THERMAL PERFORMANCE OF WIND TURBINE POWER SYSTEM'S ENGINE ROOM

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Greatly expanded use of wind energy has been proposed to reduce dependence on fossil and nuclear fuels for electricity generation. For wind turbine power generation, as a mature technology in the field of wind power utilization, its large-scale deployment is limited by the cooling technology. Therefore, the temperature distribution of the wind turbine power generation is a key issue for the design of the cooling system. It is because the characteristics of cooling system have a great effect on the performance of the wind turbine power generation. Based on some assumptions and simplifications, a thermal model is developed to describe the heat transfer behavior of wind turbine power system. The numerical calculation method is adopted to solve the governing equation. The heat generation and heat flux are investigated with a given operating boundary. The achieved results can be used to verify whether the cooling system meets the design requirements. Meanwhile, they also can reveal that among the influencing factors, the meteorological conditions, generated output and operation state as well seriously influence its thermal performance. Numerical calculation of the cooling system enables better understanding and results in performance improvement of the system.

Keywords: Wind turbine; heat transfer; numerical calculation.

1. Introduction

Energy saving and CO₂ emission reduction are among the most important issues attracting international attention. An increasing awareness of these issues has prompted the application of the renewable energy (RE) sources especially wind energy for electricity generation. The cooling system of wind turbine power generation plays an important role in ensuring the wind turbine power system operating reliability and stability¹⁻³.

The cooling system supplies cold energy for the whole wind turbine power system, responsible for carrying away the heat produced by the wind turbine power system⁴. Meanwhile, different components with the different boundary condition have heat transfer with the air of cabin, which results in the high temperature of cabin air. Inhomogeneous temperature and unreasonable design of the cooling system would make the air of cabin exceed the designed value. More serious, some unpredictable

consequences may occur since the some components in cabin are temperature-sensitive assembly such as electrical equipment⁵⁻⁷. In view of these facts, the thermal performances inside the cabin should be measured at the initial stages of the design. Hence, this work would have significant impacts on the comprehensive performance of wind turbine power generation besides proper system designs.

The investigation in this paper focuses on the thermal performance of a liquid cooling system of wind turbine power generation by connected simulation of convection, conduction and radiation. Over the past years, many researchers and experts in the wind turbine field have devoted considerable efforts to study the operation of liquid cooling system. Yuan *et al.*⁸ studied different cooling systems of the wind turbine wind generation. They found that the high-power wind turbine power generation has a preference for liquid cooling system. Similar work was done by Ni⁹, who studied the liquid cooling system of the generator cooling and gave some introduction of the basic control strategies. In these studies, the heat transfer between the key components and the air inside cabin and solar radiation on heat transfer was not considered. But for the realistic performance estimation, the thermal contact between the surface of the key components and the air of the cabin, the temperature distribution caused by solar radiation is of paramount importance to conduct the cooling system and optimize the thermal performance.

The objective of present work is to numerically analyze the thermal performances of the wind turbine power generation. In this paper the numerical calculation method is used to investigate the characteristics of heat load of the cabin. The factors such as geometric configuration, physical parameters, the meteorological conditions and operation state that impact thermal performance of the wind turbine power generation would be considered.

2. Mathematical Model

Fig. 1. Schematic diagram of the wind turbine cabin.

The core components of the wind turbine power generation are concentrated layout. The sketch of the wind turbine cabin is shown in Fig. 1¹⁰. The wind turbine carbine consists of gearbox, generator, inverter, transformer, control cabinet, and ejector mainly can be seen in Fig. 1. They are very important heat source in the inner cabin, while the solar radiation, the heat load of the fresh air used to maintain the positive pressure of cabin and the

convective heat transfer between the cabin and external environment contributes to the external heat source of the cabin.

There are several typical heat transfer behaviors proceeding in the wind turbine cabin, such as solid conduction, gas conduction, and gas natural convection in the inner cabin and radiation heat transfer between the surface of the core components and internal surface of the cabin. The heat transfer mechanism of the wind turbine cabin is shown in Fig. 2. In order to simplify the mathematical model, the following assumptions are considered¹¹⁻¹⁴:

(1) there are many components in the cabin of wind turbine and each component's surface is isothermal; (2) the heat transfer process between the outer wall of the cabin and the external environment can be assumed to be flat flow; (3) the specific heat and thermal conductivity of air and different components are constant.

Fig. 2. The heat transfer mechanism of the wind turbine.

