FIG. 1.

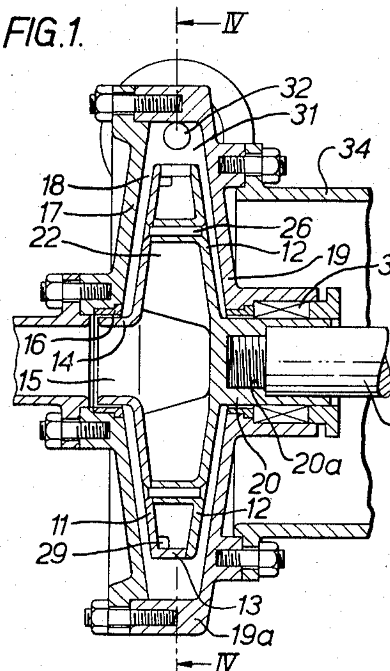

FIG. 2.

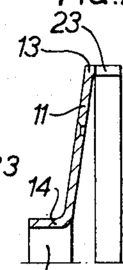

FIG. 3.

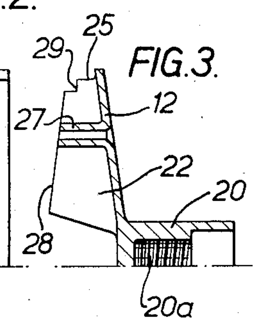

FIG. 4.

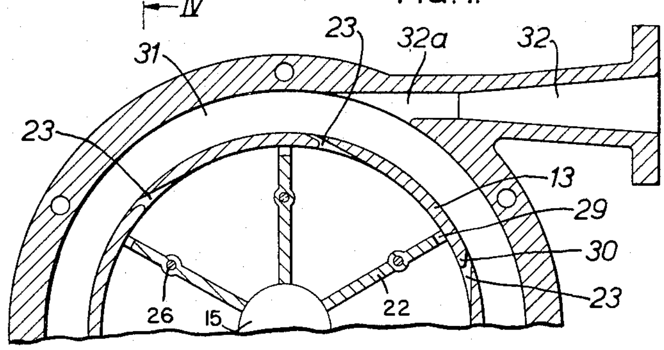

FIG. 4a

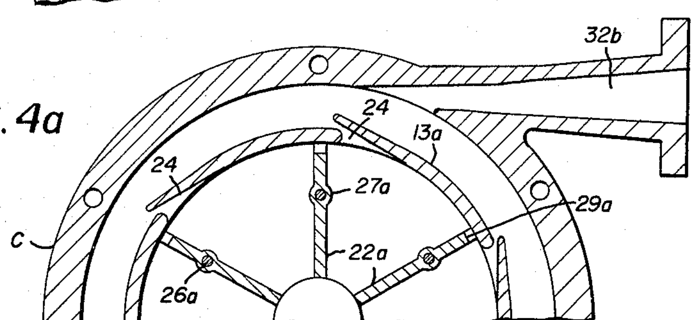

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FIG. 5

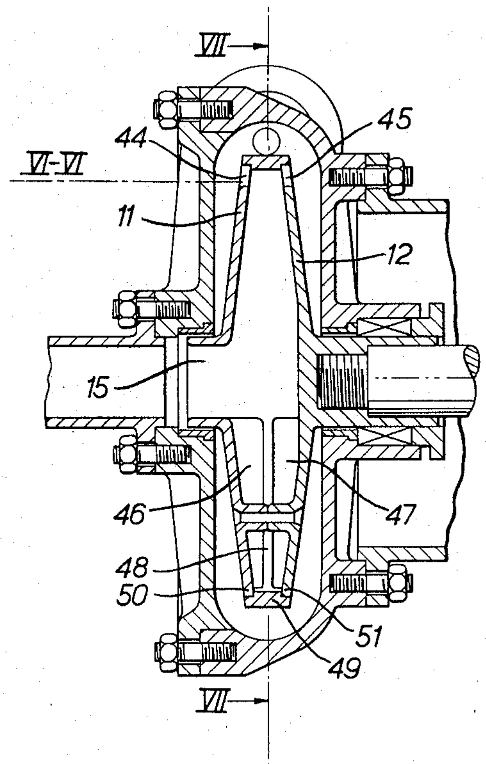
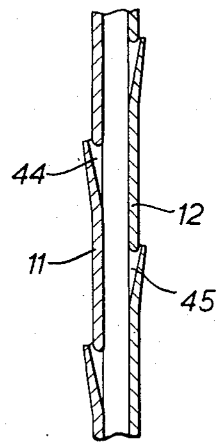


FIG. 6.



