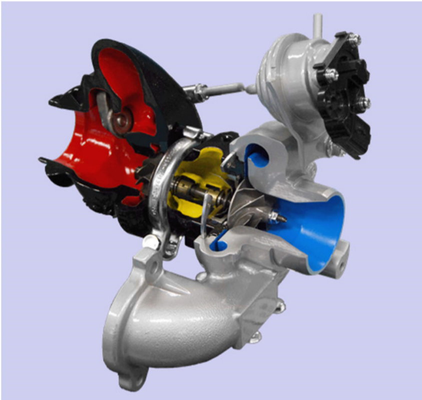
Aerodynamic Design Optimization of Centrifugal Compressor Impeller Based on Genetic Algorithm and Artificial Neural Network



SEIICHI IBARAKI[*1] ISAO TOMITA[*2]

KOICHI SUGIMOTO[*2]

The centrifugal compressor in an automotive turbocharger is required to have a high pressure ratio, high efficiency, and in particular, wide operating range. Performance improvement has been carried out utilizing computational fluid dynamics and experiments, and further performance improvement through conventional design methods has become increasingly difficult, requiring a significant amount of time for aerodynamic design. Therefore, we developed an optimized design method with computers using a genetic algorithm and artificial neural network in place of the conventional design method. This method was applied to the design of the centrifugal compressor impeller. The performance results of the two designed impellers attained higher efficiency and a significant extension of operating range, respectively, compared with the baseline impeller.

1. Introduction

The downsizing of automotive engines through the installation of turbochargers is acknowledged as an assured method for fuel efficiency enhancement, and is expanding its application. The centrifugal compressor in an automotive turbocharger is required to have wide operating range to cover automotive driving, a high pressure ratio, and high efficiency. Performance improvement has been carried out utilizing computational fluid dynamics and experiments to measure the performance and internal flow.[(1)] As the impeller in the centrifugal compressor has a complicated three-dimensional geometry, the internal flow forms a very complicated three-dimensional vortex flow field due to the geometry, and the loss generation has diverse mechanisms. There are many design parameters and the improvement of efficiency has become difficult with the conventional manual design method having the problem of requiring a significant amount of time. Therefore, we have developed an optimized design method with computers using a genetic algorithm and artificial neural network as an advanced design method. This method was applied to the design of the centrifugal compressor impeller in a turbocharger, and two types of impellers were created. As a result of a performance test, the efficacy of this optimized design method was confirmed to have attained higher efficiency of 1% or more, and an extended operating range at least twice as large in each impeller compared with the baseline impeller.

This report describes the optimized design method, the resulting design of the centrifugal compressor impeller and its performance test results. In addition, an understanding of the internal flow phenomena and the mechanism for better performance using computational fluid dynamics are briefly explained.

*1 Chief Staff Manager, Nagasaki Research & Development Center, Technology & Innovation Headquarters
*2 Nagasaki Research & Development Center, Technology & Innovation Headquarters

2. Optimized Design Method

Figure 1 shows the optimized design process consisting of a genetic algorithm (GA) and artificial neural network (ANN). GA is an optimizing algorithm simulating biological evolution, and ANN simulates the function of the human brain. In GA, design parameters such as the blade geometry coordinates shown in **Figure 2** are regarded as genes, and the new blade geometry is generated by crossing blade geometries in each generation. In ANN, the performance is estimated based on a database of blade geometry and performance without conducting detailed fluid dynamics. The high-performance blade geometry estimated in ANN is evaluated in the detailed three-dimensional viscous fluid dynamics (3D NS Analysis) shown on the left in Figure 1, and the results with the geometrical data are stored in the database. The accuracy of ANN performance prediction is improved in accordance with the expansion of the database with the learning function of ANN.

The optimized design of the centrifugal compressor impeller using this method generated 200 generations and 10 individuals (blade geometries) in one generation with GA, and the process was repeated approximately 70 times. There were 27 design parameters to define the blade geometry including the number of blades, meridional plane coordinates and blade angle. Nine penalty functions, such as efficiency, flow rate, blade load, maximum Mach number and acceleration/deceleration rate to Mach number were defined to evaluate the performance in ANN(2).

