Development of High-Efficiency Centrifugal Compressor for Turbo Chiller

TAMAKI Hideaki : Doctor of Engineering, P. E. Jp, Manager, Turbo Machinery and Engine Technology Department, Products Development Center, Corporate Research & Development

KAWAKUBO Tomoki : Manager, Turbo Machinery and Engine Technology Department, Products Development Center, Corporate Research & Development

TSUKAMOTO Minoru : Manager, Refrigerant Compressor Project Department, Compressor & Machinery Operations, IHI Compressor and Machinery Co., Ltd.

NUMAKURA Ryusuke : Doctor of Engineering, Turbo Machinery and Engine Technology Department, Product Development Center, Corporate Research & Development

A high efficiency 2-stage compressor was developed for large capacity air conditioning equipment. The goal of COP (Coefficient of Performance) of the turbo chiller was more than 6.0, COP of 6.4 for the chiller size of more than 1 000 USRt, which is the highest COP in turbo chillers. In order to satisfy COP of more than 6.0, it was required to develop the new centrifugal compressors the efficiency of which was much higher than conventional centrifugal compressors' efficiency. CFD was successfully applied to the new compressor designing. The turbo chiller with the new 2-stage compressor achieved COP of 6.3 for the chiller size of up to 700 USRt. It is expected that the turbo chiller with the new 2-stage compressor will satisfy the target performance. The aerodynamic design of the 2-stage compressor is explained with a brief introduction of centrifugal compressors.

1. Introduction

Turbo chillers are used for a wide range of applications including large capacity air conditioners for large buildings and other facilities, local cooling and heating, process cooling in chemical and other plants, and cooling for large-scale facilities. There is a demand for energy-saving air conditioners accompanying the increasing awareness of environmental problems in recent years. Increasing the efficiency of high power consumption turbo chillers saves energy and thereby has a great effect on reducing environmental load and reducing running cost.

We have developed a high efficiency 2-stage centrifugal compressor for the world's highest efficiency turbo chillers developed by Daikin Industries, Ltd. This paper outlines the aerodynamic design of the centrifugal compressor.

2. Chiller heat cycle

Figure 1 shows a general chiller heat cycle. The transition from 1 to 2 indicates that vapor (saturated) refrigerant generated by the evaporator is compressed by the compressor and its temperature rises. In the condenser (transition from 2 to 4), vapor (superheated) is cooled and becomes saturated vapor (transition from 2 to 3), and is further cooled until liquefied (transition from 3 to 4). This cooling process uses water (cooling water) at a temperature of 30 to 40°C. The transition from 4 to 5 indicates that the liquefied refrigerant is expanded by an
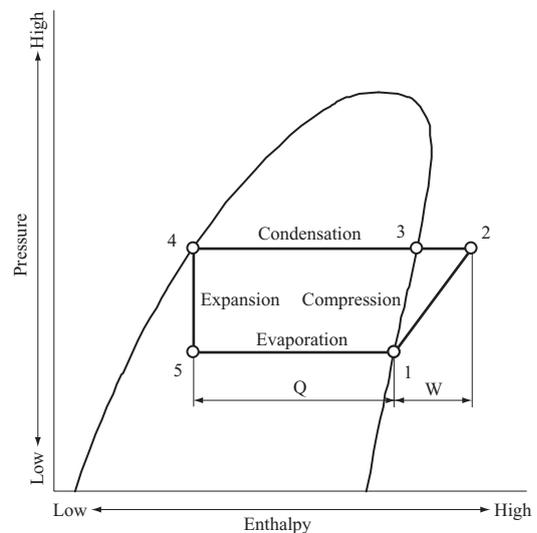
Fig. 1 Chiller heat cycle

expansion valve. Part of the refrigerant liquid evaporates and enters the evaporator with the remaining refrigerant liquid. This refrigerant liquid absorbs external heat (or is heated externally) in the evaporator (transition from 5 to 1) to become (saturated) vapor. The evaporator is heated by water (chilled water) of about 10°C. Turbo chillers use turbo compressors in the compression process mentioned above. The refrigerating capacity of a chiller is evaluated at the chilled water temperature of 12°C at the inlet and 7°C

at the outlet of the evaporator and the cooling water temperature of 32°C at the inlet and 37°C at the outlet of the condenser, and it is expressed in units of refrigeration tons (RT). One (1) RT corresponds to approx. 3.5 kW. The performance of a chiller is evaluated in terms of coefficient of performance or COP, and is expressed as the rate of Q/W, where Q is the quantity of heat taken from the chilled water in the evaporator and W is the work added externally (the work required to drive the compressor).

Figure 2 shows a schematic diagram of a refrigerating cycle adopted for the target chiller in our development. This cycle is called a 2-stage economizer cycle, which consists of 2-stage centrifugal compressor and an economizer (intermediate cooler) downstream of the 1st-stage compressor. The economizer pressure is kept equal to the pressure at the outlet of the 1st-stage compressor, $P_{1'}$, so that refrigerant liquid coming from the condenser partly evaporates and the remaining refrigerant liquid is cooled to the saturation temperature that corresponds to $P_{1'}$ in order to increase the refrigerating effect.

