A NEW METHOD FOR PREDICATING THE PERFORMANCE MAP OF A SINGLE STAGE OF A CENTRIFUGAL COMPRESSOR

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ABSTRACT

Industrial compressor manufacturers have developed optimized compressors for satisfying various customer requirements with maximizing their system efficiency and minimizing development cost and time. In order to perform these conditions, the accurate performance prediction of a specific design condition is very important. Many researches about the performance prediction methodology have been suggested for improving the accuracy of predicted results compared to real test results. This study suggested new method of the performance of a single stage of a centrifugal compressor. Firstly, in order to verify the reliability of the numerical analysis method performed in this study, simulated performance results were compared with experimental results performed at component test rigs. It was found that the predicted performance map was good agreement with the measured performance map. Secondly, the performance and flow characteristics of a single stage for various impeller design parameters such as flow-cut, machined Mach number, tip clearance, impeller scaling were analysed. Next, based on the numerical results, a new method for predicting the performance of a single stage was developed. The generation method of performance map for design parameters consists of generating two different analytic equations. A first equation was the analytic equation for determining the variation of on-design performance for design parameters. A second equation was the analytic equation for determining the variation of off-design performance. Finally, in this study, the predicted performance results were compared with measured results in order to verify the accuracy. It was found that the performance map predicted by a new method suggested in this study was good agreement with the experimental results.

INTRODUCTION

Industrial centrifugal compressors are used in a wide variety of area of the industry. The compressor is used for supplying a pneumatic gas to an industrial machinery, for recompressing the gas boiled at LNG tank to be reliquefied and for compressing fuel gas in order to supply the gas to turbine engine. Common customer requirements of a centrifugal compressor are the gas composition ratio, the inlet / outlet pressure, temperature conditions and cooling condition, which are very diverse. In order to satisfy the diverse requirements, industrial compressor manufacturers have their standard compressors. However, the standard compressor is usually operated at an off-design condition because the design condition of a standard compressor is different with the real operating condition in site, which means the performance operated at the site condition is not maximum efficiency. Due to the drawback of the standard compressor, many customer request an optimized compressor suitable for their operating condition. The demand for a very high efficient compressor has become stronger in recent as energy efficiency is increasingly important. Many researches and developments were performed for satisfying the customer's requirement. These efforts, however, increase development costs and time, which reduce company profits. Therefore, in order to reduce the development time and cost, most of manufacturer are using specific design process based on the geometry and performance database developed during a long period. The geometry and performance database of aerodynamic parts (for example, impeller, diffuser, scroll and in/out pipe) are very important because the aerodynamic performance affects dominantly on compressor performance.
In this paper, the performance and flow characteristics of a single stage for various impeller design parameters such as flow-cut, machined Mach number, tip clearance, impeller scaling were analyzed for predicting an accurate performance for a specific design. In order to verify the reliability of the numerical analysis method performed in this study, simulated performance results were compared with experimental results performed at component test rigs. It was found that the predicted performance map was good agreement with the measured performance map, which illustrated computational results can predict accurately the performance of a single stage for various design parameters. Compressor performance for a wide flow range from the choking point to surge point was determined in terms of non-dimensionless parameters (flow coefficient, head coefficient, isentropic efficiency and work coefficient) in order to eliminate the influence of geometric and operating conditions.

The numerical and experimental results performed in this study indicated the performance characteristics depended highly on the design parameters and showed very different patterns. The different changes in performance and flow characteristics for each design parameter make it difficult to predict the performance map of single stage. The difficulty causes the difference between predicted design performance and the actual test results, which acting on the main cause of degrading the reliability of the entire compressor system, and which increased develop cost. Therefore, many numerical and experimental researches were performed in order to predict more accurately the performance characteristics for various design parameters. Some researches investigated various 1D loss models for each design parameters of a centrifugal compressor. Other researches have tried to suggest some analytic equations with the correction of physical effect for predicting stage performance characteristics. In this study, based on a lot of experimental and computation results performed for more than 15 years, a new method for predicting the performance of a single stage was developed.

