March 12, 1968 C. F. KOENIG III 3,372,862

CENTRIFUGAL COMPRESSOR

Filed Oct. 22, 1965 4 Sheets-Sheet 1

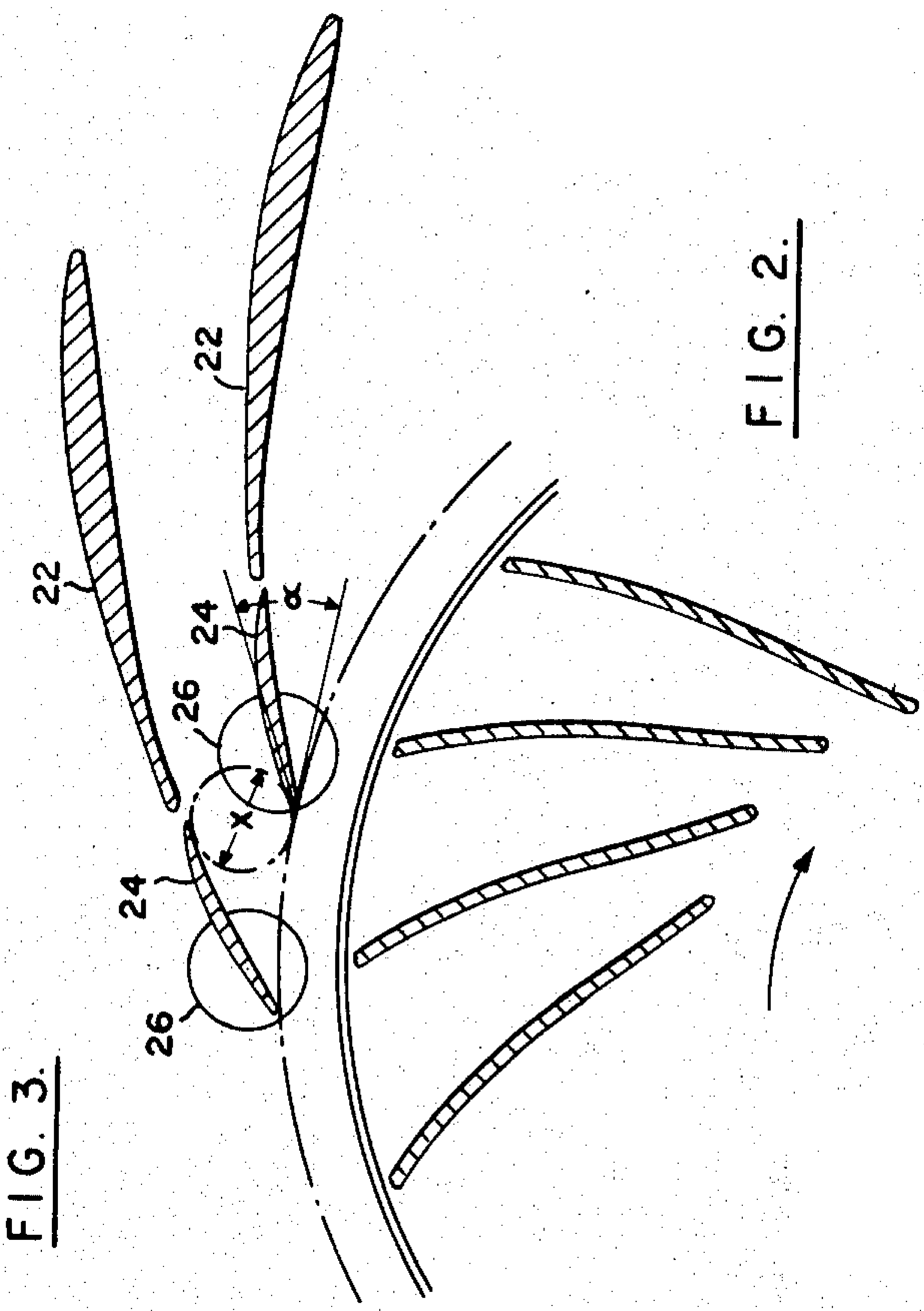

FIG. 2.

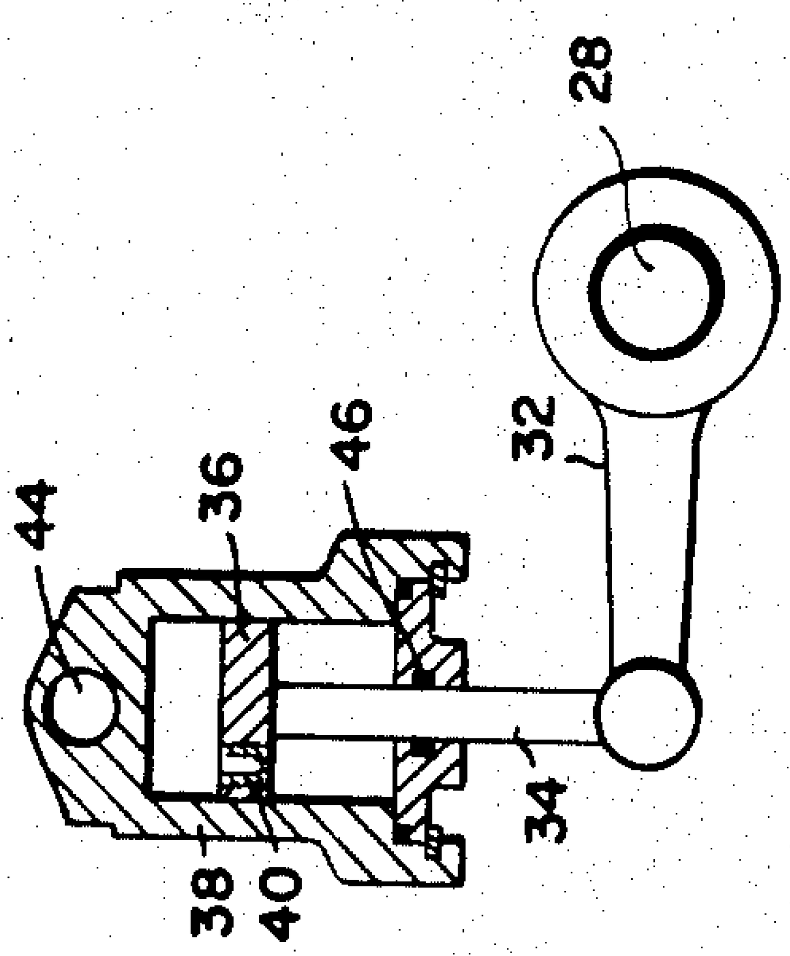

FIG. 3.

