Solutions for Reduced Life Cycle Costs of Centrifugal Compressors in Oil and Gas Industry

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OVERVIEW: High reliability and reduction of life cycle costs are valued in centrifugal compressors used in the oil and gas industry. These centrifugal compressors are required to maintain high efficiency and obtain wide operating range to cope with changes in load based on future demands. This paper provides an impeller development that has achieved high efficiency and a wide operating range. It also outlines IGV control system development to achieve the power saving at low load requirement. This IGV technology developed can be applied in a wide variety centrifugal compressors as well as for cryogenic application.

INTRODUCTION

PROCESS centrifugal compressors are widely used in gas processing plants and gas pipelines in the oil and gas industry, and are booming increasing demand in the wave of recent soaring crude oil and natural gas prices.

Reduction of life cycle costs is a key issue for compressors used in this application, which are required to have high efficiency and sufficient operating range to be suitable for changes in load including the future demand. This is because use of highly efficient compressors can reduce power consumption, which results in the reduced CO_2 emission and allows plant capacity expansion without replacement or modification of the existing compressors. In order to satisfy these requirements, an efficient capacity control method needs to be established in addition to the improvement of impellers with higher efficiency and a wider operating range.

This article discusses an impeller that was developed to meet the above requirements and a capacity control method by the IGV (inlet guide vane).

DEVELOPMENT OF HIGHLY EFFICIENT IMPELLER WITH WIDE OPERATING RANGE

Many centrifugal compressors in the oil and gas industry are of multistage impellers and it is known that the latter stages determine the overall operating range from surge to choke. Due to the compressibility of the gas nature, the latter stage the gas flows through, the higher its density becomes. Therefore, an impeller

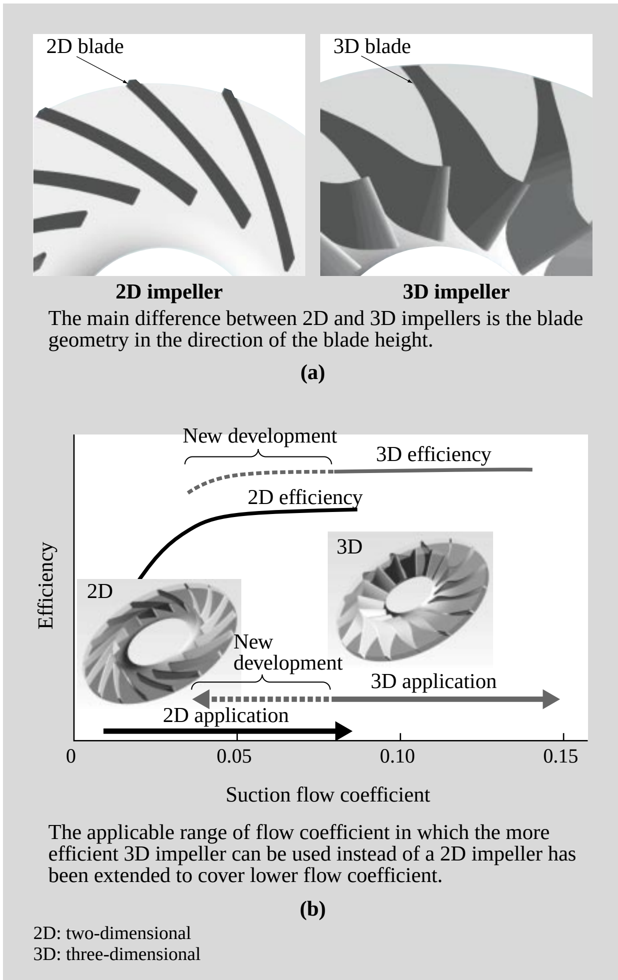

Fig. 1—Blade Geometry and Applicable Range of 2D and 3D Impellers.
Comparisons of the blade geometries (a) and applicable ranges (b) of 2D and 3D impellers are shown.

in the latter stage handles smaller volume flow and results in a lower flow coefficient. Accordingly, it is important to enlarge the operating range of the impellers with a lower flow coefficient used in the latter stages in order to ensure the overall operating range from surge to choke.

