DEVELOPMENT OF A HIGH-PERFORMANCE TURBO CHILLER

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ABSTRACT

Turbo chillers are suitable for large-scale air conditioning systems, and very important equipment for reducing environmental loads in that substantial reduction in electrical power consumption can be expected by enhancing their performance to higher levels. The MHI's turbo chillers (AART series) with HFC134a as a refrigerant, has achieved performance enhancement on the world's highest level through improvements in the refrigeration cycle, aerodynamic performance of the compressor and heat exchanger performance, as well as optimum design of the control system. Their high performance characteristics contribute toward substantially reducing electrical power consumed at factories by expanding major applications including district cooling and heating, thermal storage, factory process cooling, and especially factory process cooling for full-year continuous operation. In addition, it has been made possible to reduce the year-round annual electrical power consumption further through the application of optimum variable control using an inverter. Furthermore, the centrifugal chiller technology has also been applied to heat pump chillers that were in the field of screw chillers in the past, which has led us to provide super-high performance small turbo heat pumps.

Key Words: ***heat pumps, compressors, variable speed, HFC134a***

1 INTRODUCTION

The refrigerant HFC134a used in this turbo chiller has been used in various kinds of chiller equipment since around 1994 as an alternative refrigerant in place of CFC12, HCFC22, and HCFC123 due to the ozone layer depletion problem. At present, the refrigerant HFC134a, among HFC, has become the most often used refrigerant as a result of its adoption to automotive air conditioners. In 2001, the Law Concerning the Recovery and Destruction of Fluorocarbons was established to restrain the discharge of fluorocarbons, and handling of fluorocarbons has been made clear. As a result, stable supply of HFC refrigerants has been secured.

By the way, the situation surrounding turbo chillers has been changing greatly in recent years. Due to the world-wide recession, the total demand has a tendency to decrease, and air conditioning for buildings tends to shift from central heating to the inexpensive individual air conditioning method. In the case of factory air conditioning, however, the move to replace air conditioning units burning fossil fuels is strongly rooted due to the CO_2 emission problem and subject to performance enhancement and saving of energy. In this case, the condition includes full-year continuous operation, and it has become necessary to enhance the performance not only at the design point, but particularly during the intermediate period and winter as well, and to evaluate the life-cycle cost of the entire equipment over 15 to 20 years. These are all challenges to be addressed.

In addition, the year 2005 is a very important year in considering the global environmental problem. This is because it is a year in which we are supposed to start studying the framework in and after 2013 with just ahead the first promised period of 2008 to 2012 in the Kyoto Protocol committed to reduce greenhouse gas emissions.

Turbo chillers are used for a district cooling and heating application and factory heat source application involving large-scale central air conditioning, namely in the so-called “segment of industry”. In this application area, thermal energy costs are directly connected to operation costs and product costs. So we have pursued product development based on the customers’ demanding needs for performance enhancement that we have received since the oil crisis in 1973. In our company’s products, COP6.4 has been realized in the AART series, and furthermore, in the factory heat source that requires the supply of cooling and heating over the full year including the wintertime, performance enhancement has been realized in an inverter turbo chiller through the compressor variable speed technology enabling substantial energy savings during the intermediate period and wintertime (October through May). Then, in the NART- series, we have upgraded this inverter turbo chiller to a product that can achieve a maximum of COP17.8 and a yearly average of COP12, and in this way, turbo chillers have become products occupying the position as the “leading runner” in the category of chillers in the domestic market in Japan.

For this reason, we aimed at applying turbo chillers to medium-scale air conditioning in order to expand the performance enhancement effect from the “segment of industry” to the “segment of operation.”

Turbo chillers are large-capacity equipment exceeding 600 USRt as the average capacity of equipment shipped in Japan. In medium-scale air conditioning applications, turbo chillers are mainly adopted as heat source equipment for a refrigeration capacity of 200 USRt or less and they are typically screw chillers and absorption refrigeration machines. Although efforts have been made in recent years to enhance the performance of equipment with such a capacity, there is still much room for performance enhancement judging from the turbo chiller technology (Figure. 1).