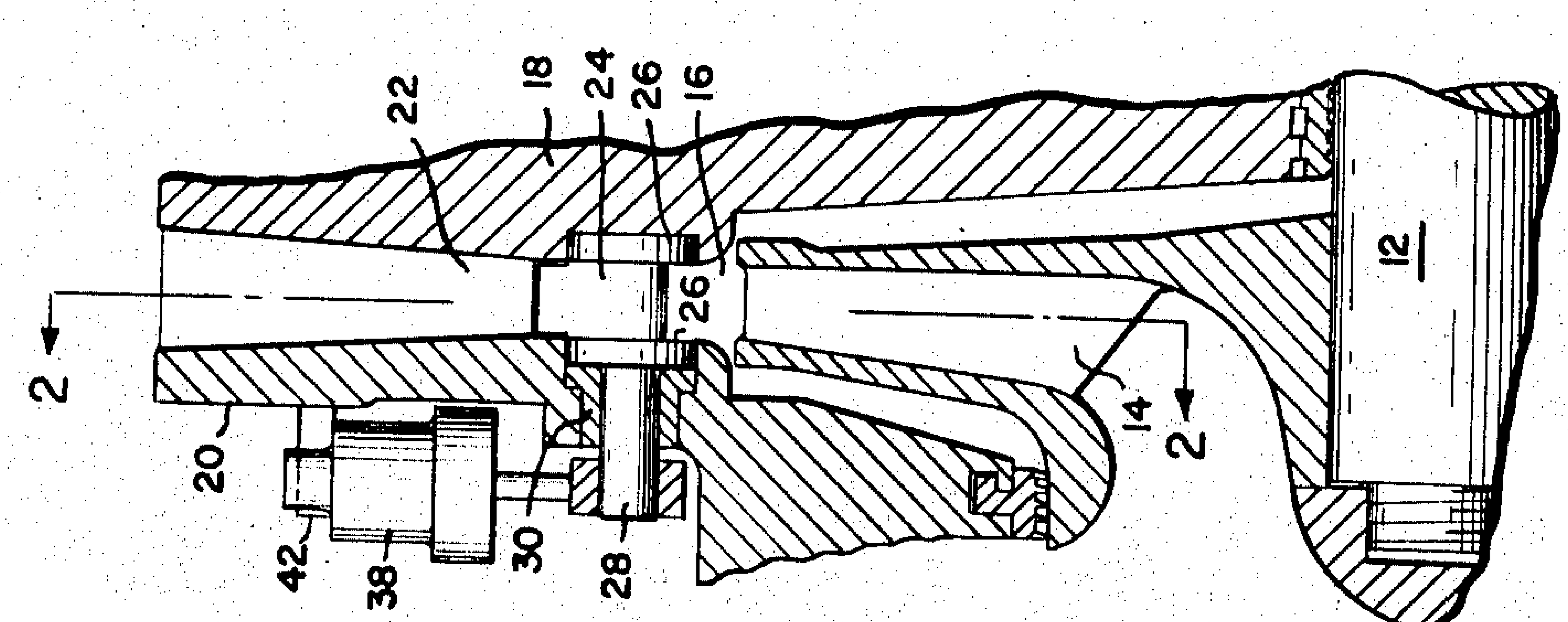

FIG. 1.

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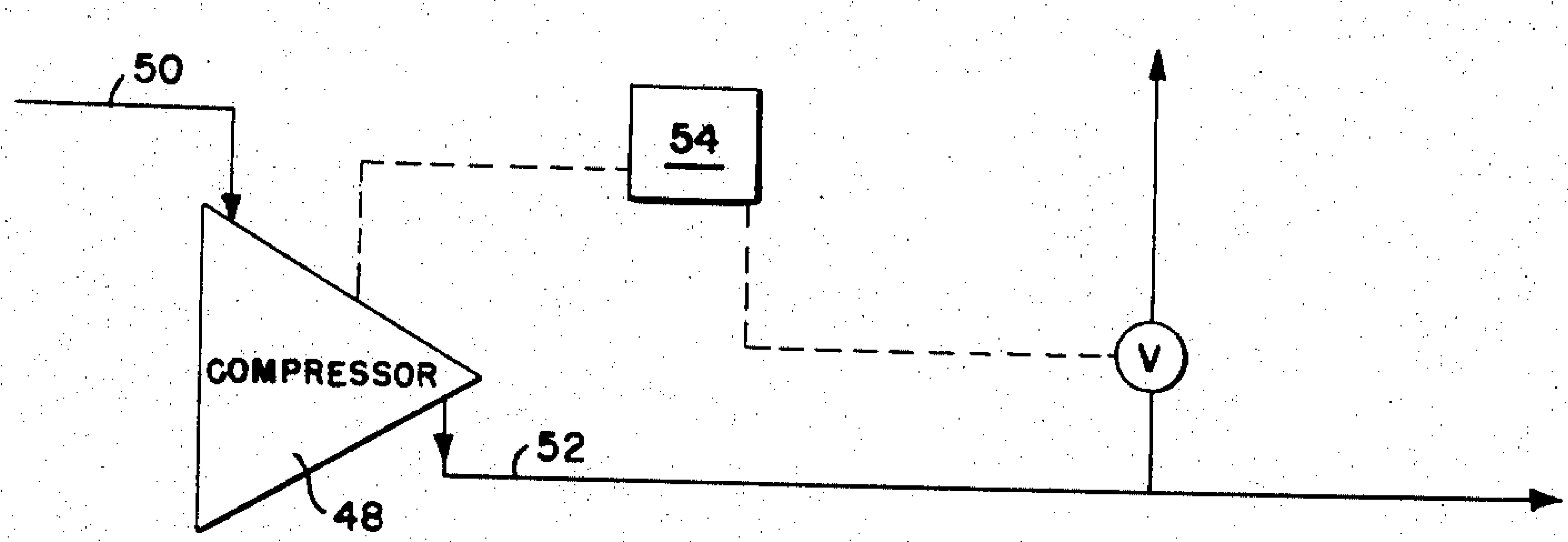

FIG. 7.

