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Davis

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[54] **VARIABLE DIFFUSER CENTRIFUGAL PUMP**

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[73] Assignee: **General Electric Company**

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[51] **Int. Cl.** **F04d 15/00,** F04d 29/46

[58] **Field of Search** 415/157, 158, 151, 415/150, 164, 165

[56] **References Cited**

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[57] **ABSTRACT**

A centrifugal pump is supplied with a discharge shutter valve which provides a variable diffuser for the pump. The shutter valve includes a hollow, slotted cylinder positioned for movement into a variety of operative positions between the impeller and the diffuser vane passages. Each operative position completely closes one or more of the diffuser passages while leaving the remaining passages completely open, thereby providing a pump with high head rise over the complete operating range of the pump.

8 Claims, 5 Drawing Figures

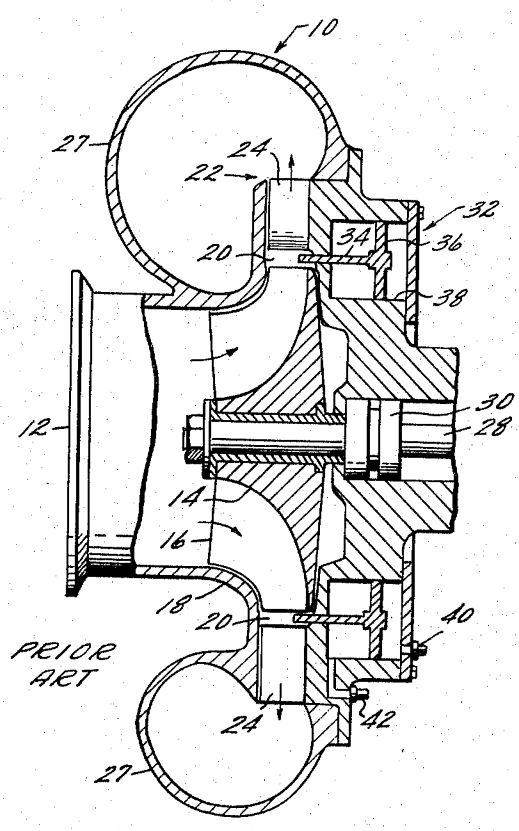
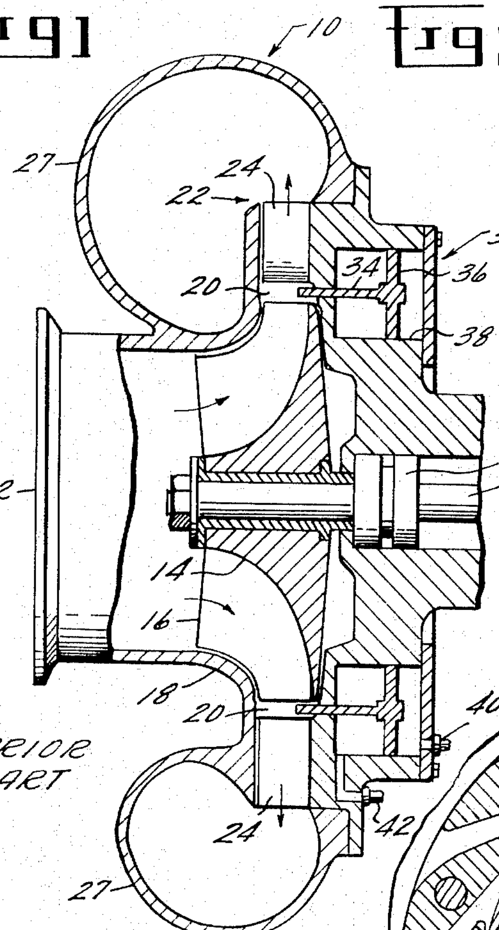


