United States Patent [19]

Yu

[11] 3,887,295

[45] June 3, 1975

[54] **COMPRESSOR INLET CONTROL RING**

[75] Inventor: **Mason Kwok Yu**, Birmingham, Mich.

[73] Assignee: **General Motors Corporation**, Detroit, Mich.

[22] Filed: **Dec. 3, 1973**

[21] Appl. No.: **421,033**

[52] **U.S. Cl.** ..... **415/116**; 415/157; 415/213 R

[51] **Int. Cl.** **F04d 31/00**

[58] **Field of Search** .. 415/11, 53, 116, 145, DIG. 1, 415/147, 146, 213, 157, 144, 159

[56] **References Cited**

UNITED STATES PATENTS

2,210,155	8/1940	Szydlowski et al.	415/147
2,405,282	8/1946	Birmann	415/159
2,798,658	7/1957	McDonald	415/53
2,981,461	4/1961	Murphy	415/213
3,011,693	12/1961	Downs et al.	415/DIG. 1
3,019,963	2/1962	Eck	415/DIG. 1
3,217,655	11/1965	Sercy et al.	415/116
3,462,071	8/1969	Garve	415/116
3,484,039	12/1969	Mittelstaedt	415/144
3,627,440	12/1971	Wood	415/DIG. 1
3,643,675	2/1972	Wetterstad	415/116

FOREIGN PATENTS OR APPLICATIONS

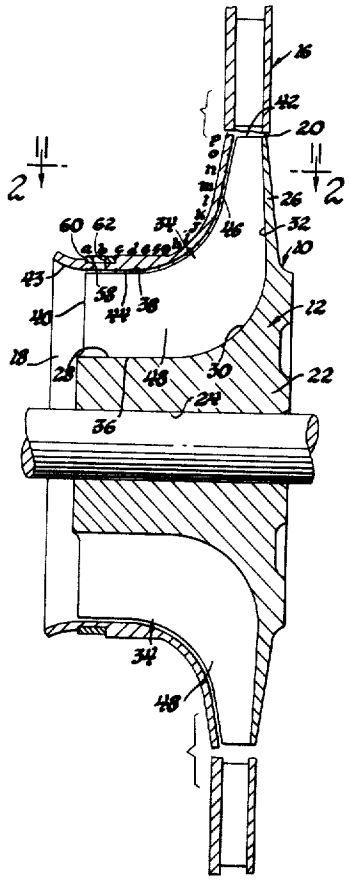
963,540	1/1950	France	415/53
1,224,445	2/1960	France	415/157
1,086,558	1/1957	Germany	415/53
106,869	9/1924	Switzerland	415/144

Primary Examiner—Henry F. Raduazo
Attorney, Agent, or Firm—J. C. Evans

[57] **ABSTRACT**

A centrifugal compressor includes a stationary outer shroud and a centrifugal rotor with an inner hub wall located radially inwardly of the stationary shroud to define an axial inlet and a radial outlet. A plurality of centrifugal blades located on the hub wall define a plurality of circumferentially spaced flow passages for compression of working fluid drawn from the axial inlet for discharge through the radial outlet. Means are included at the inlet end of the stationary shroud to direct additional flow into the flow passages between each of the centrifugal blades on the rotor to relieve partial vacuum conditions existing between the stationary shroud wall and the rotor tips at the inlet end thereof to improve the rotor exit pressure and velocity gradient thereby to increase efficiency of a following diffuser located downstream of the radial outlet of the compressor.