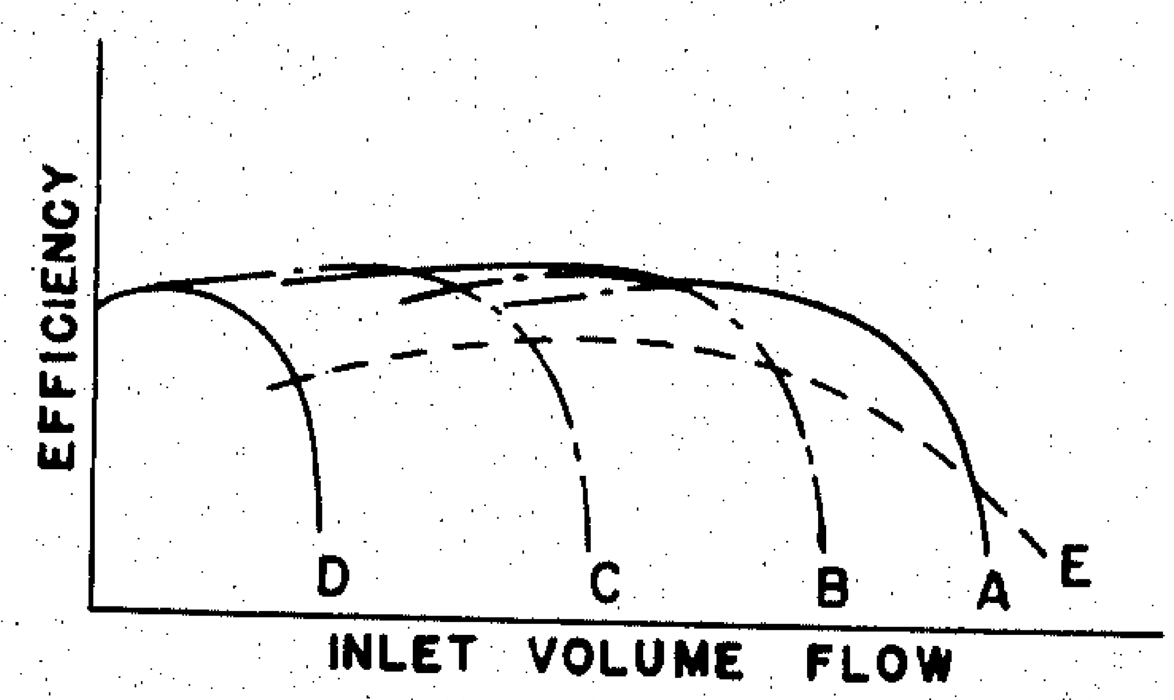

FIG. 4.

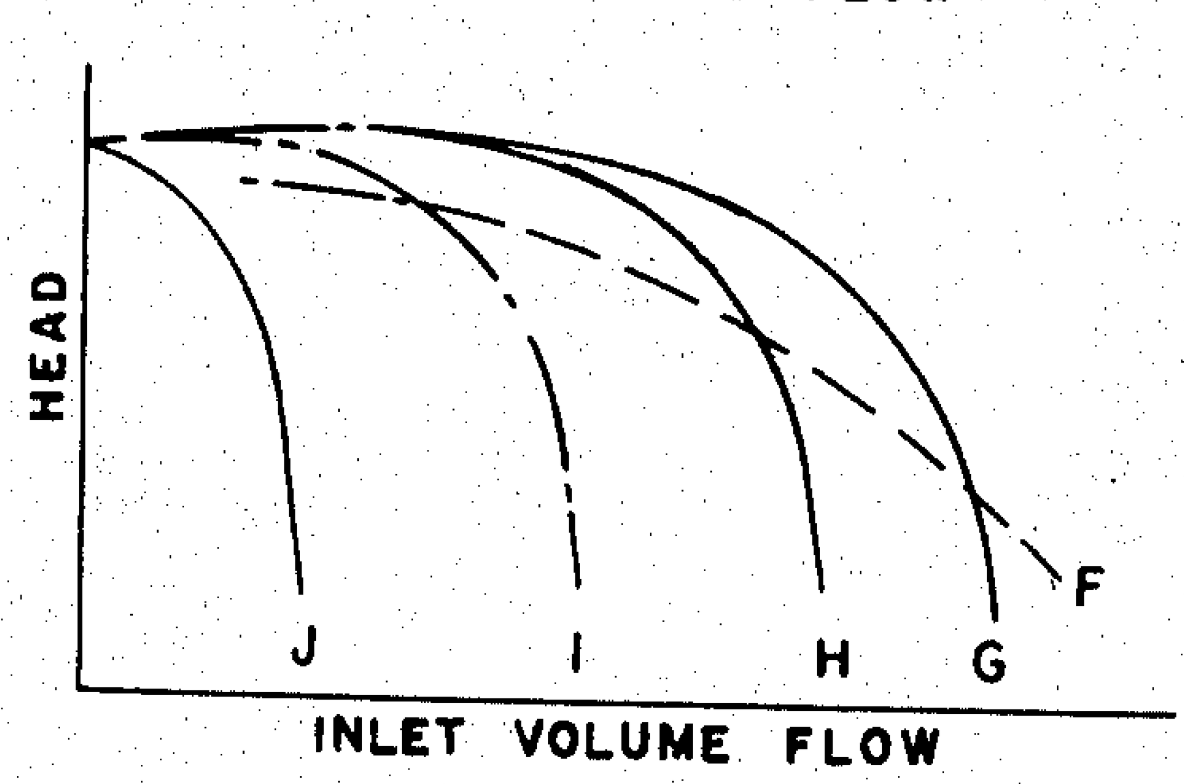

FIG. 5.

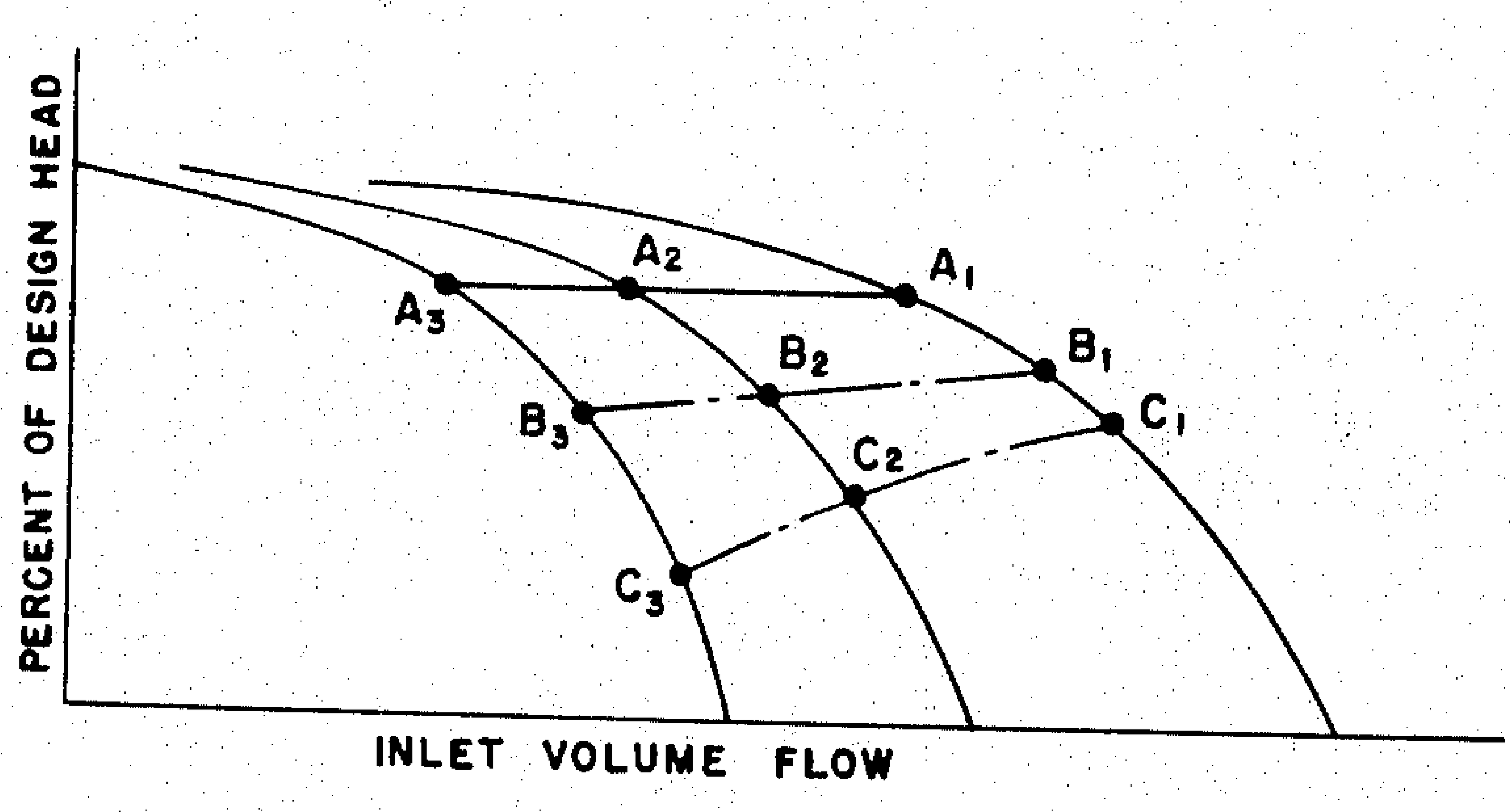

FIG. 11.

