A STUDY ON DESIGN OPTIMIZATION OF CENTRIFUGAL FAN OF HIGH VOLTAGE GENERATOR BY NUMERICAL ANALYSIS

Chungman JANG, Joonyeob LEE, Moojong JEON

Hyundai Electric & Energy Systems Co., Ltd., Rotating Machinery R&D Department., 17-10 Mabuk-Ro, 240beon-Gil, Giheung-Gu, Yongin-Si, Gyeonggi-Do, Republic of Korea

SUMMARY

The purpose of this study is to derive the optimal centrifugal fan design for a maximum 5 MW class high voltage generator. Centrifugal fans serve to limit temperature rise due to generator losses. For optimum cooling fan selection, flow resistance was calculated through an equivalent circuit program and numerical analysis. Four kinds of fans according to the blade shape considering the manufacturability, cost, and performance were examined. Numerical analysis and fan performance tests were conducted to verify the performance of the fan. As a result, it was possible to secure the reliability of the analysis results and to select the optimum cooling fan for the target generator.

INTRODUCTION

A generator is a device that converts the mechanical energy provided by an engine, a gas turbine, a steam turbine, etc. to electric energy. Heat is generated during the energy conversion process and the internal temperature of the generator is increased. A cooling fan is used to suppress the generator temperature characteristic within the numerical value in the standard. Choosing a cooling fan in a generator cooling design is a key core technology. The required air volume and required static pressure must be satisfied and the influence of the mechanical loss and noise must be considered. If the fan selection is not appropriate, it will affect the generator performance and increase the. Therefore, an effective cooling fan design is important for the selection of an optimal cooling fan. As shown in Figure 1, the high voltage generator consists of a rotor, a stator, a winding, a cooling fan, and a heat exchanger. The internal structure is complicated, and the flow resistance is generated as the cooling channel is either enlarged or reduced. Flow resistance is an important factor in determining the operating point of the fan. The system resistance is generally calculated by an equivalent circuit program or a numerical analysis based on the empirical formula, and Figure 5 is the system resistance curve of the fan selected for the generator. The air flow rate of the fan operating point of a 5MW class high voltage generator is about 5.94 m^3/s and the static pressure is about 1087Pa. We provided the system resistance curve according to the fan operating point to a fan specialist company and designed four kinds of fans according to the shape of the blades. A numerical analysis was carried out using the widely used ANSYS FLUENT to examine the performance of the four supplied fans, and the results were compared through an ANSI/AMCA standard 210-99 fan performance test. Figures 2 to 4 show the distribution of temperature rise, wind speed and pressure drop through the numerical analysis and a centrifugal fan with relatively good static pressure performance is used considering the flow resistance and heat exchanger design characteristics.

