

June 27, 1961

R. C. EDWARDS

2,990,107

COMPRESSOR

Filed Nov. 30, 1956

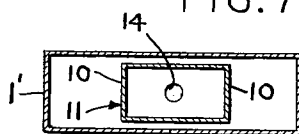
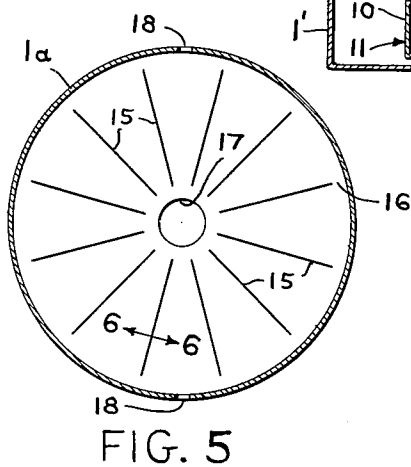
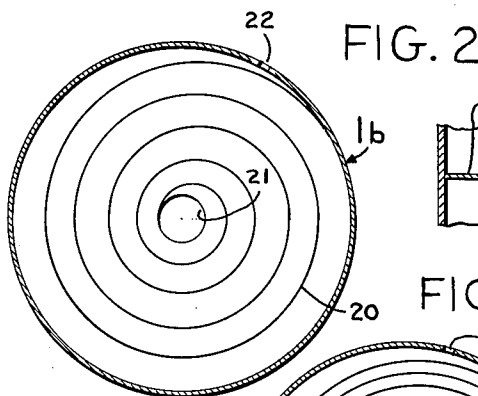
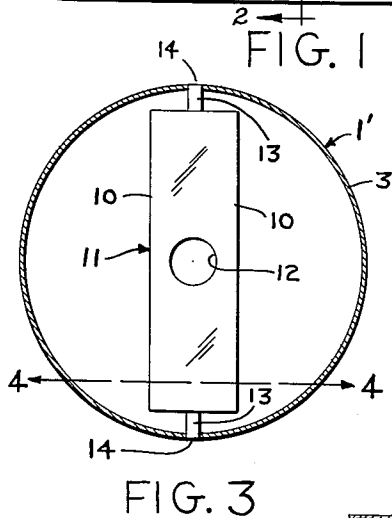
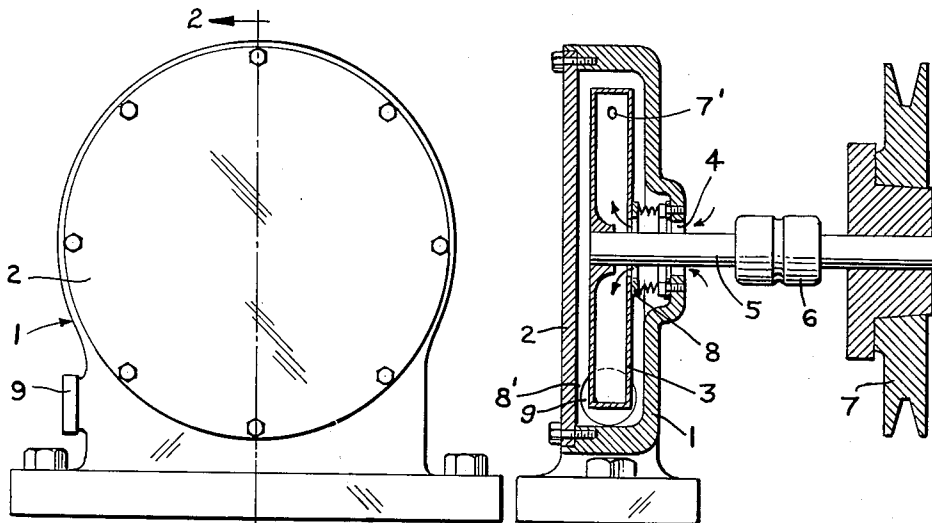


