Aug. 21, 1956 W. H. CARRIER 2,759,662

CENTRIFUGAL COMPRESSORS

Filed April 26, 1950

FIG. 1

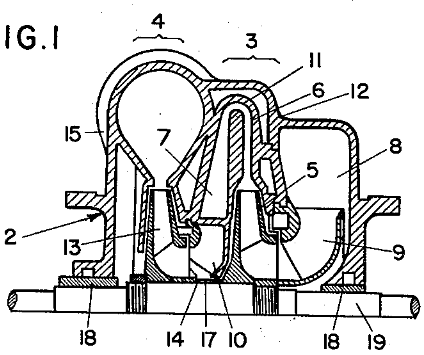
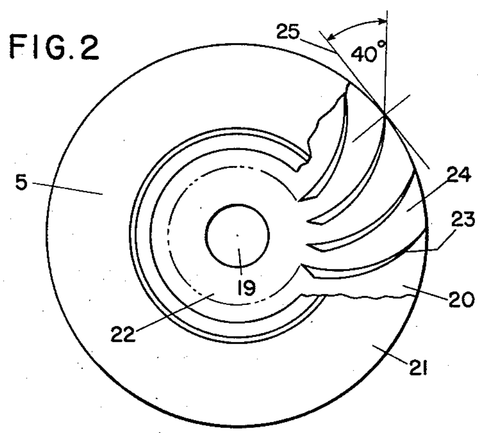


FIG. 2



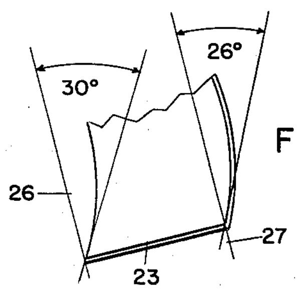


FIG. 4

FIG. 3

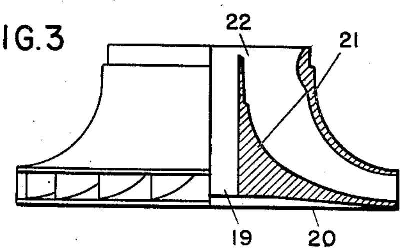


FIG. 6

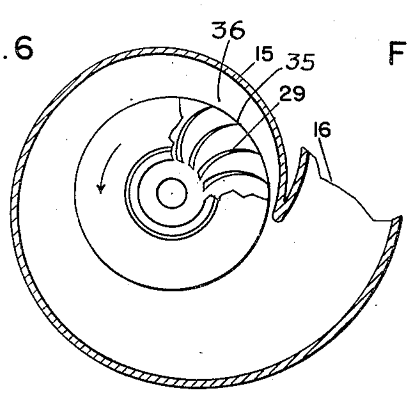


FIG. 5

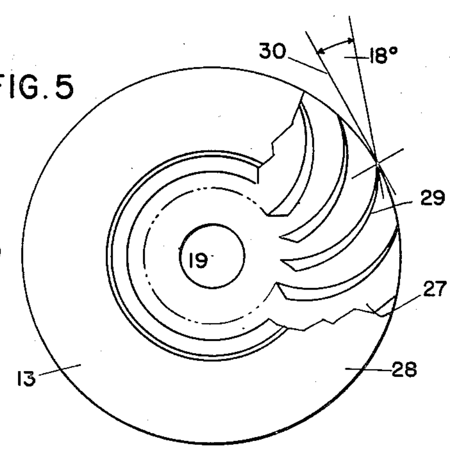
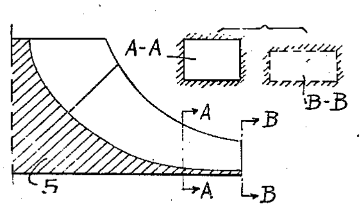


FIG. 7



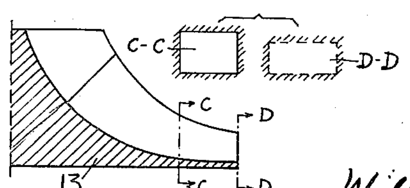


FIG. 8

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CENTRIFUGAL COMPRESSORS

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3 Claims. (Cl. 230—130)

This invention relates to centrifugal compressors and more particularly to centrifugal compressors for use in refrigeration for air conditioning applications.

In air conditioning applications in which a centrifugal compressor is employed in cooling water to a temperature not lower than 33° F., the normal temperature spread is about 70° F.; by "temperature spread" I mean the difference between the evaporating and condensing temperatures in the refrigeration machine. Employing trichloromonofluoromethane as a refrigerant, a ratio of compression of slightly over 4 is required for a 70° F. range. This is too great for good efficiency performance with a single stage machine having high pressure characteristics; it is too great for the successful application of a high efficiency volute type compressor without employing more than two stages.