2.1. Governing equations

Based on the above assumptions, a model of thermal analysis is developed to describe the physical phenomena. Fig. 3 gives the physical model involved in the operation of the solar radiation, the inner heat source, the fresh air load and the cooling coils load. Taking the cabin air temperature as variable, the thermal equation of the system shown in Fig. 3 is given by

$$mc_p \frac{dT}{dt} = Q_r + Q_j + Q_e + Q_x, \quad (1)$$

where Q_r is the inner heat source load of cabin, it consists of the gearbox heat dissipating Q_{r1} , the generator heat dissipating Q_{r2} , the inverter heat dissipating Q_{r3} , the transformer heat dissipating Q_{r4} , the pump of the cooling system heat dissipating Q_{r5} , the fan of the ejector heat dissipating Q_{r6} , the fan of the positive pressure system heat dissipating Q_{r7} , the control cabinet heat dissipating Q_{r8} and cooling cabinet heat dissipating Q_{r9} (see Fig. 3); Q_j is the thermal load emitted from the cooling coils; Q_e is the heat load absorbed from the external environment; Q_x is the thermal load from ventilation, respectively.

Fig. 3. Schematic diagram of the physical model.

The above loads can be given depending on Fourier's law and thermal radiation laws:

$$\begin{aligned}
 Q_r &= Q_{r1} + Q_{r2} + Q_{r3} + Q_{r4} + Q_{r5} + Q_{r6} + Q_{r7} + Q_{r8} + Q_{r9} \\
 Q_{r1} &= k_{in} A_1 (t_1 - t_c) \quad Q_{r2} = k_{in} A_2 (t_2 - t_c) \quad Q_{r3} = k_{in} A_3 (t_3 - t_c) \\
 Q_{r4} &= k_{in} A_4 (t_4 - t_c) \quad Q_{r5} = P_5 (1 - \eta_5) \quad Q_{r6} = P_6 (1 - \eta_6) \\
 Q_{r7} &= P_7 (1 - \eta_7) \quad Q_{r8} = Q_c \quad Q_{r9} = P_{b9} l_{19} + P_{b9} l_{29} c + P_{s9} \\
 Q_e &= K A_e (t_s - t_c) \quad Q_s = c_p G \rho_a (t_c - t_a) \quad Q_j = \dot{m} c_p (t_w - t_c)
 \end{aligned} \quad (2)$$

Substitute Eq. (2) into Eq. (1), then Eq. (3) can be acquired:

$$\begin{aligned}
 mc_p \frac{dT}{dt} &= Q_r + Q_j + Q_e + Q_x \\
 &= \sum_{i=1}^4 k_{in} A_i (t_i - t_c) + \sum_{i=5}^7 P_i (1 - \eta_i) + Q_c \\
 &\quad + P_{b9} l_{19} + P_{b9} l_{29} c + P_{s9} + \dot{m} c_p (t_w - t_c) + K A_e (t_s - t_c) + c_p G \rho_a (t_c - t_a)
 \end{aligned} \quad (3)$$

where t_1 , t_2 , t_3 , and t_4 are the surface temperatures of gearbox, generator, inverter, transformer, respectively; A_1 , A_2 , A_3 , and A_4 are the equivalent mean heat transfer coefficients inside cabin; t_c is the ambient temperature inside cabin; P_i is the power of pump and fan; η_i is the efficiency of pump and fan; Q_c is the thermal load of control cabinet; $\dot{m}$ is the air quantity used to cool the cabin; c_p is the specific heat capacity of air; K is the equivalent mean heat transfer coefficient between the cabin and outside environment; t_s is outside surface temperature of cabin; G is the air volume of fresh air; ρ_a is the density of air; t_a is the outside ambient temperatures.

Considering the heat transfer process reaching steady-state, the steady-state thermal balance equation is established as follows: $mc_p dT/dt = 0$, namely

$$Q_r + Q_j + Q_e + Q_x = 0. \quad (4)$$

When Eq. (4) is established, the wind turbine power generation can work stably and efficiently.

In this paper, the details on every item of thermal load stated in Eq. (4) will be discussed in following sections. Q_r , Q_e and Q_x will be analyzed numerically, which means that the total heat load of cabin $Q = Q_r + Q_e + Q_x$ will be discussed firstly. Then based on Eq. (4), Q_j can be obtained.