2 Claims, 7 Drawing Figures



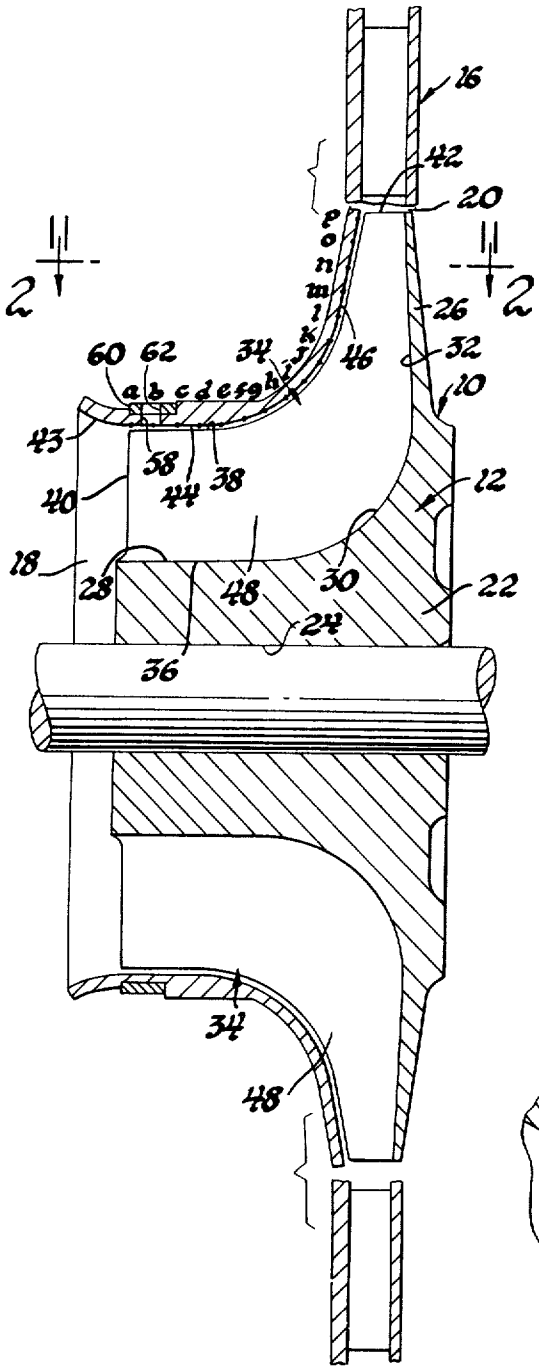


Fig. 1

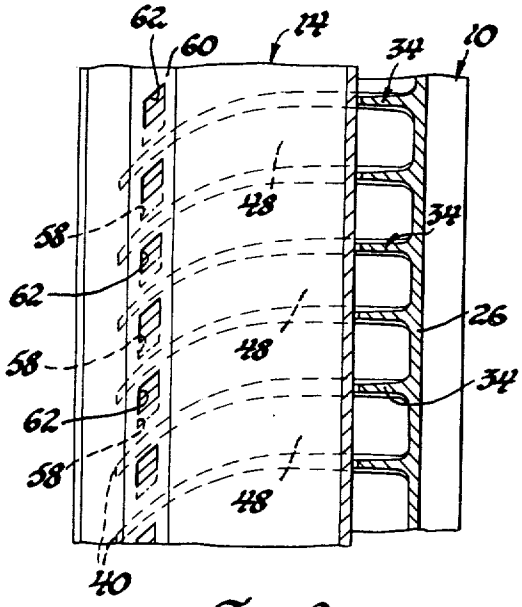


Fig. 2

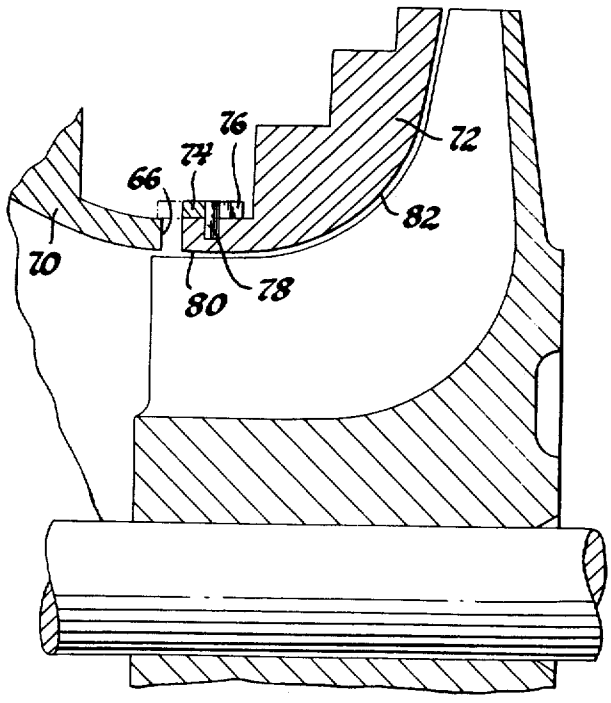


Fig. 3