The chief object of the present invention is to provide a centrifugal compressor for use in refrigeration for air conditioning applications which is highly efficient in operation and which operates over a considerably wider range without surge than centrifugal compressors heretofore obtainable.

An object of my invention is to provide a two-stage centrifugal compressor suitable for use in air conditioning applications capable of obtaining a ratio of compression between 3½ and 4½ in which the first stage has a relatively high pressure coefficient and the second stage has a relatively low pressure characteristic and less sensitivity to volume changes with greater stability.

A further object is to provide a two-stage air conditioning refrigeration compressor capable of obtaining a ratio of compression between 3½ and 4½ in which the first stage includes an impeller of improved design, an open diffuser imparting a flexibility of performance required in air conditioning applications and a return channel which in cooperation with the diffuser permits a highly efficient conversion of velocity into pressure and in which the second stage includes an impeller of improved design and a multi-discharge volute in which the change in velocity is caused in gradual steps by changing the area in an amount greater than that corresponding to volume change due to compressibility.

A still further object is to provide a two-stage centrifugal compressor so designed that a much smaller reduction in speed is required to maintain the same head or temperature difference at reduced capacities thereby effecting economies at partial loads by increasing the efficiency of the compressor and by increasing the efficiency of the variable speed drive. Other objects of my invention will be readily perceived from the following description.

This invention relates to a centrifugal compressor for use in refrigeration for air conditioning preferably consisting of two stages of compression. The preceding stage includes an impeller or rotor, an open diffuser and a return channel, which in conjunction with the diffuser, permits a highly efficient conversion of velocity into pressure. The second stage includes an impeller or rotor having a relatively low pressure characteristic and a volute for conversion of velocity into pressure at high efficiency thus gaining the same advantage in efficiency as in the first stage without the employment of a return channel. Preferably the first stage is adapted to produce about 54 to 57% of the total work done by the two impellers.

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The attached drawing illustrates a preferred embodiment of my invention, in which:

Figure 1 is a view partly in elevation and partly in section of the centrifugal compressor;

Figure 2 is a view in elevation of the first stage impeller partly broken to illustrate its construction;

Figure 3 is a view in section of the impeller shown in Figure 2;

Figure 4 is an exploded view of a vane or blade at the inlet to the wheel;

Figure 5 is a view in elevation similar to Figure 2 of the second stage impeller illustrating its construction;

Figure 6 is a diagrammatic view of the second stage volute illustrating the manner in which the change of velocity to pressure is accomplished by gradual steps;

Figure 7 is a diagrammatic view illustrating the passage areas at various points in the first stage impeller; and

Figure 8 is a diagrammatic view illustrating the passage areas at various points in the second stage impeller.

Referring to the drawing, there is shown a centrifugal compressor 2 primarily for use in refrigeration for air conditioning applications having two stages designated at 3 and 4, and adapted to be driven by any suitable means (not shown). The compressor receives gaseous refrigerant such as trichloromonofluoromethane from a cooler, raises the same to a higher pressure, and forwards the compressed refrigerant to a condenser in which it is liquefied, the liquid refrigerant being forwarded to the cooler. It will be appreciated the centrifugal machine may include the usual economizer to permit flashed refrigerant to be supplied to the second stage in addition to refrigerant from the first stage.

As shown in Figure 1, the first stage 3 includes an impeller or rotor 5 hereinafter described, an open diffuser 6 and a return channel 7 connecting the first stage to the second stage 4. An inlet 8 connects the first stage with the cooler or other fluid supply means (not shown). Suitable guide or spin vanes 9 are provided to impart a desired spin or direction of rotation to the gas entering the eye of the impeller 5. Likewise, suitable spin or guide vanes 10 are provided in the return channel. The open diffuser and return channel are formed by the diaphragm 11 placed within the casing 12 of the compressor.

The second stage 4 of the compressor receives fluid under pressure from return channel 7, such fluid entering the impeller 13 at its inlet 14. The fluid from impeller 13 is discharged into a single or multi-discharge volute 15 in which the velocity of the gaseous stream is converted into pressure. Fluid under pressure passes from the volute 15 to the condenser or other receiving means (not shown) through discharge openings 16 (refer to Figure 6).

The usual labyrinths 17 between the stages are provided to seal the stages from one another to assure that fluid from the first stage passes to the second stage substantially through return channel 7. Labyrinths 18 are also provided about the compressor shaft 19 at the ends of the stages to prevent leakage from the compressor casing. Suitable thrust bearings, shaft seals, etc. are also provided but are not described since such provision is well known.

