Optimization of a Recirculation Flow Type Casing Treatment with Guide Vanes for Centrifugal Compressors

Binbin Cheng[1,2,3]

1. Graduate School of Engineering, Nagasaki University
Nagasaki, Japan
2. Shanghai Ocean University
Shanghai, China
3. Fishery Machinery and Instrument Research Institute,
Chinese Academy of Fishery Sciences
Shanghai, China
13248172695@163.com

Daisaku Sakaguchi[1]

1. Graduate School of Engineering, Nagasaki University
Nagasaki, Japan
daisaku@nagasaki-u.ac.jp

***Abstract* - A wider operating range as well as higher performance are necessary for turbocharger compressors so as to meet demands of modern engines. Recirculation flow type casing treatment is one of the most effective devices that have been widely investigated. Recirculating flow type casing can extend the operating range of the compressor to the low flow rate region by sucking the reverse flow at shroud wall side at small flow rate condition. The advantages of the recirculation flow type casing treatment are well known as (1) a low energy fluid is sucked out from the down-let slot located near a throat of a rotating impeller (2) flow incidence is reduced because that recirculation flow merges with incoming main flow upstream of the impeller inlet. However, increase in friction loss and mixing loss resulted from the casing treatment can be considered as the disadvantages of the casing treatment. In the present study, an application of three-dimensional guide vane with 14 blades inside the casing treatment is presented to overcome the disadvantages of the casing treatment. And it shows improvement compared with the guide vanes with 7 blades. The shape of the guide vanes were optimized by using a surrogate model assisted optimization code. Numerical results showed that the optimized guide vanes hold a possibility to improve off-design performance by reducing the excessive pre-whirl at the compressor inlet.**

Keywords - Global optimization, Centrifugal Compressor, Turbocharger, Casing treatment, Flow range enhancement

I. INTRODUCTION

Application of the turbocharging technology in the internal combustion engine is spread widely in various areas such as automobile, locomotive and marine vehicles. Turbocharger could reduce engine emission, increase fuel economy, generate more power, hence downsize the engine. Thus high pressure ratio and wide operating range become indispensable for turbocharger compressor to meet various engine load conditions. However, it challenges to improve static pressure rise while maintaining adiabatic efficiency under various load conditions. When the mass flow rate becomes smaller, the unstable flow occurs because of the flow separation in the impeller and diffuser and reverse flow on the shroud wall, finally resulting in surge.

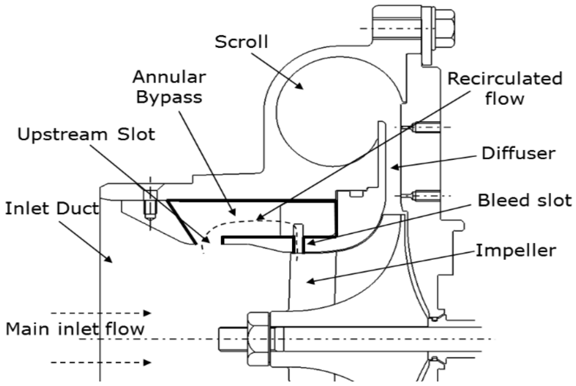
A casing treatment makes it possible to improve the near surge characteristics and extend the operating range in a turbocharger compressor. A casing treatment generally consists of a bleed slot, a guide vane, an annular bypass and an upstream slot fabricated into the compressor housing. When the compressor is operated near surge margin, the reverse flow with low momentum on the shroud wall side is sucked into the bleed slot and annular bypass to the upstream of the compressor inlet through the upstream slot. A casing treatment was investigated firstly by Fisher in 1989 [1]. In 2009, Sakaguchi [2] proposed the effect of the recirculation casing treatment with or without guide vane inside. It is found that the multi-groove with the half height guide vane is effective for increasing the mass flow at the shroud side of the impeller inlet. Moreover, the flow incidence distortion could be improved successfully in whole flow rate range. Recently non-axisymmetric casing treatment was investigated by B. T. Wang [3] which improves the asymmetric flow field using the insert ring to partially impact the counter-swirl flow at compressor inlet. The insert ring feature in the upstream position of casing treatment is a kind of guide vanes that is used to gain more surge margin as proposed by Yamaguchi [4]. Authors studied the application of surrogate-model assisted global optimization on turbomachinery components recently. Reference [5] shows the improvement in adiabatic efficiency of the optimized recirculation flow type casing treatment over conventional one which is used in a small turbocharger compressor. It was found that recirculated flow rate is one of the dominant parameters for improving adiabatic efficiency. In the study [6] of the optimization of casing treatment for the compressor used in current study, it was numerically showed that the improvement in near surge condition. Authors [7] also studied the optimization of guide vane in recirculation flow type casing treatment. A large circumferential component of velocity of the recirculation flow leads to the velocity distribution at the impeller inlet fairly distorted especially at the shroud side when the casing treatment is applied to a high speed centrifugal compressor. Too large pre-whirl makes decrease of pressure rise in the impeller, and results in a smaller recirculation flow rate. Pre-whirl can be reduced by applying a proper shape of guide vanes and off-design performance could be improved. However, the shape of guide vane in Reference [7] was optimized in two-dimension and the shape remained same in radial direction. Min Thaw Tun and Sakaguchi [8] have compared characteristics of compressor between the cases of without guide vane and optimized guide vane which has 7 blades. Application of the optimized guide vane shows better performance in adiabatic efficiency and static pressure rise by suppressing the excessive pre-whirl at impeller inlet at off-design condition.

