EFFECT OF AREA RATIO OF INLET TO OUTLET OF AN IMPELLER ON PERFORMANCE OF A TWO-DIMENSIONAL CENTRIFUGAL COMPERSSORS

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Abstract: This study is performed to understand the effect of the variation in the passage area of an impeller on its performance characteristics. The commercial computational fluid dynamics (CFD) code was used to determine the performance characteristics of a two-dimensional impeller A comparison between the CFD code and the experimental results was performed for a similar configuration in order to verify the reliability of the CFD code. Overall characteristics in the passages of impeller were analyzed in detail including streamline, pressure distribution and polytropic efficiency. When the passage area ratio exceeds 2, the pressure ratio is high. An area ratio of 2.3 showed the highest efficiency.

Key Words: MGT, two-dimensional centrifugal compressor, performance improvement, CFD

1. INTRODUCTION

Over the years, demands from customers for the development of portable products with diversified functions have been increasing. As a result, there has been an increase in the interest in new electric technology to provide an increased amount of electricity. This study is actively conducted for small energy sources such as micro-fuel cells and micro-gas turbines (MGTs).

Since Epstein *et al.* [1] introduced the first concept of MGT with MEMS technologies, MGT has been vigorously researched [2, 3, 4]. Several researchers have also proceeded with the experimental and numerical studies for performance characteristics of the MGT through scale-up models [5, 6, 7].

The flow characteristics of two-dimensional impeller were analyzed by numerically to find the way to improve the performance. From the analyzed results, it can be concluded that these parameters can be used to determine the cause for the low performance of the two-dimension centrifugal compressor and to improve the performance. In exploring the questions of performance improvement, this paper will be limited to the consideration of the effect of the variation in the passage area of an impeller.

2. NUMERICAL ANALYSIS

2.1 Analysis Model

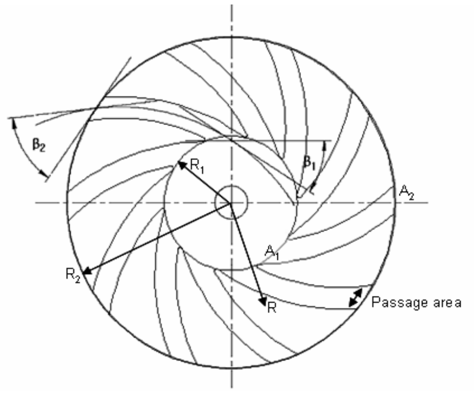


Fig. 1: Baseline design of the 2-D impeller

Fig. 1 shows major parameters on the design of a two-dimensional impeller. The passage area of the two-dimensional impeller is controlled by a thickness of the blade. Generally, the passage area is designed uniformly to control the flow separation [2]. However, in case of the changed passage area, we made an observation of performance improvement [7]. Therefore, for understanding the effect of the variation in the passage area of an impeller on its performance characteristics, we observe the results with changing the area ratio of inlet to outlet about 1~2.8.

2.2 Boundary Conditions

The computational analysis is performed for a two-dimensional centrifugal compressor whose conditions are identical with the test conditions. It is assumed that the flow is a compressible steady state and impellers have a symmetry condition due to periodic type. Therefore, the calculation is conducted only one flow passage out of ten. The

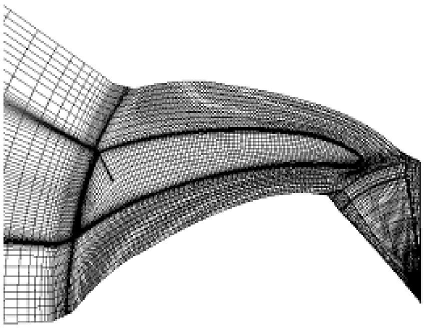


Fig. 2: Grid Structure of an impeller

inlet boundary conditions are total pressure (1*bar*) and total temperature (25˚*C*), while the outlet condition is changed by mass flow to obtain the performance curve.

2.3 Turbulence Modeling and Grid Generation

When Reynolds number is 10^5 or larger, turbulence flow occurs in turbomachinery. Therefore, the turbulence models used clearly govern the results of numerical analysis in order to solve the turbulent flows. One of the main problems in turbulence modeling is the accurate prediction of flow separation from a smooth surface. The Shear Stress Transport (SST) turbulence model can predict the scale and start of the flow separation in reverse pressure distributions because it considers the transportation of turbulent shear stress. Therefore, the SST turbulence model is employed in this simulation. The SST turbulence model is based on the k-ω model and it depends on the location closest to the walls in the predicted performance; therefore, it requires a low y+ value. y+ is organized to be grid to have the value within 2 in this study. The number of grids used in this study was determined to be about 0.4~0.3 million for the compressors with various impellers. Fig. 2 shows the grid used for the computation.