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CENTRIFUGAL COMPRESSOR

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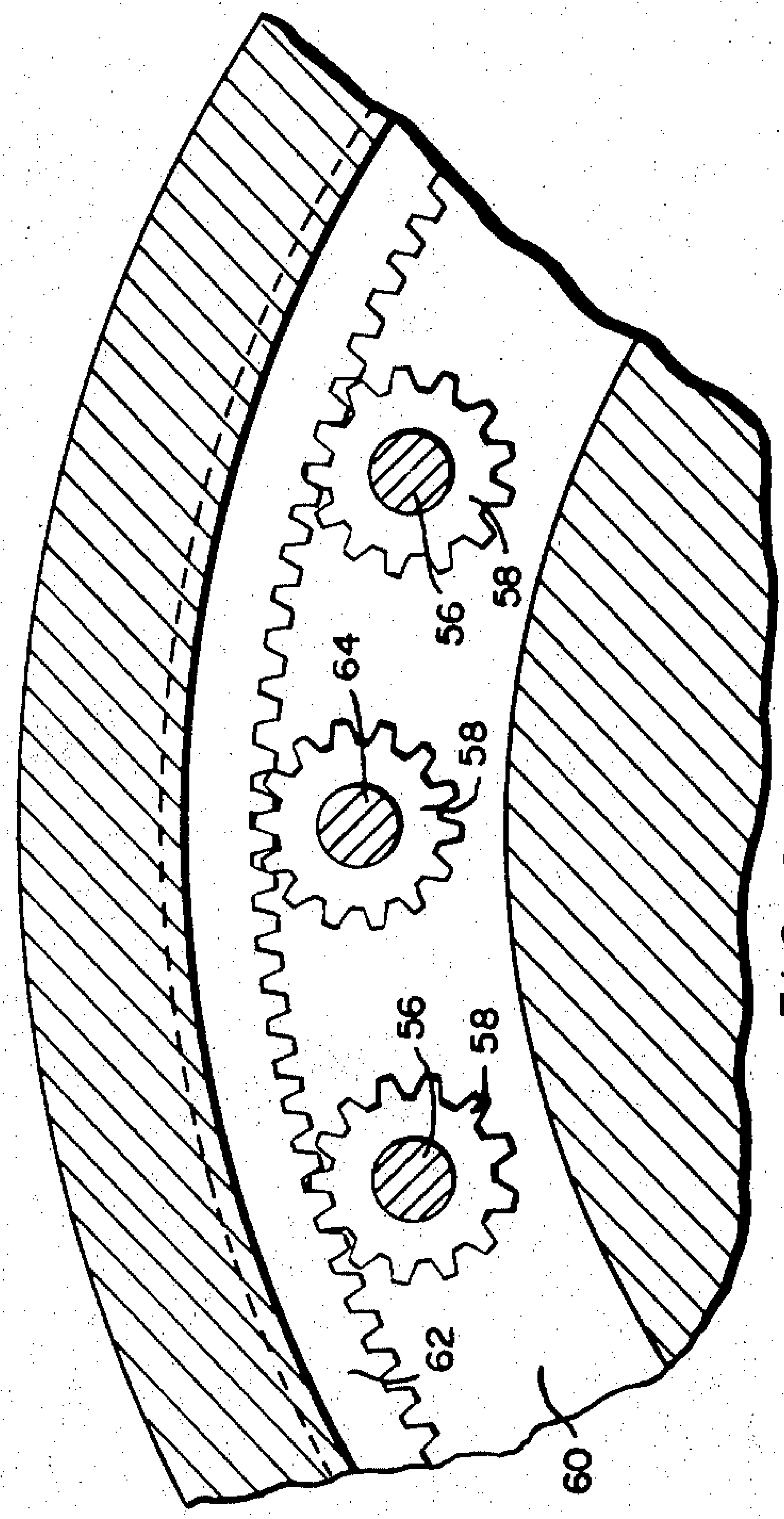

FIG. 10.