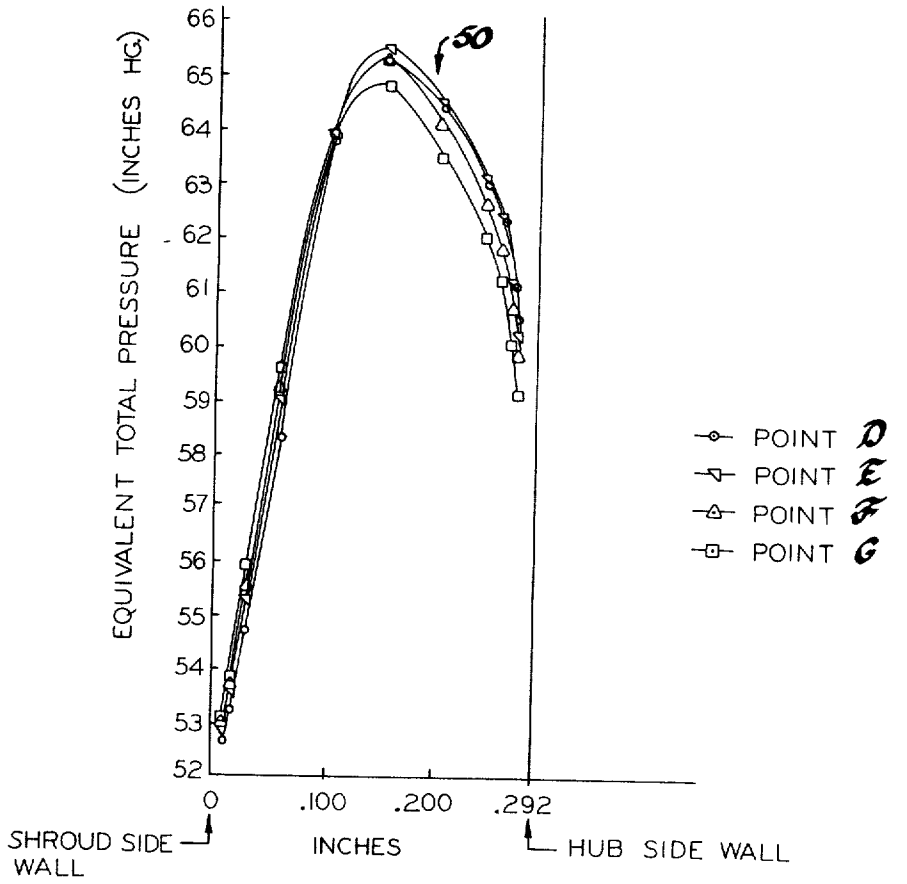

EQUIVALENT TOTAL PRESSURE (INCHES HG)
66
65
64
63
62
61
60
59
58
57
56
55
54
53
52
50
0
.100
.200
.292
SHROUD SIDE WALL
INCHES
HUB SIDE WALL
POINT D
POINT E
POINT F
POINT G

