United States Patent [19]

Kumm

[11] **3,918,828**

[45] **Nov. 11, 1975**

[54] **FLOW CONTROL FOR COMPRESSORS AND PUMPS**

[76] Inventor: **Emerson L. Kumm,** 1035 E. Laguna Drive, Tempe, Ariz. 85282

[22] Filed: **Sept. 5, 1974**

[21] Appl. No.: **503,469**

[52] **U.S. Cl.** **415/61;** 415/62; 415/143

[51] **Int. Cl.**[2] **F01D 1/24**

[58] **Field of Search** 415/60, 61, 62, 66, 67, 415/68, 143, 147

[56] **References Cited**

UNITED STATES PATENTS

2,400,240	5/1946	Lincoln	415/62
2,555,312	6/1951	Bollay	415/143
3,546,880	12/1970	Schwaar	415/62
3,692,420	9/1972	Mittelstaedt	415/143

FOREIGN PATENTS OR APPLICATIONS

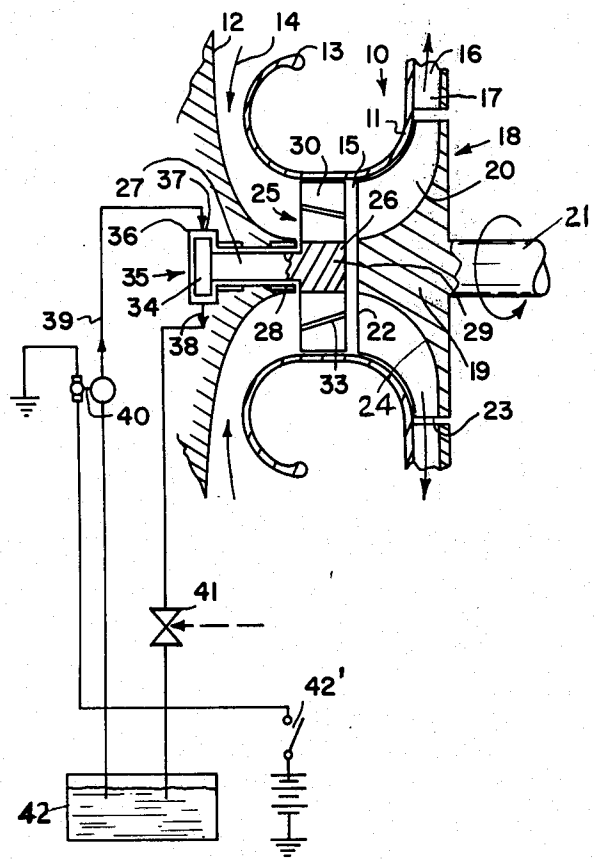
963,415	1/1950	France	415/62
399,619	10/1933	United Kingdom	415/61
520,541	2/1952	Canada	415/62

Primary Examiner—Henry F. Raduazo
Attorney, Agent, or Firm—Herschel C. Omohundro

[57] **ABSTRACT**

This flow control comprises a vane equipped guide element disposed in a fluid compressor or pump between the usual impeller and the fluid inlet. The element is mounted for rotation either by fluid drawn into the inlet by the impeller or by a positive displacement fluid energy transfer member connected with the element and communicating with a fluid system external of the compressor. Valve and pump means may be utilized in the fluid system to control the operation of the guide element. If desired, the guide element may be permitted to windmill freely in response to flow of fluid into the compressor, it may also be held stationary, or it may be driven by an external force in a selected direction at a controlled speed to secure the desired effect. The vanes on the element direct the inflowing fluid to the impeller in a manner to improve the performance of the compressor or pump.

