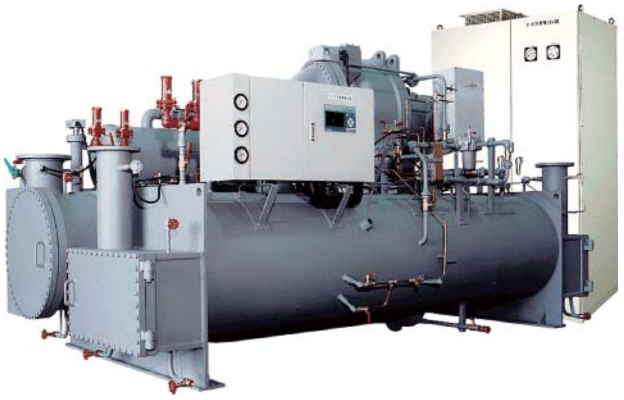

New Model Turbo Chiller, AART Series, Contributes to the Reduction of Energy Consumption under Year-Round Operation

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Among turbo chillers loaded with alternative refrigerant (HFC134a), the AART Series is one of the world's most efficient. This paper discusses the technology used in the New AART Series, a lineup of Turbo Chillers upgraded through the introduction of a two-stage inlet guide vane control mechanism and a vane-less diffuser to enhance performance during partial loads. A high-end, high-speed control panel and inverter-driven motor are combined with a variable-speed control logic. The design takes advantage of the aerodynamic characteristic of the compressor and brings about optimal operational control under changing loads and outside temperatures throughout the year. Compared to the conventional variable-speed machines of the NART-I Series, the new AART Series is expected to save considerably more energy in year-round operation.

1. Introduction

Turbo chillers are a category of cold heat source machines. Most in service today are used for heat storage, air-conditioning for factories and commercial facilities, process cooling in chemical plants, and district heating and cooling with heat-supply services. In recent years, from the view of protection of the Earth's ozone layer, the use of CFCs which was used as refrigerant has been completely eliminated, and efforts are underway to abolish the use of HCFCs in future. Since 2000, Mitsubishi Heavy Industries, Ltd. (MHI) has been trying to improve the performance of machines using HFC-134a, a specific refrigerant with an ozone depletion potential (ODP) of zero. With success in this endeavor, MHI will save more energy and help conserve the global environment.

Fig. 1 gives an overview of the trends in the performance of typical turbo chillers. The chiller performance (COP = performance of coefficient) is plotted on the vertical axis year by year (horizontal axis). The demand for improved energy saving has steadily risen since the energy crisis of the 1970's. To respond, engineers have improved the machine performance by 1.5 times over the more than two decades since 1980. The current model, the highest-performing yet, has a COP value of 6.4. This approaches almost the upper limit.

Fig. 1 shows the COP of MHI turbo chillers, the specification for rated operation, and the COP for year-round operation. The turbo chiller is rarely operated under rated load (partial-load operation accounts for 90% of the total operating time). For most customers, therefore, the performance under partial-load operation has the most important bearing on running costs for year-round operation. Conventional turbo chillers have nearly the same COP as equipment in year-round operation. The inverter-driven variable-speed chiller, however, has a higher COP. This makes it impossible to reliably evaluate the performance of the chillers based on comparison with the COP for equipment alone.

One-third of all turbo chiller orders received by MHI are orders for the inverter-driven chiller with high COP in year-round operation. Higher sales volumes are not yet anticipated, however, as the initial cost to purchase is still somewhat high. To improve the benefits of purchase, MHI has tried to enhance the energy-saving performance of a fixed-speed chiller during partial-load operation. The technology used to accomplish this has also been used to enhance the performance of MHI's inverter-driven chiller.

