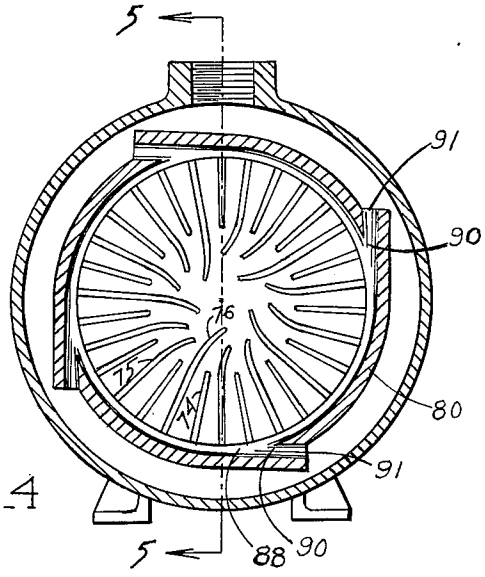


Fig. 4

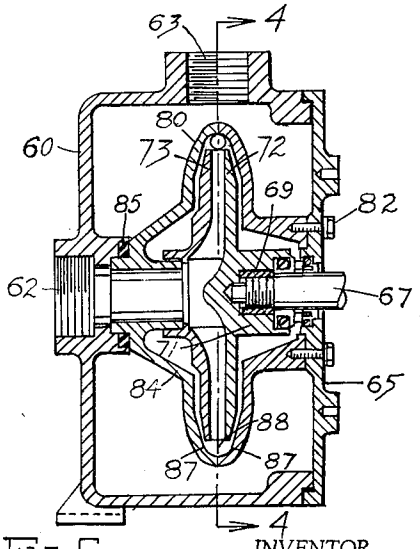


Fig. 5

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HIGH EFFICIENCY LOW SPECIFIC SPEED CONTRIFUGAL PUMP

Filed Aug. 23, 1963 4 Sheets-Sheet 3

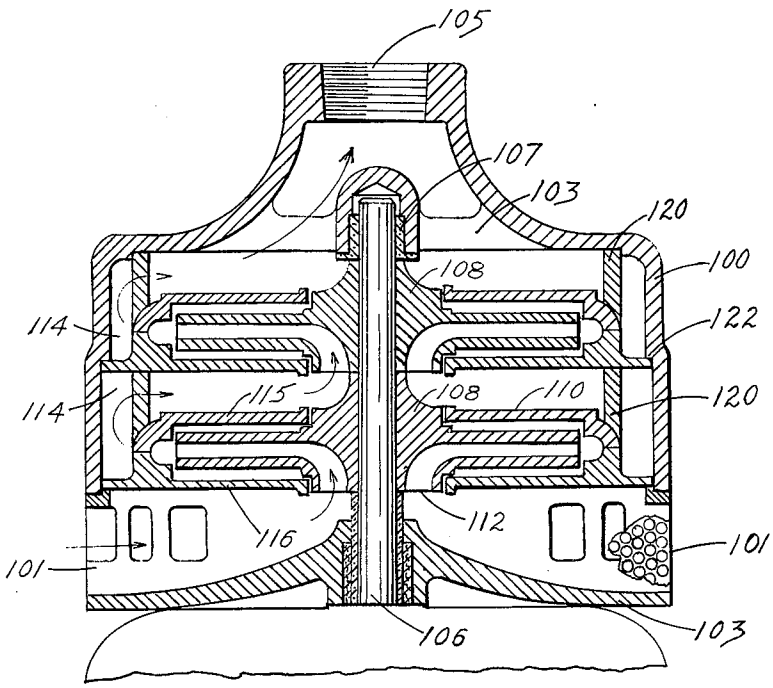


Fig. 6

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HIGH EFFICIENCY LOW SPECIFIC SPEED CONTRIFUGAL PUMP

Filed Aug. 23, 1963 4 Sheets-Sheet 4

IMPELLER - 30 VANES @ 90°
IMPELLER DISCHARGE AREA = .507 SQ. IN.
CASING THROAT AREA = .140 SQ. IN.
5¼" IMPELLER @ 4000 R.P.M.