2.2. Equivalent mean thermal coefficient k_{in}

Heat transfer process between the core components and the air of cabin mainly includes convection and radiation. The quantity of radiation among the components' surface and the air is very small in the cabin when wind turbine is working. Therefore, the radiation is neglected in the calculation, which implies that the thermal coefficient k_{in} is the convection coefficient between the key components and air. The solution for convection coefficient inside cabin can be determined by the semi-empirical formula and numerical calculation. For the closed cabin space, the average velocity of air is about 1~2.5m/s, the whole heat transfer process is similar to the process conducted in the aircraft cabin or buildings. The method for solving convection coefficient, taking into account the small volume of the cabin, can be determined with reference to the equation which is used in aircraft cabin. It can be written:

$$k_{in} = (8.23 + 4.28v_c) \sqrt{\rho_c / \rho_0}, \quad (5)$$

where v_c is the averaged velocity of air inside cabin, ρ_c is the density of cabin air, ρ_0 is the density of air at sea level. For the wind turbine power generation, $\rho_c \approx \rho_0$. Eq. (5) can be rewritten as:

$$k_{in} = 8.23 + 4.28v_c. \quad (6)$$

2.3. Equivalent mean heat transfer coefficient K

The heat transmission process between the cabin and the outside environments is a combination of several thermal behaviors, such as solid conduction, gas conduction, gas natural convection among the outside spacer and solar radiation. The quantity of gas natural conduction is very small in wind turbine working process in ambient environment, so that gas natural convection is neglected in the calculation. Therefore,

$$K = \frac{1}{1/h_{out} + e_1/\lambda_1 + e_2/\lambda_2}, \quad (7)$$

where h_{out} is the mean coefficient of convection heat transfer between the cabin skin and ambient environment; e_1 denotes the thickness of cabin skin; e_2 is the thickness of insulating layer; λ_1 and λ_2 represent the heat conductivity coefficients of cabin skin and insulating layer, respectively.

In above equation, the convection heat coefficient h_{out} is an important parameter to estimate the thermal performance of wind turbine cabin. In the present study, since the air velocity around cabin is relatively high, and according to the characteristics of the cabin structure, the form of correlation used for the convection heat transfer coefficient of the cabin skin and ambient environment is developed by Chilton-Colburn analogy:

$$Nu = 0.332 Re^{0.5} Pr^{0.33}, \quad (8)$$

where Nu , Re , and Pr are Nusselt number, Reynolds number and Prandtl number of process, respectively. Finally, the heat transfer coefficient is determined from:

$$h_{out} = Nu \frac{\lambda}{x}, \quad (9)$$

where λ is the heat conductivity coefficient of air at the design condition, x is the characteristic length of cabin.

2.4. Skin temperature of cabin t_s

According to the heat transfer theory, the skin temperature of cabin t_s , which is mainly influenced by the solar radiation, is a very important parameter in the heat process conducted from the cabin skin to the inside cabin. The solar radiation contributes to the increase of temperature of cabin skin. The governing equation can be expressed:

$$h_{out} (t_a - t_s) + q_s A_p / A = K (t_s - t_c) + 0.5 \epsilon_{os} \varphi_{os} \sigma (T_s^4 - T_{eff}^4) + 0.5 \epsilon_{og} \varphi_{og} \sigma (T_s^4 - T_g^4), \quad (10)$$

where $q_s A_p / A$ is the heat flux obtained from the solar radiation, A_p / A is ratio between the projected area and total area of fuselage, $A_p / A = 1/\pi$ is used in present study; $K (t_s - t_c)$ is the heat flux conducted from the cabin skin to the inner of cabin; σ is Stefan-Boltzmann's constant, $\sigma = 5.67 \times 10^{-8} W \cdot m^{-2} \cdot K^{-4}$; ϵ_{os} and φ_{os} are some key coefficients, $\epsilon_{os} = 0.3$ and $\varphi_{os} = 1$; T_{eff} is effective thermodynamic temperature of sky, $T_{eff} = 0K$; T_g is the averaged surface thermodynamic temperature of ground, $T_g = 284K$. The heat load contributed by solar radiation q_s is expressed as:

$$q_s = e_0 I P^m, \quad (11)$$

where e_0 is the absorption coefficient; I is the solar radiation intensity; P represents the atmospheric transparency, which is given to be 0.75; m is the atmosphere quality.

In the expressions (10) and (11), the parameters A_p / A , ϵ_{os} , φ_{os} , T_{eff} , and T_g are known. If t_a and t_c are given, K can be obtained from Eq. (7). t_s can be calculated by iterative method using these expressions. Then the heat flux conducted from the cabin skin to the inside cabin can also be calculated.

3. Results and Discussion

The thermal performance of wind turbine cabin is very important to the working behavior of wind turbine. The generated output, meteorological conditions, geometric and physical parameters influence the thermal performance. That is a key problem for the designer to be aware of such influencing factors. In this section, the influences of parameters on the thermal performance of wind turbine cabin are investigated.

3.1. Analysis of thermal coefficient

The convection heat transfer coefficient between the key components inside cabin and the air of cabin, k_{in} , is given in Fig. 4. k_{in} is related to the velocity of the inner cabin air. It can be seen that the convection heat transfer coefficient increases with the ambient velocity at the same temperature.