The 2nd-stage compressor draws the refrigerant gas at temperature $T_{1'}$ coming from the 1st-stage compressor and the refrigerant gas evaporating in the economizer. As the refrigerant gas coming from the economizer has a lower temperature than $T_{1'}$, the temperature $T_{1''}$ of the refrigerant gas drawn by the 2nd-stage compressor becomes lower than $T_{1'}$. The lower the temperature of the drawn gas, the less energy the compressor requires to generate a certain pressure.

The economizer cycle can achieve a higher COP than that in a 1-stage refrigerating cycle, due to its improved refrigerating capacity and reduced compression work. This system uses R134a refrigerant which does not destroy the ozone layer.

3. Turbo compressor

This chiller uses a turbo compressor. Fluid machines that continuously transfer energy from fluid to machine or from machine to fluid through rotating impellers (or rotors) are called turbo machines. A turbo compressor is a machine that gives energy to a gas by using internally mounted impellers to feed the gas from a low-pressure side to a high-pressure side.

Turbo compressors are generally divided into the centrifugal type and the axial-flow type, according to the direction of gas flow inside the machine. In a centrifugal compressor, fluid discharged from an impeller flows out radially at a right angle to the axis of rotation. In an axial-flow compressor, gas flows in the direction of the impeller (rotor) rotation axis and flows out in the axial direction. A centrifugal compressor is usually used for a turbo chiller.

Generally, the performance of a centrifugal compressor is expressed as a relationship between the flow rate (or flow coefficient) and the pressure ratio (or pressure coefficient) on a line with a constant number of revolutions, as shown in **Fig. 3**. A characteristic curve (or performance curve) of a centrifugal compressor is obtained by driving an impeller of the compressor with a motor or by some other means and opening/closing a valve mounted downstream of the compressor at a constant number of revolutions, as shown in **Fig. 3**. When the valve mounted on a pipe downstream of the compressor is narrowed to increase the resistance at the valve, the pressure of the compressor increases to achieve balance with the resistance. On the other hand, the increase in the resistance reduces the flow rate,
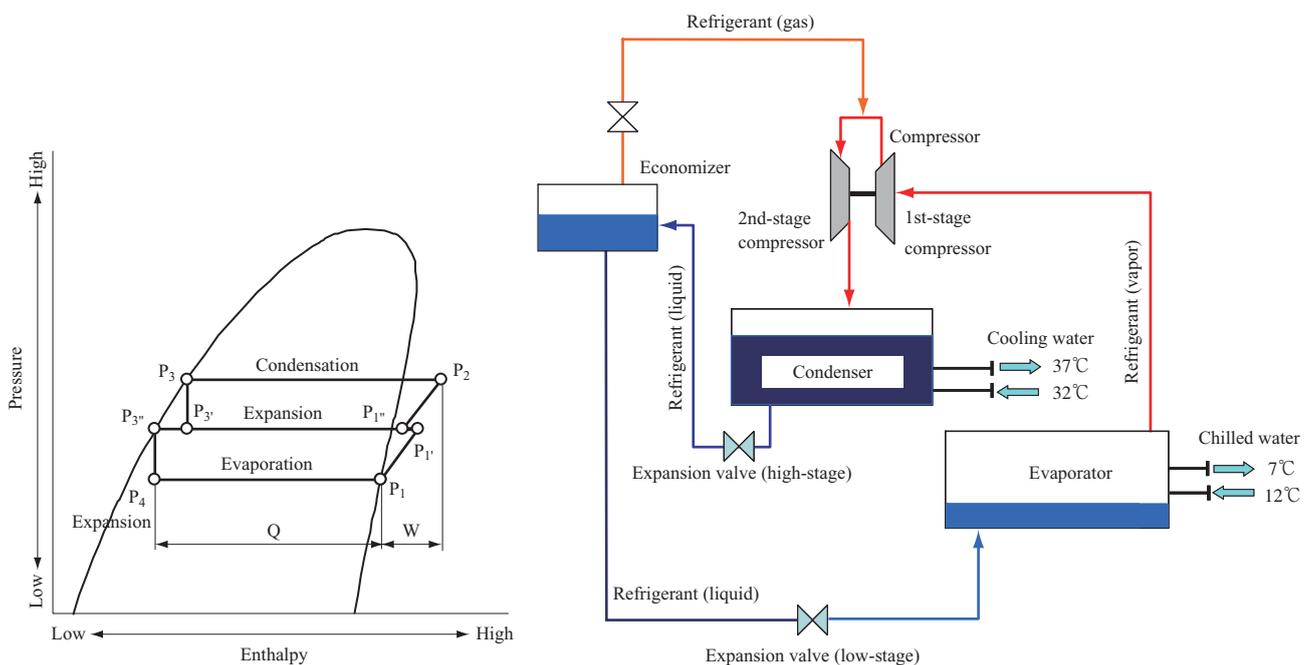
Fig. 2 Economizer cycle
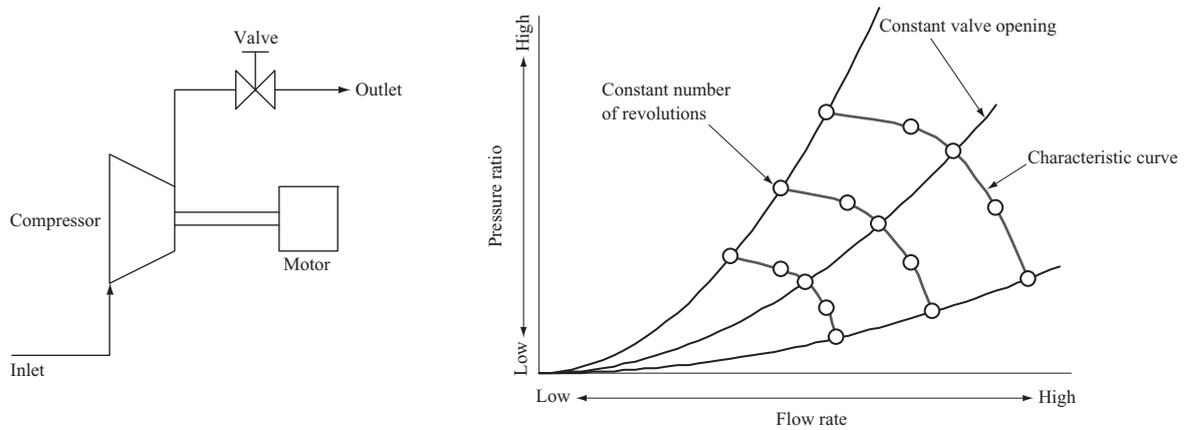
Fig. 3 Schematic of turbo-compressor test facility and compressor characteristics