FIG. 4

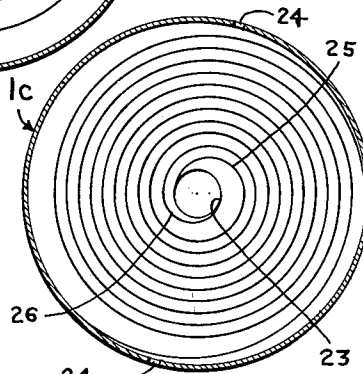


FIG. 8

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2,990,107
COMPRESSOR

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This invention relates to centrifugal compressors and more particularly to a new and novel type of centrifugal compressor whose novelty is based on the compression of the medium to be pumped within the rotating wheel. The basis of the invention is based on the premise that the material to be pumped can be compressed, such as gases, and the compressor of the present invention is particularly adaptable for compression of high density refrigerants such as Freon 12 or 22 for use in relatively small refrigeration systems, such as employ a two to five ton compressor, but it is to be understood that the invention is not so limited and that the compressor hereof may be employed in any work to which it is adaptable.

The invention is based on the phenomenon that in rotating about its axis, a completely closed cylinder containing a compressible medium such as a gas, the gas within the cylinder will eventually assume the rotative velocity of the cylinder and will be compressed by centrifugal force from the center outwardly and that by regulation of the ratio of the area of the discharge port to the area of the intake port the degree of compression may be regulated or the desired compression be obtained. Also by regulation of the discharge direction of the discharge port of the wheel, the conversion of discharge velocity pressure as the gas leaves the port in the periphery of the wheel into static pressure may be provided. Therefore, two compression phenomena may be added together to produce the total pressure of the compressor of the present invention, namely, compression within the wheel plus the conversion of velocity pressure to static pressure.

Specifically the invention consists of a cylinder rotatable about its axis at predetermined speeds, which cylinder is provided with an axial inlet and peripheral outlet for the medium compressed and with various forms of vanes formed upon and rotatable with the cylinder.

The accompanying drawings show one construction of the compressor of the present invention with various vane structures, but it is to be understood that the invention is not limited to the specific structures shown, which may be widely modified within the limits defined by the claims hereof.

In the drawings:

FIGURE 1 is an end elevation of the compressor.

FIGURE 2 is a vertical section through the compressor, taken on line 2-2 of FIGURE 1.

FIGURE 3 is a section through the cylinder of the compressor showing one form of vane.

FIGURE 4 is a cross section taken on the line 4-4 of FIGURE 3.

FIGURE 5 is a section through the cylinder showing straight radial vanes.

FIGURE 6 is a detail section taken on line 6-6 of FIGURE 5, i.e. a fragmentary section taken at right angles to the section shown in FIGURE 5.

FIGURE 7 is a section through the cylinder showing

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a single spiral vane, spiralling from the axial inlet to a single discharge port.

FIGURE 8 is a section through the cylinder showing double spiralling vanes, spiralling from the axial inlet to a pair of diametrically opposed discharge ports.

The construction of the forms of the compressor, shown will be first described, and the theory and operation thereof will follow.

The compressor is of simple construction comprising a stationary casing 1, one side of which is closed by a removable closure plate 2, to permit insertion of the rotative cylinder 3 into the casing 1.

The cylinder 3 is completely closed except for an axial opening 4 which forms the inlet for gas or the medium to be compressed, and through which the rotating shaft 5 extends. The shaft 5 may be supported by any suitable type of bearing shown at 6 and rotated by any suitable means. A sheave 7 for receiving a V-belt is shown in the drawings for rotating the shaft 5 and cylinder 3. The cylinder 3 is provided with a discharge port 7' in its perimeter and a suitable seal 8 of any approved type is provided to prevent leakage of the medium being compressed into the casing 1 surrounding the wheel at the inlet of the medium into the cylinder and also to atmosphere from the space 8' in the casing which surrounds the cylinder. The compressed gas or other medium is discharged into the space 8' from the interior of the cylinder 3 through the discharge port 7' and leaves the casing for use through an outlet 9.