7 Claims, 7 Drawing Figures



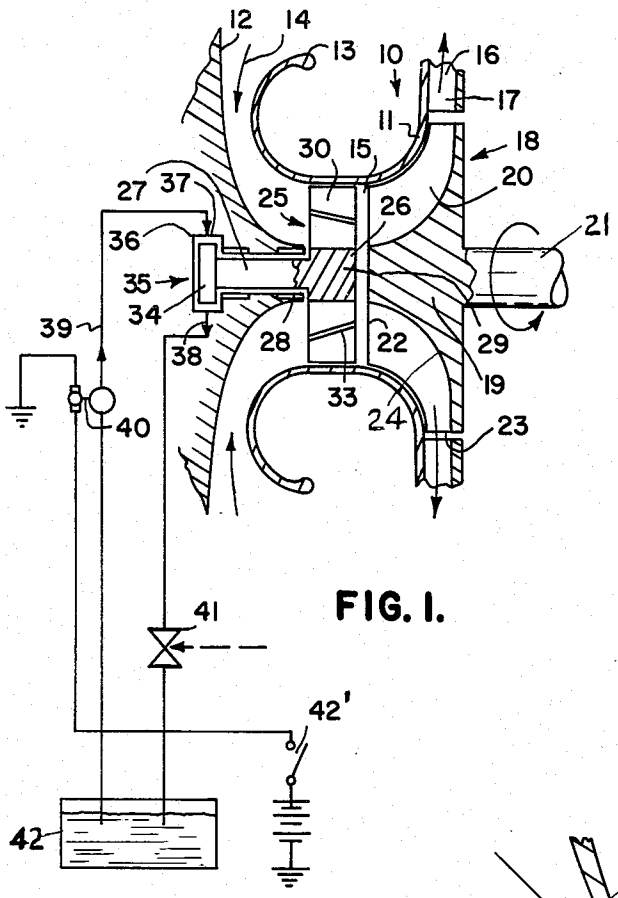


FIG. 1.

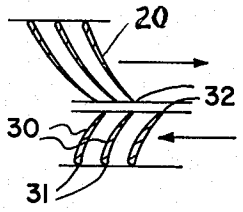


FIG. 2.

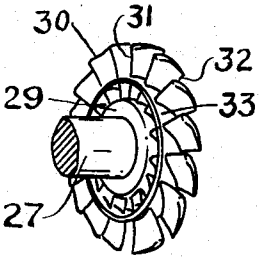


FIG. 3.

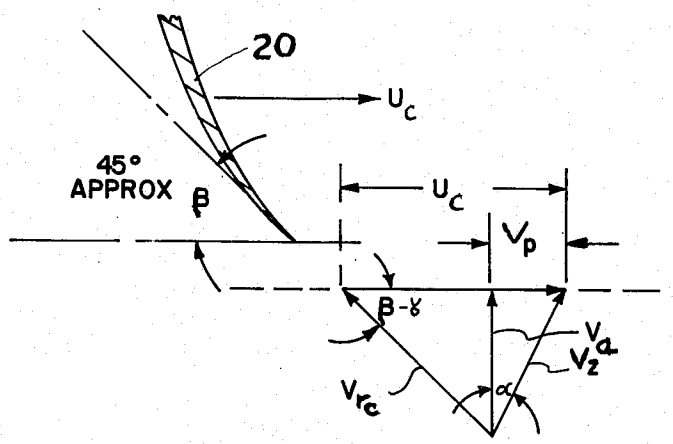
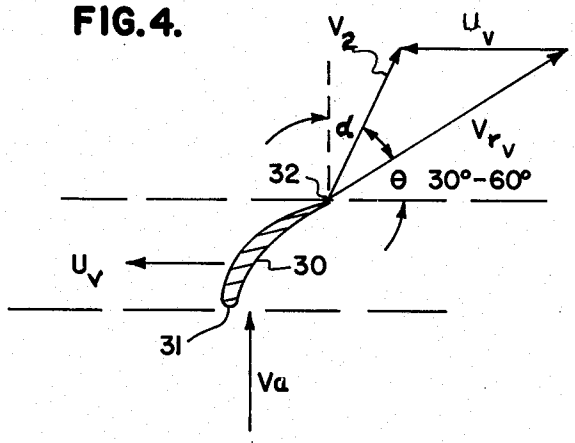


FIG. 4.



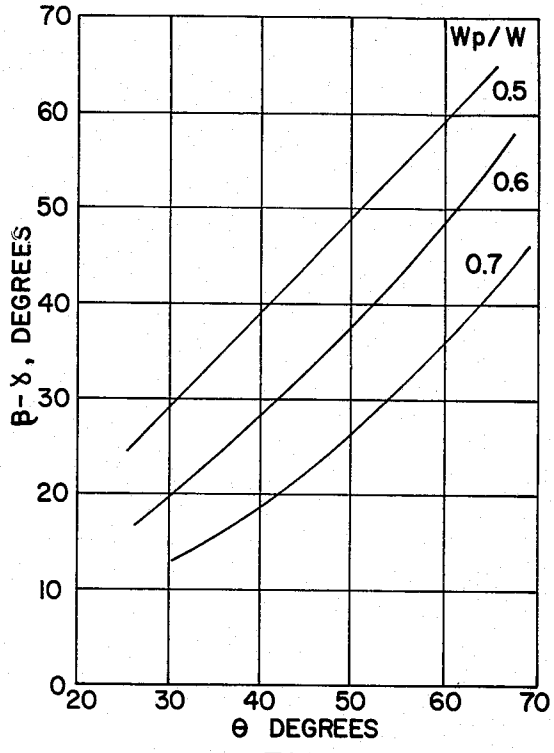


FIG. 5.

