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Study on Performance Improvement and Flow Range Enhancement in Radial Compressors
（遠心圧縮機における性能改善と運転流量範囲拡大に関する研究）

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LIST OF SYMBOLS

Nomenclature

\[ b \] Diffuser span height \([\text{m}]\)

\[ b \] Bias Unit in a Neural Network \([-]\)

\[ C_L \] Lift coefficient \([-]\)

\[ C_S \] Normalized static pressure \([-]\)

\[ F_L \] Lift force \([\text{N}]\)

\[ f \] Objective Function \([-]\)

\[ G \] Mass flow rate at outlet \([\text{kg/s}]\)

\[ G^* \] Choke mass flow rate at inlet \([\text{kg/s}]\)

\[ g_j \] Constraint Functions: Inequality Constraints \([-]\)

\[ h_j \] Constraint Functions: Equality Constraints \([-]\)

\[ i_{in} \] Input Layer in a Neural Network \([-]\)

\[ \Delta h_0/U_2^2 \] Work Coefficient, \(\Delta h_0 = (U_2^*V_{a2} - U_1^*V_{a1})\) \([-]\)

\[ l \] Chord length \([\text{m}]\)

\[ m_{IR} \] Normalized recirculation flow rate in the annular bypass \([-]\)

\[ M_i \] Mach number at inlet impeller tip \([-]\)

\[ o_{out} \] Output Layer in a Neural Network \([-]\)

\[ p_1 \] Upstream slot width \([\text{mm}]\)

\[ p_2 \] Upstream slot position \([\text{mm}]\)

\[ p_3 \] Downstream slot width \([\text{mm}]\)

\[ p_4 \] Downstream slot position \([\text{mm}]\)

\[ p_5 \] Annular bypass width \([\text{mm}]\)

\[ p_6(r) \] Annular bypass position \([\text{mm}]\)

\[ P_s, p \] Static pressure \([\text{Pa}]\)

\[ P_T \] Total pressure \([\text{Pa}]\)

\[ Q \] Flow rate \([\text{m}^3/s]\)

\[ r \] Radius \([\text{m}]\)

\[ R \] Radius ratio \([-]\)

\[ R_{LSD} \] Radius ratio of the leading edge of LSD \([-]\)

\[ RPM \] Revolutions per minute \([-]\)

\[ TF \] Transfer Function in a Neural Network \([-]\)

\[ U \] Impeller speed or Tangential velocity \([\text{m/s}]\)

\[ V_{rep} \] Vector averaged velocity \([\text{m/s}]\)

\[ V \] Absolute velocity \([\text{m/s}]\)

\[ w \] Weight Coefficient in a Neural Network \([-]\)

\[ \vec{x} \] Design Variables \([-]\)

\[ x \] Input to a Neural Network \([-]\)
\( x_l \) Side Constraints: Lower Limit [-]
\( x_u \) Side Constraints: Upper Limit [-]
\( y \) Output from a Neural Network [-]

**Greek letters**

\[ \alpha \] Absolute velocity flow angle [°]
\[ \alpha_A \] Angle of attack [°]
\[ \beta_b \] Blade angle [°]
\[ \gamma \] Stagger angle [°]
\[ \theta \] Camber angle [°]
\[ \phi \] Flow coefficient [-]
\[ \sigma \] Solidity (chord length / pitch of blade) [-]
\[ \psi_{se} \] Static pressure coefficient at diffuser exit [-]
\[ \psi_{s2} \] Static pressure coefficient at impeller exit [-]
\[ \psi_d \] Static pressure coefficient at diffuser [-]
\[ \rho \] Density of fluid [kg/m³]
\[ \eta \] Efficiency [-]

**Subscripts**

0 Reference point (Plenum tank) or Compressor Inlet Duct
1 Impeller inlet
2 Impeller exit
4 or e Diffuser exit
ad Adiabatic
CT Casing treatment
LE Leading edge of the LSD blade
m Meridional component
ref Reference Point
TE Trailing edge of the LSD blade
u Circumferential component
LIST OF PUBLICATIONS


(9) Daisaku Sakaguchi, Min Thaw Tun, Ryusuke Numakura and Baotong Wang, “Global Optimization of Recirculation Flow Type Casing Treatment in Centrifugal Compressors of Turbochargers”, *Journal of Mechanical Engineering Science, Part C: Special Issue*, (accepted), (2016.11) (Chapter 5)


INTRODUCTION

1.1 Background, Scope and Objectives

Earlier development in turbomachines was mainly based on the designer’s experience and experimental analysis. Prototypes are manufactured and tested, and possible improvement are proposed based on the newly acquired knowledge. These processes are repeated until prototypes reaches satisfactory improvement or desired performance level. Over the last three decades or so the speed of the computers has increased by about 1.5 per year. The cost has also fallen, so that the speed divided by cost doubles every year. As developing with information technology, computational tools such as Computational Fluid Dynamics (CFD) and Computational Structural Mechanics (CSM) became an alternative for designers and manufacturers to the time consuming and very expensive tests in experiment facilities. Designs can be analyzed by CFD and CSM at any time within a few hours of computation time. Repetitive nature of the design, nevertheless, remains the same.

Turbomachines employing centrifugal effects for increasing fluid pressure have been in use for more than a century. Radial compressors find the most widespread use of any type. Several ideas and design concepts has been implemented for performance improvement and map width enhancement in radial compressors. Increase in computational power allows designer to study design with higher degree of design variables and provides higher accuracy. Complicated design procedure with higher accuracy and iterative nature of turbomachines design process demand again higher computational power, time and cost. An application of multi-objectives optimization or global optimization, recently, has been attained the attention of researchers and manufactures. Multi-objectives optimization helps in searching the optimized design variables in design space. Optimization technique such as genetic algorithm or response surface method, however, demands thousands of combinations of design variables and
their target results to predict the optimized design variables. One idea to overcome the enormous amount of numerical or experimental analyses is an application of surrogate-model which provides faster prediction and lower cost, however, less accurate than traditional experimental or computational numerical analysis. Nevertheless, manufacturing and experimental analysis of the final design or prototype is still required to compare CFD and CMS predictions.

The main theme of this thesis is passive control method which focuses on stationary components of radial compressors such as casing treatment or diffuser rather than main component, impeller. This idea allows us to conduct researches without facing much difficulties in manufacturing the prototype. Improve manufacturing technology such as 3D printing technology makes possible to produce prototypes and test materials under academic research facilities.

The objective of this thesis is to study performance improvement and flow rate range enhancement in radial compressors. These are done in one centrifugal blower and two centrifugal compressors. In the first study, flow range enhancement of a centrifugal blower has been done using the low solidity circular cascade diffuser (LSD). In the second study, both performance improvement and flow range enhancement of a centrifugal compressor for small turbocharger has been carried out by an optimized recirculation flow type casing treatment. In the third study, optimization of recirculation flow type casing treatment in a centrifugal compressor for commercial turbocharger and off-design performance improvement by optimized guide vanes inside the casing treatment has been performed.

1.2 Outline

This thesis contains seven research works. These are
1) Flow Range Enhancement by Secondary Flow Effect in Low Solidity Circular Cascade Diffusers
2) Optimization and Validation of Secondary Flow Effect for Flow Range Enhancement in a Low Solidity Circular Cascade Diffuser
3) Study on the Performance Improvement of Centrifugal Compressors for Small Turbocharger
4) Study on Flow Range Enhancement of Centrifugal Compressors for Small Turbocharger
5) Study on Guide Vane in Recirculation Flow Type Casing Treatment by Applying Global Optimization Technique
6) Study on Performance Improvement in Centrifugal Compressors by a Global Optimization Approach and
7) Off-design Performance Improvement in Centrifugal Compressors with Recirculation Flow Type Casing Treatment by Optimized Guide Vane.
1.2.1 Chapter 2 Research Methodology

This chapter presents overview of research methodology used in this thesis. In this chapter, idea of optimization system, various optimization technique, meta-modeling, overview of artificial neural network, overview of genetic algorithm and optimization system used throughout in this thesis are discussed.

1.2.2 Chapter 3 Study on Flow Range Enhancement of Centrifugal Blower

Study on flow range enhancement of a low speed centrifugal blower is presented in Chapter 3. A Low Solidity circular cascade Diffuser (LSD) in a centrifugal blower is successfully designed by means of multi-objective optimization technique. The optimization is aiming at improving the static pressure coefficient at design point and at low flow rate condition while constraining the slope of the lift coefficient curve. Moreover, a small tip clearance of the LSD blade was applied in order to activate and to stabilize the secondary flow effect at small flow rate condition. The optimized LSD blade has an extended operating range of 114% towards smaller flow rate as compared to the baseline design without deteriorating the diffuser pressure recovery at design point. The diffuser pressure rise and operating flow range of the optimized LSD blade are experimentally verified by overall performance test. The detailed flow in the diffuser is also confirmed by means of a Particle Image Velocimeter. Secondary flow is clearly captured by PIV and it spreads to the whole area of LSD blade pitch. It is found that the optimized LSD blade shows good improvement of the blade loading in the whole operating range, while at small flow rate the flow separation on the LSD blade has been successfully suppressed by the secondary flow effect.

1.2.3 Chapter 4 Study on Performance Improvement and Flow Range Enhancement of a Centrifugal Compressor of a Small Turbocharger

This chapter presents three research work studying on performance improvement and flow range enhancement for a centrifugal compressor of a small turbocharger. High-pressure ratio and wide operating range are highly required for a turbocharger in diesel engines. A recirculation flow type casing treatment is effective for flow range enhancement of centrifugal compressors. Two ring slots on a suction pipe and a shroud casing wall are connected by means of an annular passage and stable recirculation flow is formed at small flow rates from the downstream slot toward the upstream slot through the annular bypass. The shape of baseline recirculation flow type casing is modified and optimized by using a multi-point optimization code with a metamodel assisted evolutionary algorithm embedding a commercial CFD code CFX from ANSYS. Objective functions in the first research are to improve adiabatic efficiency of the compressor at two different operating points, the design mass flow rate and near surge
mass flow rates. Objective functions in the second research are to improve adiabatic
efficiency of the compressor at the design mass flow rate and to enhance the surge margin.
The numerical optimization results give the optimized design of casing with improving
adiabatic efficiency in wide operating flow rate range. Sensitivity analysis of design
parameters as a function of efficiency has been performed. It is found that the optimized
casing design provides optimized recirculation flow rate, in which an increment of
entropy rise is minimized at slots and passages of the rotating impeller. In the third
research work, global optimization of two-dimensional guide vanes inside the
recirculation flow type casing treatment has been performed. This research shows that
there is a possibility to improve the adiabatic efficiency at small flow rate conditions.
Excessive pre-whirl based on recirculation flow is suppressed by the optimized guide
vane, and velocity distortion at impeller inlet has improved. It is found that improvement
of the efficiency in the case of the optimized guide vane is supported by proper
distribution of flow at impeller inlet.

1.2.4 Chapter 5 Study on Performance Improvement of a Centrifugal Compressor of a
Commercial Turbocharger

This chapter presents two research work studying on performance improvement
for a centrifugal compressor of a commercial turbocharger. A full compressor
characteristics comparisons between the cases with and without casing treatment has
experimentally been made. In the first attempt, a global optimization of a recirculation
flow type casing treatment has been done. The optimization aims to improve the adiabatic
at design flow rate condition and off-design flow rate conditions. Two individual designs
out of the several candidates on the Pareto-front has been selected based on the maximum
adiabatic efficiency at design point and off-design point. The selected individuals show
the improvement in adiabatic efficiency for entire flow rate range without static pressure
rise penalty. In the second attempt, three-dimensional parameterization of guide vanes
inside the recirculation flow type casing has been introduced. Surrogate model assisted
optimization has been performed and optimized shape of the guide vane was selected.
Application of the optimized guide vane shows improvement in adiabatic efficiency and
static pressure rise by suppressing the excessive pre-whirl at impeller inlet at off-design
condition.

1.2.5 Chapter 6 Conclusions and Perspective

This chapter summarizes the research works, discusses about major achievements,
points out important design variables and parameters for performance improvement and
flow range enhancement and describes future research directions which has potential to
improve further in designing the radial compressors.
RESEARCH METHODOLOGY

2.1 Overview

In this chapter, research methodology used in this thesis is mainly discussed. In the first place, optimization used in every research work is presented. A current trend in turbomachinery component design is to rely more and more on predictions from numerical tools in the design process. Where in the past components are designed by using rough estimation and simple correlations coming from theoretical consideration and experimental experience, nowadays computational tools such as Computational Fluid Dynamics (CFD) and Computational Structural Mechanics (CSM) are well integrated into the design process. However, the expense of computation tools is considerably high in the case of optimization of turbomachinery components as it is required a lot of trial designs which is combined with many parameters and it is difficult to confirm the optimized one. A best choice to meet these requirements, therefore, is a multi-objective and multi-point optimization technique.

In second place, an application of particle imaging velocimetry is presented. Particle Imaging Velocimetry (PIV) is a powerful measurement technique, which can be used in a wide range of research applications. The instantaneous planar velocity measurements obtained with PIV make it an attractive technique for use in the study of complex flow fields encountered in turbomachinery. Technique for optical access, light sheet delivery, CCD camera technology and particulate seeding are discussed. Results from the successful application of PIV technique to diffuser blade pitch of a centrifugal blower are presented. Experimental validation of the performance of new diffuser blade of centrifugal blower has been done by PIV measurement technique. Forming of secondary flow in the diffuser of centrifugal blower at small flow rate condition has been successfully captured by PIV measurement technique.
2.2 What is Optimization?

Optimization can be defined in different ways. Some define it as “the art of making things the best.” Definition, however, may not be reasonable, or even possible, to do something in the very best possible way. In practice, doing something as well as possible within practical constraints is very desirable. Designing a product and doing all we can to increase profit as much as is practically possible is also very desirable. Optimization provides us with the means to make things happen in the best possible practical way. Figure 2.1 illustrates the design process using (i) traditional design approaches, and (ii) using optimal design approaches.

Box A displays the input, which includes two basic issues. The first defines dream design: the desired performance levels (e.g., maximize efficiency, minimize mixing loss) and any constraints (e.g., slope of coefficient of lift curve less than 0). The second item provides an initial design that we can obtain through any conventional means. That design is simply a starting point from which improvements can be made.

Box B illustrates the analysis phase. Analysis usually gives what the output result is for a given set of input conditions under specific consideration such as numerical simulation or structural simulation.

Box C explains how the optimization process cycle begins. Using the initial design that is provided, it is very unlikely that it will satisfy all the constraints and maximize performance. Most likely, it will need to be improved by modifying it as intelligently as possible. This is where the power of optimization comes into play. This leads to the next box.
**Box D** is where the design is revised and improved in a very systematic way. In the process of optimization, Box D continually modifies the design with the expectation that the design will improve. Each modification is submitted to the analysis module. After the analysis is performed, the design performance is again evaluated to see whether it meets the objectives. If it does, the optimization is finished. If it does not, it will loop one more time.

**Box E** illustrates the human element involved in making the required improvements in a traditional way. In other words, Box E replaces Box D. Human design decision making is, however, critically needed.

### 2.3 Definition of Optimization problem

A general optimization problem [2] can be formulated using the following set of equations:

Minimize: \( f(\vec{x}) \)  
Subject to: \( g_j(\vec{x}) \leq 0 \), \( j = 1...m \)  
\( h_k(\vec{x}) = 0 \), \( k = 1...n \)  
\( x_l \leq \vec{x} \leq x_u \)  

Where \( \vec{x} = \begin{pmatrix} x_1 \\ x_2 \\ \vdots \\ x_n \end{pmatrix} \)

#### 2.3.1 Design Variables (\( \vec{x} \))

Design variables \( \vec{x} \) are entities that identify a particular design. In the search for the optimal design, the entities will change over a prescribed range. Design variables \( \vec{x} \) contain the parameters of design that can be modified, i.e. the chord length of an airfoil, stagger angle, blade angle, etc. The values of a complete set of these variables characterize a specific design. The number and type of entities belonging to this set are very important in identifying and setting up the quantitative design problem. It is essential that this choice capture the essence of the object being designed and at the same time provide a quantitative characterization of the design problem.

#### 2.3.2 Objective Function \( f(\vec{x}) \)

The function \( f(\vec{x}) \) represents the objective function to minimize. The traditional design optimization problem is defined using a single objective function. The format of this statement is usually to minimize or maximize some quantity that is calculated using some design function. This function must depend, explicitly or implicitly, on the design
variables. In turbomachinery optimization, this function can be adiabatic efficiency, pressure rise, strength or a combination of them. The optimization idea is that the design is improved by minimizing this function. The function that need to be maximized such as efficiency or strength can always be altered into a minimizing problem by taking the inverse value.

2.3.3 Constraint Functions

As design functions, these will be influenced by the design variables. The format of these functions requires them to be compared to some numerically limiting value that is established by design requirement, or the designer. This value remains constant during the optimization of the problem. A well-described design problem is expected to include several such functions, which can be represented as a vector. The comparison is usually set up using the three standard relational operators: =, ≤ and ≥.

2.3.3.1 Inequality Constraints $g_j(\mathbf{x})$

Inequality constraints are more natural in problem formulation. Inequality constraints provide more choices for design. The inequality constraint, given by Eqn. (2.2), represent constraints that need to be satisfied, such as slope of the pressure rise in compressor characteristics, operating range that needs to be achieved, etc.

2.3.3.2 Equality Constraints $h_j(\mathbf{x})$

Equality constraints are mathematically neat and easy to handle. Numerically, they require more effort to satisfy. They are also more restrictive on the design as they limit the region from which the solution can be obtained. The symbol representing equality constraints is $h_j(\mathbf{x})$ (see Eqn. 2.3).

2.3.3.3 Side Constraints $x_l$ or $x_u$

Side constraints ($x_l$ or $x_u$) are a necessary part of the solution techniques, especially numerical ones. It expresses the acceptable region for the design variables. Each design variable must be bounded by numeric values for it lower ($x_l$) and upper ($x_u$) limit. The designer makes this choice based on anticipation for an acceptable design. Equation 2.4 represents the range of design variables.
2.4 Classification of Optimization Methods

Optimization problems can be classified along seven major categories. The categories are as follows:

1. Linear vs. Nonlinear
2. Constrained vs. Unconstrained
3. Discrete vs. Continuous
4. Single vs. Multi-objective
5. Single vs. Multiple Minima
6. Deterministic vs. Nondeterministic and
7. Simple vs. Complex

2.4.1 Linear vs Nonlinear

When the objective function and the constraints are all linear, the problem is called a linear programming problem or linear optimization program. When either the objective function or any of the constraints is a nonlinear function of the design variables, the problem is called a nonlinear programming problem or nonlinear optimization problem. Generally, linear optimization problems are much easier to solve than non-linear optimization problems, particularly when the dimension of the problem is large (i.e., large number of design variables).

2.4.2 Constrained vs. Unconstrained

When the optimization problem does have constraints, we call it a constrained optimization problem. When we do not have any constraints at all, the problem is an unconstrained optimization problem. Most practical problems involve constraints. Thus, unconstrained problems generally have more theoretical than practical value. Specifically, in the case of nonlinear programming, the problem statement would only involve Eq. 2.1 – the relations 2.2 and 2.3 would not be part of the (unconstrained) optimization problem.

2.4.3 Discrete vs. Continuous

If any design variable is discrete, an optimization problem cannot be said to be continuous. the following typical cases. (i) In the first, the design variables could be restricted to take only the values of 0 or 1. This case is called 0-1 programming or binary programming. (ii) In the second case, the design variables are restricted to take on only integer values. This case is called integer programming. (iii) In the third case, the design variables are restricted to take on only a given prescribed set of real values. This case is generally called discrete optimization.
2.4.4 Single vs. Multi-objective

In general, the design or optimization of practical systems involves tradeoffs among competing objectives. Two objectives (such as efficiency and economy) compete against each other and require a compromise. Most optimization problems involve more than a single objective. Most are multi-objective in nature.

2.4.5 Single vs. Multiple Minima

Solving optimization problems that have several optima is referred to as global optimization. This optimization case is much more difficult to handle than single optimum optimization. The algorithms that are used to handle unimodal or multimodal problems are quite different. Single minimum search is referred to as local optimization, while multi-minima search is referred to as global optimization.

2.4.6 Deterministic vs. Nondeterministic

Deterministic optimization focuses on finding the global solutions of an optimization problem whilst providing theoretical guarantees that the reported solution is indeed the global one within some predefined tolerance. The term deterministic global optimization typically refers to complete or rigorous optimization methods. Rigorous methods converge to the global optimum in finite time. Deterministic global optimization methods are typically used when locating the global solution is a necessity.