Various forms of vane structures may be employed in the cylinder 3 and all such vanes, regardless of shape, type or form are formed within or attached to the inner surface of the cylinder and rotate therewith.

In FIGURE 4 of the drawings the vanes 10 are arranged to form substantially a diametrically extending cylinder 11 within the rotating cylinder 1' with the inlet 12 opening transversely into the inner cylinder 11 at its longitudinal center. Discharge passages 13 of greatly reduced area with respect to the cross-sectional area of the inner cylinder 11, and in proper ratio to the area of the inlet 12 (as hereinafter more specifically referred to) are provided at each end of the inner cylinder 11. The discharge passages 13 open out through diametrically opposed discharge openings 14 in the perimeter of the cylinder 1'.

In FIGURE 5 of the drawings a plurality of straight vanes 15 are attached to the sides 16 of the cylinder 1^a and radiate from the axis of rotation of the cylinder 1^a. The inner ends of the vanes 15 terminate a predetermined distance outwardly of the perimeter of the inlet opening 17 in one side of the cylinder 1^a and also terminate a predetermined distance inwardly of the perimeter of the cylinder 1^a as clearly shown in FIGURE 5, so as to permit flow of the medium being compressed to the two diametrically opposed discharge ports 18. The combined areas of the discharge ports 18 is considerably less than the area of the inlet port 17, and the reasons and proportions of these areas is brought out in the following discussion of the theory and operation of the compressor.

FIGURE 7 shows a deviation from the straight vane structures of FIGURES 3 and 5 in that it shows a spiral vane 20 which spirals outwardly in a uniform spiral from the perimeter of the inlet port 21 of the cylinder 1^b to the inner perimeter of the cylinder, to which it is joined at

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a point a predetermined distance beyond the discharge port 22 in the perimeter of the cylinder 1^b. Thus the spiral passage from the intake 21 first gradually increases in area for a short distance outwardly from the intake then continues in uniform area to and throughout a portion of the outermost spiral, gradually decreasing in cross-sectional area as it approaches the discharge port 22 and finally, at a predetermined distance beyond the discharge port 22 it merges to the inner periphery of the cylinder 1^b. The proportionate areas of the intake port 21 and the discharge port 22 are as described in connection with FIGURES 3 and 5. The spiral vane 20 extends from one side of the cylinder 1^b to the other, as the vanes 15 in FIGURE 5, and is attached to the cylinder to rotate therewith.

FIGURE 8 employs the same type of spiral vane as shown and described in FIGURE 7, differing therefrom in that two spiral vanes 25 and 26 are provided which spiral in uniformity from the intake port 23 in the cylinder 1^c to the two discharge ports 24 in the perimeter of the cylinder. The discharge ports 24 are diametrically opposed in the perimeter of the cylinder 1^c and are located predetermined distances forwardly of the termination of the spiral flow passages formed by the vanes 25 and 26. The proportion of the areas of the intake port 23 to the combined areas of the discharge ports is regulated as in the other forms of the cylinder above described.

While in the drawings the vanes are shown as extending from side to side of the rotative cylinders, it is to be understood that the invention embraces structures wherein the vanes extend into the cylinder only a part of its width and not completely from side to side thereof.

The invention is based upon the observed phenomenon that if one rotates a completely closed cylinder containing a compressible medium such as gas about its axis, the gas within the cylinder will eventually assume the rotative velocity of the cylinder and will be compressed by centrifugal force from the center outwardly.

In a non-rotating cylinder the gas will obey Boyle's law which states that $P\rho=RT$. Where P is the pressure in pounds per square foot, ρ is the density in pounds per cubic foot, R is the universal gas constant which is a constant for any given gas and T is the absolute temperature degree Rankine (degrees F.+460).