Traditionally, two-dimensional impellers are employed for the low and medium flow coefficient regions. The blade profiles of these impellers are unchanged in the direction of the blade height [see Fig. 1 (a)]. Use of three-dimensional impellers is limited to the regions with a high flow coefficient due to constraints of the manufacturing process, although it has been known that three-dimensional impellers can provide higher levels of performance. Hitachi has realized a new manufacturing process that allows three-dimensional blades to be used for lower flow coefficients (where the impellers have less blade height), and has successfully enlarged the applicable range of three-dimensional impellers to the low flow coefficient region [see Fig. 1 (b)].

The following chapter discusses the aerodynamic design using CFD (computational fluid dynamics) and fluid performance of a new three-dimensional impeller, which is intended for low and medium flow coefficients where two-dimensional impellers have been used in the past.

CHANGING FROM TWO-TO THREE-DIMENSIONAL IMPELLER

Aerodynamic Design of Three-dimensional Impeller

When changing blade geometry of impeller from two to three dimensions, the main dimensions of the three-dimensional impeller shown in Fig. 2 remain the same as its two-dimensional counterpart in order to maintain the interchangeability in space with existing ones.

The aerodynamic design of the three-dimensional impeller uses a method whereby the blade geometry is optimized through the iterative process of blade generation and CFD evaluation of it. The blade geometry for each successive iteration is determined based on the aerodynamic load coefficient L_d defined in equation (1):

$$L_d = \frac{w_{su} - w_p}{\overline{w}} \quad ; \overline{w} = \frac{w_{su} + w_p}{2} \quad \cdots\cdots\cdots\cdots\cdots \quad (1)$$

where w_{su} represents the relative velocity of the suction surface, and w_p represents the relative velocity of the pressure surface.

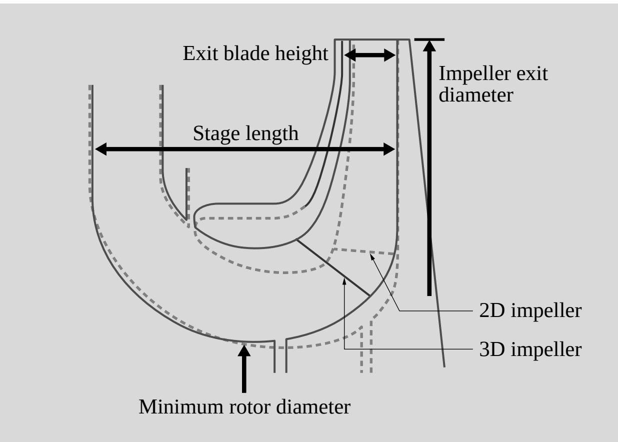

Fig. 2—Constraints on Changing from 2D to 3D Impellers.
Some constraints apply to the development of a new 3D impeller so as to keep the main dimensions of a 2D impeller.