2.4.7 Simple vs. Complex

A simple problem can be viewed as one that can be solved relatively easily by virtue of certain characteristics. This can be the case (i) because the model of the system is provided or readily created, (ii) because it only involves continuous variables, (iii) because it is not strongly nonlinear, (iv) because it is expected that local optimization will be sufficient, (v) because the computational model of the system behavior can run on a computer in seconds or minutes (not hours), (vi) because the number of design variables is not large, (vii) because all the models needed to describe the system behavior can run on a single computer, or (viii) because all the design variables are deterministic. An assessment of the above items gives us a sense of how simple or complex it will be to optimize the design or system.

2.5 Modeling System Behavior

Modeling can be defined as a process by which an engineer or a scientist translates the actual physical system under study into a mathematical model of the system.
Depending on the approach used to develop system behavior models, they can be classified into the following major categories:

1. Physics-based Analytical Models: These models are developed based on the physics of the system. If the physics of the system is defined by a set of differential equations, the analytical models represent the functional solution to those differential equations.

2. Simulation-based Models: These models generally leverage a discretized representation of the system in translating the system behavior to a set of algebraic equations that are solved using numerical techniques. Depending on modeling assumptions and the resolution of the discretization, the fidelity of these models can vary significantly. High-fidelity simulations, especially for complex systems, generally tend to be computationally expensive and more often than not require dedicated software for generating 3D geometries and performing the simulations. Examples of simulation-based models include finite element models, finite volume models, and spectral analysis models.

3. Surrogate Models: Surrogate models are purely mathematical and/or statistical models with certain generic functional forms and coefficients that can be tuned. These models are trained (i.e., the coefficients are tuned) using a set of input output data (i.e., [P, X] data) generated from a high fidelity source. The high-fidelity source could be comprised of experimental or simulations based analysis. As a result, surrogate models by themselves lack any direct physical information of the system; however, they provide the advantage of being tractable, fast, and highly portable.

### 2.6 Single-objective optimization formulation

In modern engineering applications, one single-objective function evaluation can involve a numerical simulation of the flow by a FVM, or a strength computation by a FEM. One single simulation may require several hours of computation time, and therefore sophisticated optimization methods are needed which find the optimum with a minimum number of evaluations of \( f(\vec{x}) \). Nevertheless, running one single evaluation in parallel on a cluster of computers can speed up the computation, it requires more computation power and possibly more licenses of the CFD and CSM software. Therefore, to keep the number of evaluation needed as low as possible is an important issue in an optimization algorithm.

The constrained single objective optimization problem as defined by Eqn. (2.1)-(2.5) is difficult to solve in a direct way due to the inequality constraints given by Eqn. (2.2). Usually the constrained formulation is translated into an unconstrained optimization problem for which the pseudo-objective function \( \tilde{f}(\vec{x}) \) needs to be minimized:
\[ f(\bar{x}) = f(x) + R \cdot \sum_{j=1}^{m} \delta_j (g_j(\bar{x}))^2 \]

Where

\[ \delta_j = 0 \quad \text{if} \quad g_j(\bar{x}) \leq 0 \]
\[ \delta_j = 1 \quad \text{if} \quad g_j(\bar{x}) > 0 \]

By minimizing the pseudo-objective function, an effort is made to satisfy the constraints. A penalty is given to the objective function in case of the constraints are not met. However, this is a weak formulation of the constraints, as for the optimum of \( f(\bar{x}) \) some constraints can still remain unsatisfied. The value of \( R \), the penalty multiplier, defines the weight given to satisfying the constraints. For large values of \( R \), more weight given to the constraints and the optimum of \( f(\bar{x}) \) will be closer to the constrained optimum of \( f(x) \) (see Fig. 2.1), however the pseudo-objective function becomes ill-conditioned. The value \( R \) is thus a tradeoff between numerical stability of the optimization and the degree of satisfaction of the constraints. It can also be chosen different for each constraint to put more emphasis on some constraints with respect to others.

![Figure 2.1 Influence of the penalty multiplier on the pseudo-objective function in 1D. By increasing \( R \) the location of optimum moves towards the feasible region. [1](image)](image)

2.7 Multi-objective optimization formulation

In many applications, it is required to maximize or minimize several properties at the same time, e.g. minimizing the mass of a component while increasing its strength. This type of optimization is called a multi-objective optimization and is defined by following equations.

Minimize: \[ f_i(\bar{x}) \quad i = 1...l \]
Subject to: \[ g_j(\bar{x}) \leq 0 \quad j = 1...m \]
\[ h_k(\bar{x}) = 0 \quad k = 1...n \]
\[ x_l \leq \bar{x} \leq x_u \]
The multiple objectives almost always conflict, as a result no single optimal solution exist. Instead, a Pareto front exists of non-dominated solutions. Figure 2.2, which is the results of an unconstrained problem involving two objectives which are functions of a single design variable $x$, provides the plot of each objective function on the same vertical axis and the design variable $x$ on the horizontal axis. Pareto optimal solutions, that are points between M1 and M2 in Fig. 2.2, are those for which any improvement in one objective will result in the worsening of at least one other objective. That is, a tradeoff will take place. Figure 2.3 is a plot of all the Pareto points that is identified in Fig. 2.2. All the points (M1, M2, A, B, and C) illustrated in Fig. 2.2 are plotted with their respective objective function values in Fig. 2.3. M1 and M2 form the end points of the Pareto frontier, also known as anchor points. Point M1 is where Objective 1 has the least value, while M2 is where Objective 2 has the least value. Points A and C, as dominated points, do not lie on the Pareto frontier.

Figure 2.2 Multi-objective Optimization with one single variable [1]

The weighted sum approach is the simplest and the most intuitively meaningful means of solving multi-objective optimization problems. It is also the one that is most widely used. However, it possesses some serious deficiencies. The AOF is simply a weighted linear combination of all the objective functions.

$$F(\vec{x}) = \sum_{i=1}^{l} w_{i} \cdot f_{i}(\vec{x})$$  \hspace{1cm} (2.11)

where $w_{i}$ are the weights given to the $i$-th objective function. However, the minimum of $F(\vec{x})$ will correspond with one single point of the Pareto front determined by the weights given to each objective.
2.8 Surrogate-Model Assisted Global Optimization

2.8.1 Surrogate Model Assisted Optimization Algorithm

Figure 2.4 shows the flow chart 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 the step which distributes the optimization parameters to have better analyzing the data with appropriate statistical methods. Design of experiment helps the designer in determining which variables are most influential on the objective functions and constraints and in determining 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 the current research works, ANSYS CFX is used to analyze the performance of the centrifugal compressor. In order to reduce the expensive numerical simulation cost and time, surrogate model is used. Surrogate model provides less accurate, however, very fast performance predictions to evaluate thousands of design variables sets. Artificial neural network (ANN) is chosen for surrogate model. The collected database 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 among the 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. 2.4). Differential evolution (DE), then, selects the elite sets of design variables which are sent back to analysis phase (CFD or CMS Simulation) to get accurate performance results. These processes are expressed as updating, the clockwise outer loop in the flow chart (Fig. 2.4). 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.

Figure 2.4 Surrogate model assisted optimization algorithm
2.8.2 VKI Optimization System

The optimization system used in the current research works was developed at the von Karman Institute for Fluid Dynamics and applied to a wide range of turbomachinery applications. The general idea of the optimization system is described as shown in Fig. 2.5. The top down blocks on the left of Fig. 2.5 represents the traditional design procedure in which theoretical consideration and experimental experience play a crucial role. The upper right box (Level 1) represents the optimization loop comprising of a differential evolution algorithm (DE) and a meta-model based on an artificial neural network (ANN). The artificial neural network (ANN) replaces the computationally expensive CFD simulations in the optimization loop (Level 1) and provides less accurate but very fast performance predictions to evaluate the larger number of geometries by the differential evolution algorithm during its search for the optimum. In order to train the meta-model for prediction, a database is required from the past experience through actual or numerical experiments. The database was, therefore, initialized by means of a design of experiment (DOE) comprising 32 designs in an advance optimization loop (Level 1) starts. After building a database and training the meta-model (ANN), the optimization loop (Level 1) is invoked and DE predicts the optimal geometries. However, ANN requires a validation which is then performed in the feedback loop (Level 2) in Fig. 2.5. The optimal geometries coming from the ANN prediction are analyzed by the more computationally accurate and expensive CFD calculations to verify the accuracy of the meta-model (Level 2). 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 meta-model on the enlarged database. [27-29]

![Figure 2.5 VKI Optimization system](image-url)
2.8.3 Design of Experiments

Design of experiments (DoE) techniques were originally developed to study (and empirically model) the behavior of systems through physical experiments (e.g., in experimental chemistry). DoE can be defined as the design of a controlled information gathering exercise for a system or phenomena where variation is known to be present. DoE techniques have existed in some form since the late 1700s, and is considered a discipline with very broad applications across the different natural and social sciences, and engineering.

The primary objective of DoE is to determine multiple combinations of the controlled parameters (or conditions) at which the experiments will be conducted. In mathematical terms, each combination of controlled parameters can be considered a sample. As such, DoE can also be perceived as a process of generating parameter/condition samples to conduct controlled experiments. Although, traditionally controlled experiments referred only to physical experiments, in modern times, it would include both physical and computational experiments.

DoE techniques have a large influence on the accuracy of the surrogate model developed thereof. To develop effective surrogate models, it is necessary to acquire adequate information about the underlying system. Assuming no prior knowledge of the system behavior, the typical DoE strategy is to generate a distribution of sample points throughout the design space in an uniform fashion. There are several techniques available to distribute the sample points, to provide an adequate coverage of the design space without any particular variable bias. These techniques include Factorial design, Central Composite designs, and Latin Hypercybe design, and Sobol sequence.

The most straightforward approach to uniform sampling is the factorial design method. In this technique, the range of each design variable is divided into different levels between the upper and lower limits of a design space. In a full factorial design, sample
points are located at all the combination of the different levels of all the design variables. Figure 2.6 illustrates a three-level factorial design in a three-variable space. In high-dimensional problems, the full factorial design approach may be cost/time prohibitive.

2.8.4 Overview of Artificial Neural Network

An artificial neural network is an information-processing system that has certain performance characteristics in common with biological neural networks. Artificial neural networks have been developed as generalizations of mathematical models of human cognition or neural biology, based on the assumptions that:

1. Information processing occurs at many simple elements called neurons.
2. Signals are passed between neurons over connection links.
3. Each connection link has an associated weight, which, in a typical neural net, multiplies the signal transmitted.
4. Each neuron applies an activation function (usually nonlinear) to its net input (sum of weighted input signals) to determine its output signal.

Neural networks are composed of simple elements operating in parallel. These elements are inspired by biological nervous systems. As in nature, the connections between elements largely determine the network function. Training a neural network to perform a particular function can be done by adjusting the values of the connections (weights) between elements.

Typically, neural networks are adjusted, or trained, so that a particular input leads to a specific target output. Figure 2.7 illustrates such a situation. Here, the network is adjusted, based on a comparison of the output and the target, until the network output matches the target. Typically, many such input/target pairs are needed to train a network. Neural networks have been trained to perform complex functions in various fields, including pattern recognition, identification, classification, speech, vision, and control systems.

Figure 2.7 Typical Working Principle of an Artificial Neural Network [3-5]
Neural networks come in two classes: feedforward networks and recurrent (or feedback) networks.

a) Feedforward neural networks

A feedforward neural network is a nonlinear function of its inputs, which is the composition of the functions of its neurons. A feed-forward network has a layered structure. Each layer consists of units which receive their input from units from a layer
directly below and send their output to units in a layer directly above the unit. There are no connections within a layer. Figure 2.8 shows a typical three-layer feedforward neural network.

b) Recurrent (or feedback) networks

Recurrent networks, in contrast to feed-forward networks, do have feedback elements that enable signals from one layer to be fed back to a previous layer. A basic recurrent network is shown in Fig. 2.9. A simple recurrent network is one with three layers, an input, an output, and a hidden layer. A set of additional context units are added to the input layer that receive input from the hidden layer neurons. The feedback paths from the hidden layer to the context units have a fixed weight of unity. In a recurrent network, the weight matrix for each layer contains input weights from all other neurons in the network, not just neurons from the previous layer.

Figure 2.10 ANN network layout with one hidden layer showing detailed connection

Figure 2.10 gives a schematic view of the implementation of a neural network in a mathematical form. Each element of the input layer is connected to each neuron of the first hidden layer and to each connection a weight is associated. Suppose the ANN has \( n \) input variables \( x_1, x_2, \ldots, x_n \) and one hidden layer. The input given to the \( j^{th} \) neuron of the hidden layer is given by:

\[
in_j^{\text{hidden}} = \sum_{i=1}^{n} w_{i,j}^{\text{hidden}} \cdot x_i + b_j^{\text{hidden}}
\]

2.12
where $w_{i,j}^{\text{hidden}}$ is the weight given to the connection of the $i^{th}$ input neuron to the $j^{th}$ hidden neuron, and $b_j^{\text{hidden}}$ is the bias given to the $j^{th}$ hidden neuron. This output serves as an argument of the transfer function associated to each neuron to compute the output of the hidden neuron. A variety of transfer function exist, but the sigmoid function is most used in hidden layers:

$$output_j^{\text{hidden}} = TF\left(in_j^{\text{hidden}}\right) = \frac{1}{1 + e^{in_j^{\text{hidden}}}}$$

Figure 2.11 Sigmoid function [3-5]

The output of the hidden neuron serves as an input for the next layer. Here, the next layer is output layer, however in general more hidden layer can follow each other. The output is computed as:

$$y_j = TF\left(\sum_{i=1}^{h} w_{i,j}^{\text{out}} \cdot output_i^{\text{hidden}} + b_j^{\text{out}}\right)$$

where $h$ is the number of hidden neurons, $w_{i,j}^{\text{out}}$ is the weight given to the connection of the $i^{th}$ hidden neuron to the $j^{th}$ output, and $b_j^{\text{out}}$ is the bias given to the $j^{th}$ output neuron.

2.8.5 Overview of Genetic Algorithm

Over the last decade, genetic algorithms (GAs) have been extensively used as search and optimization tools in various problem domains, including the sciences, commerce and engineering. The concept of a genetic algorithm was first conceived by John Holland of the University of Michigan, Ann Arbor. Thereafter, he and his students have contributed much to the development of this field. GAs are search and optimization procedures that are motivated by the principles of natural genetics and natural selection.
Some fundamental ideas of genetics are borrowed and used artificially to construct search algorithms that are robust and require minimal problem information.

Figure 2.12 A flowchart of the working principle of a GA [5]
Figure 2.12 shows a flowchart of the working of a GA. Unlike classical search and optimization methods, a GA begins its search with a random set of solutions, instead of just one solution. Once a random population of solutions is created, each is evaluated in the context of the underlying nonlinear programming (NLP) problem and a fitness is assigned to each solution. The evaluation of a solution means calculating the objective function value and constraint violations. Thereafter, a metric must be defined by using the objective function value and constraint violations to assign a relative merit to the solution (called the fitness). A termination condition is then checked. If the termination criterion is not satisfied, the population of the solutions is modified by three main operators and a new (and hopefully better) population is created. The generation counter is incremented to indicate that one generation (or, one iteration, in the parlance of classical search methods) of the GA is completed. The flowchart shows that the working of a GA is simple and straightforward.

1) Encoding: Encoding is a method that represent individuals in evolutionary algorithms. Typically, individuals are coded as a fixed length string (e.g., a binary number with 0’s). This string is also known as a chromosome.

2) Initial population: The algorithm begins by generating a random initial population. Important initialization choices, such as the number of individuals in each population and number of bits in the encoding, must be made by the user. These choices govern the performance of the GA.

3) Reproduction: A new generation, called child, in the genetic algorithm is created by reproduction from the previous generation, called the parent. The notion of “survival of the fittest” is usually used in genetic algorithms. There are three main mechanisms used to new generation. Different implementations of GAs use different combinations of the three ideas below. (see Fig. 2.12)
   a) Elitism: In this approach, the individuals with the best fitness values in the current generation are guaranteed to survive in the next generation. (see Fig. 2.13)
   b) Crossover: In this technique, some bits of the encoded string of one parent individual exchanged with the corresponding bits of another parent individual. (see Fig. 2.14)
   c) Mutation: Unlike the crossover operation (which requires two parents), mutation children are generated from a single parent by randomly reversing some bits from 0 to 1, or vice versa. In most GA implementations, a probability value for a mutation to occur is assumed. (see Fig. 2.14)
2.9 Overview of Particle Imaging Velocimetry

Turbomachines are used in a wide variety of engineering applications for power generation, pumping and aero-propulsion. Improving the efficiency in turbomachines requires understanding the flow phenomena occurring within rotating machinery. These detailed velocity mapping studies are required to improve the fidelity and accuracy of Computational Fluid Dynamics (CFD) code predictions. Detailed knowledge of the flow field inside impellers and diffuser are important for further improvements in the design of turbomachines. To this end, instantaneous data given by PIV have been proven as a
valuable supplement to mean data for the identification of unsteady and secondary flow phenomena.

Numerous researchers have employed various PIV techniques to study the unsteady flows in rotating machines, covering both pumps and compressors. Liquid based experimental setups have been used for pump studies and some fan studies. Rothlubbers et al. [6], used digital PIV to study the flow in a radial pump. Successful application of the PIV technique to both the blade passage region of a transonic axial compressor and the diffuser region of a high speed centrifugal compressor were presented by Wernet et al [7].

Particle image velocimetry, or PIV, refers to a class of methods used in experimental fluid mechanics to determine instantaneous fields of the vector velocity by measuring the displacements of numerous fine particles that accurately follow the motion of the fluid. The rate of particle movement is sensed by recording images of the particles or patterns related to those images at two or more precisely defined times and inferring the displacements of individual particles or the average displacements of small groups of particles from the displacements of individual particles or the average displacements of small groups of particles from the displacement of images.

Laser speckle velocimetry (LSV) is a very close relative of PIV, except that it deals with displacements of speckle patterns instead of particle images. Speckle methods for measuring displacement have their origin in solid mechanics, and the feasibility of extending them to measurement of velocity fields in fluid was first applied in 1970s. PIV was developed as a method of distinct from LSV when it was recognized that, unlike the surface of solids, the density of scattering particles in fluids would seldom be large enough to create speckle patterns in the scattered light. PIV rapidly became the dominant method, and it has remained so to this time.

The robust nature of the PIV stems in part from its conceptual simplicity. Unlike other methods of velocity measurement, PIV directly measures the two variables – displacement and time increment – that appear in the fundamental definition of velocity. In contrast, Laser Doppler velocimetry measures an intermediate physical phenomenon, the Doppler shift of light scattered by small particles, and relates the shift to the particle velocity. Similarly, thermal anemometers, such as hot-wire anemometers, infer velocity from measurement of the rate of heat transfer from heated elements to the fluid. And unlike these methods for velocity at a point, PIV gives entire fields of the velocity vector. PIV also derives robustness from using particles as markers of fluid motion. Particles are omnipresent in all but the cleanest fluids, and they scatter much more light than molecules or diffuse dye concentrations. Being micrometer sized they localize the velocity measurement, and being rigid they neither deform nor diffuse in time. Although the concept of measuring particle displacements is simple in essence, the factors that need to be addressed to design and implement PIV systems that achieve reliable, accurate and fast measurements and to interpret the results are surprisingly numerous. [9-10]
2.9.1 PIV Measurement Technique in Fluid Flow Field

PIV is one of the so-called Time of Flight (TOF) measurement techniques. The basic principle of the PIV is that a laser light sheet is used to illuminate the flow field which is seeded with small particles to visualize a flow to be measured. A double pulse YAG laser and a double shutter camera are synchronized to record two particle images with very short time separation, typically less than 100 μs (Fig. 2.15). [8]

PIV requires two particle images with very short time separation typically less than 100 micro seconds. The frame straddling technique enables to record two images with time separation of down to 100 nanoseconds. A double pulse laser and a double shutter camera are synchronized by the timing controller. Since the double pulse laser has two laser heads which can be operated independently, actual limitation is the length of dead time between two frames of the double shutter camera (Fig. 2.16). [8]

Fig. 2.15 A general optical measurement in a PIV system [8]

Fig. 2.16 Frame straddling [8]

Once the images are successfully recorded, the next step is the PIV analysis. The images are divided into small search areas, typically 32x32 pixels. These small search areas are called interrogation windows. The cross correlation is applied to these interrogation windows for both two images to obtain the correlation plane for each
interrogation window. The location of the interrogation windows in both images are same (In the Standard FFT cross correlation, the interrogation windows are shifted in the advanced algorithms). Then the peak detection and displacement evaluation are applied to obtain the dominant displacement in each interrogation window. As the size of a pixel in flow and the time separation between two images are known, the velocity can be calculated. The size of a pixel in flow is determined by the simple velocity calibration (Fig. 2.17). [8]

![Fig. 2.17 Displacement and velocity calculation](image)

2.9.2 PIV Measurement Calculation

The following Fig. 2.18 and 2.19 show the general procedure how to measure velocity flow field in PIV measurements in the diffuser of a centrifugal blower. In this measurement, PIV measurement without phase lock is presented. Figure 2.18 presents the PIV measurement images and calculated velocity flow field. The figures on the left (Camera A -1st) and middle (Camera A -2nd) are the images taken from CCD camera during the PIV measurement under difference time difference between each pair. The figures on the right show the calculated velocity flow field using the figure on the left and middle. In order to get the velocity flow field for steady state conditions, the velocity flow field calculated in previous state are averaged as shown in Fig. 2.19. Normally 165 pairs of images are taken for a single velocity flow filed image to represent the actual flow field. In order to measure the velocity flow filed with phase lock, the images are taken using the same time difference or trigger marked on rotating components of turbomachinery.
Camera A – 1st

Camera A – 2nd

Calculation

(a) Position 1

(b) Position 2

(c) Position 3

(d) Position 4

Figure 2.18 PIV measurement (calculation) (continued)
Figure 2.18 PIV measurement (calculation)

Figure 2.19 PIV measurement (average)
3

STUDY ON FLOW RANGE ENHANCEMENT IN A CENTRIFUGAL BLOWER

3.1 Overview

A high-pressure ratio and a wide operating range are highly required for compressors and blowers. The technical issue of the design is achievement of suppression of the flow separation at small flow rate without deteriorating the efficiency at design flow rate. A numerical simulation is very effective approach in design procedure, however, cost of the numerical simulation is generally high during the practical design process, and it is difficult to confirm the optimal design which is combined with many parameters. A multi-objective optimization technique is the idea that has been proposed for solving the problem in the practical design process. In this study, a Low Solidity circular cascade Diffuser (LSD) in a centrifugal blower is successfully designed by means of multi-objective optimization technique. An optimization code with a meta-model assisted evolutionary algorithm is used with a commercial CFD code ANSYS-CFX. The optimization is aiming at improving the static pressure coefficient at design point and at low flow rate condition while constraining the slope of the lift coefficient curve. Moreover, a small tip clearance of the LSD blade was applied in order to activate and to stabilize the secondary flow effect at small flow rate condition. The optimized LSD blade has an extended operating range of 114 % towards smaller flow rate as compared to the baseline design without deteriorating the diffuser pressure recovery at design point. The diffuser pressure rise and operating flow range of the optimized LSD blade are experimentally verified by overall performance test. The detailed flow in the diffuser is also confirmed by means of a Particle Image Velocimetry. Secondary flow is clearly captured by PIV and it spreads to the whole area of LSD blade pitch. It is found that the optimized LSD blade shows good improvement of the blade loading in the whole
operating range, while at small flow rate the flow separation on the LSD blade has been successfully suppressed by the secondary flow effect.