If the cylinder is rotated with an angular velocity of ω radians per second about its axis and the gas rotates about the axis of the cylinder, then the law determining the pressure in the cylinder measured along the radius of the cylinder as a function of ω , r and R and is given by the equation:

$$\left(\frac{r^2 \omega^2}{2gRT} \right)$$

$$p=p_0 e$$

This relationship is based on isothermal compression of the gas within the wheel. With the proper construction of the wheel, isothermal compression can be realized. P_0 is the pressure at the center of the wheel.

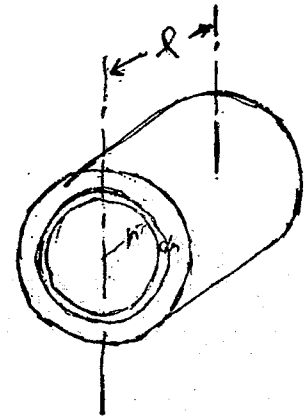
The gas will tend to be compressed along an isothermal path rather than an adiabatic path, because the discharged part is so minute in size as compared to the inlet opening to the compressor wheel, and the gas will be cooled as it is compressed. This trend toward isothermal compression will aid in reducing the H.P. requirements of the compressor for a given head.

The above relationship is derived through the following steps:

The force on the shell of the cylinder is equal to the pressure in the cylinder times the surface area. An elemental change in the surface force is equal to the surface area times an elemental change in the pressure dp or $2\pi r l dp$. This elemental change in surface force is

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equal to the elemental change in centrifugal force of an elemental section of the cylinder.



$$\frac{2\pi r dr 1\rho r\omega^2}{g}$$

Since from above $p\rho=k$, we can substitute

$$\frac{p}{k}$$

for ρ .

Now if we equate the elemental change in surface force with the elemental change in centrifugal force, we arrive at the following:

$$dp = \frac{p\omega^2}{kg} r dr$$

The equation becomes:

$$\text{at center of cylinder } \int \frac{p}{p_0} \frac{1}{p} dp = \int_{r_0}^{r_1} \frac{\omega^2}{kg} r dr$$

Integrating this equation from $r=0$ to r we have

$$[\log_e p] p r_1 = \left[\frac{r^2 \omega^2}{2 kg} \right] r = r_1$$

$$p_0 \quad r=0$$

at center

$$\left(\frac{r^2 \omega^2}{2 kg} \right)$$

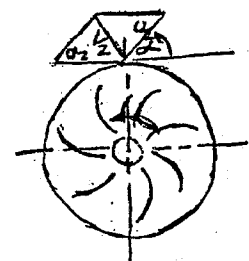
Therefore $p=p_0 e$ where p_0 is the pressure at the center of the cylinder. It also follows that

$$\rho = \rho_0 e^{\frac{r^2 \omega^2}{2 kg}}$$

Where ρ_0 is the density of the gas at the center of the cylinder.

In a centrifugal compressor the head produced is given by the following formula:

$$\text{Head} = \frac{U_2^2 - U_1^2 \cos \alpha}{g}$$



From the formula we see that the head is purely a function of the rotor velocity and the angle of discharge. However, the static pressure in pounds per sq. in. produced by the compressor is directly proportional to the

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density of the discharge gas. Therefore, if we can cause compression to take place within the wheel we can reduce the required head for a given required compression in direct proportion to the increase in density of the gas being compressed. In effect we can markedly reduce the r.p.m. at which the compressor must operate. This, in essence, is the basis of our entire invention.

A typical application for the compressor would be in the refrigeration field. The equation shows that

$$\frac{r^2 \omega^2}{2gk}$$

$$p = p_0 e$$

The ratio of compression

$$\frac{p}{p_0} = R$$

Then

$$\frac{r^2 \omega^2}{2gk}$$

$$R = e$$

For the refrigerant dichlorodifluoromethane, more commonly known as Freon 12, the value of k is—5900 when calculated from the state of the saturated gas at 40 degrees F.