Fig 1



PRIOR ART

Fig 2

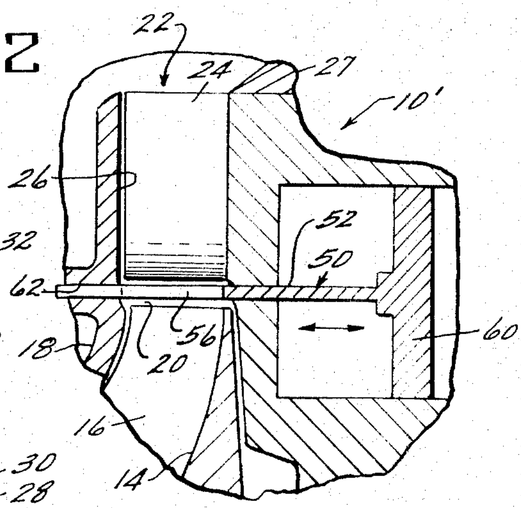


Fig 3

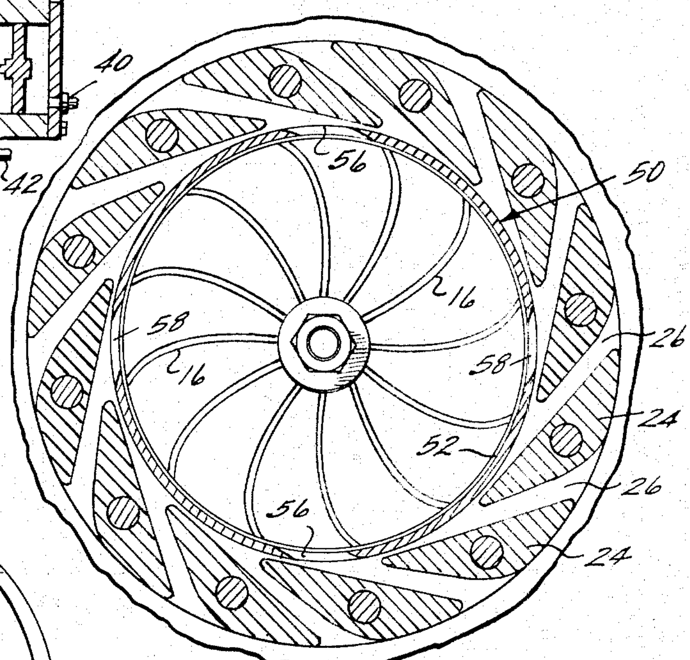


Fig 4

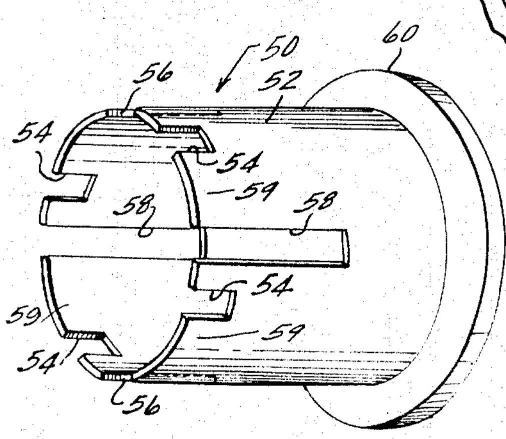


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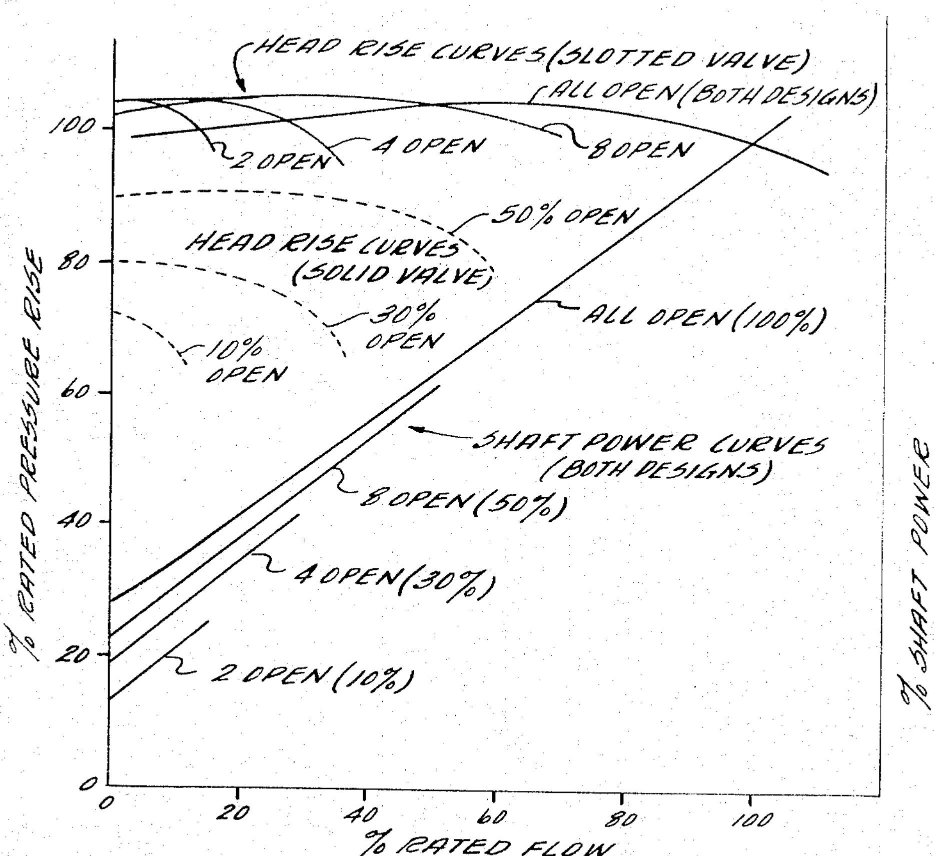
Fig 5