The generation method of performance map for design parameters consists of two steps. The first step was to define the analytic equation for the variation in performance at best efficiency point for changing design parameters. Previous results analyzed the performance characteristics by using the performance map at the flow range between the surge and the choking. However, flow conditions at the surge and the choking have very complex flow characteristics due to strong turbulence and unsteady phenomenon in the internal compressor. The complex flow characteristics reduces the accuracy in the prediction and measurement of performance. It was thus found that it was very difficult to evaluate the compressor performance for various design conditions. Therefore, this study evaluated the performance at the best efficiency point with the best stable flow characteristics, which could minimize an error and an uncertainty. The second step is to determine the analytic equation for the off-design performance. The equation was developed by analyze the performance characteristics of the compressor due to changes in the flow rate based on the best efficiency point for each design parameter. It was found that the performance map predicted by a new method suggested in this study was good agreement with the experimental results. In addition, these results illustrated that this methodology could predict the performance of a single stage of a centrifugal compressor for various design parameters of other components such as IGV (inlet guide vane) and VGD (variable geometric diffuser).

**Numerical Methodology**

In this study, two different computational models were selected in order to analyse the performance map of a single stage for various design parameters of air-ends of a centrifugal compressor as shown the figure 1. The first model, called as a partial model, consisted as a single passage impeller and a single passage vanless or vaned diffuser. The computational model could predict the effect of the design parameters of an impeller and a diffuser on the performance characteristics of a single stage by low computation cost. The interface between the rotating domain (impeller) and the stationary domain (diffuser) was specified as the non-matching and the mixing surface option, which the average properties at one side of interface were transferred to the other interface side. However, the partial model could not predict the interaction between the inlet and the impeller and between the diffuser and the scroll. The other model, called as a whole model, was thus selected. The whole model included the inlet part (consisted of an inlet pipe, an inducer and a tie-shaft), the whole passages impeller, the whole passage diffuser, and the scroll. Non-slip and adiabatic wall boundary conditions were applied at all solid wall such as impeller, diffuser vane and casing and the solid wall had roughness wall condition as well. The inlet boundary condition was defined by total pressure and total temperature. A mass flow rate was specified as the outlet boundary condition. The k-omega SST two-equation turbulence model, with modified wall function, was used in the study. The boundary layer was resolved to be less than 10 of an average y+ value. The total mesh number of the computation model was approximately 4.5 million to a partial model and 20 million to a whole model. In this study, the commercial CFD program, ANSYS CFX V15, was used as the computational solver. The computation to have converged when the root-mean-square (RMS) residual value was less than 10^-6 or the oscillation of the difference of inlet mass flow rate and outlet mass flow rate was less than 0.1%.

![Figure 1. Two different computation models. The partial model (left) and the whole model (right)](image-url)
Validation of Numerical Results

Before analysing the flow/performance characteristics of a single stage according to the design parameter, the comparisons between the experimental results tested at two different test facility for three different impeller types and the numerical results for corresponding geometries were performance in order to verify the reliability of computational results.

First validation case was a Ns 0.7 shrouded impeller. Figure 2 shows the component test rig and air-ends test section for an impeller and a vaned diffuser. The test was performed by Southwest Research Institute. As shown in figure 3, the numerical results of the shrouded impeller were good agreement with the experimental results with less than 2% difference of head and efficiency at the full range of inlet mass flow rate (From choking to surge).

Second validation case was a Ns 0.5 open impeller with a vaneless and a vaned diffuser. As shown in the figure 4, the numerical results for the case had very similar performance characteristics to that of experimental result performed at the component test rig at Hanwha Techwin plant (Figure 5).

Figure 7 shows that aerodynamic performance distributions for the flow cut factor of impeller A. The performance characteristics was expressed in the dimensionless performance parameters such as a normalized head coefficient, isentropic efficiency and flow coefficient were normalized. Because the flow cut factor is directly relative to the area of the flow channel within impeller and diffuser domains, the volumetric flow rate at modified impeller is linearly reduced as flow cut ratio gets smaller. The result shows that the head coefficient is slightly increased since impeller work is also increased (Figure 8) and that isentropic efficiency has a little change with decreasing flow cut factor due to the increase in the friction loss.