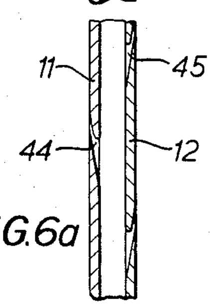


FIG. 6a

FIG. 7.

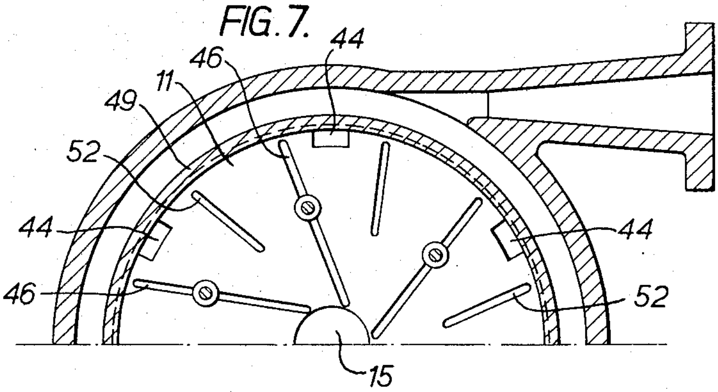


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FIG. 8.

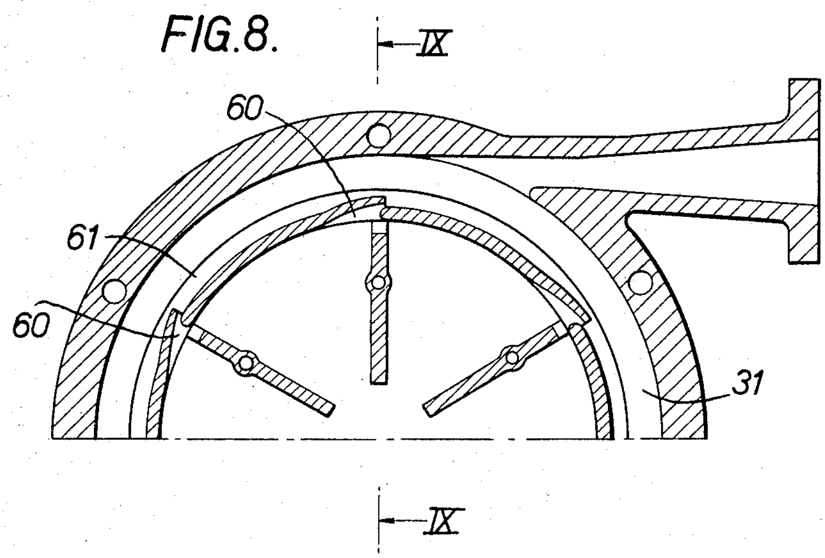

FIG. 9.

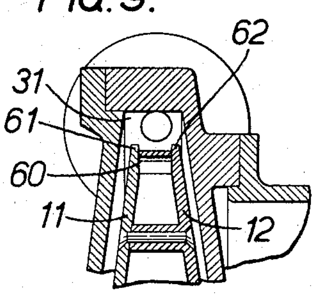

FIG. 10.

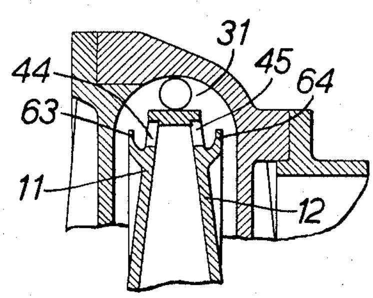

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FIG. 11.