In the present study, the comparison between the guide vane with 7 blades and the guide vanes with 14 blades was taken. Global optimization was performed on three-dimensional and the guide vanes were installed in the upstream section of the recirculation flow type casing treatment. The numerical analysis showed improvement in velocity distribution and flow incidence at impeller inlet, hence improve adiabatic efficiency and static pressure rise since the excessive pre-whirl at shroud wall side, that is carried by the recirculated flow, has been successfully suppressed by the optimized guide vanes with 14 blades.

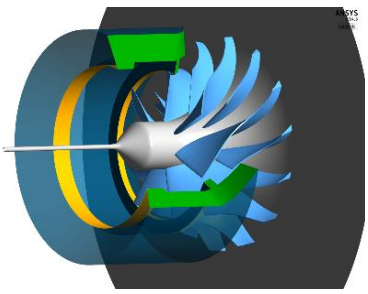
II. Basic Concept of Casing Treatment

A. Baseline Test Compressor

A centrifugal compressor with casing treatment designed by IHI Corporations was used as the baseline geometry for this optimization and end wall schematic diagram of the test compressor is shown in Fig. 1. The test compressor consists of an inlet duct, a bleed slot with annular bypass, an upstream slot, an impeller with splitter blades, a vaneless diffuser, and a non-axisymmetric scroll (Fig. 1). The compressor is from a turbocharger used on a diesel engine. The geometry details of compressor have not been exposed for reasons of commercial confidentiality. The baseline recirculation flow type treatment has been performed to improve the performance of the compressor at near surge flow rate.



(a) Schematic of end wall



(b) 3D model

Fig. 1 Test centrifugal compressor with casing treatment

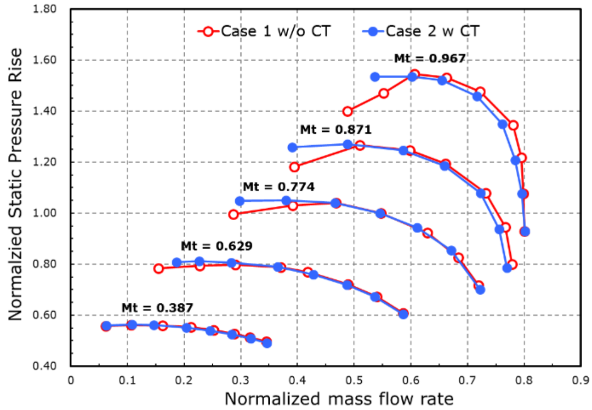


Fig. 2 Comparison of pressure ratio characteristic of the baseline compressor with and without casing treatment (Experiment)