A low solidity circular cascade diffuser (LSD), which has been proposed by Senoo [12], is one of the effective devices to improve the pressure recovery coefficient at design point while guaranteeing a wide operating range of radial compressors and blowers. The advantage of the LSD for transonic compressors was confirmed by Hayami [13]. In a large flow rate, flow in the diffuser does not show choke characteristic because of the less throat section with a design of a wide pitch. In a small flow rate, the flow separation occurred at suction surface of the LSD blade because of a large angle of attack. Unsteady motion of the flow in the vaneless space between the impeller exit and the leading edge of the vaned diffuser was investigated by Everitt and Spakovszky [14], Anish et al.[15], and Goto et al.[16]. The reversed radial flow on the end wall near the leading edge of the vaned diffuser is one of the triggers of the short-wavelength stall at small discharged flow rate. Another criteria of the reversed radial flow on the side wall was proposed by Senoo et al. [17] and Ishida et al.[18-20]. At small flow rates, a secondary flow due to the reverse flow on the diffuser end wall suppresses the flow separation of the suction side surface of the LSD blade. The secondary flow effect is a unique feature of the LSD, but yet uncertain inflow to design the LSD in order to promote this effect. Hayami et al. [21] and Oh et al.[22] suggested a design procedure using experimental result with single and tandem vanes. Shibata et al.[23-24] applied a multi-objective optimization technique to design the LSD in order to improve both, the efficiency and flow rate of operation. The optimized LSD is experimentally verified showing an extended flow range of 8 % increase in surge margin while preserving the efficiency as compared to the conventional design.

In the first study, a LSD is optimized using a multipoint and multi-objective optimization technique especially focused on the secondary effect of the LSD. The optimization is aiming at improving the diffuser static pressure coefficient at design point and at small flow rate while constraining the slope of the lift coefficient curve. Seven design parameters describing the shape and position of the LSD vane were introduced, e.g. the radial spacing between impeller exit and the LSD leading edge, the radial chord length and the mean camber angle distribution of the LSD blade with five control points. Moreover, a small tip clearance of the LSD blade is applied in order to stabilize the secondary flow effect at small flow rate. The secondary flow is directly measured and is confirmed by Particle Image Velocimetry.

In the second study, the optimization results have been compared between in the cases with and without tip clearance. The advantage of the tip clearance from the point of view of activating the secondary flow is discussed. The secondary flow effect in the diffuser is also measured by means of a PIV (Particle Image Velocimeter). The flow was compared between in the cases with and without tip clearance.
3.2 Technical Issue of Diffuser in Radial Compressors

3.2.1 Effects of diffuser and scroll casing on compressor performance and flow range

(a) Vaneless
(b) Scroll casing without vane
(c) Vaned
(d) LSD Vaned

Figure 3.1 Typical configuration of diffuser in radial compressors

![Diffuser Configurations](image)

Figure 3.2 The variation of pressure coefficient with inflow angle for three types of diffuser [11]

![Pressure Coefficient Variation](image)
Figure 3.1 shows the typical configuration of diffusers in various radial compressors. The flow in the diffuser of radial compressors may look superficially simple but this is wrong – the flow is very complicated. Any diffuser, but particularly a vaneless one is often followed by a scroll collector and this gives added pressure rise but at the expense of range.

The principal differences in performance between the vane and vaneless diffuser are shown in Fig. 3.2, taken from Rodgers (1982), which is in the form of pressure rise versus average outlet flow angle from the impeller. The flow angle may be thought of as showing the variation in mass flow rate for a given impeller speed. For applications where maximum range is required the vaneless diffuser is preferred choice, however, for maximum pressure recovery the vaned diffuser is better.

3.2.2 Suppression of Stall by Secondary Flow

Figure 3.3 shows sketch of flow separation in the LSD blade and the simplified sketch of velocity triangles at the impeller exit at design flow rate (black) and at small flow rate conditions (blue). When the flow rate becomes smaller, the absolute flow angle \( \alpha_{2D} > \alpha_{2S} \) into the diffuser becomes smaller as viewed from the absolute frame of references. The decrease in absolute flow angle \( \alpha_2 \) causes the increase in incidence which causes the flow separation on the suction side of the LSD vane.

![Simplified sketch of velocity triangles at impeller exit at design point (D) and at small flow rate (S)](image)

As visible in Fig. 3.4, the lift coefficient \( C_L \) increases with increasing angle of attack \( \alpha_A = 90 - \gamma - \alpha_2 \), i.e. decreasing flow rate, due to the smaller absolute flow angle at impeller exit (Fig. 3.3: \( \alpha_{2D} > \alpha_{2S} \)). In the case of \( R_{LSD} = 1.10 \), the lift coefficient drops at an angle of attack of \( \alpha_A \approx 11^\circ \), hence the LSD blade must be stalled. However, in the case of the baseline blade of \( R_{LSD} = 1.20 \), the sudden drop of the lift curve has disappeared and it increases gradually until the angle of attack reaches \( \alpha_A \approx 13.5^\circ \). The flow separation is
illustrated in Fig. 3.5 for both configurations. Light blue zones show the separated regions on the hub side wall and the yellow zones indicate the flow separation on the shroud side wall. The discharged flow rate is $\varphi = 0.13$, which corresponds to the angle of attack of $\alpha_A = 13.5^\circ$. In the case of $RLSD = 1.10$ (Fig. 3.5-a), the flow separation on the suction side surface of the LSD blade spreads out far downstream on the hub side wall. On the other hand, in the case of $RLSD = 1.20$ as shown in Fig. 3.5-b, the separated region near the suction side surface merges with the reverse flow region on the shroud side wall. This is the key feature of the secondary flow effect [11] since the reverse flow region on the shroud end wall reaches to the leading edge of the adjacent blade and by this means reduces the excessive angle of attack. The formation of the reverse flow propagates in circumferential direction. As a result, the flow separation in the whole LSD will be suppressed simultaneously.

![Figure 3.4 Comparison of lift coefficient curve between R_{LSD}=1.10 and R_{LSD}=1.20](image)

Figure 3.4 Comparison of lift coefficient curve between $R_{LSD}=1.10$ and $R_{LSD}=1.20$

![Figure 3.5 Reverse flow zone on the diffuser side walls ($\varphi = 0.13$)](image)

(a) $R_{LSD}=1.10$
(w/o secondary flow)

(b) $R_{LSD}=1.20$
(with secondary flow)

Figure 3.5 Reverse flow zone on the diffuser side walls ($\varphi = 0.13$)
Figure 3.5 shows the projection of the limiting streamline (blue) of flow at the hub side wall in the baseline LSD at low flow rate ($\phi = 0.13$), where Fig. 3.5-a shows the baseline LSD which is located at radius ratio of $R_{\text{LSD}} = 1.10$ while Fig. 3.5-b shows the LSD at $R_{\text{LSD}} = 1.20$. The yellow zone represents the ISO surface of radially inward flow in the diffuser. In the case of $R_{\text{LSD}} = 1.10$ (Fig. 3.5-a), the flow separation on the suction side of the LSD blade spreads out far downstream of the hub side wall. On the other hand, in the case of $R_{\text{LSD}} = 1.20$ as shown in Fig. 3.5-b, the separated region near the suction side surface merges with the reverse flow region on the shroud side wall. This is the key feature of the secondary flow effect since the reverse flow region on the shroud end wall reaches the leading edge of the adjacent blade and the excessive angle of attack can be reduced from this effect [14]. The formed reversed flow on the shroud wall side propagates in the circumferential direction and simultaneous suppression of the flow separation in the entire LSD can be achieved. This is the reason why the suction side of the LSD blade shows no reverse flow zone in the case of $R_{\text{LSD}} = 1.20$ as seen in Fig. 3.5-b.

The comparison of the mass-flow averaged streamwise velocity distribution at small flow rate for the LSD with different configurations is illustrated in Fig. 3.6. For low specific speed impellers, it is known that the high velocity region moves from one end wall to the other inside the diffuser [17]. At low flow rate, the main stream with high velocity (red zone) moves from the hub wall side to shroud wall side inside the diffuser and the low energy fluid (green zone) reaches the hub wall side (Fig. 3.6 left). This means that low energy fluid (green zone) cannot reaches the leading edge of the adjacent blade and the main stream (red zone) was disturbed by the flow separation at the diffuser outlet. However, if the diffuser could activate the secondary flow, the flow separation formed
from the large angle of attack moves to the leading edge of the adjacent blade at shroud wall side and merges with the reversed flow at the leading edge of the adjacent blade as illustrated in Fig. 3.6 right. The main stream of the flow is kept at hub side without being disturbed by the flow separation. In that manner, the suppression of the flow separation can be achieved by the secondary flow effect without deteriorating the pressure rise in the diffuser.

### 3.3 Experimental Apparatus

#### 3.3.1 Centrifugal Blower Test Rig

![Experimental apparatus of test blower](Figure 3.7)

(a) Top view

(b) Side view

Figure 3.7 Experimental apparatus of test blower
The experimental apparatus of the centrifugal blower test rig is shown in Fig. 3.7. The main components of test blower consist of (1) Flow control valve, (2) Suction plenum tank, (3) Inlet suction pipe, (4) Centrifugal blower, (5) Coupling, (6) Timing belt, (7) Impeller drive motor and (8) Torque converter. The test facility was specially designed to guarantee an axisymmetric flow field (Fig. 3.8). The impeller is a low specific speed, un-shrouded centrifugal impeller for industrial application with an exit diameter of 510
mm. It has 16 backward leaning blades with a blade exit angle of 45°. The impeller exit width is 16.5 mm and the tip clearance is 1.0 mm. The air is discharged axisymmetrically to the atmosphere from the diffuser exit at a radius ratio of about $R = 1.6$. In the experimental campaign, the impeller operated at constant $\text{RPM} = 2,000 \pm 2$. The static pressures were measured by means of manometers at the suction plenum tank located upstream of the suction pipe (Fig. 3.7, C) and at $R = 1.02$ on the shroud side wall. The flow rate was measured using the entrance nozzle at the inlet of suction pipe (Fig. 3.7, 3). The geometric parameters of the LSD blade are defined as shown in Fig. 3.8. As can be seen in Table 1, the baseline LSD blade is a USA 35-B [16] airfoil with a chord length of $l = 101.4$ mm. The number of blades is 11 with a solidity $\sigma = 0.580$. The stagger angle and camber angle of the LSD blade is $\gamma = 61.76°$ (from radial direction) and $\theta = 22.0°$, respectively. The leading edge of the LSD blade is located at the radius ratio of $R_{LSD} = 1.10$ downstream of the impeller.

Table 3.1 Design parameters of blower’s impeller

<table>
<thead>
<tr>
<th>Baseline LSD blade</th>
<th>USA 35-B</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chord length of LSD blade, $l$ (mm)</td>
<td>101.4</td>
</tr>
<tr>
<td>Number of LSD blades</td>
<td>11</td>
</tr>
<tr>
<td>Stagger angle, $\gamma$</td>
<td>61.76°</td>
</tr>
<tr>
<td>Camber angle, $\theta$</td>
<td>22.0°</td>
</tr>
</tbody>
</table>

3.3.2 PIV Measurement

Figure 3.10 shows the PIV setup to measure velocity distribution during experimental campaign. Direct measurement of internal flow of the diffuser was acquired by a Particle Image Velocimetry (PIV). PIV system is constructed with a double pulse Nd:YAG Laser (Litron Nano S50-15, 50mJ/15Hz), a PIV camera (PCO-1600,14bit) and synchronizer. All of the components are controlled by PIV software (Seika-corp. Koncerto II). The pulse timing of the laser is synchronized with the pulse of angle from the rotational impeller. The laser was shot from the outside of the diffuser to have an image of velocity field between the LSD blades. Measurement plane between the diffuser side walls was changed from 10% to 90% of the span from the hub side wall. A thickness of the laser sheet is around 1.0mm, a time interval of the double pulse laser is fixed to 20 μsec. The PIV image was taken PIV camera through the acrylic glass of the diffuser shroud casing. A time averaged velocity field was calculated from totally 167 PIV images without a phase-lock to the timing of the rotational impeller. Seeding particles from a seeding generator (PIVTec, PivPart14) has an averaged diameter of 1.0 μm. The seeding generator is installed inside of the plenum tank which is located at upstream of the inlet suction pipe.
3.4 Numerical Simulation

The steady-state numerical simulations were conducted with the commercial CFD code of ANSYS-CFX 14.5. The high resolution scheme was applied to calculate the advection terms. The blend factor of the high-resolution scheme is automatically defined from the flow in order to have a robustness of the simulation. For the turbulence closure, the k-ω turbulence model was used for the flow with a low Reynolds number of $5 \times 10^4$ in this study. To reduce the computational cost only one pitch of the impeller blade and the diffuser were modeled and periodic boundary conditions were applied. An inlet boundary of the simulation was located at far upstream from the impeller inlet, axial distance of the inlet boundary corresponds to 1.2 times of the impeller diameter. An exit boundary was located at diffuser exit which corresponds to the radius ratio of 1.8 of the impeller. The boundary condition at the inlet boundary is mass flow rate, and at the outlet boundary is static pressure. It is the same boundary condition of the experiment using the test rigs as shown in Fig. 3.8.

Figure 3.11 shows the computational grid of the diffuser. The domain interface between the rotating frame and the stationary frame is located at radius ratio of $R = 1.01$. The total number of computational grid is about 800,000 in this study. The circumferential averaging model was adopted on the interface between the rotating frame and the stationary frame. Non-uniformity of the circumferential flow distortion at the upstream exit is not retained at the downstream inlet boundary, on the other hand, the flow distortion in the meridional plane is conserved at the interface. To prove the reliability of the numerical prediction prior to the optimization, the results of the numerical simulation
were validated by experimental results (Fig. 3.12). Three static pressure coefficients are presented as a function of the discharged flow rate $\phi$: the normalized static pressure rise between impeller inlet and diffuser exit ($\psi_{se}$), the pressure rise in the impeller ($\psi_{s2}$), and the pressure rise in the diffuser ($\psi_d = \psi_{se} - \psi_{s2}$). The solid marks in Fig. 3.12 denote the surge flow rate ($\phi \approx 0.1$) at which periodic pressure fluctuation was observed by means of the semiconductor pressure sensors. As visible in Fig. 3.12 the numerical results are in fair agreement with the experimental results in the entire operating range.
3.5 Study on Flow Range Enhancement by Secondary Flow Effect in Low Solidity Circular Cascade Diffusers

3.5.1 Design Parameters

In this study, the LSD was optimized for the existing impeller of the experimental facility (see Fig. 3.7). The diffuser blade was parameterized by a blade angle and thickness distribution (Figs. 3.13) based on B-splines with an underlying set of control points. The coordinates of the control points are the optimization parameters which can be modified by the optimization program. For the blade angle distribution 5 degrees of freedom were assigned to the control points and two additional design parameters were the radial position of the LSD blade and its chord length. This results in a total of seven design parameters. The control points of thickness distribution were defined as dependent parameters on the chord length. The resulting 2-dimensional blade shape was used in the hub and tip section, which were linearly connected to build the 3D blade. The LSD is composed of 11 vanes, which remained unchanged in the optimization process. In order to activate the secondary flow effect [18], a tip clearance of 3.0% of the span was used.

![Blade angle distribution](image1.png) ![Blade thickness distribution](image2.png)

Figure 3.13 Control points of design parameters

3.5.2 Objectives and Constraints

Industrial radial compressors and blowers have to comply with the demand of high efficiency over a wide operating range. A multipoint and multi-objective optimization strategy is therefore the best choice to meet these requirements simultaneously. This optimization was aiming at improving the static pressure coefficient at diffuser section $\psi_d$ at two different operating points as shown in Fig. 3.14-(a) at the design flow rate ($\varphi = 0.27$) and at small flow rate ($\varphi = 0.10$). The objective functions are solved as a minimized problem in this paper as shown in Eqn. 3-1. Both objectives were
subject to two constraints. As it is shown in Fig. 3.14-(b), the slope of the lift coefficient curve was constrained in two operating points according to Eqn. 3-2 to guarantee an entirely negative slope and hence stable operation in the whole operating range of the blower.

Figure 3.14 Sketch of objectives and constraints

(a) Objectives
(b) Constraints

Objective

Minimize \(-\psi_{d\phi0.27}\)

Minimize \(-\psi_{d\phi0.10}\)

Constraint

\[ CL_{\phi0.10} - CL_{\phi0.13} \geq 0 \]
\[ CL_{\phi0.07} - CL_{\phi0.10} \geq 0 \]

3.5.3 Results and Discussion

In Fig. 3.15, the 2-dimensional objective space after 50 iterations is presented. It shows the static pressure coefficient at design point \((\psi_{d\phi0.27})\) with respect to static pressure coefficient at small flow rate \((\psi_{d\phi0.1})\). Each symbol represents one design which has been analyzed by CFD and is satisfying all constraints (Eqn. 4-2). The gray solid circle marks are the initial database from by the DOE process, prior to the optimization, while the red solid marks were generated during the optimization process. The yellow circle mark indicates the performance of the baseline LSD blade. Both objectives are improving towards the lower left hand corner of Fig. 3.15. Typical of multi-objective optimization problems where both objectives are conflicting, a set of non-dominated optimal geometries was found. These designs are indicated by blue solid marks and are located on the Pareto Front where one objective cannot be improved without worsening.
the other. The selected optimized LSD blade is indicated in Fig. 3.15 as a green circle, which was regarded as the best compromise between both objectives.