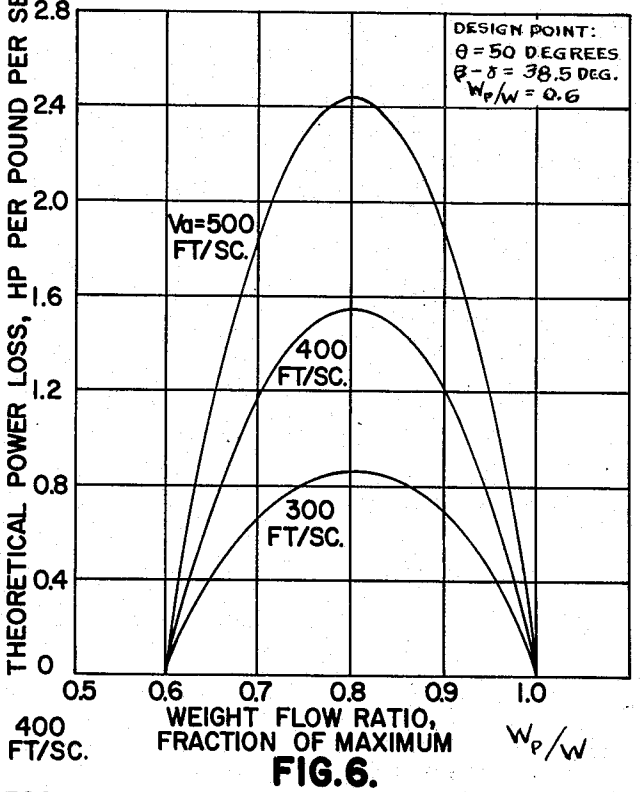


FIG. 6.

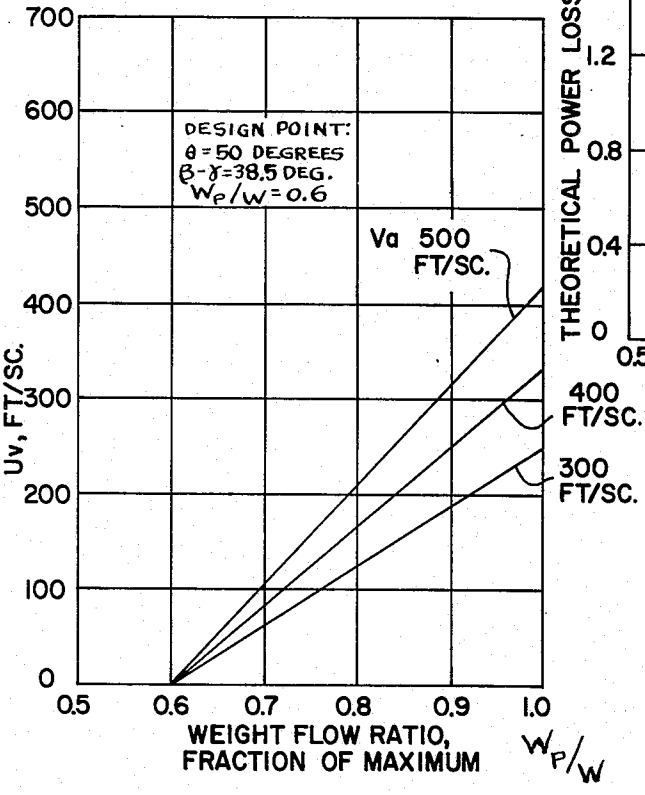


FIG. 7.

FLOW CONTROL FOR COMPRESSORS AND PUMPS

BACKGROUND OF THE INVENTION

This invention relates generally to fluid moving apparatus, such as compressors or pumps, and is more particularly directed to means in such apparatus for controlling flow of fluid therein to improve the efficiency thereof.

