Yoshifumi Nishida
e-mail: yoshifumi.nishida.cx@hitachi-pt.com

Hiromi Kobayashi
e-mail: hiromi.kobayashi.zm@hitachi-pt.com

Hideo Nishida
e-mail: hideo.nishida.cr@hitachi-pt.com

Tsuchiura Research Laboratory,
Research & Development Group,
Hitachi Plant Technologies, Ltd.
603 Kandatsu-machi,
Tsuchiura-shi,
Ibaraki-ken, 300-0013, Japan

Kazuyuki Sugimura
e-mail: Kazuyuki.sugimura.hk@hitachi.com

Department of Advanced Simulation Research,
Mechanical Engineering Center,
Hitachi Research Laboratory, Hitachi, Ltd.
832-2 Horiguchi, Hitachinaka-shi, Ibaraki-ken,
312-0034, Japan

Performance Improvement of a Return Channel in a Multistage Centrifugal Compressor Using Multiobjective Optimization

The effect of the design parameters of a return channel on the performance of a multistage centrifugal compressor was numerically investigated, and the shape of the return channel was optimized using a multiobjective optimization method based on a genetic algorithm to improve the performance of the centrifugal compressor. The results of sensitivity analysis using Latin hypercube sampling suggested that the inlet-to-outlet area ratio of the return vane affected the total pressure loss in the return channel, and that the inlet-to-outlet radius ratio of the return vane affected the outlet flow angle from the return vane. Moreover, this analysis suggested that the number of return vanes affected both the loss and the flow angle at the outlet. As a result of optimization, the number of return vane was increased from 14 to 22 and the area ratio was decreased from 0.71 to 0.66. The radius ratio was also decreased from 2.1 to 2.0. Performance tests on a centrifugal compressor with two return channels (the original design and optimized design) were carried out using two-stage test apparatus. The measured flow distribution exhibited a swirl flow in the center region and a reversed swirl flow near the hub and shroud sides. The exit flow of the optimized design was more uniform than that of the original design. For the optimized design, the overall two-stage efficiency and pressure coefficient were increased by 0.7% and 1.5%, respectively. Moreover, the second-stage efficiency and pressure coefficient were respectively increased by 1.0% and 3.2%. It is considered that the increase in the second-stage efficiency was caused by the increased uniformity of the flow, and the rise in the pressure coefficient was caused by a decrease in the residual swirl flow. It was thus concluded from the numerical and experimental results that the optimized return channel improved the performance of the multistage centrifugal compressor. [DOI: 10.1115/1.4007518]

1 Introduction

Multistage centrifugal compressors are widely used in applications in the oil and gas fields where compressors are operated for long periods, and hence their reliability is very important. For its operation to be cost effective, a compressor is required to exhibit high efficiency and a wide operating range.

To improve the aerodynamic performance of the centrifugal compressor, many investigations on impellers and diffusers have been conducted [1–5]. Previous investigations suggested that higher efficiency can be achieved by improving the blade loading distribution of the impeller in the case of high and medium flow coefficients. Other papers suggested that wedge-type impellers applied in the case of a low flow-coefficient region also provide higher efficiency. Previous investigations on diffusers reported that half-guide vane-type diffusers provided high efficiency in the case of high flow coefficients.

The velocity in the return channel is considerably lower than that in the impeller and diffuser. Therefore, the affect of total pressure loss in the return channel on the overall performance at the design flow rate is relatively small. However, it has been confirmed that the residual swirl flow at the outlet of the return channel leads to insufficient head rise in the next impeller stage [6–8].

Therefore, optimization of the return channel is necessary to minimize the loss in its passage and the residual swirl flow at its outlet.

Hildebrandt [9] optimized the return vane and return bend separately by using a multiobjective optimization method to minimize the loss coefficient. Aalburg et al. [10,11] optimized the return channel using the design of experiment (DOE) by decreasing the diffuser outlet-to-inlet radius ratio while maintaining the return channel performance. Aalburg et al. also experimentally confirmed that by using a diffuser with a smaller outlet-to-inlet radius ratio, the efficiency and power could be further decreased.