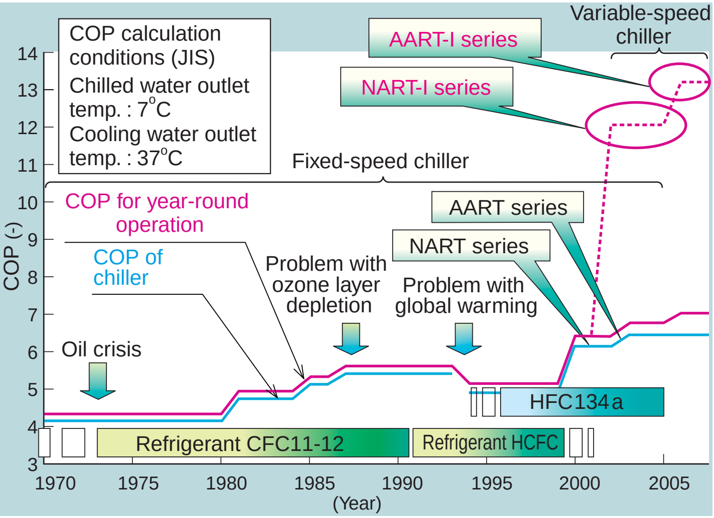

Fig. 1 Trends in turbo chiller performance and refrigerant

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2. Technical strategies for reducing power consumption under year-round operation

Compressor performance has a great influence on the performance of turbo chillers. To improve chiller performance in year-round operation, it is crucial to expand the range of operation and to enhance the compressor performance. The following sections describe recent improvements in compressors used in turbo chillers.

2.1 Improving the aerodynamic performance under partial-load operation

2.1.1 Existing problems of conventional chillers

Fig. 2 shows the structure of a compressor. The existing AART model has a diffuser vane with a small cord ratio in the discharge channel of the 2nd stage impeller to enhance the compressor efficiency. The efficiency enhancements of the diffuser vane are greatest at or near the rated flow rate for the refrigerant. As a drawback, however, the diffuser vane sometimes causes compressor failures due to peeling of the flow caused by changes in the inlet angle under partial load operation with low-flow-rate refrigerant. The use of a diffuser width control mechanism to avoid this peeling phenomenon and stabilize operation has failed to improve the compressor efficiency. The use of a movable diffuser vane has been proposed as a strategy for improving the flow status under partial-load operation, however we have rejected this proposal because of the complex construction of this vane and possible problems in sealing due to operation through the pressure partition wall inside of a compressor. The low tolerance for resonance also made it impossible to adopt a variable-speed type diffuser vane.

Noting that the diffuser width control precluded improvements in stable operation, we examined the possibility of installing a two-stage inlet guide vane (IGV) for controlling the second-stage flow, instead of a two-stage diffuser vane and a diffuser width control.

2.1.2 Examining improvement

(1) Second-stage IGV

As shown in the upper portion of **Fig. 3**, we examined two types of second-stage IGVs:

- Return-vane-combined IGV: fixed blade parts are located upstream of the two-stage impeller to enable adjustment of the inlet angle with the return vane's movable rear edge.
- Standalone IGV: the IGV is located downstream of the return vane.

Fig. 3 shows the relationships between the IGV opening and pressure loss in each type of IGV. For IGV openings of 100 - 60%, the return-vane-combined type has slightly less pressure loss than the standalone type. For an opening of 30%, the pressure loss of the standalone type is slightly higher.