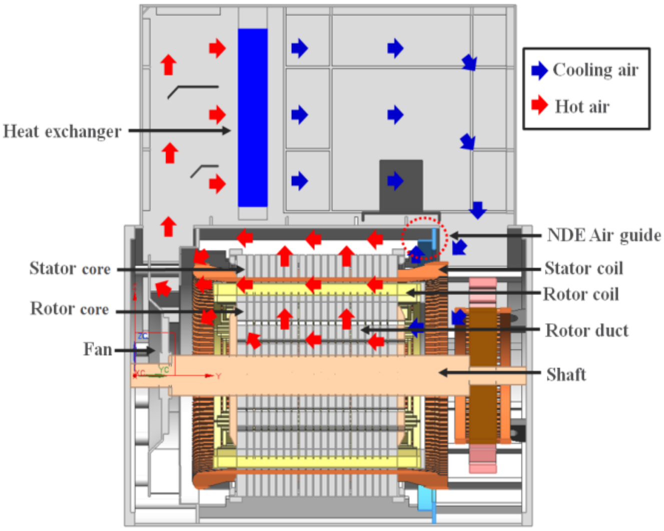


Figure 1: Schematic design of high voltage generator

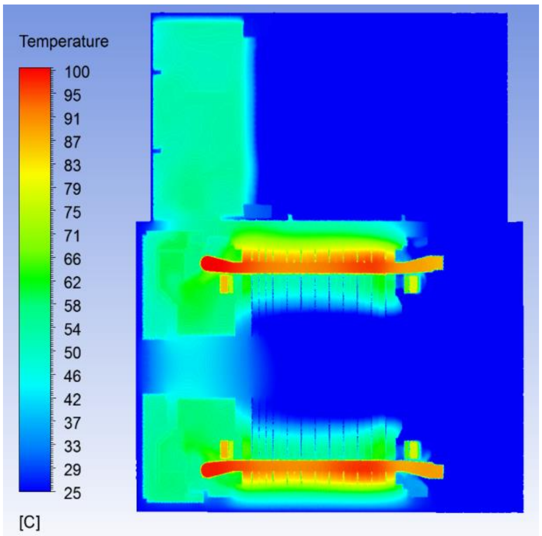


Figure 2:Temperature distribution of numerical analysis

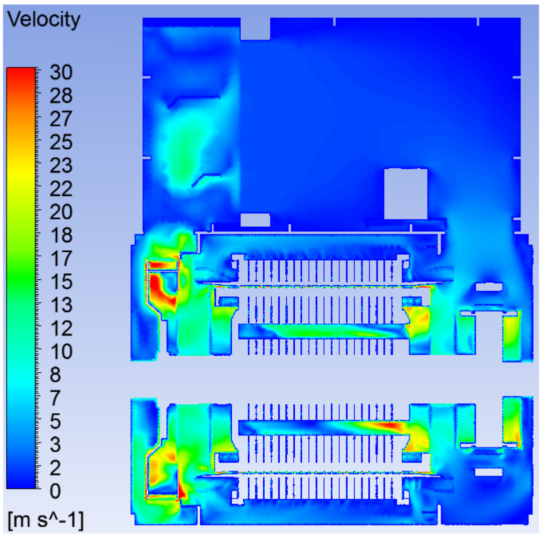


Figure 3:Velocity distribution of numerical analysis

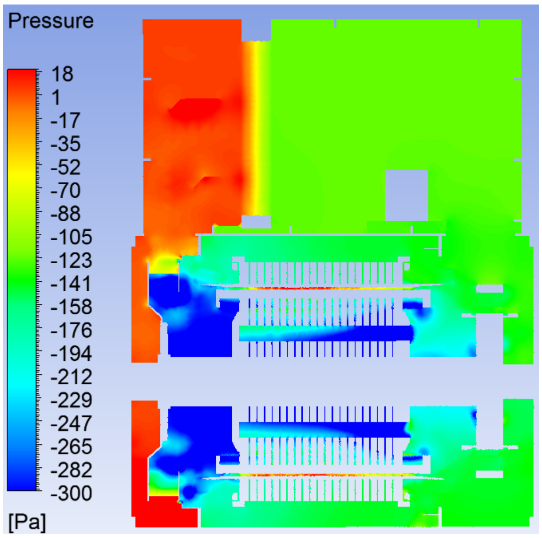


Figure 4: Static pressure distribution of numerical analysis

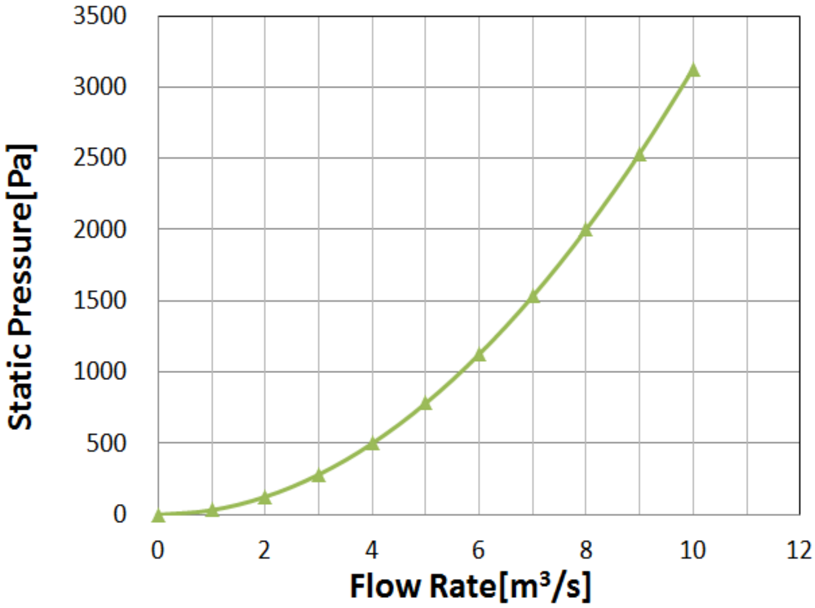


Figure 5: System resistance curve of 5MW generator

CENTRIFUGAL FAN DESIGN

Centrifugal fans normally produce more static pressure than axial-flow fans of the same wheel diameter and the same running speed because of the additional centrifugal force, which is missing in axial-flow fans. According to their blade shape, centrifugal fans be subdivided into the following six categories: AF(airfoil), BC(backward-curved), BI(backward-inclined), RT(radial-tip), FC(forward-curved), and RB (radial blade). Figure 6 shows the efficiency according to the wing shape and generally the AF type can gain high efficiency and has low noise characteristics.

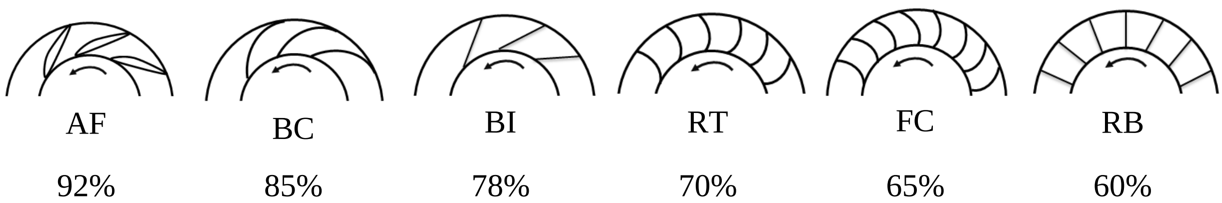


Figure 6: The approximate maximum efficiency attainable for each type is shown

As an indication of the general trend, it may be said that an increase of the blade angles β_1 and β_2 result in an increase in air volume and static pressure, but in a decrease in the fan efficiency. However, if β_1 and β_2 become too large between adjacent blades may become so wide and short that the airflow is no longer sufficiently guided and the circulatory flow will become excessive. In this case, a simultaneous increase in the number of blades will correct the situation by making the blade channel narrower. [1] As shown in Equation (1), the optimum angle of β_1 and β_2 can be selected according to the ratio d_1/d_2, and the fan efficiency and performance differ according to the criterion. Considering the easiness of preparation and cost, four kinds of fans were designed as shown in Table 1. Figure 7 shows the wing shape of each centrifugal fan.