ω in radians per second is equal to

$$\omega = \frac{2\pi \text{ r.p.m.}}{60} \text{ or}$$

$$\omega = .105 \text{ r.p.m. or } .105 n \text{ where } n = \text{r.p.m.}$$

$$\left(\frac{r^2 n^2 1045^2}{2 \times 32 \times 5900} \right)$$

$$R = e$$

$$\left(\frac{r^2 n^2}{3.46 \times 10^7} \right)$$

$$R = e$$

$$\left(\frac{r^2 n^2}{3.46 \times 10^7} \right)$$

$$R = e$$

$$r^2 n^2 = 3.46 \times 10^7 \log_e R$$

$$n = \frac{5890}{r} \log_e R$$

The following table calculated from this equation shows that very useful amount of gas compression can take place within the compressor wheel.

THEORETICAL COMPRESSION OF FREON 12 IN THE CLOSED WHEEL COMPRESSOR AS A FUNCTION OF WHEEL RADIUS AND R.P.M.

Table A

Compression Ratio R	log _e R	$\sqrt{\log_e R}$	Wheel Radius		
			r=.5 ft.	r=.75 ft.	r=1 ft.
1.5	.406	.637	7,500	5,000	3,750
2.0	.693	.833	9,810	6,440	4,900
2.5	.916	.957	11,300	7,500	5,630
3.0	1.10	1.05	12,300	8,250	6,180
3.5	1.25	1.12	13,200	8,790	6,590
4.0	1.39	1.18	13,900	9,260	6,940
4.5	1.50	1.23	14,500	9,650	7,240

From this table it can be noted that for a compression ratio of 3 to 1 and a wheel radius of .75 feet a speed of only 8250 r.p.m. is required.

If a commonly used non-over-loading type of com-

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pressor wheel were used with backward curved blades with an angle of between 35° and 60°, then the r.p.m. required for a 3 to 1 compression ratio would be:

$$\text{head} = \frac{U_1^2 - U_1 v_1 \cos \alpha}{g}$$

The equation relating head in feet, density of this gas in lbs. per cubic foot and pressure at the bottom of the column is:

$$\text{p.s.i.} = \frac{h \rho}{144}$$

$$h = \frac{144 \times \text{p.s.i.}}{\rho} = \rho = \text{density of gas in lbs./ft.}^3$$

For Freon 12 at 40° suction condition the head in feet required for a compression ratio of 3 to 1 is:

$$h = 144 \times 51.68 \times 3 \times .792 = 17,700 \text{ feet}$$

Where 51.68 is vapor pressure of Freon at 40° F. and .792 is the cubic feet per pound of gas

Assume that the $\cos \alpha = .5$ then

$$17700 = \frac{5U_1^2}{32}$$

$$U^2 = 1,130,000$$

$$U = 1065 \text{ feet/sec.}$$

Assuming a wheel radius of .75 feet and a wheel circumference of 4.7 feet the revolutions per second of the wheel would be 1065/4.7 or 227 per second or 13600 per minute. This 13600 r.p.m. compares with 8250 r.p.m. as given in the Table 1 for compression within the wheel.

However, a further and very important advantage of the invention is that the gas after being compressed within the wheel can be discharged through a port in the wheel so that its velocity pressure can be converted into static pressure. This static pressure can then be added to the compression ratio obtained within the wheel. The Freon 12 gas may be ejected through a port in the circumference of the compressor wheel. The port may be oriented in a forward, radial, or backward direction with respect to the rotational of the compressor wheel. The head produced by this discharging gas will obey the same equation as used for an open type wheel of the conventional type, namely,

$$h = \frac{U_1^2 - u_1 v_1 \cos \alpha}{g}$$

The gas leaving the wheel described above and rotating at 8250 r.p.m. will be compressed to a pressure of 172 lbs.

$$\text{p.s.i.} = \frac{\rho}{144 g} \cdot 5U^2$$

$$\text{p.s.i.} = \frac{3 \times .5 \times \left[\frac{8250}{60} \times 4.7 \right]^2}{144 \times 32 \times .792} = 172 \text{ lbs.}$$

(Cubic feet per pound at 40° suction=.792.)