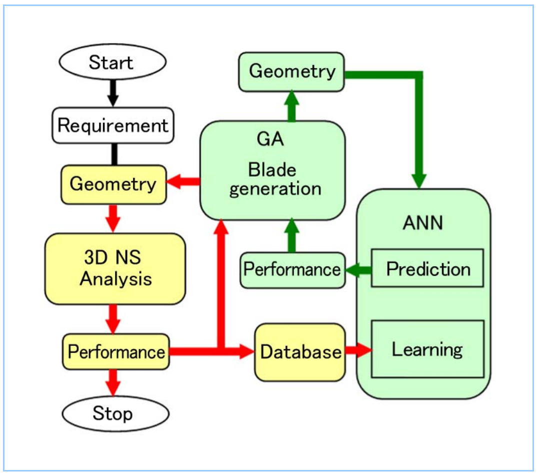


Figure 1 Process of design optimization

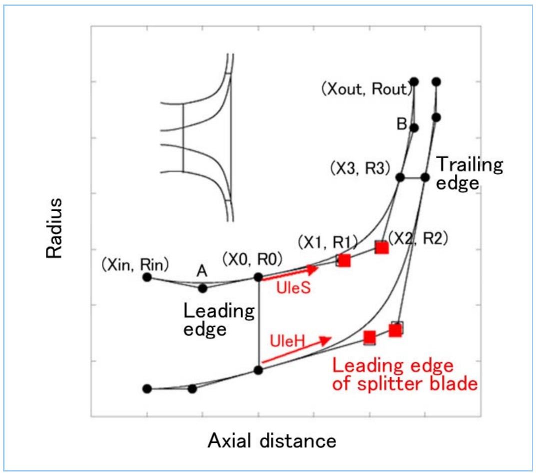


Figure 2 Difinition of blade geometry

3. Optimized Design Results of Centrifugal Compressor Impeller

The centrifugal compressor impeller is a backswept open impeller type with a splitter blade, and the design pressure ratio is 1.53. **Table 1** and **Figure 3** show the main specifications and forms of the two optimized design impellers (OPT1 and OPT2), and a baseeline impeller, respectively. **Figure 4** shows a comparison of meridional geometries. In the two optimized designs, the minimum value of the blade number, which is one of the design parameters, is the only difference. In the first case, the number of full blades and splitter blades is five each and in the second case, the number is four. OPT1 has higher efficiency by 0.5 to 1.0% compared with that of the baseline impeller, and OPT2 has an extended operating range of twice or more, although the maximum efficiency was lower by 1% compared with that of the baseline impeller, as described below.

Table 1 Specifications of baseline and optimized design impellers

	OPT1	OPT2	Baseline
Impeller diameter (mm)	50.0	49.6	49.0
Outlet width (mm)	4.14	3.75	4.20
Inlet tip diameter (mm)	39.5	37.1	37.8
Inlet hub diameter (mm)	13.9	12.6	12.6
Blade number	5+5	4+4	6+6

Figure 3 Comparison of baseline impeller and optimized impeller

The history of penalty functions of the OPT1 optimized design is shown in **Figure 5**. OPT1 is an impeller attained at the 17th iteration, and its efficiency falls within the top 10 of all results. Optimized design, however, needs to consider not only efficiency, but also other performance penalty indexes in a comprehensive manner for the selection of the impeller. **Figure 6** shows the shroud Mach number distribution from blade inlet to outlet of OPT1, which was attained by ANN and 3D NS analysis. This shows good correlation with ANN prediction and the results of 3D NS analysis. The distribution of the Mach number of OPT1 is very smooth with small local fluctuations, and the load at full blade and splitter blade are almost even with approximately the same Mach number. The final selection was OPT1 considering these results. OPT1 was thought to be advantageous in strength and productivity with less forward inclination of the splitter blade leading edge.