The invention relates to the class of structure exemplified by the following patents:

1,460,428	Moody, L. F.
1,578,843	Moody, L. F.
2,306,742	Moody, L. F.
2,561,840	Busquet, J. C.
2,733,853	Trumpler, W. E.
3,289,919	Wood, G. R.
3,657,881	Amman, C. A., et. al.
3,724,208	Welch, E. S., et. al.

SUMMARY OF THE INVENTION

An object of this invention is to provide a compressor or pump of the type having an impeller, which draws fluid into a casing and expels it therefrom, with means upstream of the impeller for guiding the fluid to the impeller in a manner to improve the operation and efficiency of the device. It has been found in some instances where pumps or compressors are driven at relatively constant speed that it is desirable to vary the output to meet certain conditions. Heretofore, this objective has been accomplished, without complete satisfaction, either by throttling the discharge with its attending power losses, or by limiting the intake or flow through the use of expensive and complicated controls such as adjustable inlet guide vanes with their inherent losses by leakage, their requirements of gearing and difficulties of operation. It is an object herein to secure the flow control in a unique inexpensive manner.

Another object of the invention is to provide a compressor or pump with flow control means having an inlet guide which may be supported for rotation in the compressor or pump and permitted to windmill, be held stationary or driven at suitable speeds in forward or reverse directions.

Still another object of the invention is to provide a compressor or pump, of the type having a centrifugal impeller, with fluid guide means between the impeller and the compressor inlet, the guide means having a vaned element supported for rotation and means for controlling the speed and directions of rotation of such element.

A further object of the invention is to provide the guide means referred to in the preceding paragraph with a shaft and a fluid pressure energy translating device connected therewith, the latter being arranged in a fluid system operative to transmit energy to or absorb energy from the guide means.

Still another object of the invention is to provide the guide means with a vane construction which will direct fluid flow in the most effective manner.

A further object is to provide the fluid guide means mentioned in several of the preceding paragraphs with flow splitting means between the outer ends of the vanes and the guide hub, the flow splitting means being shaped to obtain the desired velocity and direction profile from hub to tip ahead of the impeller.

Other objects and advantages of the invention will be obvious from a perusal of the following description of one form of the invention which has been shown in detail in the accompanying drawings.

IN THE DRAWINGS

FIG. 1 is an axial sectional view, partly schematic, of a compressor equipped with a flow control means embodying the invention;

FIG. 2 is a fragmentary sectional view taken through the guide and impeller on the line II—II of FIG. 1 to show the relationship of the blades on these two elements;

FIG. 3 is a perspective view of a portion of the flow control means shown in FIG. 1;

FIG. 4 is a view showing the entrance end of the compressor impeller and one vane of the flow control rotor with appropriate velocity triangles, well known to those skilled in the art, to illustrate the theory of incompressible fluid flow through a compressor embodying the invention;

FIG. 5 is a graph with curves showing the derived angle relationship for the exit angle of a blade on the guide member versus the inlet angle of a blade on the impeller minus the desired angle of attack;

FIG. 6 is a graph with curves showing how the vaned rotary element power loss varies with the reduction weight flow caused by changing the speed of the vaned rotary element used with an air compressing impeller; and

FIG. 7 is a graph with curves showing the vaned rotary element speed as a function of the weight flow ratio for a specific rotary vane and impeller design.

DESCRIPTION OF THE DISCLOSURE

Referring more particularly to the drawings and especially to FIGS. 1 and 2, it will be apparent that the invention has been illustrated as applied to a centrifugal pump or compressor **10**. The latter has a casing **11** with walls **12** and **13** spaced to provide an annular inlet **14**, a rotor chamber **15**, and a radially outwardly extending annular outlet **16**, the latter being provided with the usual diffuser vanes **17**.

Within the chamber **15** there is supported for rotary movement the usual impeller **18** of the centrifugal type having a hub **19**, curved blades **20**, and a drive shaft **21** through which the impeller is driven by a prime mover (not shown). The blades are substantially conventional, having axially facing leading edges **22**, and radially facing trailing edges **23**. The hub is curved, as at **24**, to turn the fluid being pumped from an axial direction to a radial direction as it flows through the passages between the blades. From the short description of the compressor thus given it will be obvious that when the shaft **21** is driven in the proper direction, fluid will be drawn into the chamber **15** via the inlet **14** and expelled through the outlet. In such flow, without other structure, the fluid changes direction from radially inwardly in the inlet to axially through the chamber **15** to the leading edges of the impeller blades after which it is whirled and turned in the impeller to flow outwardly through the outlet **16**. In the latter, diffuser vanes convert the velocity to pressure. As mentioned, the operation thus far is substantially conventional.