MOST EFFICIENT POINT - 35 G.P.M. @ 184 FT. = 55%
RIM SPEED - 91.7 FT/SEC.
THEORETICAL HEAD V_2^2/g = 261 FT.
HEAD COEFFICIENT = .84
SPECIFIC SPEED = 475

Fig. 7

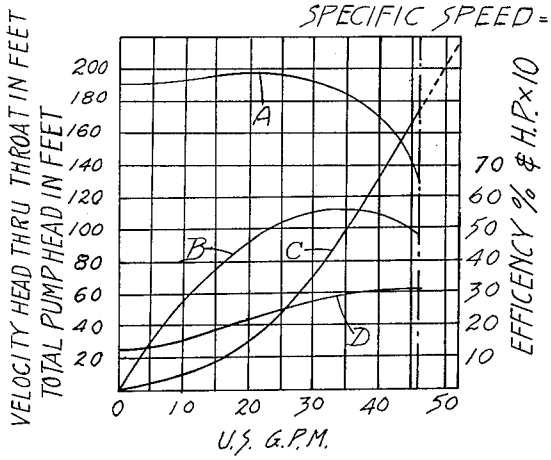


Fig. 8

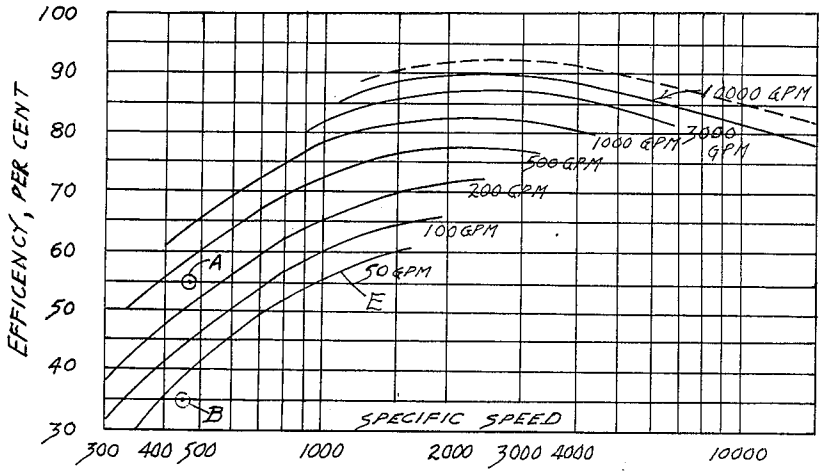
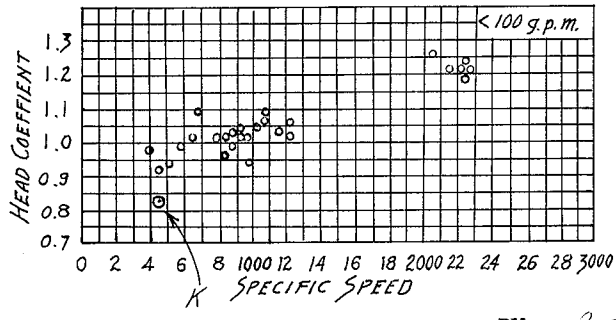


Fig. 9



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United States Patent Office 3,205,828 Patented Sept. 14, 1965

3,205,828

HIGH EFFICIENCY LOW SPECIFIC SPEED CENTRIFUGAL PUMP

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Filed Aug. 23, 1963, Ser. No. 304,072
3 Claims. (Cl. 103—103)

This invention relates to the art of centrifugal pumps and is particularly concerned with new, low specific speed, centrifugal pumps having high efficiencies and low head coefficients.