(a) Impeller A *(b) Impeller B*

Fig. 3: Photograph of impellers

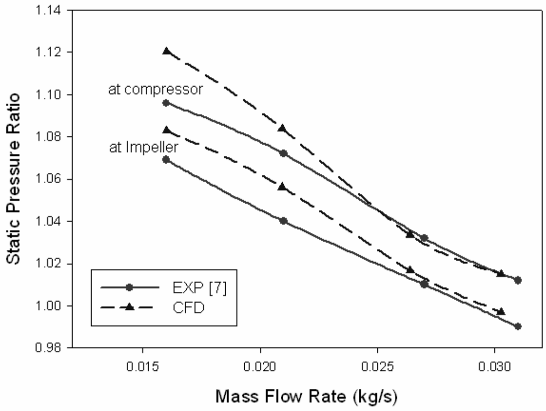


Fig. 4: Comparison with respect to experiment

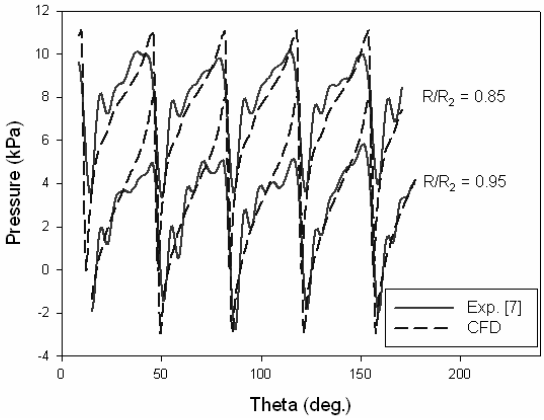


Fig. 5: Wall pressure distribution of impeller B

2.4 Verification of Numerical Code

The experiments are costly and time consuming. It must be verified whether the CFD results are consistent with the experimental results. Therefore, we compare performance curves with the experimental and computational results. Fig. 3 shows impellers using the experiment [7]. The passage area of impeller A is constant and the passage area of impeller B is uniformly increased. Fig. 4 shows pressure ratio of impeller A into changing mass flow. Fig. 5 shows the comparison the wall pressure of inside points (R/R_2=0.85, 0.95) of impeller B. Pressure distributions of both the experimental and numerical results are dependent on the impeller rotation. They exhibit a tendency to a form of a cubic equation curve. The accelerated zone is in the passage area and the decelerated zone is in the blade area. The pressure curves from the experiment have not distinct shapes and tend to have round edges because there might be a slight shift of data during the average-out process. Therefore, the computational results exhibit a tendency to consistent with experiments. From the results, the computational analysis for the techniques and the grid is considered as available and the numerical analysis result of the various shapes obtained by the same method is considered to be available. Moreover, as the detailed data was impossible to obtain in the existing experiments, it is useful to improve the performance by analyzing the flow characteristics of the two-dimension centrifugal compressors.

3. RESULTS AND DISCUSSIONS

3.1 Performance Characteristics

In order to effectively compare different shapes impellers, the flow coefficient (φ) and pressure coefficient (ψ) are defined (see equ. 1 and 2).

$$\phi = \dot{m} \big/ \rho_1 \pi r_1^2 U_2 \qquad (1)$$

$$\psi = \Delta p \big/ [1/2(\rho_1 U_2^2)] \qquad (2)$$

Fig. 6 shows the variation in the pressure coefficient with respect to the area ratio (A_2/A_1). The increase in passage area is decreased a loss part of the jet element. However, it is considered that the loss is increased as flow separation occurs easily due to a large adverse pressure gradient and low Reynolds number. When the variation in the area ratio exceeds 2, the advantage and disadvantage of this variation in passage area is offset and there is no notable variation in the performance at some level. Fig. 7 shows the variation in polytropic efficiency (see equ. 3) with respect to the area ratio.

$$\eta_{pol} = \sum_{i \to \infty} \Delta h_{is,i} \Big/ \Delta H_{real} \qquad (3)$$