$$\cos\beta_2 = \frac{d_1}{d_2}\cos\beta_1 \qquad (1)$$

Where d_1 = blade inside diameter

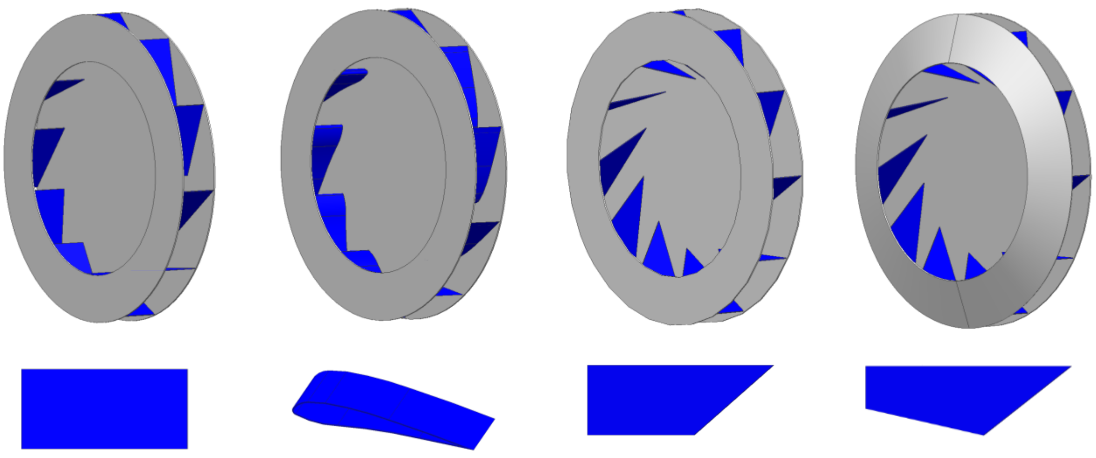
d_2 = blade outside diameter

β_1 = blade angle at the leading edge

β_2 = blade angle at the blade tip for straight blades

Table 1: Design dimensions of centrifugal fan

	Case 1	Case 2	Case 3	Case 4
d_2 [mm]	1340	1340	1340	1340
d_1 [mm]	935	935	935	935
b [mm]	160	160	160	130
β_1 [°]	20	12.7	20	20
β_2 [°]	48.65	45	58.58	58.58
Blade cross section [m^2]	0.0544	0.060	0.05536	0.5824
RPM	900	900	900	900
Blade type	BI	AF	BI	AH
Number of blades	11	10	11	11



(a) Case 1 (b) Case 2 (c) Case 3 (d) Case 4

Figure 7: Geometrical of the centrifugal fan

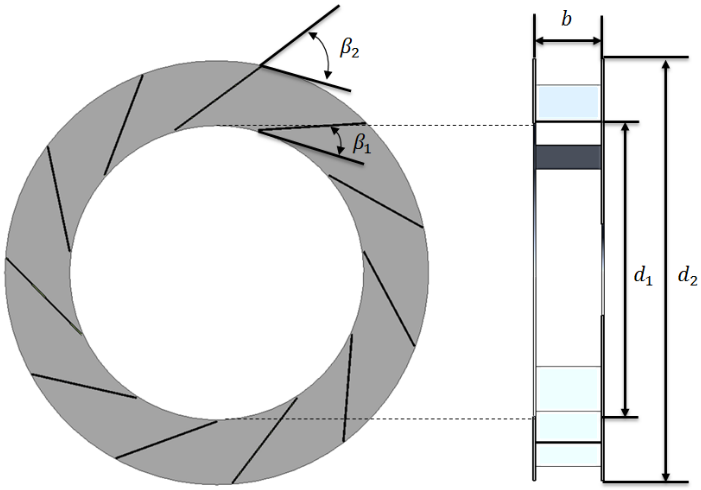


Figure 8: Schematic sketch of a typical centrifugal fan wheel

NUMERICAL ANALYSIS

We used the ANSYS ICEM-CFD grid generation program as shown in Figure 9 to analyze the fan performance characteristics through a numerical analysis, and generated about 6 million grids. The fan housing was modeled to simulate the flow channel structure of the fan installed in the actual generator. The input conditions for the numerical analysis were obtained by using the SIMPLE algorithm as shown in Table 2, and the moving wall was used on the surface of the blade and then set at the rotating condition of 900 RPM by using the MRF method in the air area around the blade. We expected turbulent flow so we used the STANDARD $k-\varepsilon$ model. The pressure condition was entered by fixing the inlet of the fan and changing the pressure value at the outlet of the fan, and the resulting air flow rate was calculated. Also, the test was repeated more than 600 times to satisfy the convergence condition within the allowable error. The fan performance curve calculated from the analysis is shown in Figure 10. As shown in Figure 11 and Figure 12, the pressure distribution and wind velocity distribution were visualized through the CFD-POST to examine the flow characteristics [2].