It has long been well known that relatively high specific speed centrifugal pumps have had efficiencies which are high enough for commercial acceptance but that low specific speed pumps have had efficiency so low as not to be acceptable commercially. As a result, it has been the general practice to use two or more high specific speed pumps in stages or series where a single low specific speed pump would serve if it could have operated at an acceptable efficiency. The present invention makes it possible to construct low specific speed pumps which are highly efficient and thereby to avoid the use of a plurality of the prior high specific speed pumps.

Moreover, pumps of the present invention may be used in multi-stage arrangement, for example in a small diameter casing in a wall where several stages of this invention will replace several times as many stages of the prior high specific pumps. Accordingly, the present invention satisfies the long existing need for an acceptably efficient low specific speed pump.

Briefly stated, these accomplishments have been achieved by observing several factors. One factor is that the areas of the flow passages in the impeller are greatly in excess of the area of the outlet passage or throat from the impeller chamber, for example, from at least ten times to as much as forty or more times the area of the outlet. Another factor is the use of the impeller discharge angle of above 27½° and preferably approximately 90° as contrasted with the much lower angle of the prior art.

Still another factor is the use of a multitude of vanes in the impeller which define substantially rectangular flow passages and are so shaped as to offer minimum impedance to the flow of fluid through those passages.

Still another factor is the formation of quite smooth surfaces in the flow passage in the casing and through the outlet and throat with resultant marked decrease in friction loss. A surface finish in the casing and throat on the order of less than about 200 microinches average is satisfactory.

In pumps embodying these several factors, the high head of the impeller which includes both velocity head and centrifugal or pressure head is converted into the maximum total output head of the pump with minimum friction loss, reduced impeller diameters and decreased power requirements with efficiencies in excess of those obtained heretofore, as far as I know.

The present invention also makes it possible to construct centrifugal pumps having those desirable characteristics and consisting of single stage or multiple stages.

The invention will be better understood by those skilled in the art from the following specification and the accompanying drawings, in which:

FIG. 1 is a top plan view of a pump embodying a preferred form of the present invention;

FIG. 2 is a sectional view taken on line 2—2 of FIG. 1;

FIG. 3 is a sectional view taken on line 3—3 of FIG. 1;

FIG. 4 is a view similar to FIG. 3 but showing the present invention embodied in a multi-diffuser casing;

FIG. 5 is a sectional view taken on line 5—5 of FIG. 4;

FIG. 6 is a view similar to FIG. 5 but showing a multi-stage pump embodying the present invention;

FIG. 7 is a chart showing the performance characteristics of a pump embodying the present invention;

FIG. 8 is a curve sheet showing the pump efficiencies at specific speeds of prior pumps and pumps embodying this invention; and,

FIG. 9 is a chart showing the head coefficients of many prior pumps of less than 100 g.p.m. capacity and the pump of FIG. 7.

In FIGS. 1, 2 and 3 is shown a preferred form of centrifugal pump embodying the present invention. The pump so illustrated includes an impeller casing 1, a rear closure plate 2 therefor and an impeller 3.

The casing 1 comprises a cylindrical wall 4, a conical wall 5 extending rearwardly therefrom to a transverse disc-like wall 6 which merges into a rearwardly extending flange 7. Wall 8 attached to flange 7 defines an outlet from the interior of the casing. Casing 1 carries a rearwardly extending sleeve 10. The axial passage 11 in wall 1 is coaxial with the passage 12 through sleeve 10, these passages constituting an inlet for fluid into the interior of the pump.

The inner surfaces of flange 7 are cylindrical and are variously shaped. Near the forward end of the flange, an annular groove 14 is formed in the flange and concentrically with part of that surface and is defined by substantially parallel surfaces 15 extending transverse to the axis of the pump and connected at their outer extremities by a semi-circular surface 16. As is better shown in FIG. 3, the wall 8 is provided with a tubular throat 17 extending tangentially from groove 14 and communicating coaxially with a nozzle 19 which is preferably conical in transverse cross section.

