



US 20170260987A1

(19) **United States**(12) **Patent Application Publication**  
**Onodera**(10) **Pub. No.: US 2017/0260987 A1**(43) **Pub. Date: Sep. 14, 2017**(54) **CENTRIFUGAL COMPRESSOR WITH  
CASING TREATMENT BYPASS**(71) Applicant: **Daikin Applied Americas Inc.,**  
Minneapolis, MN (US)(72) Inventor: **Fumiaki Onodera,** Minnetonka, MN  
(US)(21) Appl. No.: **15/067,318**(22) Filed: **Mar. 11, 2016****Publication Classification**(51) **Int. Cl.**

|                   |           |
|-------------------|-----------|
| <i>F04D 27/02</i> | (2006.01) |
| <i>F04D 29/44</i> | (2006.01) |
| <i>F25B 49/02</i> | (2006.01) |
| <i>F04D 29/28</i> | (2006.01) |
| <i>F25B 1/10</i>  | (2006.01) |
| <i>F25B 31/02</i> | (2006.01) |
| <i>F04D 29/42</i> | (2006.01) |
| <i>F04D 29/46</i> | (2006.01) |

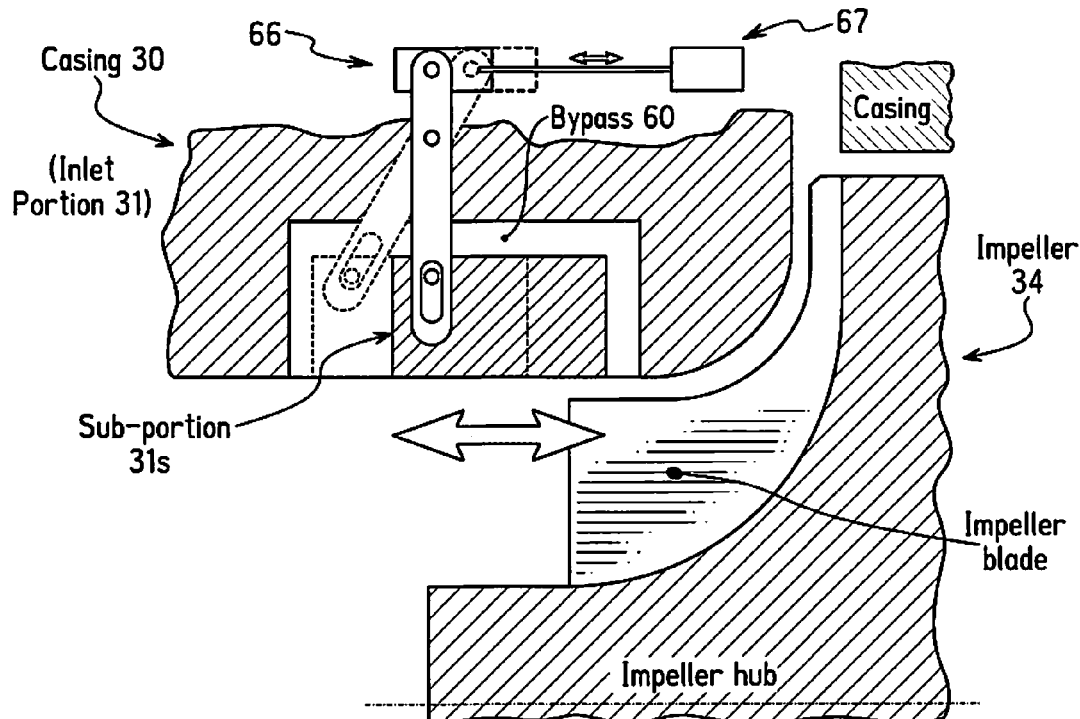
(52) **U.S. Cl.**

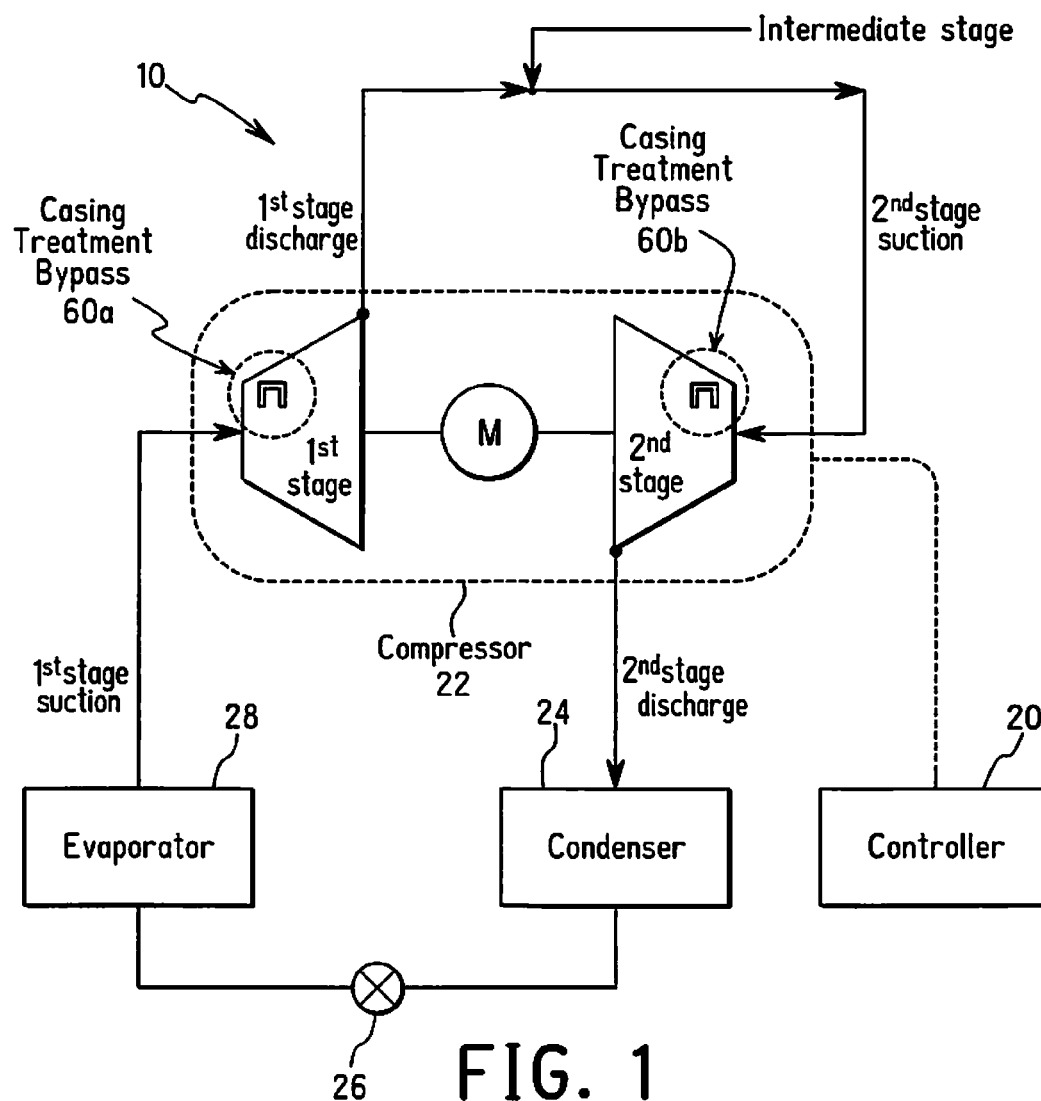
CPC ..... *F04D 27/0215* (2013.01); *F04D 29/4206*  
(2013.01); *F04D 29/444* (2013.01); *F04D*  
*29/462* (2013.01); *F04D 29/284* (2013.01);  
*F04D 27/0246* (2013.01); *F25B 1/10*  
(2013.01); *F25B 31/026* (2013.01); *F25B*  
*49/022* (2013.01)