Table 2: Analysis condition of fluent

Viscous model	Standard $k-\varepsilon$
Solution methods	SIMPLE
Equations	Flow / Turbulence
Solver type	Pressured-Based
Solver time	Steady
Velocity formulation	Absolute

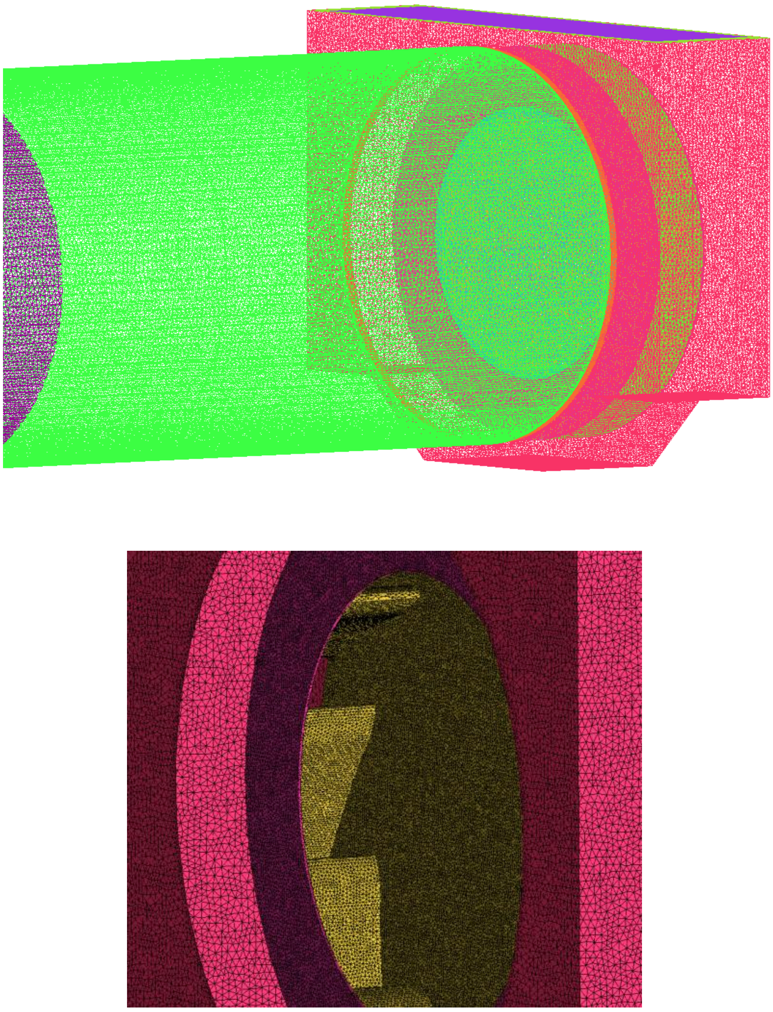


Figure 9: Computational mesh of fan housing and centrifugal fan

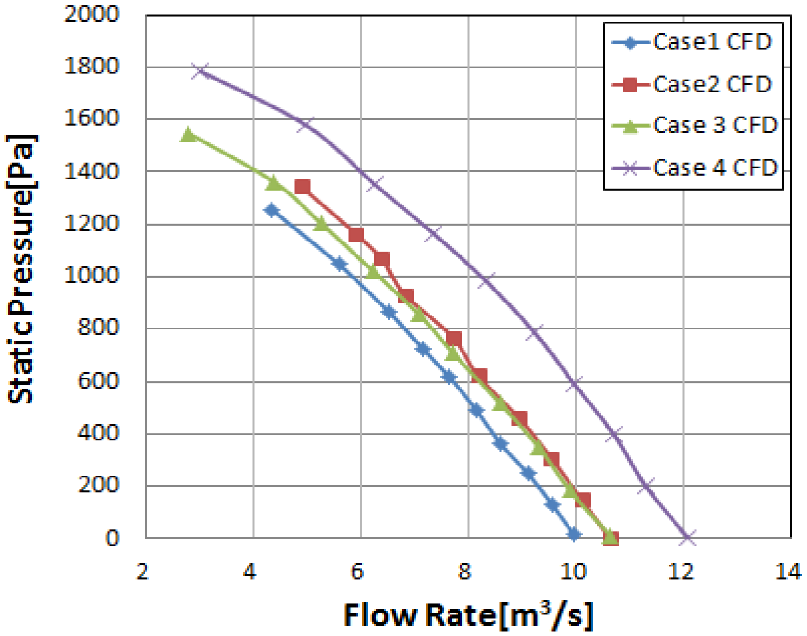
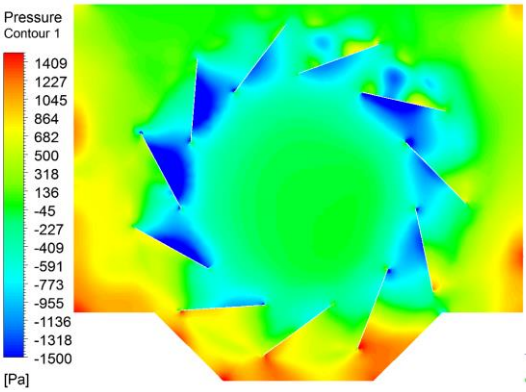
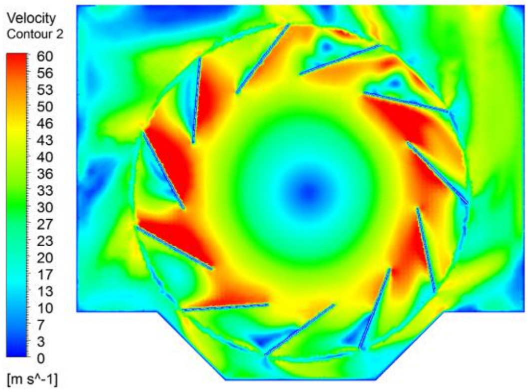