OPT2 selected in the second optimized design was also chosen using similar criteria as OPT1. OPT2 has the third highest efficiency overall, a smooth Mach number distribution at the blade tip similar to OPT1 as shown in **Figure 7.**Blade loads between the full blade and splitter blade are also even. OPT2 has only four full blades and four splitter blades as shown in Table 1 and Figure 3, and as a result, the inlet diameter of the impeller is small as shown in Figure 4. It also shows characteristics of the short axial length and forward slanted leading edge of the splitter blade.

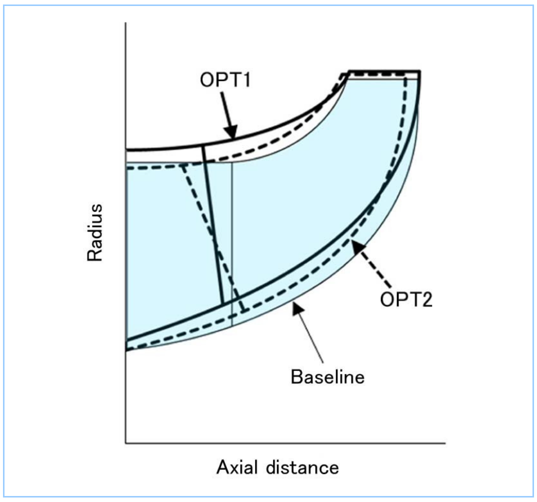


Figure 4 Comparison of meridional geometries

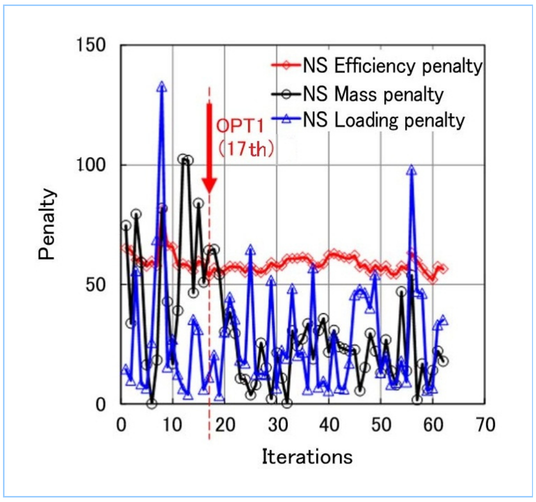


Figure 5 History of performance evaluation index in optimized design

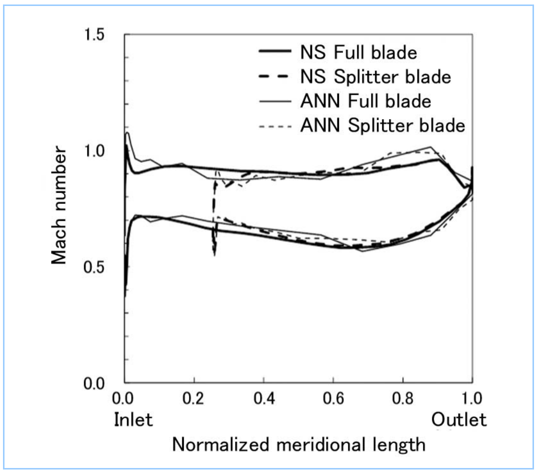


Figure 6 Shroud Mach number distribution of OPT1

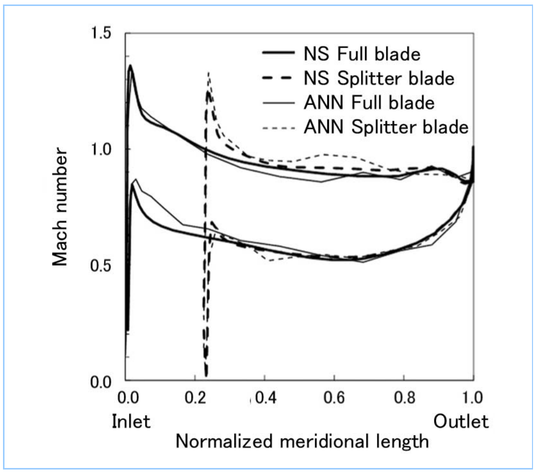


Figure 7 Shroud Mach number distribution at blade tip of OPT2

4. Performance Test Results

The performance test results of two impellers, OPT1 and OPT2, attained by the optimized design are compared with the baseline impeller. **Figure 8** shows the compressor characteristics, **Figure 9** shows the compressor efficiency, and **Figure 10** is a comparison of operating range.

Figure 8 shows the higher efficiency of OPT1, and OPT2 has a lower maximum efficiency by 1% but an extended operating range compared with the baseline impeller. In Figure 9, OPT1 shows a higher maximum efficiency by 0.5% in the proximity of pressure ratio 1.8, and higher efficiency by 1% or more at the pressure ratio 2.2 or greater. In contrast, OPT2 shows a lower efficiency by approximately 1% at all pressure ratio ranges. Figure 10 shows the normalized operating range value ((maximum flow rate – surging flow rate)/surging flow rate), and OPT1 shows a moderate reduction of operating range compared with the significant operating range reduction at a pressure ratio of 1.9 or higher in the baseline impeller. OPT2 shows a further extended operating range that is twice or more of the baseline impeller at a pressure ratio of 2.2 or higher. The compressor characteristics in Figure 8 show the stable and extended operating range of OPT2 without showing the unstable right-up characteristics of the compressor. These characteristics are ideal for the centrifugal compressor in a turbocharger that is required to have an extended operating range.