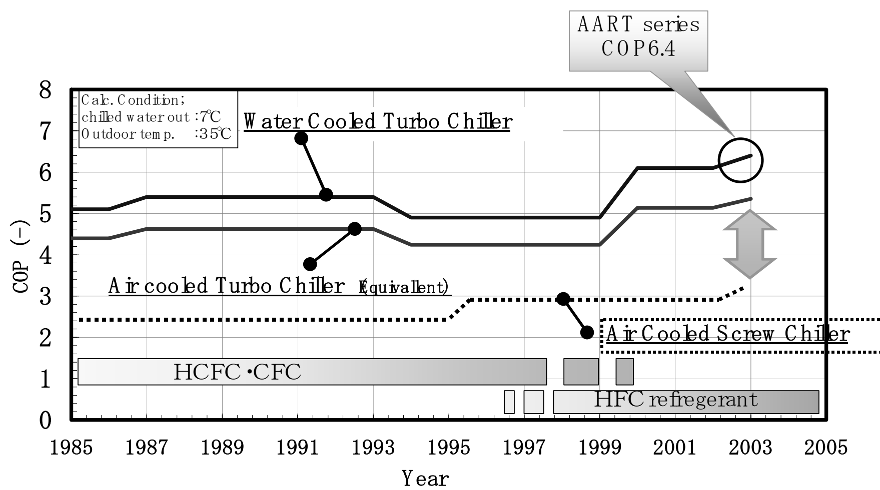


Fig. 1. Comparison between turbo chiller and screw chiller

Therefore, as a product corresponding to this field, we embarked in 2001 on the development of the “Microturbo S series,” a high-performance air-cooled small turbo heat pump to which the performance enhancement technology for turbo compressors is applied and which is turned into a heat pump. Then, in 2004, we started the sales.

2 IMPROVEMENT IN EQUIPMENT PERFORMANCE OF LARGE CAPACITY TURBO CHILLERS

An introduction is made of compressors and evaporators which have greatly contributed toward improving the performance of large equipment having a capacity of 200 USRt or more.

2.1 Improvement in Compressor Performance

In the refrigeration cycle, the cycle approaches an ideal one as the compression and expansion become more multi-stage. However, we have adopted a two-stage compression and two-stage expansion sub-cooling cycle from the viewpoint of the HFC134a properties, optimum design of blades, equipment downsizing, and cost reduction. (Figure 2) In a case where two-stage compression is adopted, the inflow Mach number for the impeller is subsonic. Therefore, we implemented the following aerodynamic design in order to improve the flow inside the impeller and downstream of the outlet and enhance the efficiency [1]. Figure 3 shows the cross-sectional structure of the compressor for the developed equipment.

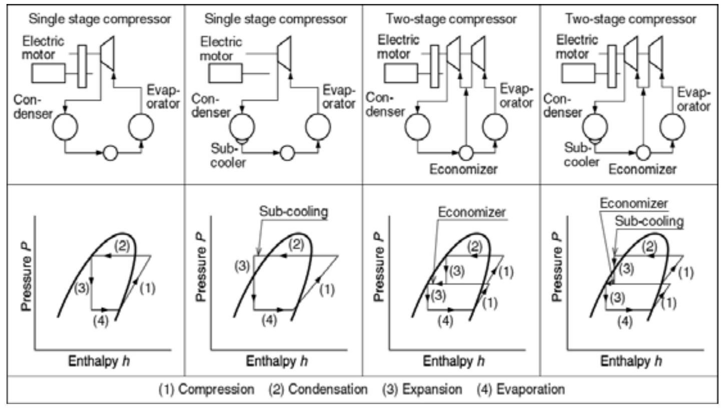


Fig. 2. Comparison of heat cycles (Four typical cycle flow diagrams and cycles on Moller charts are shown)

2.1.1 Design for high efficiency of subsonic open impeller

- The flow coefficient was made larger compared with the conventional two-stage type, and for both the first and second stages, the specific speed was made higher.
- To minimize the secondary flow inside the impeller, the blade angle distribution and blade inclination angle were determined with the aid of the latest CFD. Figure 4 shows an example under study. It shows how the Secondary flow on the negative pressure surface of the blade and boundary layer thickness decrease.
- By improving the flow inside the impeller, we were also able to improve distortion in the flow from the impeller outlet to the downstream diffuser.
- Because we were able to reduce the secondary flow inside the impeller, the number of blades was optimized in order to reduce the wetted area, increase the inlet throat area, and reduce the trailing vortex due to the rear edge thickness of the blade. The optimum number of blades was also determined based on the CFD results. Figure 5 shows actual impeller and shaft assembly.

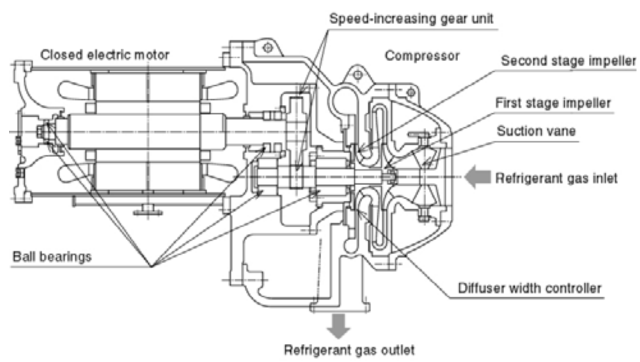


Fig. 3. Cross section of compressor

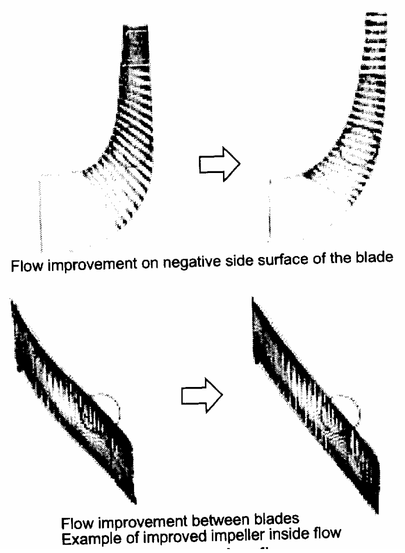


Fig. 4. Examples of improved turbulence flow by CFD analysis for impeller inside flow (Reduction of secondary flow)