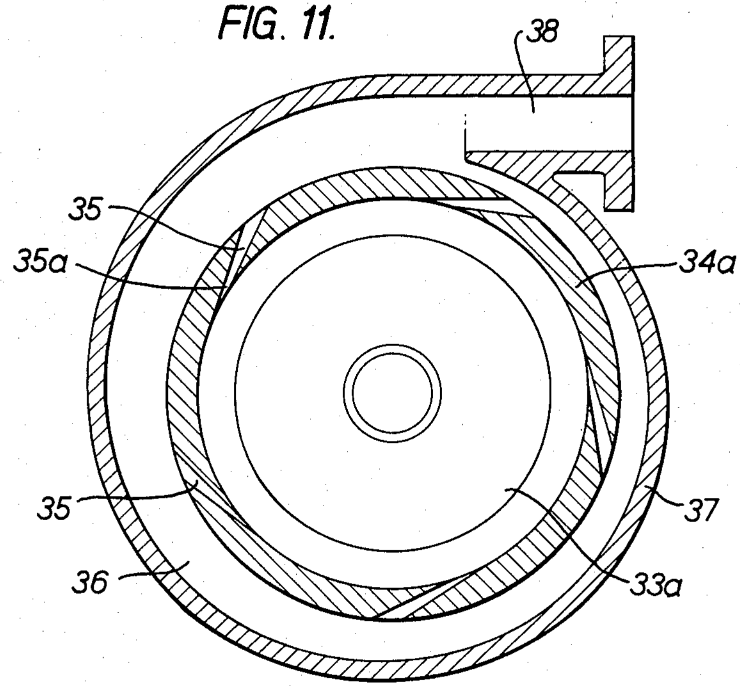

FIG. 12.

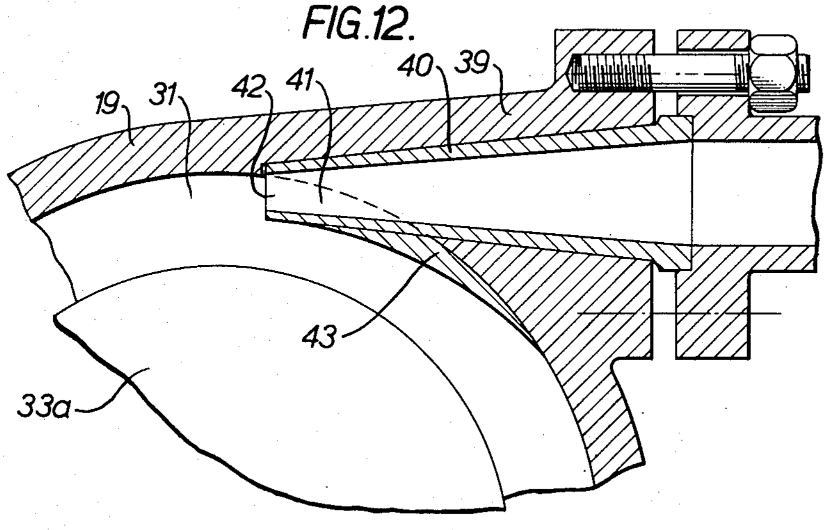

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FIG. 13.

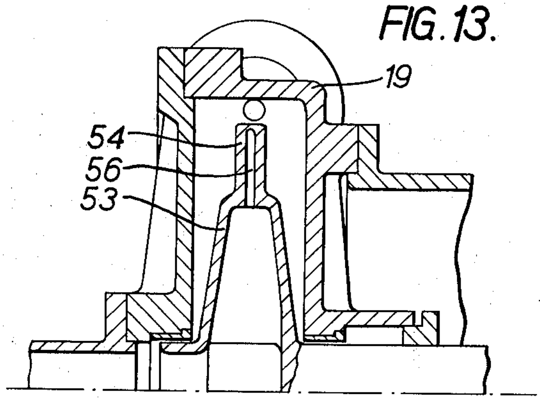


FIG. 14.

