Experimental Study of Fluid Motion in Air Gap Using Scaled Water Model*

(Winding Efficiency of Rotor Surface Shape of Air-Cooled Generators)

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In this study, we investigated the value of the loss coefficient, ζ_{23} , at the exit of the rotor ventilation ducts when air coolant flows from the rotor ventilation ducts to the rotor-stator air gap by measuring the distribution of static pressure on the surface of the rotor and the stator and on the wall of the cooling duct of air cooled generators, by using a scale water model. Protuberant type rotor wedge was proposed and they performed much better than the flat type of rotor wedge for ventilation cooling. From the experimental results, for the protuberant type rotor wedge, the fluid can flow from the exit of the rotor cooling duct into the air gap with a small pressure loss, since the fluid near the rotor surface is separated from the rotor surface by the triangular protuberance and the static pressure above the exit of the cooling duct is less than that in the rotor cooling duct. The experimental results of the pressure loss and the results of numerical analysis which will be reported in another paper⁽¹⁾, were in good agreement.

Key Words: Swirling Flow, Pressure Distribution, Turbulent Flow, Combining tees, Pressure Loss, Ventilation Cooling, Turbine Generator

1. Introduction

In recent years, the unit rating of air cooled generators has increased to reduce the cost of manufacture. An increase in the rotor winding loss occurs which means that the task of cooling the winding to maintain an acceptable winding temperature, and to make the temperature distribution symmetrical has become more critical. To attain these ends, a high performance rotor cooling system, and highly accurate methods to estimate the cooling requirements, based on flow field and heat transfer knowledge of air cooled generators are required. The pressure loss of coolant air at the exit of the rotor cooling ducts have a great influence on the capacity of ventilation cooling of rotors. The loss coefficient at the exit of the cooling duct is strongly affected by the velocity of the main flow in the air gap. The flow in the air gap is complex combining tees and flow through an annulus

with an inner rotating cylinder.

In previous studies, the loss coefficients for combining and diverging tees have been studied. Several theoretical and experimental equations which are good agreement with the experimental results have been reported^{(2),(3)}. The resistance of flow through an annulus with an inner rotating cylinder has also studied⁽⁴⁾. However the interaction of these phenomena is still unclear.

In this study, we investigated the value of loss coefficient, ζ_{23} , at the exit of the rotor cooling ducts by using a scale water model. The loss coefficient at the exit of the cooling duct is strongly affected by not only the velocity of main flow in the air gap, but also the shape of the rotor wedge surface. So we experimented in changing the shape of the rotor wedge surface. At the end of this paper, the experimental results of the pressure loss are compared with the results of numerical analysis which will be reported in another paper.

2. Experimental Apparatus and Conditions

2.1 Experimental apparatus

The schematic arrangement of the apparatus

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Fig. 1 The schematic arrangement of the apparatus

which consisted of a water tank, a rotation control unit, and water circulation units is shown in Fig. 1. Water was used as the test medium. Since water has different properties from air, Reynolds number could be matched using a relatively low rotor rotational speed and low flow rates⁽⁵⁾. This greatly simplified the testing. In a water tank a 1/3 scale model which consisted of a rotor, a stator, ventilation ducts, and an air gap, was set up vertically. The rotor shaft was connected to the drive motor by a rubber belt. The rotational speed of the rotor could be controlled from 0 rpm up to 360 rpm by an inverter controller. Water circulation units were located near the water tank. The unit was composed of a pump, a valve, and a flow meter. Each flow in the air gap and the rotor ventilation ducts could be controlled individually. As a result of these equipment, the flow field in the scale model matched the flow conditions in an actual air cooled generator. To visually observe the flow in the air gap, four windows were located on the side wall of the water tank.

The details of the test section are shown in Fig. 2. The stator and the rotor were made of acrylic resin making them transparent. The stator was divided into two section horizontally, the axial flow in the air gap was supplied from the upper part of the stator. The top of the air gap was closed, so the flow from the upper part of stator passed down through the air gap. The rotor had 36 slots with 1 ventilation duct per slot. Considering the existence of the complicated three dimensional flows which occur in rotating machinery, it is difficult to clarify the fluid motions in rotor/stator air gaps. In this paper we paid great attention to understanding the fluid motions around the exit of the rotor ventilation ducts. To simplify the flow in the air gap, the rotor cooling ducts on the model, were given

Fig. 2 The detail of test section

10 degree pitch around the rotor surface, which is different from an actual rotor construction. Rotor wedges were mounted at the exit of rotor ducts, so that the shape of duct exits could be changed by using different types of wedges.