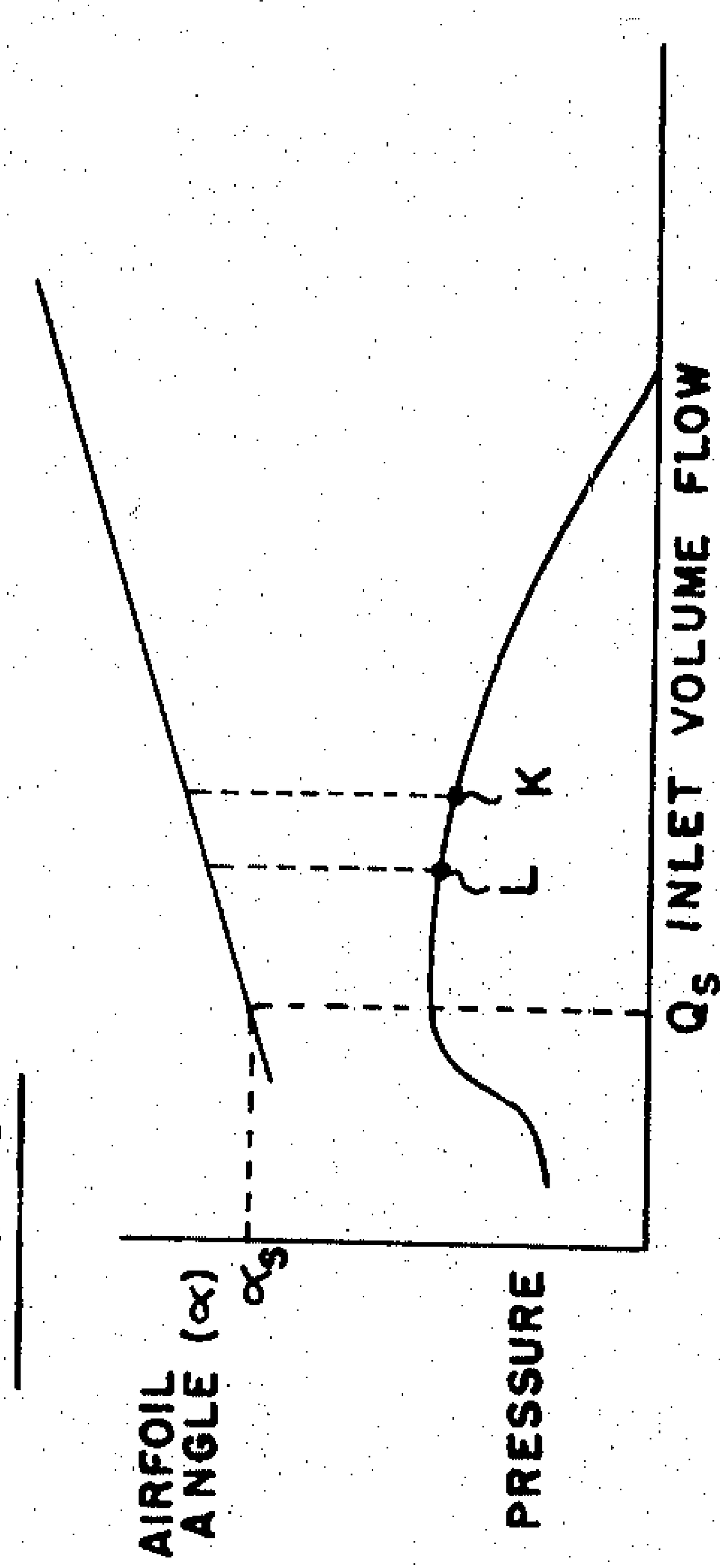

FIG. 8.

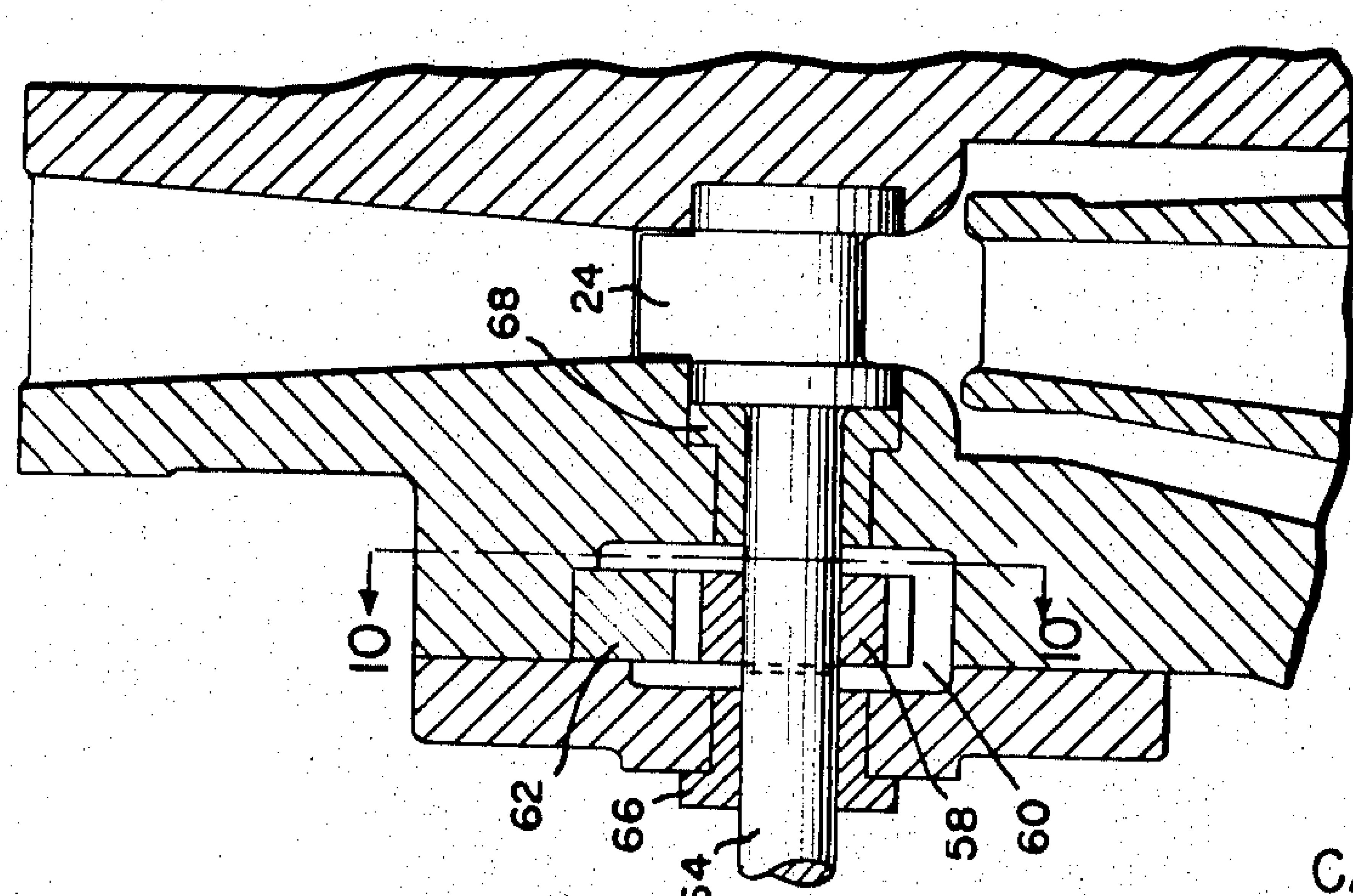

FIG. 9.