In many prior constructions, guide means are provided upstream of the impeller to initially impart whirl to the fluid immediately prior to its entry into the impeller passages. In some prior constructions the guide

means are made adjustable to secure varied results. Frequently the mechanism for accomplishing such adjustment is complicated and inefficient in operation and otherwise objectionable.

To avoid the foregoing objections and secure other additional advantages, the flow control means of the present invention, designated generally by the numeral **25,** has been provided. The flow control means **25** includes a rotary guide member **26** fixed on one end of a shaft **27** supported for rotation in bearing **28** at the outer end of and in alignment with the axis of the impeller. The guide member **26** includes a hub **29** and radially outwardly projecting blades **30.** As shown in FIGS. **2, 3** and **4,** blades **30** curve backwardly and laterally from the forward or leading edges **31** to the rear or trailing edges **32.** The blades **30** may also taper in thickness from the leading to trailing edges. The lateral direction of curvature of the blades **30** is opposite to that of the impeller blades, as will be apparent from FIGS. **2** and **4.** From FIG. **4** it will be noted that the exit angle of the blades of the guide, i.e., the angle of the blade relative to the plane of rotation, is between **30** and **60** degrees. From the same Figure it will also be noted that the entrance angle of the blade of the impeller with which the disclosed guide means is used is approximately **45** degrees to the plane of rotation, the direction of rotation of the impeller being opposite to that of the guide when the latter is driven by fluid flowing into the compressor. The blades **30** are reinforced by thin webs **33** extending between adjacent blades and such webs combine to form a frustoconical tube between the hub **29** and the peripheral ends of the blades, the base of the frustum facing toward the inlet of the compressor. In addition to reinforcing the guide blades **30,** this tube also serves to modify the radial fluid velocity profile at the impeller inlet.

As shown in FIG. **1,** the outer end of the shaft **27** is provided with one element **34** of a fluid energy translating device such as a positive displacement pump or motor designated generally by the numeral **35.** This device includes a casing **36** with combination inlet-outlet ports **37,38** which communicate with a fluid pressure system **39.** System **39** may be provided with another pump **40,** a valve **41,** and a reservoir **42.** Through the adjustment of valve **41,** pump or motor **35** may be caused to impose a load (constant or variable, as desired) on the rotary guide member **26** to govern its speed of rotation and effectiveness as a flow control for the compressor. The pump **40** may be driven, if desired, to drive pump or motor **35** (in either desired direction) and impart rotary movement to guide member **26** to create a desired flow control effect.

From the foregoing it will be recognized that means have been provided for controlling the output of a fluid pump or compressor by varying the fluid flow into the impeller or other fluid motivating means. Such flow controlling means includes a guide element of a rotary type with vanes or blades and other means for modifying the direction and speed of flow of fluid entering the pump or compressor. The flow controlling means also includes a fluid pressure energy translating device disposed in a fluid pressure system with means for loading the fluid pressure energy translating device to control the speed and direction of rotation of the guide element and additional means for causing the guide element to revolve in a desired direction and at a predetermined speed. The guide element has vanes or blades shaped to change the axial flow to a flow containing a whirl velocity most suitable for flow control from the pump or compressor with minimum expenditure of energy.

The theory of operation of the guide element may best be understood by reference to the vector diagrams in FIG. **4.** For normal compressor or pump impeller speeds and with reference to the upper vector diagram, U_r represents the impeller blade velocity and, as shown, extends to the right. V_2 represents the fluid (gas or liquid) velocity at the discharge of the blades or vanes **30** of the guide element **25.** Accordingly, V_{r_c} will then represent the fluid velocity relative to the pump or compressor impeller blade **20.** The axial fluid velocity is then represented by V_a. The velocity of the vaned rotary guide element blade is represented by U_r in the lower vector diagram with the magnitude and direction of fluid velocity V_2 the same as in the upper vector diagram. The velocity of the fluid at the trailing edge of the vaned rotary guide element blade **30** relative to such blades is then represented by V_{r_v}. The trailing or exit angle of the blade **30** of the guide element and the inlet angle of the pump or compressor blade, specify the blade geometry significant to the embodied construction. An angle of attack, , is designated as used in a well-known fashion by designers of turbomachinery blade elements. In the upper vector diagram, the assumed velocity and angular relationship result in a prewhirl fluid velocity of V_p which extends in the same direction as the pump or compressor blade velocity U_r. For efficient pump or compressor performance, it is necessary that the angle - not vary by more than **3** to **5** in either direction for efficient operation, as is well known in the design of turbomachinery blade elements. As will be shown, this fact then requires a prescribed relationship between rotating elements **18** and **25** and the axial flow velocity V_a.