The static pressure taps which were located on the stator surface, the rotor surface, and the rotor ventilation duct are shown in Fig. 2. The points 3s and 3r were located on the stator/rotor wall downstream of the axial flow $23d$ from the rotor duct exit, d is the diameter of rotor ventilation duct exit. Point 2 was located on the rotor duct wall $2.8d$ upstream from the exit. The static pressure of the rotor ventilation duct wall and on the surface were measured using differential pressure transmitters (Validyne Engineering Corp., PD 55 D 1 N 530 S 4 A, PD 55 D 1 N 534 S 4 A) which were mounted on the rotor. The power

supply and output from the differential pressure transmitters were transmitted from the rotor to the recorder using a slip ring. The static pressure was measured on the stator wall using strain gage transducers (Kyowa, PGM-02 KG). These outputs were recorded in the recorder (Yokogawa, Texela LTES-30), the sampling rate was 1 Hz.

2.2 New rotor exit shape

Since the high rotational speed of the rotor surface, the tangential velocity was fully developed into turbulence flow in the air gap. The flow resistance of the exit of rotor ventilation duct is strongly affected by the turbulent boundary-layer. Therefore to reduce the pressure loss occurs in that area, where the swirl main flows and the ejected jets of coolant air interact with each other, we propose the new rotor surfaces shapes shown in Fig. 3. In the figure, case 1 represents the simple flat surface which is common to various rotating machinery. In case 2 prism protuberances are located upstream of the coolant air exit and the ventilating duct is expanded near the exit in order to decelerate the speed of coolant air. The sectional shapes of the flat type of rotor wedge, and the protuberant type of rotor wedge are also shown in Fig. 3.

2.3 Experimental conditions

In this study, the pressure losses were measured under these flow conditions.

$$Re = 5.6 \times 10^3, 6.2 \times 10^3, Re = \frac{De V_3}{\nu}$$

$$R\omega = 0, 5.1 \times 10^5, 1.0 \times 10^6, R\omega = \frac{\omega D^2}{\nu}$$

$$2.7 \times 10^3 < Re d < 2.5 \times 10^4, Re d = \frac{d V_2}{\nu}$$

$$Q_2/Q_3 = 0.0476, 0.0526, 0.158, 0.263, 0.368, 0.474$$

where Re is the axial flow Reynolds number after combining in the air gap. $R\omega$ is the rotating Reynolds

number, $Re d$ is the rotor ventilation duct Reynolds number, De is the equivalent diameter, $2s$, s is the width of the air gap, D is the rotor diameter, d is the rotor ventilation duct diameter, V_3 is the axial mean velocity after combining in the air gap, V_2 is the mean velocity in rotor ventilation duct, ω is the speed of rotation, ν is the kinematic viscosity, Q_2 is the flow rate of rotor ventilation duct, and Q_3 is the axial flow rate after combining in the air gap.

3. Numerical Method

In order to understand the fluid motions in the air gap and predict the pressure loss through it, we utilized computerized fluid dynamics (CFD) based on the Finite Volume Method⁽⁶⁾ combined with the SIMPLE algorithm⁽⁷⁾, by using STAR-CD. The code treats the mass and momentum conservation equations for general incompressible fluid flows concerning the rotational effect of a rotor, which is added to the momentum conservation equations as a source term. In addition, the RNG $k-\epsilon$ model^{(8),(9)} was considered in evaluating turbulent fluid motions. Figure 4 shows the computational domain. To reduce the computer power required, this calculated model is based on a rotational symmetry whose section angle is 10 degrees. It consists of a rotor surface, a stator surface, an air gap, and a ventilating duct. Considering the existence of complicated three dimensional flows, which occur in general rotating machinery, it is difficult to make the fluid motions in the rotor/stator air gaps clear. Thus we paid much attention to giving the appropriate geometry shape and boundary conditions to the calculated models shown in Fig. 4. In order to consider the interactions of the swirl flow in the air gap and the ejected coolant air jet, the computational domain is expanded $23d$ downstream from the ventilation duct exit hole, d is the diameter of rotor ventilation duct exit. As a result, the model contains approximately 100,000 computational cells in order to describe the precise shapes of the rotor and stator surfaces, and to obtain high resolution images of the flow field.