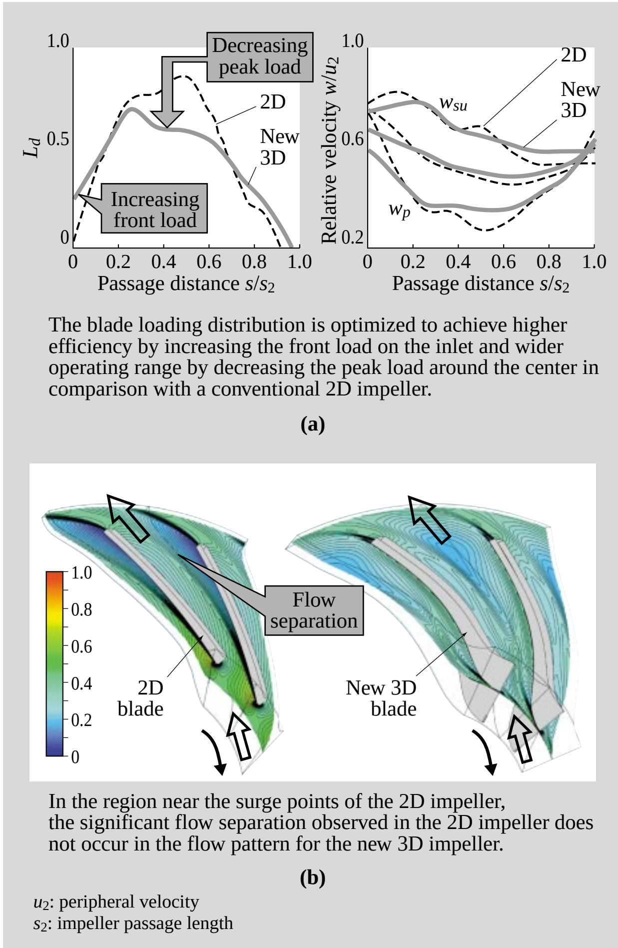

Fig. 3—Comparison of Internal Flow Pattern of 2D and New 3D Impellers.
The aerodynamic load and the relative flow velocity distribution on the shroud side (a) and the relative Mach number distribution at mid-span near surge point on the 2D impeller (b) are shown.

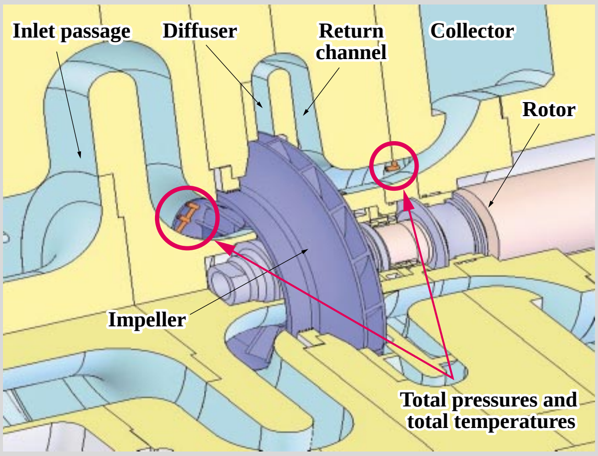

Fig. 4—Cross-sectional View of Tested Compressor.
A cross-sectional view of components of the tested compressor is shown.

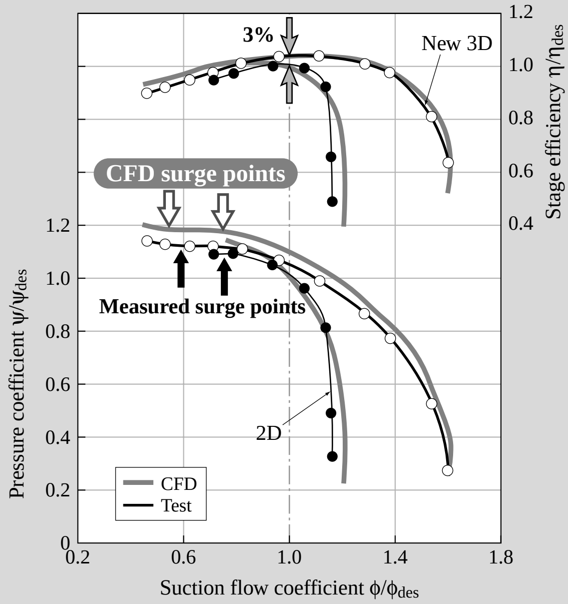

Fig. 5—Test Results and Performance Predictions Obtained Using CFD for 2D and New 3D Impellers.
The new 3D impeller achieved 3% higher efficiency and 2.8 times wider operating range compared with the 2D impeller. The prediction accuracy by using CFD (computational fluid dynamics) for these impellers is within ±5%.

Fig. 3 shows a design example obtained using optimization. Aerodynamic load distribution for this new impeller is characterized by a higher front load around the inlet and lower peak load around the center compared with previous designs [see Fig. 3 (a)]. The increasing front load around the inlet drastically decelerates the flow in the first half of the impeller and improves the impeller efficiency. In addition, the decreasing peak load around the center of the impeller increases the minimum flow velocity on the suction surface, and thus prevents flow separation inside the impeller and enlarges the surge margin [see Fig. 3 (b)].