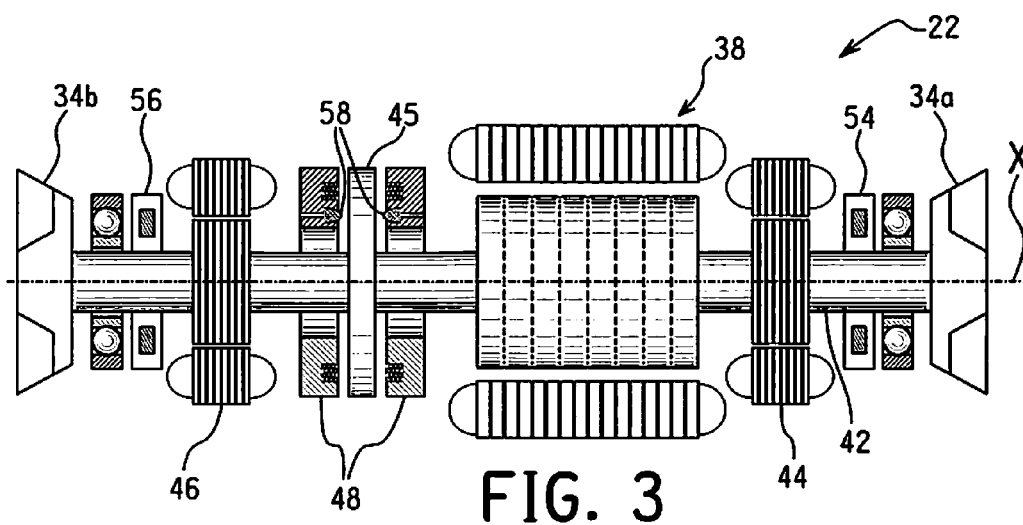
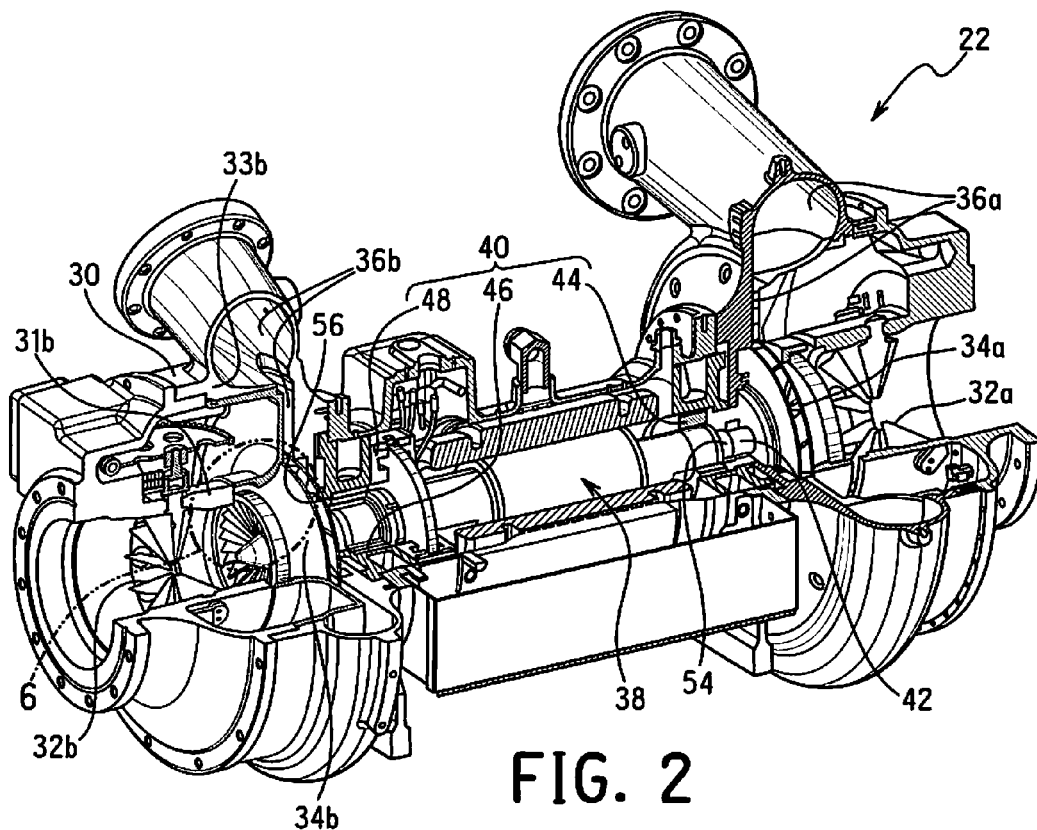
(57)

**ABSTRACT**

A centrifugal compressor for a chiller system includes a casing, an inlet guide vane, an impeller disposed downstream of the inlet guide vane, a motor and a diffuser. The casing has inlet and outlet portions with the inlet guide vane disposed in the inlet portion. The impeller is attached to a shaft rotatable about a rotation axis, and the motor rotates the shaft in order to rotate the impeller. The centrifugal compressor further includes a casing treatment bypass having an entrance port and an exit port. The casing treatment bypass injects refrigerant from a gap between the impeller and the inlet portion of the casing toward an area between the impeller and the inlet guide vane. The exit port of the casing treatment bypass is positioned upstream in a direction of a refrigerant flow with respect to the entrance port of the casing treatment bypass.