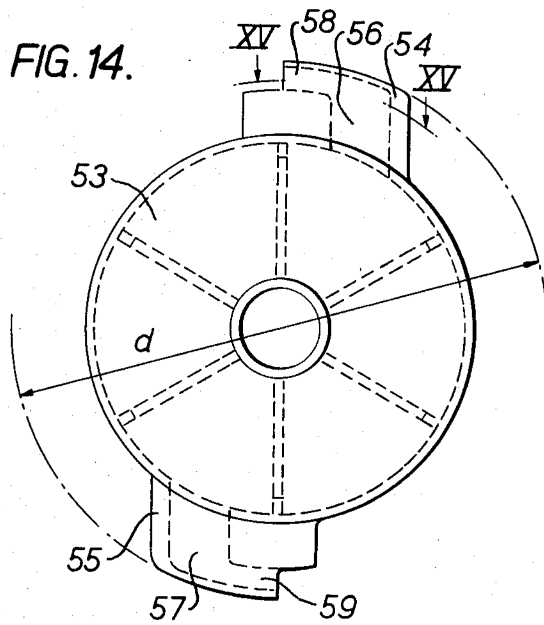
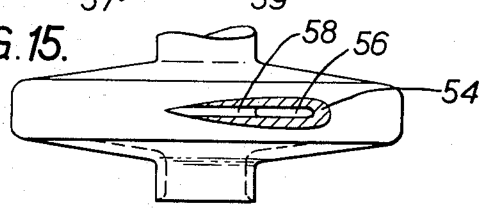


FIG. 15.



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United States Patent Office

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3,244,109
CENTRIFUGAL PUMPS

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Claims priority, application Germany, July 19, 1963, B 72,749
9 Claims. (Cl. 103—103)

The invention relates to centrifugal pumps adapted to operate at medium and low flow rates and relatively high delivery heads. Centrifugal pumps of conventional design are not suitable for such operating conditions which, according to well-known definition, are called "low specific speeds." The reason for this is that, under low flow conditions, no reasonably undisturbed flow can be maintained in the channels of the impeller and of the discharge diffuser or diffusers.

Other types of rotodynamic pumps as, for instance, side channel pumps, turbulence pumps, open impeller pumps and scoop wing pumps are somewhat better suitable for operating at low specific speeds but their efficiencies are satisfactory over a very narrow range of flow rates only and are dependent to a wide extent on maintaining close side clearances between the impeller blades and the pump casing, and the hydraulic output, particularly of the first and second types quoted above, decreases quickly as a result of wear of the mating surfaces of the impeller and of the casing. A scoop wing pump involving a rotating casing does not suffer from this disadvantage but an undesirable feature is the considerable weight of the rotating casing and it is often difficult to vent the casing properly during operation. Another limiting factor is that this type of pump is preferably suitable for very low flow rates only.

The centrifugal pump according to the invention yields considerably increased efficiences by permitting the following conditions to prevail.

(1) The relative flow velocities inside the impeller are always kept at a low level, and secondary vortices are practically avoided.

(2) A relatively undisturbed flow of the liquid discharged at the periphery of the impeller is maintained under all operating conditions, the direction of the impeller exit flow being in the tangential direction.

(3) The liquid flow in the space between the impeller and the casing remains undisturbed to a high degree.

(4) As the flow velocity of this liquid surrounding the impeller is, in the average, equal to the relative exit velocity of the impeller the skin friction between the impeller and this liquid will be considerably reduced. This is of particular importance when applying high peripheral velocities, e.g., such as are present in a liquid fuel pump operating at 18,000 r.p.m. at 30 atms.

(5) As a consequence of the undisturbed flow mentioned under (3) favourable flow conditions are promoted in the diffuser passage or passages leading to an improved conversion of kinetic energy into static pressure. This conversion may be further improved by an improved arrangement of the diffuser relative to the casing.

In order to obtain the above-mentioned improved flow conditions, the present invention comprises a centrifugal impeller pump with an axial inlet, the impeller being in the form of a circular vessel with an axial inlet eye and with smooth external surfaces and comprising internal vanes or blades, the peripheral region of the vessel being provided with an exit nozzle or nozzles so arranged as to cause the jet or jets discharged thereby to be directed tangentially or substantially tangentially to the periphery of the vessel, the pump casing also having smooth internal

walls and being provided with at least one peripheral diffuser for discharging the liquid.

The nozzle or a series of nozzles may be formed in the peripheral wall of the vessel or may be formed in one or each side wall of the vessel near to the periphery. The interior of the impeller may be provided with straight radial vanes.

The nozzles may be formed with their discharge ends situate in the circular external periphery of the peripheral wall of the impeller vessel or they may be constructed to open beyond the external periphery of the peripheral wall to the vessel to achieve a more nearly tangential discharge. They may be formed with their discharge ends in the outer faces of the side walls or they may be constructed to open beyond the said side faces.