However, in the above studies, the residual swirl flow was not considered in the objective functions used in the optimization. Therefore, in this study a multiobjective optimization based on a genetic algorithm was performed to determine the optimum shape of the return channel to minimize the loss in the passage and the residual swirl flow at the outlet of the return channel. The improved performance of the optimum return channel was experimentally confirmed using a two-stage test compressor.

2 Return Channel Optimization System

2.1 Objective Functions and Design Variables. In this study the return vane and meridional shapes were optimized by using the Latin hypercube sampling (LHS), the multiobjective genetic algorithm (MOGA), and the Kriging model [12]. The goal of the optimization was to redesign the shape of the return channel to minimize the loss and to improve the residual swirl flow by

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reducing the flow nonuniformity at the outlet. The following objective functions were selected:

- loss coefficient ζ
- mean flow angle at the return channel outlet $\bar{\alpha}$
- mean circumferential velocity at the return channel outlet $\bar{C}_\theta$
- standard deviation of the flow angle at the return channel outlet $\Delta\alpha$
- standard deviation of the circumferential velocity at the return channel outlet ΔC_θ

Here $\bar{\alpha}$ and $\bar{C}_\theta$ indicate the amplitude of the residual swirl component, and $\Delta\alpha$ and ΔC_θ indicate the flow nonuniformity. Computational fluid dynamics (CFD) was employed to simulate the flow in the return channel. The objective functions were calculated from the simulation results. The MOGA was used to explore the minimum values of the objective functions.

Figure 1 shows the design parameters of the return channel. Table 1 shows the design variables and objective functions in the original design (RCH-ORG). In the optimization of the return channel, the design parameters of the return bend (b_4 , R_h , R_s), the next-stage inlet ($A_{1,next}$), and the shape of the shroud (b_5 , r_5) were fixed. The design variables were the number of vanes (Z), the outlet-to-inlet radius ratio (r_5/r_6), the outlet-to-inlet area ratio (A_5/A_6), the vane angle at the inlet (β_5), and the wrap angle ($\Delta\gamma$). The investigated ranges of the design parameters were as follows:

- $0.5 \leq A_5/A_6 \leq 1.0$
- $13 \leq Z \leq 22$
- $1.8 \leq r_5/r_6 \leq 3.0$
- $14 \leq \Delta\gamma \leq 20$
- $25 \text{ deg} \leq \beta_5 \leq 40 \text{ deg}$.

When the value of r_5/r_6 exceeded 2.18, the return vane was expanded by adding a flat plate, which is shown as the dashed line in Fig. 1. The meridional shape between the next-stage impeller inlet and the return-vane trailing edge on the hub side was drawn as a Bezier curve.

2.2 Flow Chart of the Return Channel Optimization System. Figure 2 shows a flow chart of the optimization system. Here the design goal was to determine the optimum shape for the return channel to minimize all the objective functions. Fifty initial design points were sampled by LHS in the design space defined by the design variables. All the initial design points were evaluated by CFD to acquire the objective functions. A response

Table 1 Design variables and objective functions of the original design and optimized design

	RCH-ORG	RCH-OPT
A_5/A_6	0.71	0.66
Z	14	22
r_5/r_6	2.1	2.0
$\Delta\gamma$ (deg)	19	16.6
β_5 (deg)	55	58.2
$\bar{\alpha}$	6.7	1.5
$\Delta\alpha$	17.8	12.8
$\bar{C}_\theta$	8.6	1.2
ΔC_θ	22.4	16.1
ζ	0.239	0.222

Fig. 2 Flow chart of the return channel optimization system

Fig. 1 Parameters of return channel

Fig. 3 Calculation models

surface that was based on the design variables and objective functions was constructed using the Kriging model. The optimum design was investigated using the MOGA. At the end of the process, the obtained Pareto optimal solutions were analyzed and design candidates were manually chosen. Further CFD simulations were conducted for these candidates, and their results were added to the database. All of these processes were iterated until the changes in the Pareto optimal solutions became small.