Assume, in a first case, that the vaned rotary element **25** is substantially unrestricted in its rotation and windmills freely in the fluid flow. It is observed that the fluid flow direction will be substantially unchanged in passing through the vaned rotary element blades **30,** and consequently give a prewhirl velocity, V_p, that is very small. Assume, now in a second case, that the vaned rotary element **25** is held stationary, such as by closure of valve **41** to prevent flow from pump **35** which is connected with such element, the fluid flow direction as it passes through the vaned rotary element blades **30** will be substantially changed and will give a large value to the prewhirl velocity V_p. Now, it may be seen that to maintain the same, or nearly the same, value of the flow angle, -, it is necessary that the axial flow velocity, V_a, be greatly reduced compared to that of the first case. Thus, it is perceived that the most efficient axial flow velocity, V_a, will decrease as the vaned rotary element velocity, U_r, is decreased from its free windmilling speed to a lower velocity approaching that of a stationary vaned rotary element. The mass flow for incompressible fluid is directly proportional to the axial flow velocity, V_a. Hence, it is shown that decreasing the speed of the vaned rotary element corresponds to reducing the mass flow to the impeller of the compressor or pump for most efficient operation.

Using the velocity triangles of FIG. **4,** the geometrical design angle, -, may be specified relative to the exit angle, , of the blade **30** of the rotary element **25** and the weight flow ratio limit as given on FIG. **5.** The weight flow ratio limit, $^{W}p/W$, is defined as the ratio of the inlet weight flow with a stationary vaned rotary element **25** to the inlet weight flow with a freely windmilling vaned

rotary element **25.**

The analysis of the flow equations using continuity of flow and the geometric nomenclature of FIG. **4** gives the following equation for the weight flow ratio limit for incompressible flow:

$$W_p/W = [1 + V_v/U_c] \Big/ \left[1 + \frac{\tan ( )}{\tan }\right]$$

This equation is derived as follows:

By continuity of flow

1. $W = A V_a$

With prewhirl

$\tan = V_a/(U_r + V_p)$

(2) $V_a = (U_r + V_p) \tan$, $V_p = \frac{V_a}{\tan} U_r$

$\tan ( ) = V_a/[U_r \ V_p]$

3 $V_a = [U_r \ V_p] \tan ( )$

Let Eq^n (2) = Eq^n (3)

$(U_r + V_p) \tan = (U_r \ V_p) \tan ( )$

Solve for V_p,

$V_p [\tan + \tan ( )] = U_r \tan ( ) \ U_r \tan$

$V_p = [U_r \tan ( ) \ U_r \tan]/[\tan + \tan ( )]$

Sub. in Eq^n (1) and (2)

$$(4) \ W_p = A\left[U_r \tan + \frac{U_r \tan ( ) \ U_r \tan}{1 + \frac{\tan ( )}{\tan}}\right]$$

If $V_p = 0$. zero prewhirl

By Eq^n (2), $V_a = U_r \tan$

By Eq^n (3), $V_a = U_r \tan ( )$

5. $W = AV_a = AU_r \tan ( )$, Constant angle of attack.

Dividing (4) by (5)

$$(6) \ \frac{W_p}{W} = \frac{U_r \tan}{U_r \tan ( )} + \frac{1 \left[\frac{U_r \tan}{U_r \tan ( )}\right]}{1 + \left[\frac{\tan ( )}{\tan}\right]}$$

Note that from Eq^n (2)

$$(7) \ \tan = V_p/V_a = \frac{1}{\tan} \quad \frac{U_r}{U_a}$$

Eq^n (6) may be simplified to

$$(8) \ \frac{W_p}{W} = [1 + U_r/U_c] \Big/ \left[1 + \frac{\tan (65)}{\tan}\right]$$

The hydraulic oil pump **35,** shown in FIG. **1,** may be operated as a motor by a pressurization of the oil through the operation of pump **40** to cause the vaned rotary element to rotate at speeds greater than the free windmilling speed. It may be seen that such rotation would increase the inlet weight flow to more than that obtained with the freely rotating windmilling element **25.** Also, the hydraulic pump or motor **35** could be operated in a reverse direction by suitable pressurization of the system **39** with pump **40.** This action would cause the element **25** to rotate in a reverse direction and decrease the inlet weight flow in comparison to that resulting from a stationary vaned rotary element. With an impeller having the same value of as before, the same efficiency of impeller operation would result.