The interior space of the impeller should be of ample width in the axial direction so that the radial velocity component of the liquid shall be low and commensurate with that of the axial inlet velocity and the space between the outer circular surface area of the impeller and the circular internal wall of the casing, which may be coaxial with the impeller vessel, should also be dimensioned so that the radial velocity component of the liquid discharged by the impeller is low and that a rotating liquid ring can be established in this space and the tangential flow velocity of this ring-shaped liquid body made to equal, in the average, the difference between the impeller peripheral velocity and the impeller nozzle relative exit velocity.

The vessel may taper radially outwards to a narrow cylindrical peripheral wall and in order to retain, as far as possible, a smooth circular shape of the impeller the exit nozzles are preferably formed as flat slots accommodated in the said wall through they may be formed in the outer margins of the end walls. In the latter case the circular outer periphery of the impeller need not be interrupted and this helps to reduce skin friction losses.

The nozzles may be directed backwardly, i.e., opposite to the impeller peripheral motion or they may be directed forwardly, i.e., they point in the direction of the latter. In the conventional terminology this means that the exit angles of the nozzles are approximately 180° or 0°, respectively.

The hydraulic output of the pump will be very different in both cases, of course, a remarkable feature being that forwardly directed nozzles yield a much higher delivery head than would be obtained with the same size impeller with backwardly directed nozzles.

In pumps for very low flow rates which may require, for instance, two discharge nozzles only, skin friction can be further reduced by using a circular vessel of a small diameter from the periphery of which the discharge nozzles project radially to the full diameter required for the delivery head and terminating in tangential delivery outlets, the exterior of the nozzles having a streamlined shape.

Some examples of the centrifugal pump according to the invention are shown in the attached drawings in which:

FIG. 1 is a longitudinal section of a pump,

FIGS. 2 and 3 show an example of the components of which the impeller can be made,

FIG. 4 is a cross section IV—IV of FIG. 1, showing in particular the arrangement of the nozzles at the impeller periphery,

FIG. 4*a* is a similar cross section showing modified nozzles,

FIG. 5 is a longitudinal section of another pump with a different impeller design where the nozzles are accommodated in the end walls of the impeller vessel,

FIG. 6 is a developed cylindrical section VI—VI of FIG. 5, showing two different shapes of nozzles,

FIG. 6a is a similar view to FIG. 6 shows a modified shape of nozzle,

FIG. 7 is a cross section VII—VII of FIG. 5,

FIG. 8 shows a half longitudinal section of a pump with forwardly directed discharge nozzles of the impeller,

FIG. 9 is a section IX—IX of FIG. 8 while

FIG. 10 is a similar section of a pump in which the forwardly directed nozzles are situated in the end walls of the impeller.

FIG. 11 shows a modified pump casing with 5 diffusers for the delivery, while in

FIG. 12 an improved diffuser design is shown in a larger scale,

FIG. 13 is a longitudinal section through the upper half of an impeller showing a circular section of a reduced diameter and a streamlined nozzle projecting beyond the periphery of this section, while

FIG. 14 shows an end view of the whole impeller, and

FIG. 15 a section XV—XV through the nozzle of FIG. 14.

As will be seen from FIGS. 1 and 4, the pump impeller is represented by a circular vessel consisting of the dish-like end walls **11**, **12** and a narrow cylindrical section **13**. The front wall or shroud **11** carries at its inner portion a short cylindrical section **14** which, on the one hand, represents the inlet eye **15** of the impeller while its external surface forms, together with a wearing ring **16** fixed in the cover **17** of the casing **19**, a labyrinth seal between the internal space **18** of the casing **17**, **19** and the axial inlet duct **15**. The back wall or shroud **12** is, by way of example, integral with the hub **20** which, in turn, is screwed on to the shaft **21** by a screw thread **20a**. The interior of the impeller is of an ample axial width to keep the radial velocity component sufficiently low and is provided with six radial ribs or vanes **22** while six exit nozzles **23** (see the upper half of FIG. 4 or 24, see the lower half of FIG. 4) are arranged in the peripheral wall **13**. Generally it will be advisable to have the impeller made of two open sections which will then be fixed to each other. This is shown, by way of example, in FIGS. 2 and 3 where the front wall **11**, the peripheral section **13** and the inlet piece **14** form one component while the back wall **12** with the vanes **22** and the hub **20** represent the second component to which the first one is fixed by rivets **26**, see FIGS. 1 and 4. The co-axial position of both parts is secured by the edges **25** of the vanes **22** which fit into the inside of the section **13**. The holes for the rivets **26** are preferably drilled through the vanes which are provided for this purpose with thicker portions **27**.

