A turbo-type compressor having an improved performance which may be produced without increasing its size or cost, and the performance of a refrigeration unit is also improved when it is provided with such a turbo-type compressor.

The turbo-type compressor includes a casing (55) provided with an intake opening and a discharge opening; a rotation shaft (41) operated by a driving mechanism; an impeller (19) provided integrally with the rotation shaft (41); a diffuser section (46) constituted by a pair of a first wall section (56) and a second wall section (58), located at the outer periphery side of the impeller (19) to serve as a fluid passage for a refrigerant driven towards the outer side by the rotation action of the impeller (19). The refrigerant is drawn in through the intake opening, by the action of the impeller (19) which is rotated together with the rotation shaft (41) and driven by a motor, to be compressed and discharged through the outlet opening. In this compressor, the diffuser section (46) is constructed in such a way that the width dimension in the axial direction of the outlet opening (46b) is made larger than the width dimension of the inlet opening (46a).
FIG. 6

FIG. 7

AREA RATIO (AR-1)

IMPROVED CP

CONVENTIONAL CP

ASPECT RATIO (2ΔR/b2)
TURBO COMPRESSOR AND REFRIGERATOR WITH THE COMPRESSOR

TECHNICAL FIELD

[0001] The present invention relates to a turbo-type compressor used for compressing liquids by a rotating impeller, and relates to a refrigeration apparatus having the compressor.

BACKGROUND ART

[0002] Conventionally, a turbo-type compressor is used in refrigeration apparatuses as a means of compressing refrigerant. The turbo-type compressor compresses a fluid by rotating a rotation shaft provided with an impeller.

[0003] The structure of a conventional turbo-type compressor will be explained with reference to FIG. 10.

[0004] As shown in the figure, an impeller 3 having a plurality of vanes 2 arranged in a circumferential direction with spaces therebetween is attached to a rotation shaft 1 of the turbo-type compressor so that the vanes can be rotated with the rotation shaft 1. A rotor constituted by the rotation shaft 1 and the impeller 3 is housed inside a casing 4.

[0005] The interior of the casing 4 is divided by a partition plate 5 into a diffuser section 6 and a return passage 7, and the diffuser section 6 and the return passage 7 are communicated with a return bend section 8 having U-shaped cross section.

[0006] The diffuser section 6 is constituted by a first wall section 6a on the casing side and a second wall section 6b on the partition plate side, and the first wall section 6a and the second wall section 6b are oriented perpendicular to the rotation shaft 1 while being parallel to each other. Also, a plurality of return vanes 9 are spaced circumferentially so as to guide the flowing fluid.

[0007] In this turbo-type compressor, the fluid compressed by the impeller 3 and output to the diffuser section 6 is forwarded to the return passage 7 through the return bend section 8.

[0008] In such a compressor, the diffuser section 6 comprised by the first wall section 6a and the second wall section 5b serves to decelerate the flow of the fluid driven by the impeller 3 and recovers most of the dynamic pressure as static pressure, and the pressure recovery coefficient C_p, which is a parameter to indicate the performance of the turbo-type compressor, is influenced by the shape of the diffuser section 6.

[0009] Therefore, the pressure recovery coefficient C_p may be increased by improving the area and shape of the inlet opening 6a and outlet opening 6b of the diffuser section 6.

[0010] However, in a conventional compressor, as shown in the graph in FIG. 11, the pressure recovery coefficient C_p has not reached a value of 0.5, thus leaving room for further improvement. For this reason, there are demands to improve the performance of the turbo-type compressor by modifying the pressure recovery coefficient C_p in the diffuser section 6.

[0011] Here, the pressure recovery coefficient C_p is expressed by the aspect ratio and the area ratio of the inlet and outlet openings 6a and 6b of the diffuser section 6.