Using the well-known Euler energy loss equation,

$$E = \frac{V^2_a}{g} \left[\frac{\tan}{\tan} \quad \tan^2\right],$$

the theoretical power loss may be computed for a specific design. FIG. **6** gives such power loss by a vaned rotary element **25** using air at various flow velocities, V_a over a flow ratio range achieved by changing the vaned rotary element speed, U_r, for a construction using = 50 and = 38.5. The horsepower loss equation for one pound per second flow is:

$$HP = \left[\frac{U_r V_a}{\tan} \quad U^2_r\right] \Big/ \ 550 \ g$$

FIG. **7** shows how the vaned rotary element speed changes with weight flow reduction in order to hold constant.

From the foregoing description of the construction and theory of operation, it is evident that control of the speed of the vaned rotary element mey be used to rapidly and efficiently change the mass flow through a compressor or pump impeller which is operating at a relatively constant speed. As previously pointed out, the valve **41** and switch **42,** or other controls for the motor of pump **40,** may be remotely located and integrated as desired into an overall control system.

I claim:

1. Flow control means for compressors and pumps of the type having a casing with an inlet, an outlet, and an impeller supported for rotation between said inlet and outlet to draw fluid into said inlet and discharge it from said outlet, said flow control means comprising:

a. guide means mounted in the casing between said impeller and the inlet, said guide means being supported for rotary movement and having vanes engaged by fluid flowing to said impeller to impart direction to such fluid; and

b. fluid pressure responsive means disposed exteriorly of said casing for controlling the rotary movement of said guide means to obtain a predetermined speed ratio range relative to said impeller.

2. The flow control means of claim **1** in which the guide means has an annular collar between the ends of the vanes and the hub, said collar being conical with the base of the cone facing toward the casing inlet.

3. The flow control means of claim **1** in which the means for controlling the rotary movement of said guide means has a fluid pressure energy transfer system associated therewith.

4. The flow control means of claim **3** in which the fluid pressure energy transfer system has a reversible fluid pump connected for rotation with the guide means and means for controlling fluid flow to and from the pump.

5. The flow control means of claim **4** in which the reversible fluid pump is of the positive displacement type and the fluid flow control means has a selectively operable valve.

6. The flow control means of claim **1** in which the impeller of the compressor is a centrifugal type and the guide means has a hub in axial registration with the impeller, vanes extending radially outwardly from the

hub, said vanes being curved from the leading to trailing edges with the exit angle between thirty and sixty degrees, a flow guide collar extending from the leading to the trailing edges of said vanes between the ends of said vanes and said hub, said collar decreasing in diameter between the leading and trailing edges of said vanes, the means for controlling rotary movement of said guide means having a shaft projecting forwardly from said hub, fluid energy transferring means at the end of said shaft, and a fluid system communicating with said fluid energy transferring means.

7. The flow control means of claim 6 in which the fluid system communicating with said fluid energy transferring means has a valve for governing flow to and from said energy transferring means and means for moving fluid under pressure to said fluid energy transferring means.

* * * * *

UNITED STATES PATENT OFFICE

CERTIFICATE OF CORRECTION

Patent No. 3,918,828 Dated November 11, 1975

Inventor(s) Emerson L. Kumm

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 3, Line 20, 21
and 4. From FIG. 4 it will be noted that the exit angle, Θ, of the blades of the guide, i.e. the angle of the blade

Column 4, Line 14
tary guide element blade is represented by U_v in the

Column 4, Line 20, 21
exit angle, Θ, of the blade 30 of the guide element and the inlet angle, β, of the pump or compressor blade, specify

Column 4, Line 23
struction. An angle of attack, γ, is designated as used in

Column 4, Line 30, 31
necessary that the angle, $\beta-\gamma$, not vary by more than 3 to 5 degrees in either direction for efficient operation, as is well