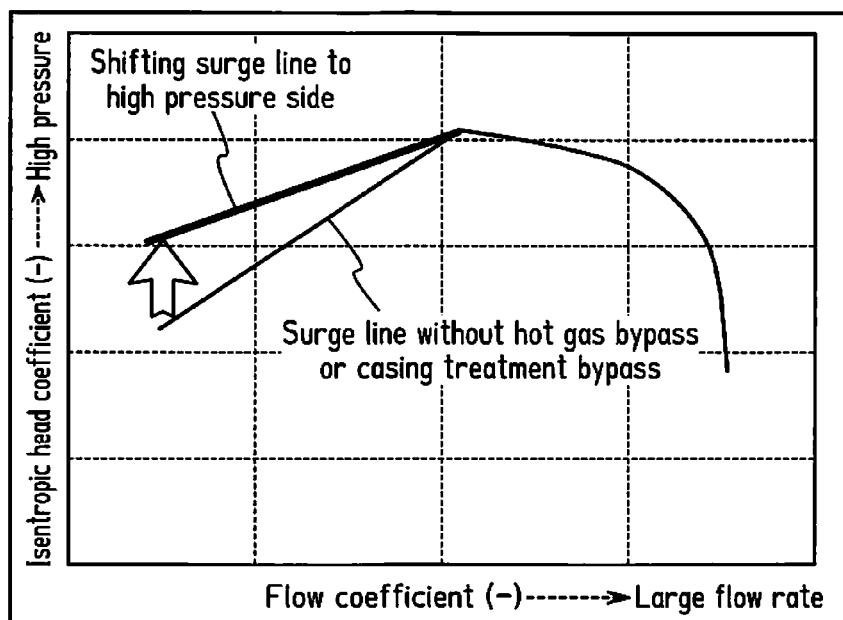


FIG. 4A

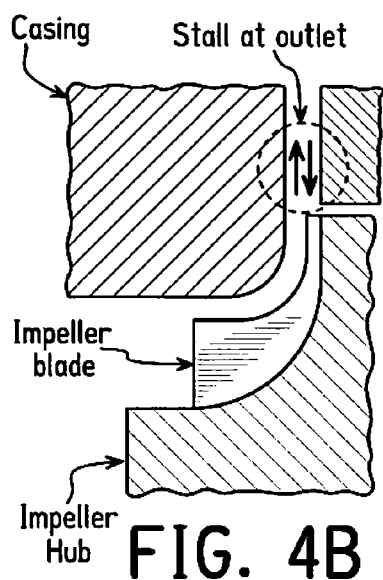


FIG. 4B

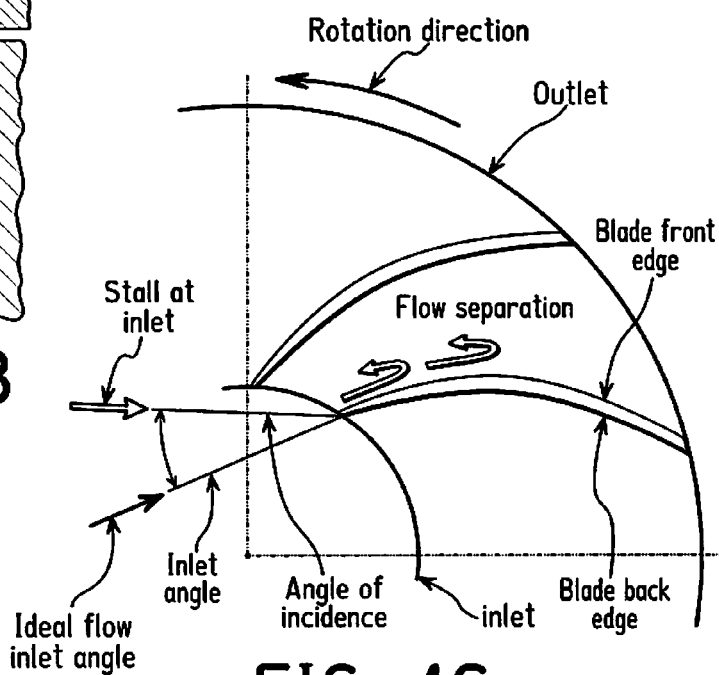


FIG. 4C

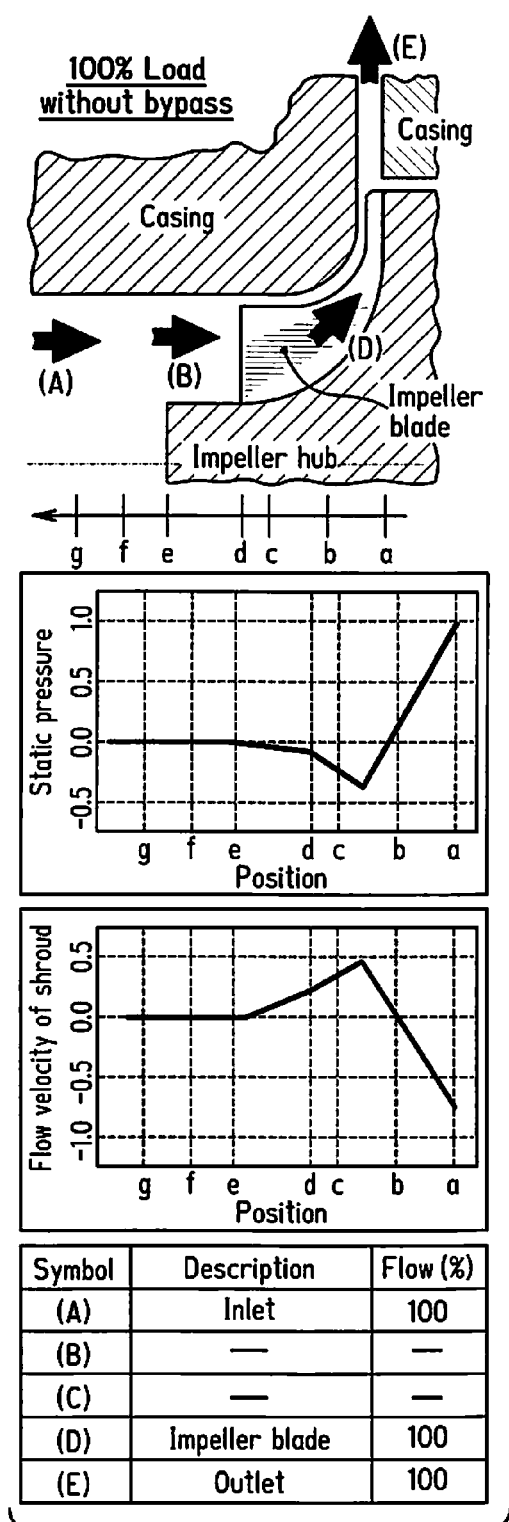


FIG. 5A

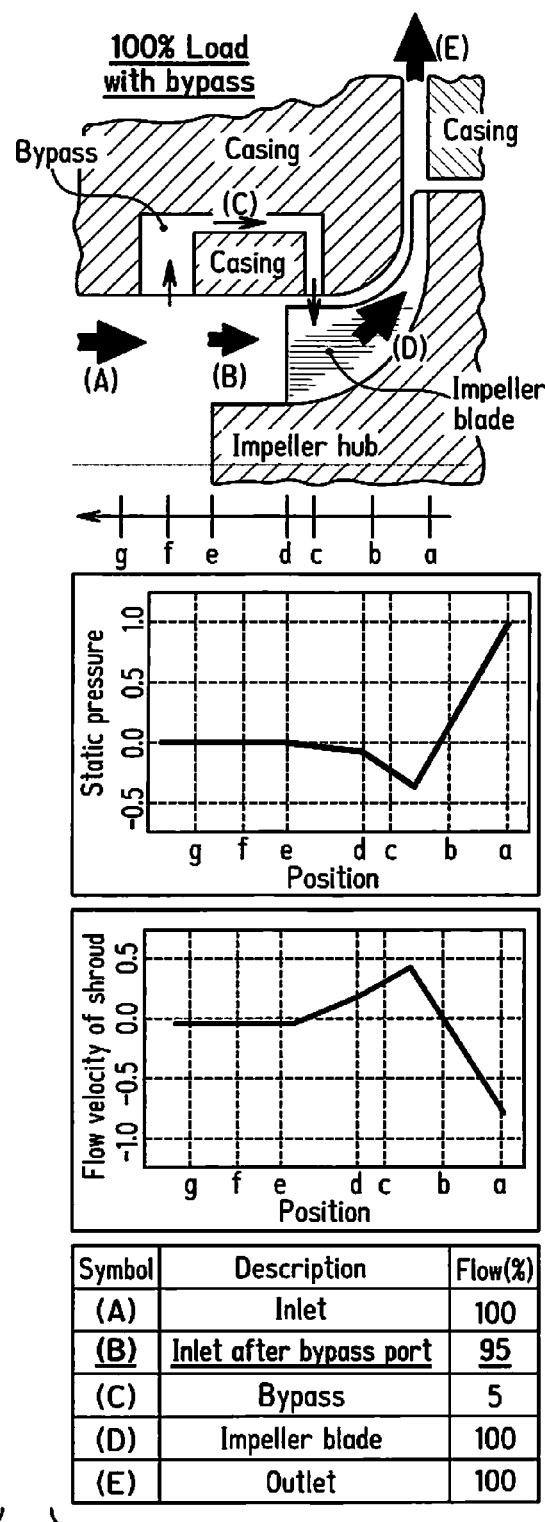


FIG. 5B

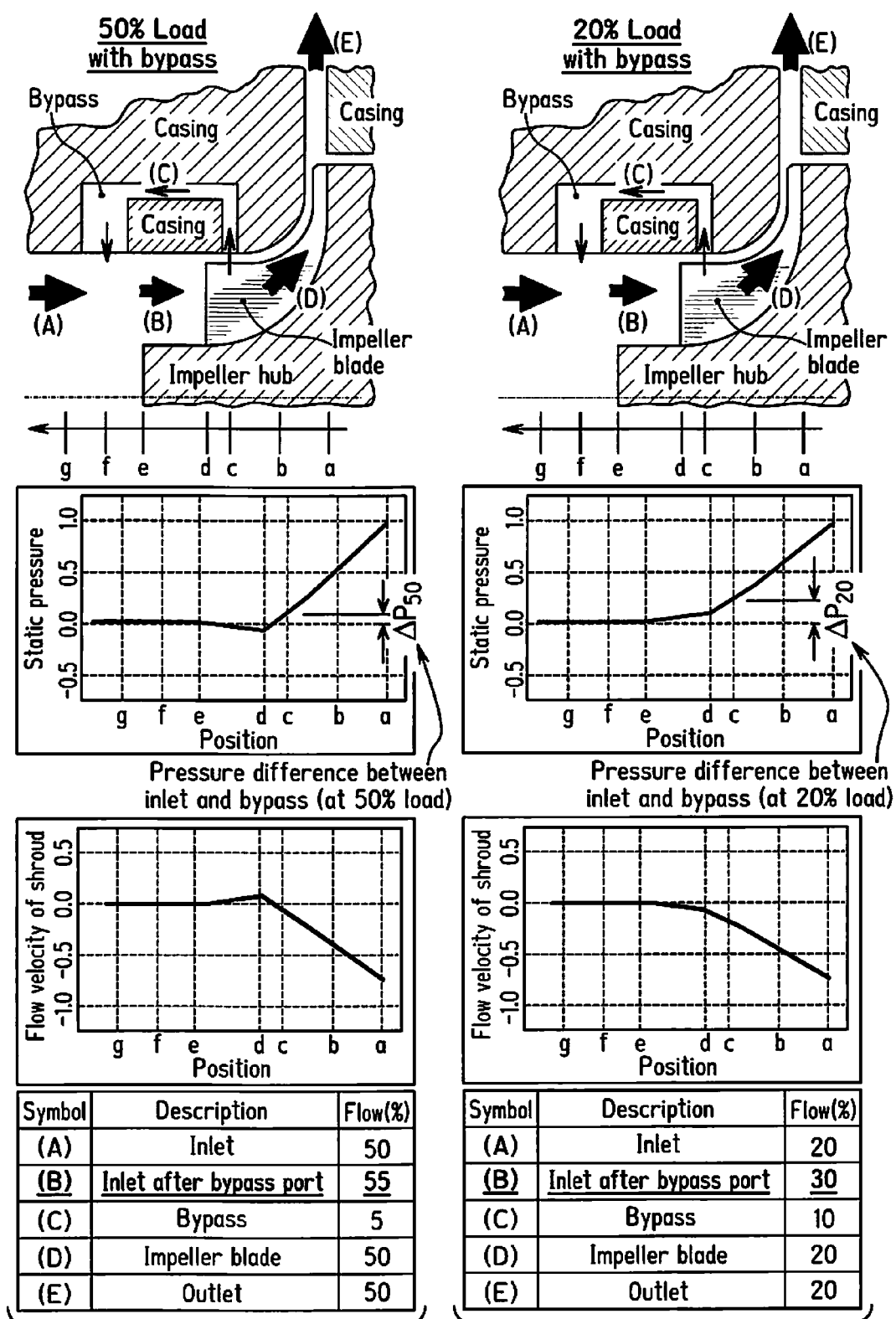


FIG. 5C

FIG. 5D

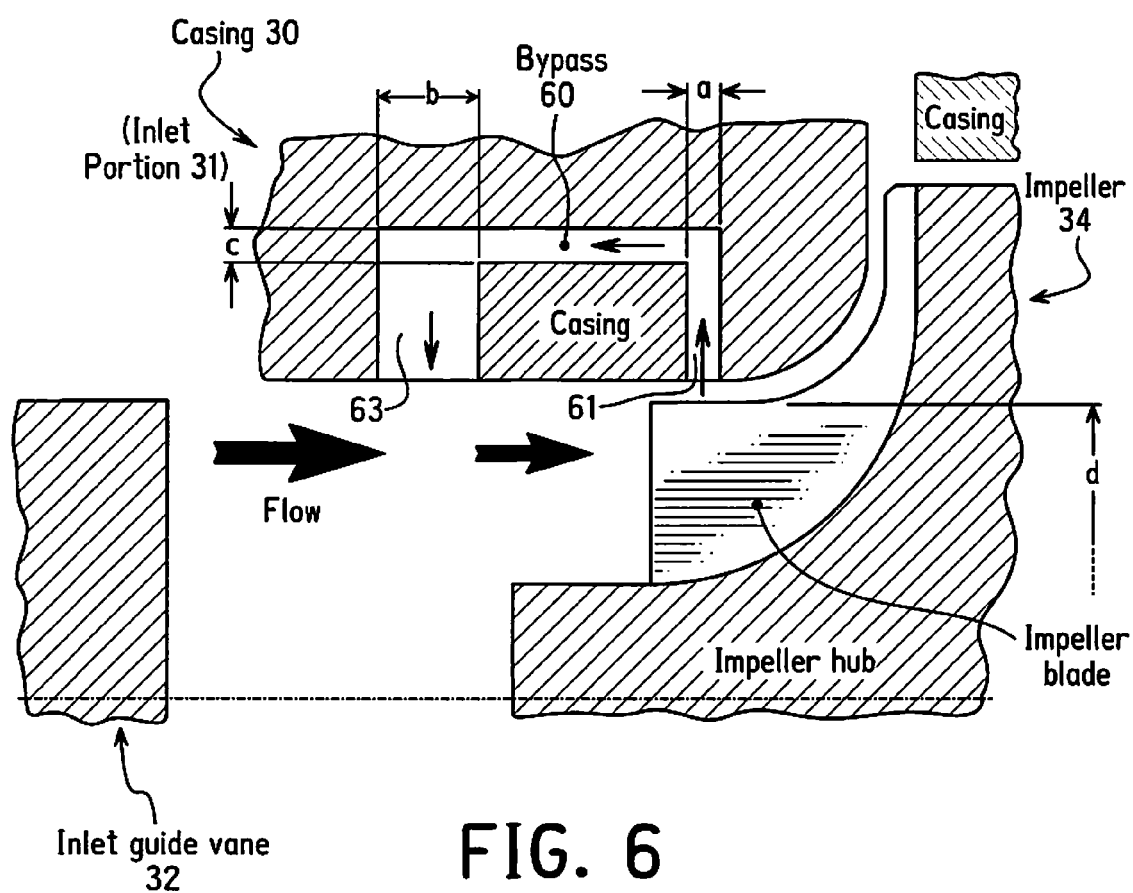


FIG. 6

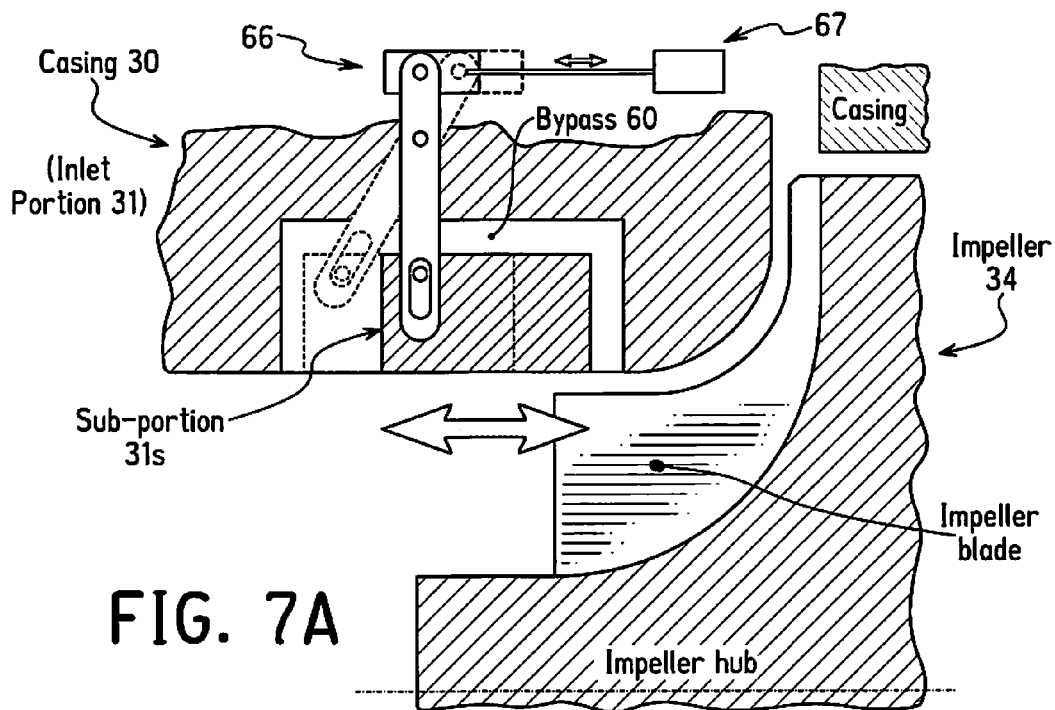


FIG. 7A

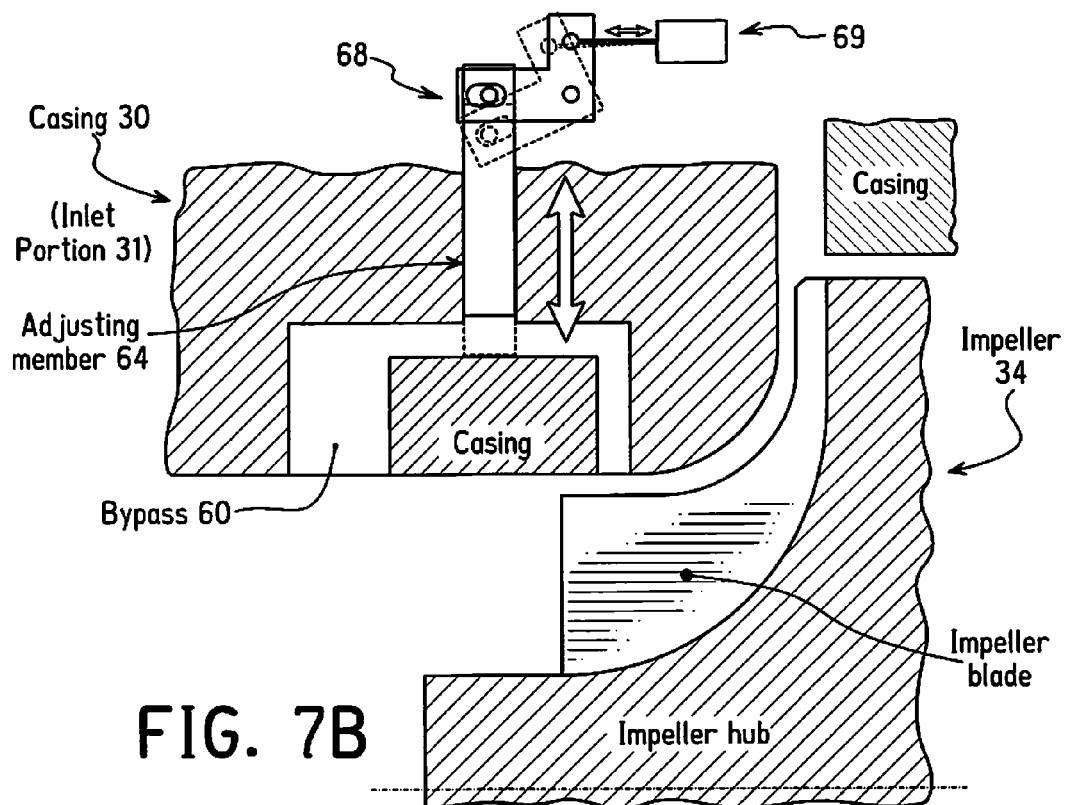


FIG. 7B

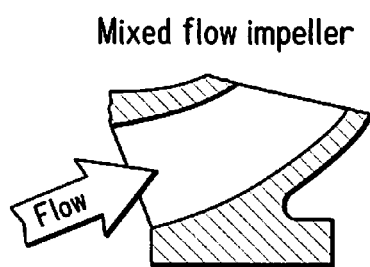


FIG. 8A

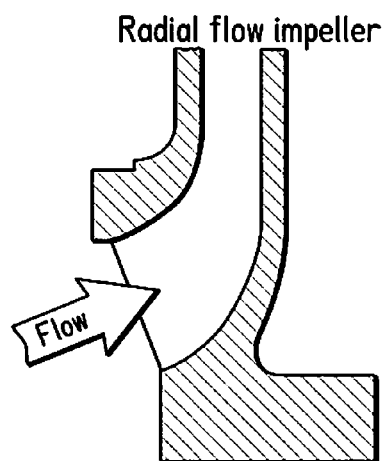


FIG. 8B

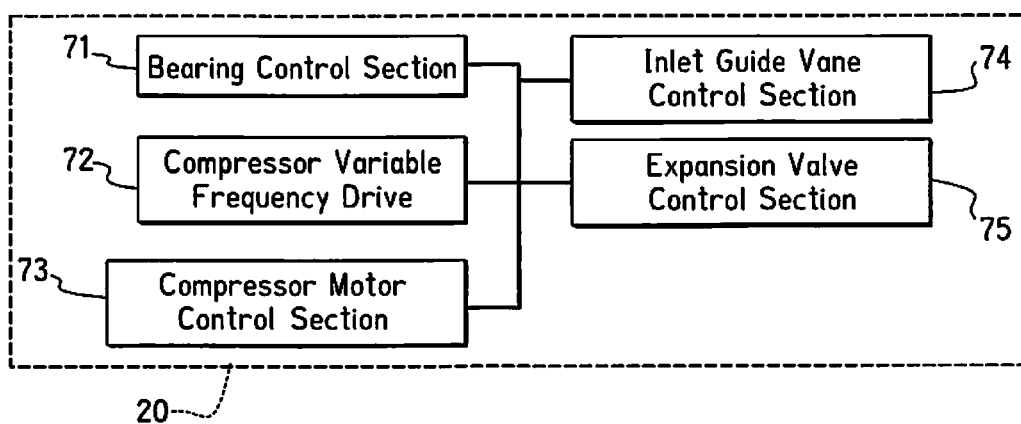
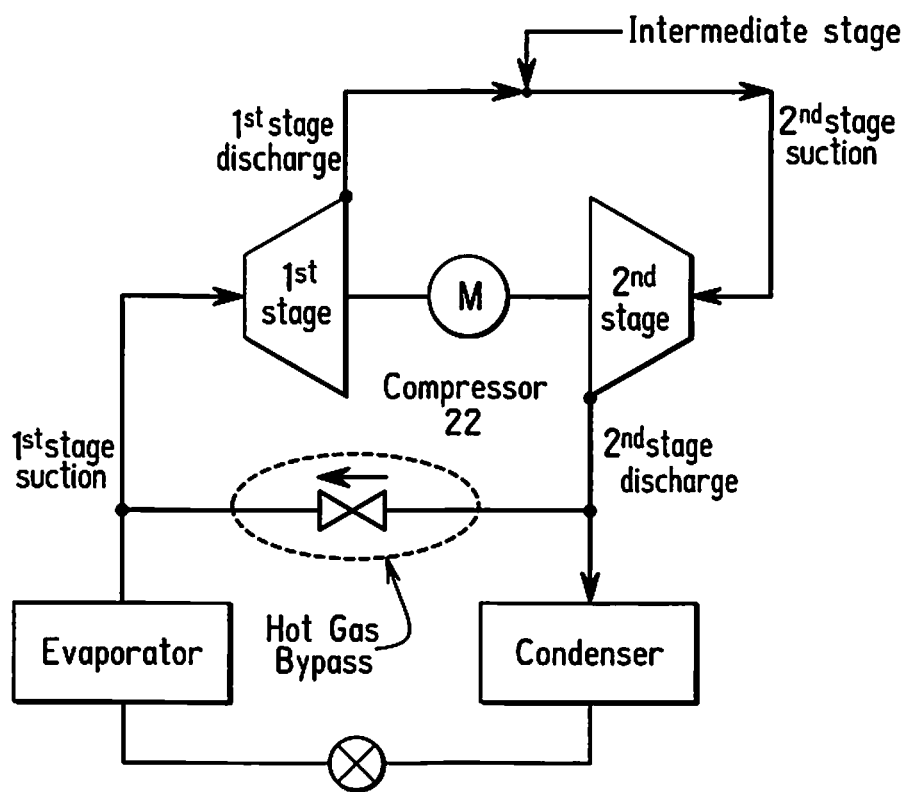


FIG. 9



**FIG. 10**  
(PRIOR ART)

## CENTRIFUGAL COMPRESSOR WITH CASING TREATMENT BYPASS

### BACKGROUND

**[0001]** Field of the Invention

**[0002]** The present invention generally relates to a centrifugal compressor used in a chiller system. More specifically, the present invention relates to a centrifugal compressor with a casing treatment bypass.

**[0003]** Background Information

**[0004]** A chiller system is a refrigerating machine or apparatus that removes heat from a medium. Commonly a liquid such as water is used as the medium and the chiller system operates in a vapor-compression refrigeration cycle. This liquid can then be circulated through a heat exchanger to cool air or equipment as required. As a necessary byproduct, refrigeration creates waste heat that must be exhausted to ambient or, for greater efficiency, recovered for heating purposes. A conventional chiller system often utilizes a centrifugal compressor, which is often referred to as a turbo compressor. Thus, such chiller systems can be referred to as turbo chillers. Alternatively, other types of compressors, e.g., a screw compressor, can be utilized.

**[0005]** In a conventional (turbo) chiller, refrigerant is compressed in the centrifugal compressor and sent to a heat exchanger in which heat exchange occurs between the refrigerant and a heat exchange medium (liquid). This heat exchanger is referred to as a condenser because the refrigerant condenses in this heat exchanger. As a result, heat is transferred to the medium (liquid) so that the medium is heated. Refrigerant exiting the condenser is expanded by an expansion valve and sent to another heat exchanger in which heat exchange occurs between the refrigerant and a heat exchange medium (liquid). This heat exchanger is referred to as an evaporator because refrigerant is heated (evaporated) in this heat exchanger. As a result, heat is transferred from the medium (liquid) to the refrigerant, and the liquid is chilled. The refrigerant from the evaporator is then returned to the centrifugal compressor and the cycle is repeated. The liquid utilized is often water.