Performance Test Apparatus and Test Method

Fig. 4 shows the cross-sectional view of the model compressor for a performance verification test. The tested compressor is a single stage whose performance characteristic curve was obtained by changing the volume flow at a machine Mach number of 0.9. As shown in Fig. 4, the pressure coefficient and efficiency were obtained by measuring total temperatures and total pressures at the impeller inlet and the return channel exit.

Test Results

Fig. 5 shows a performance comparison between the three- and two-dimensional impellers in the test. Both the efficiency η and the pressure coefficient ψ are normalized by those of the two-dimensional impeller at the design flow coefficient.

As shown in Fig. 5, the design point efficiency of the new three-dimensional impeller (○ in Fig. 5) was increased by about 3% compared with the two-dimensional impeller (● in Fig. 5). At the same time, its operating range was about 2.8 times as wide as that of the two-dimensional impeller. The combination of changing from a two- to a three-dimensional impeller and optimizing the aerodynamic load distribution can achieve significantly improved efficiency and a wider operating range.

The choke margin of the newly developed impeller is also increased remarkably. This is mainly due to the wider throat area resulting from the change in blade geometry to three dimensions. This increased choke margin makes it possible to deal with increased plant capacity in future.

Fig. 5 shows also the compressor performance characteristics obtained using CFD. CFD predicts the actually measured compressor performance with good accuracy. The prediction accuracy is also sufficient for surge points. Hitachi intends to utilize the analysis know-how and verification data obtained through this work in future development to achieve higher performance of compressors and to complete such development more quickly.

INLET GUIDE VANE CONTROL

Use of the IGV is one of the capacity modulation methods used in turbo machinery. This method has the following characteristics:

(1) Capacity can be regulated without a significant decrease in compressor head.

(2) A wider operating range at the low flow rate can be achieved compared with speed variation control.

(3) Losses are lower compared to using suction throttling.

(4) Higher power saving effect at constant discharge pressure application compared with speed variation control

These characteristics make suitable for applications that require virtually constant discharge pressure even when plant load is changed. A typical example is the BOG (boil-off gas) compressor in a LNG (liquefied natural gas) plant.

This application involves the BOG compressor handling gas with extremely low suction temperature (approximately –160°C) and requires that capacity be controlled based on the LNG evaporation capacity within the storage tank. The IGV must be able to deal with wide variations in load as gas evaporation varies depending on the ambient temperature and also during loading from LNG tankers. The IGV system requires a highly reliable drive mechanism to withstand use in such harsh environments.

The following sections describe the results of performance prediction at partial load and reliability verification test on a simulated compressor equipped with an IGV in a cryogenic condition for the purposes of long-term maintenance-free operation.

IGV Component

Fig. 6 shows a typical structure of the IGV equipment. The guide vanes are placed in parallel flow channel of the compressor suction. The flow from the suction nozzle is distributed in the circumferential direction through an annular channel in the casing. The flow velocity angle of pre-whirl is generated by the variable guide vanes.

Spur gears are fixed on the end of each guide vane shaft and engage with the internal teeth of the ring gear. The external teeth of the ring gear engages with a spur gear fixed on the end of a drive shaft that passes through the compressor casing and is oriented in the axial direction. In this mechanism, the external drive

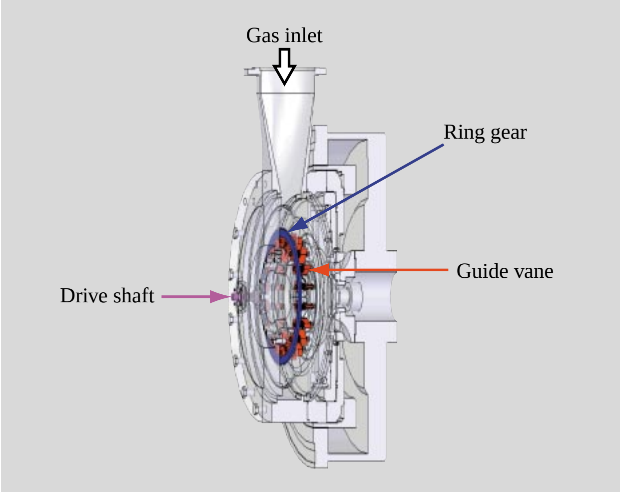

Fig. 6—Structure of the IGV Equipment.
An example structure of the IGV (inlet guide vane) equipment is shown.

mechanism rotates the drive shaft to turn the ring gear, and this then turns the IGV shafts.