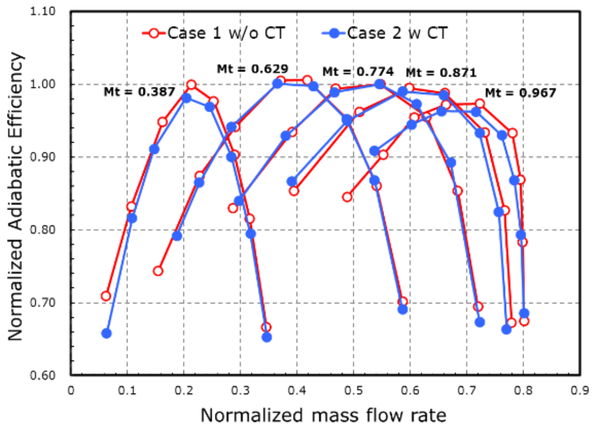


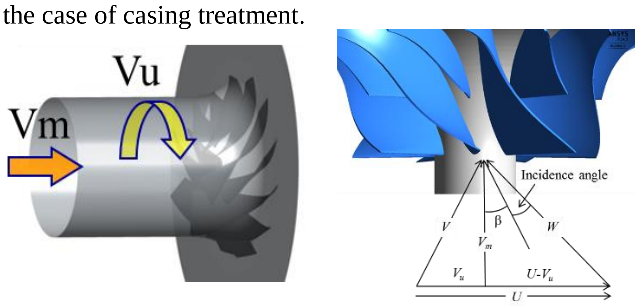
Fig. 3 Comparison of adiabatic efficiency characteristic of the baseline compressor with and without casing treatment (Experiment)

The experimental analysis of baseline centrifugal compressor between the cases with and without casing treatment was showed in Fig. 2 and Fig. 3. The analysis was performed at five different speeds of impeller. It can be seen that the pressure characteristics of the compressor are compared between the cases with and without casing treatment in Fig. 2. In these comparisons, the static pressure rise and efficiency were normalized by respective ones at the design flow rate (G/G^* = 0.479) when the impeller inlet tip's Mach number (M_t) equals 0.774, whereas the mass flow rate (G) was normalized by the choke mass flow rate (G^*) at the compressor inlet. As we can see that the casing treatment for baseline compressor successfully improved the static pressure rise at near surge flow conditions with increase in the impeller rotation speeds (M_t = 0.774, M_t = 0.871 and M_t = 0.967) without serious deterioration at other flow rate conditions (Fig. 2). However, there is no considerable surge improvement in the condition of low speed.

The efficiency characteristics (Fig. 3) shows same characteristics with small deterioration of adiabatic efficiency under various mass flow rates and rotational speeds. However, in the case with casing treatment, small improvement in adiabatic efficiency can be seen at two high impeller speeds (M_t = 0.871 and M_t = 0.967) at near surge flow rate. However, in both cases, the major drawback of the high speed turbocharger centrifugal compressor was significant drop in adiabatic efficiency at near surge condition.

B. Technical Issue of the Casing Treatment

The recirculation flow type casing treatment shows improvement of static pressure rise near surge. However, it remains that technical issue of an efficiency drop. Applying the casing treatment to a centrifugal compressor would increase in friction loss and mixing loss. Increase in such losses is the reason for drop in the compressor efficiency in the case of casing treatment.

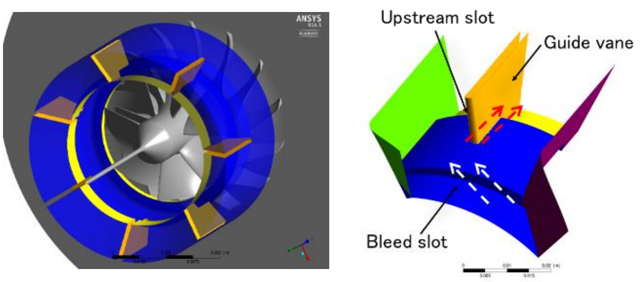


(a) Meridional and circumferential velocity components

(b) Velocity triangle at impeller inlet

Fig. 4 Meridional and circumferential components of velocity in a centrifugal compressor

Another technical issue is high pre-whirl motion by recirculated flow. Figure 4 (a) shows the meridional (V_m) and circumferential (V_u) components of velocity looking from the stationary frame of reference. The circumferential components of velocity (V_u) before the impeller inlet is defined as the pre-whirl. At design operation point, the flow, relative velocity (W), approaches the blade leading edge with zero or close to zero incidence. However, at off-design conditions, the flow approaches the blade leading edge at an incidence angle as shown in Fig. 4(b). In order to reduce the large pre-whirl or incidence angle, guide vanes were installed inside the annular bypass at the upstream slot section as shown in Fig. 5. The detailed explanations about the reasons for the use of guide vanes inside the casing treatment were presented in previous research work in Reference [7]. However, the complicated flow phenomena inside the casing makes huge difficulty to design a guide vane with a large number of design of freedom.



