LOW SPECIFIC SPEED COMPRESSOR

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Field of Search 415/212, 213, 416/186, 184, 416/183, 181, 179

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ABSTRACT

Wedge-type vanes, or vanes with increasing thickness from inlet to exit facilitate the layout of aerodynamically favorable flow channels in application to small diameter low specific speed compressor wheels. The channels may have various cross sections including rectangular, square, oval or circular. The channels may be backwardly bent, straight, or s-shaped in form. Streamlining the pressure side of the vanes at the exit helps to reduce exit wake losses. A local restriction may be provided in the channel axial width at the exit to correct for the area increase created by the streamlining at the exit. An extension at the trailing side of the exit to follow the shape of the streamlined pressure side of the following vane may be provided for better control of the channel exit area and better direction of the flow.

2 Claims, 10 Drawing Figures
LOW SPECIFIC SPEED COMPRESSOR

BRIEF SUMMARY OF THE INVENTION

This invention relates to centrifugal compressors, and particularly to an impeller vane configuration giving advantageous performance in low flow, high head applications of small diameter, low specific speed compressor wheels.

As a matter of background, low specific speed compressor wheels are relatively low speed, low flow, high head radial wheels with long, narrow meridional flow passages. The last stages of multi-stage compressors are examples of low specific speed wheels. The last stages of low flow compressors, in particular, have low specific speed wheels of small diameter.

Specific speed \( N_s \) is defined as:

\[
N_s = \frac{N \sqrt{Q_1}}{H^{3/4}}
\]

where:

\( N \) = speed in RPM;
\( Q_1 \) = capacity at inlet conditions in CFM; and
\( H \) = adiabatic head in feet.

Mixed flow and axial flow wheels are typically operated in the high specific speed range, while radial flow wheels are typically operated in the lower ranges of specific speed.

Low specific speed wheels exhibit inherently low efficiency as a result of relatively high leakage, disc friction and channel friction losses.

The cross sections of the relatively long, narrow flow passages of conventional low specific speed wheels have small hydraulic diameters \( d_h \) and give rise to low "channel Reynolds numbers" \( Re_{ch} \) defined respectively as follows:

\[
d_h = 4A/P
\]

where:

\( A \) is the area of the channel cross section, defined as \( A = b_d \), (see FIGS. 1 and 2); and \( P \) is the perimeter of the channel cross section defined as \( P = 2 \left( b + d \right) \), (see FIGS. 1 and 2.)

\[
Re_{ch} = \frac{w \cdot d_h}{\nu}
\]

where:

\( w \) = relative velocity in f.p.s.;
\( d_h \) = hydraulic diameter in ft.; and \( \nu \) = kinematic viscosity in ft. \( ^2 \)/sec.

It should be recognized that a value of hydraulic diameter for a given wheel is merely a representative value, since hydraulic diameter varies somewhat from inlet to exit.

The low values of \( d_h \), the relatively long channel length \( L \), and the high values of the friction coefficient \( \lambda \), in conventional low specific speed wheels give rise to high friction losses in the impeller channels as can be seen from the following equation for head loss:

\[
H = \frac{\lambda L/d_h}{W^2/2g} \cdot W^2/2g
\]

where:

\( H \) = loss of head in feet
\( \lambda \) = friction coefficient (depends on \( Re_{ch} \) and relative wall roughness)
\( L \) = channel length in feet
\( d_h \) = hydraulic diameter of the channel cross section in feet
\( w \) = relative velocity in f.p.s., and
\( g \) = gravitational constant, 32 ft/sec².

The values of \( d_h \) in the conventional low specific speed wheels of small diameter are inherently low.

The values of \( L \) are relatively high to avoid prohibitively narrow impeller passages at discharge leading to flat blade discharge angles \( \beta \) (see FIG. 2) and thus to the necessity for excessively long passages.

The friction coefficient \( \lambda \) is affected by the channel Reynolds number \( Re_{ch} \) in two ways. In very smooth narrow passages, at low channel Reynolds numbers, \( \lambda \) is determined by the Blasius equation; \( \lambda \) increases as \( Re_{ch} \) decreases. In narrow passages having physically rough walls, "relative roughness" varies approximately inversely with \( Re_{ch} \) so that \( \lambda \) again increases with decreasing \( Re_{ch} \).

