Applied Thermal Engineering 129 (2018) 1002–1009

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

Research Paper

Design analysis and improvement of cooler in positive-pressure explosion-proof low-speed high-capacity induction motors

Yu Xiwei *, Meng Dawei

Harbin University of Science and Technology, Department of Electrical and Electronic Engineering, Xuefu Road 52, 150080 Harbin, China

HIGHLIGHTS

- Paper studies a low-speed induction motor with the largest capacity in the world.
- Cooler with a special structure was designed according to the product features.
- The fluid and temperature fields of the cooler were simulated and analysed.
- We improve the cooling performance by optimizing the structure of the motor cooler.

ARTICLE INFO

Article history:
Received 16 August 2017
Revised 19 September 2017
Accepted 19 October 2017
Available online 20 October 2017

Keywords:
Positive-pressure explosion-proof motor
Low-speed high-capacity induction motor
Cooler
Fluid field
Temperature field

ABSTRACT

In this paper, we study a low-speed induction motor with the largest capacity in the world. This is a positive-pressure explosion-proof motor designed to be operated in harsh environments. First, a cooler with a special structure was designed according to the product features. Then, the fluid and temperature fields of the cooler were simulated and analysed using a global coupling method. The prototype was then subjected to a temperature rise test, and the measured temperature and speed of the flow were found to be in good agreement with the calculation results. This indicated that the design of the special structure of the cooler was reasonable and that the solution method and modelling proposed in the paper were appropriate. Finally, the internal construction of the cooler was further improved to serve as a theoretical basis for future optimisation designs.

0. Introduction

With the development of the power industry, the capacity of motors has continued to increase [1]. On the basis of the similarity principle of motors, the cooling of the motor is one of the main obstacles in the increase of motor capacity. Therefore, the design of the cooler is very important.

Three basic modes of cooling for motors are currently in use: air–air cooling, air–water cooling and open style cooling. In air–air cooling, the rotation of the motor drives the fan that dissipates heat [2–4]. However, the prototype motor studied here has a low rotational speed; therefore, it is not suitable for this mode of cooling. In the air–water cooling mode, heat is dissipated by an externally installed water circulation system [5–7]; therefore, the capacity of the motor may be increased. However, the prototype motor studied here is deployed in regions that lack water resources. Thus, the air–water cooling mode cannot be applied. In the open style cooling mode, the air in the interior of the motor is connected to the air external to the motor so that heat may be easily transferred to the exterior. However, the prototype is a positive-pressure explosion-proof motor; therefore, it must be of the closed type. For these reasons, this study designed a special cooler with forced air in both the internal and external air flow to satisfy the heat dissipation requirement of the motor based on the actual characteristics of the product and the environment in which the motor is deployed.

In fluid field and temperature field analysis of motors home and abroad is focused mainly on the internal structure of the motor [8–19] and rarely on the ventilation and heat dissipation of the motor cooler. For example, a numerical analysis was conducted for the external air flow of a medium-sized asynchronous motor in Ref. [20], and a thermal impedance calculation was performed for an air–air cooler in Ref. [21]. However, neither had involved the study of forced ventilation of the cooler. This type of cooler has an internal fan and the cooling zone contains a highly complicated three-dimensional unsteady viscous flow that posed difficulty in the

* Corresponding author.
E-mail addresses: 88580882@163.com (X. Yu), mengdawei2017@163.com (D. Meng).

https://doi.org/10.1016/j.applthermaleng.2017.10.101
1359-4311/

accurate fluid field analysis. In this study, the basic assumptions and boundary conditions are based on the actual engineering situation, and the numerical simulation of the fluid field and temperature field of the cooler employs a global coupling method based on computational fluid mechanics principles and heat transfer theories. The simulation results are compared with the experimental test results. The comparison has validated the calculated results and provided a theoretical basis for future analysis of this type of coolers.

1. Physical model of the prototype motor cooler

Table 1 lists the basic parameters of the motor.

The Explosion-proof class of the motor is divided into Positive-pressure "p", Flameproof enclosures "d", Increased safety "e" and so on. Positive-pressure "p" and Flameproof enclosures "d" are usually used in Zone 2, The Flameproof enclosures "d" motor is realized through the heavy casing and the small flameproof gap, so it can't meet the explosion-proof requirements of the larger volume of motor, so the final choice the Positive-pressure "p" motors, such as "ExpxdeIIT3 Gb", in which, the "Ex" indicates Explosion-proof, the "px" indicates that the main body of motor is Positive-pressure type, The "d" indicates that the forced ventilation motor is Flameproof enclosures, the "e" indicates that the temperature measuring terminal box is Increased safety type, the "II" indicates Zone 2, the "T3" indicates the temperature class (the maximum surface temperature is less than 200 °C), the "Gb" indicates equipment protection level.

The protection grade of the Positive-pressure "p" motor usually should be more than IP4X, in this paper, the motor adopts the higher grade IP55 (The dust grade 5, namely: completely prevent intrusion of foreign objects, although not completely prevent the intrusion of dust, but the amount of invading dust will not affect the normal operation of the motor; The waterproof grade 5, namely: which can prevent from all directions emitted by the nozzle of water into the motor causing damage.

Based on the basic data and operating conditions of the motor, a special cooler with forced ventilation for both the internal and external air flow was designed. At the top of the cooler, two forced blowers were installed for the internal air flow. Two additional forced blowers were installed on the side of the cooler for the external air flow. Fig. 1 shows the prototype and the physical cooler.