Fig. 4

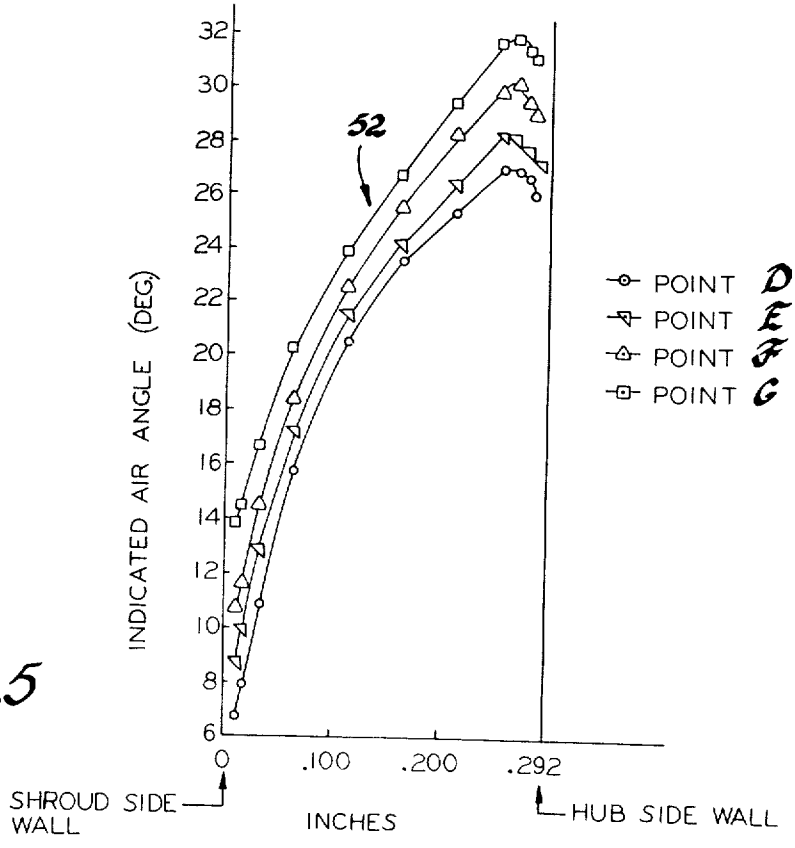

INDICATED AIR ANGLE (DEG.)
32
30
28
26
24
22
20
18
16
14
12
10
8
6
52
0
.100
.200
.292
SHROUD SIDE WALL
INCHES
HUB SIDE WALL
POINT D
POINT E
POINT F
POINT G

Fig. 5

COMPRESSOR INLET CONTROL RING

This invention relates to centrifugal compressors and more particularly to means in association with such compressors for controlling the pressure build-up therein.

In the design of centrifugal or mix flow compressors there are two principle objectives. The first is to achieve a certain flow rate at a predetermined pressure ratio to maximize efficiency at various operating speeds. The second objective is to achieve a broad flow range from the inlet to the outlet of the compressor rotor at any speed to provide satisfactory operation during transient compressor conditions.

In accordance with these objectives, it is assumed that there is an ideal uniform flow from the inlet through the outlet of the compressor. Other parameters such as inlet and outlet areas, diameter, pump and shroud contour, blade number distribution and blade thickness are designed to obtain a proper balance between preformance, the design material of the compressor components and manufacturing requirements.

However, under real environmental conditions there are viscous losses, compressibility build-up within the flow passages of the compressor rotor and boundary layer effects that cause a compressed working fluid to be distorted from the inlet to the outlet of the compressor thereby resulting in non-uniform pressure and velocity gradients at different points in a working flow passage through the rotor of the compressor.

Accordingly, an object of the present invention is to improve the uniformity of compressed fluid at an exit passage of a centrifugal rotor to obtain overall higher efficiency and broader flow range through the compressor.

Still another object of the present invention is to provide means for maintaining a uniform flow distribution at the inlet of a centrifugal compressor rotor so as to improve the uniformity of the compressed fluid at the radial outlet of a centrifugal compressor to maintain higher efficiency and broader flow range therethrough.

Yet another object of the present invention is to provide an improved centrifugal type compressor including a rotor and blade passages wherein a partial vacuum is produced between a stationary shroud wall and tips of blades defining the blade passages wherein means are provided to selectively direct an additional flow of ambient air into the inlet of the compressor to produce an increase in flow range between the inner shroud wall and the blade tips thereby to improve the rotor exit pressure and velocity gradients.

Further objects and advantages of the present invention will be apparent from the following description, reference being had to the accompanying drawings wherein a preferred embodiment of the present invention is clearly shown.

In the Drawings:

FIG. 1 is a vertical sectional view of a centrifugal compressor shown in association with a downstream diffuser;

FIG. 2 is a fragmentary sectional view taken along the line 2—2 of FIG. 1 looking in the direction of the arrows;

FIG. 3 is a fragmentary vertical sectional view of a centrifugal compressor including another embodiment of the invention;

FIG. 4 is a graph showing the pressure profile across a flow passage in the compressor of FIG. 1 at select points thereon;

FIG. 5 is a graph of traverse points taken to indicate air angle across the width of a flow passage at select points in the flow passages of the compressor in FIG. 1;

FIG. 6 is a graph showing the pressure ratio between the shroud wall static pressure and the inlet pressure to the compressor of FIG. 1; and

FIG. 7 is a graph showing air flow characteristics and pressure ratios at various rotor speeds of a typical centrifugal compressor without the present invention.

Referring now to the drawings, in FIG. 1 a centrifugal compressor **10** is illustrated including a rotor **12** and a stationary shroud **14** for directing compressed fluid to a vane or vaneless type diffusers **16.** Working fluid is introduced from an inlet **18** and compressed and pushed out by a combination of centrifugal and lift forces toward a radial annular exit passage **20** leading to the diffuser **16.**

The rotor **12** more particularly includes a hub **22** having a central opening **24** through which a drive shaft is directed and connected to the rotor **12** for driving it at selected speeds of rotation. The hub **22** includes a radially outwardly directed rear wall **26** and an axial inlet surface **28** thereon joined through an annular curved surface **30** to a radially outwardly directed inner surface **32.**