When assembling the impeller it is by no means required that the edges **28** of the vanes **22** fit tightly against the internal face of the wall **11** nor is it essential to have a fully leak-proof fit at the periphery as very low leakage at this portion would not affect the correct operation of the impeller. Recesses **29** are provided at the outer edges of the vanes **22** so that the internal spaces of the impeller formed by the vanes communicate with each other. This ensures a safe vent of the impeller when priming the pump and a full internal pressure balance during operation.

Because of the low relative velocities inside the impeller smooth internal surfaces are not needed but it is most important to provide smooth internal surfaces at the mouths of the nozzles in order to reduce friction losses at these points where the liquid is ejected at a high relative velocity. It is therefore advantageous that the nozzles which are accommodated in the section **13** (FIG. 2) be well accessible before assembly of the impeller. It will be seen that FIG. 4 shows two nozzles **23**. Nozzles **23** can be cut into the section **13** as narrow slots in which case the "nosepiece" **30** must be well rounded. The jets ejected by these nozzles will not be directed fully tangentially but it is a great advantage of this design that the external surface of **13** can be worked smoothly by turning. FIG. 4a is a view similar to FIG. 4 but showing nozzle **24** of a modified form. The nozzles **24** are more difficult to machine but they will give a tangential or more nearly tangential direction to the jets. Also easy to machine are the internal surfaces of the pump casing **19** and the cover **17** as the peripheral wall **19a** is circular and co-axial with the impeller. It is most important to have these surfaces and the exterior of the impeller machined as smoothly as possible in order to reduce skin friction losses of the impeller and of the liquid rotating in the spaces **18** and **31**. This liquid will be discharged into the pressure line by means of a conical (divergent) diffuser **32** arranged tangentially, in well-known way, at the peripheral wall **19a** of the casing.

In FIG. 4a, the parts designated **31a**, **22a**, **26a**, **27a**, **29a** and **32b** represent respectively the peripheral section of the vessels, the blades, the rivets, the thicker portions and the recesses corresponding to those described with reference to FIG. 4.

A gland **33** or other suitable sealing means is arranged in the usual way between the hub **20** and the pump casing **19**. The latter is fixed to a bearing housing **34** or the like of which only a small section is shown.

The example shown in FIGS. 5 and 7 includes an impeller the nozzles **44**, **45** of which are located at the margins of its end walls **11**, **12**. The arrangement of the nozzles is shown in more detail in the two sections of FIG. 6, the shapes of the nozzles **44** and **45** corresponding to those of the nozzles **23** and **24**, respectively, of FIG. 4. The two components forming the impeller are modified insofar as each end wall **11**, **12** carries one half of the blades, i.e., the end wall **11** carries the blade sections **46** and the end wall **12** carries the blade sections **47** the blade sections **46**, **47** being opposed to one another virtually to form single blades, but being separated by narrow slots **48** left between the adjacent edges of these half blades **46**, **47**. The cylindrical periphery of the impeller is formed by a smooth ring **49** the recesses **50**, **51** of which accommodate the outer edges of the walls **11**, **12**. It will also be seen from FIG. 7 that, in addition to the vanes **46**, **47** extending from the impeller periphery to the inlet eye **15** radially shorter ribs **52** which may extend across the width of the impeller vessel are provided which help to reduce secondary vortices still further.