which gives the characteristic curve as shown in **Fig. 3**.

A centrifugal compressor can operate within a limited range of flow rate, which means that there are maximum and minimum operational flow rate. The range of operational flow rate is called an operating range or a stable operating range. A compressor cannot operate at any flow rate exceeding the flow rate where the relative velocity in an impeller reaches sound velocity, or the (absolute) velocity of flow in a channel downstream of the impeller reaches sound velocity (these are called choke). When the valve downstream of the compressor is gradually narrowed to decrease the flow rate, a sudden vibration in pressure and flow occurs accompanied by noise at a certain flow rate. This is called a surge (or surging). When surge occurs, compressor components could be damaged by excitation forces occurring in surge cycles. The flow immediately

before this surge occurs is the minimum flow at which the compressor can operate. A compressor impeller is shown in **Fig. 4**. Generally, the larger the impeller outlet blade angle, the lower the flow rate at which surge occurs. Surge is mainly caused by separation at a low-flow rate.

The centrifugal compressor for chillers that we have developed adopts a 2-stage structure in which two impellers are mounted to a single rotating shaft. In conventional 2-stage centrifugal compressors, the impeller inlets face in the same direction as shown in **Fig. 5-(a)**. On the other hand, our compressor adopts a structure with impellers mounted back to back as shown in **Fig. 5-(b)**.

Impellers in an operating centrifugal compressor are subject to the static pressure distribution on the hub side and the back side as shown in **Fig. 6**. The difference in the axial static pressure distribution becomes an axial thrust
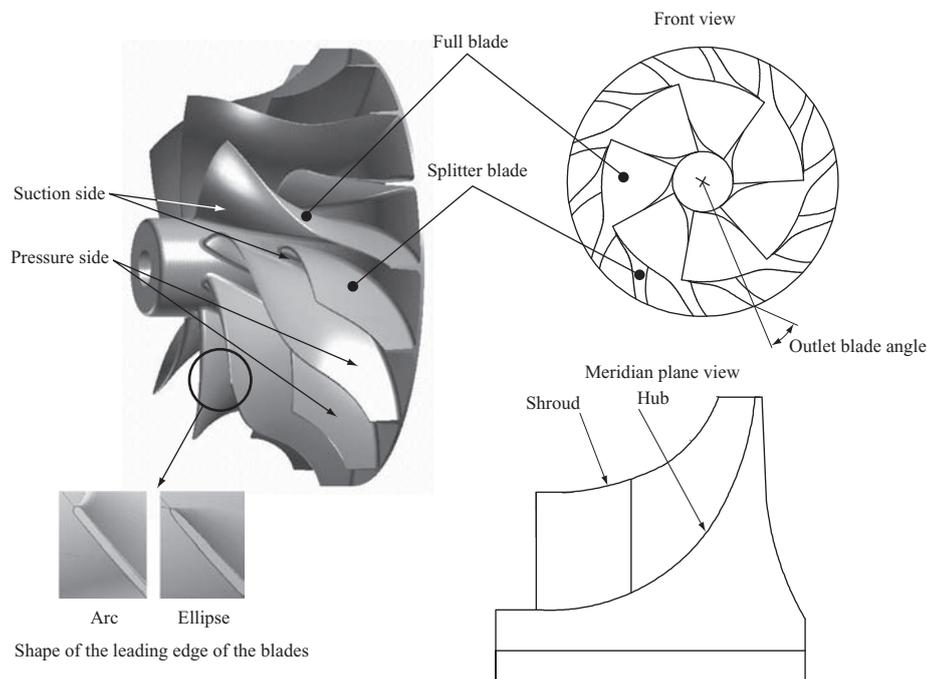
Fig. 4 Centrifugal compressor impeller