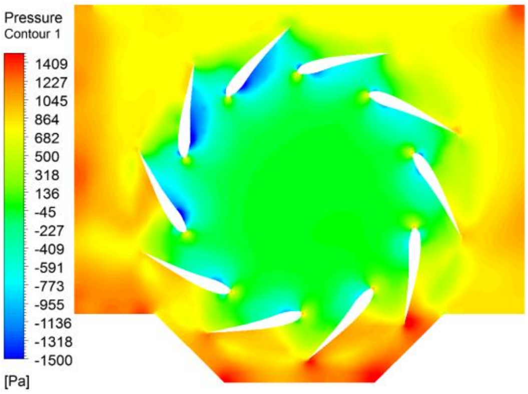
Figure 10:Fan performance curve of CFD results



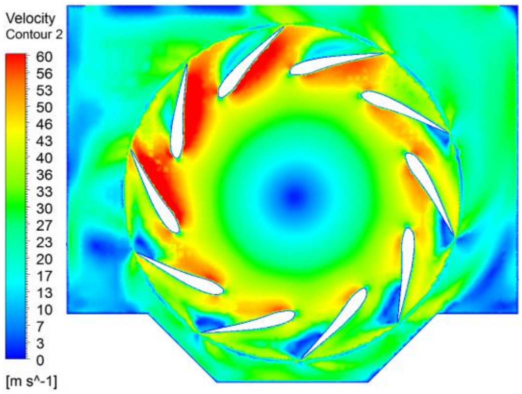
(a) Case 1



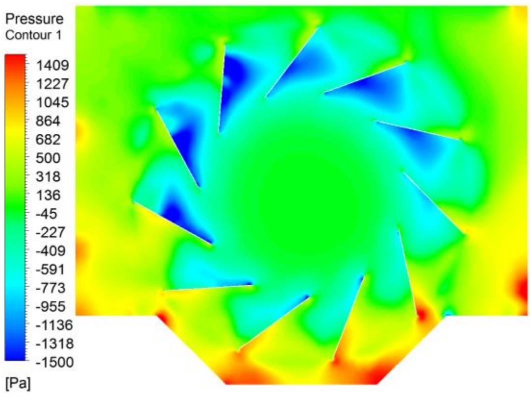
(b) Case 1



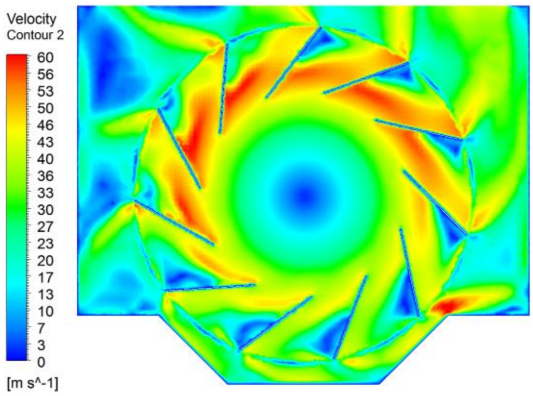
(c) Case 2



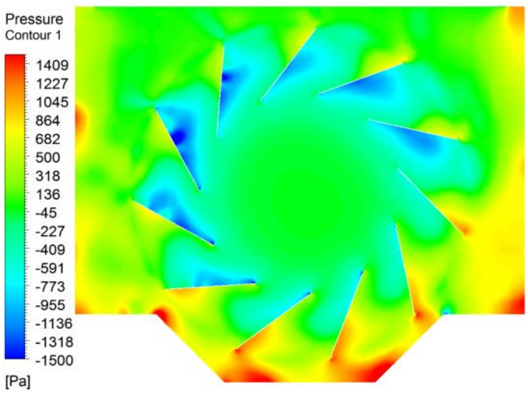
(d) Case 2



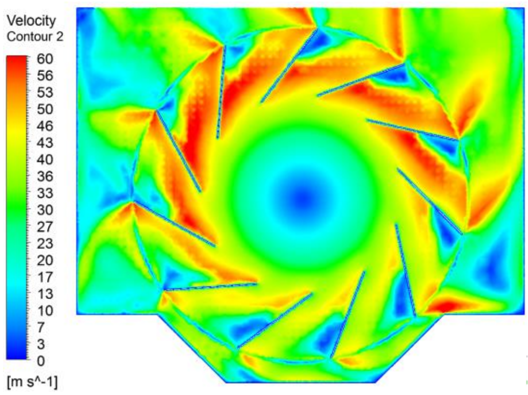
(e) Case 3



(f) Case 3



(g) Case 4



(h) Case 4

Figure 11: Static pressure distribution

Figure 12: Velocity distribution

FAN PERFORMANCE TEST

The performance of a fan can be presented in a curve sheet in which all or some of the following seven variables are plotted against air volume (cfm): static pressure (SP), total pressure (TP), brake horsepower (bhp), motor input (kW), mechanical efficiency (ME), static efficiency (SE), and sound level (dB). The fan performance test was conducted using the ANSI/AMCA standard 210-99 specification as shown in Figure 13. The pressure and voltage are measured through the pressure gauge located in the chamber, and the air velocity and volume can be calculated by the dynamic pressure and the density of the air as shown in Equation (3). Figure 14 shows the type of the cooling fan according to the delivery method and type C was selected in consideration of the installation conditions of the actual cooling fan. A fan jig was manufactured considering the characteristics of the flow structure of the cooling fan as shown in Figure 15. The test was conducted with the atmospheric pressure of 100.75 kPa, the dry bulb temperature of 19.2 ℃, the wet bulb temperature of 11.6 ℃, the air density of 1.20 kg/m^3 and the rotation speed of 900 rpm as the test conditions.