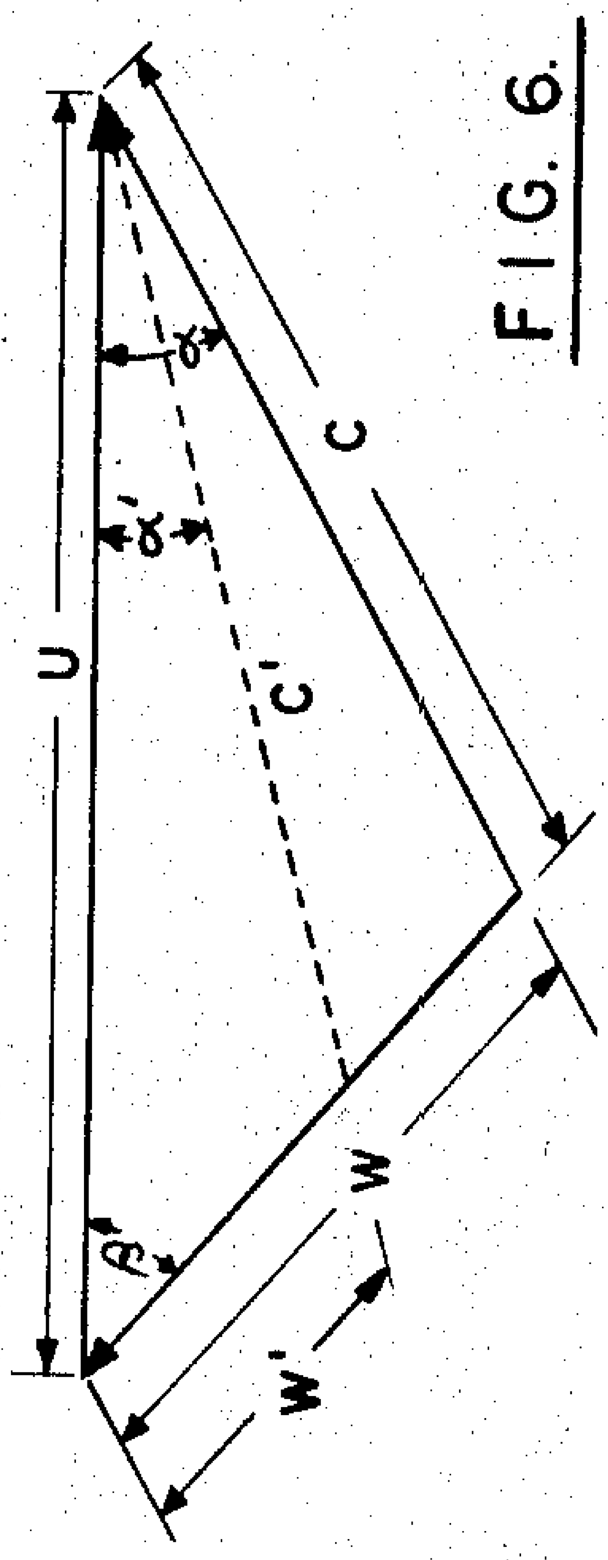

FIG. 6.

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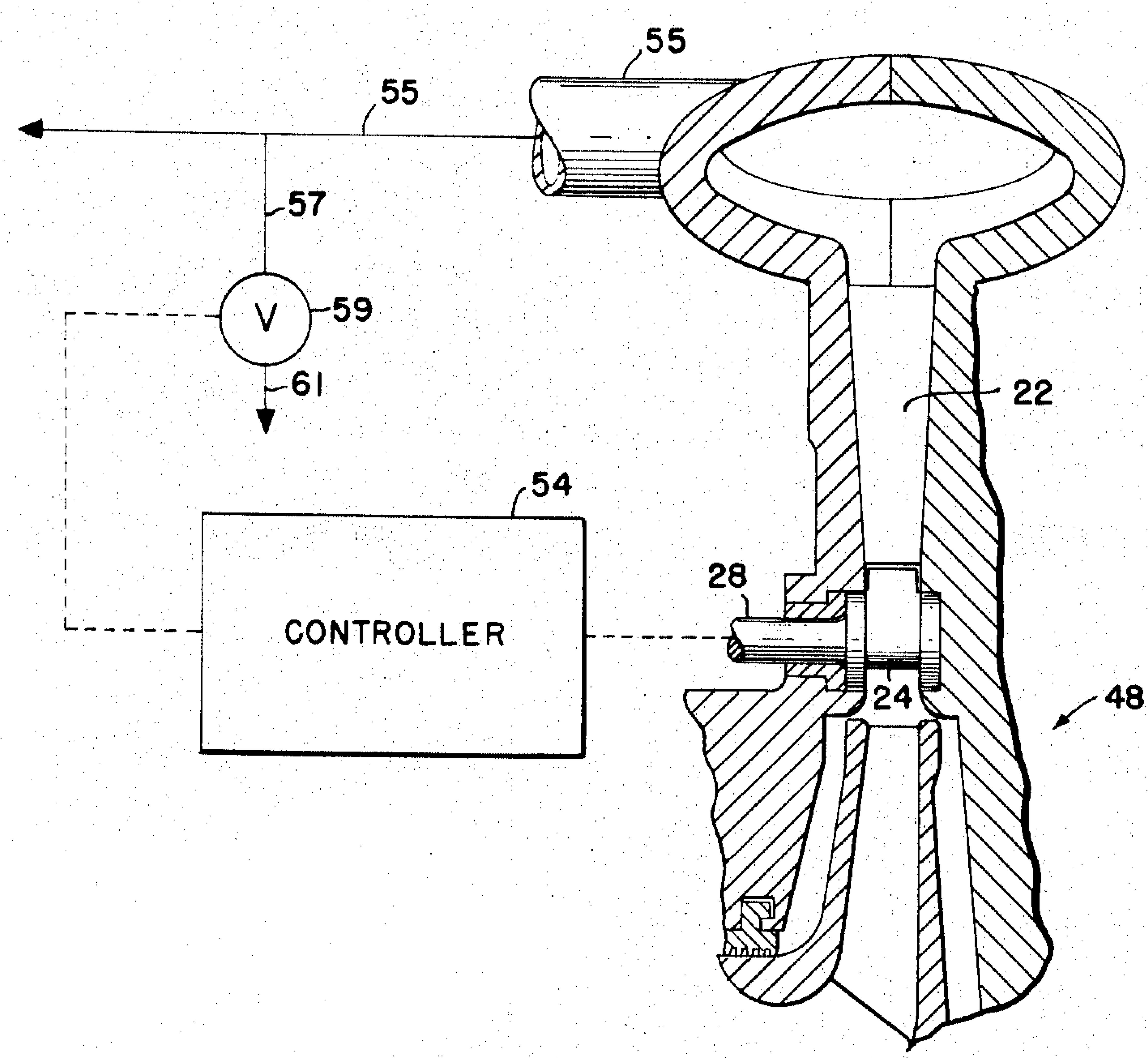

FIG. 12.

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CENTRIFUGAL COMPRESSOR

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2 Claims. (Cl. 230—114)

ABSTRACT OF THE DISCLOSURE

A centrifugal compressor is provided with fixed and movable diffuser vanes disposed around the periphery of its impeller. The movable diffuser vanes are in the form of airfoils pivoted and free to rotate about axes in order to provide variable throat areas between the leading and trailing edges of adjacent vanes. The vanes are pivoted so that this throat area is directly proportional to flow rate and inversely proportional to absolute discharge velocity so that optimum pressure and efficiency are obtained. The movable vanes may be provided with dampers, and may be connected to control an outlet by-pass valve in order to prevent the occurrence of a surge condition.

This invention relates to centrifugal compressors, and particularly to those having movable diffuser vanes.

In compressors having constant impeller speeds, the performance characteristics for a given flow rate can be improved considerably by the vaned diffusers. For a given flow rate, both the efficiency and the discharge pressure can be increased by the use of vaned diffusers. Compressors with vaneless diffusers, of course, are capable of operating over a wider range of flow rates with a more or less constant efficiency and a more or less flat pressure-flow characteristic over a wide range of flow rates. Compressors having fixed vane diffusers, on the other hand, can operate with high efficiency and against a higher head, but the range of flow rates which they are capable of handling is severely limited since they have a high surge flow and a very low maximum flow.

The efficiency-flow and the pressure-flow characteristics of a centrifugal compressor can be varied by providing adjustable diffuser vanes, and the efficiency and pressure can be optimized for a given flow rate.

It is the general object of this invention to provide a centrifugal compressor having adjustable vanes, in which the adjustment occurs automatically as flow conditions change.