2.3 CFD Calculation Method. CFD calculations for return channel optimization were carried out to acquire the objective functions. CFD analysis was performed utilizing the commercial code ANSYS® CFX™ 12.1 (ANSYS® CFX™ 12.1 is a registered trademark of ANSYS, Inc.). The calculation utilized 3D steady Reynolds-averaged Navier–Stokes (RANS) simulations. Air was used as the working fluid and was considered to be an ideal gas. The shear stress transport (SST) turbulence model was used in collaboration with a standard wall function. The total pressure, total temperature, and flow angle were specified as the inlet boundary conditions, and the mass flow rate was specified as the outlet boundary condition.

Our previous experimental results indicate that the outlet flow distribution of the return channel is strongly affected by the return channel bend flow rather than the inlet boundary flow. Therefore, the inlet flow distribution was set as a uniform flow.

Figure 3(a) shows the CFD calculation domain used in return channel optimization, which consisted of a return bend and a return channel of one vane passage. Hexahedral meshes were generated so that Y^+ was approximately 30 at the closest mesh to the wall. The total number of nodes was about 400,000.

2.4 Optimization Results. Figure 4 shows the results of the sensitivity analysis for the design variables and the objective functions. These figures show the contributions from the main effect and the interaction effect of the two design variables as percentages [13]. The results of sensitivity analyses for $\bar{\alpha}$ and $\bar{C}_\theta$ are shown in Figs. 4(a) and 4(b), respectively. According to these figures, the radius ratio r_5/r_6 was found to have the greatest impact on the behavior of $\bar{\alpha}$ and $\bar{C}_\theta$ among the design parameters. Figure 5(a) shows the correlation between $\bar{\alpha}$ and r_5/r_6 and Fig. 5(b) shows the correlation between $\bar{C}_\theta$ and r_5/r_6 . When r_5/r_6 was less than 2.0, $\bar{\alpha}$ and $\bar{C}_\theta$ increased but when r_5/r_6 was much larger, $\bar{\alpha}$ and $\bar{C}_\theta$ became negative; in other words, a reverse swirl flow was generated. The result of sensitivity analysis for $\Delta\alpha$ is shown in Fig. 4(c), which shows that A_5/A_6 was the parameter with the greatest effect on $\Delta\alpha$. The correlation between A_5/A_6 and $\Delta\alpha$ is shown in Fig. 5(c), which suggests that the optimum value of A_5/A_6 is between 0.50 and 0.8. The result of sensitivity analysis for ΔC_θ is shown in Fig. 4(d), which shows that Z was the parameter with the greatest effect on ΔC_θ . The correlation between Z and ΔC_θ is shown in Fig. 5(d), which suggests that ΔC_θ decreased as Z increased. When Z was greater than 23, ΔC_θ decreased, whereas ζ

increased. No clear relation between Z and ζ was observed here. The result of sensitivity analysis for ζ in Fig. 4(e) indicates that A_5/A_6 had the greatest effect on ζ , and Fig. 5(e) indicates that the optimum value of A_5/A_6 is between 0.65 and 0.8.

The optimum design parameters for RCH-OPT were chosen from the Pareto optimal solutions. The optimum design candidates were fulfilled in above-mentioned optimum value of design parameter. The optimized design parameter was adopted for the minimum loss coefficient in the optimum design candidates, and the optimized design parameters were manually chosen. The design variables and objective functions of RCH-OPT and RCH-ORG are shown in Table 1. Figure 6 illustrates the meridional and vane shapes of the return channel in RCH-ORG and RCH-OPT. The optimized design has more vanes than the original design because of the need of guide the flow. Moreover, A_5/A_6 for RCH-OPT was smaller than that for RCH-ORG to reduce the flow velocity in the return channel. $\Delta\gamma$ and r_5/r_6 for RCH-OPT were smaller than those for RCH-ORG. RCH-OPT was superior to RCH-ORG in terms of all of its objective functions. Therefore, these results show that the optimized design contributed to flow redistribution to minimize the loss in the return channel passage.

2.5 Integrated Analysis of Return Channel and Impeller. The inflow to the next-stage impeller was investigated by CFD for RCH-OPT and RCH-ORG. The calculation domain consisted of the return channel with the impeller and the vaneless diffuser as shown in Fig. 3(b). The calculation conditions were the same as those used in the optimization. The total number of nodes was about 680,000.