In accordance with this invention, channel friction losses are reduced by providing flow channels which are of relatively higher axial width "\( b \)" and smaller plan view width "\( d \)" (measured perpendicular to the flow) resulting in higher \( d_h \) values. Shorter channel lengths "\( L \)" can be advantageously used with steeper discharge angles resulting both in further reduction of the friction loss and higher generated impeller head. The higher hydraulic diameters result further in higher \( Re_{ch} \) and lower relative wall roughness. The higher channel Reynolds numbers lead to a lower friction coefficient for one of two reasons: lower \( \lambda \) as determined by the Blasius equation for smooth walls or lower \( \lambda \) for relative roughness of the walls having the same physical roughness. This is preferably accomplished by using "wedge-shaped" impeller vanes arranged so that the vane thickness increases gradually but continuously throughout substantially all the length of the vanes from the inlet to the exit. An aerodynamically more favorable channel can be obtained from the inlet to the exit, but the improvement in the channel cross section is more pronounced at the channel exit where prohibitively low axial widths "\( b \)" and high plan view widths "\( d \)" are avoided.

Wedge-type impeller vanes have been heretofore proposed for relatively high specific speed compressor wheels. However, in normal applications with moderate or high specific speeds, an impeller having wedge-type vanes does not offer any particular advantages over a conventional wheel. On the contrary, it has the disadvantage of high wake losses. In fact, such impellers have lower efficiencies than other more conventionally designed impellers, and are not regarded as having practical value.

The flow channels of impellers in accordance with the invention are aerodynamically superior to those of conventionally designed impellers. The channel Reynolds number \( Re_{ch} \) is higher, the surface of the channel walls is relatively smoother, and the hydraulic diameter \( d_h \) of the channel is larger. As a result, friction losses are lower, leading to a higher impeller efficiency.

Also, in accordance with the invention, the pressure sides of the wedge-type vanes at the exit are streamlined, that is, each vane is provided with a curved contour at its pressure side at the exit. The streamlining reduces the severity of the exit wakes and reduces the losses attributable thereto.

The streamlining of the vanes would ordinarily create an excess area in the flow channel cross sections near the exit. Correction for this excess area can be achieved by reducing somewhat the axial width of the channel near the exit, with a resulting net improvement in efficiency.
The invention accomplishes significant improvements in the performance of compressors having wheels with diameters of less than approximately 20 inches and operated at specific speeds of less than approximately 350.

The principal object of the invention, therefore, is to provide a low specific speed compressor having a small diameter wheel in which friction losses are minimized.

A further object of the invention is to provide a small diameter compressor wheel for low specific speed applications in which friction losses and exit wake losses are minimized, thereby achieving still higher efficiency in the compressor in which the wheel is incorporated.

Other objects of the invention will be apparent from the following description when read in conjunction with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a meridional section illustrating particularly one vane and its associated parts exemplifying the prior art;

FIG. 2 is a fragmentary radial section through the vanes and gas passages of the conventional impeller of FIG. 1;

FIG. 3 is a section corresponding to FIG. 1 but showing an impeller in accordance with one embodiment of the invention;

FIG. 4 is a fragmentary radial section of the impeller of FIG. 3:

FIG. 5 is a section corresponding to FIG. 1 but showing an impeller constructed in accordance with a second embodiment of the invention;

FIG. 6 is a fragmentary radial section of the impeller of FIG. 5:

FIG. 7 is an enlargement of a possible modification of the wheel of FIGS. 3 and 4, showing in detail a restriction of the flow channel at the exit;

FIG. 8 shows a possible modification of the trailing side of the exit edge of the vanes of the wheel shown in FIG. 5;

FIG. 9 is a section corresponding to FIG. 1 showing an impeller constructed in accordance with still another embodiment of the invention; and

FIG. 10 is a fragmentary radial section of the impeller of FIG. 9.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring first to FIGS. 1 and 2, which illustrate the prior art, there is shown a conventional housing 8 with a conventional low specific speed compressor wheel adapted for rotational mounting in the housing. The wheel comprises a hub 10, a shroud 12 and a conventional set of narrow vanes including vanes 14, 16 and 18, vane 14 being shown in both figures as having an entrance edge 19 and an exit edge 21.

From FIG. 1, it will be apparent that the elements just described define long, narrow passages with decreasing width “b” (which, for all practical purposes can be measured in the axial direction for the particular compressor described herein) from inlet to exit.

From FIG. 2 it will be apparent that the plan view channel width “d” (measured perpendicularly to the mean flow line) increases greatly from inlet to exit. As a result of its long flow channel lengths, the inherent low values of the channel Reynolds number $Re_{ch}$, and the low value of hydraulic diameter $d_h$, the wheel just described is subject to high friction losses in its channels.

Each compressor wheel described herein is adapted to be rotationally mounted within a conventional housing to produce a complete compressor. The compressor may be a single stage compressor or any stage of a multiple-stage compressor. A conventional housing is indicated at 22 in FIG. 3, at 23 in FIG. 5, and at 25 in FIG. 9. These figures show a diffuser passage of the kind used in a single stage compressor or in the last stage of a multiple-stage compressor.