In centrifugal compressors, where losses in the impeller and in the diffuser become high enough so that the pressure varies directly rather than inversely with the flow rate, a surge condition will occur. Before this condition occurs, it is desirable to by-pass the output of the compressor to direct it away from the process being supplied by the compressor. The movable vanes which accomplish automatic adjustment of the characteristics of the compressor are also capable of detecting the onset of a surge condition. Accordingly, a further object of the invention is to provide a control whereby the output of a compressor is directed away from the process which is supplied by the compressor for the duration of a surge condition.

A still further object of the invention is to provide positive means whereby the delivery of a centrifugal compressor can be adjusted.

Other objects will be apparent from the following description read in conjunction with the accompanying drawings in which:

FIGURE 1 is a section of a centrifugal compressor stage showing a movable vane in accordance with the invention;

FIGURE 2 is a partially diagrammatic section taken on the plane 2—2 of FIGURE 1;

FIGURE 3 is a sectional view of a damping mechanism in accordance with the invention;

FIGURE 4 is a diagram showing efficiency vs. flow curves for a compressor employing a vaneless diffuser and for a compressor employing an adjustable vaned diffuser in various positions of adjustment;

FIGURE 5 is a diagram illustrating pressure-flow characteristics for a compressor employing a vaneless diffuser and for a compressor employing an adjustable vaned diffuser in positions of adjustment corresponding to the diffuser positions from which FIGURE 4 was obtained;

FIGURE 6 is a vector diagram illustrating the operation of the automatically adjustable diffuser vanes in accordance with the invention;

FIGURE 7 is a diagram of a control system for redirecting the output of a compressor in the event of a surge condition;

FIGURE 8 is a diagram illustrating the variation of diffuser vane angle with flow rate and the variation of outlet pressure with inlet volume flow of a compressor equipped with the control system illustrated in FIGURE 7;

FIGURE 9 is a section of a centrifugal compressor stage provided with a positive means for adjusting the position of a diffuser vane in accordance with the invention;

FIGURE 10 is a sectional view of the positive diffuser vane adjusting means taken on the plane **10—10** of FIGURE 9;

FIGURE 11 is a diagram showing the stage head-volume characteristic for a typical three-stage compressor which is not provided with the automatically adjustable diffuser vanes in accordance with the invention; and

FIGURE 12 is a partially diagrammatic illustration of a centrifugal compressor and a system for redirecting its output in the event of a surge condition.

Referring to FIGURE 1, a driving shaft **12** is shown on which is mounted in a suitable manner a radial impeller **14**. A passage **16** is formed between housing members **18** and **20**, and within this passage there is provided a plurality of stationary diffuser vanes **22** spaced from one another uniformly around the circumference of the compressor stage.

Adjacent the leading edge of each of the fixed diffuser vanes **22**, there is provided a movable diffuser vane **24** fixed to a pair of cylindrical members **26** mounted to rotate in bearings provided in housing members **18** and **20** so that vanes **24** can rotate freely about their respective axes parallel to the axis of rotation of drive shaft **12**. Vanes **24** are in the form of highly cambered airfoils, and are pivoted so that the axes about which they rotate pass through them near and parallel to their leading edges. The distance between the leading edge and the axis of rotation is desirably from 20 to 25% of the chord length of the airfoil. The axis of rotation of each of the airfoils **24** passes through the centers of its corresponding member **26**. The pivot position with respect the airfoil is a design parameter and determines the airfoil position as a function of the flow rate.

Each movable diffuser vane **24** is fixed to a shaft **28** which passes through a bearing **30** in housing member **20**. Referring to FIGURES 1 and 3, shaft **28** is suitably keyed to a crank arm **32**, which is pivoted at its other end to a shaft **34** driving a piston **36** within a dashpot **38**. Dashpot **38** is desirably oil-filled, and a restricted orifice **40** in piston **36** provides a strong damping force to resist rapid movement of the diffuser vanes **24**.

In this embodiment of the invention the dashpot 38 is provided for each movable van 24, and each dashpot is mounted by means of a pin 42 fixed to housing member 20 and passing through a hole 44 provided at the top of each dashpot. Suitable seals 46 are provided at the opening through which piston rod 34 passes. The attitude of the vanes 24 is affected by the direction of flow of the stream entering the diffuser, which, in turn, is affected by the quantity of flow. Vanes 24 tend to line up with the direction of the stream entering the diffuser, and thus the attitude is dependent on the quantity of flow. The degree of damping and the rate of movement of an airfoil for a given torque can be adjusted by altering the size of the orifice 40 in the associated dashpot.

The relationship between the position of the movable vanes and the flow will best be understood from reference to FIGURES 2 and 6. α is an angle representing the direction of flow. U represents the peripheral speed of the impeller and is constant as long as the impeller speed is unchanged. W is the relative velocity vector and varies in magnitude directly as the rate of flow, but remains at essentially a constant angle β with the peripheral component U. C is the absolute discharge velocity.

As the rate of flow decreases, W decreases to W', α decreases to α' and C, the absolute velocity, changes both in magnitude and direction to C'. To obtain optimum performance, the diffuser throat area should now be decreased by the ratio $Q'C/QC'$ where Q is the original flow rate, and Q' is the new flow rate. The throat area of the diffuser is represented by the diameter X of the circle interposed between the trailing edge of one vane 24 and the leading edge of an adjacent vane.

The pivot axes of the vanes 24 are located so that the change in the diffuser throat area varies according to the above relationship. The position of the pivot axis which accomplishes this relationship is ordinarily at a distance between 20 to 25% of the chord length of the airfoil from its leading edge. It will be apparent that, as flow increases, the diffuser throat area increases, and as flow decreases, the area decreases.

Reference should now be made to FIGURE 4 which shows typical curves representing efficiency vs. flow. Curve E represents the characteristic of a typical vaneless diffuser. Each of curves A, B, C and D represents the characteristic of a compressor having a vaned diffuser in which the vanes are in various positions of adjustment. It will be apparent that the peak of each efficiency vs. flow curve is in a different position, and that if the characteristics of a compressor were made continuously variable according to the flow, the compressor can be made to operate near maximum efficiency throughout a wide range of flow rates. The maximum efficiency is considerably above that obtainable from a compressor having a vaneless diffuser.

The automatic adjustment of the diffuser throat area described above changes the characteristic efficiency vs. flow curve in accordance with the flow rate so that, at high rates of flow, the characteristic curve is, for example, the position of curve A corresponding to high rate of flow since the diffuser throat area is relatively large. At low rates of flow, the diffuser throat area automatically becomes small, and consequently the effective efficiency curve is, for example, curve B. The resulting effective efficiency curve is, therefore, the envelope of the individual curves.

FIGURE 5 shows a pressure-flow characteristic F for a typical compressor having a vaneless diffuser for the purpose of comparison, and curves G, H, I and J which represent the pressure-flow characteristics corresponding to different adjustable vane positions. The position of the vane represented by curve G corresponds to the vane position represented by curve A in FIGURE 4, H corresponds to B, I to C and J to D. By the present invention, the compressor is made to operate on the envelope of the pressure-flow characteristics. Pressure is therefore made practically independent of flow rate, the envelope having an approximately flat portion over a large range of flow axis.