Figure 3.15 2-dimensional objective space

Figure 3.16 Comparison of characteristics curve between baseline blade and optimized blade (CFD)
Table 1 Comparison of design parameters between baseline blade and optimized blade

<table>
<thead>
<tr>
<th></th>
<th>( R_{LSD} ) [-]</th>
<th>Solidity [-]</th>
<th>Chord length [m]</th>
<th>Camber [deg]</th>
<th>Stagger [deg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Baseline</td>
<td>1.20</td>
<td>0.580</td>
<td>0.1014</td>
<td>22.00</td>
<td>61.76</td>
</tr>
<tr>
<td>Optimized</td>
<td>1.08</td>
<td>0.486</td>
<td>0.0764</td>
<td>28.85</td>
<td>59.85</td>
</tr>
</tbody>
</table>

Figure 3.17 Comparison of the shape of blade between baseline blade and optimized blade

In Fig. 3.16, the blower characteristics of the optimized LSD are compared with the baseline geometry based on numerical simulations. As already seen in Fig. 3.16, the diffuser static pressure coefficient \( \psi_d \) has been slightly improved at design flow rate and at small flow rate condition. However, the main improvement was achieved for the lift coefficient \( C_L \) of the LSD blade in the whole range of operation from design flow rate of \( \phi = 0.27 \) to small flow rate of \( \phi = 0.07 \). On the other hand, according to the numerical prediction the static pressure coefficient at impeller exit \( \psi_{s2} \) deteriorates by 3.6 % and the global performance of the blower deteriorates by 2.8 %. This is mainly due to the smaller distance between impeller exit and diffuser leading edge of the optimized design. The impeller exit performance was however not targeted as objective in this optimization, as only the global performance was addressed, with success. As shown in Table 1 the radius ratio of the optimized LSD was reduced from \( R_{LSD} = 1.2 \) to \( R_{LSD} = 1.08 \). This is also visible in Fig. 3.17 in which both LSD geometries are compared. The leading edge location of the optimized blade is 10 % closer to the impeller exit and the chord length is 25 % shorter as compared to the baseline blade. The camber angle of the optimized blade is 31 % larger, while the stagger angle has been reduced by 3.1 %. A comparison of the blade angle distribution is shown in Fig. 3.18. From this plot it is apparent that most of the increase in camber angle (from 22.0° to 28.85°) has been realized in the rear part of the blade.
3.5.3 Experimental validation of the optimized LSD

Figure 3.18 Comparison of blade angle distribution between baseline blade and optimized blade

Figure 3.19 Comparison of blower characteristics between baseline blade and optimized blade (EXP)
The characteristics of the optimized LSD blade was confirmed by experiments. This is shown in Fig. 3.19. The stall onset was found at the flow coefficient of $\phi = 0.078$, which means that the optimized geometry has an extended operating range by 114% towards small flow rate condition as compared to the baseline design. Furthermore, the diffuser performance was not deteriorated in the whole flow rate range, which is in agreement with the numerical predictions.

The reason why the optimized blade has even an extended operating range must be related to the secondary flow effect, which is illustrated in Fig. 3.20. It shows the reverse flow zones at small flow rate for $\phi = 0.13$. The yellow zones show the reverse flow on the shroud end wall. There is no reverse flow on the hub side wall. The reverse flow zone on the shroud side wall acts as the secondary flow, which suppresses the flow separation of the suction side surface of the LSD blade.

Figure 3.20 Reverse flow zone on the diffuser side walls (CFD, Optimized, $\phi = 0.13$)

The detail velocity distribution between LSD blades is measured by PIV. Measured velocity field is compared with CFD result as shown in Fig. 3.21 in the case of baseline blade, and Fig. 3.22 in the case of optimized LSD. In both figure, velocity vector is plotted with color of flow angle. The red color shows the radially outward flow, and blue color shows the radially inward flow. In Fig. 3.21, the measurement plane is 10% span which close to the hub side wall. Flow separation at the suction surface of the blade which is shown in blue color is clearly captured by PIV. In Fig. 3.22, the measurement plane is 90% span which close to the shroud side wall. At the suction surface of the optimized LSD blade, the velocity vector shows red color which indicates that flow is not separated. However, at the middle of the LSD pitch, the blue color region spreads in the circumferential direction. Low energy fluid, which passes through the tip clearance at the
shroud side of the LSD blade, changes the vectors toward the radially inward at the middle of the LSD pitch, and it reaches to the leading edge of the adjacent blade. At the adjacent blade, incoming flow rate increases by means of the low energy fluid and result in the reduction of the flow incidence, and finally, the flow separation at the suction surface of the blade is successfully suppressed. The formation of the reverse flow propagates in circumferential direction, and as a result, the flow separation in the whole LSD will be suppressed simultaneously. It is found that the secondary flow was generated by the optimized LSD with tip clearance is clearly captured by the PIV image.

Figure 3.21 Comparison of velocity distribution between PIV image and CFD result (Baseline RLSD=1.10, φ = 0.13, span 10% : Hub side)

Figure 3.22 Comparison of velocity distribution between PIV image and CFD result (Optimized LSD, φ = 0.13, span 90% : Shroud side)
The final discussion is the advantage and dis-advantage of the secondary flow. An excessive secondary flow only marginally decreases the impeller performance. The static pressure coefficient at impeller exit $\psi_s$ is deteriorated at small flow rate condition as shown in Fig.14. This may be related to the reverse flow reaching the impeller exit, which reduces the deceleration of the main flow at impeller exit. It is the important point of the present study that the optimizer could find the stable secondary flow with the minimum reverse flow rate maximizing the deceleration of main flow in the vaneless space as possible.

3.5.4 Conclusions

A multipoint and multi-objective optimization is applied to design a low solidity cascade diffuser in centrifugal blowers. The fully automated design approach uses a differential evolution algorithm, a metamodel and a steady-state CFD solver to improve the diffuser static pressure rise at design and off-design conditions. The optimized blade was validated by the experimental test rig at the point of view of the pressure rise and velocity fields. The concluding remarks in this study are follows,

1. It is effective to apply the tip clearance at the LSD blade in order to stabilize the secondary flow effect.
2. In spite of the high camber angle, the optimized LSD shows good anti-stall characteristics at small flow rate conditions. It is based on the secondary flow effect which suppresses the flow separation on the suction side surface of the LSD blade.
3. The secondary flow near the diffuser side wall is clearly captured by the PIV image.
4. It is found that the optimized shape shows good improvement of the operating flow rate range without deteriorating the diffuser performance.

3.6 Optimization and Validation of Secondary Flow Effect for Flow Range Enhancement in a Low Solidity Circular Cascade Diffuser

A Low Solidity circular cascade Diffuser (LSD) in a centrifugal blower is designed by means of multi-objective optimization technique. Multi-objective optimization is done by a meta-model assisted evolutionary algorithm in which a commercial CFD code CFX from ANSYS is embedded. The aim of optimization is to improve the static pressure rise at design point and at off-design point condition and the slope of lift coefficient curve of the LSD vane is defined as the constraint. The shape and location of the LSD vane is parametrized by seven design parameters. The secondary flow effect at small flow rate condition is activated by applying a small tip clearance at the LSD blade. The comparison of characteristics of optimized LSD and the baseline is done
numerically and experimentally. The experiment result from the blower test rig shows that the optimized LSD has an extended operating range of 114% towards smaller flow rate as compared to the baseline design without deteriorating the diffuser pressure recovery at design point. The detailed flow phenomena in the LSD blade has been studied by means of a Particle Image Velocimetry (PIV) technique. The flow was compared between in the cases with and without tip clearance. In spite of the fluctuating flow at the diffuser inlet, secondary flow spreads to the whole area of LSD blade pitch stably. Stable secondary flow suppresses flow separation at the suction surface of the LSD. It is found that the optimized LSD blade shows good improvement of the blade loading in the whole operating range, while at small flow rate the flow separation on the LSD blade has been successfully suppressed by the secondary flow effect.

3.6.1 Results and Discussions

The main aim of the optimization in this study is operating flow range enhancement. It is possible to apply the flow range as the objective function, however, process of exploring the stall limit costs very high. The strategy of this study is that the target of the flow rate of stall limit was decreased step by step. Three steps of optimization results will be discussed here. The static pressure coefficient of the diffuser section $\psi_d$ at 2 points of the flow rate (Design and Near-surge) were selected as objectives, and negative slope of the lift coefficient of the diffuser $\Delta C_L$ was applied as constraints. The combination of these objectives and constraints means that the optimizer tries to find the diffuser shape with better performance without stall conditions.

3.6.2 Results of 1st optimization

In Fig. 3.23, the results of 1st optimization is shown with 2-dimensional objective space. Two points of objectives were applied as design flow rate condition of $\phi = 0.27$ and near surge flow rate condition of $\phi = 0.13$. Slope of lift coefficient $\Delta C_L$ between the flow rate of $\phi = 0.13$ and 0.10 was applied as constraints. In the figure of 2-dimensional objective space, one circle mark indicates one design which has been analyzed by CFD. The light gray circle (DOE) and dark gray circle (DOECnstr) marks are the initial database of the DOE process, prior to the optimization, while the blue (IT) and red (ITCnstr) circle marks presented the LSD design predicted by evolutionary algorithm during the optimization process. The dark gray circle and red circle marks represent the designs which satisfy the constraints. As the optimization was treated as minimization problem, the better design tended to approach to lower left hand corner. During the 1st optimization process, the optimizer generates many candidates as shown in red circles. Moreover, the Pareto front where both objective are conflicting beyond it forms clearly by the red circles in the figure.
Figure 3.23 1\textsuperscript{st} optimization results of 2-dimensional objective space (w/o tip clearance)

Figure 3.24 2\textsuperscript{nd} optimization results of 2-dimensional objective space (w/o tip clearance)

Figure 3.25 3\textsuperscript{rd} optimization results of 2-dimensional objective space (with tip clearance)
3.6.3 Results of 2nd optimization

The second step of the optimization in this study is exploring the possibility of improvement under smaller flow rate condition. One of the objective functions was applied at small flow rate of \( \phi = 0.10 \), and smaller flow rate of \( \phi = 0.07 \) was simulated in order to calculate slope of lift coefficient \( C_L \). The optimization result is shown in Fig. 3.24. There are no red circles in Fig. 3.24. It means there are no candidates of the optimized shape of blade which satisfied the constraints. Moreover, the blue circles (IT) indicated that there might be the better geometries although these did not satisfy the constraints. However, the optimal set of geometries could not be found while the operation range has been extended to smaller flow rate (\( \phi = 0.10 \)) and did not satisfy the constraints.

This must be coming from the fact that predicted geometries could not activate the secondary flow effect near small flow rate. The criterion of the previous study [18] is that the ideal flow pattern for activating secondary flow is as shown in Fig. 3.6. The high velocity region kept the position near one side of the diffuser side wall. In order to realize the ideal flow pattern for activating the secondary flow effect, blade loading balance between hub and shroud should be changed. The tip clearance of the LSD is one of the simple idea to make an unbalanced blade loading.

3.6.4 Results of 3rd optimization

In the 3rd optimization, small tip clearance of the LSD blade at shroud side was applied. The clearance is 0.5 mm which corresponds to 3% height of the span. It is possible to include tip clearance height as an optimization design parameter, however, in order to avoid the computational mesh problem, the fixed tip clearance was applied in this study. The objectives and constraints are the same as the 2nd optimization case with only difference of application of the tip clearance in the 3rd optimization. The optimizer found many candidates which satisfied the constraints as shown in red circles in Fig. 3.25. The selected optimized LSD blade is indicated in Fig. 3.25, which was regarded as the best compromise between both objectives.

The comparison of the characteristics between the baseline LSD and optimized ones (with and without tip clearance) is shown in Fig. 3.26. The slight improvement of the static pressure coefficient of the diffuser \( \psi_d \) at design flowrate, \( \varphi = 0.27 \), can be seen in the case of optimized LSD without tip clearance compared to the baseline. In the case of optimized LSD with tip clearance, the static pressure coefficient \( \psi_d \) shows extension of operating range to the small flow range side without much deterioration in static pressure coefficient \( \psi_d \) of the diffuser. However, the main difference was clearly seen in the comparison of lift coefficient \( C_L \). Although LSD blade without tip clearance shows the higher lift coefficient near design flow rate, it falls into the stall region when the flow coefficient becomes less than 0.13.
Figure 3.26 Comparison of blower characteristics from numerical simulation between baseline and optimized LSD blades

3.6.5 Experimental validation of the optimized LSD

Figure 3.27 Comparison of blower characteristics between baseline blade and optimized blade (CFD)
The experimental analysis of the characteristics of the optimized LSD blade was illustrated in Fig. 3.27. The stall onset was found at the flow coefficient $\phi = 0.091$ in the case of optimized LSD without tip clearance and at the flow coefficient $\phi = 0.078$ in the case of optimized LSD with tip clearance, meanwhile stall onset was found at the flow coefficient $\phi = 0.10$ in the case of the baseline. This means that the operating range was extended toward surge side by 105% in the case of optimized LSD without tip clearance and by 114% in the case of optimized LSD with tip clearance. As predicted by the numerical simulation, no significant deterioration was found on the static pressure coefficient of optimized diffuser in the whole flow range for both cases.

![Front view](image1)

![Meridional view](image2)

Figure 3.28 Reversed flow zone on the diffuser side walls (CFD, Optimized $\phi = 0.13$)

Figure 3.28 shows the forming of secondary flow effect on the shroud wall side of the diffuser vane, which is the main reason of the extended operating range of the optimized LSD diffuser. The figure on the left shows the reversed flow (yellow) zone at the flow coefficient $\phi = 0.13$, while the figure on the right shows the meridional views of the optimized LSD diffuser with tip clearance at the flow coefficient $\phi = 0.13$, $\phi = 0.10$ and $\phi = 0.07$. In spite of increase in area of reversed flow zone on the shroud wall side with decrease in the flow rate, the optimized LSD blade shows good suppression of flow separation and stable secondary flow on the shroud wall side.

3.6.6 PIV measurement of the optimized LSD blade

The detailed velocity distribution between LSD blades is measured by PIV. PIV images of velocity distribution between the cases of LSD blade with and without tip clearance are compared under various small flow rates as shown in Fig. 3.29. In these figures, velocity vector is plotted with color of absolute flow angle measured from circumferential direction. The red color shows the radially outward flow, and blue color shows the radially in-ward flow. In these figures, the measurement plane is 90% span which close to the shroud side wall.
Figure 3.29 Comparison of velocity distribution between the cases with TC and w/o TC
(Optimized LSD, span = 90%, shroud side)

In Fig. 3.29, the comparison of velocity distribution of PIV images of LSD blade between the cases with and without tip clearance for small flow rate from flow coefficient $\phi = 0.14$ to $\phi = 0.11$ is shown. The left side of each pair represents the velocity field of LSD blade without tip clearance while the right side represents the case with tip clearance.
Both cases are compared using optimized blade under same operation conditions except the tip clearance. The effect of tip clearance on the secondary flow, however, is quite significant near the small flow rate. In the case without tip clearance, at the suction side of the LSD blade, the velocity vector shows red color indicating that the primary flow reaches the shroud wall. This means that the primary flow is disturbed at the diffuser exit by the flow separation due to large angle of attack at small flow rate and forming of secondary flow by the LSD blade is unsuccessful. In the case with tip clearance, velocity vectors of blue color, which indicate the low energy fluid due to flow separation, leaving from the suction side reaching the leading edge of adjacent blade for the whole range of small flow rate. It can be concluded that the forming of secondary flow is successful in the case of optimized LSD blade with tip clearance. The primary flow is undisturbed by the flow separation resulting in the continuous rise in lift coefficient (CL) without falling into stall region.

In Fig. 3.30 and Fig. 3.31, the comparisons of velocity distribution between PIV image and CFD result in the case of LSD blade with tip clearance is shown. The measurement locations were 90% span near shroud side for Fig. 3.30 and 10% span near hub side for Fig. 3.31 respectively at the flow coefficient \( \phi = 0.13 \). At the suction surface of the optimized LSD blade, the velocity vector shows red color at both shroud side and hub side which indicates that flow is not separated. However, at the middle of the LSD pitch near shroud side (Fig. 3.30), the blue color re-ion spreads in the circumferential direction. Low energy fluid, which passes through the tip clearance at the shroud side of the LSD blade, changes the vectors radially inward at the middle of the LSD pitch, and it reaches the leading edge of the adjacent blade (Fig. 3.30). In spite of difference velocity flow field between CFD and PIV measurement near hub side (Fig. 3.31), both results showed radially outward downstream after LSD blade without being disturbed by flow separation. The blue region at the leading edge of adjacent blade in Fig. 3.31 indicated the movement of low energy fluid from shroud wall side to the hub wall side along the leading edge. At the adjacent blade, incoming flow rate increases by means of the low energy fluid and result in the reduction of the flow incidence, and finally, the flow separation at the suction surface of the blade is successfully suppressed. The formation of the reverse flow propagates in circumferential direction, and as a result, the flow separation in the whole LSD will be suppressed simultaneously. It is found that the secondary flow generated by the optimized LSD with tip clearance is clearly captured by the PIV image.

Final discussion is the stability of secondary flow in LSD blade by PIV measurement with phase lock. Previous PIV measurements were presented as an average velocity distribution over a specific duration without taking into account the location of impeller position. Figure 3.32 shows the time dependent PIV measurement of velocity field in the optimized LSD blade with tip clearance at 90 percent span near shroud side at the flow coefficient \( \phi = 0.13 \). The description of the PIV images is illustrated in Fig. 3.32-
The velocity fields in Fig. 3.32 (a to k) are from eleven different impeller position for one LSD blade pitch. The impeller builds pressure against the blockage in the diffuser until the impeller can longer pump against the developed head. At small flow rate the high-pressure fluid (blue zone) built up in the diffuser flows back upstream to the vaneless space. It can be seen that stable movement of low-momentum fluid (blue zone) from the suction side to the leading edge of the adjacent blade in all the impeller positions (Fig. 3.32, a to k). The pressure gradient, with respect to the impeller position, in the diffuser passage is changing with time. The separated nature of the flow (jet-wake) model at the impeller outlet also can be seen from the PIV measurement.

Figure 3.30 Comparison of velocity distribution between PIV image and CFD results with TC (Opt: LSD, \( \phi = 0.13 \), span = 90%, shroud side).

Figure 3.31 Comparison of velocity distribution between PIV image and CFD results with TC (Opt: LSD, \( \phi = 0.13 \), span = 10%, hub side).
3.6. Conclusions

A multipoint and multi-objective optimization is applied to design a low solidity cascade diffuser in centrifugal blowers. The fully automated design approach uses a differential evolution algorithm, a metamodel and a steady-state CFD solver to improve the diffuser static pressure rise at design and off-design conditions. Three steps of optimization were carried out with taking into consideration the secondary flow effect. The optimized blade was validated by the experimental test rig focusing on the pressure rise and velocity fields.

The concluding remarks in this study are follows:

Figure 3.32 Velocity distribution measured by PIV technique with a phase lock (Opt: LSD with TC, $\varphi = 0.13$, span = 90%, shroud side)
1. It is difficult to optimize the LSD with a wide operating flow rate range without applying the tip clearance. The tip clearance is one of the simple ways to activate the secondary flow at small flow rate conditions.

2. The optimized LSD shows good anti-stall characteristics at small flow rate conditions being the high camber angle. The secondary flow effect is the main reason which suppresses the flow separation on the suction side surface of the LSD blade.

3. The secondary flow near the diffuser side wall is clearly captured by the PIV image.

4. It is found that the optimized shape shows good improvement of the operating flow rate range without deteriorating the diffuser performance.
4

STUDY ON THE PERFORMANCE IMPROVEMENT AND FLOW RANGE ENHANCEMENT IN CENTRIFUGAL COMPRESSORS FOR SMALL TURBOCHARGER

4.1 Overview

Turbocharging has become indispensable for the reciprocating engines of vehicles. The technology was first used as a mean to increase the engine power density, but its benefits are highlighted more in downsizing and emission control nowadays. Turbocharging makes it possible to downsize vehicle engines significantly, thereby lowering the fuel consumption and reducing the carbon dioxide (CO$_2$) emissions. Furthermore, it is important to facilitate the treatment of nitrogen oxide (NO$_x$) emissions with high exhaust gas recirculation rates, helping to satisfy future rigid emission regulations. With increasing low-end torque and downsizing requirements, turbocharger centrifugal compressors need a wider stable flow range, especially at high pressure ratios.

A casing treatment for flow range enhancement in a centrifugal compressor was investigated firstly by Fisher et al. [31]. Effect of the casing treatment was investigated experimentally in the turbocharger for a marine diesel engine. Recently, numerical investigation has succeeded for the recirculation device by Tamaki et al. [32]. Effect of recirculation device with a counter swirl vane was investigated in a centrifugal compressor with high pressure ratio. The new designed counter swirl vane which was installed inside of the recirculation flow type casing to provide the recirculation flow with counter swirl, and operational flow rate range is
successfully enhanced. Ishida et al. [33] studied effects of the inlet recirculation arrangement on inducer stall and the diffuser width on diffuser stall in a high-specific-speed-type centrifugal impeller. Unstable flow range of a blower was significantly reduced about 45% without deteriorating impeller characteristics by implementing the recirculation flow type casing and the narrowed diffuser width. Sakaguchi et al. [34] also investigated effect of the recirculation casing treatment with or without guide vane inside. 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. Chi-Yong Park et al. [35] numerically showed that the recirculation flow type casing without the guide vane increased the operating range of the compressor and the recirculation flow type casing with the guide vanes improved both performance of the compressor at small flow rates and stall margin of the compressor. The study of Xinqin Z. et al. [36] showed that the self-circulation casing treatment reduced the accumulation of the low-energy fluid in the blade tip, thus delaying impeller stall. Subenuka et al. [37] investigated the map width enhancement and the performance improvement of a turbocharger compressor using straight and curve vanes in the annular cavity and significantly improved the map width by 14.3% and the peak pressure ratio by 2.25%. Xinqian et al. [36, 40] thoroughly studied the non-axisymmetrical flow in a centrifugal compressor and widened the operating range by developing an asymmetrical flow control method using non-axisymmetrical self-recirculation casing treatment. Their results show that the non-axisymmetrical casing treatment has a certain influence on the performance and has a larger potential for stability improvement than the conventional casing treatment.