(a) Conventional 2-stage centrifugal compressor

(b) Centrifugal compressor for chiller
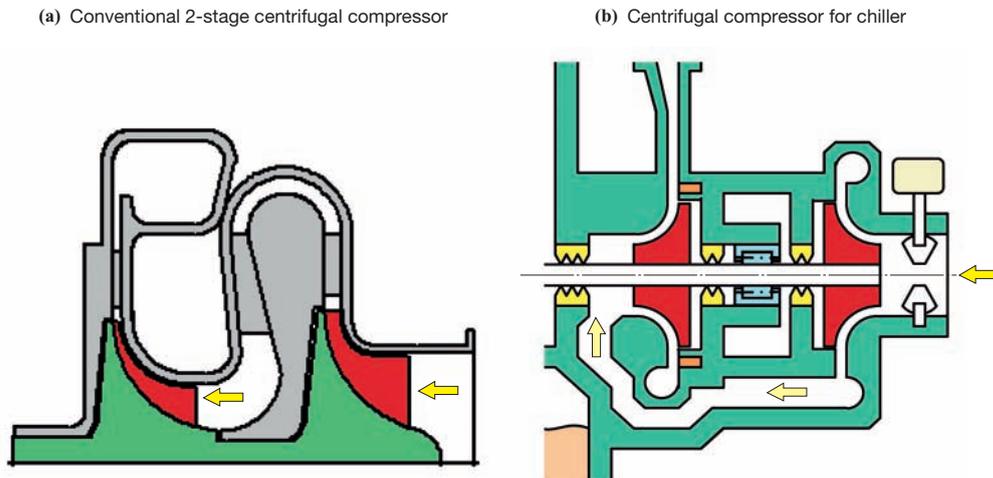
Fig. 5 Impeller arrangement of 2-stage centrifugal compressor

Static pressure

Static pressure
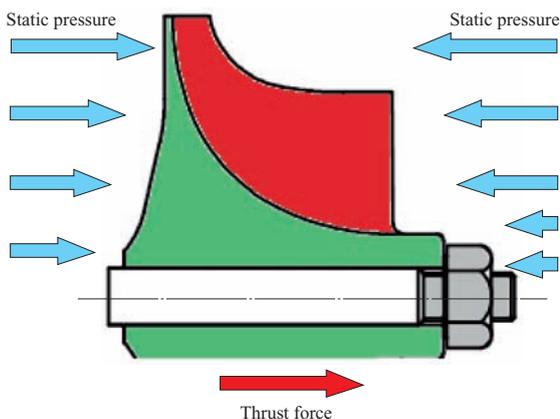
Thrust force

Fig. 6 Pressure force on compressor axial load

that acts on the impeller inlet side. This axial thrust is supported by a thrust bearing. When impellers are arranged into a structure as shown in **Fig. 5-(b)**, the axial thrusts acting on the 1st- and 2nd-stage impellers cancel each other. This has advantages of reducing the load acting on the thrust bearing, increasing the durability of the bearing and reducing mechanical loss.

(1) Development of impellers

To achieve the high efficiency of a COP of 6.0 or higher, we took up the challenge of developing a new compressor superior in efficiency to conventional compressors.

The 1st-stage compressor requires a wide stable operating range to easily match the 2nd-stage compressor. Our examinations of the thermal cycle indicated that the 2nd-stage compressor was subject to a high loading (high pressure coefficient).

An impeller increases pressure by redirecting the flow into the direction of rotation using its blades (see **Fig. 4**). When the flow direction is changed by blades with a limited thickness, the flow velocity on the suction side becomes higher (the pressure becomes lower) and the flow velocity on the pressure side becomes lower (the pressure becomes higher). This difference in velocity between the suction side and the pressure side (the difference in pressure between the pressure side and the suction side) becomes larger as the blades become thicker. To achieve a larger pressure rise, the difference in velocity between the suction side and the pressure side (the difference in pressure between the pressure side and the suction side) becomes larger. At the outlet of an impeller, the pressure side and the suction side have the same velocity (pressure). This leads to rapid deceleration (adverse pressure gradient) on the suction side, resulting in separation and loss. One of the keys to ensuring a stable operating range and higher efficiency is reducing the blade loading (the difference in pressure between the pressure side and the suction side) applied to each blade of the impellers for both the 1st- and 2nd-stage compressors. Therefore, we reduced the thickness and increased the number of blades for both the 1st- and 2nd-stage impellers. We also increased the length of splitter blades on the shroud side by inclining them upstream.

To increase the stable operating range of the 1st-stage impeller, we increased the impeller outlet blade angle (backward angle).

To improve the flow pattern inside impellers, we changed the shape of the leading edge of the blade from arc to ellipse for both the 1st- and 2nd-stage impellers to suppress sudden acceleration/ deceleration at the leading edge of the blade (see **Fig. 4**). We also adopted a splitter blade shape that is different from the full blade shape to smooth out the flow between splitter blades and full blades.