[0012] The aspect ratio and the area ratio are obtained according to the following expressions:

\[
\text{Aspect ratio} = \frac{2R_b h_b}{2(R_2 - R_1) t_2} \quad (1)
\]

\[
\text{Area ratio} = \frac{A_R}{1 + \left(\frac{R_2 h_2}{R_1 h_1}\right)} - 1 \quad (2)
\]

where

[0014] R1 is a radius at the inlet opening 6a of the diffuser section 6;

[0015] R2 is a radius at the outlet opening 6b of the diffuser section 6;

[0016] b1 is a width dimension of the inlet opening 6a of the diffuser section 6; and

[0017] b2 is a width dimension of the outlet opening 6b of the diffuser section 6.

[0018] Also, the pressure recovery coefficient C_p for the diffuser section 6 is represented by the following expression:

\[
\text{pressure recovery coefficient } C_p = \frac{P_{31} - P_{32}}{P_{31} - P_{T_1}}
\]

where

[0020] P_{31} is a static pressure at the inlet opening 6a of the diffuser section 6;

[0021] P_{32} is a static pressure at the outlet opening 6b of the diffuser section 6; and

[0022] P_{T_1} is a total pressure at the inlet opening 6a of the diffuser section 6.

[0023] It can be seen that the level of recovery of the dynamic pressure into static pressure of the fluid, which is compressed and output by the impeller 3, can be improved by an amount corresponding to an increase in the pressure recovery coefficient C_p.

[0024] The present invention takes into consideration the above-mentioned circumstances, and objects thereof include providing a high efficiency turbo-type compressor having improved performance without increasing the size of the compressor, and providing a refrigeration apparatus including such a turbo-type compressor.

DISCLOSURE OF INVENTION

[0025] In order to achieve the above objects, the present invention provides a turbo-type compressor comprising a casing provided with an intake opening and a discharge opening; a rotation shaft operated by a driving mechanism; an impeller provided integrally with the rotation shaft; a diffuser section formed by a pair of wall sections at outer peripheries of the impeller to serve as a fluid passage for a fluid driven towards the outer periphery side by the rotation of the impeller so that the fluid is drawn in from the inlet opening, by the action of the impeller which is rotated together with the rotation shaft and driven by the driving mechanism, to be compressed and discharged from the discharge opening through the diffuser section; wherein the diffuser section is formed in such a way that the width dimension in the axial direction of an outlet opening located at the outer periphery side is made larger than the width dimension of an inlet opening for the fluid which is driven by the impeller.

[0026] Accordingly, because the width dimensions of the openings in the axial direction of the inlet side and the outlet side are made so that the outlet side is larger relative to the
inlet side, the aspect ratio of the inlet side and the outlet side is made somewhat smaller and the area ratio is made larger, so that the pressure recovery coefficient in the diffuser section can be made larger. This design enables effective recovery of the dynamic pressure of the compressed fluid output from the impeller into a static pressure in the diffuser section without making the structure larger or more complex, and accordingly, a high compression efficiency is realized in the turbo-type compressor of the present invention.

[0027] In another aspect of the invention, in the turbo-type compressor described above, the width dimension of the total opening is made larger than the width dimension of the inlet opening without altering the area ratio of the inlet opening to the total opening of the diffuser section.

[0028] Accordingly, without altering the area ratio of the inlet opening to the outlet opening in the diffuser section, the width dimension of the outlet opening is made larger relative to the width dimension of the inlet opening. That is, because the radius of the outlet opening is reduced by an amount corresponding to the increase in the width dimension of the outlet opening, the outer radius can be made smaller and the cost of the compressor can be reduced.

[0029] Also, even if the area ratio of the inlet opening to the outlet opening is not altered, because the width dimension of the outlet opening is made larger relative to that of the inlet opening, the aspect ratio is reduced, and the pressure recovery coefficient is made larger so as to reliably improve the performance.

[0030] In yet another aspect of the invention, in the turbo-type compressor described above, the pair of wall sections forming the diffuser section is tapered so as to separate gradually from each other from the inlet opening towards the outlet opening.