2.1.2 Evaluation of shaft system

Compared with the conventional two-stage type, the specific speed for the impeller was made larger. Therefore, the shaft length of the impeller became longer and the overhang portion from the bearing support position became longer. The soundness of these changes was verified by the rotor dynamics analysis. Furthermore, the natural frequency of the blade as installed was measured and avoidance of coupled vibration with the shaft and resonance with the upstream stator blade was confirmed. It was also confirmed in the actual equipment verification testing machine that no abnormal vibration was generated under the operating conditions, including surging, with the shaft and casing vibrations monitored.

2.1.3 Design for reducing mechanical loss

The mechanical loss was reduced to 50% or less compared with the conventional value by changing the bearing from the conventional plain bearing to a rolling bearing. As for the bearing life, 50,000 hours or more were ensured. (2)

Regarding the gear, a power transmission efficiency of 99% or more was achieved by optimizing the pressure angle. (2)

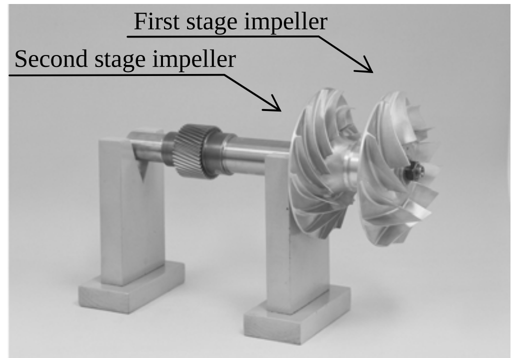


Fig. 5. Impeller and shaft assembly (The first and second impellers are installed on the compressor shaft)

2.2 Enhancement of Evaporator Performance

In order to improve the heat transfer performance of the evaporator, it is essential to improve the overall heat transfer performance by optimizing the heat transfer tube arrangement because the enhancement of the heat transfer tube performance alone has a limit. Especially in the upper portion of the tube bundle, air bubbles produced in the lower heat transfer tube accumulate, causing the void ratio to go up. This restrains nucleate boiling on the heat transfer tube surface, reducing the heat transfer performance. In order to improve this problem, “tube-removed zones” were provided in the tube bundle to remove produced air bubbles by collecting them in these zones. In addition, the pitch in the horizontal direction of the heat transfer tube was increased.

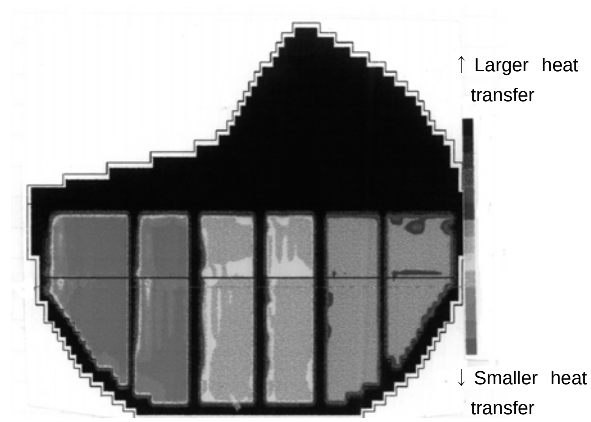


Fig. 6. Example of evaporator fluid analysis (An example analyzing the distribution of shell side boiling heat transfer coefficient in the cross section of an evaporator is shown)

Furthermore, based on the basic data, such as the correlation between the heat transfer coefficient and void ratio, obtained in the factor test, the heat flow analyzing general code was improved. This improvement made it possible to predict the distribution of heat transfer coefficients of an evaporator using HFC134a, and it was applied to the design of actual equipment. Figure 6 shows an example of the analysis of the evaporator. (4)

2.3 Application of Variable Speed Control

By applying the variable speed control to the compressor through the combination with an inverter, it was made possible to save a substantial amount of energy in the full-year operation. Generally, variable speed control is common in small-capacity equipment using a positive-displacement compressor. However, the performance improvement achieved through the application of variable speeds to the compressor for turbo chillers was not sufficient in the previous application.

2.3.1 Compressor performance

Control of chillers (compressors) includes the variable speed control for the impeller's number of revolutions, in addition to the IGV control, diffuser variable mechanism and hot gas bypass valve control performance. Other types of control, excluding the variable speed control, include losses, and therefore, it is made possible to improve the performance by increasing the control area based on the variable speed control.