On the rear side of groove 14, flange 7 has an annular surface 21 which is cylindrical, a shoulder recess 23 and a snap ring recess 25 for purposes presently to appear.

The rear closure 2 of the pump includes a disc 27, a conical wall 29 and a cylindrical wall 31. The disc 27 has a cylindrical outer surface 33 to lie close to surface 21 of flange 7 and a flange 35 to seat in recess 23. A snap ring 37 seated in recess 25 bears against the rear face of disc 27 and serves to press shoulder 35 against the opposed surfaces of flange 7. The outer periphery of disc 27 is provided with an annular groove 39 in which an O-ring 41 is seated. This ring serves to prevent escape of fluid from the interior of the casing between flange 7 and disc 27. An annular web 63 extends inwardly from walls 29 and 31 of rear plate 2 and a fluid seal 61 is seated in this web.

The impeller 3 comprises front and rear shrouds 43 and 45 and a plurality of vanes 47, 48 and 49 (see FIG. 3).

Front shroud 43 has an axial passage coaxial with the passage through wall 4 and sleeve 10 to admit fluid to be pumped into contact with the vanes. Sleeve 51 is fixed to the front shroud 43 around the opening through the latter and surrounds sleeve 10 with close radial clearance to minimize the return of high pressure fluid from the outer portion of the impeller chamber into the intake of the impeller.

The rear shroud 45 is provided with a hub 53 which projects rearwardly and which has an axial recess 55

provided with a ring 56 secured therein. This ring may be composed of any metal or other material capable of serving as a suitable connection between the impeller and the impeller shaft. This ring 56 is fixed to the hub 53 preferably by being cast therein when the ring is composed of metal. Ring 56 is interiorly threaded to receive the threaded front end 57 of impeller shaft 59. This shaft extends through the back plate and is threaded into the ring 56. A balanced mechanical shaft seal surrounds shaft 59 and serves to prevent escape of fluid from the interior of the casing between the rear plate and the shaft. This mechanical seal comprises an annular stationary seal 61 which is supported in web 63 projecting inwardly from the cylindrical wall 31 of back plate 2; and a rotating ring 65 which is seated in the rear recess of hub 53 and is arranged to rotate with the shaft 57. The running engagement of the two seal rings constitutes the said fluid seal.

The outer peripheries of the impeller shrouds 43 and 45 are concentric with and lie close to the inner surfaces of flange 7 and the vanes between the shrouds are aligned with the groove 14. As is better shown in FIG. 3, the several vanes 47, 48 and 49 have their outer ends substantially flush with the outer peripheries of the shrouds and each vane extends inwardly radially for some distance. Vanes 47 are radial for their entire length, vanes 48 are radial for a distance comparable to the length of vanes 47, and are then curved forwardly in the direction of rotation of the impeller; and vanes 49 are radial for a distance similar to the length of vanes 47 and are then curved forwardly in the direction of rotation of the impeller to points beyond the inner ends of vanes 48. It will be noted that in each group of six vanes, there is one long vane 49, two intermediate length vanes 48 and three short vanes 47. This arrangement of numbers, sizes and shapes of vanes makes it possible to employ a large number of vanes which are radial at their outer ends without affording any substantial restriction on the flow of fluid from the eye of the impeller into the spaces between adjacent vanes.

The vanes are substantially equally spaced from one another at the periphery of the impeller and the circumferential length of the flow spaces between the adjacent pairs of vanes are substantially equal at their peripheries of the impeller. These flow spaces are also substantially equal as measured axially of the impeller. These flow spaces are substantially rectangular and preferably are approximately square at the periphery of the impeller.

It will be understood that other obvious arrangements of the vanes may be made and that their inner ends may be variously shaped as desired, but it is important to provide adequate flow space between the vanes from the eye of the impeller to its periphery.