These calculations are based on the same angle of discharge as used in the calculations for a standard wheel. If a forward discharge port were used, the theoretical head produced would, of course, be far greater.

Typical types of wheel that could be used to produce these results are shown in the drawings.

The above shows that gas to be compressed can be compressed within the cylinder and then discharged after the compression has taken place; therefore the present invention consists of an entering opening at the axis of a closed cylinder containing vanes to cause the gas to rotate with the cylinder and a discharge port on the outer perimeter of the cylinder as is shown in the drawings and hereinbefore described.

The ratio of the area of the discharge port to that of the area of the intake port at the axis of the cylinder must be quite small so that the gases are forced to compress within the cylinder before they are discharged. Compression within the cylinder may be caused to occur when this ratio is in the neighborhood of 1 to 10 or less. It is the present practice in constructing centrifugal compressors to build them with the ratio of port areas in the order of magnitude of five or six to one, and thus the compressors have, if anything, caused a decrease in pressure within the rotating wheel rather than an increase.

One of the many advantages of the present invention is that of the construction of a centrifugal gas compressor with a low rotative velocity.

In standard existing types of centrifugal compressors using Freon 11 (suction 40° F. condensing 84° F.) of the best design where a head of approximately ten (10) pounds is required, with a wheel having a diameter of one foot the peripheral velocity of the wheel must be 725 feet per second. However, if compression is caused to take place within the wheel, as in the cylinder of the present invention, the peripheral velocity of the wheel may be reduced to 475 feet per second. This reduction in peripheral velocity, is of particular importance in the refrigeration industry where it is highly desirable to provide a low tonnage refrigeration compressor of the order of magnitude of two to five tons. To date this has been impossible from a practical standpoint because rotative wheel speeds of the order of 15,000 to 25,000 revolutions per minute and greater are required. However, if compression is caused to take place within the wheel, as in the cylinder of the present invention, the rotative speed may be reduced to approximately 10,000 r.p.m. which is well within a practical range for use in practical equipment now available on the market.

Also, in the construction of a two to five ton centrifugal compressor of present day best design using a conventional type of wheel, a low density refrigerant gas must be used in order to obtain a good hydraulic efficiency. Low density refrigerant gas does not have the frictional losses that a high density gas has within a wheel. However, the low density gases still have sufficient functional losses within the wheel that makes them impractical for use in small tonnage machines of present day design.

With the compressor of the present invention, in which the compression takes place within the wheel, friction losses within the wheel can be almost ignored because they will so small when high density refrigerants, such as Freon 12 and 22 are employed. High density refrigerants are ideal for use in a compressor of this new design.

Table A (above) shows the compression ratios that may be obtained within the wheel, using 40° F. suction temperature with Freon 12 and various r.p.m. It will be noted that with moderate speeds, compression ratios of two to three to one can be obtained within the wheel. Added to this compression ratio within the wheel, is (in the present invention) the conversion of the discharge velocity pressure as the gas leaves the discharge port in the periphery of the wheel to static pressure. Therefore, two compression phenomena are added together to produce the total pressure of the centrifugal compressor of the present invention, namely, compression within the wheel plus the conversion of the velocity pressure to static pressure.

The conversion of the velocity pressure to static pressure can be controlled by the direction in which the gas is discharged from the periphery of the cylinder 3. The discharge port may be directed forwardly with respect to the direction of rotation of the cylinder, radially or backward, depending on the necessity for using the velocity pressure to augment the compression within the wheel.