Aerodynamic performance. In the case of fixed speed compressors, the application range of the number of revolutions is 115% to 85% of the design point. In the case of variable speed compressors, the number of revolutions is 115% to 50% of the design point, which means substantial expansion of the off-design area.

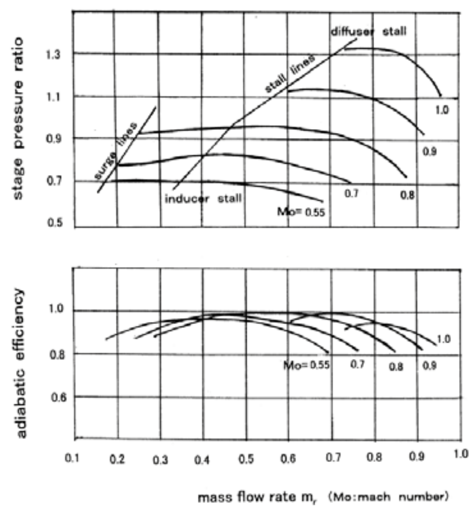


Fig. 7. Compressor map.

In order to utilize variable speeds effectively, it is essential to understand the aerodynamic stability area of a compressor. Because the stability area is an area surrounding by the surge limit, diffuser stall limit and inducers stall limit, it is important to expand each limit area. In this application, we adopted a vane-less diffuser and inducers-less blade designed at subsonic speed and we were able to expand the stability area to the surge limit line and choke limit.

Furthermore, it is known that the surge limit expands more toward the lower air quantity side as the number of revolutions goes from the design point to a lower speed area. In this application, however, the stability area expanded to the minimum air quantity area in the 50% to 70% range, making it possible to expand the high-efficiency operation area substantially. Also, as an aerodynamic basic property, the extremely small reduction in the stage efficiency at the off-design Mach number was also a satisfactory property (Figure 7).

2.3.2 Resonance strength design

Because the operation range is expanded, it is necessary to give full consideration to the shaft resonance design and blade resonance design. It is especially important to avoid resonance for the number of revolutions, the number of impeller rotor blades, and the number of stator blades.

2.3.3 Mechanical loss

It is confirmed that the loss of a rolling bearing is expressed by the load term and speed term [Eq. 1] and the speed term is dominant in the application in the high speed area. This is estimated to be a loss due to a collision between drops of mist-supplied lubricating oil and a rotating ball. In a case where variable speed control is implemented, collision losses decrease, which has made it possible to reduce mechanical losses.

As for the loss of a speed increasing gear, losses due to acceleration and pressurization of lubricating oil make up 80%, and therefore, it has become possible to substantially reduce losses in the low speed area.

2.3.4 Control system for variable drive

In order to carry out optimum variable speed control, it is necessary to always control the aerodynamic operation point of a compressor to the stable operation area. In this application, we were able to calculate, on a steady basis, the compressor air quantity and compressor head with the help of an external sensor by using the central processing unit, and make comparisons with the operation area data. From the data compared, it has been made practicable by calculating optimum values to carry out compressor head variation follow-up control based on cooling water temperature and chilled water temperature, load control, control of the number of revolutions of a compressor, IGV control, diffuser width control and hot gas bypass valve control.

In order to realize this control, it was necessary for us to have a central processing unit capable of performing calculations at high speed and a memory, and therefore, we developed them newly and applied them.

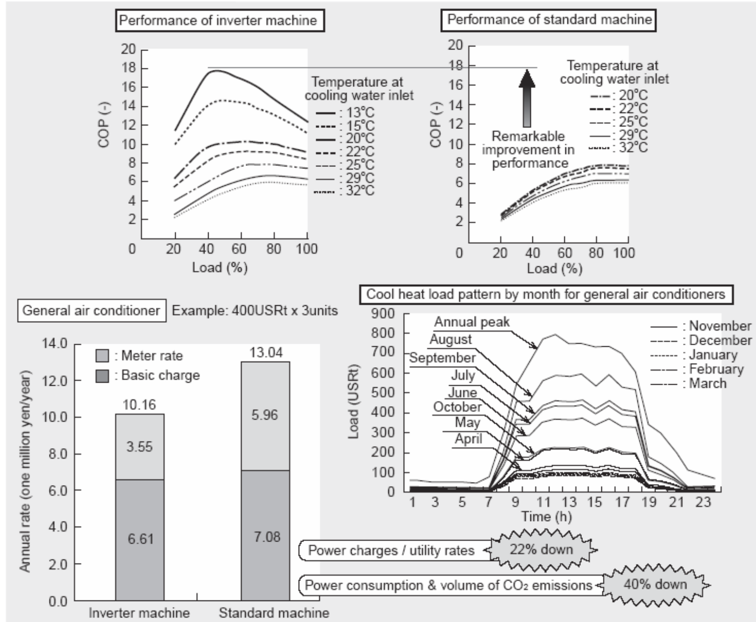


Fig. 8. Performance comparison of inverter and standard machine; annual energy saving effect at actual load pattern.