[0031] That is, by tapering the pair of wall sections forming the diffuser section, the performance of the compressor is readily improved by enlarging the width dimension of the outlet opening relative to that of the inlet opening. Also, because the wall sections are tapered, it is possible to avoid the problem of stream separation in the diffuser section of the refrigerant output from the impeller.

[0032] In yet another aspect of the invention, in the turbo-type compressor described above, one of the wall sections forming the diffuser section is formed into a tapered shape so as to separate gradually from the other one of the pair of wall sections from the inlet opening towards the outlet opening.

[0033] In other words, by making only one of the wall sections that constitute the diffuser section, into a tapered shape, the performance of the compressor can be improved quite readily by increasing the width dimension of the outlet opening relative to the inlet opening. Also, because only one of the wall sections needs to be made into a tapered shape, the performance can be improved even more easily.

[0034] In particular, by tapering the wall section in the front side which has more space as compared with the rear side of the impeller having a downstream side passage that communicates with the diffuser section, the dimension in the axial direction can be made smaller.

[0035] In yet another aspect of the invention, in the turbo-type compressor described above, the compressor is a multi-stage compressor having a plurality of the impellers and compresses the fluid from the intake opening sequentially by using, first, an upstream-side impeller first, and successive impellers afterward.

[0036] In other words, the present multi-stage impeller provides a superior turbo-type compressor of high performance because the pressure recovery coefficient is improved in each stage of compression by respective impellers at the respective diffuser sections serving as the fluid path for the fluid output by the respective impellers.

[0037] The refrigeration apparatus according to the present invention includes a compressor for compressing a refrigerant admitted from an inlet opening and discharging the refrigerant from a discharge opening; a condenser for condensing and liquefying the refrigerant and forwarding a liquefied refrigerant; a throttling mechanism for reducing the pressure of the liquefied refrigerant; a vaporizer for cooling an object to be cooled by exchanging heat between the object to be cooled and a resultant condensed and pressure-reduced liquefied refrigerant, and evaporating and vaporizing the liquefied refrigerant, wherein the compressor is a turbo-type compressor described above.

[0038] In the present refrigeration apparatus, because a high efficiency turbo-type compressor, having a diffuser section that exhibits high performance and produces a high recovery coefficient, is used as a compressor to compress the refrigerant and to output the compressed refrigerant to the condenser, the cooling efficiency can be improved significantly, and accordingly, the refrigeration apparatus can produce superior cooling performance.

BRIEF DESCRIPTION OF DRAWINGS

[0039] FIG. 1 is a perspective view to explain the structure and construction of a turbo-type compressor in an embodiment of the present invention and a refrigeration apparatus having the compressor.

[0040] FIG. 2 is a schematic diagram to explain the structure of the turbo-type compressor and the refrigeration apparatus having the compressor in the embodiment of the present invention.

[0041] FIG. 3 is a cross sectional view of the turbo-type compressor to explain the construction of the turbo-type compressor according to the embodiment of the present invention.

[0042] FIG. 4 is a cross sectional view of a compression section to explain the construction of the turbo-type compressor according to the present invention.

[0043] FIG. 5 is a graph showing the performance of a diffuser section of the turbo-type compressor according to the embodiment of the present invention.

[0044] FIG. 6 is a cross sectional view of the compression section to explain the construction of a turbo-type compressor according to another embodiment of the present invention.

[0045] FIG. 7 is a graph showing the performance of the diffuser section of the turbo-type compressor according to the another embodiment of the present invention.
FIG. 8 is a cross sectional view to explain the construction of the turbo-type compressor according to yet another embodiment of the present invention.

FIG. 9 is a cross sectional view of the compression section of the turbo-type compressor according to still another embodiment of the present invention.

FIG. 10 is a cross sectional view to explain the construction of a conventional turbo-type compressor.

FIG. 11 is a graph showing the pressure recovery coefficient at the diffuser section.

BEST MODE FOR CARRYING OUT THE INVENTION

A turbo-type compressor and a refrigeration apparatus provided with the turbo-type compressor according to an embodiment of the present invention will be described with reference to the attached drawings.