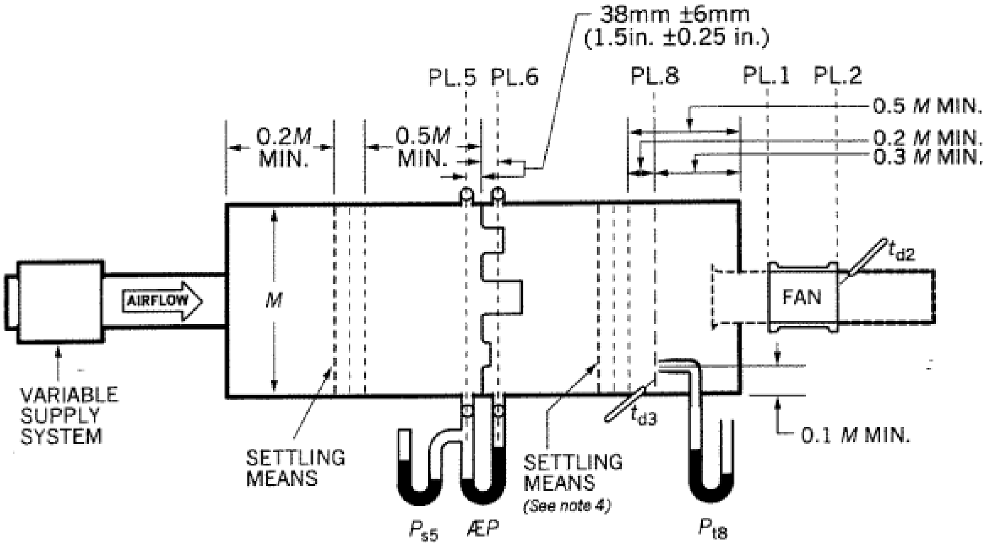
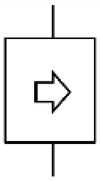
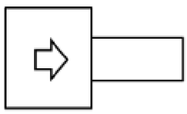


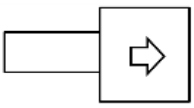
Figure 13: Inlet chamber setup – multiple nozzles in chamber (ANSI/AMCA 210-99)



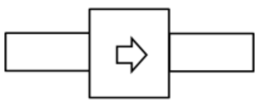
TYPE A : Free Inlet, Free Outlet



TYPE B : Free Inlet, Ducted Outlet



TYPE C : Ducted Inlet, Free Outlet



TYPE D : Ducted Inlet, Ducted Outlet

Figure 14: Types of standard fan installation

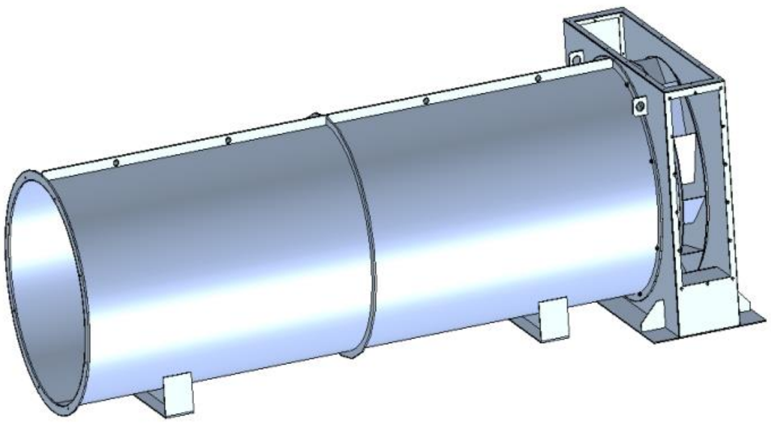


Figure 15: Centrifugal fan jig

$$Q_5 = \sqrt{2}\, Y \sqrt{\frac{\Delta P}{\rho_5}}\ \Sigma(CA_6) \tag{2}$$

$$Q = Q_5 \left(\frac{\rho_5}{\rho}\right) \tag{3}$$

Figure 16 shows the results of the fan characteristics test based on each case. Especially in Case 4, the fan performance showed a relatively large distribution and we determined that it was caused by the blade angle and blade cross-sectional area designed relatively higher than those of the other cases. In Case 2, as the fan performance was relatively low and the number of the blades was designed to be 10, we determined that the pressure drop was relatively low at the same flow rate. As a result, we could confirm the sequential fan performance characteristics according to the blade angle, blade cross-section, and the number of blades.