The testing focused on the following points to ensure that the IGV mechanism could operate for long period of time:

(1) Review the guide vane structure and suction flow channel geometry to improve the effect of pre-whirl flow and lower pressure loss.

(2) Select suitable materials with superior wear-resistance in cryogenic and completely lubricant-free condition.

(3) Review the structure to prevent overcooling of the shaft bearings and seals.

Performance Prediction Accuracy on Partial Load Characteristics under IGV Control

It is necessary to predict the relation between the IGV angle and compressor partial load performance with a high degree of accuracy in order to meet the partial load of process operation. A single-stage compressor was used in this new development program to confirm how prediction corresponds to experiment in fluid performance. First, a three-dimensional viscosity flow analysis was conducted from the suction nozzle to the diffuser passage to review the optimized flow channel geometry, the calculated flow velocity angle of pre-whirl characteristics in the guide vanes and the pressure loss in the suction channel. Then, the guide vane characteristics were verified by use of the single-stage compressor unit with attached IGVs. Fig.7 shows the relation between the guide

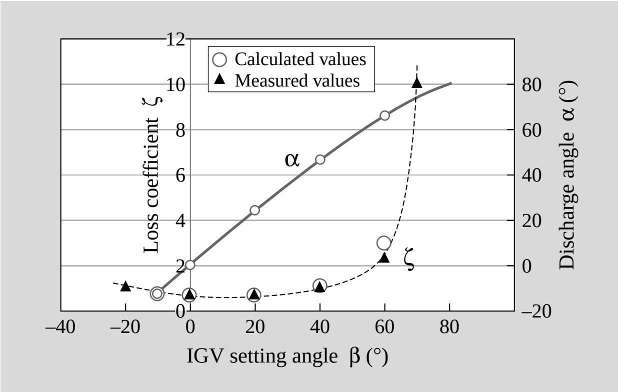

Fig. 7—Pre-whirl Characteristics and Loss Coefficient for IGV Equipment.
This figure shows the relationship between the guide vane regulating angle, flow angle of pre-whirl, and pressure loss coefficient.

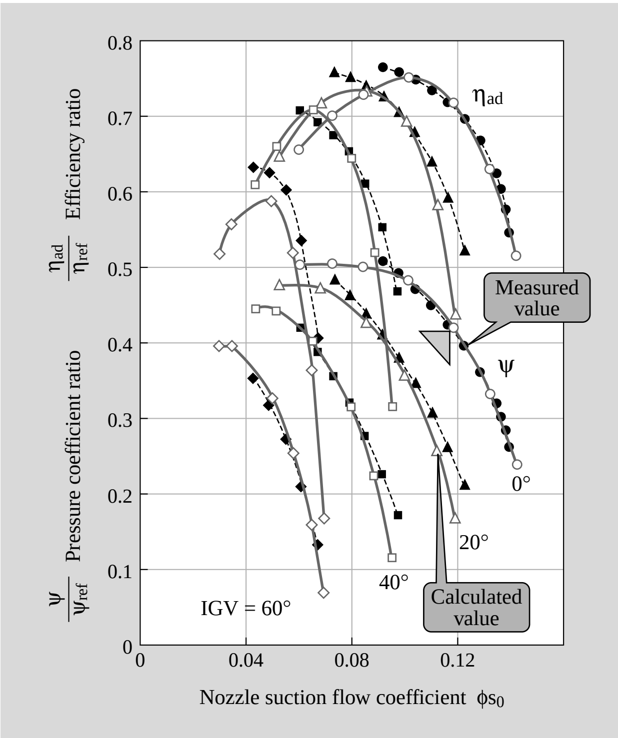

Fig. 8—Prediction and Actual Measurement of Fluid Performance of Compressor with IGV.
The predicted compressor part load performance curve is closely matched to the actual measurement.

vane regulating angle, flow angle of pre-whirl, and pressure loss coefficient. The calculated pressure loss coefficient values are in good agreement with the experiments. Since the predicted compressor partial load performance characteristics are almost consistent with the observed results as shown in Fig. 8, these results also verify the prediction accuracy for the pre-whirl angle by IGVs.