In the machine contemplated, the first stage is adapted to produce from 54% to 57% of the total work per pound of refrigerant done by the two impellers. The design contemplates operation at a peripheral speed of approximately 525 ft./sec.

Considering the first stage 3 of the compressor 2, such stage is designed to permit a highly efficient conversion of velocity into pressure as well as to provide the flexibility of performance desired for utilization in air conditioning applications. The impeller 5 of predetermined diameter comprises a disk or shroud 20 secured to the shaft 19, a cover plate 21 in which the inlet or eye 22 is provided, and a plurality of backwardly extending blades 23 disposed between the shroud and cover forming passageways 24 therebetween.

In the impeller passageways 24, the area normal to the radial component of gas flow decreases as the radius of the impeller increases (refer to Figure 7), note the cross-sections illustrated in Figure 7 wherein the area represented by symbol A—A is greater than the area represented by symbol B—B. The blades are so disposed that the angle between the blade and the tangential direction increases as the radius of the impeller increases. Such design provides a relatively high radial component of velocity of the gas leaving the first impeller; that is the impeller is narrow at the periphery relative to the quantity of gas being handled. I have found that impeller 5 should be designed to possess a blade tip angle of about 40°±5° from a line 25 drawn perpendicular to the radius through the blade tip and tangential to the circumference of the disk or shroud as shown in Figure 2. It is desirable to so proportion the blade that it possesses a continuous variation of from about 30°±3° blade angle at the tip radius at the inlet decreasing monotonically to about 26°±3° at the shaft side for the vane inlet. In other words, the blade should possess an inlet angle at the shroud of about 30°±3° from a line 26 drawn normal to the radius through the vane leading edge and an inlet angle at the hub of about 26°±3° from a line 27 drawn normal to the radius through the vane leading edge. The angles recited are obtained by projecting the blades from the selected points and considering the angles such projections make with lines 25, 26, 27 respectively.

Preferably, the variation in angle is such as to maintain an approximately constant angle of attack along the heel of the blade when the velocity distribution into the eye of the impeller is considered in connection with the local peripheral speed so as to achieve substantially shock-free entry at the inlet or eye. The area ratios between the vane leading edges and the impeller tip are so proportioned as to reduce the actual through-put component of velocity by from about 10% to 15%. It will be appreciated the actual physical areas will change due to the increase in density of the gas when different refrigerants are employed depending upon the proportion of work in the rotor, the blade tip angle, the amount of inlet handling, etc.

The open diffuser 6 provides flexibility of performance and in cooperation with the return channel 7 permits a highly efficient conversion of velocity into pressure.

The second stage 4 of compressor 2 differs from the first stage in that it is designed to provide a smaller ratio of compression than the first stage. The shape of the impeller passageway between the blades or vanes is different since the area normal to the radial component of gas flow for at least the last half of the passageway increases with the radius note the cross-sections illustrated in Figure 8 wherein the area represented by symbol C—C is less than the area represented by the symbol D—D while the angle between the blade and the tangential direction decreases with increasing radius over approximately the last half of the length of the passageway thereby providing a relatively low radial component of the velocity of gas leaving the impeller (refer to Figure 7). In the impeller of the second stage, the ratio

of the radial component to the tangential component is lower than in the first impeller. The velocity head representing the kinetic energy leaving the impeller of the second stage is less than ⅔ of the corresponding velocity head in the first stage.

The rotor or impeller 13 includes a disk or shroud 27 attached to shaft 19, a cover plate 28 and a plurality of vanes or blades 29. I have found that impeller 13 should be designed to possess a blade tip angle of about 18°±3° from a line 30 drawn perpendicular to the radius through the blade tip and tangential to the circumference of the disk 27, as shown in Figure 5. The impeller blade trailing edge angle and leading edge angle are proportioned approximately the same as the similar angles of the first stage impeller. The area ratios between the vane leading edges and the impeller tip are so proportioned as to reduce the actual through-put component of velocity by about 15% to 20%. The diameter of rotor 13 is approximately the same as the diameter of rotor 5 but the axial distance at the inlet between the shroud and cover plate is about 80%±5% of the axial distance at the inlet between the shroud 20 and cover plate 21 of impeller 5 while the axial distance between the shroud and cover plate at the periphery is within the range of 75 to 100% of the axial distance at the periphery between the shroud and cover plate of impeller 5.

The volute 15 may be provided with a single or multiple discharges 16. This volute is so designed that the change in velocity to pressure occurs by gradual steps; preferably, the velocity in the volute decreases as the cross section of the volute increases to provide greater efficiency than may be obtained with a constant velocity as in conventional designs or with an open diffuser and a return channel as in the first stage described above.