An overall structure of the refrigeration apparatus will be explained first with reference to FIGS. 1 and 2.

The refrigeration apparatus shown in the figures includes a vaporizer 11 for cooling the cold water by means of heat exchange between the refrigerant and the cold water and for evaporating and vaporizing the refrigerant; a compressor 12 for compressing the refrigerant vaporized in the vaporizer 11; a condenser 13 for condensing and liquefying the refrigerant compressed in the compressor 12; a throttle valve 14 for reducing the pressure of the refrigerant liquefied in the condenser 13; an intermediate cooler 15 for temporarily storing and cooling the refrigerant liquefied in the condenser 13; and an oil cooler 16 for cooling the lubricating oil for the compressor 12 by utilizing a portion of the refrigerant cooled in the condenser 13.

Also, a motor (a driving mechanism) 17 is connected to the compressor 12 for operating the compressor 12.

The vaporizer 11, the compressor 12, the condenser 13, the throttle valve 14 and the intermediate cooler 15 are connected via a primary piping 18 to constitute a closed system in which the refrigerant is circulated.

The compressor 12 is based on a 2-stage (multi-stage) centrifugal compressor, a so-called turbo compressor, and this turbo compressor 12 is provided with a plurality of impellers 19. The refrigerant is compressed in a first stage impeller 19a situated in the upstream side of the impeller 19, and the compressed refrigerant is led into the second stage impeller 19b to be compressed further and is then sent to the condenser 13.

The condenser 13 includes a main condenser 13a and a sub-cooler 13b which is an auxiliary compressor, and the refrigerant is introduced first to the main condenser 13a and then to the sub-cooler 13b. However, a portion of the refrigerant cooled in the main condenser 13a is introduced into the oil cooler 16, without passing through the sub-cooler 13b, to cool the lubricating oil.

Also, apart from the above process, a portion of the refrigerant cooled in the main condenser 13a is introduced into the casing 31 of the motor 17, which will be explained later, without passing through the sub-cooler 13b, and cools stators and coils which are not shown in the diagram.

The throttle valve 14 is disposed between the condenser 13 and the intermediate cooler 15, and the intermediate cooler 15 and the vaporizer 11, and they are used for stepwise reduction of the pressure of the refrigerant liquefied in the condenser 13.

The structure of the intermediate cooler 15 is equivalent to a hollow vessel, and the refrigerant which has been cooled in the main condenser 13a and the sub-cooler 13b and reduced in pressure in the throttle valve 14, is temporarily stored therein and is subjected to further cooling. Here, the vapor phase components in the intermediate cooler 15 are introduced into the second stage impeller 19b of the compressor 12 through the bypass piping 23, without passing through the vaporizer 11.

The turbo-type compressor 12 provided for the above-mentioned refrigeration apparatus will be further explained in detail below.

As shown in FIG. 3, the motor 17 is provided integrally with the turbo-type compressor 12, which is operated by the rotational driving power of the motor 17.

The rotational power of the rotation shaft 35 of the motor 17 is transmitted to the rotation shaft 41 which constitutes the turbo-type compressor 12 by means of engaged transmission gears 36 and 37, thereby operating the rotation shaft 41 of the turbo-type compressor 12.

The turbo-type compressor 12 is designed so that one end side thereof is designated as the intake opening 42, and the refrigerant from the vaporizer 11 is thereby output to the intake opening 42. Intake vanes 40 are disposed at the intake opening 42 so that the intake vanes 40 control the intake volume of the refrigerant at the intake opening 42.

The turbo-type compressor 12 is provided with a first stage compression section 43, and a second stage compression section 44, in that order, from the intake opening 42 side, and provided on the first stage compression section 43 and the second stage compression section 44 are the first stage impeller 19a and the second stage impeller 19b described above.