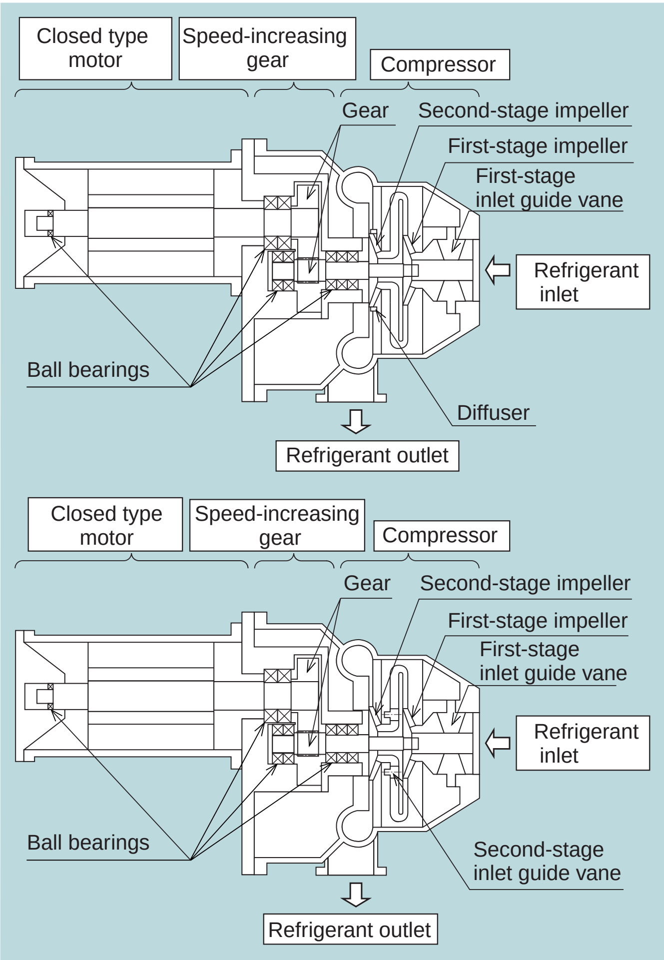

Fig. 2 Structure of second-stage inlet vane
The upper portion shows the conventional type; the lower portion, the improved type.

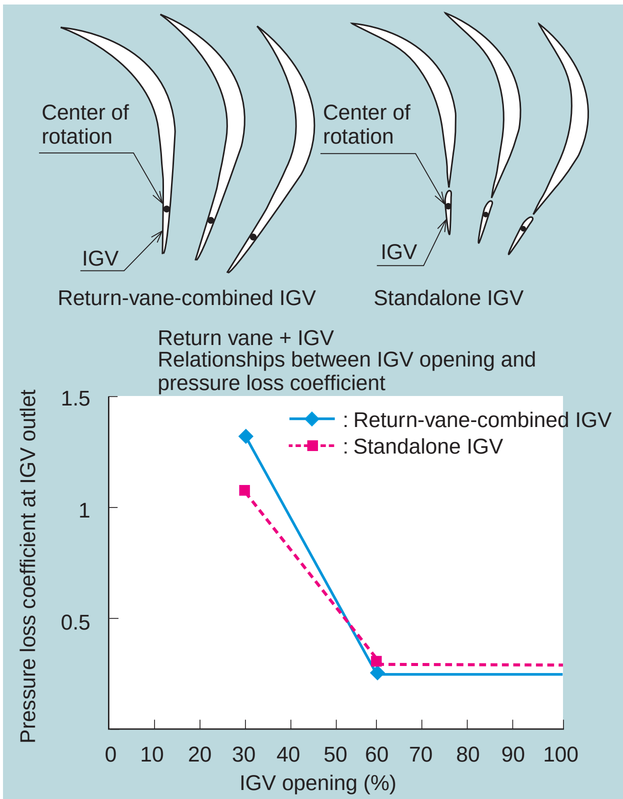

Fig. 3 Comparison of two-stage IGV location and pressure loss in compressor

Fig.4 shows the distribution of the Mach number for an opening of 30%. The standalone type has a smaller overall pressure loss for two reasons: first, the flow doesn't stagnate between the return vane and IGV; second, the uniform tailing vortex induces very little pressure loss. Based on these results, we decided to adopt the standalone type IGV. In selecting the position of the circumferential direction of the IGV, we kept the pressure surface of the rear edge of the return vane on the same position as the negative pressure surface of the front edge of the IGV.