Performance characteristics of the impeller obtained by CFD are shown in Fig. 7. These performance characteristics were calculated using the mass-averaged total pressure and total temperature at the inlet and exit evaluation planes. The vertical axis indicates the pressure coefficient (ψ/ψ_{ref}), the theoretical pressure coefficient ($\tau_2/\tau_{2\text{ref}}$), and the adiabatic efficiency (η/η_{ref}), each by the value for RCH-ORG. The horizontal axis indicates the flow coefficient. RCH-OPT had a 2.5% higher theoretical pressure coefficient and a 0.8% higher adiabatic efficiency than RCH-ORG. These improvements were caused by the decreased residual swirl flow and the decreased flow nonuniformity. As a result, the pressure coefficient was improved by 3.3%.

3 Performance Tests

3.1 Experimental Apparatus. The performances of the RCH-ORG and RCH-OPT were evaluated using the experimental apparatus shown in Fig. 8, which was composed of a closed loop with air used as the working fluid. A model compressor was driven by a variable-speed motor through gears. Circulating air was cooled by a gas cooler, and the total temperature at the compressor inlet was kept constant at about 293 K by automated control of the coolant flow rate of gas cooler. The total pressure at the compressor inlet was also maintained at about 121.3 kPa (absolute) by injecting or bleeding the working air. The flow rate of the circulating air was controlled by a flow control valve and was measured by an orifice-type flowmeter based on the Japanese Industrial Standard. The pressure difference upstream and downstream of the orifice plate was measured with dynamic pressure sensors to capture fluctuations in the flow such as surging.

A cross section of the test apparatus is shown in Fig. 9. The design flow coefficients of the impellers in the first and second stages were 0.105 and 0.073, respectively. The exit blade width was 17.5 mm for the first-stage impeller and 12.65 mm for the second-stage impeller. For both impellers, the outlet diameter was 0.3 m, the peripheral speed was 300 m/s, the number of blades was 15, and the outlet blade angle was 60 deg.

3.2 Instrumentation and Test Method. The overall performance and individual stage performances were measured. The

total pressure and total temperature were measured using Kiel probes, total temperature probes, and a five-hole pitot tube, respectively. The performance was calculated from the total pressure and total temperature in the suction nozzle, discharge nozzle, and first-stage return channel outlet (second-stage impeller inlet). The flow rate was measured by an orifice flowmeter.

The above-mentioned instrumentation is well established and the estimated uncertainties for relative and repeated measurements at a 95% confidence level were as follows: efficiency $\pm 0.1\%$; pressure coefficient $\pm 0.12\%$; flow coefficient $\pm 0.53\%$.

The mean total pressure and three-dimensional (3D) flow pattern at the first-stage return channel outlet were measured by

five-hole pitot tube traverse equipment set at three circumferential positions in the first-stage return channel outlet. The measurement points are shown in Fig. 10. The five-hole pitot tubes were moved radially and circumferentially.

3.3 Test Results and Discussion. Flow angle distributions at the return channel outlet in one pitch of the return vane measured by the five-hole pitot tubes at the design flow rate are shown in Fig. 11. The flow angle distributions for RCH-OPT and RCH-ORG were similar. The flow angle distribution was uniform in the circumferential direction but not in the radial direction. The outlet

(a) Sensitivity to $\bar{\alpha}$

(b) Sensitivity to $\bar{C}_\theta$

(c) Sensitivity to $\Delta\alpha$

(d) Sensitivity to ΔC_θ

(e) Sensitivity to ζ

Fig. 4 Results of sensitivity analysis

Fig. 5 Relationships between the design variables and objective functions

flow swirled in one direction in the center region and in the reversed direction near the hub and shroud sides. The average flow angle in each case was 6 deg and 10 deg, respectively. This difference in the exit flow angle of 4 deg caused a 2% difference in the theoretical head in the next-stage impeller. Therefore, it is considered that the pressure and efficiency in the second stage were increased by the reduction in the residual swirl flow and the flow nonuniformity.

The CFD calculation results for the flow distribution in the return channel outlet are shown in Fig. 12. The calculated values of $\bar{\alpha}$ were smaller than those obtained experimentally, but the

calculated flow distribution pattern was the same as that observed experimentally. The CFD results showed improved residual swirl flow and flow nonuniformity for RCH-OPT.