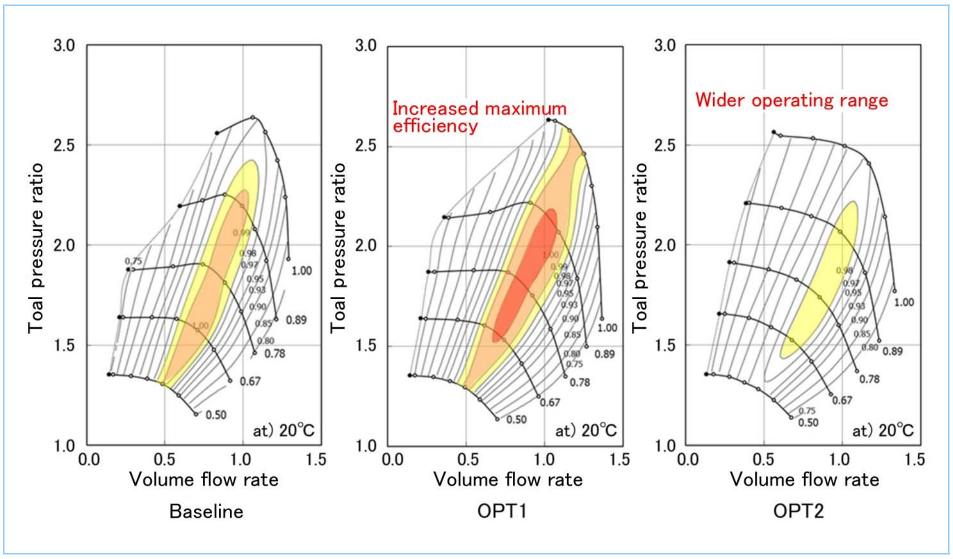


Figure 8 Comparison of compressor characteristics

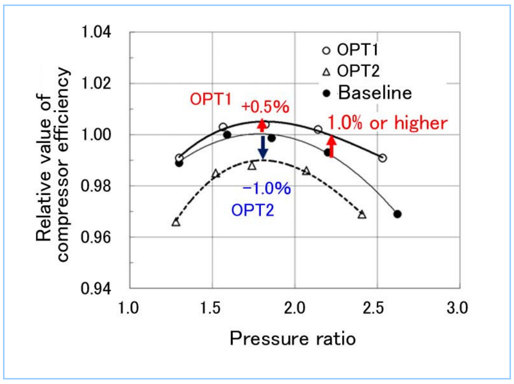


Figure 9 Comparison of compressor efficiencies

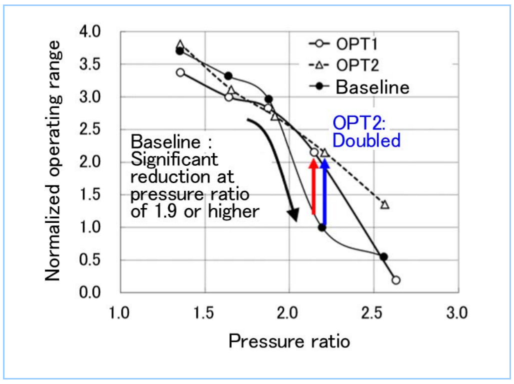


Figure 10 Comparison of operating range

5. Consideration of Internal Flow

The clarification of the mechanism of performance improvement of OPT1 and OPT2 was conducted with three-dimensional fluid dynamics. ANSYS CFX ver. 12 was used as the analysis code, and the k-ε model was used for the turbulent model for the steady analysis of the impeller, diffuser and scroll. The number of computational grids is 2,470,000 in the impeller, 410,000 in the diffuser and 250,000 in the scroll. Computations were made at the rotational speed of 0.89 (maximum speed: 1.0) and the peak efficiency point of the flow rate as shown in Figure 8.

The results of fluid dynamics of OPT1, OPT2 and the baseline impeller are shown in **Figure 11** (a) to (c) respectively. These are visualizations of the vortical structure of the impeller, and the combination of the blade surface limiting stream line, stream line and loss distribution are shown. Figure 11 (a) shows the baseline impeller, and it shows the strong secondary flow rising up from the hub to blade tip that is generated at the blade inlet in the vicinity of the leading edge. With this flow, low energy fluids such as blade surface boundary layer accumulate on the blade tip, move down the stream while sucking-in the tip leakage vortex, and create a large loss area at the right side passage of the splitter blade near the outlet. The tip leakage vortex is generated at the immediate downstream of the full blade leading edge, creating a large loss area by moving into the left side passage of the splitter blade. In contrast, OPT1 in Figure 11 (b) is free from the secondary flow at the blade inlet, and the accumulation of low energy fluid at the tip end is controlled. In addition, the tip leakage vortex near the leading edge is generated farther downstream than that of the baseline impeller. With the small number of blades, the blade load is high and the leakage vortex is strong, but the vortex moves into the right side passage of the splitter blade. As a result, the total loss is reduced by suppressing the secondary flow in spite of the higher maximum loss compared with the baseline impeller at the right side passage of the splitter blade outlet. This shows that the cause of the higher efficiency of OPT1 is the suppression of secondary flow, the effect of the blade load distribution in the blade height and streamwise direction attained in the optimized design.

Figure 11 (c) shows the results of fluid dynamics of OPT2. The flow structure and loss generating mechanism of OPT2 are almost the same as OPT1. The secondary flow, which is more than that of OPT1 and less than that of the baseline impeller, is generated at the blade inlet. A significant amount of low energy fluid is accumulated, and is believed to result in lower efficiency. The reason for the significant extension of operating range in OPT2 is due to the vortex breakdown of tip leakage vortex, and the details are described in another report[3].