We optimized the blade shape by calculating flows inside impellers through numerical simulation using computational fluid dynamics (CFD) codes for both the 1st- and 2nd-stage impellers.

Figure 7 shows the shapes of the conventional and new designs of 1st- and 2nd-stage impellers. **Figure 8** shows the relative Mach number on the shroud side (85% span position) of the 2nd-stage impellers of the conventional type and the new type, as an example of CFD calculation. Slight separation occurs at the leading edge of full and splitter blades of the conventional type, while no separation occurs in these regions of the new type. **Figure 9** shows a comparison of the performance of impellers of the new type and the conventional type estimated by using CFD. The efficiency of the new type of impeller is higher than that of the conventional type.

(2) Compressor performance

The kinetic energy of flow going out of an impeller corresponds to 30 to 40% of the total input work of the impeller. To design a high efficiency centrifugal compressor, it is necessary to efficiently recover this kinetic energy to a static pressure through an enlarging passage (diffuser) provided downstream of the impeller.

Diffusers used for centrifugal compressors are roughly divided into two types; vaneless diffusers and vaned diffusers. Vaneless diffusers are widely used in applications where a wide operating range is required. Vaned diffusers are used in applications where a high pressure ratio or high efficiency is required. Compressors for chillers need a wide flow rate range depending on refrigerating capacity, so we adopted a vaneless diffuser with a wide operating range. **Figure 10** shows a comparison between the

(a) Conventional impellers   **(b)** New impellers

1st-stage   2nd-stage   1st-stage   2nd-stage
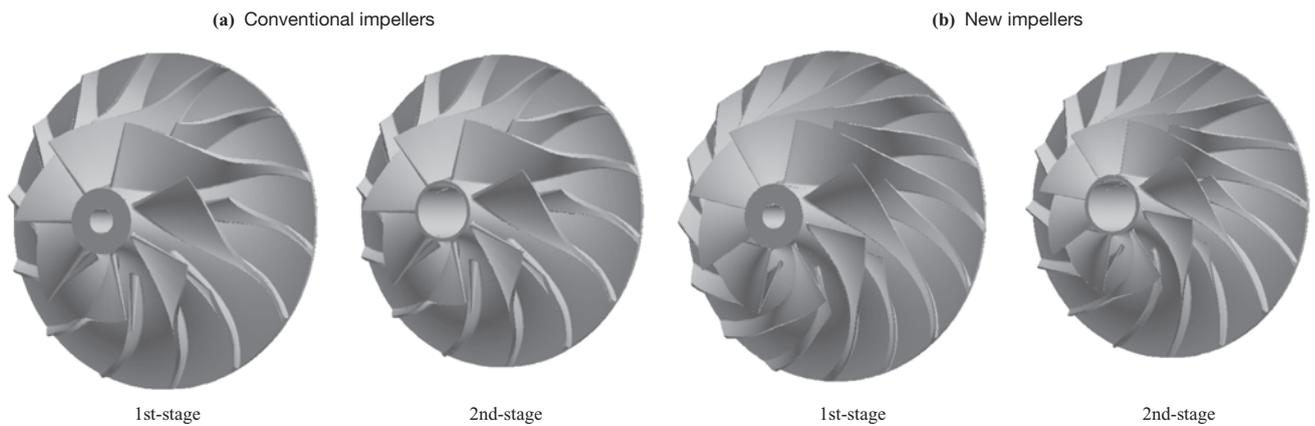
Fig. 7   Conventional and latest impeller shapes

(a) Conventional type   **(b)** New type

◯ : Area with improved flow
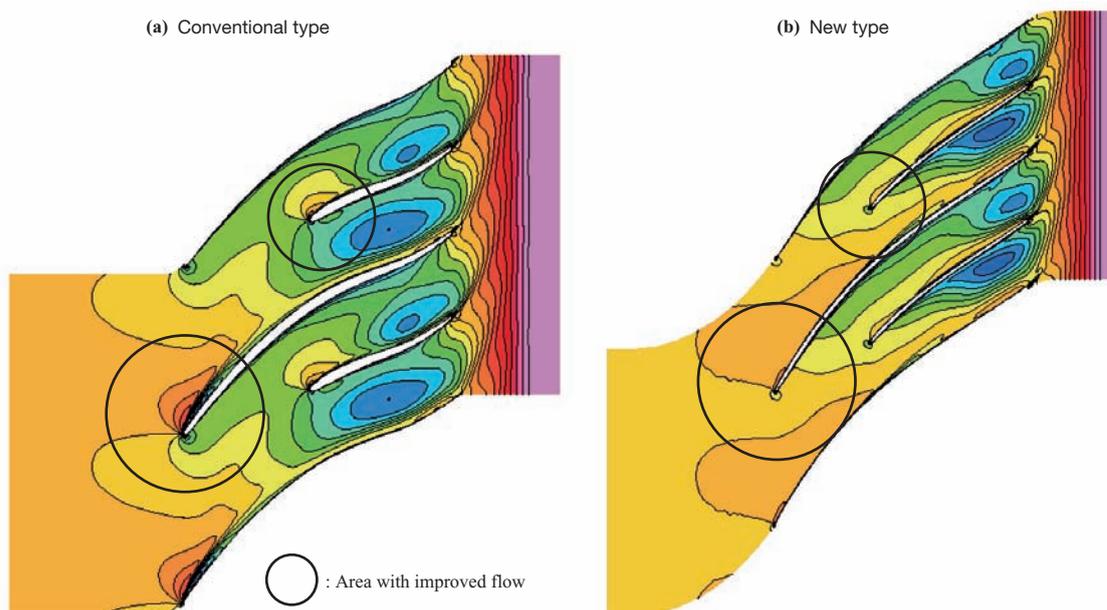
Fig. 8   2nd-stage compressor impeller relative Mach number distributions at shroud section