The ventilation structure of the cooler is shown in Fig. 2, where the dashed line shows the air flow direction of the internal flow. Under the action of two centrifugal fans on top of the cooler, the hot air in the motor enters the cooler from the lower central portion of the cooler and then returns to the motor after heat exchange from the two side outlets at the lower portion of the cooler. The solid line in Fig. 2 shows the flow direction of the external air flow of the cooler. Under the action of two axial flow fans at the tail of the cooler, the external air enters the cooling tube of the cooler and after heat exchange, returns to the ambient air. Fig. 3 shows the solution model based on the actual ventilation structure.

Table 1
The basic parameters of the prototype.

Motor parameters	Rating
Model	YZYKK6500-20
Power (MW)	6.5
Voltage (kV)	11
Speed (r/min)	297
Frequency (Hz)	50
Efficiency	96%
Protection class	IP55
Explosion-proof class	ExpxdeIIT3 Gb
Installation site	Zone 2 (IIC)

2. Global coupling simulation analysis

2.1. Mesh division

The model is divided using a structural mesh. Owing to the thinness of the cooling tube wall and the fan blades, the mesh size must be small for better mesh quality. A total of 55.92 million elements and 13.40 million nodes were installed. Fig. 4 shows the split of the local mesh.

2.2. Model analysis

2.2.1. Mathematical description

(1) Fluid field

When solving the physical model of the fluid flow in the cooler, the fluid flow must be controlled by the mass conservation equation. In addition, owing to the viscosity of the fluid, the fluid flow must satisfy a momentum conservation equation. Owing to the presence of rotating fans in the cooler, the flow is turbulent and requires the use of the standard turbulent flow κ-ε model [22,23].

The mass conservation equation is given here.

$$\frac{\partial\rho}{\partial t}+u\frac{\partial(\rho u)}{\partial x}+v\frac{\partial(\rho v)}{\partial y}+w\frac{\partial(\rho w)}{\partial z}=0 \tag{1}$$

The momentum conservation equation is given here.

$$\left.\begin{aligned}&\frac{\partial(\rho u)}{\partial t}+\frac{\partial(\rho uu)}{\partial x}+\frac{\partial(\rho uv)}{\partial y}+\frac{\partial(\rho uw)}{\partial z}=\\&-\frac{\partial p}{\partial x}+\frac{\partial\tau_{xx}}{\partial x}+\frac{\partial\tau_{yx}}{\partial y}+\frac{\partial\tau_{zx}}{\partial z}+F_x,\\&\frac{\partial(\rho v)}{\partial t}+\frac{\partial(\rho vu)}{\partial x}+\frac{\partial(\rho vv)}{\partial y}+\frac{\partial(\rho vw)}{\partial z}=\\&-\frac{\partial p}{\partial y}+\frac{\partial\tau_{xy}}{\partial x}+\frac{\partial\tau_{yy}}{\partial y}+\frac{\partial\tau_{zy}}{\partial z}+F_y,\\&\frac{\partial(\rho w)}{\partial t}+\frac{\partial(\rho wu)}{\partial x}+\frac{\partial(\rho wv)}{\partial y}+\frac{\partial(\rho ww)}{\partial z}=\\&-\frac{\partial p}{\partial x}+\frac{\partial\tau_{xz}}{\partial x}+\frac{\partial\tau_{yz}}{\partial y}+\frac{\partial\tau_{zz}}{\partial z}+F_z.\end{aligned}\right\} \tag{2}$$

Here p is the pressure on the micro-element of the fluid. τ_{xx}, τ_{xy} an τ_{xz} are the components of the viscosity stress τ along th x, y an z directions, respectively, F_x, F_y an F_z are the body forces on the micro-element.

The turbulent flow equation is given here.

$$\left.\begin{aligned}&\frac{\partial(\rho k)}{\partial t}+\frac{\partial(\rho k u_i)}{\partial x_i}=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+\\&G_k+G_b-\rho\varepsilon-Y_M+S_k,\\&\frac{\partial(\rho\varepsilon)}{\partial t}+\frac{\partial(\rho\varepsilon u_i)}{\partial x_i}=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial x_j}\right]+\\&\frac{\varepsilon}{k}(G_k+C_{3\varepsilon}G_b)-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}+S_\varepsilon.\end{aligned}\right\} \tag{3}$$

k is the kinetic energy of the turbulence, ε is the rate of dissipation of the turbulent kinetic energy, μ_i is the velocity in the i direction, μ is the kinetic viscosity, G_k is the generation term of the turbulent kinetic energy k caused by the average velocity gradient, G_b is the generation term of the turbulent kinetic energy k caused by buoyancy, Y_M is the contribution of pulsation expansion of the compressible turbulent flow and S_k and S_ε the self-defined source terms of k and ε, respectively. If the flow is incompressible and there are no user-defined source terms, then $G_b=0$, $Y_M=0$, $S_k=0$, $S_\varepsilon=0$. The turbulent viscosity μ_t may be expressed as a function of k and ε, namely, $\mu_t=\rho C_\mu\varepsilon^2/k$. According to the values recommended by Launder and Spalding and the subsequent experimental verification, $C_{1\varepsilon}=1.44$, $C_{2\varepsilon}=1.92$, $C_\mu=0.09$, $\sigma_k=1.0$, $\sigma_\varepsilon=1.3$.