Figure 9 indicates that the variation of the performance according to the change of impeller scale factor. It was found that the performance of a single stage for impeller scale factors are almost same. As the impeller scale factor (impeller size) is increased, the head coefficient is increased due to the increase in the impeller work.
diameter) is greater, the head coefficient and the isentropic efficiency of a single stage are also increased. The increase in the performance for increasing impeller scale factor is caused by the increase in the Reynolds Number, which reduces the friction losses generated within flow channel. The results are a good agreement with previous the results for the effect of Reynolds number on the centrifugal compressor as shown in figure 10. As shown in these results, the performance characteristics for each geometric condition have very different pattern.

Variation of performance characteristics of a single stage for operating conditions

Other methods for satisfying customer’s performance requirements is the change of operating conditions of a compressor such as rotating speed, inlet pressure and temperature and coolant conditions. In this study, the numerical simulation for various rotation speed was performed as shown in figure 11. In this study, the performance variation for rotating speed of impeller is expressed by the rotation speed factor, Machined Mach number in order to consider only the effect of rotating speed on the performance. The results shows that head coefficient and the peak efficiency is linearly increased with increasing Machined Mach number. It is also found that a surge margin of a single stage is reduced and that the flow coefficient at peak efficiency is also reduced. The reduction in surge margin and flow coefficient is caused by increasing the relative velocity within flow channel.

Figure.7 Variation of the performance for flow cut

Figure.8 Variation of the normalized work coefficient for flow cut

Figure.9 Variation of the performance for scale factor

Figure.10 Variation of the performance for scale factor

Figure.11 Variation of the performance for various Machined Mach number

Variation of performance characteristics of a single stage for impeller types

As mentioned at previous part, many researches investigated various 1D loss models for each design parameters of a single stage in order to predict the performance characteristics of a centrifugal compressor. However, the performance of a single stage of a centrifugal compressor is affected by not only modification parameters but also impeller type and base design point as shown in from figure 12 to figure 15. Figure 12 and figure 13 show that the performance variations according to flow cut factor for two different impeller type. As shown in these results, the performance variation for changing flow cut factor have different pattern.
In this study, in order to compare the performance variation for the same design parameter of different impeller type and diffuser type, the ratio of non-dimensional performance parameters, which determined the ratio of non-dimensional performance for modified design to these for a base design, was used. Figure 13 and figure 14 indicates the ratio of performance variables of two different impeller type with/without vaned diffuser for various flow cut ratios. As shown in these figures, the performance variation for the same design parameter have different pattern, which illustrates it is difficult to predict performance map by 1D losses models for each design parameters without design concept for each impeller.

![Figure 13 Variation of the performance according to flow cut factors for the B impeller](image)

**Figure 13** Variation of the performance according to flow cut factors for the B impeller

The ratio of flow coefficient of impeller type and diffuser type for flow cut factors

![Figure 14 The ratio of flow coefficient of impeller type and diffuser type for flow cut factors](image)

**Figure 14** The ratio of flow coefficient of impeller type and diffuser type for flow cut factors

The ratio of isentropic efficiency of impeller type and diffuser type for flow cut factors

![Figure 15 The ratio of isentropic efficiency of impeller type and diffuser type for flow cut factors](image)

**Figure 15** The ratio of isentropic efficiency of impeller type and diffuser type for flow cut factors

Method to predict the performance map for design parameters

As shown in previous results, the performance of a single stage of a centrifugal compressor is highly depended upon the design parameters and impeller type. These results indicated that it is very difficult to predict the performance for a specific design. Therefore, previous studies evaluated the performance characteristics at the whole operating ranges from surge to chock and suggested the method for predicting the performance map for design parameters. However, the flow within the internal channel at impeller and diffuser at the surge and chocking is very complex due to a strong turbulence and unsteady phenomenon. This causes the very low accuracy in the numerical prediction and the measuring data performed at experiments. In order to improve the prediction accuracy, this study analyzed the performance characteristics by the variation of the operating condition at best efficiency point (BEP, peak efficiency point) and the performance at the condition. The performance variation was evaluated in terms of the ratio of non-dimension performance parameters at a specific design to these at base design. Based on these analysis results, the new method for predicting the performance of a single stage of a centrifugal compressor.