(a) 1st-stage impellers

(b) 2nd-stage impellers
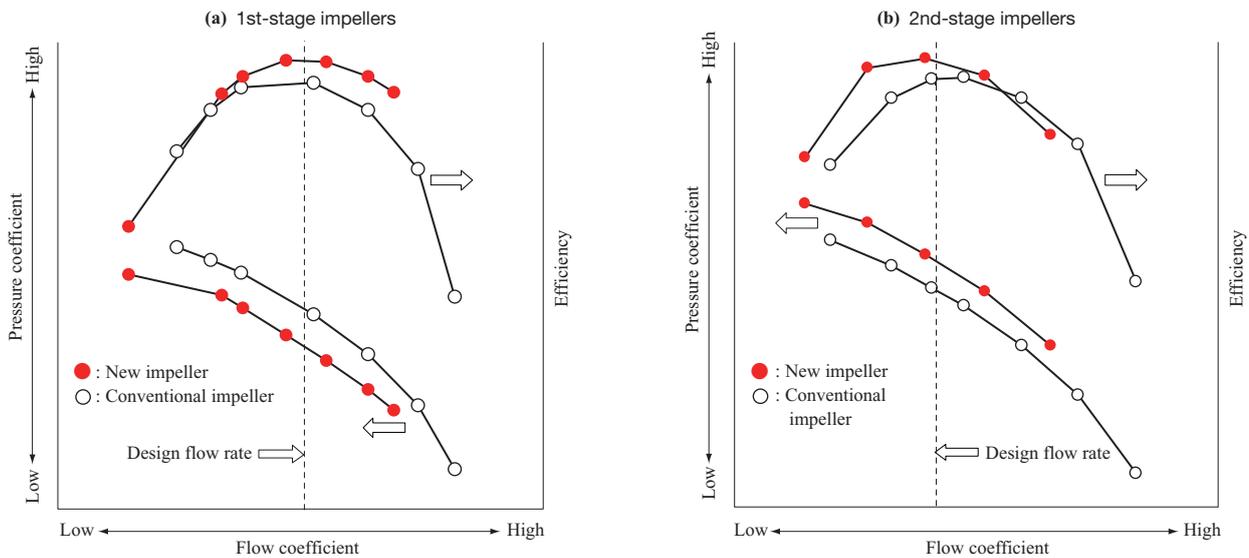
Fig. 9   Impeller performance
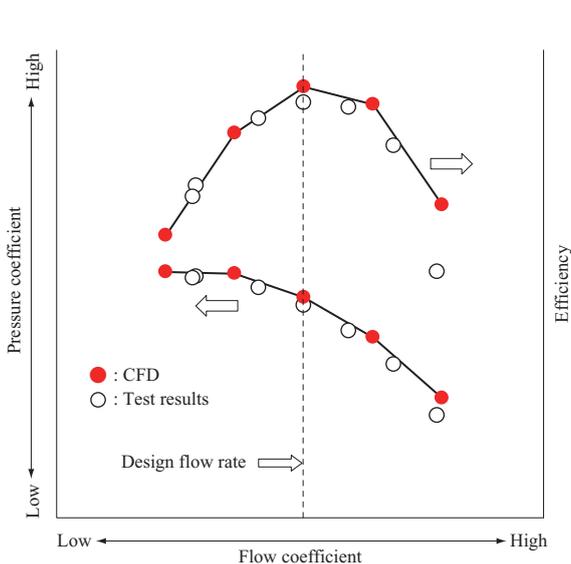
Fig. 10   1st-stage compressor performance (comparison between CFD and test results)
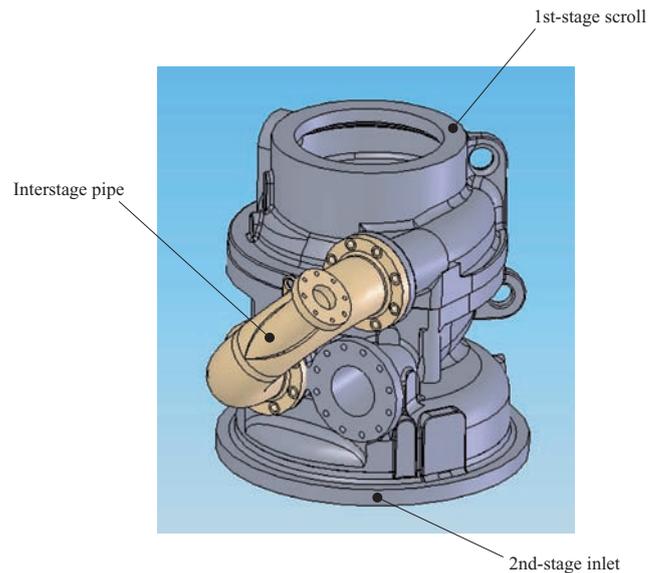
Fig. 11   Connecting conditions of the 1st- and 2nd-stage compressors

CFD and the test results for the 1st-stage compressor. The improved performance was achieved as designed.