The rotor **12** includes a plurality of circumferentially located blades **34** thereon each including a root **36** thereof secured to the surfaces **28, 30, 32.** Each of the blades **34** further includes an outer radial tip **38** which is curved from an axial direction to a radial direction as best seen in FIG. **1.** As seen in FIG. **2,** each of the blades **34** has a curvature or camber from a leading edge **40** directed generally perpendicularly with respect to the hub surface **28** at the inlet **18.** Each blade **34** further includes a trailing edge **42** that is directed generally perpendicularly to the surface **32** adjacent the radial outlet **20.** The stationary shroud **14** includes an annular, inwardly converging inlet lip **43** thereon that is joined to an axial wall surface **44** of the shroud **14** that overlies the rotor surface **28** in spaced parallelism therewith and in closely spaced relationship with the rotor blade tips **38.** The inner wall surface of stationary shroud **14** is then directed radially outwardly at **46** in spaced relationship with the radially outwardly directed hub wall **26** and closely adjacent blade tips **38.**

The inner wall surfaces **44, 46** of the stationary shroud **14** and the hub wall surfaces **28, 30, 32** along with the individual rotor blades **34** define a plurality of enclosed flow passages **48** each being opened at opposite ends thereof to the inlt **18** and the outlet passage **20.**

Compressors of this type, under real environmental conditions, have viscous fluid losses and compressibility effects as well as boundary layer effects therein which can cause a compressed fluid to be distorted from the inlet **18** to the outlet **20** as the working fluid passes through passages **48.**

As shown in FIG. **1,** a plurality of pressure taps a through p directed through the stationary shroud **14** are utilized to demonstrate the degree of this non-uniform flow condition. As shown in FIG. **4,** the pressure tap located at **20** and pressure traverses from the inner surface **46** of the shroud **14** to the hub surfaces

30, 32 at different flow rates D-G results in a plurality of pressure gradient curves represented by the family of curves **50.** They show a reduced pressure in the vicinity of the interface between the shroud surface **46** and the blade tips **38** and an increased pressure toward the midpoint of the passages **48** for each rate D-G. The pressure is reduced toward the hub wall with this pressure still exceeding the pressure at the stationary shroud wall. As shown in FIG. **5,** angle probes located at **20** produce indicated air angle from the shroud surface **46** toward the hub wall as shown by the family of curves **52.** The angle increases from the region between the inner surface **46** and the blade tip **38** in the direction of the hub wall.

Furthermore, as is depicted in FIG. **6** by the family of curves **54,** the pressure ratio between the shroud wall static pressure and the inlet pressure at **18** at different points along the camber line distance of the blades **34** increases from a minimum at the inlet edge of the passages **48** to a maximum at the outlet thereof.

Reference line **56** in FIG. **6** indicates ambient pressure. The data points were measured at a rotor speed of 44,000 rpm at the inlet end of the blades at a temperature of 784°F. At this speed of operation, representative of compressor speeds found in turbine engines, it can be seen that for a range of flow rates represented by the family of curves **54** that there will be a negative pressure at the static pressure taps *a* through *p* through a substantial operating speed range. This discontinuity in the stream tube through the passages **48** will produce an equivalent total pressure and indicated air angle which is much lower at the outlet of the shroud wall **46** compared with the rest of the outlet opening **20** at the trailing edge **42** of each of the blades **34.** The measurements also indicate that a smaller portion of the flow is passed through the stream tube at points close by the shroud surfaces **44, 46.**

As best shown in FIG. **2,** a plurality of slanted ports **58** shaped as parallelograms are provided in the stationary shroud **14** immediately downstream of the lip **43** thereof. Each of the ports **58** are located where the static pressure ratio is always less than unity as shown in the graph of FIG. **6.** A control ring overlies the slanted ports **58** and includes a plurality of slanted ports **62** therein shaped like ports **58.** Rotation of the ring **60** on the outer surface of the shroud **14** will cause ports **62** to overlap ports **58** to a greater or lesser degree to produce a variable control of communication between ambient air and the inner wall surface **44** of the shroud **14** at locations where the partial vacuum conditions exists as shown in FIG. **6.** As a result, a controlled amount of additional flow will be sucked into the rotor stream at the shroud surface **46** due to pressure differential between atmosphere and the partial vacuum condition. This extra introduction of flow will increase the flow range of a given rotor and will also improve the rotor exit pressure and velocity gradient.

Because of this improved pressure profile and velocity gradient at the outlet passageway **20,** the following diffuser **16** will have increased efficiency by virtue of the improvement of the entrance flow uniformity thereto.