It will be noted that since the outer ends of the vanes are disposed radially of the impeller, each one makes an angle of approximately 90° with a line tangent to the outer end of the vane. This radial positioning of the vanes together with the unusual width of the flow passages between the vanes and a substantially rectangular cross sectional area of these flow passages and also the relatively small cross sectional area of the outlet 17 from groove 14 results in the peripheral flow area of the impeller being many times, for example about 30 times, the area of the outlet from the pump. As a result of this combination of structural elements or factors, the throat 17 controls the maximum flow capacity of the pump in contrast to conventional pumps in which the capacity is determined by the size of the eye of the impeller. It is to be understood that the outer ends of the vanes need not make angles of 90° with a tangent to their outer ends but that results superior to those obtainable with conventional pumps having vanes at about 27½° may be realized at the various angles from 27½ to 90° but varying with the angularity.

An examination of FIGS. 2 and 3 will show that the impeller casing 1 is capable of being provided with extremely smooth surfaces with which the liquid comes into contact. The cylindrical surfaces of groove 14 may be machined and given a smooth finish as is also true of the cylindrical surfaces 21 of the flange. Similarly, the surfaces of the tube 17 and throat 19 in wall 8 may be made quite smooth. Surface finishes less than about 200 micro-inches are suitable. Since the friction created by fluid moving over metal surfaces varies directly with the roughness of the surfaces, it will be understood that the friction traceable to contact of flowing fluid with the extremely smooth surfaces just mentioned in the illustrated pump will be reduced in proportion to the reduction in roughness of those surfaces.

FIG. 7 shows the performance characteristics of a pump embodying the present invention and illustrated in FIGS. 1 to 3. The casing throat area is the cross sectional area of the outlet 17 of FIG. 3. Curve A shows the variation in total head with variations in capacity. As this curve indicates, the total head, which simply means the height to which the pump will elevate the liquid assuming no hindrance from friction, varied from about 190 to 200 with the increase in pumping capacity from 0 to 20 gallons per minute (g.p.m.) and then gradually decrease to about 130 at which point the pump was delivering 46 g.p.m. This was the maximum capacity of the pump and is sharply limited there because no more liquid could be forced through the throat. In conventional pumps the maximum capacity is not controlled by the throat and, hence, follows a gradually descending curve as contrasted with the vertical part of curve A.

Curve B on FIG. 7 shows the efficiency of the pump at different capacities with its maximum efficiency being about 55% at 35 g.p.m. and approximately 49% at the maximum capacity. Curve C shows the variation in velocity head at the throat (tubular passage 17, FIG. 3) with variations in the gallonage to a maximum of about 170 at the maximum capacity of 46 g.p.m. Curve D indicates the power input and shows that the power rises only slightly from the point of maximum efficiency to the point of maximum capacity.

FIG. 8 shows the optimum efficiences of conventional pumps at various specific speeds of from 50 to 15,000 and for capacities from 50 to 10,00 g.p.m. This chart is based on tests of hundreds of conventional pumps. Curve E shows the optimum efficiencies of conventional pumps of 50 g.p.m. and having speeds of from about 350 to about 1500.

Point B on FIG. 8 indicates the optimum efficiency to be expected from a conventional pump designed to deliver 35 g.p.m. at 184 feet head at 4000 r.p.m. Point A on FIG. 8 indicates the optimum efficiency actually realized from a pump embodying this invention and designed to deliver 35 g.p.m. at 184 feet head and 4000 r.p.m. Points A and B show that the maximum efficiency of the pump of FIGS. 1–3 and 7 was 55% as contrasted with 35% for a similar but conventional pump. This difference of 20% is startling and highly important. For the first time low specific speed pumps can be made to operate at commercially acceptable efficiencies.