In the first study, the baseline shape of recirculation flow type casing is modified and optimized. An optimization code with a meta-model assisted evolutionary algorithm is used with a commercial CFD code ANSYS-CFX. The objective of the optimization is to find the design with higher efficiency in wide flow rate range. The optimized recirculation flow rate is discussed by the sensitivity analysis of the optimization output.

In the second study, the baseline shape of casing treatment is modified and optimized for enhancement of the flow rate range. An optimization code with a meta-model assisted evolutionary algorithm is used with a commercial CFD code ANSYS-CFX. The objectives of the optimization are to find the design with higher efficiency at design flow rate and wider operating range. The sensitivity analysis of design parameters as a function of operating flow range and adiabatic efficiency at design flow rate are discussed in detail.

In the third study, in order to suppress too large pre-whirl at the shroud side of impeller inlet, guide vanes were installed in the annular passage of the recirculation device. The recirculation flow type casing treatment with guide vane is applied to the high speed centrifugal compressor, in addition, global optimization has performed to find out the optimized shape of the guide vane. It is found that the optimized guide vane has a possibility to reduce an excessive pre-whirl without deteriorating the adiabatic efficiency.
4.2 Experimental Apparatus

Figure 4.1 shows the diagram of the experimental facilities of diesel engine test rig. The test rig is composed of a super charged diesel engine, a dynamo meter to study the engine performance and a turbocharger to study the centrifugal compressor performance. The specification of diesel engine is shown in Table 4.1. The exhaust gas from the engine is used to drive the radial turbine of turbocharger, which drives the centrifugal compressor. Detailed of the test compressor is discussed in the numerical simulation section. In the present study, only the performance of compressor has been studied. The effect of turbocharger on the engine performance, however, has not been studied yet.

![Figure 4.1 Experimental facilities of diesel engine test rig](image)

Table 4.1 Specifications of Test Engine

<table>
<thead>
<tr>
<th>Specifications</th>
<th>Specifications</th>
<th>Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
<td>4 Cylinder 4 Stroke Water-cooled Diesel</td>
<td>Bore × Stroke [mm]</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>17.5</td>
<td>Max. output [KW(PS)/rpm]</td>
</tr>
<tr>
<td>Combustion chamber</td>
<td>Direct injection</td>
<td>Max. torque [Nm(kgf/m)/rpm]</td>
</tr>
<tr>
<td>Valve mechanism</td>
<td>Double overhead camshaft</td>
<td>Total emission [cm³(L)]</td>
</tr>
</tbody>
</table>
Measuring apparatus for the compressor consists of the three statics pressure sensors, three thermocouples, a flow meter, a gap sensor, a flow control valve controlled by a stepper motor, a data acquitting unit and a personal computer for data sampling. One statics pressure sensors and one thermocouple were placed at the compressor inlet. Two statics pressure sensors and two thermocouples were placed at upstream and downstream of the flow meter before the flow control valve. Flow control value is used to control the mass flow rate of the compressor.

The atmospheric air enters the compressor of the turbocharger and compressed air leaved for the atmosphere after passing through the flow control valve. In the experimental campaign, the revolution of the engine was increased step by step until the necessary revolution of the compressor reached. The compressor was operated at a constant impeller tip Mach number ($M_t$) of 0.46, 0.579 and 0.695 respectively. The flow rate was decreased to surge limit from the fully open valve condition.

4.3 Research Background

A three-dimensional model of test compressor with the recirculation flow type casing and its photograph is shown in Fig. 4.2. In the previous research [34], numerical simulation showed the effectiveness of the impeller of an automotive turbocharger with the recirculation flow type casing over that of without and proposed a baseline the recirculation flow type casing arrangement. Three-dimensional flow separation zone was projected to the meridional plane as shown in Fig. 4.3 in the case of the mass flow rate of $G/G^*=0.382$, in which $G^*$ is the choke flow rate at the inlet suction pipe.

![3D Model of the test compressor with the recirculating flow type casing and its photo](image)

In the case without the recirculation flow type casing shown in Fig. 4.3-(a), the reverse flow zone is seen on the shroud casing wall near the splitter blade leading and the leading edge separation of the main blade. In the case with recirculation flow type casing shown in Fig. 4.3-
(b), the reverse flow region on the shroud casing near the splitter blade leading edge is sucked into the casing. The sucked flow merges with the incoming main flow at upstream of the impeller inlet and the leading edge separation is reduced successfully.

Figure 4.3 Projection of the reverse flow zone at G/G*=0.382 and M_t = 0.695

4.3.1 Experimental Analysis of Recirculation Flow Type Casing Treatment in Centrifugal Compressors

Figure 4.4 Comparison of the characteristics curves between the cases with and without a baseline recirculation flow type casing (Experiment)
Figure 4.4 shows the experimental comparison of the characteristics curve between the cases with and without baseline recirculation flow type casing. The total pressure ratios of the compressor are presented as a function of the discharge mass flow rate under three different impeller inlet tip Mach number (Mt): 0.46, 0.579 and 0.695 respectively. As visible in Fig. 4.4 the baseline recirculation flow type casing was successfully replaced the normal casing of the turbocharger compressor without deteriorating the compressor’s performance. The flow rate range of compressor, moreover, has increased to the side of small flow rate by 102.5% at Mt = 0.695 after installing the baseline recirculation flow type casing.

The reason why there was small range enhancement to the small flow rate is explained in Fig. 4.5. Figure 4.5 shows the comparison of the projection of reverse flow zone between the cases with and without recirculation flow type casing shape at very small flow rate (G/G* = 0.297). Although the baseline recirculation flow type casing was successfully enable to suck the low energy fluid when the mass flow rate decreases, the numerical simulation shows that it does not succeed for the entire flow rate range. As shown in Fig. 4.5, when the mass flow rate become very small (G/G* = 0.297), the recirculation flow type casing was unable to suck the most of the low energy fluid and hence made very small flow rate range enhancement (Fig. 4.4). The previous study, hence, shows possibility of the application of the recirculation flow type casing treatment in small turbocharger’s compressors. An optimization is required to search the optimum shape of the recirculation flow type casing to improve the performance and enhance the flow rate range.

4.3.2 Technical Issues in Casing Treatment Technique

Recirculation flow type casing treatment is widely applied in centrifugal compressors in order to achieve a wider flow rate range. Two ring grooves on a suction pipe and a shroud casing wall are connected by an annular bypass passage. At small discharged flow condition, recirculation flow is formed based on pressure difference between two ring slots. Reverse flow
on the shroud wall is successfully sucked into the bleed slot, and recirculation flow merges with incoming flow at impeller inlet, excessive flow incidence is reduced by an increased flow. Thus, recirculation flow type casing treatment is expected to have two effects simultaneously, that is suppression of reverse flow on the shroud wall and improvement of flow incidence at impeller inlet. The recirculation flow type casing treatment has advantages for flow range enhancement, however, technical issues such as an efficiency drop still remain. Adiabatic efficiency of a compression system is defined as following equation.

\[
\eta_{ad} = \frac{H_{2r} - H_1}{H_2 - H_1}
\]  

(1)

where \( \eta_{ad} \) is adiabatic efficiency, \( H \) is enthalpy, and subscript states the condition of pressure level in the compression system as shown in Fig. 4.6. It is very important to have a pressure rise without having entropy rise for the compression system. However, in the case of the casing treatment, recirculation flow needs to take an irreversible process, such as, friction and mixing. These losses are a source of entropy rise, and thus, the compressor meets efficiency drop in the case of the recirculation flow type casing treatment.

Another technical issue is high pre-whirl motion by recirculation flow. Velocity distributions at the impeller inlet are shown in Fig. 4.7 in the case of the small mass flow rate of \( G/G^*=0.382 \). The meridional component of velocity \( V_m \) and the circumferential component of the absolute velocity \( V_u \) are normalized by the impeller tip speed \( U_2 \). In the case with casing treatment, meridional component of velocity is increased in the whole span region. The increment of the meridional velocity is based on the recirculation flow from the upstream groove and a large circumferential velocity is extended from the tip to the 80% span. Suitable pre-whirl is acceptable to suppress the leading edge separation, however, too large pre-whirl causes negative flow incidence at the shroud side. It is clear that, in order to obtain a flat inlet velocity distribution, the circumferential component of velocity due to the recirculation flow should be suppressed.

Figure 4.6 Ideal and real processes in compression system
4.4 Numerical Simulations

The flow was simulated for the model impeller of an automotive turbocharger. Meridional section of the test compressor with a standard recirculation flow type casing is shown in Fig. 4.8. The open-shrouded impeller has six main blades and six splitter blades; the total number of blade is twelve. The exit and inlet radius of the impeller are 25.5 and 19.2 mm respectively. The simulation was carried out at a constant impeller rotational speed of 120,000 rpm which corresponds to the inlet tip Mach number of \( M_t = 0.695 \), and under the condition
without scroll casing for simplicity. The inlet blade angles are 37 and 68 degrees at the blade root and at the blade tip respectively from the circumferential direction. The exit blade angle is 45 degrees and the exit blade height is 4.5 mm. The clearance between the impeller blade tip and the shroud wall is 0.3 mm. The vaneless diffuser is located at the downstream of the impeller with the exit radius of \( R = 2.0 \). The pinch diffuser was applied until the radius ratio of \( R = 1.1 \) in order to suppress the reverse flow on the shroud casing wall.

With respect to the recirculation flow type casing, the upstream slot is placed at 18 mm upstream of the impeller inlet with the width of 5 mm. The downstream slot is placed near the splitter blade leading edge with 2 mm width. The two slots are connected by the annular bypass with 5 mm height.

The three-dimensional turbulent flow was calculated by using the commercial CFD code CFX from ANSYS together with the k-\( \omega \) based SST (Share Stress Transport) turbulence model. In the calculation, one of the impeller passages with the main and the splitter blade is selected. The impeller passage is referenced to the relative frame. The 3D computational mesh is shown in Fig. 4.9-(c).

![Figure 4.9 3D Computational Mesh](image)

(a) Main passage (R1)  (b) Groove (S1)  (c) Combined

4.5 Validation of Numerical Simulations

To prove the reliability of the numerical prediction prior to the optimization, the results of the numerical simulation were validated by experimental results (Fig. 4.10). Static pressure ratio between diffuser exit and inlet boundary \( (P_{s4}/P_{t0}) \) are presented as a function of the normalized mass flow rate \( (G/G^*) \). As visible in Figure 4.10 the numerical results are in fair agreement with the experimental results in the entire operating range. Percentage of error at the design flow rate \( G/G^* = 0.425 \) is 1.5%.
Figure 4.10 Validation of numerical simulation comparing with characteristics curves between experiment and CFD at Mt = 0.695

4.6 Study on the Performance Improvement of Centrifugal Compressors for Small Turbocharger

In this study, the recirculation flow type casing shape was optimized based on the existing baseline shape. The recirculation flow type casing shape was parameterized by upstream slot width \( p_1 \), upstream slot position \( p_2 \), downstream slot width \( p_3 \), downstream slot position \( p_4 \), annular bypass height \( p_5 \) and annular bypass position \( p_6 \) as shown in Fig. 4.11. These six independent sets of parameters (Table 4.2) are the optimization parameters which can be modified by the optimization program. This optimization aimed at improving the adiabatic efficiency of the compressor at two different operating points as shown in Fig. 4.12 at the design mass flow rate \( G/G^* = 0.425 \) and at near surge mass flow rates \( G/G^* = 0.297 \). The objective functions are solved as a minimized problem as shown in Eqn. 4.1, which means that both adiabatic efficiencies were treated as negative quantities. Both objectives were subject to a constraint that computed residue of numerical simulation must be less than \( 10^{-3} \) as described in Eqn. 4.2.

Objective

Minimize \(-\eta_{ad} \) at \( G/G^* = 0.425 \)  
Minimize \(-\eta_{ad} \) at \( G/G^* = 0.297 \)

Constraint

\[
\text{RMS Residual} < 10^{-3}
\]
Table 4.2 Optimization parameters of design variables for recirculating flow type casing

<table>
<thead>
<tr>
<th>Design Variables</th>
<th>Initial</th>
<th>Minimum</th>
<th>Maximum</th>
</tr>
</thead>
<tbody>
<tr>
<td>$p_1$ [mm]</td>
<td>4.0</td>
<td>2.0</td>
<td>8.0</td>
</tr>
<tr>
<td>$p_2$ (z) [mm]</td>
<td>37.5</td>
<td>45.0</td>
<td>30.0</td>
</tr>
<tr>
<td>$p_3$ [mm]</td>
<td>2.5</td>
<td>1.0</td>
<td>4.0</td>
</tr>
<tr>
<td>$p_4$ (z) [mm]</td>
<td>16.5</td>
<td>18.3</td>
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<td>$p_5$ [mm]</td>
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<td>2.0</td>
<td>6.0</td>
</tr>
<tr>
<td>$p_6$ (r) [mm]</td>
<td>24.5</td>
<td>25.0</td>
<td>26.0</td>
</tr>
</tbody>
</table>
4.6.1 Result of Optimization Process

In Fig. 4.13, the 2-dimensional objective space after iteration process is presented. It shows the adiabatic efficiency at design point ($\eta_{ad}$ at $G/G^* = 0.425$) with respect to adiabatic efficiency at small flow rate ($\eta_{ad}$ at $G/G^* = 0.297$). Each symbol represents one design which has been analyzed by CFD. The white circle (DOE) and black circle (DOECnstr) marks are the initial database of the DOE process, prior to the optimization, while the yellow (IT) and red (ITCnstr) circle marks were generated during the optimization process. The black circle and red circle marks represent the designs which satisfy the constraints (Eqn. 4.2). Both objectives are improving towards the lower left hand corner of Fig. 4.13. The selected optimized recirculation flow type casing shape is indicated in Fig. 4.13, which was regarded as the best compromise between both objectives.

![Figure 4.13 Optimization results of 2-dimensional objective space](image)

4.6.2 Comparison of the Performance between Baseline and Optimized Casing Treatments

The optimized configuration of the casing treatment comparing with that of baseline is shown in Fig. 4.14-(a) and (b). Comparing with the baseline, the width of optimized recirculation flow type casing shows smaller width, the upstream slot is located on the farther upstream, and the downstream slot is located on the farther downstream. Table 4.3 shows the comparison of the design variables between the baseline and optimized recirculation flow type casings as well as the minimum and maximum ranges during the optimization process.
Figure 4.14 Comparison of meridional shape between baseline and optimized recirculation flow type casing arrangement

Figure 4.15 Comparison of adiabatic efficiency and recirculation flow rate between baseline and optimized (CFD)
Table 4.3 Comparison of design variables between baseline and optimized recirculation flow type casing shape

<table>
<thead>
<tr>
<th>Design Variables</th>
<th>Initial</th>
<th>Minimum</th>
<th>Maximum</th>
<th>Optimized</th>
</tr>
</thead>
<tbody>
<tr>
<td>p₁ [mm]</td>
<td>4.0</td>
<td>2.0</td>
<td>8.0</td>
<td>2.9</td>
</tr>
<tr>
<td>p₂ (z) [mm]</td>
<td>37.5</td>
<td>45.0</td>
<td>30.0</td>
<td>44.9</td>
</tr>
<tr>
<td>p₃ [mm]</td>
<td>2.5</td>
<td>1.0</td>
<td>4.0</td>
<td>1.0</td>
</tr>
<tr>
<td>p₄ (z) [mm]</td>
<td>16.5</td>
<td>18.3</td>
<td>14.3</td>
<td>14.3</td>
</tr>
<tr>
<td>p₅ [mm]</td>
<td>4.5</td>
<td>2.0</td>
<td>6.0</td>
<td>4.2</td>
</tr>
<tr>
<td>p₆ (r) [mm]</td>
<td>24.5</td>
<td>25.0</td>
<td>26.0</td>
<td>26.0</td>
</tr>
</tbody>
</table>

The adiabatic efficiency at impeller exit $\eta_{ad2}$ and the recirculation flow rate $m_{IR}$ under various discharged mass flow rate are shown in Fig. 4.15. The adiabatic efficiency shows good improvement in wide operating condition, the recirculation flow rate becomes smaller in the case of optimized casing than in the case of baseline.

In Fig. 4.16, comparison of the projection of reverse flow zone between baseline and optimized casing treatment for small flow rate from $G/G^*=0.425$ to $G/G^*=0.297$ is shown. The left side of each pair represents the reverse flow zone of the baseline recirculation flow type casing shape while the right side represents the optimized recirculation flow type casing shape. It can be clearly seen from Fig. 4.16 that the optimized shape was successfully able to suck the low energy fluid into the casing.

4.6.3 Sensitivity Analysis of the Parameters as a Function of Adiabatic Efficiency

Sensitivity analysis of the optimization parameters of the recirculation flow type casing shape has been performed as a function of adiabatic efficiency at both design and small flow rate. Figure 4.17 shows the sensitivity analysis at design flow rate and Figure 4.18 shows at small flow rate conditions. The analysis is based on the results which satisfy the constraints Eqn (2), that are $DOECnstr$ and $ITCnstr$.

At design flow rate condition, downstream slot width ($p₃$) and downstream slot position ($p₄$) has higher influence on the efficiency than other parameters. As can be seen clearly in Fig. 4.17-c, the efficiency at the design flow rate is maximum when the downstream width is minimum. The highest efficiency at design flow rate occurs when the downstream position ($p₄$) is 15.43 mm from impeller back plate, which is located at the midway between the leading edges of main and splitter blades, although the optimized value is chosen at $p₄ = 14.3$ mm as shown in Fig. 4.17-d. It can be concluded that there is an optimum position for downstream slot to have a higher adiabatic efficiency at design flow rate. Another possible potential parameter is annular bypass position ($p₆$). Figure 4.17-f shows that the larger radius of annular bypass position provides higher efficiency at design flow rate. From this fact it can be conclude that more investigations are required to get an optimum radius for annular bypass position.

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Baseline Optimized

(a) $G/G^* = 0.425$

(b) $G/G^* = 0.318$

(c) $G/G^* = 0.297$

Figure 4.16 Comparison of projection of the reverse flow zone between baseline and optimized at $Mt = 0.695$ (CFD)
Figure 4.17 Sensitivity analysis of the parameters as a function of adiabatic efficiency at design flow rate ($G/G^* = 0.425$) and $Mt = 0.695$
At small flow rate condition, upstream slot position ($p_2$), downstream slot width ($p_3$) and downstream slot position ($p_4$) are seem to be dominant parameters on the adiabatic efficiency. From Fig. 4.18-b, upstream slot position ($p_2$) provides highest efficiency at $p_2 = 42$.
mm from the impeller back plate which closes to the chosen optimized position. The highest efficiency at small flow rate can be obtained when the downstream slot width \( p_3 \) become less than 1.6 mm as shown in Fig. 4.18-c. It will be difficult to confirm the smallest possible value due to manufacturing difficulty and 1 mm is chosen for the optimized shape combining with other parameters. Downstream slot position also has similar trend as seen in Fig. 4.18-d. When the downstream slot position \( p_4 \) is 15.43 mm from impeller back plate, the midway between the two leading edges, the higher efficiency can be obtained. Further moving up to the position of leading edges of splitter blade shows not much changes in efficiency.

4.6.4 Effect of Recirculation Flow Rate on the Efficiency

![3D Normalized entropy and recirculation flow rate graph](image)

Figure 4.19 3D Normalized entropy and recirculation flow rate \((G/G^* = 0.297 \text{ and } M_t = 0.695)\) (CFD)

The recirculation flow rate seems one of a dominant parameter on the adiabatic efficiency. A sensitivity analysis is performed in order to make clear the dependency of the recirculation flow rate on the adiabatic efficiency. All the optimization candidates are plotted in Fig. 4.19. Black solid circles show the case of DOE (Design of Experiment) which is for the initial database set of optimization process. Open circles show the result from optimization iteration process. The recirculation flow rate shows maximum efficiency at \( m_{IR} = 0.09 \). It is
found that the larger flow rate or smaller flow rate of the recirculation flow deteriorates the adiabatic efficiency, the optimizer found the optimized recirculation flow rate for maximizing the adiabatic efficiency.

Figure 4.20 Projection of reverse flow region on meridional plane with different casing arrangements ($G/G^*=0.297$) at $Mt = 0.695$ (CFD)

Three configurations of recirculation flow type casing treatment are selected in order to compare the reverse flow region. As shown in Fig. 4.20, the reverse flow region is projected on
meridional plane. Figure 4.20-a "IT001_IND004" is the case of optimized recirculation flow rate, Figure 4.20-b "IT002_IND000" is the case of over recirculation flow is formed, and Figure 4.20-c "IT007_IND002" is the case of under recirculation flow is formed.