(3)   Design of the non-rotating channel

Figure 11 shows the connection between the 1st- and 2nd-stage compressors. The outlet of the 1st-stage (scroll) and the inlet of the 2nd-stage (inlet) are connected with a pipe (interstage pipe). This interstage pipe is connected to a pipe from an economizer. The flow entering the 2nd-stage inlet is turned to the axial direction, with circumferential velocity components removed by a fixed inlet guide vane, and then enters the 2nd-stage impeller. In order to successfully develop a high performance 2-stage centrifugal compressor, it is necessary to select a shape that will not cause any backflow and will generate as little loss as possible

in the flow in the interstage pipe and at the inlet. We calculated flow in each element using CFD in the same way as for impellers. **Figure 12** shows calculation results for the 1st-stage scroll and the interstage pipe. There is no significant backflow or loss region at the joint of the high-temperature refrigerant gas coming from the scroll of the 1st-stage compressor and the low-temperature refrigerant gas coming from the economizer. **Figure 13** shows the calculation results for the 2nd-stage inlet. Separation occurs at the fixed inlet guide vane (IGV). As shown in **Fig. 14**, it is possible to prevent separation from occurring by changing the setting angle of the guide vane where separation occurs.

(a) 1st-stage compressor scroll and interstage pipe

(b) Computational grid

1st-stage scroll

Interstage pipe

(c) 1st-stage scroll (Mach number distribution)

(d) Interstage pipe (streamline)
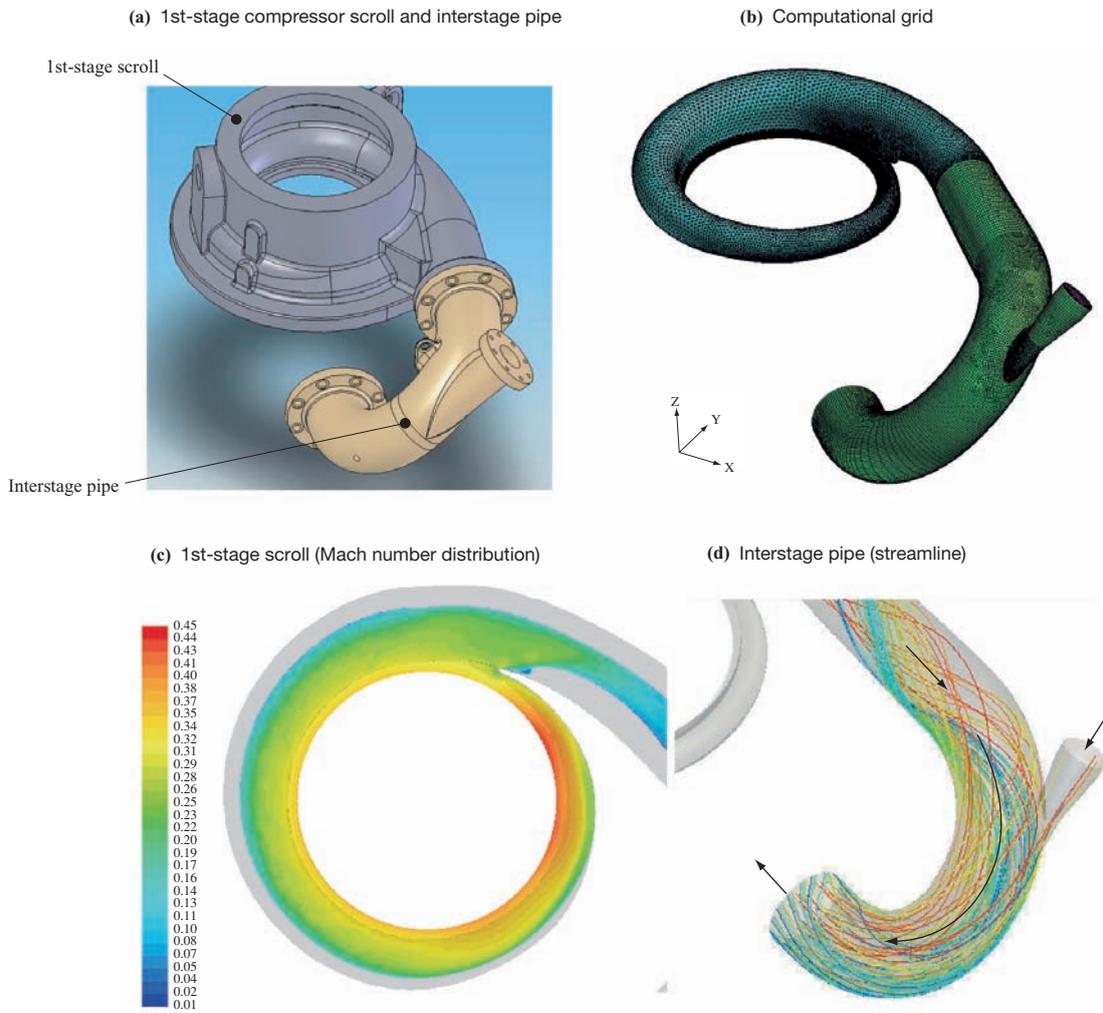
Fig. 12   CFD results of 1st-stage compressor scroll and interstage pipe