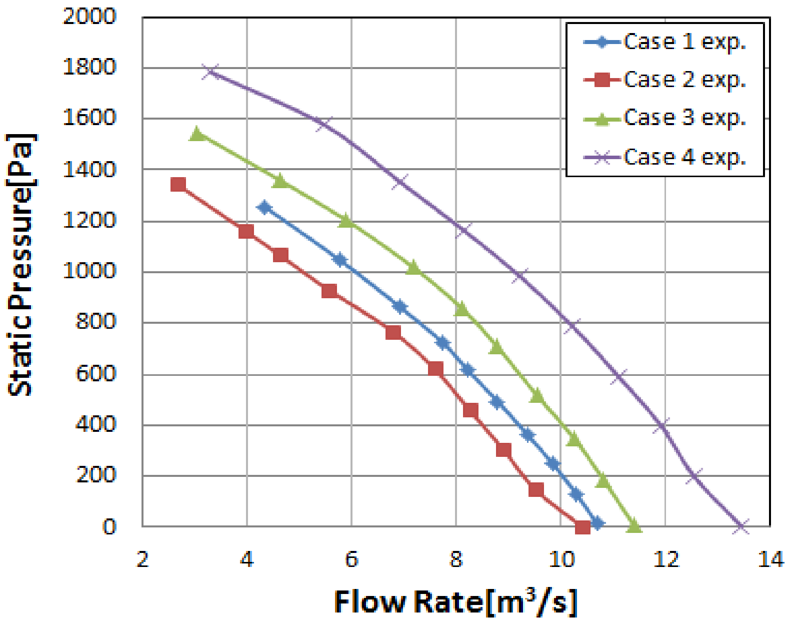
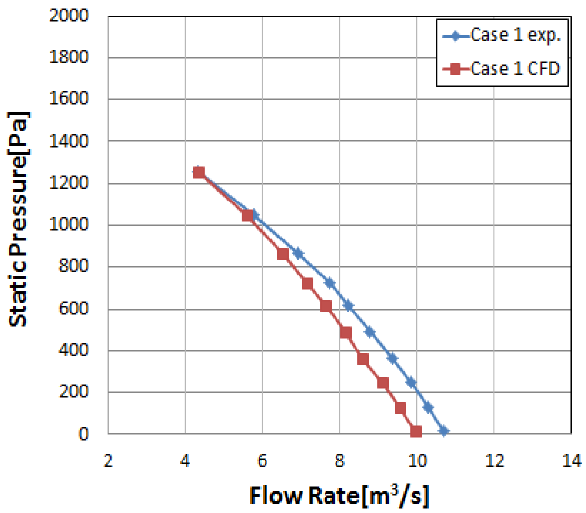


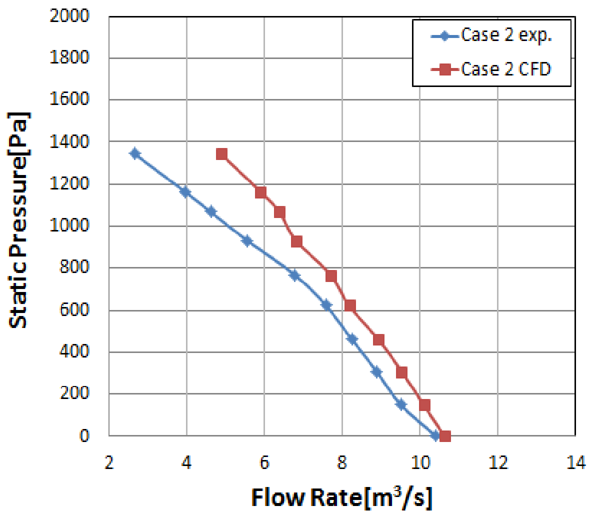
Figure 16: Fan performance curve of experiment results

COMPARE RESULTS

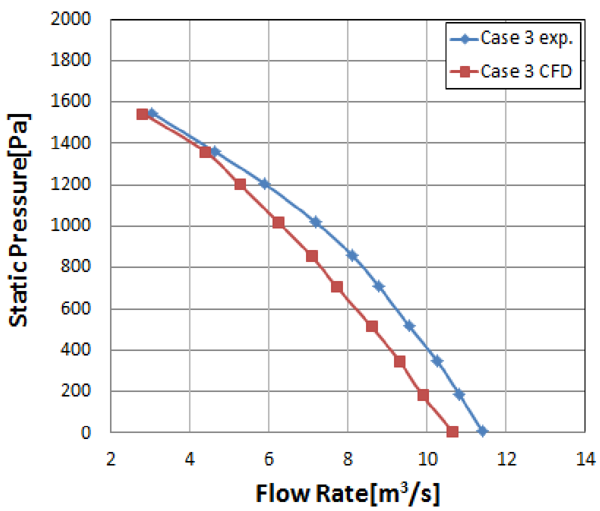
Figure 17 shows the comparison of the performance curves through the numerical analysis and the performance tests. As a result of the comparison, we could obtain a reliable analysis result. However, in Case 2, the analytical results showed higher performance than the test results, and we confirmed that the error range gradually decreased as the flow rate increased and in all cases except Case 2, the analytical results showed lower performance than the test results and finally as the flow rate increased, the error range increased. In order to reduce the error range of future analysis and tests, we plan to increase the accuracy of analysis by examining the grid density and analysis technique.