FIGS. 3 and 4 illustrate a first form of a compressor wheel in accordance with the invention, which compressor wheel comprises a hub 20 and includes vanes, two of which are indicated at 24 and 26, and a shroud 28. A driving motor is indicated diagrammatically at 27.

As will be apparent from FIG. 3, the flow channel, unlike that in the conventional impeller of FIG. 1, has a large axial width from the inlet to the exit of the impeller and does not become very narrow at any point from the inlet to the exit. Preferably, the axial width is substantially constant from inlet to exit. The flow channel, even though substantially constant in axial width, may have a slight local restriction in its axial dimension near the exit. If warranted by aerodynamic considerations, the channel may be made wider and curved at the inlet in meridional view.

As seen in FIG. 4, each vane gradually but continuously increases in thickness throughout substantially all of its length from inlet to exit, and is generally wedge-shaped. This vane configuration gives rise to flow channels which, in contrast with those of FIG. 2, do not increase as rapidly or do not increase at all in width “d” from inlet to exit. The axial width “b” of the channels, however, in comparison with that of the conventional channel shown in FIG. 1, is made relatively large at least near the exit in order to compensate for the reduction of the width caused by thicker vanes. The resultant flow channels are more nearly constant in axial width “b” and nearly constant in width “d”. These channels, having higher hydraulic diameters, lower relative roughness and higher channel Reynolds numbers are aerodynamically more favorable than channels designed along conventional lines. Friction losses are reduced significantly over those existing in a conventional wheel capable of producing a comparable flow.

Wedge-shaped vanes in themselves provide improved performance because of the reduction of friction loss, but the performance of the compressor wheel having wedge-shaped vanes can be optimized by modifications to the flow channel configuration.

In a wheel having rudimentary wedge-shaped vanes, high wake losses would result from severe wakes set up at the thick exit ends of the vanes. Accordingly, as shown in FIG. 4, the pressure sides of the exit ends of the vanes, as exemplified by pressure side 30 of vane 24 are "streamlined" or provided with a smooth curvature to reduce wake loss.

Streamlining as described accomplishes a significant reduction in wake loss but also increases the flow channel width at and near the exit. This increase would tend to counteract the advantages herebefore described as attributable to the increasing vane thickness. The axial width of the flow channel, however, may be slightly decreased as shown in FIG. 7, from location 32 where widening of the channel due to streamlining begins, to
the exit. The restriction corrects for the increased flow channel width resulting from the streamlining; it may reduce flow, leading to a further reduction of the specific speed at which the impeller can operate at a reasonable efficiency.

It will be recognized that, to achieve improved performance, the radial location at which the restriction begins need not be precisely the location at which widening of the channel due to streamlining begins, but the location should be at least approximately the same in order to achieve effective correction and in order to avoid abrupt changes in the cross sectional area of the flow channel.

FIGS. 5 and 6 illustrate another embodiment of the invention, in which the compressor wheel comprises a hub 34, a shroud 36 and wedge-type impeller including vane 38 and 40. A driving motor is shown at 29.

The flow channels defined by the hub, shroud and vanes have large axial widths from inlet to exit in contrast to the flow channels of the conventional impeller shown in FIGS. 1 and 2. Preferably, the axial width is substantially constant from inlet to exit. The leading edge of each vane at the exit, as illustrated by leading edge 42, is streamlined in a manner similar to the streamlining of the leading edges of the vanes in FIG. 4 and the flow channels may be slightly reduced in width from location 44 to the exit in order to correct for the widening of the flow channel resulting from the streamlining of the leading edges of the vanes.

The vanes are wedge-shaped and gradually but continuously increase in thickness from their inlet to exit ends to provide flow channels which do not increase rapidly in plan view width or do not increase at all in plan view width up to the point at which the streamlining begins.

The wheel of FIGS. 5 and 6 differs from the wheel of FIGS. 3 and 4 primarily in that the flow channel in FIGS. 3 and 4 is curved, while the flow channel in FIGS. 5 and 6 is straight throughout substantially all its length. Whether the flow channel is curved or straight depends on considerations irrelevant to the invention, and the invention is equally applicable to either configuration.

The vanes of FIGS. 5 and 6, being somewhat thicker near the wheel exit than those of FIGS. 3 and 4, are made hollow, as shown in FIG. 6, to reduce the weight of the impeller.