Then, by rotating the rotation shaft 41, the first stage impeller 19a and the second stage impeller 19b are respectively rotated, and the refrigerant from the vaporizer 11 is withdrawn into the first stage compression section 43 from the intake opening 42, compressed by the first stage impeller 19a of the first stage compression section 43, output to the second compression section 44 via the return passage 49 which includes the diffuser section 46, return bend section 47 and the return vane 48, and is compressed by the second impeller 19b of the second compression section 44. After that, the refrigerant passes through the diffuser section 46 and is discharged from the discharge opening 53 via the scroll section 52, which is a fluid passage formed in the circumference direction, to be sent to the condenser 13.

As described above, the refrigerant sent from the intermediate cooler 15 is output to the second stage compression section 44, and is compressed together with the refrigerant output from the first stage compression section 43 by the second stage impeller 19b of the second stage compression section 44. Then, the refrigerant, as described above, passes through the diffuser section 46 and is dis-
charged from the discharge opening 53 via the scroll section 52, to be sent to the condenser 13.

[0067] Next, the structure of the diffuser section 46 in the first stage compression section 43 and the second compression section 44 will be explained by using the structure of the diffuser section 46 in the first stage compression section 43 as an example.

[0068] As shown in FIG. 4, in the diffuser section 46, the first wall section 56 includes the casing 55 of the turbo-type compressor 12, and the second wall section 58 that includes the partition plate 57 are formed in such a way to separate from each other in the radial direction, and in so doing, the diffuser section 46 is formed in a tapered shape so as to widen gradually from the inlet opening 46a towards the outlet opening 46b, with the result that the width dimension of the diffuser section 46 in the axial direction becomes gradually wider towards the outer radial direction.

[0069] As described above, the turbo-type compressor 12 is formed in such a way that, in the diffuser section 46, the width dimension b2 of the outlet opening 46b is wider than the width dimension b1 of the inlet opening 46a (b1>b2), and therefore, the aspect ratio of the inlet opening 46a and the outlet opening 46b of the diffuser section 46 is somewhat reduced and the area ratio of the inlet opening 46a to the outlet opening 46b of the diffuser section 46 is increased.

[0070] By adopting such a design, as shown in FIG. 5, in the case of the turbo-type compressor 12 having the diffuser section 46, the pressure recovery coefficient Cp is increased to a point located at the left and above the 0.5 value which is higher than the Cp exhibited by the conventional turbo-type compressor having the diffuser section 6 in which the width dimension of the inlet opening 46a is the same as that of the outlet opening 46b, and therefore, the performance of the diffuser section 46 is enhanced and the efficiency of the turbo-type compressor 12 is improved.

[0071] Similarly, the pressure recovery coefficient Cp is also improved in the diffuser section 46 of the second stage compression section 44.

[0072] Note that although only a two-stage (multi-stage type) turbo-type compressor having the first stage impeller 19a and the second stage impeller 19b is explained above, it is obvious that the present invention may be applied to a single-stage turbo-type compressor having one impeller.

[0073] As explained above, according to the turbo-type compressor 12 having the structure described above, because the width dimensions in the axial direction at the inlet opening 46a side and the outlet opening 46b side of the diffuser section 46 are made in such a way that the width of the outlet opening 46b side is larger than that of the inlet opening 46a side, the aspect ratio at the inlet opening 46a side and the outlet opening 46b side is made somewhat smaller and the area ratio thereof is made larger, so that it is possible to increase the pressure recovery coefficient Cp at the diffuser section 46.

[0074] In other words, without making the structure complex, it becomes possible to efficiently recover a dynamic pressure of the compressed refrigerant, which is output from the first stage impeller 19a and the second stage impeller 19b, in the form of a static pressure in the diffuser section 46, and accordingly, the turbo-type compressor 12 having a superior compression efficiency may be realized without making the apparatus larger or more complex.

[0075] Also, the tapered shape of the pair of wall sections, comprised by the first wall section 56 and the second wall section 58, which constitute the diffuser section 46, readily enables improvement in the performance by enlarging the width dimension of the outlet opening relative to that of the inlet opening. Also, because the first wall section 56 and the second wall section 58 are tapered, it is possible to eliminate a problem of separation of the refrigerant in the diffuser section 46, which is output from the first stage impeller 19a and the second stage impeller 19b.