The polytropic efficiency initially increases with

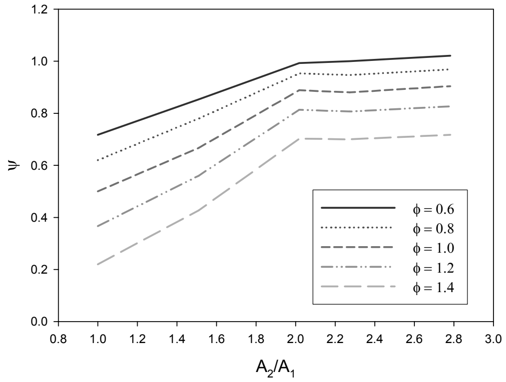


Fig. 6 Pressure coefficient with area ratio

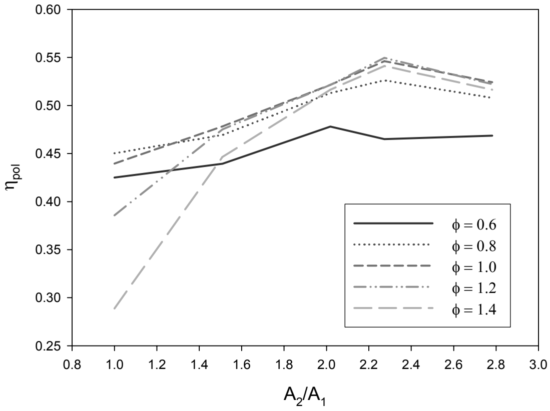
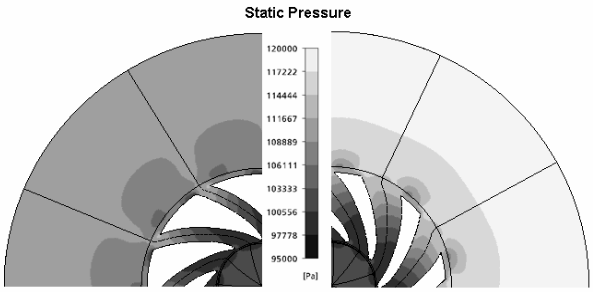


Fig. 7 Polytropic efficiency with area ratio

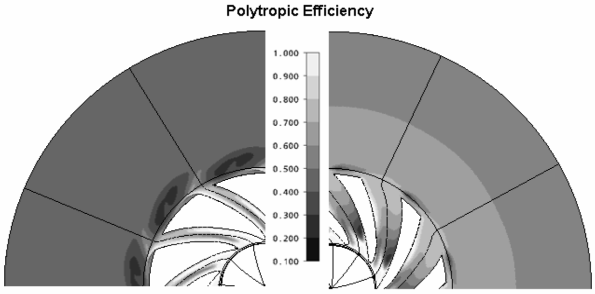
area ratio like the results of pressure curve, however it gradually decreases in most of cases when area ratio exceeds 2.3. In the case of low flow rate, area ratio 2 presents the highest efficiency and area ratio 2.3 presents the highest efficiency in the other cases. If we consider for an overall compressor performance, the best shape of the variation in the passage area is impeller B (A_2/A_1=2.3).

In order to compare the performance characteristics of the two-dimensional centrifugal compressor with regard to the area ratio, computational results of impeller A and B are monitored for same flow coefficients. Figs. 8 and 9 show the pressure and polytropic efficiency



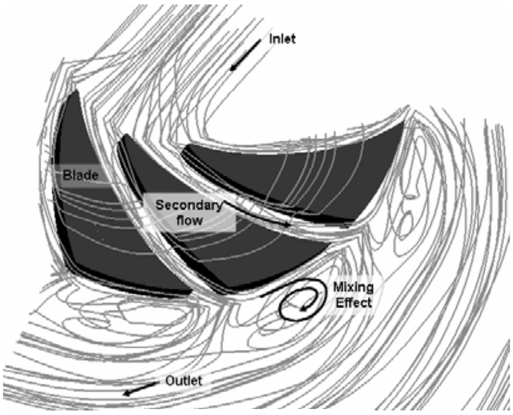
(a) Impeller A (b) Impeller B

Fig. 8: Static pressure distribution at mid-span

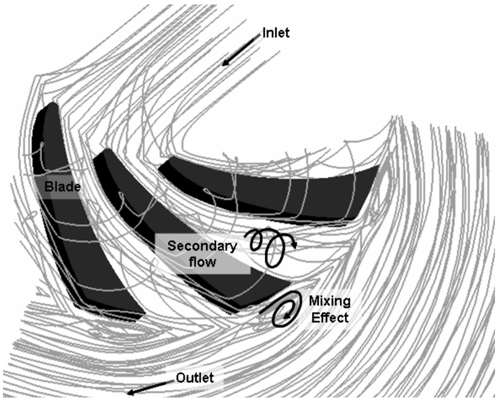


(a) Impeller A (b) Impeller B

Fig. 9: Efficiency contours at mid-span



(a) Impeller A



(b) Impeller B

Fig. 10: Streamlines of 2-D compressors

distribution of each impeller. The smooth connection of the characteristics of impeller B without rapid variation can be observed than impeller A.

3.2 Flow Characteristics

Understanding the flow characteristics from the experiment is difficult. However, the numerical investigation is useful for an observation all over. Therefore, the computational investigation is suited to understand the flow characteristics. Fig. 10 shows the streamlines inside of each impeller. The flow is bent by 90 degree rapidly to a radial direction in impeller inlet due to two-dimensional impeller. Therefore, the inlet loss of the impeller is inevitable; however the exits loss of the impeller should be reduced. Impeller A has a consistent passage area; hence, the loss due to secondary flow is smaller than that in impeller B; however, it is found that the flow from the impeller has a large pressure loss in the diffuser entrance due to a rapid diffusion.

4. CONCLUSIONS

Up to now, we have looked at the effect on changing the flow passage area of two-dimensional impeller. As the passage area is constant, the loss by secondary flow was low. However the heavy loss happened by mixing effect in the impeller exit. When the passage area increases, the loss was covered at the impeller exit but increased by secondary flow. The optimum area ratio of inlet to outlet is 2.3 in veiw of the results so far achieved. In the future, the study on other major parameters for two-dimensional impeller design should be conducted. This result will establish the concept design of the two-dimensional impeller and will be used as reference of performance improvement.

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