2.3.5 Performance evaluation (Evaluation of energy-saving effect)

The improvement effect was verified based on the yearly load pattern in the actual building air conditioning. As shown in Figure 8, we obtained the prospect of being able to reduce the yearly electric power consumption by about 22% compared with the fixed speed in the conventional standard turbo chiller series (NART series).

3 DEVELOPMENT OF HIGH-PERFORMANCE SMALL-CAPACITY TURBO CHILLER

In developing the high performance turbo chiller (product name: Microturbo series) to be applied to medium-scale heat source systems with 200 USRt or less, there were two challenges as shown below.

- Reduction in capacity as a centrifugal turbo chiller
- Conversion into a heat pump

3.1 Development Challenges

3.1.1 Downsizing in capacity

Air conditioning systems are classified into "individual air conditioning" and "central air conditioning" according to the type of heat source system. The former is a system similar to household air-conditioners, while the latter is a system made up of an air-conditioner which carries out ventilation and humidity control as well as room temperature control, and heat source equipment. Because the turbo chiller is heat source equipment for central air conditioning, the lower limit of the application was set as the capacity for the Microturbo development. The minimum capacity of the conventional turbo chillers was about 100 USRt (352 kW) for general-purpose models and about 200 USRt (703

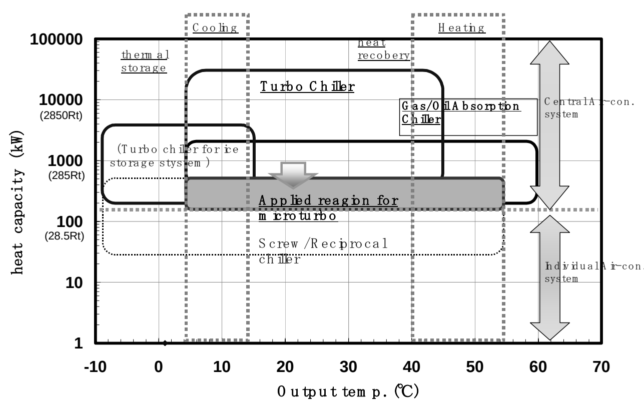


Fig. 9. Scope of heat source equipment for central air conditioning

kW) for high performance models. In contrast, the minimum capacity of the Microturbo series, high performance equipment, is set at 50 USRt, which means a virtual reduction to 1/4. This capacity reduction is the first priority challenge. (Figure 9)

3.1.2 Conversion into a heat pump

As heat source equipment for central air conditioning in the 50 USRt (175 kW) class, screw chillers and absorption refrigeration machines are typical. COP of the former is in the cooling COP range of 4.1 to 3.0 as of 2004, while COP of the latter is in the range of 1.0 to 1.35 (2.7 to 3.7; as converted by the total heating-power average receiving end efficiency of 36.6% in the "Law Concerning the Rational Use of Energy as revised"). Thus, products can be selected according to the type of power source.

In the case of the Microturbo to be driven by an electric motor, the screw chiller is a product against which a comparison is made directly. Screw chillers are available in the three-model lineup shown in Table 1 according to the intended purpose. The domestic demand for screw chillers is about 3000 units/year and the rate of shipment is almost 1/3 (about 1000 units/year) for each model. In the case of water-cooled chillers and air-cooled chillers which are exclusively for cooling, mainly factory heat source applications make up the majority. As for air-cooled heat pumps capable of producing output of chilled water and hot water, the air conditioning applications for office buildings, hospitals, hotels, etc. exceed the majority.

In the case of absorption refrigeration machines, on the other hand, dominated in the small equipment category by the direct firing type capable of producing output of chilled water and hot water, the air conditioning applications for office buildings, hospitals, hotels, etc. exceed the majority in the area with 80 USRt (281 kW) or less.

If absorption refrigeration machines and screw chillers are combined, the domestic demand is about 4000 or more units/year. In this capacity range, however, heat pumps capable of taking out chilled water and hot water make up about half of the total, and therefore, the development of an air-cooled heat pump was the second priority challenge.

3.2 Development Specification for the Microturbo Series

3.2.1 Development of the Microturbo series

To expand the high performance technology for large equipment to the Microturbo most surely, we developed and commercialized the Microturbo "W series", a high performance, small, water-cooled turbo chiller using the same component elements, and we confirmed the accomplishment of the challenge of achieving both size reduction and performance enhancement. Next, we fulfilled the procedure for developing the Microturbo "S series" adopting air cooling and conversion into a heat pump. In this document, the "S series" is introduced.