In order to simplify the choice of the optimized recirculation flow rate of selected casing treatments, comparison of entropy distribution of these casing treatments at the impeller inlet was introduced as shown in Fig. 4.21. The blue line represented the steam wise entropy distribution of "IT001_IND004" while the green line represented for "IT002_IND000" and the purple line for "IT007_IND002". In the upstream of the splitter blades the entropy distribution was directly proportional to the recirculation flow rate which means the low recirculation flow rate seemed to be better. However, the entropy distribution in the downstream of splitter blades showed different result. The entropy distribution of both casing treatments with over and under recirculation flow rates became larger than that of optimized shape. With considering the entropy distribution at impeller inlet, high entropy fluid based on the large amount of recirculation flow merged with main flow at impeller inlet, on contrary, large entropy rise could be seen at main blade. In the case of under recirculation flow rate, it must be coming from separated flow region as shown in Fig 4.20-(c) without enough sucking low energy fluid from downstream slot.

4.6.5 Conclusions

The recirculation flow type casing treatment for the centrifugal compressor of turbocharger is optimized in order to improve the efficiency. The optimization system successfully predicted the geometry of optimized recirculation flow type casing shape without costing tons of simulation time. The optimized casing shows good improvement of adiabatic efficiency in wide flow rate range. From the sensitivity analysis of the recirculation flow type casing design parameters, downstream slot width, downstream slot position has large impact on the adiabatic efficiency at both design and small flow rate. Optimum position can be found near the leading edge of the splitter blade. Upstream slot position and annular bypass position also has greater role than upstream slot width and annular bypass width in the effect of adiabatic efficiency. The sensitivity analysis of $m_{IR}$ showed that the recirculation flow rate is one of the dominant parameters for improving the adiabatic efficiency of the compressor. It is found that the optimized shape of recirculation flow type casing has optimized recirculation flow rate in order to maximize the adiabatic efficiency.

4.7 Study on Flow Range Enhancement of Centrifugal Compressor for Small Turbocharger

In this study, the recirculating flow type casing shape was optimized based on the existing baseline shape. The recirculating flow type casing shape was parameterized by upstream slot width (p1), upstream slot position (p2), downstream slot width (p3), downstream slot position (p4), annular bypass height (p5) and annular bypass position (p6) as shown in Fig.
4.11. These six independent sets of parameters (Table 4.4) are the optimization parameters which can be modified by the optimization program.

Table 4.4 Optimization parameters of design variables for recirculating flow type casing

<table>
<thead>
<tr>
<th>Design Variables</th>
<th>Baseline</th>
<th>Minimum</th>
<th>Maximum</th>
</tr>
</thead>
<tbody>
<tr>
<td>p1, [mm]</td>
<td>5.0</td>
<td>2.0</td>
<td>7.0</td>
</tr>
<tr>
<td>p2, (z) [mm]</td>
<td>-35.6</td>
<td>-50.0</td>
<td>-30.0</td>
</tr>
<tr>
<td>p3, [mm]</td>
<td>2.0</td>
<td>1.0</td>
<td>7.0</td>
</tr>
<tr>
<td>p4, (z) [mm]</td>
<td>-16.0</td>
<td>-18.3</td>
<td>-13.0</td>
</tr>
<tr>
<td>p5, [mm]</td>
<td>5.0</td>
<td>3.0</td>
<td>8.0</td>
</tr>
<tr>
<td>p6, (r) [mm]</td>
<td>23.5</td>
<td>25.7</td>
<td>30.0</td>
</tr>
</tbody>
</table>

This optimization aimed at increasing the operating flow range to small flow region and improving the adiabatic efficiency of the compressor as shown in Fig. 4.22 (a) and (b). In search of the operating flow range for each design point, pressure ratio was used as an indicator of the surge occurrence. This is illustrated in Fig. 4.22-a showing the pressure ratio as a function of the mass flow rate and the ratio of two successive pressure ratios was applied as constraint. The objective functions are solved as a minimized problem as shown in Eqn. 4.3, which means that adiabatic efficiency at design flow rate was treated as negative quantities. Both objectives were subject to a constraint that computed residue of numerical simulation must be less than $10^{-3}$ as described in Eqn. 4.4. The combination of these objectives and constraints means that the optimizer tries to find the recirculating flow type casing shape with better performance in wider operating flow rate range.
Objective

Minimize $-\eta_{ad}$ at $G/G^* = 0.425$

Maximize operating flow range OFR

Constraint

RMS Residual $< 10^{-3}$

4.7.1 Result of Optimization Process

In Fig. 4.23, the 2-dimensional objective space after iteration process is presented. It shows the adiabatic efficiency at design point ($\eta_{ad}$ at $G/G^*=0.425$) with respective to the operating flow range. Each symbol represents one design which has been analyzed by CFD. The black circle (DOE) marks are the initial database of the DOE process, prior to the optimization, while the red (IT) circle marks were generated during the optimization process. Both represent the designs which satisfy the constraints (Eqn. 4.4). Both objectives are improving towards the lower left side corner of Fig. 4.23. The selected recirculation flow type casing shape with widest operating flow range (OFR) and that with maximum efficiency (EFF) at design flow rate is indicated in Fig. 4.23.

Figure 4.23 Optimization results of 2-dimensional objective space
4.7.2 Comparison of the Performance between Baseline, OFR and EFF Casing Treatment

![Diagram of baseline, OFR, and EFF casing treatments]

Figure 4.24 Comparison of meridional shape between baseline, OFR and EFF recirculation flow type casing treatment

<table>
<thead>
<tr>
<th>Table 4.5 Comparison of design variables between Baseline, OFR and EFF</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Design Variables</strong></td>
</tr>
<tr>
<td>----------------------</td>
</tr>
<tr>
<td>p1, [mm]</td>
</tr>
<tr>
<td>p2, (z) [mm]</td>
</tr>
<tr>
<td>p3, [mm]</td>
</tr>
<tr>
<td>p4, (z) [mm]</td>
</tr>
<tr>
<td>p5, [mm]</td>
</tr>
<tr>
<td>p6, (r) [mm]</td>
</tr>
</tbody>
</table>

![Diagram of projection of reverse flow zone]

Figure 4.25 Comparison of projection of reverse flow zone between baseline, OFR and EFF casing treatment \( \frac{G}{G'} = 0.236 \) at \( M_r = 0.695 \) (CFD)

The configurations of the OFR and EFF casing treatment with that of baseline is shown in Fig. 4.24. Comparing the baseline shape, the upstream slot width (p1) of OFR shows slightly
smaller width, while EFF casing shapes shows a bit wider width, the upstream slot is located on farther upstream in the same position (p2) for both OFR and EFF shape. The downstream slot width (p3) of both OFR and EFF casing shapes shows same width with minimum possible value, however opposite preference shows for the downstream slot position (p4). The OFR shape demands larger annular bypass position (p6), while demanding smaller annular bypass width (p5) than the baseline and EFF shape. Table 4.5 shows the comparison of the design variables between baseline, OFR and EFF recirculating flow type casing as well as the minimum and maximum limits during the optimization process.

In Fig. 4.25, comparison of the projection of reverse flow zone between baseline, OFR and EFF casing treatment for small flow rate at \( \frac{G}{G^*} = 0.297 \) is shown. It can be seen from Fig. 4.25 that the EFF casing shape has upper side in sucking the low energy fluid into the slot than OFR casing shape at near surge flow rate. The adiabatic efficiency at impeller exit (\( \eta_{ad2} \)) and the static pressure ratio (\( \frac{P_s}{P_t} \)) under various discharged mass flow rate are shown in Fig. 4.26. The optimizer has successfully enhanced the operating flow range to 107% for EFF and 110% for OFR casing treatment. The adiabatic efficiency of EFF casing treatment shows good improvement in wide operating condition comparing to the OFR casing treatment which shows fair adiabatic efficiency distribution.

![Figure 4.26 Comparison of adiabatic efficiency and static pressure ratio between baseline, OFR and EFF (CFD)](image-url)
4.7.3 Sensitivity Analysis of the Parameters as a Function of Operating Flow Range and Adiabatic Efficiency

- (a) Upstream slot width
- (b) Upstream slot position
- (c) Downstream slot width
- (d) Downstream slot position
- (e) Annular bypass width
- (f) Annular bypass position

Figure 4.27 Sensitivity analysis of the parameters as a function of operating flow range (Mt = 0.695)
Figure 4.28 Sensitivity analysis of the parameters as a function of adiabatic efficiency at design flow rate (Mt = 0.695)
Sensitivity analysis of the optimization parameters of the recirculation flow type casing shape has been performed as a function of operating flow range and adiabatic efficiency at design flow rate ($G/G^* = 0.425$). Figure 4.27 shows the Sensitivity analysis as a function of operating flow range and Figure 4.28 shows as a function of adiabatic efficiency. The analysis is based on the results which satisfy the constraints Eqn. 4.4. Each symbol represents one design which has been analyzed by CFD. The black circle (DOE) marks are the initial database of the DOE process, prior to the optimization, while the red (IT) circle marks were generated during the optimization process.

As can be seen in Fig. 4.27 upstream slot width ($p_1$), upstream slot position ($p_2$) and downstream slot width ($p_3$) seem to have higher influence on the operating flow range (OFR) than other parameters. The operating flow range is found to be widest when the upstream slot position is farthest from impeller back plate, which is located at the leading edge of the main blade as shown in Fig. 4.27-b and Fig. 4.24-b. The widest operating flow range is found when the upstream slot width ($p_1$) is 4.7 mm, which is midway between the predefined limit as shown in Fig. 4.27-a. It can be concluded that there is an optimum position for upstream slot width to obtain the wider operating flow range. The downstream slot width ($p_3$) has the similar trend with upstream slot position ($p_2$) and the widest operating flow range is found at the minimum downstream slot width ($p_3$). It should be noticed that farthest downstream slot position ($p_4$) also provides the widest operating flow range.

The adiabatic efficiency at the design flow rate was influenced by one more parameter, downstream slot position ($p_4$) than the influence parameters on the OFR. The influences on the efficiency of upstream slot position and downstream slot width are same as that of on the OFR. The highest efficiency is denoted at the farthest upstream slot position ($p_2$) and at the minimum downstream slot width ($p_3$) as can be seen in Fig. 4.28-b and 4.28-c. For the smallest downstream slot width, 1 mm is chosen for both OFR and EFF casing treatment and choosing further smaller value will make the difficulty in manufacturing. The wider upstream slot width ($p_1$), however, provides the higher efficiency, as shown in Fig. 4.28 (a), unlike to the influence on the OFR and further study might be required in order to obtain the optimum width. The higher efficiency is denoted when the downstream slot position ($p_4$) move to the downstream slot side as in Fig. 4.28-d. In other words, the nearest downstream slot position ($p_4$) from the compressor back plate provides the highest adiabatic efficiency which is reverse effect to the operating flow range.

4.7.4. Conclusions

The recirculating flow type casing treatment for the centrifugal compressor of turbocharger is optimized in order to improve the efficiency and operating flow range. The optimization system successfully predicted the geometry of two recirculating flow type casing shape without costing tons of simulation time. The optimizer has successfully enhanced the operating flow range to 107% for EFF and 110% for OFR casing treatment. The EFF casing shows good improvement of adiabatic efficiency in wide flow rate range. From the sensitivity
4.8 Study on Guide Vane in Recirculation Flow Type Casing Treatment by Applying Global Optimization Technique

In the present study, in order to suppress too large pre-whirl at the shroud side of impeller inlet, guide vanes were installed in the annular passage of the recirculation device as shown in Fig. 4.29. However, it is difficult to design the guide vane that is because flow inside of the annular passage is complicated, and the blade has a large number of design of freedom.

Global search system with GA (Genetic Algorithms) supported by meta-model of ANN (Artificial Neural Network) is applied in this study for design of the guide vane. The optimization aims at improving efficiency at design point and at small flow rate. Four design parameters describing the blade angle of the guide vane were introduced. Internal flow of the compressor with optimized guide vane was discussed at the point of view the inlet distortion and recirculation flow rate.

Figure 4.29 Guide vane inside of the recirculation flow type casing treatment

4.8.1 Objectives and Design Parameters

The optimization is aiming to improve the adiabatic efficiency at two different operating points of the design flow rate (G/G*=0.425) and small flow rate (G/G*=0.297). Multi-point optimization has performed in this study, optimizer try to find the efficiency at two different flow rate conditions as much as possible as shown in Fig. 4.30. Four design variables of the blade angle are selected. Table 4.6 shows design of freedom in four blade angles. $\beta_1$, $\beta_2$ and $\beta_3$
are the blade angle from a leading edge to a trailing edge measured from the rotating axis. The blade angle $\beta_i$ is an inclined angle measured from radial direction. The guide vane blade assumed as a two-dimensional blade, that means the blade angle distribution is same between inner and outer side walls of the annual passage. The baseline blade is flat plate type blade installed in radial direction. A minimum and a maximum values of design parameters are set within a feasible region.

![Diagram showing adiabatic efficiency at different flow rate conditions](image)

Figure 4.30 Objective function at different flow rate conditions

<table>
<thead>
<tr>
<th>Design Variables</th>
<th>$\beta_1$</th>
<th>$\beta_2$</th>
<th>$\beta_3$</th>
<th>$\beta_4$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Baseline</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Minimum</td>
<td>-15</td>
<td>0</td>
<td>-15</td>
<td>-27</td>
</tr>
<tr>
<td>Maximum</td>
<td>30</td>
<td>15</td>
<td>15</td>
<td>50</td>
</tr>
</tbody>
</table>

4.8.2 Optimization Results

In Fig. 4.31, the 2-dimensional objective space after 20 iterations is presented. It shows the adiabatic efficiency at design point ($G/G^*=0.425$) with respect to adiabatic efficiency at small flow rate ($G/G^*=0.297$). Each symbol represents one design which has been analyzed by CFD and is satisfying all constraints. The grey solid circle marks are the initial database from by the DOE process, prior to the optimization, while the red solid marks were generated during the optimization process. The green circle mark indicates the performance of the baseline guide vane. Both objectives are improving towards the lower left hand corner of Fig. 4.31. Typical of multi-objective optimization problems where both objectives are conflicting, a set of non-dominated optimal geometries was found. These designs are indicated by blue solid marks and are located on the Pareto front where one objective cannot be improved without worsening the
other. The selected optimized guide vane is indicated in Fig. 4.31 as a blue circle, which was regarded as the best compromise between both objectives. Three-dimensional shape of the selected optimum guide vane is shown in Fig. 4.32. The optimized guide vane has the blade angle of a counter direction of the rotating impeller.

Figure 4.31 Optimization result in two objective design space

Figure 4.32 Optimized guide vane
4.8.3 Flow Characteristics of Optimized Guide Vane

Comparison of adiabatic efficiency $\eta_{ad2}$ and recirculation mass flow rate $m_{IR}$ as a function of a discharged mass flow rate are obtained by numerically as shown in Fig. 4.33. At small flow rate conditions, adiabatic efficiency is successfully improved in the case of the optimized guide vane. The recirculation flow rate is almost same between the cases of baseline and optimized guide vane. From the database of the optimization system, sensitivity of the adiabatic efficiency at small flow rate is sorted out as a function of the recirculation flow rate as shown in Fig. 4.34. It is noticed that the recirculation flow rate is possible to change by the shape of the guide vane. The adiabatic efficiency at small flow rate is depends of the recirculation flow rate, and there seems to be an optimum recirculation flow rate for the efficiency. By chance, the recirculation flow rate in the case of the baseline is close to the optimized recirculation flow rate, however, the adiabatic efficiency shows the difference of 0.7%.

It is indicated that the efficiency is not only depended on the recirculation flow rate. Figure 4.35 is velocity distribution at impeller inlet. Pre-whirl based on recirculation flow is successfully improved in the case of the optimized guide vane. It is noticed that the improvement of the adiabatic efficiency must be based on changing in angler momentum (Euler’s Head). Moreover, flow incidence distribution, as shown in Fig. 4.36, improved between hub and shroud. Improvement of the efficiency is supported by proper distribution of flow at impeller inlet by applying the optimized guide vane.

![Figure 4.33 Comparison of efficiency and recirculation flow rate](image-url)
Figure 4.34 Sensitivity of the adiabatic efficiency at off-design condition to the recirculation flow rate $m_{IR}$

Figure 4.35 Comparison of velocity distribution between hub and shroud at impeller inlet ($G/G^*=0.297$)
Figure 4.36 Comparison of flow incidence distribution between hub and shroud at impeller inlet (G/G*=0.297)

4.8.4 Conclusions

Global optimization has performed in the design process of the guide vane installed in the recirculation flow type casing treatment for a centrifugal compressor of turbochargers. In the case of the optimized guide vane, it is clearly indicated that there is a possibility to improve the adiabatic efficiency at small flow rate conditions. Excessive pre-whirl based on recirculation flow is suppressed by the optimized guide vane, and velocity distortion at impeller inlet has improved. It is found that improvement of the efficiency in the case of the optimized guide vane is supported by proper distribution of flow at impeller inlet.
5

STUDY ON CENTRIFUGAL COMPRESSOR COMPONENTS OPTIMIZATION IN A COMMERCIAL TURBOCHARGER

5.1 Overview

Recently casing treatment for centrifugal compressors of automobile turbochargers receives considerable attention among the turbomachinery manufactures and researchers. However, the complex flow phenomena inside the casing treatment and interaction between the impeller and casing treatment has not been fully studied yet. Adiabatic efficiency, pressure rise and operating flow range enhancement of centrifugal compressor with casing treatments are conflicting and improving one parameter could cause a depletion in others. Therefore, it is challenging to design a compressor which provide a broader map width and higher pressure ratio, while simultaneously increasing the efficiency characteristics.

Many attempts so called active control methods such as installing inlet guide vane, using variable geometry or taking motor assist, have been made to enhance the map width and to improve the performance of centrifugal compressors. However, these are impractical for centrifugal compressors for turbochargers due to limited space and small scale geometry and casing treatment becomes one of the methods that has been widely investigated. A successful application of recirculation flow type casing treatment, comprising two grooves connected by annular bypass, for turbocharger in a marine diesel engine was first investigated by Fisher [31] in the late 1980s and improved the performance in surge flow and stabilized the inducer flow considerably. Recently, Dickmann, H. P. et al [32] developed the inducer casing bleed system which is one kind of recirculation type casing treatment, analyzed the internal flow of the casing treatment
numerically and the compressor characteristics were compared with the experimental results under several flow rate condition. Iwakiri, Y. and Uchida, H. et al [33] also developed the self-recirculation flow type casing treatment for turbochargers. Their results showed that the inlet velocity distortion could be improved by the combination with the variable inlet guide vane. Zheng X. et al. [36, 40] thoroughly studied the non-axisymmetrical flow in a centrifugal compressor and widened the operating range by developing an asymmetrical flow control method using non-axisymmetrical self-recirculation casing treatment. Their results show that the non-axisymmetrical casing treatment has a certain influence on the performance and has a larger potential for stability improvement than the conventional casing treatment. Sivagnanasundaram S. et al [37-38] numerically and experimentally investigated the flow rate range enhancement and studied the inducer flow field under various bleed slot widths and vanes arrangement in the annular bypass. The wider bleed slot width could provide the wider flow rate range with adiabatic efficiency penalty. Chen, H. and Lei, V. M. et al [44] discussed the development of the casing treatment over two decades long research work and explored the underline mechanism that deliver the performance improvement. Stability near surge could be improved by removal of low momentum fluid near the shroud region and reducing strength of shock wave on the blade suction surface while compressor flow capacity near choke could be increased by additional flow entering the blade passage through annular bypass.

In the first study, an optimization design approach to recirculation flow type casing treatment for centrifugal compressors of turbochargers has been performed. The optimization code is a global exploring system based on a meta-model assisted evolutionary algorithm. The objective functions of the optimization are adiabatic efficiency at design flow rate and near surge flow rate condition. Several optimized shapes on a “Pareto front” are selected in order to discuss the effect of casing treatment at the point of view of suppression of flow separation and a reduction of leading edge flow separation. The influence of optimized recirculation flow rate on the optimization output is also discussed. Increase in adiabatic efficiency of optimized casing treatment at near surge flow rate is discussed by the streamwise relative velocity distribution, reversed flow zone, static entropy distribution and flow incidence at impeller inlet. Detailed sensitivity of design variables on the adiabatic efficiency at design and off-design conditions was carried out.

In the second study, global optimization was performed on three-dimensional shape of the guide vanes 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.
5.2 Baseline Test Compressor and Casing Treatment

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. 5.1. 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. 5.1). 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.