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VARIABLE DIFFUSER CENTRIFUGAL PUMP

BACKGROUND OF THE INVENTION

This invention relates generally to centrifugal pumps and, more particularly, to a variable diffuser exit passageway for such a pump.

Many fluid pumping applications require a very large flow range with low pump temperature rise throughout the flow range. For example, current and advanced gas turbine engines and variable thrust liquid rocket engines require very large fuel flow ranges with minimum fuel temperature rise from the pumping system. This minimum fuel temperature rise requirement is rather easily attained at high flow rates because of the high flows and the short dwell times within the pumping system. The low pump temperature rise requirement during low flow operation is not as easily attained. In fact, fuel pump performance considerations show that the requirement of a high fuel flow range (high flow turndown) is inconsistent with the requirement of low fuel temperature rise across the pump at low pump outputs.

As a result of the above consideration, traditional gas turbine engine and variable thrust liquid rocket engine fuel systems utilize combinations of several fuel pumping elements operating in parallel to solve the flow range problem. The number of pumping elements operated at any one time depends upon the required flow output, with the remaining pumps being inactivated until needed. This approach to obtaining a wide fuel flow range, while successful in providing low fuel temperature rise at low pump flows, presents problems of added size, complexity and control.

Centrifugal pumps, while capable of providing the high output requirements of gas turbine engine and variable thrust liquid rocket engine systems, have historically been limited to relatively narrow flow ranges because of the excessive temperature rise and stability problems associated therewith at low percentages of rated flow. Previous attempts at providing a centrifugal pump with a relatively wide flow range have met with various degrees of success. For example, variable speed drive centrifugal pumps can achieve the necessary high flow turndowns but the reduced speed results in substantial head dropoff from the pump. Pump inlet throttling, such as by using the inlet vapor core approach, or pump discharge throttling with a diffuser inlet shutter can each be utilized to provide the reduced temperature rise at low flow. Each of these approaches, however, also results in significant head rise dropoff at high flow turndown ratios.

One further proposed approach consists of providing a pump with a variable angle diffuser vane, which can successfully provide the low temperature rise without significantly causing head rise dropoff. The complexity of such a pump, however, in most cases far outweighs the advantages gained therefrom.

SUMMARY OF THE INVENTION

It is an object of this invention, therefore, to provide a relatively high flow range centrifugal pump having low temperature rise throughout the range of operation. It is a further object of this invention to provide such a pump without the necessity of variable angle diffuser vanes.

Briefly stated, the above and other related objects are attained by providing a centrifugal pump with a variable diffuser which utilizes a slotted valve to selectively close off diffuser vane entry passages as the fuel rate of the pump is decreased. In one embodiment, the slotted valve takes the form of a slotted cylindrical sleeve radially located between the pump impeller tip and the pump diffuser entry passages. Means are provided for moving the valve axially as a function of pump-delivered flow in such a manner that selective diffuser vane entry passages are completely closed off while the desired flow is delivered through the remaining fully open diffuser vane entry passages. The net effect of the slotted valve is to increase overall efficiency at low delivered flows while maintaining near rated head rise of the pump throughout its range of operation.