3.2.2 Specification for the Microturbo S series

The S series (see Table 1) is a high efficiency heat pump capable of supplying chilled water and hot water. It is made up of a compressor, water heat exchanger, sprinkling type fin & tube air heat exchanger, intercooler, water cooling system, fan, control panel and inverter

Fig. 11. External appearance of the Microturbo S series

panel. Figure 11 shows the external appearance of the Microturbo S series.

To achieve performance substantially exceeding that of the conventional models, we improved the heating COF by a newly developed high performance small turbo compressor; furthermore, we developed an air heat exchanger optimized as the sprinkling type, thereby improving the cooling COP substantially.

3.3 Development of High Performance Small Turbo Compressor

Technically, it is highly difficult to reduce the size while maintaining the high performance characteristics of a turbo compressor suitable for large capacities. The following four items in particular are important challenges.

- High efficiency impeller and aerodynamic design
- Low mechanical loss
- Excellent controllability
- Durability

Table 1. Development specification (Microturbo S series)

Type	MTSH175	
Operation mode	Cooling	Heating
Compressor type	MCM50S	
Capacity	175kW	175kW
Temp. of chilled and hot water	7℃out/12℃in	40℃in/45℃out
Outside air temp.	35℃DB/24℃WB	7℃DB/6℃WB
Power consumption	35kW	43.75kW
COP	5.0	4.0
COP of conventional machine	(2.7)	(3.2)
Control system	Variable speed control, inlet guide vane control, hot gas by-pass control, Fan control, sprinkling control	

In the development of the compressor for the Microturbo series, we carried the development forward in accordance with the procedure for the AART series and NART series in which performance enhancement was realized in large-capacity equipment (Figure 12).

Fig. 12. Small high performance compressor

3.3.1 High efficiency impeller and aerodynamic design

Even in the case of impellers having a diameter of 100 mm or less, data on actually measured performance of large equipment is effective because the law of similitude is applicable in the aerodynamic design. Based on the database of actual measurements which had been taken, we implemented the aerodynamic design optimizing the shape of the stationary flow channel in which a refrigerant gas flows and the shape of the impeller blade by performing an analysis with the aid of CFD (flow analysis), while paying attention to the increase in the influence of the boundary layer occurring in the flow field with the adoption of a smaller diameter. Furthermore, a performance verification test on the actual equipment scale was carried out and tuning of the detailed shape was done based on the actually measured performance and properties (Figure 13).

Fig. 13. Impeller (1st and 2nd stage)

3.3.2 Reduction in losses with the capacity reduction

Realization of high efficiency. With the size reduction, the leak loss increases relatively and the volume efficiency decreases. In order to reduce the leak loss of the impeller, the shape minimizing the gap with the mating part on the stationary side (shroud) was determined based on the following calculated values.

- Analysis accurately predicted deformation of the impeller blade tip due to centrifugal and fluid force.
- Bearing support rigidity was properly ensured by bearing selection and tuning, and the amount of rotary deflection of the impeller blade tip was calculated by a shaft vibration analysis.
- The maximum amount of shaft movement was carefully examined based on the axial support elasticity and axial load.

Machining accuracy. The impeller was a component to be machined by the NC 5 shaft processing machine and reproducibility of the required shape was ensured. However, with the reduction in diameter to 100 mm or less, restrictions in the shape occur due to the machining accuracy.

First, from the viewpoint of the production technology, we implemented basic optimization, such as tool selection, cutter path and the amount of feed of tooth, thereby minimizing the effect of the machining accuracy and surface roughness. Next, we incorporated constraining conditions in machining into the design, and then, planned the impeller shape again to minimize the reduction in efficiency.

Furthermore, we adopted a high-strength aluminum alloy with good machinability and minimum processing strain as raw material to achieve both cutting time reduction and high strength.

3.3.3 Excellent controllability (Variable speed control)

In the turbo compressor, the compression ratio can be controlled by making the impeller's number of revolutions variable.

By taking advantage of this property not available in the screw type or reciprocating type, it was made possible to follow the outside air temperature and cooling water temperature optimally. Because the variable speed control range extends to 100% to 40% of the number of revolutions, attention was paid to the aerodynamic characteristics not only at the rated design point but at the off-design point as well in the determination of the aerodynamic shape.

3.3.4 Durability performance

As the overhaul interval for large turbo compressors, 50,000 hours or eight years, whichever interval is shorter, is recommended as the standard. This is determined based on the life of the rolling bearing, a consumable part. Even in the case of screw compressors, the bearing is similarly a consumable part, although the overhaul interval differs.

The bearing life depends on the load, number of revolutions, bearing material and degree of contamination in the operation environment. In the case of bearings for chillers, we realized a long life and eliminated the need for replacement by utilizing the fact that they operate in a hermetically sealed environment isolated from the outside environment, such as air, maintaining a very clean environment and avoiding contamination, and adopting a combined bearing to reduce the load.

In addition, we paid attention to the bearing shape and lubrication, thereby achieving low losses at the same time.