(2) Improving the return vane

In installing inlet guide vanes at the inlet of the second stage impeller, it is necessary to optimize the gas flow from the first stage to the second stage impeller. To reduce the pressure loss, we decreased the range of peeling (i.e., the unstable range) of refrigerant gas flow at the return vane located in the static stream toward the second-stage inlet by improving the distribution of the blade angle with the aid of flow analysis. In selecting the blade shape, we considered the pressure loss induced by the flow under both rated-load and partial-load operation. With the adoption of the return vane with an improved shape, the peeling range for the negative-pressure surface can be narrowed and moved downstream, as shown in **Fig. 5**.

And to reduce the pressure loss under partial-load operation, the front edge of the return vane is configured as a large-radius curve.

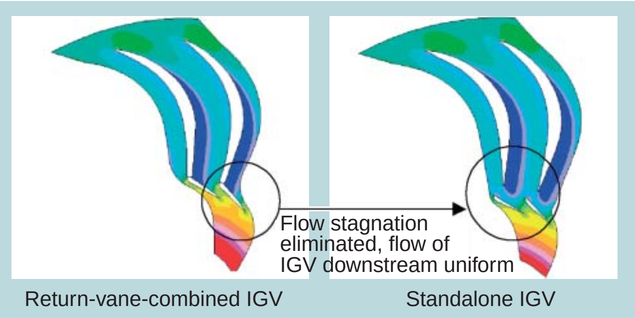

Fig. 4 Examination of two-stage IGV location based on flow analysis
The Figure shows the Mach number distribution.

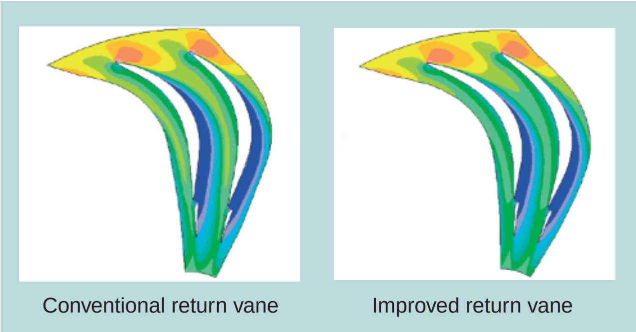

Fig. 5 Return vane improved based on flow analysis

(3) Improving the seal mechanism

The NART Series and its predecessors use a metal labyrinth seal at the drive shaft and static parts to avoid gas leaks from the rear of the two-stage impeller. To minimize gas leak from the rear side of the impeller and reduce leak loss, the AART Series adopts an abradable seal mechanism which uses a plastic ring for static parts to completely eliminate clearance, taking deformation of driving shaft parts through the centrifugal force associated with high speed into account. This structure reduces the refrigerant gas leaks in the tail of the second-stage impeller to about zero.

2.2 Enhancing the aerodynamic properties of the compressor

Fig. 6 is a map for explanation of the aerodynamic properties. The data illustrate the effects of the IGV installed downstream of the return vane mentioned above. The adoption of the second-stage IGV expands the operation range up to the rotating stall range previously unavailable, thereby expands the range of operation in which the diffuser width control and the hot gas by-pass control are left unused. In the previous compressor, use of the diffuser width control reduces the compressor efficiency where the refrigerant flow rate is low and the hot gas by-pass control reduces the efficiency of the overall system. As a result, COP is improved under partial-load operation.

In addition, in order to enhance the performance of partial load operation, the improved compressor has eliminated the diffuser width control required in the previous compressor for the narrowing of the flow channel to avoid the rotating stall. This results in enhancement of the compressor efficiency under partial-load operation.