**[0006]** A conventional centrifugal compressor basically includes a casing, an inlet guide vane, an impeller, a diffuser, a motor, various sensors and a controller. Refrigerant flows in order through the inlet guide vane, the impeller and the diffuser. Thus, the inlet guide vane is coupled to a gas intake port of the centrifugal compressor while the diffuser is coupled to a gas outlet port of the impeller. The inlet guide vane controls the flow rate of refrigerant gas into the impeller. The impeller increases the velocity of refrigerant gas. The diffuser works to transform the velocity of refrigerant gas (dynamic pressure), given by the impeller, into (static) pressure. The motor rotates the impeller. The controller controls the motor, the inlet guide vane and the expansion valve. In this manner, the refrigerant is compressed in a conventional centrifugal compressor.

**[0007]** When the pressure next to the compressor discharge is higher than the compressor discharge pressure, the fluid tends to reverse or even flow back in the compressor. This happens when the lift pressure (condenser pressure-evaporator pressure) exceeds the compressor lift capability. This phenomenon, called surge, repeats and occurs in cycles. The compressor loses the ability to maintain its lift when surge occurs and the entire system becomes unstable. A collection of surge points during varying compressor speed

or varying inlet gas angle is called a surge surface. In normal conditions, the compressor operates in the right side of the surge surface. However, during startup/operation in part load, the operating point will move towards the surge line because flow is reduced. If conditions are such that the operating point approaches the surge line, flow recirculation occurs in the impeller and diffuser. The flow separation will eventually cause a decrease in the discharge pressure, and flow from suction to discharge will resume. Surging can cause damage to the mechanical impeller/shaft system and/or to the thrust bearing due to the rotor shifting back and forth from the active to the inactive side. This is defined as the surge cycle of the compressor.