3.4 High Efficiency Sprinkling Type Air Heat Exchanger

Enhancing the performance of a heat exchanger is very effective and important for improving the chiller performance because it can reduce the required pressure

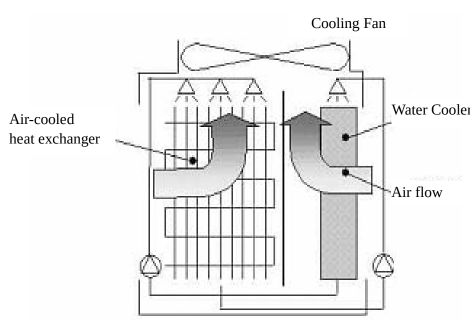


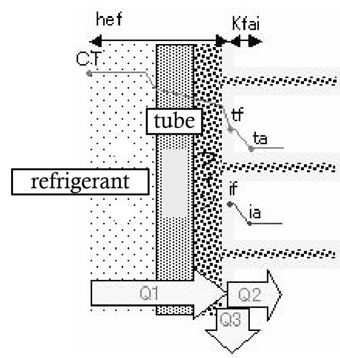
Fig. 14. Explanatory diagram of sprinkling water system

ratio for a compressor. In a water-cooled chiller, such as large turbo chiller, it is possible to discharge latent heat into air with a cooling tower by using cooling water at the 30℃ to 32℃ level under the 35℃ outside air condition. However, the conventional air-cooled chillers and air-cooled heat pumps have a structure in which sensible heat is directly discharged into 35℃ air. Even if a pure aluminum fin superior in the heat conduction property is used to ensure a sufficient heat transfer area, the performance of discharging heat into air falls behind the performance of discharging heat into cooling water in the shell & tube type or plate type water heat exchanger applied in water-cooled chillers. Therefore, we carried out development of a sprinkling type air heat exchanger which enables higher efficiencies by using the heat transfer property of water (Figure 14).

3.4.1 Improvement in air heat exchanger performance by evaporation heat transfer

We realized a substantial improvement in the air heat exchanger performance through the transfer of evaporation heat by allowing water to flow down to the fin and circulate during cooling, thereby forming a water liquid film on the surface.

In order to maximize the heating value of the transfer of evaporation heat on a water liquid film surface, Q_2, it is better for the liquid film temperature to be higher. However, when the liquid film temperature exceeds optimum temperature, it noticeably reduces the heating value of the heat transfer to a liquid film, Q_3, locally (Figure 15). Therefore, we worked out a proper value of sprinkling water temperature maximizing the total amount of heat transfer, Q_1, through simulation and verified it in the factor test, thereby optimizing the air heat exchanger performance.



Q_1: Total amount of heat transfer
Q_2: Amount of evaporation heat transferred from liquid film
Q_3: Amount of sensible heat transferred to liquid film

Fig. 15. Explanatory diagram of sprinkling water system

Because sprinkling water is circulating, its temperature goes up by heat input from the air heat exchanger fin. In order maintain proper temperature, we exercised our ingenuity and provided the cooler in the sprinkling system.

Furthermore, in a case where water is sprinkled on an air heat exchanger, it is important to give consideration to rust prevention for the fin; therefore, the fin was protected by triple rust prevention measures which are using rustproof and hydrophilic fin, control system designed to the function of restraining and controlling condensation, and using an organic inhibitor.

3.5 Unit Performance Enhancement

Enhancing the performance of only the compressor and heat exchanger with a high rate of contribution to the performance is not enough to enhance the performance of the unit. It is important to enhance the performance of the fan and fan-driving motor which make up of 10% of the total electrical power consumption. Furthermore, another important development challenge was a control panel enabling the control of the number of revolutions and optimum control by high speed calculation control for the expansion valve, etc.

3.5.1 Development of fan and fan motor

The fan blade was designed in line with the new design of the blade for outdoor equipment for intended business purposes. The outside diameter was made optimum for the Microturbo S series, and both low noise and high efficiency were achieved.

3.5.2 Development of new type of fan motor

During cooling, the sprinkling type air heat exchanger utilizes the transfer of evaporation heat of water, and therefore, requires a less amount of air compared with the conventional air heat exchangers. During heating, on the other hand, it requires the amount of air equivalent to that required in the conventional models. Therefore, we developed an exclusive DC blushless motor, making it possible to handle variable amounts of air and enhance the efficiency. In this way, we improve the efficiency and power factor at the specified point by 20% to 10% compared with the conventional induction motor, thereby realizing a substantial reduction in the power consumption of the fan.

3.5.3 Exclusive microcomputer board and control

In addition to the high speed calculation control (variable speed control, expansion valve control, inlet vane control and hot gas bypass control) time-proven in the high efficiency inverter turbo NART-I, we newly designed an exclusive board to which fan control and heat pump operation control were added, thereby realizing high performance characteristics equivalent to those of large equipment. In addition, we also gave consideration to the convenience by enabling the operation, status display and scheduled operation with the exclusive remote control.