The newly developed compressor impeller is designed to avoid the resonance in the operational rotating speed range caused by interference from excited vibration power arising from structures such as the static IGV. With the elimination of the movable diffuser as above

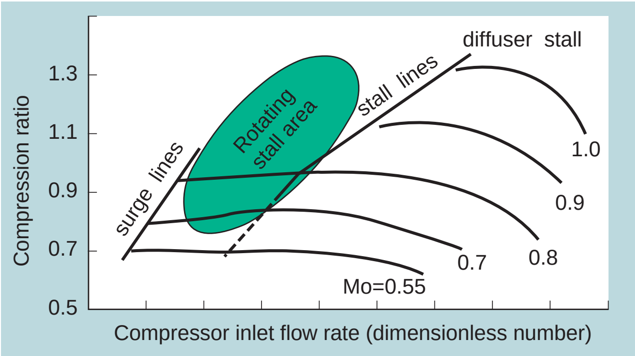

Fig. 6 Map of Aerodynamic Properties to Illustrate the Improvements
With our improvements, the compressor now operates in the elliptic region previously unused.

mentioned, the stalled cell in the improved compressor is also designed to eliminate resonance to enable normal operation of the improved compressor in the rotating stall area (elliptic region shown in Fig. 6). Strength analyses and practical unit pressure measurements by our group have confirmed that the compressor has adequate strength to resist forced vibration.

Detailed measurements of pressure changes at the compressor outlet during verification tests have identified the number of stalled cells and the rotating speed in the rotating stall range. The data have also been used to confirm that no resonance is induced by the impeller vibration frequency at any revolution speed or under any other condition of operation.

3. Optimum control to allow variable-speed operation and enhance efficiency under partial-load operation

The control logic has been adjusted to accommodate the change in the aerodynamic performance accompanying the switchover from the second-stage IGV control (the previous method) to the new method of controlling the diffuser width of the second-stage impeller outlet used for the AART Series. Next, we measured the aerodynamic properties in detail during practical operation to identify the aerodynamic changes resulting from the addition of the second-stage IGV and elimination of the diffuser vane. We developed a method of arithmetic control for identifying the optimum aerodynamic properties by digitizing the measurement results and processing them in a control panel. The AART Series is equipped with a control panel configured for this purpose. **Fig. 7** illustrates the concept of the control system and the expanded range of operation. The AART-I Series, a variation of the standard AART Series with variable-speed control, performs significantly better than the variable-speed NART-I Series, our previous model, under partial-load operation. The improvement is greatest under operation loads of 50% or less.

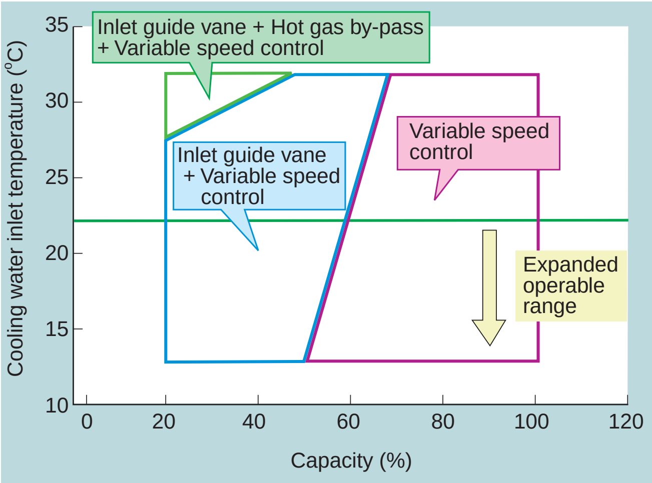

Fig. 7 Schematic diagram for operation range and control system

4. Control panel and monitoring function

Optimal control requires the rapid processing of extremely complex characteristic expressions and data collected from many detection points. Without high-speed calculation capacity, the control panel used would fail to accurately keep track of the changes in load. In the AART Series, a control panel equipped with a high-end CPU is built in as standard equipment, together with a 7-inch TFT color LC display to ensure high visibility and operability. The 24-hour monitoring function is also equipped as standard to ensure optimal control of operation and rapid response to customer requirements. The following information is sent to a tele-monitoring center over phone lines;

- Daily operation data,
- Storage of data on abnormalities and automatic notification to service staff, and
- Real-time data collection.

You can also add optional communication and monitoring functions such as connection to automation systems for commercial use, central monitoring using specialized communication equipment, and web-based monitoring using network system.