Reliability Test for Shaft Bearing and Gear Components in Cryogenic Condition

Lubricant oil and agents such as grease cannot be used at extremely low temperatures because they solidify. It is important to select appropriate materials and establish the selection criteria to ensure the long-term reliability of the IGV components which include sliding and meshing parts that must function without lubrication exposed to cryogenic temperatures. For this reason, component tests were conducted on the shaft bearings and gears in a cryogenic condition (–160°C) using each test apparatus that worked by cryogenic nitrogen.

As prospective tribo-material for sliding parts, eleven kinds of materials such as PTFE (polytetrafluoroethylene), copper, and PEEK (polyetheretherketone) materials commonly used in cryogenic and dry condition are considered. The results showed that PEEK material has the best wear resistance under low-sliding velocity and high-load conditions.

A 10^5 cycle test rotating forward and backward alternately up to 90 degrees each direction was conducted and a PEEK material resulted a specific wear rate of 2×10^7 [mm^3/(N•m)]. This is sufficiently low enough to meet the target wear life.

On the other hand, austenitic, precipitation-hardened, and ferritic stainless steel materials for gear components were selected to conduct an endurance test under low-sliding velocity and high-meshing load conditions at a temperature of –160°C. The test results clearly indicated that the combination of materials had a significant impact on wear characteristics and showed that austenitic stainless steel has particularly good wear resistance. Although material hardening techniques are widely used to enhance wear resistance at room temperatures, the results showed that some characteristics in cryogenic condition are different to what would normally be expected. A 10^5 cycle wear testing for meshing was also conducted using these materials and this confirmed that the target life could be obtained.

Reliability Test for Cryogenic IGV Equipment

A reliability verification test was conducted by making a complete cryogenic IGV equipment able to simulate actual conditions consisting of the suction casing, IGV components drive unit, and hot oil supply unit of a BOG compressor. Cryogenic gas at –160°C was supplied by flushing liquid nitrogen upstream of the suction nozzle, passing it through the vane part by a induced fan, and then circulating it back to the suction nozzle. Hot oil was supplied to the suction casing to simulate the prevention of overcooling at the DGS (dry gas seal) of the compressor shaft seal and journal bearing housing section. A spring was used to load equivalent maximum force and moment in the fluid condition to each vane shaft .

By use of this simulated test equipment, rotations forward and backward directions alternately to zero to 100% of the vane opening were repeated 10^4 times to determine the rate of wear in sliding parts, meshing parts, the seal leakage rate and the change in the driving force. Fig. 9 shows the reliability test system during operation. Ice coating was observed on casing surfaces affected by cryogenic gas. The suction side is located on the near side, and the discharge side on the far side in the figure. Cold insulation was installed on the casing surface on the discharge side.

Fig. 10 shows the change in casing surface temperature with time. The horizontal axis represents the time duration from the test start and the vertical axis represents the temperature at several points of the casing. It took several hours for the whole equipment to reach a state of temperature equilibrium although the gas temperature in the nozzle part decreased rapidly. The casing surface temperature on the discharge side (1) is close to the cryogenic gas supply temperature from the nozzle at temperature equilibrium. The casing surface temperature on the suction side (5) indicates a phenomenon whereby the temperature drop is limited due to heat transfer from the surrounding atmosphere. It was also confirmed that the casing temperature (13) at locations where DGS is installed will not be overcooled because of the hot oil supply. The tests confirmed that the hot oil will not freeze inside the casing as the 2°C temperature drop of the hot oil from 40°C supply.