**[0008]** Therefore, techniques have been developed to control surge. See for example U.S. Patent Application Publication No. 2014/0260385 and U.S. Patent Application Publication No. 2014/0260388.

### SUMMARY

**[0009]** In a conventional centrifugal compressor as shown in FIG. 10, a hot gas bypass may be provided to connect the discharge side of the compressor and the suction side of the compressor to expand the operation range of the compressor. While this technique works relatively well, this system requires a pipe of a large diameter for the hot gas bypass, which results in increased costs. Especially in a case where a centrifugal compressor uses low pressure refrigerant such like R1233zd, the specific volume of the refrigerant is large compared to a case of conventional refrigerant such like R134a. This requires a large-diameter pipe for the hot gas bypass, which results in increased costs.

**[0010]** Also, a conventional centrifugal compressor with a hot gas bypass requires sensitive control using a variable frequency drive (VFD) to prevent surge.

**[0011]** Therefore, one object of the present invention is to provide a centrifugal compressor that expands the operation range of the compressor to prevent surge without increased costs.

**[0012]** Another object of the present invention is to provide a centrifugal compressor that prevents surge from occurring without performing sensitive control.

**[0013]** One or more of the above objects can basically be attained by providing a centrifugal compressor adapted to be used in a chiller system, the centrifugal compressor including a casing having an inlet portion and an outlet portion, an inlet guide vane disposed in the inlet portion, an impeller disposed downstream of the inlet guide vane, the impeller being attached to a shaft rotatable about a rotation axis, a motor arranged and configured to rotate the shaft in order to rotate the impeller, a diffuser disposed in the outlet portion downstream of the impeller, and a casing treatment bypass having an entrance port and an exit port, the casing treatment bypass being arranged and configured to inject refrigerant from a gap between the impeller and the inlet portion of the casing toward an area between the impeller and the inlet guide vane, and the exit port of the casing treatment bypass being positioned upstream in a direction of a refrigerant flow with respect to the entrance port of the casing treatment bypass.

**[0014]** These and other objects, features, aspects and advantages of the present invention will become apparent to those skilled in the art from the following detailed description, which, taken in conjunction with the annexed drawings, discloses preferred embodiments.