Fig. 4. Variation of inside heat transfer coefficient with the inside air velocity.

Fig. 5 presents the variation of h_{out} with the ambient velocity. We can obtain that the k_{in} is related to the velocity of ambient cabin air, and the convection heat transfer coefficient increases with the ambient velocity at the same temperature.

Fig. 5. Variation of outside heat transfer coefficient with the ambient air velocity.

3.2. Analysis of thermal performance

The geometry of wind turbine cabin is similar to cuboids. The wind turbine cabin, which is insulated by polyurethane, foams with its thickness of 15 mm. Cuboids capsule, made of high-strength aluminum, alloys with its thickness of 10 mm. The refrigerant is aqueous ethylene glycol solution of 60% mass concentration.

This section concerned with the thermal performance of the wind turbine cabin from the numerical point of view. Considering the components layout and boundary conditions, the thermal model is developed by the VC++ program. In addition, the meteorological condition and the material properties are taken into account in the calculation. The results are shown as follows.

Fig. 6. Variation of the total heat loads with ambient air velocity.

Fig. 6 shows the variation of total heat load of the cabin with the air velocity of ambient environment. It is obtained from the figure that the total heat load of the cabin Q decreases nonlinearly with increasing of the air velocity of ambient environment. And it increases with the ambient temperature if the air velocity of ambient environment v is given. And it is observed that the total heat load decreases with increasing of the air velocity of ambient environment. The results show that the influence of thermal coefficient between the cabin and ambient environment grows larger due to the larger air velocity of ambient environment, which will enhance the heat transfer between the cabin and ambient environment.

Fig. 7 gives the total heat load variation of the cabin with the air velocity of internal cabin air. When the air temperature of ambient environment is 35°C , the air velocity of ambient environment is 10m/s , and the solar radiation intensity is 1353W/m^2 . When the temperature of inside cabin air is 52°C , the total heat load increases from 10039.8 W to 16402.5W while the inside cabin velocity increases from 1m/s to 2.9m/s . And it can be seen from Fig. 7 that the influence of air velocity of inside cabin on the total heat load is large owing to the convection heat transfer coefficient dependent on the air velocity of inside air. It can be also obtained from the Fig. 7 that the total heat load Q decreases nonlinearly with the control temperature of the inside cabin t_c . The reason is that the

higher control temperature indicates the smaller temperature difference between the inside cabin air and the surface of the key components, the smaller temperature difference means the weaker heat transfer driving force, which results in the decreasing of the total heat load.

Fig. 7. Variation of the total heat loads with inside cabin air velocity.

Fig. 8. Variation of the total heat loads with the temperature of key components' surface.

Fig. 8 shows the heat load variation with the temperature of core components' surface. We assume that the ambient environment parameters are constant. It is observed that the heat load linearly increases with the temperature of core components' surface. The heat transfer driving force is larger when the temperature of the core components' surface is larger, which lead to the increase of the inner heat load produced by the core components. It is obvious from Fig. 8 that the variation rate of heat load caused by the inverter is the largest of the three kinds of components in the same scale of temperature variation, which is caused by the largest surface area of the inverter.

Fig. 9. Variation of the total heat loads with the ambient temperature.

Fig. 9 shows the variation of heat load with ambient environment in different solar radiation absorption rate. From Fig. 9, it is known that at a certain ambient temperature, the heat load increases with the absorption rate of solar radiation. Since the cabin surface temperature increases with the absorption rate of solar radiation, the temperature difference increases with the cabin surface temperature, which results in more heat flux transported from the ambient environment.

Fig. 10 presents the variation of the heat load with time in one day. It is observed from figure that the heat load of the cabin varies obviously with time. Heat load decreases firstly, then increases, and decreases again; the variation trend is the same as the ambient temperature variation with the time. The maximum heat load appears at 14 o'clock while the minimum heat load occurs at about 4 o'clock. The result is determined by the different solar radiation of each hour and the property of the insulation layer.

Fig. 10. Variation of the total heat loads with time.

4. Conclusions

In this paper, the thermal performance of a 3MW wind turbine cabin is numerically conducted. The influences of geometrical configuration and physical parameters, the

meteorological conditions and operation state on thermal behaviors are investigated. The obtained results show that all these factors have a great effect on the heat load of the cabin. As a consequence, to design the cooling system of the wind turbine power generation, not only the heat production of each components should be considered, but also the surface temperature of the core components, the meteorological conditions and the operate state are important parameters. The present work is helpful for the optimization of cooling system in wind turbine.

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