The experimental analysis of baseline centrifugal compressor between the cases with and without casing treatment was illustrated in Fig. 5.2 and Fig. 5.3. The analysis was performed at five different speeds of impeller. The pressure characteristics of the compressor are compared between the cases with and without casing treatment as can be seen in Fig. 5.2. In these comparisons, the static pressure rise and efficiency were normalized by respective ones at the design flow rate \( \frac{G}{G^*} = 0.479 \) 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. It can be seen 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 \( (M_t = 0.774, M_t = 0.871 \text{ and } M_t = 0.967) \) without serious deterioration at other flow rate conditions (Fig. 5.2). However, there is no considerable surge improvement at lowest speed.

![Schematic of end wall](image1.png)

![3D model](image2.png)

Figure 5.1  Test centrifugal compressor with casing treatment
The efficiency characteristics (Fig. 5.3) 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 (Mt = 0.871 and Mt = 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.

5.3 Numerical Method

In the optimization process, numerical analysis has been carried out for a single flow passage of centrifugal compressor with casing treatment using CFX and ICEM CFD from ANSYS. The compressor consists of an inlet duct, baseline recirculating flow type casing treatment, vaneless diffuser and scroll casing. The open-shrouded impeller has six main blades and six splitter blades; the total number of blade pairs is six. The impeller was designed with backward exit angle in order to have stability in wider flow rate condition. The vaneless diffuser is located at the downstream of the impeller and the pinch diffuser is applied in order to suppress the reverse flow on the shroud wall casing.

The simulation was carried out at a constant impeller rotation speed of the inlet tip Mach number $M_t = 0.774$, and under the condition without scroll casing for simplicity. For this CFD setup, the inlet boundary conditions have been defined as the total atmospheric pressure and temperature. The discharge mass flow rate was applied at the pinch diffuser exit. The three dimensional turbulent flow was calculated by using the commercial CFD code ANSYS-CFX together with the k-ω based SST (Shear Stress Transport) turbulence model. One of the impeller passage with one main and one splitter blade was selected. The impeller passage is referenced to the relative frame.

![Figure 5.4 Computation grid viewed from meridional section](image)

The axisymmetric components of the compressor stage such as the suction pipe, impeller and vaneless diffuser consist of a structured mesh using Turbogrid, whereas the recirculation casing was developed with an unstructured mesh using ICEM CFD. The total number of elements for the impeller passage is about 2.5 million with a hexahedral mesh.
mesh. In order to have a smooth mesh distribution, an O-grid was applied around the blade. Two grooves and the annular passage are referenced to the absolute frame. The total number of elements for the groove passage is about 400,000 with a tetrahedral mesh to achieve a smooth mesh for a complex geometry. The average first cell spacing of $y^+ \leq 8$ was accepted because it was difficult to reach under the value of 5 and minimum mesh size was chosen based on balance of meshing time and cost. The multi-frame of reference consisting of the rotating domain of the impeller section and the stationary domain of the recirculation flow type casing section were adopted. Interfaces between two domains are coupled with the GGI (General Grid Interface) connection of ANSYS-CFX feature. The stage (mixing plane) interface was set between the stationary and rotating domains. The meridional section of the computational grid is shown in Fig. 5.4.

5.4 Validation of Numerical Simulation

![Figure 5.5 Comparison of compressor characteristics between experiment and CFD at Mt = 0.774](image)

Figure 5.5 Comparison of compressor characteristics between experiment and CFD at Mt = 0.774

Numerical results of static pressure rise and adiabatic efficiency over a wide flow rate range of the compressor with baseline casing treatment is compared with the experiment ones (Fig. 5.5). Static pressure rise ($P_{sd}$) and adiabatic efficiency were normalized ($\eta_{ad}$) by respective parameters at design flow rate condition as well as discharge mass flow rate ($G$) by choke mass flow rate ($G^*$) at the impeller inlet. Static pressure rise of the numerical simulation shows good agreement with the experimental results in the entire operating range. Percentage of error at the design flow rate $G/G^* =$
0.475 is 3.2%. Despite over estimation, adiabatic efficiency shows the same trend with the experimental one in the whole flow rate range. The reason why there is over estimation of adiabatic efficiency could be coming from the different configurations between the experiment and numerical simulation, such as symmetrical steady simulation and no scroll casing which can reduce friction loss, mixing loss and unsteady loss that occurs during the experiment campaign.

5.5 Study on Performance Improvement in Centrifugal Compressors by a Global Optimization Approach

In this study, the optimization of the shape of recirculation flow type casing treatment for a turbocharger centrifugal compressor (Fig. 5.1) has been performed in order to improve the adiabatic efficiency at design and low flow rate condition. The extended hub in front of the impeller’s spinner can be seen in Fig. 5.1-b because a small radius, which has no effect on the computational results, was defined on the hub surface in order to avoid meshing error in Turbogrid. The optimization aims at the improvement of the adiabatic efficiency at design and low flow rate conditions. Two individuals shape of the casing treatment have been selected out of the candidates which have been numerically investigated during the optimization process. The performance of the selected individuals are compared with the baseline casing treatment. Sensitivity of design parameters influence on the improvement of performance, effect of recirculation flow rate on the adiabatic efficiency are discussed. The improvement in performance at low flow rate condition is discussed by inlet velocity distribution and flow incidence at the impeller inlet, static entropy distribution and relative velocity deceleration in streamwise location.

5.5.1 Design Parameters, Objective Function and Constraint

In order to balance the recirculation flow rate throughout the entire flow rate range, the shape of recirculation flow type casing was optimized based on the existing baseline shape. The selected independent parameters in the process of optimization are upstream slot width, upstream slot position, downstream slot width, downstream slot position, bypass height and bypass position as shown in Fig. 5.6. Upstream slot position and downstream slot position are the axial positions measured from the impeller back plate while bypass position is the radial position measured from the rotating axis of the impeller. The optimization parameters normalized by the impeller inlet tip radius except upstream slot and downstream slot positions by the impeller axial length of the baseline casing treatment are expressed in Table 5.1 showing the range of parameters in the optimization process.
Figure 5.6 Control points of design parameters of the recirculation flow type casing

<table>
<thead>
<tr>
<th>Design Parameter</th>
<th>Minimum</th>
<th>Baseline</th>
<th>Maximum</th>
</tr>
</thead>
<tbody>
<tr>
<td>Norm: upstream slot width (p1)</td>
<td>0.141 (58.3%)</td>
<td>0.242</td>
<td>0.423 (174.9%)</td>
</tr>
<tr>
<td>Norm: upstream slot position (p2)</td>
<td>1.790 (95.5%)</td>
<td>1.874</td>
<td>1.931 (103.0%)</td>
</tr>
<tr>
<td>Norm: downstream slot width (p3)</td>
<td>0.028 (33.3%)</td>
<td>0.085</td>
<td>0.141 (166.7%)</td>
</tr>
<tr>
<td>Norm: downstream slot position (p4)</td>
<td>0.803 (82.6%)</td>
<td>0.972</td>
<td>1.057 (108.7%)</td>
</tr>
<tr>
<td>Norm: bypass height (p5)</td>
<td>0.223 (63.2%)</td>
<td>0.352</td>
<td>0.409 (116.0%)</td>
</tr>
<tr>
<td>Norm: bypass position (p6)</td>
<td>1.071 (100.0%)</td>
<td>1.071</td>
<td>1.304 (121.7%)</td>
</tr>
</tbody>
</table>

The optimization of the shape of recirculation flow type casing treatment aimed at improving the adiabatic efficiency of the compressor at two different operating points as shown in Fig. 5.7 at the design mass flow rate (G/G* = 0.479) and at off-design mass flow rates (G/G* = 0.259). Multi-point optimization has been performed in this study and both objective functions are solved as a minimized problem as shown in Eqn. 5.1, which means that both adiabatic efficiencies were treated as negative quantities. Both objectives were subject to a constraint that computed residual (RMS) of numerical simulation must be less than 10^-4 as described in Eqn. 5.2.

Objective

\[
\text{Minimize } -\eta_{\text{ad}} \text{ at } G/G^* = 0.479 \\
\text{Minimize } -\eta_{\text{ad}} \text{ at } G/G^* = 0.259
\]

Constraint

\[
\text{RMS Residual } < 10^{-4}
\]
5.5.2 Selection of Optimized Individuals

2-dimensional objective space after twelve iteration process of optimization is shown in Fig. 5.8. The adiabatic efficiency of each individual at off-design point ($\eta_{ad}$ at $G/G^* = 0.259$) is presented as a function of the adiabatic efficiency at design point ($\eta_{ad}$ at $G/G^* = 0.479$). All the adiabatic efficiency is nondimensionalized by the adiabatic efficiency of baseline configuration at design point ($\eta_{ad\_ref}$). The initial database from the DOE process prior to the optimization is represented by the black solid circles (DOE), while optimal individuals predicted by the meta-model assisted evolutionary algorithm...
during the optimization process by the red solid circle (IT) marks. Each circle mark represents each design which has been analyzed by the numerical simulation. Both objectives are improving towards the lower left hand corner of Fig. 5.8 (shaded region). Typical multi-objective optimization problems where both objectives are conflicting, a set of non-dominated optimal geometries was found. Two individuals Opt_1 (green) and Opt_2 (red), which lie on the Pareto front where one objective cannot be improved without worsening the other, were selected based on the maximum adiabatic efficiency at design \((G/G^* = 0.479)\) and off-design \((G/G^* = 0.259)\) flow rate conditions respectively as shown in Fig. 5.8.

Figure 5.9 shows the comparison of meridional shapes between baseline and selected individuals. Comparing the baseline shape, the upstream slot width of the individual with maximum efficiency at off-design point Opt_2 shows biggest width, while the individual with maximum efficiency at design point Opt_1 shows slightly bigger than that of baseline. Upstream slot position of all three candidates seems to possess the optimum location and the individual Opt_1 has slightly farther upstream position. Downstream slot width and downstream slot position play crucial roled to maximize the adiabatic efficiency. Downstream slot width of the individual Opt_1 is minimum which provides the maximum efficiency at design point, while the individual Opt_2 has slightly smaller downstream slot width compare to baseline. Both selected individuals share the same downstream slot position which moves farther downstream and possesses the same bypass width and same bypass height which is higher than that of the baseline (Table 5.2).

Figure 5.9 Comparison of meridional shapes between baseline and selected individuals

![Figure 5.9 Comparison of meridional shapes between baseline and selected individuals](image)

(a) Baseline
(b) Opt_1 \((\eta_{\text{max}} \text{ at design point})\)
(c) Opt_2 \((\eta_{\text{max}} \text{ at off-design point})\)

Table 5.2. Comparison of normalized design variables
5.5.3 Results and Discussion on Efficiency, Pressure and Work Input Characteristic of Compressor

The normalized total pressure rise and normalized adiabatic efficiency ($\eta_{ad}/\eta_{ad_{ref}}$) at the impeller exit under various discharge mass flow rate are numerically compared as shown in Fig. 5.10. On the left vertical axis, the static pressure rise ($P_{T4}$) between the inlet duct and the diffuser exit is normalized by the total pressure rise ($P_{T4_{ref}}$) at design point in the case of baseline configuration, as well as the adiabatic efficiency ($\eta_{ad}$) is normalized by the adiabatic efficiency ($\eta_{ad_{ref}}$) at design point in the case of baseline on the right. Compared to the baseline, the total pressure rise of both selected individuals has no significant different over the whole flow rate range except the improvement at the off-design point. The adiabatic efficiency of the individual Opt$_1$ shows good improvement over the whole flow rate range compared to the baseline, while the other shows no difference. The adiabatic efficiency ($\eta_{ad_{02}}/\eta_{ad_{02_{ref}}}$) of individual Opt$_2$ shows 102% improvement at design mass flow rate condition compared to the baseline. At the off-design point, however, the individual Opt$_2$ has upper hand while the adiabatic efficiency of individual Opt$_1$ falls close to that of the baseline. The individual Opt$_2$ shows 104% improvement at off-design point while its opponent shows drop in efficiency close to baseline.

Comparison of normalized static pressure rise ($P_{S4}/P_{T4_{ref}}$) between the inlet duct and the diffuser exit, work coefficient of the impeller ($\Delta h_{0}/U_{2}^{2}$) and normalized recirculation flow rate $m_{IR}$ as a function of normalized discharge mass flow rate ($G/G^{*}$) are numerically obtained as shown in Fig. 5.11. The pressure rise ($P_{S4}$) are normalized by total pressure rise ($P_{T4_{ref}}$) at the design flow rate condition. The recirculation flow rate $m_{IR}$ of the individual Opt$_2$ rises significantly compare to the baseline when the mass flow rate become smaller, while the individual Opt$_1$ shows more uniform distribution over the entire flow rate range. The work input requirement for the impeller of Opt$_2$ dropped when the mass flow rate became smaller while the Opt$_1$ was demanding more. The increase in adiabatic efficiency of the individual Opt$_2$ at off-design point is the result of smaller work input requirement and higher static pressure rise. It can be concluded that the rise in adiabatic efficiency ($\eta_{ad}/\eta_{ad_{ref}}$) of the individual Opt$_2$ at the off-design point is due to the
fact that enough low energy fluid is sucked into the casing owing to wider downstream slot width and farther downstream slot position.

Figure 5.10 Comparison of the total pressure rise and adiabatic efficiency between the selected individuals and baseline at $M_t = 0.774$ (CFD)

Figure 5.11 Comparison of the static pressure rise, work coefficient and recirculation flow rate between the selected individuals and baseline at $M_t = 0.774$ (CFD)
Figure 5.12 Comparison of the reverse flow zones between baseline and selected individuals at off-design point (G/G* = 0.259)

Figure 5.13 Comparison of the static entropy distributions between baseline and selected individuals at off-design point (G/G* = 0.259)

The relation between the efficiency improvement and increase in recirculation flow rate at off-design point in the case of individuals Opt2 can be concluded from the comparison of reversed circumferential average streamwise velocity distribution between the selected individuals and the baseline at off-design point (G/G* = 0.259) for impeller
inlet tip Mach number $M_t = 0.774$ as shown in Fig. 5.12. It can be seen that individual Opt$_2$ shows almost no reverse flow zone (green area) upstream of downstream slot in the impeller, while the others show significant reverse flow zones near the shroud wall side. It can be concluded that enough low momentum fluid is sucked into the casing owing to wider downstream slot width and farther downstream slot position. This is one of the reasons why the individual Opt$_2$ has the maximum adiabatic efficiency at off-design point. However, the reverse flow zones at design point are not presented since there were no significant reversed flow for all candidates at design point condition.

Another reason for the adiabatic efficiency improvement in Opt2 at off-design point was discussed with the normalized static entropy distribution as shown in Fig. 5.13. It can be seen that the static entropy rise at the compressor inlet was coming from the recirculated low energy fluid which merges with the incoming main stream flow. In the case of baseline the low energy fluid existing from the upstream slot was not merged well with the incoming flow, thus shows the circulation of low energy fluid between the upstream and downstream slots at the compressor inlet (Fig. 5.13-a). Thus, leaded to increase in static entropy rise, in other words decrease in adiabatic efficiency. In the cases of Opt$_1$ and Opt$_2$, the low energy fluid existing from the upstream slot was merged with the incoming fluid which leaded to decrease in static entropy rise (Fig. 5.13-b and 5.13-c). Opt$_2$ (Fig. 5.13-c) shows the better mixing of recirculated fluid and main stream fluid which provided adiabatic efficiency improvement at off-design point.

5.5.4 Sensitivity of Recirculation Flow Rate on Compressor Performance

In order to give the clear understanding of the effect of recirculation flow rate on the compressor performance at off-design point further investigation has been made. Figure 5.14 and 5.15 show the influences of recirculation flow rate at design point and at off-design point respectively for all the candidates which have been numerically investigated. Three individuals: Opt$_1$, Opt$_2$ and Opt$_3$ were selected from the database based on the recirculation flow rate at off-design point (Fig. 5.15) and their $m_{IR}$ were 15.33%, 36.29% and 52.62% respectively. Normalized adiabatic efficiency are presented over normalized recirculation mass flow rate by the main inlet flow. At design point (Fig. 5.14), adiabatic efficiency increases when recirculation flow rate approaches to a smaller amount and it is found that Opt$_1$ with maximum adiabatic efficiency possesses very small amount recirculation flow rate ($m_{IR} = 4.14\%$). At off-design point (Fig. 5.15), the optimizer attempted to find the individual (Opt$_2$) with a certain amount of recirculation flow rate ($m_{IR} = 36.29\%$). Minimum recirculation flow rate is required to maximize the adiabatic efficiency at design point to reduce the mixing and friction losses. It can be concluded that optimum recirculation flow rate is required to suck the required amount the reverse flow at the shroud wall side at off-design point.
Figure 5.14 Influence of the recirculation flow rate on the adiabatic efficiency at design point

Figure 5.15 Influence of the recirculation flow rate on the adiabatic efficiency at off-design point

5.5.5 Flow Characteristics of Selected Individuals

Figure 5.16 shows the distribution of relative velocity normalized by the impeller inlet tip speed over the streamwise location in the compressor at off-design point. It can be seen that Opt₂ shows better deceleration in the impeller than Opt₁ with under \( m_{IR} \) and Opt₃ with over \( m_{IR} \).

Another detailed discussion of the inducer flow field using the numerical analysis is also presented since the Opt₂ shows improvement due to the influence of mixing behavior of recirculation flow rate with main inlet flow. This analysis includes the variation of meridional and circumferential velocities that results from the annular bypass recirculation (Fig. 5.17) and its effect on incidence angle. The flow incidence was computed using the mass flow average meridional and circumferential velocities.
components just upstream of the main blade leading edge from the CFD analysis (Fig. 5.18).

It can be seen that there is large circumferential flow near the shroud wall at the impeller inlet when the flow rate approaches to the off-design point. Both Opt_1 (under \( m_{IR} \)) and Opt_3 (over \( m_{IR} \)) show larger circumferential components than Opt_2 (optimized \( m_{IR} \)) at the shroud wall side. As discussed in Fig. 5.13, low momentum fluid with large circumferential components reaches to the upstream of impeller in the inlet duct due to not enough recirculation flow rate. The meridional and circumferential velocity distributions of the individuals Opt_2 show more uniform distribution than other recirculation flow type casing treatments.

Figure 5.16 Relative velocity distribution over streamwise location

Figure 5.17 Velocity distributions at the impeller inlet at off-design point
As was found with the baseline casing treatments, no mixing of the recirculation flow occurs at the core area of the inducer with any casing treatments. The recirculating flow mixes with only 40% of the span at shroud side in all the selected casing treatments (Fig. 5.17). Removing some of the inlet swirl should give an increase in pressure ratio. However, in both selected individuals, numerical results did not show a significant increase in pressure over the baseline.

It should be noted that flow incidence of Opt\textsubscript{2} shows uniform distribution from hub to shroud at the impeller inlet while Opt\textsubscript{1} shows larger positive flow incidence and Opt\textsubscript{2} shows negative ones near the shroud wall side as can be seen in Fig. 5.18. Flow incidence tends to negative moving from hub to shroud in all the candidates at off-design point. The negative incidence near shroud wall side was suppressed by the reverse flow from surge with large negative axial velocity components which led to large positive incidence in the case of Opt\textsubscript{1} (Fig. 5.17 and 5.18). However, in the case of Opt\textsubscript{3}, the negative incidence increased due to a large component of circumferential velocity and a small component of axial velocity coming from an exceeded amount of recirculating flow. In the case of Opt\textsubscript{2}, it is the indication of a proper recirculation flow rate which provides the uniform flow incidence and better mixing of recirculating flow, and hence leading edge flow separation could be suppressed.

5.5.6 Sensitivity of Optimization Parameters on the Objective Functions

Final discussion is the sensitivity of input (optimization) parameters on the objective functions (adiabatic efficiencies at design and off-design conditions). Sensitivity of design variables on the adiabatic efficiency at the design condition was illustrated in Fig. 5.19 and sensitivity of design variables on the adiabatic efficiency at
the off-design condition in Fig. 5.20. The black circle (DOE) marks are the initial database from the design of experiment process prior to optimization, while the red (IT) circle marks were generated during the optimization process.

Figure 5.19 Sensitivity of design variables on the adiabatic efficiency at design point
As can be seen in Fig. 5.20, downstream slot width ($p_3$), downstream slot position ($p_4$), bypass height ($p_5$) and bypass position ($p_6$) seem to have higher influence on the...
adiabatic efficiency at design condition. The adiabatic efficiency was increasing when
downstream slot width ($p_3$) becomes smaller and the maximum value can be seen when
downstream slot width ($p_3$) was minimum (Fig. 5.19-c). It was clear that minimum or no
recirculation flow rate was preferred at the design mass flow rate which leaded to
minimum downstream slot width. The increase in adiabatic efficiency could be obtained
when downstream slot position ($p_4$) moved to the farther downstream positon (Fig. 5.19-
d). In spite of small influences, larger bypass height ($p_5$) and higher bypass position ($p_6$)
were preferred to obtain the higher efficiency at design condition (Fig. 5.19-e and 5.19-
f).