## BRIEF DESCRIPTION OF THE DRAWINGS

[0015] Referring now to the attached drawings which form a part of this original disclosure:

[0016] FIG. 1 is a schematic diagram illustrating a chiller system in accordance with an embodiment of the present invention which includes a casing treatment bypass;

[0017] FIG. 2 is a perspective view of the centrifugal compressor of the chiller system illustrated in FIG. 1, with portions broken away and shown in cross-section for the purpose of illustration;

[0018] FIG. 3 is a schematic longitudinal cross-sectional view of the impeller, motor and magnetic bearing of the centrifugal compressor illustrated in FIG. 2;

[0019] FIG. 4A is a graph illustrating expanding the operation range of a compressor by shilling a surge line, FIG. 4B is a schematic diagram illustrating surge and a stall at the outlet, and FIG. 4C is a schematic diagram illustrating a stall at the inlet;

[0020] FIG. 5A illustrates the static pressure, the flow velocity, and the flow rate of refrigerant without a casing treatment bypass, and FIGS. 5B-5D illustrate the static pressure, the flow velocity, and the flow rate of refrigerant with a casing treatment bypass of various load;

[0021] FIG. 6 is an enlarged schematic diagram inside circle 6 in FIG. 2, illustrating the inlet guide vane, the impeller and the casing of the centrifugal compressor of FIGS. 1-3 with a casing treatment bypass;

[0022] FIGS. 7A and 7B are schematic diagrams illustrating other embodiments of the casing treatment bypass;

[0023] FIG. 8A is a schematic diagram illustrating a mixed flow impeller and FIG. 8B is a schematic diagram illustrating a radial flow impeller;

[0024] FIG. 9 is a schematic diagram of the chiller controller of the chiller system of FIG. 1; and

[0025] FIG. 10 illustrates a conventional chiller system which includes a hot gas bypass.

## DETAILED DESCRIPTION OF EMBODIMENT(S)

[0026] Selected embodiments will now be explained with reference to the drawings. It will be apparent to those skilled in the art from this disclosure that the following descriptions of the embodiments are provided for illustration only and not for the purpose of limiting the invention as defined by the appended claims and their equivalents.

[0027] Referring initially to FIG. 1, a chiller system 10, which includes a casing treatment bypass 60 (60a, 60b), is illustrated in accordance with an embodiment of the present invention. The chiller system 10 is preferably a water chiller that utilizes cooling water and chiller water in a conventional manner. The chiller system 10 illustrated herein is a two-stage chiller system. However, it will be apparent to those skilled in the art from this disclosure that the chiller system 10 could be a single stage chiller system or a multiple stage chiller system including three or more stages.

[0028] The chiller system 10 basically includes a chiller controller 20, a compressor 22, a condenser 24, an expansion valve 26, and an evaporator 28 connected together in series to form a loop refrigeration cycle. In addition, various sensors (not shown) are disposed throughout the circuit of the chiller system 10. The chiller system 10 is conventional

except that the chiller system has the casing treatment bypass 60 (60a, 60b) in accordance with the present invention.

[0029] Referring to FIGS. 1-3, the compressor 22 is a two-stage centrifugal compressor in the illustrated embodiment. The compressor 22 illustrated herein is a two-stage centrifugal compressor which includes two impellers. However, the compressor 22 can be a multiple stage centrifugal compressor including three or more impellers. The two-stage centrifugal compressor 22 of the illustrated embodiment includes a first stage impeller 34a and a second stage impeller 34b. The centrifugal compressor 22 further includes a first stage inlet guide vane 32a, a first diffuser/volute 36a, a second stage inlet guide vane 32b, a second diffuser/volute 36b, a compressor motor 38, and a magnetic bearing assembly 40 as well as various conventional sensors (only some shown). A casing 30 covers the centrifugal compressor 22. The casing 30 includes an inlet portion 31a and an outlet portion 33a for the first stage of the compressor 22. The casing 30 also includes an inlet portion 31b and an outlet portion 33b for the second stage of the compressor 22.

[0030] The chiller controller 20 receives signals from the various sensors and controls the inlet guide vanes 32a and 32b, the compressor motor 38, and the magnetic bearing assembly 40 in a conventional manner, as explained in more detail below. Refrigerant flows in order through the first stage inlet guide vane 32a, the first stage impeller 34a, the second stage inlet guide vane 32b, and the second stage impeller 34b. The inlet guide vanes 32a and 32b control the flow rate of refrigerant gas into the impellers 34a and 34b, respectively, in a conventional manner. The impellers 34a and 34b increase the velocity of refrigerant gas, generally without changing pressure. The motor speed determines the amount of increase of the velocity of refrigerant gas. The diffusers/volutes 36a and 36b increase the refrigerant pressure. The diffusers/volutes 36a and 36b are non-movably fixed relative to the casing 30. The compressor motor 38 rotates the impellers 34a and 34b via a shaft 42. The magnetic bearing assembly 40 magnetically supports the shaft 42. Alternatively, the bearing system may include a roller element, a hydrodynamic bearing, a hydrostatic bearing, and/or a magnetic bearing, or any combination of these. In this manner, the refrigerant is compressed in the centrifugal compressor 22.

[0031] In operation of the chiller system 10, the first stage impeller 34a and the second stage impeller 34b of the compressor 22 are rotated, and the refrigerant of low pressure in the chiller system 10 is sucked by the first stage impeller 34a. The flow rate of the refrigerant is adjusted by the inlet guide vane 32a. The refrigerant sucked by the first stage impeller 34a is compressed to intermediate pressure, the refrigerant pressure is increased by the first diffuser/volute 36a, and the refrigerant is then introduced to the second stage impeller 34b. The flow rate of the refrigerant is adjusted by the inlet guide vane 32b. The second stage impeller 34b compresses the refrigerant of intermediate pressure to high pressure, and the refrigerant pressure is increased by the second diffuser/volute 36b. The high pressure gas refrigerant is then discharged to the chiller system 10.

[0032] Referring to FIGS. 2 and 3, the magnetic bearing assembly 40 is conventional, and thus, will not be discussed and/or illustrated in detail herein, except as related to the present invention. Rather, it will be apparent to those skilled

in the art that any suitable magnetic bearing can be used without departing from the present invention. The magnetic bearing assembly 40 preferably includes a first radial magnetic bearing 44, a second radial magnetic bearing 46 and an axial (thrust) magnetic bearing 48. In any case, at least one radial magnetic bearing 44 or 46 rotatably supports the shaft 42. The thrust magnetic bearing 48 supports the shaft 42 along a rotational axis X by acting on a thrust disk 45. The thrust magnetic bearing 48 includes the thrust disk 45 which is attached to the shaft 42.

[0033] The thrust disk 45 extends radially from the shaft 42 in a direction perpendicular to the rotational axis X, and is fixed relative to the shaft 42. A position of the shaft 42 along rotational axis X (an axial position) is controlled by an axial position of the thrust disk 45. The first and second radial magnetic bearings 44 and 46 are disposed on opposite axial ends of the compressor motor 38. Various sensors detect radial and axial positions of the shaft 42 relative to the magnetic bearings 44, 46 and 48, and send signals to the chiller controller 20 in a conventional manner. The chiller controller 20 then controls the electrical current sent to the magnetic bearings 44, 46 and 48 in a conventional manner to maintain the shaft 42 in the correct position.

[0034] The magnetic bearing assembly 40 is preferably a combination of active magnetic bearings 44, 46, and 48, which utilizes gap sensors 54, 56 and 58 to monitor shaft position and send signals indicative of shaft position to the chiller controller 20. Thus, each of the magnetic bearings 44, 46 and 48 are preferably active magnetic bearings. A magnetic bearing control section 71 uses this information to adjust the required current to a magnetic actuator to maintain proper rotor position both radially and axially.

[0035] Referring to FIGS. 1 and 9, the chiller controller 20 includes a magnetic bearing control section 71, a compressor variable frequency drive 72, a compressor motor control section 73, an inlet guide vane control section 74, and an expansion valve control section 75. The compressor variable frequency drive 72 and the compressor motor control section 73 can be a single section.

[0036] In the illustrated embodiment, the control sections are sections of the chiller controller 20 programmed to execute the control of the parts described herein. The magnetic bearing control section 71, the compressor variable frequency drive 72, the compressor motor control section 73, and the inlet guide vane control section 74, and the expansion valve control section 75 are coupled to each other, and form parts of a centrifugal compressor control portion that is electrically coupled to an I/O interface of the compressor 22. However, it will be apparent to those skilled in the art from this disclosure that the precise number, location and/or structure of the control sections, portions and/or chiller controller 20 can be changed without departing from the present invention so long as the one or more controllers are programed to execute control of the parts of the chiller system 10 as explained herein.

[0037] The chiller controller 20 is conventional, and thus, includes at least one microprocessor or CPU, an Input/output (I/O) interface, Random Access Memory (RAM), Read Only Memory (ROM), a storage device (either temporary or permanent) forming a computer readable medium programmed to execute one or more control programs to control the chiller system 10. The chiller controller 20 may optionally include an input interface such as a keypad to receive inputs from a user and a display device used to

display various parameters to a user. The parts and programming are conventional, and thus, will not be discussed in detail herein, except as needed to understand the embodiment(s).

[0038] As mentioned above, the chiller system 10 has the casing treatment bypass 60 (60a, 60b) in accordance with the present invention. In the illustrated embodiment, the compressor 22 is a two-stage centrifugal compressor. A first stage casing treatment bypass 60a and a second stage casing treatment bypass 60b are provided in the first stage and the second stage of the compressor 22, respectively, as shown in FIG. 1. It will be apparent to those skilled in the art from this disclosure that the structures of the first stage casing treatment bypass 60a and the second stage casing treatment bypass 60b are identical, except that they are mirror images of each other. Therefore, the first stage casing treatment bypass 60a and the second stage casing treatment bypass 60b are collectively referred to as the casing treatment bypass 60 hereinafter.