In FIG. 9 the dot K indicates the head coefficient of the centrifugal pump whose performance is indicated on FIG. 7. It will be noted that its head coefficient is about 0.84 while the lowest coefficient for prior pumps of this size was over about 0.9.

Conventional pumps having low specific speeds require high heads combined with relatively low capacity and high peripheral speeds of the impeller. These high speeds require large impeller diameters and usually require small fluid passages. Since the friction loss, due to the spinning of a disc in a body of liquid, is, for a fixed speed, proportional to about the fifth power of the diameter, it is evident that the impeller diameter plays a most important

part in determining the power lost due to disc friction in a centrifugal pump. In other words, if, at a given shaft speed, the desired head could be attained by a smaller diameter impeller, than a substantial reduction in disc friction would result and the efficiency would be substantially increased.

By the present invention, I have been able to attain substantially the same head with the 5¼" diameter impeller pump, described above, as can be attained with a conventional pump having a 6" diameter impeller. This reduction of ¾", or about 8% in the diameter of the impeller accounts for the saving of about 50% in the horsepower required to overcome disc friction in a conventional pump.

It is well known that the head coefficient of a centrifugal pump is directly proportional to the impeller diameter. Hence, if the impeller diameter of conventional pumps can be reduced, as has been accomplished by this invention as stated above, a corresponding reduction in head coefficient will follow and a reduction in disc friction and an increase in efficiency will be obtained. The head coefficient of the pump above disclosed in which the impeller diameter was 5¼" was .84. This is in contrast with a head coefficient of .90 which is the lowest known to me for conventional pumps and which is shown in FIG. 9. That figure shows the head coefficient of a plurality of conventional centrifugal pumps having capacities of less than 100 g.p.m. The decrease in head coefficient from .90 to .84 is a decrease of 15%; and a decrease of 15% in impeller diameter amounts to a 55% decrease in disc friction. Hence, my invention has made it possible greatly to reduce the head coefficient and the disc friction and correspondingly to increase the overall pump efficiency.

FIGS. 4 and 5 show a modified form of a single stage pump embodying the present invention. In this modification a housing **60** is provided with a fluid inlet **62** and a fluid outlet **63**. The housing has a back plate **65** through which an impeller shaft **67** projects. The inner end of the shaft has screw threaded engagement in a ring **69** which is cast or molded in the hub **71** of an impeller. This impeller includes rear shroud **72**, front shroud **73** and a plurality of vanes **74**, **75** and **76** similar in shape, size and number to vanes **47**, **48** and **49** of FIG. 3. An impeller casing **80** surrounds the impeller within housing **60** and is attached to back plate **65**, as by cap screws **82**, and has a front portion **84** which projects into a cylindrical space in housing **60** around inlet **62**. Gasket **85** carried on the outer side of portion **84** is pressed against housing **60** and serves to prevent escape of fluid from within casing **60** back into the inlet **62**.

The casing **80** has a cylindrical surface **87** opposed to the outer periphery of the impeller and a groove **88** which has an open inner side between surface **87** and extends around the impeller and concentric therewith. This groove **88** has a plurality of, in this case, four, outlets **90** and each outlet communicates with a diffuser **91**.

It will be understood that when the impeller is being rotated counter-clockwise as seen in FIG. 4, fluid will be forced through each of the several outlets and diffusers and that the fluid so discharged from all the diffusers may flow out of casing **60** through opening **63**.

Casing **80** may be cast in a single piece, if desired, but preferably, it is split transversely through the groove **88** so that the surfaces **87** and the surface of groove **86** may be more readily made extremely smooth as by plastic molding or die casting to eliminate machining.

FIG. 6 shows a multistage pump embodying the present invention and including several pumps quite like that shown in FIGS. 4 and 5. In this modification, the parts are quite similar to those in FIGS. 4 and 5 and described above.

In transverse cross section, each stage of the pump of FIG. 6 would correspond closely with the transverse sectional view in FIG. 4. The axial view in FIG. 6 corresponds to that in FIG. 5.