DESCRIPTION OF THE DRAWINGS

While the specification concludes with a series of claims which particularly point out and distinctly claim the subject matter which Applicant regards as his invention, a complete understanding of the invention will be gained from the following description of a preferred embodiment. This description is given in connection with the accompanying drawings in which:

FIG. **1** is a generally sectional view, with portions deleted, describing a centrifugal pump constructed in accordance with the prior art;

FIG. **2** is an enlarged partial sectional view of a centrifugal pump, similar to FIG. **1**, describing a pump constructed in accordance with the present invention;

FIG. **3** is a sectional view, with portions deleted, showing the cross section of the pump of FIG. **2** with the slotted valve in one of its operative positions;

FIG. **4** is a perspective view of the slotted valve of FIGS. **2** and **3**; and

FIG. **5** is a graphical plot showing the outputs of the pumps illustrated in FIGS. **1** and **2**.

DESCRIPTION OF A PREFERRED EMBODIMENT

Referring now to the drawings wherein like numerals correspond to like elements throughout, attention is directed initially to FIG. **1** wherein a centrifugal pump constructed in accordance with the prior art is designated generally by the numeral **10**. The pump **10** is shown to include an axial inlet **12**, a rotating impeller wheel **14** which includes a plurality of impeller vanes **16**, and a casing **18** which defines a radial outlet **20** surrounding the tips of the impeller vanes **16**. A diffuser **22** surrounds the radial outlet **20** and includes a plurality of stationary diffuser vanes **24** at its inlet. Each pair of the diffuser vanes **24** defines a diffuser entry passage **26** from the radial outlet of the pump **10** to a toroidal-shaped collector **27**. The impeller wheel **14** extends from a rotatable shaft **28**, which is journaled for rotation in bearings **30**.

As previously mentioned, the elements described above are typical of prior art centrifugal pumps. For this reason, the structure of the centrifugal pump **10** is shown somewhat schematically in FIG. **1**. As will become apparent from the following description, Applicant's inventive concept may be applied to any type centrifugal pump and the structure shown in FIG. **1** is merely meant to be illustrative and not limiting in any manner.

Centrifugal pumps, such as that shown in FIG. **1**, have been limited to relatively narrow flow ranges because of the excessive temperature rise and stability problems at low percentages of rated flow. These problems may be substantially reduced or eliminated by reducing low flow power loss. For this reason it is known to provide the centrifugal pump **10** with a diffuser inlet shutter **32** which throttles the pump discharge by partially closing all of the diffuser entry passages **26**. In its simplest form, as shown in FIG. **1**, the diffuser inlet shutter **32** takes the shape of a hollow cylinder positioned between the radial outlet **20** of the pump **10** and the diffuser vanes **24** as shown in FIG. **1**. The cylinder **34** is provided with some actuating mechanism, such as a piston **36** formed integrally with one end of the cylinder **34**. The piston **36** is positioned within a chamber **38** which is supplied with actuating fluid by means of ports **40** and **42**, respectively.

In operation, fluid is supplied to the centrifugal pump **10** at the inlet **12** in any known manner, while the shaft **28** and impeller wheel **14** are driven by some suitable driving mechanism (not shown). The fluid is acted upon by the impeller vanes **16** and is delivered through the radial outlet **20** at a much higher pressure and velocity than its inlet pressure and velocity. The fluid then passes through the diffuser entry passages **26** of the diffuser **22**, where its velocity pressure is converted to static pressure. From the diffuser **22** the fluid is directed into collector **27**. The fluid may then be directed to a suitable control device (not shown) or to some other component, depending upon the use of the pump **10**.

In cases where the maximum output of the pump **10** is desired, sufficient servo fluid is delivered to the rod end of the piston **36**, through the port **42**, to position the cylinder **34** completely outside of the diffuser inlet passages **26**.

Being a fluid dynamic device dependent upon fluid velocity and velocity vectors for adequate flow, pressure and pressure stability, the centrifugal pump **10** can only be designed to produce optimum effiency at a single value of output flow. This optimum or best efficiency point is usually obtained at or near the maximum flow point in order to satisfy the required pressure rise with minimum impeller size. This necessity of sizing the pump for high flow results in poor efficiency at low flows since the pump is grossly oversized in terms of fluid volume during low flows. In addition, the blade and diffuser shapes are mismatched for low flow operation. At these low flows, excessive power loss to the fuel can result in large fuel temperature rise, and numerous forms of pump and pump/system related pressure instabilities can occur.