[0039] In the same manner, the elements of the first stage and the second stage of the compressor 22 are collectively referred to hereinafter without being distinguished. For example, the inlet portion 31a of the casing 30 for the first stage and the inlet portion 31b of the casing 30 for the second stage are collectively referred to as the inlet portion 31 of the casing 30. The first stage inlet guide vane 32a and the second stage inlet guide vane 32a are collectively referred to as the inlet guide vane 32. The first stage impeller 34a and the second stage impeller 34b are collectively referred to as the impeller 34.

[0040] In accordance with the present invention, the casing treatment bypass 60 is provided in the chiller system 10 to inject refrigerant from a gap between the impeller 34 and the inlet portion 31 of the casing 30 toward an area between the impeller 34 and the inlet guide vane 32, as explained in more detail below.

[0041] Referring to FIG. 4A, the operation range of a compressor is expanded at low load (i.e., at a small flow rate) by shifting a surge line to the high pressure side. The surge line is a line connecting pressure limit values in which the compressor cannot be operated at a predetermined flow rate. As shown in FIG. 4B, when surge occurs, the flow moves forward or backward repeatedly at the outlet, which results in severe vibration of the pressure and the flow rate. The severe vibration causes a stall of the flow at the outlet. As shown in FIG. 4C, a stall of the flow occurs also at the inlet in a case where the flow rate is reduced. More specifically, when the flow rate is reduced, the flow goes into the blade of the impeller with a larger angle of incidence compared to the ideal flow inlet angle, which causes the flow separation from the front edge of the blade. The flow separation causes the stall of the flow at the inlet. The casing treatment bypass 60 in accordance with the present invention is provided to prevent the flow separation and the stall of the flow at the inlet so as to expand the operation range of the compressor 22, as explained in more detail below.

[0042] FIG. 5A illustrates the static pressure, the flow velocity, and the flow rate of refrigerant without a casing treatment bypass, and FIGS. 5B-5D illustrate the static pressure, the flow velocity, and the flow rate of refrigerant with a casing treatment bypass of various load. In each of FIGS. 5A-5D, the upper graph shows the static pressure of the refrigerant at positions "a" to "g" illustrated in the diagram at the top. Here, the static pressure at the inlet is 0.0,

and the static pressure at the outlet is 1.0. The negative value means that the static pressure is lower than that at the inlet. In each of FIGS. 5A-5D, the lower graph shows the flow velocity of the refrigerant at positions "a" to "g" illustrated in the diagram at the top. Here, the flow velocity at the inlet is 0.0. The negative value means that the flow velocity is lower than that at the inlet. In each of FIGS. 5A-5D, the table at the bottom shows the flow rate of the refrigerant at positions (A) to (E) illustrated in the diagram at the top.

[0043] Referring to FIG. 5A, when the compressor is operated at 100% load without a casing treatment bypass, the static pressure of the refrigerant decreases, and then increases as the flow approaches the outlet. Referring to FIG. 5B, when the compressor is operated at 100% load with a casing treatment bypass, the static pressure of the refrigerant decreases, and then increases as the flow approaches the outlet, similarly to the case shown in FIG. 5A. In the case shown in FIG. 5B, the exit port of the casing treatment bypass is arranged at the edge of the impeller blade, and thus, the ability of increasing the pressure does not significantly change even with the casing treatment bypass. In this manner, when the compressor is operated at 100% load, the static pressure of the refrigerant changes similarly regardless of whether a casing treatment bypass is provided or not.

[0044] As mentioned above, the stall of the flow at the inlet will occur when the flow rate of the refrigerant is reduced. In other words, the stall of the flow at the inlet will not occur when the flow rate of the refrigerant is sufficiently large as in the cases shown in FIGS. 5A and 5B. The casing treatment bypass 60 in accordance with the present invention is provided to prevent the stall of the flow at the inlet when the flow rate of the refrigerant is reduced as in the cases shown in FIGS. 5C and 5D.

[0045] Referring to FIG. 5C, when the compressor is operated at 50% load with a casing treatment bypass, the static pressure of the refrigerant at position "c" is larger than the static pressure of the refrigerant at the inlet. The pressure difference between the static pressure at position "c" and the static pressure at the inlet at 50% load refers to  $\Delta P_{50}$ . Here, 50% is an estimated value. In this instance, it is assumed that surge partially occurs at the inlet because the flow rate of the refrigerant is small. As a result of the pressure difference  $\Delta P_{50}$ , the flow toward the inlet (toward position "f" in the diagram) is created in the casing treatment bypass. Referring to FIG. 5D, when the compressor is operated at 20% load with a casing treatment bypass, the static pressure of the refrigerant at position "c" is much larger than the static pressure of the refrigerant at the inlet. The pressure difference between the static pressure at position "c" and the static pressure at the inlet at 20% load refers to  $\Delta P_{20}$ . Here, 20% is an estimated value. As seen from FIGS. 5C and 5D, the pressure difference  $\Delta P_{20}$  is larger than the pressure difference  $\Delta P_{50}$ .

[0046] When the compressor is operated at low load such as 50% or 20% with the casing treatment bypass, the flow rate of the refrigerant increases at the inlet due to the flow introduced by the casing treatment bypass, which prevents the stall at the inlet from being caused by the flow separation at the front edge of the impeller blade. In this manner, the operation range of the compressor 22 is expanded when the flow rate of the refrigerant is small.

[0047] Referring to FIGS. 6, FIGS. 7A and 7B, the casing treatment bypass 60 in accordance with the present invention will be explained in more detail.

[0048] FIG. 6 is an enlarged schematic diagram inside circle 6 in FIG. 2, illustrating the inlet guide vane 32, the impeller 34 and the casing 30 of the centrifugal compressor 22 of FIGS. 1-3 with the casing treatment bypass 60.

[0049] In the illustrated embodiment, the casing treatment bypass 60 is a hole formed in the inlet portion 31 of the casing 30 of the compressor 22. The casing treatment bypass 60 may include a plurality of holes, with each hole as illustrated in FIG. 6 being circumferentially disposed of the inlet portion 31 of the casing 30. For example, the number of the holes may be eight, and the diameter "a" of each hole may be 46.1 mm.

[0050] The casing treatment bypass 60 includes an entrance port 61 and an exit port 63. The entrance port 61 of the casing treatment bypass 60 is connected to a gap between the impeller 34 and the inlet portion 31 of the casing 30. The exit port 63 of the casing treatment bypass 60 is connected to an area between the impeller 34 and the inlet guide vane 32. As shown in FIG. 6, the exit port 63 of the casing treatment bypass 60 is positioned upstream in a direction of the refrigerant flow with respect to the entrance port 61 of the casing treatment bypass 60. In this manner, the casing treatment bypass 60 injects the refrigerant from the gap between the impeller 34 and the inlet portion 31 of the casing 30 back to the area between the impeller 34 and the inlet guide vane 32. In the illustrated embodiment, the exit port 63 of the casing treatment bypass 60 is located in the area between the impeller 34 and the inlet guide vane 32. In other words, the exit port 63 of the casing treatment bypass 60 is located downstream of the inlet guide vane 32 in the refrigerant flow direction. Alternatively, the exit port 63 of the casing treatment bypass 60 may be located upstream of the inlet guide vane 32 in the refrigerant flow direction.

[0051] As shown in FIG. 6, the impeller 34 includes an impeller hub 35 and impeller blades 37. The impeller blades 37 are disposed to surround the impeller hub 35. The entrance port 61 of the casing treatment bypass 60 faces the impeller blade 37. The exit port 63 of the casing treatment bypass 60 opens to the area between the inlet guide vane 32 and the impeller 34. The diameter "a" of the entrance port 61 of the casing treatment bypass 60 is determined based on the diameter "d" of the impeller blade 37. It is preferable, however, that the diameter "a" of the entrance port 61 of the casing treatment bypass 60 does not exceed 25% of the inlet area of the impeller 34.

[0052] It is also preferable that the cross-sectional area of the exit port 63 of the casing treatment bypass 60 is equal to or greater than the cross-sectional area of the entrance port 61 of the casing treatment bypass 60. For example, the diameter "b" of the exit port 63 of the casing treatment bypass 60 can be arranged to be greater than the diameter "a" of the entrance port 61 of the casing treatment bypass 60 as illustrated in FIG. 6. With this arrangement, the refrigerant can stably flow in the casing treatment bypass 60 from the entrance port 61 toward the exit port 63.

[0053] Referring to FIGS. 7A and 7B, the casing treatment bypass 60 in accordance with another embodiment will be explained.