In FIG. 6 the housing **100** corresponds generally with housing **60** of FIG. 5 but fluid is admitted radially through passages **101** in a rear or lower back plate **103**; and fluid is withdrawn from the housing through outlet **105**. A shaft **106** extends through the back or lower plate **103** and has a bearing therein and also is supported at its inner or upper end in bearing **107** which is supported by members **103** extending inwardly from the housing **100**. On this shaft **106** a plurality of impellers **108** is mounted and keyed thereto for rotation thereby. Each impeller **108** is quite like the impeller shown in FIGS. 5 and 6 and described above and each impeller rotates in a casing **110** which is generally quite similar to casing **80** of FIGS. 4 and 5. It will be understood, as indicated by the arrows, that fluid flows into the casing **100** through passages **101** and enters the first impeller through the opening in the ring-like shroud **112** and after passing between the vanes and into the circumferential groove is discharged through a plurality of outlets and throats into the space **114** around the impeller casing which is formed by members **115** and **116**. The fluid being pumped flows from the last of such spaces **114** through the outlet **105**. The rearmost or lowermost casing is positioned within the housing **100** by engaging the opposed surfaces of back or rear plate **103**. Adjacent casings are spaced from one another by struts **120**. The front or upper impeller casing is spaced from the adjacent end of the housing by similar struts **120** and may be additionally positioned by engagement of members **116** with shoulders **122**.

While FIG. 6 shows only a two-stage pump, it will be understood that any suitable and desired number of stages may be employed in a single housing if desired, each stage being substantially the same as those shown in FIG. 6.

Having thus described this invention in such full, clear, concise and exact terms as to enable any person skilled in the art to which it pertains to make and use the same, and having set forth the best mode contemplated of carrying out this invention, I state that the subject matter which I regard as being my invention is particularly pointed out and distinctly claimed in what is claimed, it being understood that equivalents or modifications of, or substitutions for, parts of the above specifically described embodiment of the invention may be made without departing from the scope of the invention as set forth in what is claimed.

What is claimed is:

1. A centrifugal pump comprising:

a casing and an impeller, the casing having an impeller chamber provided with a cylindrical outer surface, a cylindrical groove having substantial radial extent surrounding and substantially coaxial with said surface and communicating with the impeller chamber through an annular opening in said surface, at least one discharge throat extending from said groove and a diffuser nozzle communicating with said throat, said throat extending tangentially from said groove and substantially coaxially with said throat and nozzle, whereby fluid discharged into said groove will flow smoothly therein therearound and to and through the throat and its velocity will be converted into pressure in said nozzle,

said impeller being rotatably mounted in said chamber substantially coaxial with the said outer surface and with the said groove, said impeller having at least one shroud with its outer periphery lying adjacent to said outer surface of the impeller chamber, vanes attached to said shroud substantially equally spaced apart throughout the circumference of the shroud and extending inwardly from substantially its outer periphery, the outer ends of said vanes partly defining therebetween flow passages approximately equal in circumferential length and axial width and substantially aligned with said annular opening, the total area of the flow passages at the periphery of

the impeller being at least ten times the total throat area taken at right angles to the longitudinal axis of the throat at the intersection of the innermost part of the throat surface with the outer surface of the groove, whereby the volume of the fluid in the impeller will create and apply high velocity to the fluid in the groove and the velocity of that fluid will be converted into pressure in the diffuser.

2. The combination of elements set forth in claim 1 in which the casing is provided with a plurality of discharge throats and diffusers and in which the casing is enclosed in a housing, the areas of the flow passages at the periphery of the impeller being at least ten times the combined areas of the said plurality of throats.

3. The combination of elements set forth in claim 1 in which a plurality of said impellers is mounted on a single shaft and each of said impellers is surrounded by one of said casings, a housing enclosing said casings and means providing communication between said casings through said housings in series.

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