FIG. 8 shows a possible modification to the trailing side of the exit end of a vane. The trailing side 46 is curved backwardly as shown in order to follow the shape of the streamlined pressure side of the following blade. This backward curvature provides a restriction which helps to balance out the excess area created by streamlining the pressure side 48 of the exit end of the following vane, and which provides better control of the direction of flow at the exit. This modification may be used by itself or in conjunction with a restriction such as that shown in FIG. 7.

Finally in FIGS. 9 and 10, there is illustrated a motor 49 driving an impeller 51 having vanes 50 and 52, both of which are generally wedge-shaped, and which are shaped so that the flow passage is curved forwardly at the inlet. Again the flow passages have large axial widths which are preferably substantially constant from inlet to exit, and the vanes gradually but continuously increase thickness from inlet to exit in order to provide flow channels which do not increase rapidly in plan view width or do not increase at all in plan view width.

The inclination of the flow channel with respect to the radial direction may vary widely in the various embodiments of the invention. In the case of the embodiment shown in FIG. 10, this inclination may extend down to zero degrees.

A comparison of the performance of the first three impellers described herein is made in the following table of test results wherein it will be noted that flow is held constant in order to enable a valid comparison of wheel performance in terms of obtainable head and efficiency.

<table>
<thead>
<tr>
<th>Conventional Wheel</th>
<th>Wheel of FIGS. 3 &amp; 4</th>
<th>Wheel of FIGS. 5 and 6 or 300</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow (CFM)</td>
<td>430</td>
<td>430</td>
</tr>
<tr>
<td>Adiabatic Head (ft.)</td>
<td>6950</td>
<td>8500</td>
</tr>
<tr>
<td>Adiabatic Efficiency (percent)</td>
<td>52.0</td>
<td>59.5</td>
</tr>
<tr>
<td>Specific Speed</td>
<td>343</td>
<td>294</td>
</tr>
</tbody>
</table>

With respect to the wheel of FIGS. 5 and 6, additional data is given for a lower specific speed of 221 at which the flow is 300 CFM.

As will be apparent from the table, the wheels in accordance with the invention are capable of operating with significantly greater efficiency against higher heads, all of which results from the new configuration of the vanes coupled with the streamlining near the exit.

It will be apparent that the invention is capable of various modifications with respect to the shapes of the flow channels and the particular vane structure; that it can be embodied in unshrouded as well as shrouded wheels; and that it can be used in conjunction with various forms of diffusers, both vaned and vaneless.

While the impeller in accordance with the invention may be manufactured in any accepted manner, e.g., by casting or by fabricating using separately machined parts, it may also be manufactured by machining, electrically eroding or otherwise forming passages in a solid wheel. These passages may be circular, square, oval, or of any desired cross sectional shape.

While the impeller in accordance with the invention finds particular utility in the later stages of multiple-stage compressors, it will be apparent that, in some applications, it may be used in a single-stage compressor as well.

I claim:

1. A low specific speed centrifugal compressor comprising a housing having a fluid receiving outer chamber and an inner impeller receiving chamber, an impeller having vanes rotatably supported in said impeller chamber for discharging into said outer chamber, said impeller having axially spaced annular discs supporting said vanes to define therewith a plurality of pumping flow channels, said vanes extending from a central eye to adjacent the periphery of said discs, each of said vanes being defined by a continuously smooth inclined pumping face and a continuously smooth inclined suction face, each of said vanes having continuously increasing cross-sectional thickness in a radially outward direction to present a wedge-shaped configuration in radial section, said channels being of substantially constant cross-sectional area throughout their length, said vanes having rounded entrance edges and a rounded convex curvature in the pumping face adjacent the periphery of said impeller, and said inclined surfaces
merging into a trailing end, whereby the fluid pumped by said vanes moves from the central eye to the periphery of the impeller smoothly and with a minimum of turbulence through the pumping channels.

2. A low specific speed centrifugal compressor, comprising a housing having a fluid receiving outer chamber and an inner impeller receiving chamber, an impeller having vanes rotatably supported in said impeller chamber for discharging into said outer chamber, said impeller having axially spaced annular discs supporting said vanes to define therewith a plurality of pumping flow channels, said vanes extending from a central eye to adjacent the periphery of said discs, each of said vanes being defined by a continuously smooth convex pumping face and a continuously smooth concave suction face, each of said vanes having continuously increasing cross-sectional thickness in a radially outward direction to present a wedge-shaped configuration in radial section, said channels being of substantially constant cross-sectional area throughout their length, said vanes having rounded entrance edges and a rounded portion with a pronounced increase in convex curvature in the pumping face adjacent the periphery of said impeller, and said convex and concave surfaces merging into a rounded trailing end whereby the fluid pumped by said vanes moves from the central eye to the periphery of the impeller smoothly and with a minimum of turbulence through the pumping channels.