Fig. 3 Exit of rotor ventilation duct
Case 1: the flat type of rotor wedge
Case 2: the protuberant type of rotor wedge

Fig. 4 Calculation model and mesh structure

4. Result and Discussion

To evaluate the pressure losses at the exit of the rotor ventilation duct from the experimental results, we treated the measured data as follows. To cancel the effect of the Euler head, we modified the measured data with initial data which measured when the rotor was rotating with all water circulation units stopped. We use the treated data as static pressure, P_s at each measurement points. The total pressure P_t is calculated from P_s and the mean velocity at the measurement section, V , as follows,

$$P_{t2} = P_{s2} + \frac{\gamma V_2^2}{2g} \quad (1)$$

$$P_{t3} = \frac{P_{s3s} + P_{s3r}}{2} + \frac{\gamma V_3^2}{2g} \quad (2)$$

where, subscripts 2, 3s, and 3r are measurement

Coefficients for flow from rotor ventilation ducts.

$R\omega = 0$, $Re = 5.6 \times 10^3$

1. theoretical equation developed from the momentum theorem.
2. experimental equation form reference 3

Fig. 5 Loss coefficient when the rotor was stopped

Coefficients for flow from rotor ventilation ducts.

$Q2/Q3 = 0.473$, $Re = 5.6 \times 10^3$

(a) $Q2/Q3 = 0.473$

Coefficients for flow from rotor ventilation ducts.

$Q2/Q3 = 0.0526$, $Re = 5.6 \times 10^3$

(b) $Q2/Q3 = 0.0526$

Fig. 6 Rotational effect on loss coefficients. $\zeta_{23(F1)}$ is the value of flat type wedge at $R\omega = 5.1 \times 10^5$

points, the mean velocities V_2 and V_3 are calculated from mass flow rate divided by sectional area at each measurement point.

The pressure loss of coolant jet from the rotor ventilation duct to the air gap is defined as:

$$\Delta P_{23} = (P_{t2} - \Delta P_{\text{Friction}}) - P_{t3} = \zeta_{23} \frac{\gamma V_3^2}{2g} \quad (3)$$

where $\Delta P_{\text{Friction}}$ is friction loss in the rotor cooling duct, since the Reynolds number in the air gap is assumed to be $2.7 \times 10^3 < Re < 2.5 \times 10^4$, treating the diameter of the ventilating duct as the criterion of the length scale. Here $\Delta P_{\text{Friction}}$ is calculated using Blasius's fluid friction coefficient,

$$\Delta P_{\text{Friction}} = 0.3164 Re^{-1/4} \frac{l_{12}}{d} \frac{\gamma V_2^2}{2g} \quad (4)$$

where d is the diameter of the rotor ventilation duct, and l_{12} is the distance between measurement point 2 and the exit of the ventilating duct.

When the rotor was stopped, the loss coefficients plotted in Fig. 5 show the experimental results when the rotor is not rotating. Loss coefficients which are divided by m^2 , ζ_{23}/m^2 are plotted to the flow rate ratio $Q2/Q3$. Here m is sectional area ratio, $m = A_1/A_2$, A_1 is the sectional area of the air gap per slot, A_2 is the sectional area of the rotor ventilation duct, in this paper $m = 40.19$. To compare the experimental results with the previous studies^{(2),(3)}, two curves are plotted, curve 1 is theoretical equation developed from the momentum theorem and curve 2 is experimental equation of the loss coefficients for combining tees with circular cross section.

As shown in Fig. 5, the experimental results were in very good agreement with curve 2. The coolant flow around the exit of rotor ventilation duct was considered very similar to the flow in combining tees, when the rotor was not rotating.