The adiabatic efficiency at the off-design condition was seem to be influenced by
all design variables. Figure 5.20-a shows wider upstream slot width ($p_1$) could provide
better adiabatic efficiency. This could be concluded from static entropy analysis (Fig.
5.13) that wider upstream slot was able to provide better mixing of recirculated low
energy fluid with main stream fluid and hence reduce the static entropy rise. Figure 5.20-
b and 5.20-c show that optimizer found the optimum position for upstream slot ($p_2$) and
optimum width for downstream slot ($p_3$). Other three parameters, downstream slot
position ($p_4$), bypass height ($p_5$) and bypass position ($p_6$), show same influences on the
adiabatic efficiency at design point and off-design point. Farther downstream slot position,
larger bypass height and higher bypass positon could provide better adiabatic efficiency
at both design and off-design conditions (Fig. 5.20-d, 5.20-e and 5.20-f). It can be
concluded that further investigations are required in order to get the optimum value of
these three parameters.

5.5.7 Conclusions

The performance of a turbocharger compressor with baseline recirculation flow
type casing treatment was investigated using both CFD modeling and experimental
measurements. Based on the baseline CFD model, an optimization of the casing treatment
has been performed. In search of optimized casing treatment, model assisted evolutionary
algorithm was introduced for global optimum exploration and for reduction of expense
of computational tools and computation time. The optimized result successfully shows
“Pareto front” on the two objective design spaces. Two optimized shapes on a “Pareto
front” are selected in order to discuss the effect of casing treatment at the point of view
of suppression of flow separation and a reduction of leading edge flow separation. The
prevalent of inducer reversed flow toward small flow rate region could be suppressed by
increasing recirculating flow into the casing. However, the exceed amount of the
recirculating flow also could lead to decrease in the compressor performance too. The
proper amount of recirculation flow rate could provide the uniform flow incidence and
better mixing of recirculating flow and hence improve the compressor performance at the
off-design point. It is noticed that despite being preferred small recirculation flow rate at
design point, optimized recirculation flow rate is essential at the off-design point to
maximize the adiabatic efficiency. Another criterion was mixing of recirculated low energy fluid and main stream fluid. Low energy fluid tends to circulated between the upstream slot and downstream slot and hence increases the static entropy rise in the compressor inlet. Thus, phenomena could be suppressed by enlarging the upstream slot width. From sensitivity of design parameters influence on the performance, larger upstream slot width (p1) and larger bypass height (p5) were preferred for adiabatic efficiency improvement at off-design point condition. Farther downstream slot position (p4) and higher bypass position (p6) were likely to provide better performance at off-design point. Optimizer found optimum upstream slot position (p2) and downstream slot width (p3) in the range of optimization parameters at off-design point.

5.6 Off-design Performance Improvement in Centrifugal Compressors with Recirculation Flow Type Casing Treatment by Optimized Guide Vane

![3D Model Compressor Model with guide vanes inside casing treatment](image1) ![Redirection of flow inside the casing treatment using guide vane](image2)

Figure 5.21 Application of guide vanes inside the annular bypass

A turbocharger compressor requires a wider operating range as well as higher performance in order to meet demands of modern engines. Recirculation flow type casing treatment is one of the most popular methods 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 merits of recirculation flow type casing treatment are 1) reduction of pre-whirl at impeller inlet, 2) improvement of flow incidence 3) higher blade loading and 4) proper deceleration of velocity. However, increase in friction loss and mixing loss can be considered as the demerit of the casing treatment. In the present study, an application of three-dimensional guide vane inside the casing treatment (Fig. 5.21-a) is presented to overcome the demerits of the casing treatment. The shape of a guide vane was optimized
using a surrogate model assisted optimization code. Numerical results showed that the optimized guide vane hold a possibility to improve off-design performance by reducing the excessive pre-whirl at the compressor inlet.

5.6.1 Objective Functions and Design Variables

The objective functions in this research work aim to improve adiabatic efficiency at two operating points at design mass flow rate \((G/G^* = 0.479)\) and at off-design mass flow rate \((G/G^* = 0.330)\) under same impeller tip speed \((M_t = 0.774)\) as shown in Fig. 5.22. 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, six design variables are defined as shown in Fig. 5.23 and their minimum and maximum limits are expressed in Table 5.3. Six design variables are angular (bow intensity) \((X)\) and spanwise \((Y)\) positions for mid-section stacking point, chord length \((C)\) and three blade angles \((\beta_1, \beta_2, \text{and } \beta_3)\) of guide vane.

Objective

\[
\begin{align*}
\text{Minimize } -\eta_{\text{ad}} \text{ at } G/G^* &= 0.479 \\
\text{Minimize } -\eta_{\text{ad}} \text{ at } G/G^* &= 0.330
\end{align*}
\]

Constraint

\[
\text{RMS Residual } < 10^{-4}
\]

Figure. 5.22 Objective Functions
![Figure 5.23 Definition of design variables](image)

**Table 5.3 Design Variables for Guide Vanes**

<table>
<thead>
<tr>
<th>Description</th>
<th>Max</th>
<th>Min</th>
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<td>X position X/R₁₀ [-]</td>
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<td>0</td>
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<tr>
<td>Y position Y/R₁₀ [-]</td>
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<td>0.156 (Span80%)</td>
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<tr>
<td>Chord C/R₁₀ [-]</td>
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<td>β₁ [deg]</td>
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<td>0°</td>
</tr>
<tr>
<td>β₂ [deg]</td>
<td>30°</td>
<td>-30°</td>
</tr>
<tr>
<td>β₃ [deg]</td>
<td>0°</td>
<td>-30°</td>
</tr>
</tbody>
</table>

5.6.2 Optimization Results and Three-dimensional Guide Vanes

![Figure 5.24 Two-dimensional objective space for global optimization](image)
The result of the optimization is presented as the two-dimensional objective space in Fig. 5.24. The two objective functions, adiabatic efficiency at design flow rate \( (G/G^* = 0.479) \) and at off-design flow rate \( (G/G^* = 0.334) \), are presented as a minimizing problem which means that both parameters are improving approaching to the lower left corner. The efficiencies are normalized by adiabatic efficiency of baseline compressor with casing treatment at design flow rate \( (G/G^* = 0.479) \). Each circle mark indicates one individual design which has been analyzed by numerical simulation and satisfies all constraints. The grey 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). The selected optimized guide vane design, as indicated in Fig. 5.24, are illustrated in Fig. 5.25 and design variables are expressed in Table 5.4.

![Single guide vane viewed from axial direction](image1)  ![3D Model](image2)

(a) Single guide vane viewed from axial direction  (b) 3D Model

Figure 5.25 Optimized guide vanes

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
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<tbody>
<tr>
<td>X position X/R₁₃ [-]</td>
<td>0.156</td>
</tr>
<tr>
<td>Y position Y/R₁₃ [-]</td>
<td>0.436 (Span20%)</td>
</tr>
<tr>
<td>Chord C/R₁₃ [-]</td>
<td>1.246</td>
</tr>
<tr>
<td>( \beta_1 ) [deg]</td>
<td>30°</td>
</tr>
<tr>
<td>( \beta_2 ) [deg]</td>
<td>30°</td>
</tr>
<tr>
<td>( \beta_3 ) [deg]</td>
<td>0°</td>
</tr>
</tbody>
</table>

Table 5.4 Design Variables for Optimized Guide Vanes
5.6.3 Performance of Optimized Guide Vanes

The comparison of characteristics of compressor between the cases of with straight guide vanes, optimized guide vanes and without guide vane has been performed based on the numerical simulation. The adiabatic efficiency and static pressure rise are compared between the cases with optimized guide vanes and without guide vanes as shown in Fig. 5.26, whereas work input coefficient and recirculated mass flow rate in Fig. 5.27. At off-design flow rate condition, both adiabatic efficiency and static pressure rise are successfully improved in the cases of straight and optimized guide vanes (Fig. 5.26) while recirculated mass flow rate ($m_{IR}$) is same for the cases of straight and optimized guide vanes except at off-design point (Fig. 5.27). Work input coefficient also shows significant increase in the cases with straight and optimized guide vanes at off-design point which means that the impeller needs more energy when the guide vane is applied (Fig. 5.27). When the mass flow rate become larger than the design flow rate, it can be seen as the drop in adiabatic efficiency in the cases with guide vanes. This can be discussed from pressure rise from Fig. 5.26 and work input coefficient from Fig. 5.27. All the cases show same work input coefficient and static pressure rise drops in the cases with guide vanes. The negative value of recirculated flow rate means that some of the mainstream flow goes bypassing the casing treatment when the mass flow rate become larger than design flow rate. The rate of bypass flow is higher and hence causes more friction losses inside the casing in the cases with guide vanes. Comparison of reverse flow zone on the meridional plane at off-design condition ($G/G^*=0.334$) between the cases with optimized guide vane and without guide vane shows that the casing with optimized guide vane suck more low momentum fluid on the shroud wall side of the impeller (Fig. 5.28).
5.6.4 Flow characteristics of Optimized guide vane

Figure 5.29 shows the distribution of relative Mach number over the streamwise location from the inlet duct to the impeller outlet averaged at mid span (span 50%) at off-design point. The sudden raise of the relative Mach number after the upstream slot in the cases with guide vanes means the merging of recirculated flow with lesser pre-whirl to the main flow stream. The cases with the guide vanes shows better deceleration and smaller relative Mach number in the impeller despite higher relative velocity at the main blade leading edge. Thus, the application of guide vane could provide better pressure rise.
Figure 5.29 Relative Mach number distribution from inlet to outlet \((G/G^*=0.331)\) 

(CFD)

The comparison of entropy distribution between the cases with guide vane and without is shown in Fig. 5.30. In the upstream of impeller after the upstream slot, static entropy has dramatically raised in all cases due to merging of recirculated flow. The cases with guide vane shows higher entropy rise compare to the case without guide vane. This means the cases with guide vane provides higher recirculated flow rate. The highest entropy rise in the case with straight guide is due to high pre-whirl motion carried by the recirculated flow. However static entropy in the case without guide is highest in the downstream of downstream slot. This could be explained that not enough low momentum fluid is sucked into the casing treatment in the case without guide vane. It can be concluded that the cases with optimized guide vane provided smallest entropy rise at the impeller exist because of more correct amount of recirculated flow rate, better redirection of recirculated flow inside the casing and better merging with the main inlet flow.

The flow characteristics upstream of the compressor inlet has been numerically investigated at off-design condition \((G/G^*=0.334)\). Meridional \((V_m)\) and Circumferential \((V_u)\) velocity components measured 5mm before the impeller leading are compared in Fig. 5.31 between the cases with guide vanes and without guide vane. In the cases with straight and optimized guide vanes, circumferential velocity \((V_u)\) shows better distribution and the pre-whirl at the shroud wall side has successfully been suppressed. Despite larger flow incidence, the cases with guide vane provide more smooth incidence at the impeller inlet (Fig. 5.32) and all casing treatments with and without guide vane show good surge characteristics. This could be assumed that flow separation due to large flow incidence
has been suppressed simultaneously owing to balancing of pressure between the inlet duct and downstream of the impeller inlet.

Figure 5.30 Normalized static entropy distribution from inlet to outlet (G/G*=0.331) (CFD)

Figure 5.31 Span-wise meridional and velocity at off-design condition (G/G*=0.334) (CFD)
The detailed numerical analysis of pressure ratio characteristic under four impeller speed has been carried out as shown in Fig. 5.33. For all impeller speed, the case with optimized guide vane shows improvement in surge margin despite small deterioration of static pressure ratio for larger flow rate. It can be concluded that an application of guide vane has possibility to improve surge margin and the performance in small flow rate conditions.
5.6.4 Conclusions

The characteristics of the centrifugal compressor for turbocharger between the cases with and without casing treatment has experimentally been analyzed and compared. Three-dimensional parameterization of guide vanes inside the recirculation flow type casing has been introduced. Surrogate model assisted optimization has been performed and optimized shape of the guide vane was selected. Application of the optimized guide vane shows improvement in adiabatic efficiency and static pressure rise by suppressing the excessive pre-whirl at impeller inlet at off-design condition. It can be concluded from an application of guide vane inside a recirculation flow type casing treatment as following:

1. A casing treatment with guide vane holds strong possibility to improve pressure rise and adiabatic efficiency at off-design point, however, it can deteriorate the performance when the mass flow rate becomes larger than design condition.
2. A casing treatment with guide vane demands more the work input when the flow rate become smaller.
3. An application of guide vane can reduce the circumferential velocity component from the recirculated flow at impeller inlet and properly designed guide vane can suppress all the circumferential velocity component.
4. Despite large flow incidence to the impeller inlet, both casing treatments with and without guide vane shows good surge characteristics.
5. An application of guide vane also has possibility to improve surge margin.
6.1 Conclusions

This thesis presents various optimizations with respect to performance improvement and flow rate range enhancement in radial compressor. These include optimizations of two components of radial compressors with three different test compressors.

6.1.1 Low Solidity Circular Cascade Diffuser

Secondary flow is the cross-flow which is deviated from the primary flow in a turbomachinery. It should be suppressed to improve an efficiency of turbomachinery in design procedure. In small flow rate region, however, it should be noticed that secondary flow suppresses flow separation. Despite a low solidity circular cascade diffuser (LSD), which has been proposed by Senoo, has an advantage to overcome the trade-off between vane and vaneless diffuser, it is difficult to design a LSD blade which can activate the secondary flow effect to suppress the flow separation.

In these research works, a two-dimensional LSD blade has been optimized, manufactured and the characteristic of optimized LSD is confirmed by the experimental test rig. Optimized LSD has an extended operating range of 114 % towards smaller flow rate as compared to the baseline design without deteriorating the diffuser pressure recovery at design point. Detailed discussion about the effect of tip clearance in performing the optimization of a LSD blade is also presented. The detailed flow in the diffuser is also confirmed by means of a Particle Image Velocimetry (PIV). The flow structures in a LSD captured by PIV have been reported under several small flow rate conditions for both cases with and without tip clearance. The flow separation and
secondary flow in a LSD diffuser are clearly captured by PIV. The stability of secondary flow effect of optimized LSD under different interaction between impeller and diffuser has been reported by PIV using phase lock.

In the research works, the optimized LSD shows good anti-stall characteristics at small flow rate conditions being the high camber angle. The secondary flow effect is the main reason which suppresses the flow separation on the suction side surface of the LSD blade. It is difficult to optimize the LSD with a wide operating flow rate range without applying the tip clearance. The tip clearance is one of the simple ways to activate the secondary flow at small flow rate conditions. An excessive secondary flow only marginally decreases the impeller performance. Deterioration of the static pressure coefficient at impeller exit $\psi_{c2}$ at small flow rate condition is related to the reverse flow reaching the impeller exit, which reduces the deceleration of the main flow at impeller exit. The reverse flow rate should be minimized to maximize the deceleration of main flow in the vaneless space as possible.

6.1.2 Recirculation Flow Type Casing Treatment

It has been a long time since the first attempt of the casing treatment to turbocharger compressor in a marine diesel engine by Fisher, however, casing treatment is less interested by researchers and manufacturers in earlier years due to difficulties in component matching and lack of aerodynamics knowledge interacting between the casing treatment and impeller. The research works in this thesis intended to fill the gap in between. The optimizations of the shape of the recirculation flow type casing treatment has been conducted to centrifugal compressors for small turbocharger and commercial turbocharger. At first, an application of casing treatment was intended to enhance the compressor map width, the results show that a well-designed casing treatment can improve pressure rise and adiabatic efficiency in the whole flow rate range.

In the research of study on performance improvement in centrifugal compressor for small turbocharger (academic research), adiabatic efficiency is improved in wide operating condition, and the recirculation flow rate and bypass flow becomes smaller in the case of optimized casing than in the case of baseline. It has been confirmed that the optimized casing treatment possesses optimized recirculation flow rate ($m_{IR} = 0.09\%$) at off-design and least entropy rise at the impeller exit.

In the research of study on flow range enhancement in centrifugal compressor for small turbocharger (academic research), the operating flow range has been enhanced to 107\% (EFF) and 110\% (OFR) for two different casing treatments respectively. The EFF casing treatment shows good improvement of adiabatic efficiency in wide flow rate range.

In the research of study on global optimization of guide vane in recirculation flow type casing treatment for small turbocharger (academic research), it is clearly indicated
that there is a possibility to improve the adiabatic efficiency at small flow rate conditions. Excessive pre-whirl based on recirculation flow is suppressed by the optimized guide vane, and velocity distortion at impeller inlet has improved. It is found that improvement of the efficiency in the case of the optimized guide vane is supported by proper distribution of flow at impeller inlet.

In the research of study on performance improvement in centrifugal compressor for commercial turbocharger (industrial collaboration research), two casing treatments has been selected and compared based on the maximum efficiency at design and off-design conditions. The selected individual based on the maximum efficiency at off-design conditions shows significant adiabatic efficiency improvement at off-design condition. The prevalent of inducer recirculation toward small flow rate region is suppressed and velocity distortion at the impeller inlet has improved.

In the research of study on off-design performance improvement by optimized guide vane in centrifugal compressor with casing treatment for commercial turbocharger (industrial collaboration research), optimization of three-dimensional guide vane has been introduced. Application of optimized three-dimensional guide vane shows improvement in adiabatic efficiency and static pressure rise by suppressing the excessive pre-whirl at impeller inlet at off-design condition.

In the studies, which focus on the performance improvement, it has been confirmed that recirculation flow rate is one of the dominant parameters for improving the adiabatic efficiency of the compressor. It is found that the optimized shape of recirculation flow type casing has optimized recirculation flow rate to maximize the adiabatic efficiency. From the sensitivity of the recirculation flow type casing design parameters, downstream slot (bleed slot) width, downstream slot positon has large impact on the adiabatic efficiency at both design and small flow rate. Downstream slot width should be minimum to improve adiabatic efficiency at design flow rate condition, however, it should be optimized to maximize adiabatic efficiency at off-design (near surge) condition. Although it is difficult to confirm downstream slot position, downstream slot should locate between middle or downstream of inducer of the impeller for both design and off-design conditions. A phenomenon observed during these studies is a circulation of low momentum fluid (recirculated flow) between the casing treatment and the impeller inducer. The recirculated flow injected from the upstream slot should merge well with the incoming main stream flow, otherwise, the low energy fluid locating on the shroud wall side of the inlet duct, will reach to the impeller and this will cause smaller effective area at the impeller inlet. The area effected by the low momentum fluid (recirculated flow) should be as small as possible to maximize the efficiency. This phenomenon can be observed by studying static entropy distribution from the end wall view.

The study on flow range enhancement by optimizing the casing treatment in turbocharger compressor is one of the challenging research works which has been studied
only once in this thesis period. This study demands higher computational power, which requires at least three numerical simulations or more for each individual design, compare to the study on performance improvement, which requires only two numerical simulations for one. Moreover, optimized design obtained from study on performance improvement shows improvement in flow rate range too. From the sensitivity analysis of the recirculating flow type casing design parameters, upstream slot width, upstream slot position and downstream slot width position has large influence on both the operating flow range and the adiabatic efficiency at design flow rate. Minimum downstream slot width and farther upstream slot position provide both widest operating flow range and highest adiabatic efficiency at design flow rate. However, the downstream slot position has opposite influence on the operating flow range and the adiabatic efficiency.

In the studies concerned with optimization of guide vane, a casing treatment with guide vane holds strong possibility to improve pressure rise and adiabatic efficiency at off-design point, however, it can deteriorate the performance when the mass flow rate becomes larger than design condition. A casing treatment with guide vane demands more work input when the flow rate become smaller. An application of guide vane can reduce circumferential velocity component from the recirculated flow at impeller inlet and properly designed guide vane can suppress all the circumferential velocity component. An application of guide vane also has possibility to improve surge margin. In optimization of two-dimensional guide vane, the optimized guide vane has the blade angle of a counter direction of the rotating impeller.

6.2 Perspectives

The research conducted in this thesis brings the better understanding of flow phenomenon in LSD diffuser and recirculation flow type casing treatment. Following presents some suggestions for further improvement.

The optimization of low solidity cascade diffuser can be performed with more higher freedom of design variables such as:

1) Considering the effect of lean and rake in the design variables in a LSD diffuser
2) Considering the effect of stacking line and adding control parameters to it
3) Considering the effect of relative thickness distribution and
4) Considering the other type of diffuser such as wedge and tandem type.

The optimized recirculation flow type casing treatments have not been experimentally confirmed yet. The full pitch unsteady numerical investigation of recirculation flow type casing treatment with scroll casing will bring better understanding of flow phenomena and more realistic interaction between the impeller and casing treatment. Optimization on unsymmetrical distribution of guide vane inside the recirculation flow type casing treatment should be studied. Optimization of design
variables of guide vane and casing treatment simultaneously should also be considered. Study on the optimization of impeller design parameters become popular among the researchers.

The optimizations in this thesis mainly focuses on the aerodynamics approach. Multidisciplinary optimization of impeller considering aerodynamics and stress should also be considered for further improvement of optimization algorithm. Extending the application of optimization idea in other research area such as vibration analysis or noise reducing should be conducted.
REFERENCES


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