The volute 15 is so designed as to provide a reduction in velocity from the impeller tip 35 to the volute throat 36 by making the width of the throat somewhat greater than the width of the impeller tip to obtain a reduction in the neighborhood of 5–15% as a sudden expansion. A second reduction in velocity may be obtained from the volute throat 36 to the cut-off by making the lower portion of the volute in the form of a frustum of a triangle having side angles less than 30°; approximately 15–20% reduction in velocity may be obtained to the point of cut-off. A third reduction may be obtained from the cut-off to the area of the volute proper. In the volute itself a reduction of about 27% may be obtained, while in the discharge pipe after the cut-off a further reduction of about 38% may be obtained. It will be appreciated the percentages of reduction of velocity so recited are approximate and that a 10% plus or minus toleration of the figures recited above may be included. These percentages are recited on the basis of 80% reduction of the original velocity. In addition to the losses caused by these successive steps, friction of the volute creates a further loss which brings the total loss at design operating condition of the machine to approximately 20% of the energy corresponding to the velocity of the gas leaving the rotor which is better than the first stage. The reduction in velocity is such that the effective angle of discharge is on the average 14° at the rotor tip continuously decreasing to 8% at the cut-off of the volute. These flow angles may be maintained with a small range of tolerance rather than with the first percentage-wise stages of reduction in velocity. The cut-off point should be located at about 112% of the impeller radius, preferably a tolerance is provided of about +10% and −5%.

The present invention provides a centrifugal compressor of greater efficiency than compressors heretofore obtainable and which is adapted for operation over wide ranges of capacity and pressure, which conditions are encountered in air conditioning refrigeration applications. This invention provides a two-stage compressor having a ratio of compression between 3½ and 4½ in which the first stage has a relatively high pressure coefficient while the second stage has a relatively low pressure coefficient and greater sensitivity to volume changes with greater stability. The compressor is designed in such manner that surge becomes a negligible factor in the operation of the machine. A smaller reduction in speed is required to maintain the same head or temperature difference at reduced capacities than with a three stage volute type machine designed for the same performance. This smaller reduction in speed effects savings at partial load by reducing the power and by increasing the efficiency of the variable speed drive since the percent of power losses in such a drive are approximately proportional to the percent of speed reduction.

While I have described the present invention with particular reference to a two-stage machine, it will be appreciated the two stages so designed may be included in a machine having more than two stages in which these stages comprise the last stage and the preceding stage.

While I have described a preferred embodiment of my invention it will be understood my invention is not limited thereto since it may be otherwise embodied within the scope of the following claims.

I claim:

1. In a centrifugal compressor, the combination of a preceding stage including an impeller having a cover, a shroud and a plurality of blades forming passages through the impeller, and an open diffuser surrounding the impeller, and a final stage including an impeller having a cover, a shroud and a plurality of blades forming passages through the impeller, and a volute, the impellers having approximately the same diameters, the passage area normal to the radial component of gas flow of the preceding stage impeller decreasing as the radius of the impeller increases, while the angle between the blade and the tangential direction increases as the radius of the impeller increases, thereby giving a relatively high radial component of velocity of gas leaving the impeller, the passage area for the final stage impeller for at least the outer portion of the passage normal to the radial component of gas flow increasing as the radius increases while the angle between the blade and the tangential direction decreases with increasing radius over approximately the last half of the length of the passageway thereby providing a

relatively low radial component of velocity of gas leaving the impeller.

2. In a centrifugal compressor, the combination of a preceding stage including an impeller and an open diffuser surrounding the impeller, said impeller having a cover, a shroud and a plurality of blades, and a final stage including an impeller and a volute, said final stage impeller including a cover, a shroud and a plurality of blades, the impellers having approximately the same diameters, the blades of the preceding stage impeller having outlet angles of about $40°\pm5°$ from a line drawn perpendicular to the radius through the blade tip and tangential to the circumference of the shroud and having an inlet angle at the shroud of about $30°\pm3°$ from a line drawn normal to the radius through the blade leading edge, and an inlet angle at the hub of about $26°\pm3°$ from a line drawn normal to the radius through the blade leading edge, the blades of the final stage impeller having outlet angles of about $18°\pm3°$ from a line drawn perpendicular to the radius through the blade tip and tangential to the circumference of the shroud.

3. A centrifugal compressor according to claim 2 in which the axial distance between the cover and the shroud of the final stage impeller is $80\%\pm5\%$ of the axial distance between the cover and the shroud of the preceding stage impeller and the axial distance at the periphery between the cover and the shroud of the final stage impeller is within the range of 75% to 100% of the axial distance at the periphery between the cover and the shroud of the preceding stage impeller.

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