For this reason, whenever the particular application of the pump **10** calls for low flow outputs, servo fluid is delivered to the head end of the piston **36** through the port **40** and the piston **36** is moved partially into the radial outlet **20**. By thus throttling the pump discharge, the low flow power losses can be substantially reduced and the fluid temperature rise associated with the power losses can be minimized. Unfortunately, by partially blocking each of the diffuser entry passages **26**, the effectiveness of the diffuser **22** is decreased, thereby resulting in significant head rise dropoff at high flow turndown ratios. This drawback to the use of diffuser inlet shutter is shown graphically by the dotted line curves of FIG. **5**. As shown therein, as the percent of rated flow decreases the effect of partially closing the diffuser inlet passages **26** reduces shaft power somewhat, but the effect on the Q-H curves is also significant with the percent of rated head rise dropping significantly near the lower end of the rated flow spectrum.

Referring now to FIGS. **2** through **4**, Applicant's inventive concept for diminishing the fluid temperature rise while maintaining near rated head rise throughout the range of operation of a pump **10'** will be described. The basic construction of the pump **10'** remains unchanged from that previously described. That is, the pump **10'** still includes an inlet **12**, an impeller wheel **14**, a diffuser **22**, a plurality of diffuser vanes **24** which define diffuser inlet passages **26**, and a collector **27**. The primary difference between the pump **10** described in connection with FIG. **1** and the improved pump **10'**, shown in FIGS. **2** through **4**, lies in the fact that the diffuser inlet shutter **32** of the pump **10** is replaced with a variable diffuser valve **50**.

The variable diffuser valve **50**, as shown most clearly in FIG. **4**, consists of a cylinder **52** having a plurality of various length slots **54, 56** and **58**, located therein. The cylinder **52** is further shown to include a piston member **60**, similar to the piston **36** of FIG. **1**, which may be formed integrally with the cylinder **52** or joined thereto in any known manner.

Each of the slots **54 – 58** begins in the end of the cylinder **52** opposite the piston **60** and extends axially toward the piston **60**. The slots **54 – 58** are of approximately equal width but are of varying axial lengths, with the slots **54** being the shortest and the slots **58** the longest, as shown in FIG. **4**.

The slots **54 – 58** are spaced around the periphery of the cylinder **52** and positioned in such a manner that each of the slots lies between one of the diffuser inlet passages **26** and the tips of the impeller vanes **16**. This arrangement is shown in the perspective view of FIG. **4** wherein the eight slots are positioned as if there were twelve equally spaced slots, i.e. the centerlines of the slots lie 30° apart. Since there are twelve diffuser passages **26** and only eight of the slots **54 – 58**, there are four solid portions of the cylinder **52**, generally designated by the numeral **59**, which block four of the diffuser passages **26** as soon as the valve **50** is moved into a first operative position within the radial outlet **20**. While the cylinder **52** is shown to include a total of eight slots of three various lengths, the total number of slots and the sizing thereof will, of course, depend upon the number of diffuser passages and the design of the centrifugal pump, as will become apparent from the following description of the operation of the variable diffuser centrifugal pump **10'**.

The basic operation of the pump **10'** is identical to that of the pump **10** with fluid being delivered through an inlet **12** to the impeller wheel **14** and being directed radially outwardly while being acted upon by the impeller vanes **16**. The fluid is then delivered through the radial outlet **20** to the diffuser entry passages **26**, where its velocity pressure is converted to static pressure, and then it passes into the collector **27**.

When the pump **10'** is operating near its rated maximum output, the variable diffuser valve **50** is held in an inoperative position such that all diffuser entry passages **26** are fully open to receive fluid from the radial outlet **20** and to deliver the same to the collector **27**. As the desired output flow of the pump is decreased, the valve **50** is moved axially into a first operative position between the tips of the impeller vanes **14** and the diffuser vanes **24**. The diffuser valve **50** is stopped in this first position in which each of the slots **54, 56** and

58 lies between the tips of the impeller vanes 14 and one of the diffuser inlet passages 26. In this manner, because there are eight slots positioned opposite the twelve diffuser vane passages 26, eight of the passages 26 will remain fully open while the remaining four diffuser passages 26 will be completely closed by the solid portion 59 of the cylinder 52. In this manner, the eight fully opened diffuser inlet passages 26 operate as if the pump 10' were delivering its total rated flow, while the remaining four passages have absolutely no effect on the pump output. That is, the eight diffuser inlet passages 26 are still operating near their peak design efficiency in contrast to the structure of FIG. 1 which partially closes down each of the diffuser inlet passages causing throttling and resulting loss of diffuser pressure recovery.