(a) 3D Model Compressor Model with guide vane inside casing treatment

(b) Redirection of flow inside the casing treatment using guide vane

Fig. 5 Application of guide vanes inside the annular bypass

III. Research Methodology

A. Surrogate Model Assisted Optimization Algorithm

Figure 6 shows the flowsheet of surrogate model assisted optimization algorithm which is used in this research work. Firstly, the design variables such as optimization parameters and dependent parameters, objective functions and constraints are defined. Design of experiments (DOE) is a step which makes the optimization parameters analyze the data better with appropriate statistical methods. Design of experiment helps the designer determine which variable is the most influential variable on the objective functions and constraints and determine where to set the influential design variables. After that, numerical or experimental analysis has been performed based on the set of design variables distributed by the design of experiments (DOE). In this research work, ANSYS CFX is used to analyze the performance of the centrifugal compressor. Surrogate model is used to reduce the expensive numerical simulation cost and time. Surrogate model provides less accurate but very fast performance predictions to evaluate thousands of design variables sets. In this research work, artificial neural network (ANN) is chosen for the surrogate model. The database collected from the analysis phase is used to construct a surrogate model or in the case of artificial neural network, the collected database is used to train the neural network. Genetic Algorithm - Differential Evolution (DE) model, one of the most popular optimization techniques, has been chosen to predict the sets of design variables which hold the desired performance. Differential evolution (DE) generates thousands sets of design variables and trained surrogate model (ANN) predicts their performance. These processes can be seen as the counter clockwise inner loop in the flow chart (Fig. 6). Differential evolution (DE) selects the elite sets of design variables which are sent back to analysis phase (CFD Simulation in this case) to get accurate performance results. These processes are expressed as updating, the clock wise outer loop in the flow chart (Fig. 6). The results of the accurate performance analysis are added to the database and a new optimization loop is started after a new training of the surrogate model (ANN) on the enlarged database.