3.6 High Performance Characteristics and Economical Efficiency

3.6.1 Equipment performance

In the Microturbo S series, substantial performance improvements over the conventional equipment were realized by improving the mechanical loss, auxiliary machinery power, and performance of each element of the compressor and heat exchanger.

Compared with our company's conventional equipment, a reduction of 46% in the power consumption was achieved and the reduction in the compressor power due to the improvement in the heat exchanger and compressor performance contributed most. Figure 16 shows the comparison with the conventional equipment in the power consumption at the specified point.

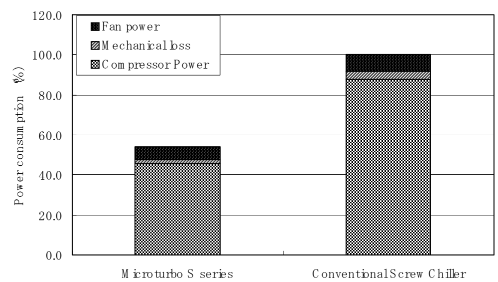


Fig. 16. Comparison between Microturbo and conventional equipment

The cooling performance characteristics of the Microturbo S series are shown (Figure 17). In both of them, substantial performance improvement was realized in the entire area compared with the conventional equipment, making it possible to save energy and reduce the running cost.

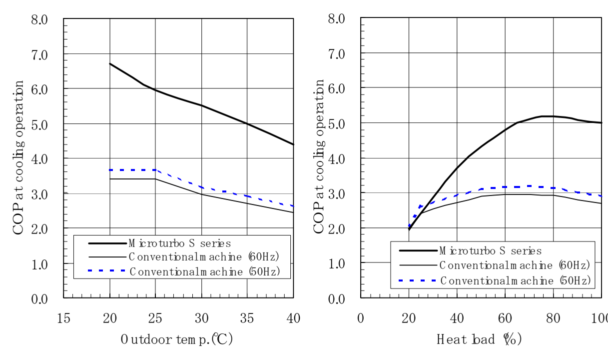


Figure 17 Comparison of cooling performance characteristics

3.6.2 Energy-saving and running cost reducing effect

Assuming combined load models focusing on office buildings, including the food-service industry and accommodation facilities, we assessed the yearly running cost based on the control of four pieces of heat source equipment. The equipment with which comparisons were made was the conventional equipment; screw chiller and absorption water chilling/heating equipment (including cooling tower). Table 2 We did a trial calculation of the electricity rate, gas rate, water rate for cooling water, and cost of chemicals added to cooling water. (As for the outside air temperature, electricity rate, gas rate, and water rate, the data in the Tokyo district was adopted.)

Table 2. Heat source equipment for trial calculation of running cost

Item	Microturbo	Screw chiller	Absorption chiller
Cooling capacity	175kW Note1)		
Cooling water temp.	7℃out/12℃in		
Outside air temp.	35℃DB/24℃WB		
Cooling COP	5.0	2.7	1.1 (3.0) Note3)
Heating capacity	175kW Note1)		
Hot water temp.	45℃out/40℃in		
Outside air temp.	7℃DB/6℃WB		
Heating COP	4.0	3.2	0.95 Note2) (2.6) Note3)

Notes: 1) As for the cooling capacity and heating capacity, the values are the same for all the units for the same of convenience; 2) Boiler efficiency, 3) Converted by the total heating-power average receiving end efficiency of 36.6% according to the "Revised Law Concerning the Rational Use of Energy"

In the case of the Microturbo S series, it is possible to reduce the running cost by about 30% compared with screw chillers. Especially, the reduction effect reaching about 40% during cooling is large and a cost reduction effect results even if the increment, a water rate and cost of chemicals to be charged newly, is added. Also, in comparison with the absorption refrigeration machine, it is possible to reduce the running cost by about 20%. In addition to the advantage in cooling, the difference in the water rate contributes to the total difference (Figure 18).

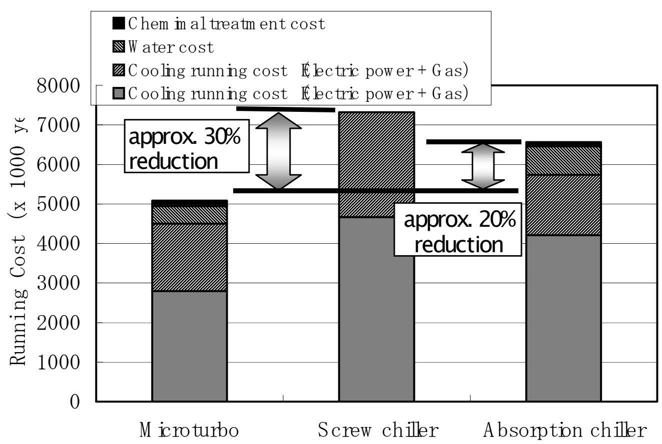


Fig. 18. Comparison of running cost