Having shown embodiments of the invention it will be useful to describe the operation of the pump. Referring more particularly to FIGS. 1 to 4 (although the operation is generally similar in the pump shown in FIGS. 5 to 7), the liquid entering the impeller at **15** will be brought into rotation by the vanes **22**. The flow inside the impeller will by no means be a smooth one but due to the wide internal space the relative flow velocities are low and no measurable loss of energy will occur during this phase. At the internal periphery of the impeller the liquid has obtained the full static pressure and dynamic head which corresponds to the peripheral velocity. As long as the outlet is closed, only the static pressure will theoretically be transferred to the liquid filling the space **31** through the nozzles **23**, **24**, otherwise the liquid will rotate at a low speed as energy is transferred to it by the skin friction of the impeller only. When the outlet valve is now slightly opened a small quantity of liquid will be discharged through the impeller nozzles into the space **31** and further through the outlet diffuser **32**. The absolute flow velocity c in the space **31** can be expressed with a good approximation by the equation $c=u-w$ as the radial component of the flow velocity is negligibly small. Taking into account that, at a low flow rate, the relative exit velocity w of the impeller nozzles is low it will be seen from the above equation that c is approximately equal to the impeller peripheral velocity u, but in spite of this high velocity the skin friction at the static walls surrounding the space **31** will be moderate because of the circular shape and the smooth surfaces of these walls. On the other hand the skin friction of the impeller will be very low as the surrounding liquid rotates at nearly the same speed, therefore the hydraulic efficiency will be fairly high though the output is low.

If the flow rate is further increased, at a constant impeller speed, the velocity w also increases while the average velocity c in the space **31** decreases. Therefore the skin friction at the static walls will go down while that of the impeller will rise. However, the increased impeller friction losses are more than balanced by the increase in hydraulic output so that the efficiency rises. According to a more elaborate theoretical investigation high efficiencies can be expected at flow rates corresponding to values of $w=0.3u$ to $0.6u$.

FIGURE 8 illustrates a form with forwardly directed impeller nozzles **60**. This does not affect the flow conditions inside the impeller but the average flow velocity in the space **31** has to be expressed now by $c=u+w$.

It will be seen from this equation that c is always higher than u and increases with w, i.e., with the flow rate. Consequently the skin friction at the casing will increase but, on the other hand, a positive (driving) skin friction torque will be exerted to the impeller.

Attention has to be paid to the fact that, theoretically, when pumping an "ideal" fluid, i.e., one with zero viscosity, an impeller with forwardly directed nozzles would not develop a flow at all for the following reason:

When starting the impeller while the fluid filling the space **31** is at rest (as it always would be in the beginning) a ram pressure would be developed at the mouth of each nozzle, the magnitude of this ram pressure being exactly equal to the centrifugal pressure developed inside the impeller. Both pressures would always balance each other so that the nozzles would appear to be blocked, and no flow could ever be obtained. However, a real fluid having a certain viscosity will always form a boundary layer which rotates more or less with the impeller, thus reducing the ram pressure below the level of the internal centrifugal pressure, and a flow can develop when an outlet valve is opened.

In order to improve the conditions for initiating an exit flow through forwardly directed nozzles means are provided to increase the external skin friction in the vicinity of the exit ports of the nozzles. In the example shown in FIGS. 8 and 9 the outer edges **61**, **62** of the end walls **11**, **12** of the impeller extend for this purpose in the radical direction slightly beyond the exit ports of the nozzles **60**.

FIG. 10 shows an impeller with forwardly directed nozzles **44**, **45** situated in the margins of the end walls **11**, **12** corresponding to the example shown in FIGS. 5, 6 and 7. In this case additional circular rings **63**, **64** extending in the radial direction outside the nozzles **44**, **45** are provided.

In pumps operating at medium flow rates one or more further outlet diffusers may be usefully provided in the casing. This is shown in FIG. 11 where five diffusers **35** are arranged in the ring part **34**a of the pump casing surrounding the impeller **33**a. The liquid is discharged by the diffusers **35** into a space **36** which, by way of example, is formed as a volute **37** carrying the outlet **38**.

FIG. 12 shows, on a larger scale, a diffuser **39** provided with a conical insert **40** the inner end **41** of which projects to a certain extent into the space **31** but so that the diffuser entry **42** is mostly located in the region of the undisturbed flow. This will result in an improved ram effect at low flow rates and consequently in increased static delivery pressure. At higher flow rates an undisturbed entry flow to the diffuser throat and a correspondingly improved conversion of kinetic energy into static pressure can be expected which, according to experience, is not so well obtainable with a simple tangential bore **32**a and **35**a as shown in FIG. 4 and FIG. 11, respectively.

The projecting section **41** of the insert **40** itself has

a reasonably favourable external shape so as not unduly to disturb the passing liquid flow. However, when applying high peripheral velocities it may be advisable to provide an additional section **43** of an improved streamlined shape for the projecting part **41**.