Column 4, Line 50
angle, $\beta-\gamma$, it is necessary that the axial flow velocity, V_a,

Column 4, Line 63, 64
design angle, $\beta-\gamma$, may be specified relative to the exit angle, θ, of the blade 30 of the rotary element 25 and the

Column 4, Line 66
flow ratio limit, W_p/W, is defined as the ratio of the inlet

Column 5, Line 6-10

$$W_p/W = \left[1 + U_v/U_c\right] \Big/ \left[1 + \frac{\tan(\beta-\gamma)}{\tan\Theta}\right]$$

UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,918,828 Dated November 11, 1975

Inventor(s) Emerson L. Kumm

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 5, Line 13-30

(1) $W = \rho A V_a$

With prewhirl

$$\tan \Theta = V_a / (U_v + V_p)$$

(2) $$V_a = (U_v + V_p) \tan \Theta \ , \ V_p = \frac{V_a}{\tan \Theta} - U_v$$

$$\tan (\beta - \gamma) = V_a / \left[U_c - V_p\right]$$

(3) $$V_a = \left[U_c - V_p\right] \tan (\beta - \gamma)$$

Let Eq^n (2) = Eq^n (3)

$$(U_v + V_p) \tan \Theta = (U_c - V_p) \tan (\beta - \gamma)$$

Solve for V_p,

$$V_p \left[\tan \Theta + \tan (\beta - \gamma)\right] = U_c \tan (\beta - \gamma) - U_v \tan \Theta$$

$$V_p = \left[U_c \tan (\beta - \gamma) - U_v \tan \Theta\right] / \left[\tan \Theta + \tan (\beta - \gamma)\right]$$

Sub. in Eq^n (1) and (2)

(4) $$W_p = \rho A \left[U_v \tan \Theta + \frac{U_c \tan (\beta - \gamma) - U_v \tan \Theta}{1 + \frac{\tan (\beta - \gamma)}{\tan \Theta}}\right]$$

If $V_p = 0$, zero prewhirl

UNITED STATES PATENT OFFICE

CERTIFICATE OF CORRECTION

Patent No. 3,918,828 Dated November 11, 1975

Inventor(s) Emerson L. Kumm

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 5, Line 30-55

By Eq^{n} (2), $V_a = U_v \tan\Theta$

By Eq^{n} (3), $V_a = U_c \tan(\beta-\gamma)$

(5) $W = \rho A V_a = \rho A U_c \tan(\beta-\gamma)$, Constant angle of attack.

Dividing (4) by (5)

$$(6)\quad W_p/W = \left[U_v/U_c\right] \frac{\tan\Theta}{\tan(\beta-\gamma)} + \frac{\left[1 - \dfrac{U_v}{U_c}\dfrac{\tan\Theta}{\tan(\beta-\gamma)}\right]}{\left[1 + \dfrac{\tan(\beta-\gamma)}{\tan\Theta}\right]}$$

Note that from Eq^{n} (2)

$$(7)\quad \tan\alpha = V_p/V_a = \frac{1}{\tan\Theta} - U_v/V_a$$

Eq^{n} (6) may be simplified to

$$(8)\quad W_p/W = \left[1 + U_v/U_c\right] \Big/ \left[1 + \frac{\tan(\beta-\gamma)}{\tan\Theta}\right]$$

UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,918,828 Dated November 11, 1975

Inventor(s) Emerson L. Kumm

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 6, Line 1

With an impeller having the same value of $\beta - \gamma$ as before

Column 6, Line 4-8

$$E = \left[V_a^2 / g\right] \left[\frac{\tan\alpha}{\tan\theta} - \tan^2\alpha\right]$$

Column 6, Line 13, 14

Rotary element speed, U_v, for a construction using θ = 50 degrees and $\beta - \gamma$ = 38.5 degrees. The horsepower loss equation for one

Column 6, Line 16-20

$$HP = \left[\frac{U_v V_a}{\tan\theta} - U_v^2\right] \Big/ 550\, g$$

Column 6, Line 22

Changes with weight flow reduction in order to hold $\beta - \gamma$

Column 6, Line 30

the valve 41 and switch 42', or other controls for the

Signed and Sealed this

eleventh Day of *May 1976*

[SEAL]

Attest:

RUTH C. MASON
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents and Trademarks