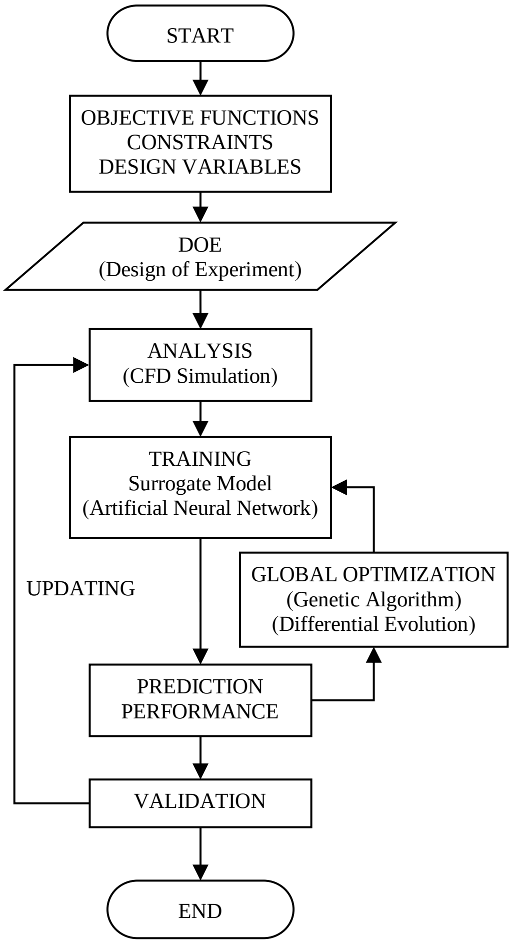


Fig. 6 Surrogate model assisted optimization algorithm

B. *Objective Functions and Design Variables*

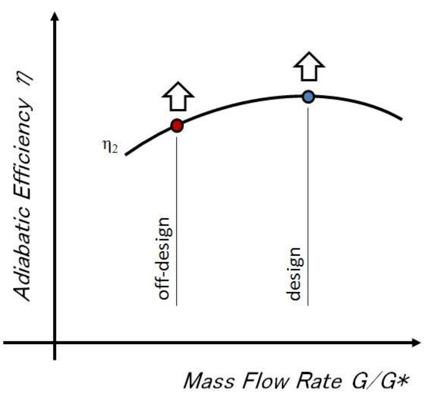


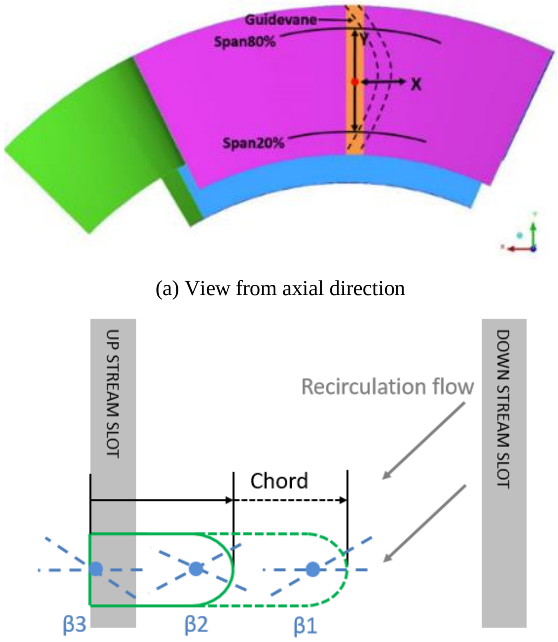
Fig. 7 Objective Functions

The objective functions in this research work was aiming to improve the adiabatic efficiency at two operating points at design mass flow rate ($G/G^* = 0.474$) and at off-design mass flow rate ($G/G^* = 0.258$) under same impeller tip speed ($M_t = 0.774$) as shown in Fig. 7. The optimization is performed as minimizing problem which means that objective functions are treated as negative quantities as shown in Eqn. (1). In order to control the three-dimensional shape of the guide vane, eleven design variables are defined as shown in Fig. 8. Eleven design variables are angular (bow intensity) (X) and spanwise (Y) positions for mid-section stacking point, three blade angles (β_1, β_2 and β_3) of guide vane and six parameters of the casing treatment (p_1-p_6).

Objective

$$\text{Minimize } -\eta_{ad} \text{ at } G/G^* = 0.474 \qquad \textbf{(1)}$$

$$\text{Minimize } -\eta_{ad} \text{ at } G/G^* = 0.258$$



(a) View from axial direction

(b) Blade to blade view

Fig. 8 Definition of design variables

IV. Results and Discussion

A. *Optimization Results and Three-dimensional Guide Vanes*

The result of the optimization is presented as the two-dimensional objective space in Fig. 8. The two objective functions, adiabatic efficiency at design flow rate (G/G^*=0.474) and at off-design flow rate (G/G^*=0.258), are presented as a minimizing problem which means that both parameters are improving approaching to the lower left hand corner. The efficiencies are normalized by adiabatic efficiency of baseline compressor with casing treatment at design flow rate (G/G^*=0.474). Each circle mark indicates one individual design which has been analyzed by numerical simulation. The white circle marks are generated during the design of experiment (DOE) phase while the red ones are generated during the optimization loop selected by the differential evolution (DE) based on the prediction of surrogate model (ANN). Two individuals Opt1 (green) and Opt2 (blue) lie on the Pareto front where one objective cannot be improved without worsening the others were selected based on the maximum adiabatic efficiency at design and off-design flow rate conditions respectively. The selected optimized guide vane designs, as indicated in Fig. 9, are illustrated in Fig. 10 and design variables are expressed in Table 1.