[0054] The casing treatment bypass 60 illustrated in FIG. 7A has an annular ring shape extending the whole circumference of the inlet portion 31 of the casing 30. In this embodiment, the inlet portion 31 of the casing 30 includes a sub-portion 31s which is separated from the inlet portion 31 of the casing 30 by the ring-shaped casing treatment

bypass 60. For example, the sub-portion 31s is connected to the inlet portion 31 with a linkage mechanism 66 which is attached to the outside of the inlet portion 31. The linkage mechanism 66 may include a connecting ring, and a connecting rod which is rotatably attached to the connecting ring. The linkage mechanism 66 is driven by a driving mechanism 67 such as a stepper motor or a hydraulic cylinder. In operation, the connecting ring of the linkage mechanism 66 is driven by the driving mechanism 67 to move in the axial direction corresponding to the rotation axis of the impeller 34, and rotate the connecting rod of the linkage mechanism 66. The connecting rod of the linkage mechanism 66 then moves the sub-portion 31s in the axial direction corresponding to the rotation axis of the impeller 34 as shown with the dotted line in FIG. 7A. In this manner, the sub-portion 31s is arranged to be movable relative to the non-movable inlet portion 31 in the axial direction corresponding to the rotation axis of the impeller 34. In this embodiment, the flow path area of the casing treatment bypass 60 can be adjusted by moving the sub-portion 31s in the axial direction corresponding to the rotation axis of the impeller 34. Alternatively, the flow path area of the casing treatment bypass 60 may be fixed. For example, the width in the radial direction of the ring-shaped casing treatment bypass 60 may be 15.75 mm in a case where the diameter at the inlet of the impeller 34 is 270 mm.

[0055] The casing treatment bypass 60 illustrated in FIG. 7B includes an adjusting member 64. The adjusting member 64 is a movable ring disposed in the inlet portion 31 of the casing 30 so as to at least partly block the flow path area of the casing treatment bypass 60. For example, the adjusting member 64 is connected to the inlet portion 31 with a linkage mechanism 68 which is attached to the outside of the inlet portion 31. The linkage mechanism 68 may include a connecting ring which is rotatably attached to the adjusting member 64. The linkage mechanism 68 is driven by a driving mechanism 69 such as a stepper motor or a hydraulic cylinder. In operation, the connecting ring of the linkage mechanism 68 is driven by the driving mechanism 69 to move the adjusting member 64 in the radial direction perpendicular to the axial direction corresponding to the rotation axis of the impeller 34 as shown with the dotted line in FIG. 7B. In this manner, the adjusting member 64 is arranged to be movable relative to the non-movable inlet portion 31 in the radial direction perpendicular to the axial direction, and the flow path area of the casing treatment bypass 60 can be adjusted by moving the adjusting member 64 in the radial direction perpendicular to the axial direction. The adjusting member 64 can be applied to both of the case in which the casing treatment bypass 60 has a single hole and the case in which the casing treatment bypass 60 has a plurality of holes.

[0056] As mentioned above, the flow path area of the casing treatment bypass 60 can be optimized by moving the sub-portion 31s in the axial direction corresponding to the rotation axis of the impeller 34 (see FIG. 7A) or by moving the adjusting member 64 in the radial direction perpendicular to the axial direction (see FIG. 7B). The casing treatment bypass 60 in accordance with the present invention can use either one of the movable sub-portion 31s and the movable adjusting member 64 to optimize the flow path area of the casing treatment bypass 60.

[0057] Referring to FIGS. 8A and 8B, various types of impellers can be used as the impeller 34 of the compressor

22 in accordance with the present invention. FIG. 8A is a schematic diagram illustrating a mixed flow impeller and FIG. 8B is a schematic diagram illustrating a radial flow impeller. However, the present invention is not limited to the type of the impeller 34. The case treatment bypass 60 in accordance with the present invention can be applied to various types of impellers including the mixed flow impeller as shown in FIG. 8A or the radial flow impeller as shown in FIG. 8B.

[0058] In terms of global environment protection, use of new low GWP (Global Warming Potential) refrigerants such like R1233zd, R1234ze are considered for chiller systems. One example of the low global warming potential refrigerant is low pressure refrigerant in which the evaporation pressure is equal to or less than the atmospheric pressure. For example, low pressure refrigerant R1233zd is a candidate for centrifugal chiller applications because it is non-flammable, non-toxic, low cost, and has a high COP compared to other candidates such like R1234ze, which are current major refrigerant R134a alternatives. Especially in a case of using low pressure refrigerant, the compressor 22 including the casing treatment bypass 60 in accordance with the present invention has advantages because the operation range of the compressor 22 can be expanded to prevent surge without requiring a large-diameter pipe for a conventional hot gas bypass.

#### General Interpretation of Terms

[0059] In understanding the scope of the present invention, the term “comprising” and its derivatives, as used herein, are intended to be open ended terms that specify the presence of the stated features, elements, components, groups, integers, and/or steps, but do not exclude the presence of other unstated features, elements, components, groups, integers and/or steps. The foregoing also applies to words having similar meanings such as the terms, “including”, “having” and their derivatives. Also, the terms “part”, “section”, “portion”, “member” or “element” when used in the singular can have the dual meaning of a single part or a plurality of parts.

[0060] The term “detect” as used herein to describe an operation or function carried out by a component, a section, a device or the like includes a component, a section, a device or the like that does not require physical detection, but rather includes determining, measuring, modeling, predicting or computing or the like to carry out the operation or function.

[0061] The term “configured” as used herein to describe a component, section or part of a device includes hardware and/or software that is constructed and/or programmed to carry out the desired function.

[0062] The terms of degree such as “substantially”, “about” and “approximately” as used herein mean a reasonable amount of deviation of the modified term such that the end result is not significantly changed.

[0063] While only selected embodiments have been chosen to illustrate the present invention, it will be apparent to those skilled in the art from this disclosure that various changes and modifications can be made herein without departing from the scope of the invention as defined in the appended claims. For example, the size, shape, location or orientation of the various components can be changed as needed and/or desired. Components that are shown directly connected or contacting each other can have intermediate structures disposed between them. The functions of one

element can be performed by two, and vice versa. The structures and functions of one embodiment can be adopted in another embodiment. It is not necessary for all advantages to be present in a particular embodiment at the same time. Every feature which is unique from the prior art, alone or in combination with other features, also should be considered a separate description of further inventions by the applicant, including the structural and/or functional concepts embodied by such feature(s). Thus, the foregoing descriptions of the embodiments according to the present invention are provided for illustration only, and not for the purpose of limiting the invention as defined by the appended claims and their equivalents.

What is claimed is:

1. A centrifugal compressor adapted to be used in a chiller system, the centrifugal compressor comprising:

- a casing having an inlet portion and an outlet portion;
- an inlet guide vane disposed in the inlet portion;
- an impeller disposed downstream of the inlet guide vane, the impeller being attached to a shaft rotatable about a rotation axis;
- a motor arranged and configured to rotate the shaft in order to rotate the impeller;
- a diffuser disposed in the outlet portion downstream of the impeller; and
- a casing treatment bypass having an entrance port and an exit port,

the casing treatment bypass being arranged and configured to inject refrigerant from a gap between the impeller and the inlet portion of the casing toward an area between the impeller and the inlet guide vane, and the exit port of the casing treatment bypass being positioned upstream in a direction of a refrigerant flow with respect to the entrance port of the casing treatment bypass.

2. The centrifugal compressor according to claim 1, wherein

- the impeller includes an impeller hub and an impeller blade surrounding the impeller hub,
- the entrance port of the casing treatment bypass faces the impeller blade, and
- the exit port of the casing treatment bypass opens to the area between the inlet guide vane and the impeller.

3. The centrifugal compressor according to claim 2, wherein

- a diameter of the entrance port of the casing treatment bypass is determined based on a diameter of the impeller blade.

4. The centrifugal compressor according to claim 1, wherein

- a cross-sectional area of the exit port of the casing treatment bypass is equal to or greater than a cross-sectional area of the entrance port of the casing treatment bypass.

5. The centrifugal compressor according to claim 1, wherein

the casing treatment bypass includes a hole formed in the inlet portion of the casing.

6. The centrifugal compressor according to claim 1, wherein

the casing treatment bypass includes a plurality of holes formed in the inlet portion of the casing.

7. The centrifugal compressor according to claim 1, wherein

the casing treatment bypass has a ring shape.

8. The centrifugal compressor according to claim 1, wherein

a flow path area of the casing treatment bypass is fixed.

9. The centrifugal compressor according to claim 1, wherein

a flow path area of the casing treatment bypass is adjustable.

10. The centrifugal compressor according to claim 9, wherein

a movable adjusting member is disposed in the inlet portion of the casing so as to at least partly block the flow path area of the casing treatment bypass, and the flow path area of the casing treatment bypass is adjusted by moving the movable adjusting member in a radial direction perpendicular to an axial direction corresponding to the rotation axis of the impeller.

11. The centrifugal compressor according to claim 7, wherein

an flow path area of the casing treatment bypass is adjustable.

12. The centrifugal compressor according to claim 11, wherein

the inlet portion of the casing includes a movable sub-portion which is separated from the inlet portion of the casing by the casing treatment bypass, and the flow path area of the casing treatment bypass is adjusted by moving the sub-portion in an axial direction corresponding to the rotation axis of the impeller.

13. The centrifugal compressor according to claim 1, wherein

the refrigerant is low global warming potential refrigerant.

14. The centrifugal compressor according to claim 13, wherein

the low global warming potential refrigerant is low pressure refrigerant.

15. The centrifugal compressor according to claim 14, wherein

the low pressure refrigerant includes R1233zd.

16. The centrifugal compressor according to claim 1, wherein

the impeller is a mixed flow type impeller.

17. The centrifugal compressor according to claim 1, wherein

the impeller is a radial flow type impeller.

\* \* \* \* \*