[0076] Also, since the turbo-type compressor 12 described above is a two-stage type (multi-stage type) having the first stage impeller 19a and the second stage impeller 19b, and the pressure recovery coefficient Cp is made larger at the diffuser section 46, which is the passage for the refrigerant output from the first stage impeller 19a and the second stage impeller 19b, an extremely high efficiency turbo-type compressor may be realized, in which the efficiency has been increased in the first stage impeller 19a as well as in the second stage impeller 19b.

[0077] According to the refrigeration apparatus having the turbo-type compressor 12 explained above, because the highly efficient turbo-type compressor 12 having the diffuser section 46, which exhibits a superior performance in terms of the high pressure recovery coefficient Cp, is used, it becomes possible to significantly increase the cooling efficiency to provide a refrigeration apparatus having superior cooling characteristics.

[0078] Next, other embodiments according to the invention will be explained.

[0079] As shown in FIG. 6, in the case of this diffuser section 46 also, the first wall section 56 comprised by the casing 55 of the turbo-type compressor 12 and the second wall section 58 comprised by the partition plate 56 are disposed in such a way to separate from each other towards the outer radial direction to form a tapered diffuser section 46 that widens from the inlet opening 46a towards the outlet opening 46b, so that the width dimension of the diffuser section 46 gradually becomes wider towards the outer radial direction.

[0080] However, in this diffuser section 46, by making the radius R2 smaller at the outlet opening 46b, the area ratio of the inlet opening 46a to the outlet opening 46b is kept the same as the area ratio prior to the improvement.

[0081] In other words, the area ratio is left unchanged in this turbo-type compressor 12 while the aspect ratio is reduced, and therefore, in the case of the turbo-type compressor 12 having the diffuser section 46, the pressure recovery coefficient Cp shifts, as shown in FIG. 7, to the left so as to be above 0.5 as compared with the Cp of a conventional turbo-type compressor having a conventional diffuser section, thereby improving the performance of the diffuser section 46 and increasing the efficiency of the turbo-type compressor 12.

[0082] Accordingly, in the case of the turbo-type compressor 12 having the above-mentioned structure, because the width dimensions in the axial direction are such that the outlet opening 46b side is made larger relative to the inlet
opening 46a side without altering the area ratio of the inlet opening 46b side to the outlet opening 46b side in the diffuser section 46, the aspect ratio of the inlet opening 46a side and the outlet opening 46b side is reduced so that the pressure recovery coefficient \( C_p \) in the diffuser section 46 can be made larger.

[0083] In other words, without making the structure complex, it is possible to efficiently recover a dynamic pressure of the refrigerant, which is output from the first stage impeller 19a and the second stage impeller 19b, in the form of a static pressure in the diffuser section 46, and accordingly, the turbo-type compressor 12 may have a superior compression efficiency without making the apparatus larger or more complex.

[0084] Further, without altering the area ratio of the inlet opening 46a to the outlet opening 46b in the diffuser section 46, the width dimension b2 of the outlet opening 46b is made larger relative to the width dimension b1 of the inlet opening 46a, that is, because the radius R2 of the outlet opening 46b is reduced by an amount corresponding to the increase in the width dimension b2 of the outlet opening 46b, the outer radius can be made smaller and the cost of the compressor can be reduced.

[0085] Also, because the width dimension b2 of the outlet opening 46b is made larger relative to that of the inlet opening 46a without altering the area ratio of the inlet opening 46a to the outlet opening 46b, it becomes possible to assuredly improve the performance of the compressor 12 by reducing the aspect ratio and increasing the pressure recovery coefficient \( C_p \).