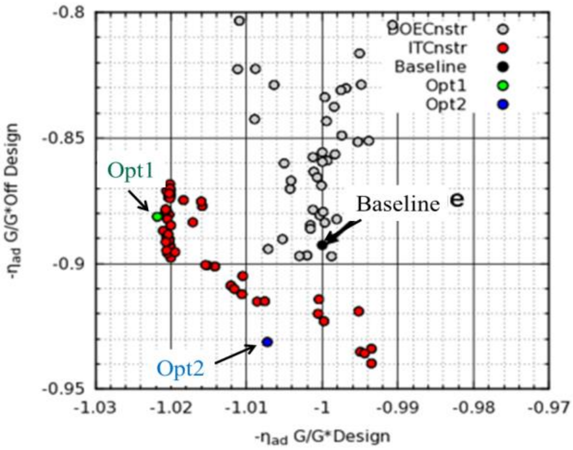
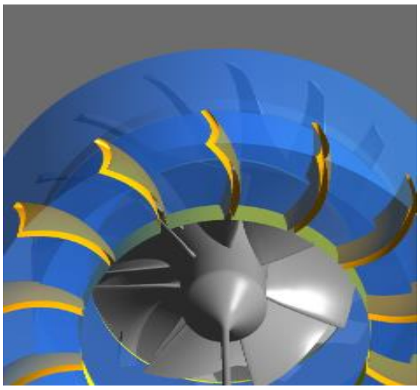
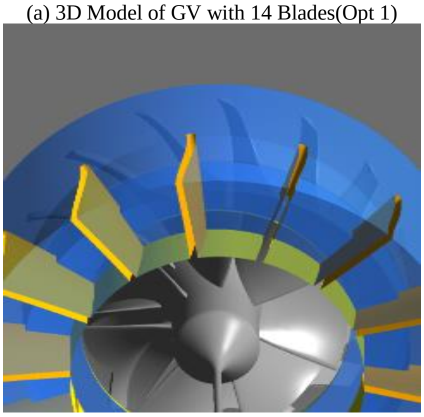


Fig. 9 Two-dimensional objective space for global optimization



(a) 3D Model of GV with 14 Blades(Opt 1)



(b) 3D Model of GV with 14 Blades(Opt 2)

Fig. 10 Optimized guide vanes

TABLE I

DESIGN VARIABLES FOR OPTIMIZED GUIDE VANES (14 BLADES)

Design Parameters	Opt1	Opt2
X position	-0.078	-0.003
Y position	0.516	0.202
β_1 [deg]	21.1	53.4
β_2 [deg]	-9.5	-29.9
β_3 [deg]	-30	-0.1
Upstream slot width (p_1)	0.128	0.233
Upstream slot position (p_2)	1.930	1.895
Downstream slot width (p_3)	0.016	0.016
Downstream slot position (p_4)	0.803	1.014
Bypass height (p_5)	0.226	0.226
Bypass position (p_6)	0.717	0.716

B. *Performance of Optimized Guide Vanes*

The comparison of characteristics of compressor between the optimized guide vane with 7 blades and the optimized guide vanes with 14 blades has been performed based on the numerical simulation. The adiabatic efficiency and static pressure rise are compared between optimized guide vanes with 14 blades and the guide vane with 7 blades as shown in Fig.11, whereas work input coefficient and recirculated mas flow rate in Fig. 12. When the mass flow rate becomes larger than the value of the design point, both adiabatic efficiency and static pressure rise are successfully improved in the cases of optimized guide vanes with 14 blades (Fig. 11) while recirculated mass flow rate (m_{IR}) of the guide vanes with 14 blades become better than that of 7 blades when the mass flow rate is larger than the value of design point (Fig. 12). Work input coefficient also shows improvement in the cases with optimized guide vanes over the entire mass flow rate which means that the impeller imparts more energy to the working fluid in the cases with 14 blades (Fig. 12).

The flow characteristics upstream of the compressor inlet has been investigated at off-design condition (G/G^*=0.258). Meridional (V_m) and Circumferential (V_u) velocity components measured 5mm before the impeller leading are compared in Fig. 13 between the cases with 17 blades and 7 blades. In the case with 14 blades opt2, circumferential velocity (V_u) shows a better distribution and the pre-whirl at the shroud wall side has successfully been suppressed. Hence, the intensity of flow incidence has been reduced in the case with 14 blades opt2 as shown in Fig. 14