The test results for the overall, second-stage, and first-stage performance are shown in Figs. 13–15, respectively. Each value is normalized by the measured value for RCH-ORG.

As shown in Fig. 13, the overall theoretical pressure coefficient of RCH-OPT was 0.7% larger than that of RCH-ORG and the pressure coefficient of RCH-OPT was 1.5% larger than that of RCH-ORG. Moreover, the polytropic efficiency of RCH-OPT was 0.7% higher than that of the RCH-ORG. As shown in Fig. 14, the

Fig. 6 Meridional and vane shapes of return channel

Fig. 7 Performance characteristics of impeller (CFD)

Fig. 8 Compressor test apparatus

Fig. 9 Model compressor cross section

Fig. 10 Pitot tube measurement positions

first-stage efficiency of RCH-OPT was 0.5% higher than that of RCH-ORG. It is considered that this improvement was caused by the reduced loss in the return channel.

As shown in Fig. 15, the theoretical pressure coefficient of RCH-OPT was 2.2% larger than that of RCH-ORG in the second stage, and the pressure coefficient of RCH-OPT was 3.2%

Fig. 11 Measured flow angle distribution in first-stage outlet

Fig. 12 Calculated flow angle distribution in first-stage outlet

Fig. 13 Measured overall performance characteristics

larger than that of RCH-ORG. The polytropic efficiency of RCH-OPT was also 1.0% higher than that of RCH-ORG. The performance test results were similar to the CFD results shown in Fig. 7.

Fig. 14 Measured first-stage performance characteristics

Fig. 15 Measured second-stage performance characteristics

4 Conclusions

1. To improve the shape of the return channel in a centrifugal compressor, the return channel was optimized using a multi-objective optimization method based on a genetic algorithm. The loss coefficient, residual swirl flow, and flow nonuniformity were minimized by optimization. The results of

sensitivity analysis indicated that the loss coefficient was affected by the return vane outlet-to-inlet area ratio, and that the residual swirl flow was affected by the return vane outlet-to-inlet diameter ratio.

2. The results of the performance test indicated that the optimized return channel improved the pressure coefficient by 3.2% and raised the polytropic efficiency by 1.0%.
3. The flow angle distribution was uniform in the circumferential direction, but not in the radial direction. The outlet flow swirled in one direction in the center region and in the reverse direction near the hub and shroud sides. The optimized return channel further improved the residual swirl flow and the flow nonuniformity, compared with the original return channel.
4. The flow distribution in the return channel outlet exhibited a similar distribution in the performance test and the CFD calculation. The performance characteristics determined by the performance test and CFD calculation were in good qualitative agreement.

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Nomenclature

A	= area
b	= width of passage
C_θ	= circumferential velocity (m/s)
D	= diameter in radial direction (m)
H_{ad}	= adiabatic head (J/kg)
H_{th}	= theoretical head (J/kg)
P_s	= static pressure
P_t	= total pressure (Pa)
Q	= suction volume flow rate (m ³ /s)
r	= radius from the rotation axis
T_t	= total temperature (K)
Z	= number of vanes
α	= flow angle from meridional direction (deg)
β	= vane angle from meridional direction (deg)
$\Delta\gamma$	= wrap angle (deg)
ζ	= loss coefficient
η	= efficiency
$\tau = H_{th}/u_2^2$	= theoretical pressure coefficient
$\psi = H_{ad}/u_2^2$	= pressure coefficient
$\bar{C}_\theta$	= mean circumferential velocity
$\bar{\alpha}$	= mean flow angle
ΔC_θ	= standard deviation of circumferential velocity
$\Delta\alpha$	= standard deviation of flow angle

R = radius of return bend (m)
 $\phi = Q/\pi r_2^2 u_2$ = flow coefficient

Subscripts and Superscripts

1	= impeller inlet
2	= impeller outlet
4	= diffuser outlet
5	= return vane inlet
6	= return vane outlet
ad	= adiabatic
h	= hub
next	= next-stage value
pol	= polytropic
ref	= original value
s	= stage inlet, shroud

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