The improved flow characteristics are set forth in the graph of FIG. 7 which shows equivalent air flow in pounds per second versus the pressure ratio across the compressor. There is a family of curves **64** shown thereon which represent various speeds of rotation of the rotor **12.** Each of the curves **64** run from a surge limit line **68** and represent a fairly flat pressure ratio through most of the extent thereof through a wide range of air flow rates. By use of the present invention, each of the curves **64** will have a higher pressure ratio and flow range than shown in FIG. **7.** The control ring **60** will also produce a substantially uniform pressure condition at the inlet of the vaneless diffuser throughout the full transverse extent thereof. FIG. **3** shows another embodiment with a slot **66** formed in a two piece shroud **70, 72** circumferentially, continuously thereabout to define a gap in communication with atmosphere. A continuous ring **74** around shroud **72** has grooves **76** therein at circumferential points thereabout which receive dowel pins **78** fixed to the shroud piece **72** to permit axial movement of ring **74** with respect to slot **66** to open and close the gap. This regulates additional inlet flow into the rotor stream at surfaces **80, 82,** corresponding to surfaces **44, 46,** respectively in the embodiment of FIG. **1.**

While the embodiments of the present invention, as herein disclosed, constitute a preferred form, it is to be understood that other forms might be adopted.

What is claimed is:

1. A centrifugal compressor comprising a stationary shroud with an inner wall having an axial inlet and a radial outlet, a continuously open inwardly converging inlet lip on said shroud fixedly connected with respect to said axial inlet to form a flow transition therebetween, a rotor including a hub adapted to be connected to a drive shaft, said hub including a hub wall having an axial inlet and a radial outlet a plurality of rotor blades supported on said hub wall including a radially outer tip and an axial leading edge and a radial trailing edge thereon, each of said rotor blades defining a flow passage therebetween bound by the inner wall of said stationary shroud and the hub wall and having a flow discontinuity negative pressure region at said tip from the axial leading edge through a part of the chamber length of said tip, a gap in the inner wall of said stationary shroud located circumferentially therearound at the flow transition between said inlet lip and said axial inlet, said gap being in communication with atmospheric pressure exteriorly of said stationary shroud and overlying the rotor blade tips immediately downstream of the leading edge of each of said rotor blades, an annular control ring movable with respect to said gap for regulating the communication between atmosphere and the inner wall of said stationary shroud to produce a controlled bleed of atmosphere to said negative pressure region to produce a uniform pressure and velocity gradient from the inlet to the outlet of each of the said flow passages with the total pressure and indicated air angle of compressed flow through the passages being uniform from the radial outlet of the shroud inner wall to the hub wall.

2. A centrifugal compressor comprising a stationary shroud having an inner wall with an axial portion and a radial portion, a continuously open inwardly converging inlet lip on said shroud fixedly connected with respect to said axial inlet to form a flow transition therebetween, a rotor having a hub wall thereon with an axial portion located radially inwardly of said inner wall axial portion to define an annular axial inlet and a radial portion thereon in spaced relationship to the radial portion of said inner wall of said shroud to define an annular radial outlet, a plurality of centrifugal blades sup-

ported on said hub at circumferential points thereabout, each of said centrifugal blades having a root secured to the rotor hub wall and including a tip portion thereon located in close spaced relationship to the inner wall of said shroud from the axial inlet thereof to the radial outlet thereof, each of said blades including a leading edge at the axial inlet and a trailing edge thereon at the radial outlet, each of said blades defining a flow passage therebetween into which working fluid is drawn from the axial inlet for centrifugal and lift discharge through the radial outlet, said blades producing a partial vacuum through a predetermined portion of the axial portion of the inner shroud wall which distorts the rotor exit pressure and velocity gradient across the radial outlet of said compressor, means forming a gap in said inner shroud wall at the flow transition between the inlet lip and the axial portion at a flow passage location wherein the static pressure ratio between the blade tip portions and the inner wall of said shroud is always less than unity during compressor operation, a control ring supported on the exterior of said stationary shroud radially outwardly of said gap movable to vary the flow area between atmospheric pressure and the flow passage to produce additional air flow into the regions of static pressure ratio less than unity to produce a uniform rotor exit pressure and velocity gradient at the radial outlet therefrom thereby to produce an improved efficiency of entrance flow to a diffuser located downstream of the radial outlet from said centrifugal impeller.

* * * * *