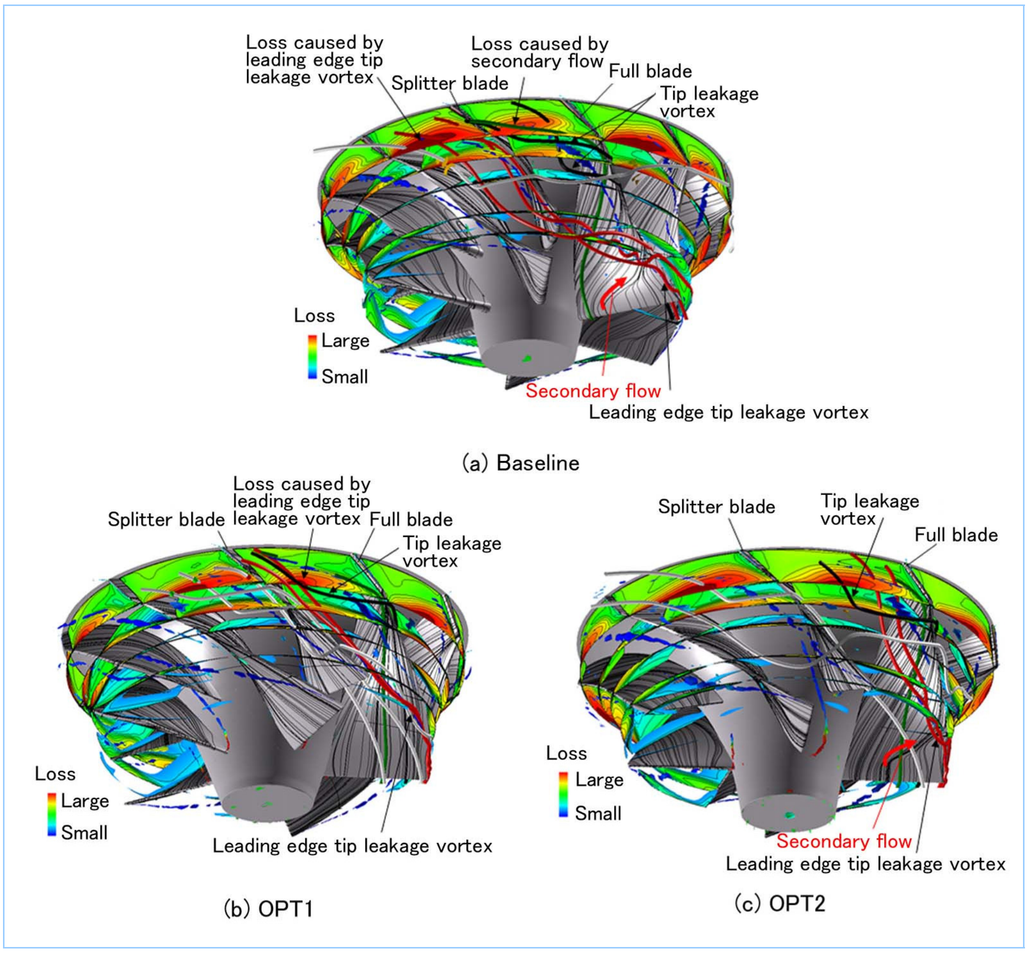


Figure 11 Comparison of internal flow

6. Conclusion

An advanced optimized design method was developed with a genetic algorithm and artificial neural network for performance innovation and design time reduction. The method was also applied to the design of the centrifugal compressor impeller in a turbocharger. The test results of two types of impellers created using the design method showed the improvement of efficiency and the extension of operating range, and this method was verified to be valid. In addition, with the reverse analysis of the obtained impellers using three-dimensional viscous fluid dynamics, the flow structure and mechanism of efficiency improvement were understood.

This optimized design method, which stimulates the utmost usage of computer, can be a powerful tool for the increasingly difficult performance improvement and the acceleration of development speed. We will proceed with upgrades of the optimized design method such as the simultaneous optimization of multiple operating points and constituting parts, aerodynamic performance and reliability and the improvement of the optimization algorithm. Furthermore, we would like to create a new performance improvement concept by utilizing fluid dynamics and experiments for the reverse analysis of the attained optimized design solutions, and utilize it for the formulation of guidelines for performance improvement design.

This optimized design method is a joint development with the Von Karman Institute for Fluid Dynamics in Belgium, and we would like to express our gratitude to Honorary Professor Prof. René Van den Braembussche, Associate Professor Prof. Tom Verstraete and Dr. Zuheyr Alsalihi.

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