It is the practice in the field to use reciprocating compressors for refrigeration systems of small tonnage due to the inability to heretofore provide a practical centrifugal compressor for such use. In such reciprocating compressors the obtaining of capacity control is difficult and requires expensive relative, delicate and sensitive mechanisms for such control, but with a compressor of the present invention accurate capacity control is provided merely by changes in speed of rotation of the cylinder 1. With a centrifugal compressor the compressor capacity varies with the square of the wheel velocity.

It will be understood that the invention is not to be limited to the specific constructions shown, but that they may be widely modified within the invention defined by the claims.

What is claimed is:

1. A centrifugal compressor including a closed cylinder having flat ends and means for rotating said cylinder wherein a compressible gas within the cylinder will assume the rotative velocity of the cylinder and will be compressed by centrifugal force from the rotative center of the cylinder outwardly, the inner surface of the perimeter of said cylinder being in the form of a continuous circle and free from irregular surface areas, at least one vane within said cylinder and rotatable with the cylinder, said vane spaced with relation to the inner surface of the perimeter of said cylinder as to permit unimpeded rotary movement of the compressed gas along the inner surface of the perimeter of the cylinder, the perimeter of said cylinder having at least one discharge port therein, and said cylinder provided with an inlet port at its axis the total discharge area of the cylinder with respect to the total intake area of the inlet to the cylinder being of a ratio in the neighborhood of one to five for forcing gas within the cylinder to compress before being discharged.

2. A centrifugal compressor as claimed in claim 1, wherein said outlet port is directed forwardly with respect to the direction of rotation of said cylinder to convert a predetermined percent of velocity pressure of the compressed gas to static pressure.

3. A centrifugal compressor as claimed in claim 1, wherein said outlet port is directed backward with respect to the direction of rotation of the cylinder to convert a predetermined percent of velocity pressure of the compressed gas to static pressure.

4. A centrifugal compressor as claimed in claim 1, wherein said outlet port is directed radially of the rotative axis of the cylinder to convert a predetermined percent of the velocity pressure of the compressed gas to static pressure.

5. A centrifugal compressor as claimed in claim 4, wherein the ratio of the area of said discharge port to that of said inlet port is in the neighborhood of one to ten.

6. A centrifugal compressor as claimed in claim 1, wherein said vanes constitute a plurality of straight vanes radiating from the axis of said cylinder and said inlet port with their inner ends spaced from the inlet port to provide an obstruction free annular space between the inner ends of the vanes and the outer edge of the inlet opening.

7. A centrifugal compressor including a closed cylinder having flat ends, means for rotating said cylinder wherein a compressible gas within the cylinder will assume the rotative velocity of the cylinder and will be compressed by centrifugal force from the rotative center of the cylinder outwardly, the inner surface of the perimeter of said cylinder being in the form of a continuous circle free from irregular surface areas, a plurality of vanes in said cylinder and rotatable with the cylinder, said vanes spaced with relation to the inner surface of

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the perimeter of the cylinder as to permit unimpeded rotary movement of the compressed gas along the inner surface of the perimeter of the cylinder, said cylinder provided with an inlet port at its axis and at least one discharge port therein, said vanes spaced outwardly of the inlet port so as to provide a curved obstruction free passage for gas about said inlet, the edges of said vanes extending to and engaging the inner surfaces of the ends of said cylinder, whereby the only obstruction free curved passages for gas within the cylinder are at the inner and outer ends of said vanes, the total discharge port area of the cylinder with respect to the total inlet area of the cylinder being such that the compressible gas within the cylinder will assume the rotative velocity of the rotating cylinder and will be compressed along an isothermal path.

8. A centrifugal compressor as claimed in claim 7, wherein the discharge port of said cylinder is disposed at an angle to the direction of rotation of the cylinder so as to convert a portion of the discharge velocity of the compressed gas to static pressure.

9. A centrifugal compressor as claimed in claim 7, wherein the ratio of the area of the discharge port to that of the inlet port is greater than one to four.

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