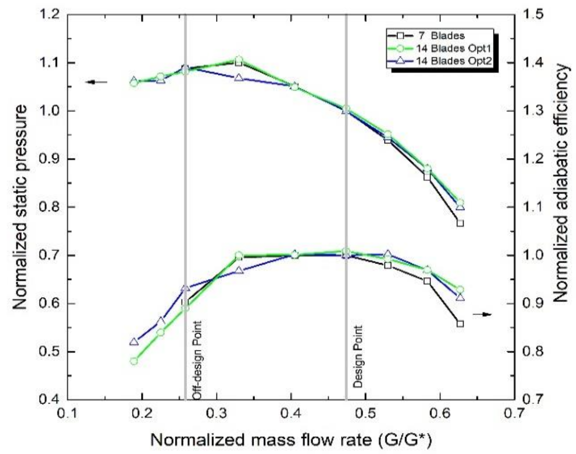


Fig. 11 Efficiency and static pressure rise characteristics

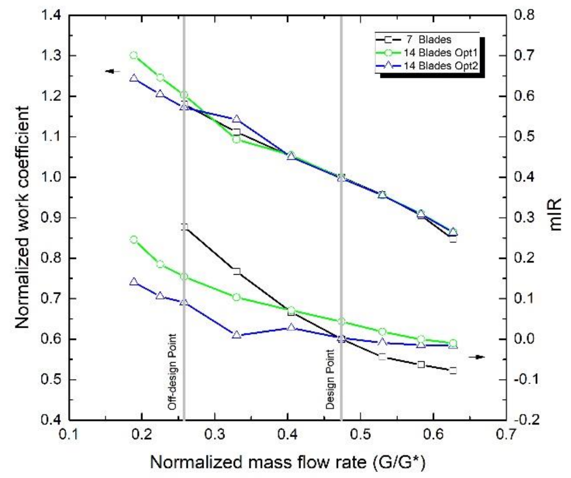


Fig. 12 Work coefficient and recirculated mass flow characteristics

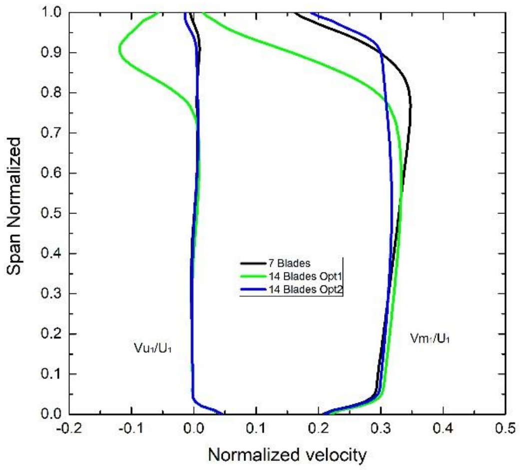


Fig.13 Span-wise meridional and velocity at off-design condition (G/G*=0.258)

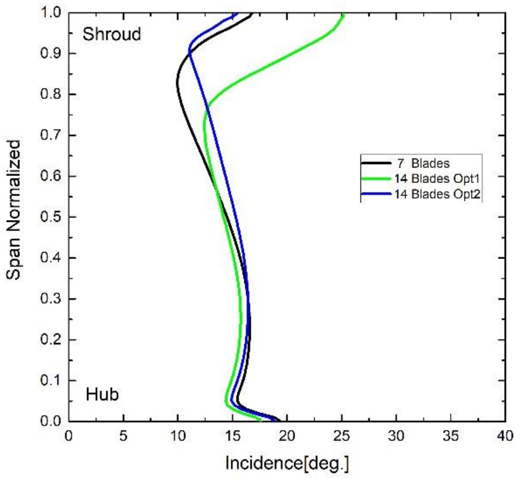


Fig. 14 Flow incidence at off-design condition (G/G*=0.258)

V. Conclusions

The characteristics of the centrifugal compressor for turbocharger between the guide vanes with 14 blades and with 7 blades has experimentally been analyzed and compared. Three-dimensional parameterization of guide vanes inside the recirculation flow type casing has been improved. Surrogate model assisted optimization has been performed and optimized shape of the guide vanes were selected. Application of the optimized guide vane in the case with 14 blades opt2 shows improvement in adiabatic efficiency and static pressure rise by suppressing the excessive pre-whirl at impeller inlet at off-design condition.

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