When the desired output flow from the variable diffuser centrifugal pump 10' further decreases, the diffuser valve 50 is moved further to the left (as shown in FIG. 2) to a second position in which the slots 54 are no longer positioned between the impeller tips and the diffuser vanes 24. That is, the diffuser valve 50 is moved to a second operative position in which that portion of the cylinder 52 which includes the slots 54 lies within a groove 62 (FIG. 2) formed in the casing 18. In this position, only the slots 56 and 58 are positioned between the impeller wheel 14 and the diffuser vanes 24, and thus only four of the diffuser vane passages 26 are fully open while the remaining eight are fully closed. The four fully open passages 26 continue to operate near their design efficiency, but the required shaft power of the pump is reduced significantly. FIG. 3 is a sectional view of the pump with the diffuser valve 50 located in this second position. As clearly shown in this view, the slots 56 and 58 deliver fluid from the impeller vanes 16 to four of the diffuser vane passages 26, while the remaining eight passages are completely blocked.

Similarly, if the desired output of the variable diffuser centrifugal pump 10' is further decreased, the diffuser valve 50 is moved axially to a third operative position in which only the slots 58 lie between the impeller wheel 14 and the diffuser vanes 24. In this position, only two of the diffuser vane passages 26 receive fluid from the impeller wheel 14, while the remaining ten passages are completely blocked.

The output of a pump constructed as shown in FIGS. 2 through 4 is plotted as solid lines in FIG. 5. As shown therein, the net effect of the variable diffuser is to increase overall efficiency at low delivered flows while maintaining near rated head rise. That is, the shaft power curves for the fully open and three alternative positions of the diffuser valve 50 are identical to those for the centrifugal pump 10. However, the head rise curves show that the percent of rated head rise actually increases as the number of open diffuser inlet passages 26 decreases, as opposed to the dotted head rise curves which show significant decreases in head rise for the pump 10 of FIG. 1 at low rated flow levels. The increase in head rise is caused by the increasing pressure as the number of diffuser inlet passages is decreased.

As described above, Applicant has provided an improved centrifugal pump which has a number of basic advantages. The primary advantage of this pump is the high head rise over the full operating range of the pump. This phenomenon, when combined with the further advantage of low fluid temperature rise due to low power loss at low flow rates, provides a centrifugal pump capable of usage in many areas heretofore thought impossible. Other advantages become readily apparent when one compares the simplicity of the described design with the complexity of previously known variable diffuser geometry-type pumps. Furthermore, the pump described above provides the potential of maintaining stable pump operation at lower delivered flow rates than pump 10.

Various changes could be made in the structure shown in FIGS. 2 through 4 without departing from the broader aspects of Applicant's invention. For example, the means for causing axial movement of the diffuser valve 50 could take many forms other than the piston 60 described above. For example, the diffuser valve 50 could be positioned mechanically by some linkage arrangement. Likewise, the shape and number of the slots within the cylinder 52 could vary substantially from those shown while still providing the basic function of completely closing a certain number of inlet diffuser passages while leaving the remaining passages completely open. These and other similar changes are meant to be covered by the appended claims.

What I claim is:

1. In a centrifugal pump of the type including an impeller adapted to propel fluid through a radial outlet to a diffuser which includes a plurality of stationary diffuser vanes defining diffuser passages therebetween, the improvement comprising:

a discharge shutter valve positioned between said impeller and said diffuser vanes and capable of movement between an inoperative position and at least one operative position, said valve including meand for completely closing the inlets to individual diffuser passages when in an operative position while leaving the inlets to the remaining passages completely open.

2. The improved centrifugal pump recited in claim 1 wherein said valve comprises a hollow cylinder.

3. The improved centrifugal pump recited in claim 2 wherein said cylinder has a plurality of axially extending slots located therein, said slots being separated by solid portions of said cylinder.

4. The improved centrifugal pump recited in claim 3 wherein each of said slots is positioned opposite the inlet of one of said diffuser passages.

5. The improved centrifugal pump recited in claim 4 wherein at least one of said slots is of a different axial length from the remaining slots.

6. The improved centrifugal pump recited in claim 5 wherein said valve includes actuating means for moving said cylinder between its operative and inoperative positions.

7. The improved centrifugal pump recited in claim 6 wherein said actuating means includes a piston extending from one end of said cylinder, a chamber for receiving said piston, and means for delivering servo fluid to opposite sides of said piston.

8. The improved centrifugal pump recited in claim 7 wherein said slots extend from the end of said cylinder opposite the end which includes said piston, and said slots extend toward said piston.

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