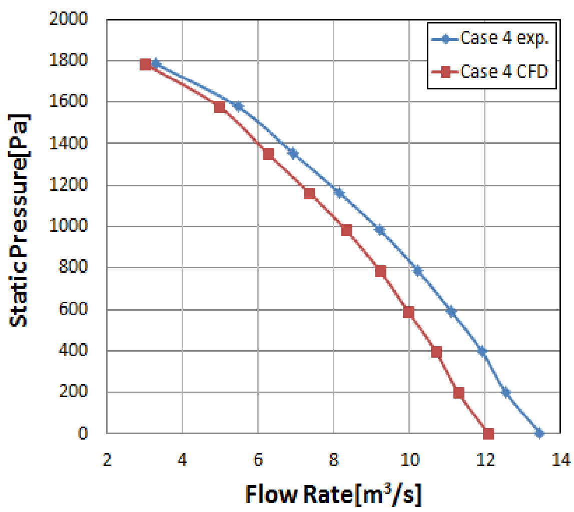
(a) Case 1



(b) Case 2



(c) Case 3



(d) Case 4

Figure 17: Centrifugal fan performance curve Q-H

CONCLUSIONS

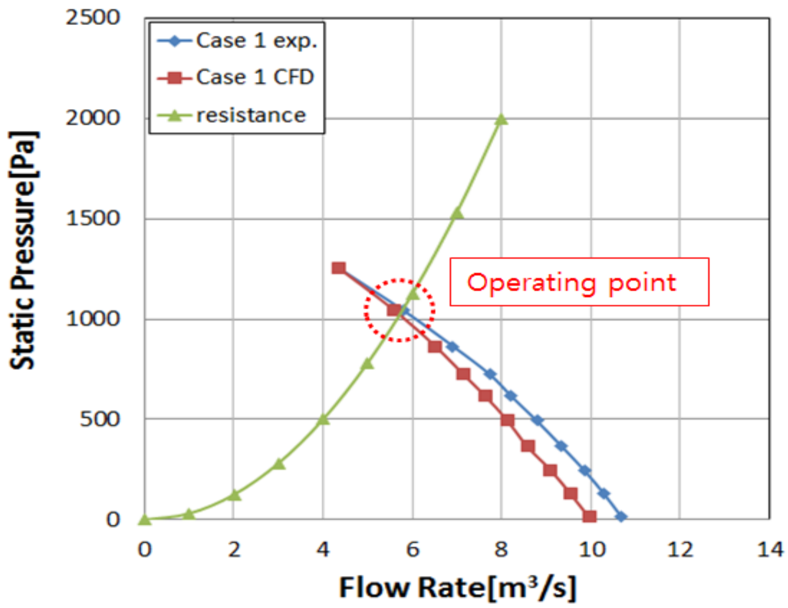


Figure 20:Fan operating point

The purpose of this study is to derive the optimal centrifugal fan design for a maximum 5MW class high voltage generator. I calculated the flow resistance of the generator to select the optimal cooling fan and reviewed the design of four cooling fans based on the shape of the blades considering the manufacturability, cost and performance. We performed a numerical analysis and an ANSI/AMCA standard 210-99 specification fan performance test to verify the performance of the fans. We used the ANSIS ICEM-CFD grid generation program as shown in Figure 9 to analyze the fan performance characteristics through a numerical analysis, and generated about 6 million grids. We expected turbulent flow so we used the STANDARD $k-\varepsilon$ model. Figure 17 shows the comparison of the performance curves through the numerical analysis and the performance tests. As a result of the comparison, we could obtain a reliable analysis result. The air flow rate of the fan operating point of a 5 MW class high voltage generator is about 5.94 m^3/s and the static pressure is about 1087 Pa. Based on the calculated system resistance curves and the results of the fan performance test, Case 1 was selected as the optimal cooling fan, as shown in Figure 20, which was judged to satisfy the required air flow rate and pressure of the target generator. In addition, if the reliability of the performance test is improved by examining the grid density and the analysis method of the numerical analysis, not only the improvement can save much time and cost during the examination of the cooling performance for the additional fan selection, and but also contribute to improve the product performance.

BIBLIOGRAPHY

[1] Frank P. belier - *Fan Handbook: Selection, Application, and Design*, Mc Graw-Hill, **1998**

[2] ANSYS Inc., release16.1, ANSYS FLUENT User’s Guide, **2015**

[3] ANSI/AMCA 210-99 *Laboratory Methods of Testing Fans for Aerodynamic performance Rating. An American National Standard*, **1999**