The above described improvement in the design of the outlet diffuser or diffusers is of vital importance to pumps with forwardly directed impeller nozzles (FIG. 8) as most of the energy of the liquid discharged by the impeller into the space **31** is of the kinetic nature and the pump efficiency is practically determined by that of the exit diffuser or diffusers.

In FIGS. 13, 14 and 15 the circular section **53** of the pump impeller has a reduced diameter relative to the internal size of the pump casing **19**. This design is intended to deal with very low flow rates so that two nozzles are required only for normal operating conditions. The nozzles are represented by two streamlined members **54**, **55** projecting from the periphery of the circular section **53** in the radial direction. They are provided with internal channels **56**, **57**, respectively, which have a relatively wide cross sectional extent in the circumferential direction, and they communicate with the actual outlet nozzles **58**, **59**, respectively, which are so located that the jets will have a tangential direction as required. Compared with an impeller body of the full diameter d, the two streamlined nozzles cause a considerably lower skin friction so that satisfactory efficiencies can still be obtained in spite of small flow rates.

For exceptionally low flow rates only one nozzle of this type can be used. The impeller will then have to be mechanically balanced accordingly.

What I claim is:

1. A centrifugal impeller pump comprising a casing and an impeller therein in the form of a circular vessel having a peripheral wall and side walls, an inlet eye in a said side wall, and internal fluid rotating means, said vessel being constructed of separate parts and being disposed in the casing so as to form around the vessel an annular chamber, said chamber having smooth surfaces and a discharge outlet means; at least one of said side walls having in the region of its periphery at least one tangentially-discharging exit nozzle opening into the said smooth surfaced chamber for producing a non-turbulent flow in said chamber to said outlet means.

2. A centrifugal impeller pump according to claim **1**, said tangentially-discharging nozzle having a rounded junction with the interior surface of said side wall of said vessel where an acute angle would normally be formed by the tangential location of the nozzle.

3. A centrifugal impeller pump according to claim **1**, having a plurality of said tangentially delivering nozzles in each said side wall near to the said peripheral wall, each said nozzle merging into a discharge end of restricted cross section situate in the outer face of the relevant said side wall and having a relatively large inlet opening in the inner face of the said wall.

4. A centrifugal impeller pump according to claim **1**, having a plurality of said tangentially delivery nozzles in each said side wall near to the said peripheral wall, each said nozzle opening at the inner face of said side wall and merging into a restricted discharge end opening beyond the outer face of the relevant side wall.

5. A centrifugal impeller pump according to claim **1**, having a plurality of said nozzles each converging outwardly in at least one of said side walls of the vessel, said side wall having an annular outer channel into which the said nozzles discharge tangentially.

6. A centrifugal impeller pump according to claim **1**, said discharge output means comprising a diffuser member the inner end of which projects into the said annular chamber near to the outer periphery thereof and provides a restricted inlet extending transversely across part of the width of said chamber to provide a ram effect at low flow rates and consequently an increased delivery pressure.

7. A centrifugal impeller pump according to claim 1, said discharge outlet means comprising a diffuser member the inner end of which projects into the said annular chamber with its inlet and providing a restricted outlet extending transversely across part of the width of said chamber near to the outer periphery thereof and comprising a curved fillet interposed between the said inner projecting end of the said member and the outer periphery of the said chamber to fill the angle the projecting end of the diffuser member would otherwise make with the said outer periphery and to enable the liquid material not entering the diffuser member to have a smooth flow round the said outer periphery.

8. A centrifugal impeller pump with an axial inlet, comprising a casing and an impeller therein in the form of a circular vessel with an axial inlet eye, a peripheral wall and side walls and internal fluid-rotating means carried by and strengthening said side walls; the said vessel being disposed in the said casing so as to form around the vessel an annular chamber, said chamber having smooth surfaces and outlet means, the peripheral region of the vessel being provided with discharging nozzles each projecting outwards beyond the outer periphery of said vessel into the said annular chamber and being shaped first with a radial passage opening into the vessel and then with a continuing tangential exit passage opening into the said chamber.

9. A centrifugal pump according to claim 8, in which the said outwardly projecting nozzles are of streamlined

cross section with nose leading in the direction of rotation and with said continuing tangential exit passage in the tapering tail end of said nozzle.

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