[0086] Note that although a tapered diffuser section 46 that widens from the inlet opening 46a towards the outlet opening 46b, and the width dimension of the diffuser section 46 which gradually becomes wider towards the outlet radial direction are formed by separating the first wall section 56, which includes the casing 55 of the turbo-type compressor 12, away from the second wall section 58, which includes the partition plate 57 towards the outlet radial direction, in all of the above embodiments, the efficiency of the turbo-type compressor 12 can be increased by increasing the pressure recovery coefficient \( C_p \) so long as the width dimension of the outlet opening 46b side is made wider than that of the inlet opening 46a side to an extent that would not cause a separation of fluid stream in the diffuser section 46.

[0087] Here, in the embodiment shown in FIG. 8, only the second wall section 58, which includes the partition plate 57, is inclined to form a tapered shape, and in the embodiment shown in FIG. 9, only the first wall 56, which includes the casing 55, is inclined to form a tapered shape. In either case, the degree of tapering is restricted so as not to cause a stream separation in the diffuser section 46.

[0088] According to the diffuser section 46 shown in FIG. 8 or 9, one of the wall sections, i.e., either the first wall section 56 or the second wall section 58, is tapered and the other wall section is oriented at substantially right angles to the rotation axis 41, so that the structure of the compressor may be simplified and the cost thereof may be reduced as compared with the case where both of the first wall section 56 and the second wall section 58 are tapered.

[0089] Here, when the second wall section 58 comprised by the partition plate 57 is tapered, it is necessary to incline the partition plate 57 itself towards the rear side so as to secure a curvature that does not generate a stream separation in the return bend section 47. However, in the embodiment shown in FIG. 9, since the second wall section 58 comprised by the partition plate 57 is inclined at a substantially right angle to the rotation axis 41, problems such as an increase in the dimension in the axial direction, resulting from slanting the partition plate 57 towards the rear side to secure a curvature that does not produce a stream separation in the return bend section 47, may be eliminated. Accordingly, it is possible to increase the efficiency of the compressor without increasing the dimension of the turbo-type compressor 12 in the axial direction.

[0090] That is, according to the turbo-type compressor 12 explained above, by tapering only one of the pair of wall sections, i.e., either the first wall section 56 or the second wall section 58, which constitute the diffuser section 46, the performance thereof can be improved quite easily by increasing the width dimension b2 of the outlet opening 46b relative to the inlet opening 46a. Also, because only one of the wall sections, either the first wall section 56 or the second wall section 58, needs to be tapered, the performance of the compressor 12 can be improved even more simply.

[0091] In particular, in the embodiment shown in FIG. 9, as described earlier, since the first wall section 56 is tapered, which constitutes the front side wall section that is more spacious as compared with the rear side of the impeller 19 that has a downstream side passage communicating with the diffuser section 46, the dimension in the axial direction can be made smaller.

[0092] Moreover, it is obvious that the structures of the diffuser section 46 shown in FIGS. 8 and 9 can be adapted to either of the structures of the diffuser section 46 shown in FIGS. 4 and 6.

[0093] Further, in the above embodiments, although the diffuser section 46 is illustrated using a vaneless type diffuser that has no vanes, the diffuser section 46 used in the present invention may be provided with vanes.

INDUSTRIAL APPLICABILITY

[0094] As explained above, according to the turbo-type compressor and the refrigeration apparatus provided with the turbo-type compressor of the present invention, the following effects may be obtained.

[0095] According to the turbo-type compressor of claim 1, since the width dimensions of the openings in the axial direction of the inlet side and the outlet side are made so that the outlet side is larger relative to the inlet side, the aspect ratio of the inlet side and the outlet side is made somewhat smaller and the area ratio is made larger, so that the pressure recovery coefficient in the diffuser section can be made larger. This design enables effective recovery of the dynamic pressure of the compressed fluid output from the impeller into a static pressure in the diffuser section without making the structure larger or more complex, and accordingly, a high compression efficiency is realized in the turbo-type compressor of the present invention.

[0096] According to the turbo-type compressor of claim 2, without altering the area ratio of the inlet opening to the outlet opening in the diffuser section, the width dimension of the outlet opening is made larger relative to the width
dimension of the inlet opening. That is, because the radius of the outlet opening is reduced by an amount corresponding to the increase in the width dimension of the outlet opening, the outer radius can be made smaller and the cost of the compressor can be reduced.