An off-design flow rate, the change in attitude of the movable vanes 24 will result in an opening between the trailing edges of the vane and the leading edges of the corresponding stationary diffusers. These openings form effective boundary layer control slots. Flow separation is retarded, and the performance of the stationary diffuser passage at off-design flows is greatly improved over that to be expected with conventional adjustable diffusers.

Since the movable vanes change their attitude with flow rates, they are capable of detecting the onset of the surge condition in the compressor and of controlling operation to prevent surge as follows:

Referring to FIGURE 7, the symbolized compressor 48 is provided with an inlet 50 and an outlet 52 leading to a process. A controllable, normally closed by-pass valve V is provided to direct the discharge of the compressor away from the process for the duration of the surge condition. A suitable valve controller 54 is provided with the signal obtained from and corresponding to the position of a vane 24 in the compressor. The signal might, for example, be obtained from a potentiometer (not shown) driven by shaft 28 to which the movable vane is attached. The controller 54 is desirably continuous in its operation so that the degree of opening of valve V corresponds to the attitude of the adjustable vane driving the controller 54. Such controllers are well-known in the art and need not be described in detail.

In FIGURE 12, the connection of the controller and by-pass valve to the compressor are shown. Compressor 48 is provided with a fixed diffuser vane 22, and a movable vane 24 adapted to rotate shaft 28. Outlet passage 55 from the compressor is shown partly diagrammatically and line 57 connects line 55 through valve 59 to an additional outlet line 61. Controller 54, which is of the well-known servo type, is arranged to open valve 59 proportionately in correspondence with the position of shaft 28 so that, when vane 24 is in a position corresponding to the onset of a surge condition in the compressor, valve 59 is opened in order to provide an additional outlet for gas from the compressor to counteract the surge condition.

Referring to FIGURE 8, a typical pressure-flow characteristic is shown, and the surge point is indicated at Q_s. The corresponding vane angle vs. flow curve is shown. The controller is designed so that valve V begins to open when the flow reaches point K, and is fully opened when the flow reaches point L at the onset of the surge condition. The surge condition occurs at a flow rate Q_s, and the vane angle at this time is α_s.

(It will be apparent that various suitable and known electric, hydraulic or pneumatic control devices can be used to control valve V in response to variations in the position of the movable vane.)

It is often desirable to provide positive means within a compressor for adjusting the delivery independently of the process supplied by the discharge of the compressor. An alternative arrangement illustrated in FIGURES 9 and 10 accomplishes this result.

A plurality of movable vanes 24 are fixed to rotatable shafts disposed circumferentially around the periphery of the impeller. On each of these shafts there is suitably fixed a pinion 58 in an annular space 60 provided in the housing. A ring gear 62, also provided in space 60 meshes with each of pinions 58 so that the vanes are no longer capable of moving independently. At least one of pinions 58 is keyed to a control shaft 64 rotatable within bearings 66 and 68 in the housing. Shaft 64 extends to the exterior of the housing for connection to an external actuator. Thus, by operation of the actuator, the throat area of the diffusers can be varied so that any desired pressure-flow characteristic can be obtained. The actuator of shaft 64 may be made to operate in response to signals corresponding to the process flow, process pressure or any other parameter to which the compressor delivery is related. Alternatively, the position of shaft **64** can be adjusted manually.

The invention, in the embodiment in which the movable diffuser vanes are free to align themselves with the fluid stream, produces particularly advantageous results in multiple stage compressors wherein each of the stages is equipped with movable diffuser vanes in the form of airfoils. The difficulties involved in ordinary multiple stage compressors result from the fact that the stages can be perfectly matched to one another at only one flow rate. These difficulties will be apparent from FIGURE 11 which shows the pressure-flow characteristics for each stage of a typical three-stage compressor with fixed diffusers. The abscissa is the inlet volume flow, and the ordinate represents percentage of the design head. Where the operating point for stage one is A_1, at the design inlet volume flow, the operating points of stages **2** and **3** are A_2 and A_3, respectively. The stages are perfectly matched at these particular operating points, since each stage is operating against its design head. If the inlet volume flow to the first stage is increased so that the operating point is B_1, because of the reduced compression in the first stage, the second stage operates at the point B_2, so that the compression in the second stage is somewhat below that in the first stage, and the compression in the third stage is likewise below that in the second stage. Each of the stages is operating at a different percentage of its design head as is represented by the operating points B_1, B_2 and B_3. As the inlet volume flow to the first stage is further increased, the separation of the operating points of the stages becomes greater, and the mismatching of the stages increases as the inlet volume flow increases.

The problem of mismatching at off design flow rates is avoided in a multiple stage compressor in which each of the stages is equipped with the self-adjusting diffuser vanes of the invention, since each of the stages will have a broad operating range. Since the operating pressure is made less dependent on flow rate by the self-adjusting diffuser vanes, the stages will remain matched over a broad range of flow rates. The overall performance of a multiple stage compressor is thus greatly improved and the provision of the self-adjusting vanes of the invention is very desirable where inlet volume flow is likely to vary.

Another problem inherent in centrifugal compressors is caused by uneven flow distribution around the periphery of the impeller which results in unequal flow angles (α) in the fixed vane diffusers. The self-adjusting diffusers of the invention adjust themselves to whatever flow angle (α) is required at any point on the periphery of the impeller. Losses due to unequal flow distribution are thus minimized.

The invention is further applicable to compressors comprising a plurality of stages operating in parallel. If the flow rate is not equal in all of the parallel stages, in ordinary compressors the overall efficiency is lowered and the overall operating range is reduced. Where self-adjusting diffuser vanes are provided in each of the stages, each stage adjusts itself to optimum performance,

and the overall performance is not affected by small differences in flow in the paralleled stages.

It will be apparent that the invention is not limited to use in conjunction with closed radial impellers and that it may be applied to compressors having various types of impellers and various configurations other than those disclosed. It will also be apparent that various modifications can be made to the invention without departing from its scope as defined in the following claims.

What is claimed is:

1. In combination, a centrifugal compressor having inlet and outlet passages for each of its stages, a rotatable vane in the form of an airfoil disposed in at least one of said outlet passages and pivoted near its leading edge so that it aligns itself with the direction of flow of gas through said one of said outlet passages, control means responsive to the position of said rotatable vane and valve means operable by said control means and connected to provide an additional path for the discharge from said compressor when flow of gas through said one of said outlet passages is in the direction accompanying a surge condition.

2. A centrifugal compressor comprising at least one stage having inlet and outlet passages, an impeller disposed within said stage for pumping a gas introduced at said inlet passage through said outlet passage and producing a substantial radial component of flow through said outlet passage, a plurality of vanes disposed within said outlet passage circumferentially about said impeller means, said vanes being in the form of airfoils, the trailing portion of each said vane cooperating with the leading edge of the next adjacent vane to provide a variable throat area for the escape of gas through said outlet passage, and means pivoting said vanes for free rotation about individual axes solely in response to the flow over said vanes, said pivoting means being positioned with respect to said vanes so that the distance between the leading edge of each of said vanes and its axis of rotation is between 20 and 25 percent of the chord length of the vane, whereby said throat area varies directly with discharge flow rate and inversely with absolute discharge velocity.

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