The simulated casing surface temperature by FEM (finite element method) of this reliability system is also described in Fig. 10 comparing with the experiment ones. These calculated temperature values are almost in good agreement with the experiment ones. The large temperature difference between the casing surface of the suction side and the discharge

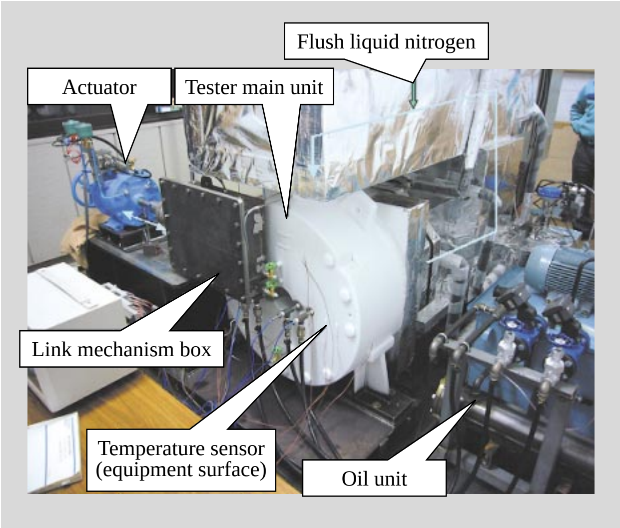

Fig. 9—Cryogenic Reliability Test of IGV Equipment Equivalent to Actual System.
The IGV equipment during reliability testing is shown.

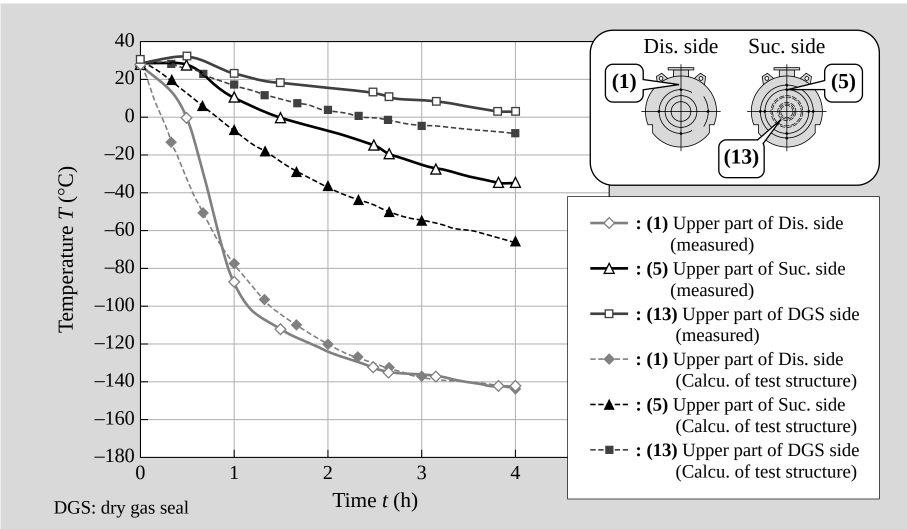

Fig. 10—Temperature Distribution and Change in Suction Casing.
The chart shows the changing surface temperature of the casing during operation comparing with the calculated ones.

side will make the misalignment between the drive shaft and the ring gear axis. The misalignment angle was about 0.1 degrees by the simulation at temperature equilibrium.

With conditions setup in this way to simulate the real equipment including transient changes that will occur at startup and shutdown of the actual operation, an acceleration test was conducted by repeating an 8 seconds operation cycle 10^4 times. Changes in the internal conditions were examined by measuring the stress on the actuator rod during the test but no significant changes were observed. Normal action of rotation and fluctuation of the guide vane were confirmed.

The wear rates at various different parts were checked after the test and were found to be within the allowable range.

CONCLUSIONS

A world-leading improvement in efficiency (3% improvement on previous design) and a significant wider operating range (2.8 times improvement on previous own performance) of the impellers were achieved by optimizing aerodynamic load distribution using CFD. Hitachi has established and utilized its own compressor automated design system covering from compressor model selection including detail design of components to manufacturing drawing. Outcomes from this new development program are already incorporated into this system. The system allows engineers to select a highly efficient compressor with a wide operating range according to the customer’s specifications very quickly. In addition to the aerodynamic design improvements, the IGV technology developed is proved to be efficient energy saving at partial load operation under harsh condition. Taking advantage of these development outcomes to aim the reduction of life cycle cost, Hitachi will provide highly efficient centrifugal compressors with a wide operating range suitable for the oil and gas market in a swift and timely manner.

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