[0097] Also, even if the area ratio of the inlet opening to the outlet opening is not altered, because the width dimension of the outlet opening is made larger relative to that of the inlet opening, the aspect ratio is reduced, the pressure recovery coefficient is made larger so as to reliably improve the performance.

[0098] According to the turbo-type compressor of claim 3, by tapering the pair of wall sections forming the diffuser section, the performance of the compressor is readily improved by enlarging the width dimension of the outlet opening relative to that of the inlet opening. Also, because the wall sections are tapered, it is possible to eliminate a problem of stream separation in the diffuser section of the refrigerant output from the impeller.

[0099] According to the turbo-type compressor of claim 4, by making only one of the wall sections that constitute the diffuser section, into a tapered shape, the performance of the compressor can be improved quite readily by increasing the width dimension of the outlet opening relative to the inlet opening. Also, because only one of the wall sections needs to be made into a tapered shape, the performance can be improved even more easily.

[0100] In particular, by tapering the wall section in the front side which has more space as compared with the rear side of the impeller having a downstream side passage that communicates with the diffuser section, the dimension in the axial direction can be made smaller.

[0101] According to the turbo-type compressor of claim 5, because the pressure recovery coefficient is improved in each stage of compression by respective impellers at the respective diffuser sections serving as fluid paths for the fluid output by the respective impellers, a superior multi-stage turbo-type compressor of high performance may be provided.

[0102] According to the refrigeration apparatus of claim 6, because a high efficiency turbo-type compressor, having a diffuser section that exhibits high performance and produces a high recovery coefficient, is used as a compressor to compress the refrigerant and to output the compressed refrigerant to the condenser, the cooling efficiency can be improved significantly, and accordingly, the refrigeration apparatus can produce superior cooling performance.

1. A refrigeration apparatus, comprising: a compressor for compressing a refrigerant admitted from an intake opening and discharging the refrigerant from a discharge opening; a condenser for condensing and liquefying the refrigerant and forwarding a resultant liquefied refrigerant; a throttling mechanism for reducing the pressure of the liquefied refrigerant; and a vaporizer for cooling an object to be cooled by exchanging heat between the object to be cooled and a resultant condensed and pressure-reduced liquefied refrigerant, and evaporating and vaporizing the liquefied refrigerant, wherein

the compressor is a turbo-type compressor comprising: a casing provided with an intake opening and a discharge opening; a rotation shaft operated by a driving mechanism; an impeller provided integrally with the rotation shaft; a diffuser section comprised of a pair of wall sections at outer peripheries of the impeller to serve as a fluid passage for a fluid driven towards the outer periphery side by the rotation of the impeller so that the fluid is drawn in from the intake opening, by the action of the impeller which is rotated together with the rotation shaft and driven by the driving mechanism, to be compressed and discharged from the discharge opening through the diffuser section; wherein

the diffuser section is formed in such a way that the width dimension in the axial direction of an outlet opening located at the outer periphery side is made larger than the width dimension of an inlet opening for the fluid which is driven by the impeller.

2. A refrigeration apparatus according to claim 1, wherein the width dimension of the outlet opening is made larger than the width dimension of the inlet opening without altering the area ratio of the inlet opening to the outlet opening of the diffuser section.

3. A refrigeration apparatus according to claim 1, wherein the pair of wall sections forming the diffuser section is made in a tapered shape so as to separate gradually from each other from the inlet opening towards the outlet opening.

4. A refrigeration apparatus according to claim 1, wherein one of the pair of wall sections forming the diffuser section is made into a tapered shape so as to separate gradually from the other one of the pair of wall sections from the inlet opening towards the outlet opening.

5. A refrigeration apparatus according to claim 1, wherein the turbo-type compressor is a multi-stage compressor having a plurality of the impellers and compresses the fluid from the intake opening sequentially by using, first, an upstream-side impeller, and successive impellers afterward.

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