Fig. 7 Loss coefficients when the rotor was rotating, $\zeta_{23F(2)}$ is the value of flat type wedge at $Q_2/Q_3 = 0.0526$.

The dependence of the loss coefficients on the rotational speed were plotted to $R\omega$ at the flow rate ratios $Q_2/Q_3 = 0.473$, 0.0526 , in Fig. 6. Loss coefficients were normalized by $\zeta_{23F(1)}$. $\zeta_{23F(1)}$ is the value of flat type wedge at $R\omega = 5.1 \times 10^5$. On a flat rotor wedge, when $R\omega$ increases, loss coefficients become higher, on the other hand, on protuberant type rotor wedge, loss coefficients become lower, for any flow rate ratio.

From the experimental results, when $R\omega$ increases, the speed of the rotor surface increases so that the shear layer become stronger, so in the case of the flat rotor wedge, penetrating the strong shear layer requires more tremendous energy from the jet. It was estimated that for the protuberant rotor wedge, the fluid can flow from the exit of the rotor cooling duct into the air gap with only a small pressure loss, since the fluid near the rotor surface is separated from the rotor surface by the triangular protuberance and the static pressure above the exit of the cooling duct becomes less than that in the rotor cooling duct. It is a remarkable result that the ζ_{23} of the flow rate ratio $Q_2/Q_3 = 0.0518$ has a negative sign.

The experimental results are plotted of flow rate ratio Q_2/Q_3 in Fig. 7. In this case, experimental condition is equivalent to an actual air cooling generator. Loss coefficients were normalized by $\zeta_{23F(2)}$, $\zeta_{23F(2)}$ is the value of flat type wedge at $Q_2/Q_3 = 0.0526$. As shown in the figure, protuberant type has constant effect on reducing pressure loss through all flow rate ratio as compared with flat type wedge. Therefore we estimated the effect of the protuberant type rotor wedge on an actual cooling system.

In the actual air cooled generator, when the radial vent cooling system is used for the rotor cooling, and

Fig. 8 Comparison with numerical analysis results. $\zeta_{23F(e)}$ is the experimental result of flat type wedge.

the stator coolant flows radially outwards through ducts in the stator core from the air gap, the flow rate Q_2/Q_3 increases from the end to the center of the air gap. The value of Q_2/Q_3 varies from 0 to 0.5 linearly along the air gap from the end to the center. To estimate the effect of protuberant rotor wedge in the actual cooling system, we define an integral average of loss coefficients, $IA(\zeta_{23})$, as follows:

$$IA(\zeta_{23}) = \frac{\int_0^{0.5} \zeta_{23} d\left(\frac{Q_2}{Q_3}\right)}{0.5} \quad (5)$$

The difference in the integral average of the loss coefficients between the protuberant type and the flat type means that the effect of protuberant type on the ventilation cooling system, $E(P)$. It calculated as follows;

$$E(P) = \frac{IA(\zeta_{23P}) - IA(\zeta_{23F})}{IA(\zeta_{23F})} \times 100 \quad (\%) \quad (6)$$

where subscript F is the flat type, and P is the protuberant type. The experimental result $E(P) = -13.2$ (%) is an effective value for the entire pressure loss of the ventilation cooling system.

Finally we compared the coefficient ζ_{23} which was given experimentally, to the given numerical coefficient in order to validate the numerical approach. Taking into account the measurement error, estimated at 1.5 in normal deviation, the numerical results show very good agreement with the experimental data plotted in Fig. 8. Loss coefficients were normalized by $\zeta_{23F(e)}$, where $\zeta_{23F(e)}$ is the value of flat type.

5. Conclusion

The main objective of this research was to clarify the loss coefficient at the exit of rotor ventilation duct quantitatively. A 1/3 scale water model was utilized

here as the experimental method. It worked very well, with the experimental results in very good agreement with the previous study on the loss coefficients for combining tees with circular cross section when rotor was stopped. We also obtained the dependencies of the loss coefficients on the rotational speed.

The second objective of this research was to assess the effects of rotor surface shapes of turbine generators on the ventilating systems. The new rotor surface shape which includes a protuberance and an expansion duct is highly effective for ventilation.

Finally the experimental results were compared with the numerical analysis and were in good agreement.

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