(a) 2nd-stage compressor inlet

(b) Total pressure distribution

IGV

Flow direction

IGV

Separation

Separation

Separation
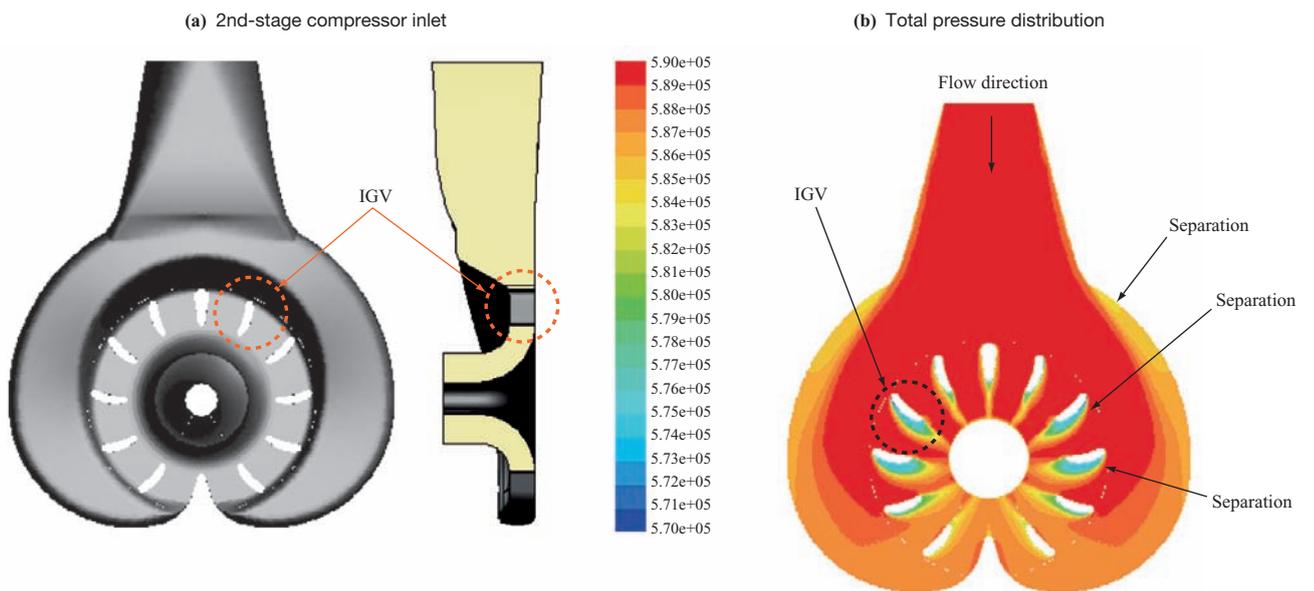
Fig. 13   CFD results of the 2nd-stage inlet

(a) Original IGV **(b)** Improved IGV

Improved IGV restricts separation
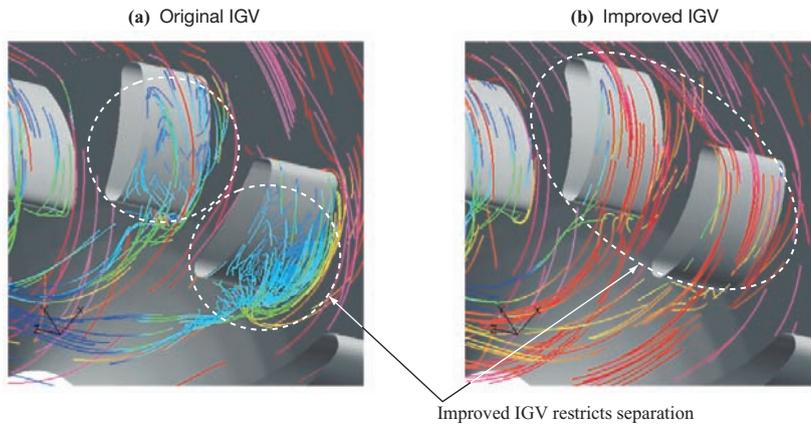
Fig. 14 Streamline around IGV in the 2nd-stage inlet

4. Conclusion

The turbo chiller equipped with the new compressor achieved COP of 6.3 for chiller sizes of up to 700 USRt, contributing to the reduction of the environmental load.

We will continue to develop high performance turbo chiller compressors in consideration of environmental loads.