5. Performance verification results and estimation of cost-effectiveness

Fig. 8 compares the COP (projections) values of the previous series with those of the new AART Series. The improvement in the COP of the new model is significant at load ratios of 60% or less.

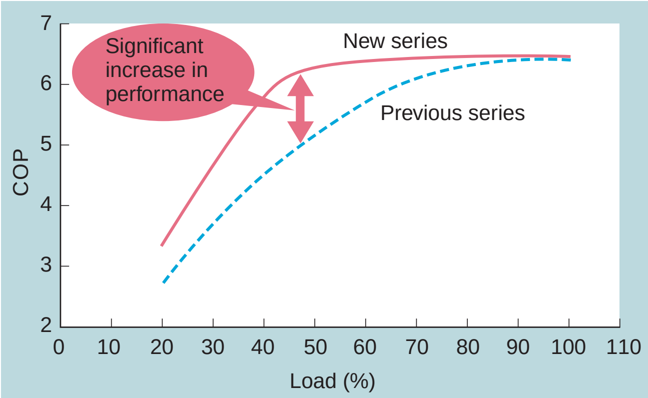

Fig. 8 Comparisons of estimated COP values for new and previous AART models
Comparisons are based on the fixed-speed model.

Table 1 Comparison of estimated annual operation power consumption

Comparison of estimated annual operation power consumption				
	AART previous model	NART-I previous model	AART new model	AART-I new model
Variable speed	Fixed speed	Variable speed	Fixed speed	Variable speed
Business facility	100 %	94.4 %	98.3 %	86.7 %
Commercial facility	100 %	93.0 %	98.5 %	84.0 %
Culture/Training	100 %	94.5 %	99.3 %	97.8 %
Factory	100 %	96.0 %	97.0 %	94.9 %

Note: Cooling water temperature is based on JIS specifications

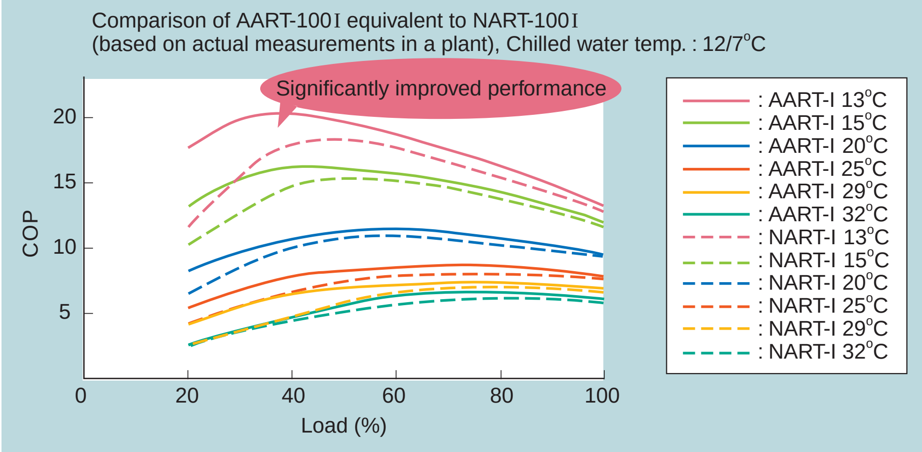

Fig. 9 Comparison of estimated COP for previous model (NART-I) and new model (AART-I) (variable-speed models)

Fig. 9 compares the COP for variable-speed models. Under partial-load operation with low cooling water inlet temperature, the addition of variable-speed control significantly enhances the performance (maximum COP of more than 20). **Table 1** shows the estimated annual power consumption by load pattern (the cooling water inlet temperature is applicable to the load specified in Japanese Industrial Standards [JIS]). The new variable-speed AART-I Series is estimated to consume 15% less power annually than the previous fixed-speed AART Series under the same load conditions. (Assuming a minimum cooling water inlet temperature of 27 ?C, as the condition specified by JIS. (Actual power consumption will improve more when the actual water temperatures drop lower in winter.)

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