The first step for predicting the performance map is to evaluate the variation in performance at best efficiency point according to design parameters. Figure 17 and figure 18 shows the flow coefficient and isentropic efficiency distributions at best efficiency point for changing impeller scale factor for two different impeller type. The change in performance for a specific design can predicted by the loss models for each design parameter. Hanwha techwin is developing the loss model for each parameter as well as the interaction losses between design parameters by many numerical and experimental researches.

![Figure 16 A first step of performance map generation : Defining the change of the operating conditions at best efficiency point (BEP)](image)

**Figure 16** A first step of performance map generation : Defining the change of the operating conditions at best efficiency point (BEP)

![Figure 17 The change of best efficiency point (BEP) for different two impeller with vaneless diffuser](image)

**Figure 17** The change of best efficiency point (BEP) for different two impeller with vaneless diffuser
The second step is to predict the performance at off-design conditions, expressed at performance map, at the operating ranges between the surge to the chock. In order to generate the performance map of a single stage of a centrifugal compressor, this study developed the analytic equation of the variation of head coefficient and isentropic efficiency in accordance with flow coefficient for each different impeller types and different design parameters as shown in figure 19. The variation of dimensionless performance variables was normalized by the performance value of base model.

Figure 20 and figure 21 shows that the analytic equation of the variation in the head coefficient and isentropic efficiency with the variation in the flow coefficient for two different design parameters such as flow cut factor and impeller scale factor. As shown in these results, the equation is not depended upon these design parameters because the change rate of work coefficient to flow coefficient is not varied with the design parameter. However, the equations for performance variation are linearly changed with the value of machined Mach number as shown in figure 22 and figure 23. The change is caused because the change rate in work coefficient is highly proportioned to the value machined Mach number.

In this study, firstly, the variation of the performance at best efficiency point for machined Mach numbers and tip clearances was defined by numerical results. The analytic equation for on-design performance was then corrected by experimental results. Next, the analytic equation for off-design performance was confirmed by only numerical results. Final step is comparison between predicted results performed by a new method and experimental results (Head vs Flow) in order to evaluate the accuracy of a new method as shown in figure 22 and figure 23. These figures indicate that the predicted results are good agreement with the experimental results. These results illustrates that the new method for generating performance map can predict the performance of a single stage of a centrifugal compressor.
CONCLUSIONS
In this study, firstly, numerical analysis for flow characteristics of a single stage for various impeller design parameters such as flow-cut, machined Mach number, tip clearance, impeller scaling have been performed in order to analyze the performance characteristics of these design parameters. From these results, it was found that the performance of a single stage of a compressor was highly depended upon the design parameters and impeller type. It was also indicated that it is very difficult to predict the performance for a specific design. Therefore, this study suggested a new method for predicting the performance of a single stage by using many computational results as well as the experimental results performed. The generation method of performance map for design parameters consists of two different analytic equations. A first equation was the analytic equation for determining the variation of on-design performance for design parameters. A second equation was the analytic equation for determining the variation of off-design performance. In this study, finally, comparison between predicted results performed by a new method and experimental results. These results showed that the predicted results are good agreement with the experimental results. These results illustrates that a new method for generating performance map can predict the performance of a single stage of a centrifugal compressor.

NOMENCLATURE
- $\psi$: Head Coefficient
- $\phi$: Flow Coefficient
- $\eta$: Isentropic Efficiency
- $q_s$: Work Coefficient
- $N_s$: Specific Speed
- FC: Flow Cut Factor
- SF: Impeller Scale Factor

REFERENCES.