REAL TIME NUMERICAL SIMULATION OF THERMAL CONDUCTIVITY OF MARINE GAS TURBINE LUBRICATING OIL UNDER COMPLEX SEA CONDITIONS

by

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Due to the influence of the marine environment, marine gas turbine lubricants are required to have good thermal conductivity. However, the current research on marine gas turbine lubricants mainly focuses on its corrosion resistance. Therefore, a real-time numerical simulation of thermal conductivity of marine gas turbine lubricating oil under complex sea conditions is proposed. In this paper, the turbulent kinetic energy equation and the loss transport equation are obtained by introducing the turbulent column and the k-ε model. The Nusselt number in the k-ε model can be regarded as the ratio of convective heat transfer between fluid and solid wall and internal heat transfer of fluid. According to the equations, the transient heat transfer model of vertical moving contact parts is established, and the heat transfer relationship between lubricating oil and plug group is obtained. The experimental results show that the area of low temperature in this scheme is larger than that in the traditional scheme, which means that the efficiency of impingement cooling is improved after the introduction of the pin structure in the impingement cooling design. The heat transfer process on the suction surface is stronger than that on the pressure surface, and the heat transfer coefficient of the blade increases continuously, reaching the local maximum value at about $X/L = 0.65$.

Key words*: marine gas turbine, lubricating oil, heat conduction, real time numerical simulation*

Introduction

Improving the thermal conductivity of gas turbine lubricants on ships is conducive to the further development of technology in the field of ship power. In the future, the power of naval ships in China will be mainly gas turbine [1]. The working process of gas turbine combustor has the characteristics of high temperature, high speed, high combustion heat intensity, high excess air coefficient, and drastic changes of operating parameters. These characteristics affect the stability, economy, and reliability of the combustion chamber. The internal flow and chemical reaction process of combustion chamber is very complex, including complex turbulent flow, fuel injection, two-phase flow, fuel atomization, combustion, heat transfer, *etc*.

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Yang *et al*. [2] analyzed the flame transfer function in modern gas turbines through principal component analysis and dynamic mode decomposition to overcome the limitations of FTF. Confirmed the relationship between the mode shape and FTF. In [3], SiC-reinforced HfB2 and ZrB2 ultra-high temperature ceramics are proposed as gas turbine stator blades. The heat transfer and stress-strain equations are numerically solved by the finite element method to obtain the temperature and stress distribution. The results show that the maximum thermal stress occurs near the cooling pipe with the largest temperature gradient. But currently, the measurement of the complex working process in the combustion chamber of heavy duty gas turbine is still subject to great limitations, and the test cost is very high and the test period is relatively long.

This paper presents a real-time numerical simulation of thermal conductivity of marine gas turbine lubricating oil under complex sea conditions. In this paper, the turbulent kinetic energy equation and the loss transport equation are obtained by introducing the turbulent column and the k-ε model. The Nusselt number can be regarded as the ratio of convective heat transfer between fluid and solid wall and internal heat transfer of fluid. Based on the theories of CFD, computational heat transfer, and computational combustion, the non-linear simultaneous conservation differential equations of mass, momentum, energy, and components are solved by numerical method. According to the equations, the transient heat transfer model of vertical moving contact parts is established, and the heat transfer relationship between lubricating oil and plug group is obtained.

Thermal conductivity simulation of marine gas turbine lubricating oil

Heat conduction of marine gas turbine lubricating oil

In the turbulent gas-liquid two-phase flow of combustion chamber combustion, mass, momentum, and energy transfer always exist in the air and oil droplet two-phase, and there is also the interaction of mass, momentum, and energy between the two phases [4]. In addition, the internal structure of combustion chamber is very complex. There are a lot of gas turbulent flow in the combustion chamber and transition section of gas turbine. Therefore, in order to study the gas flow state in the transition section, it is necessary to study the turbulent flow of gas.

The purpose of introducing the pin is to improve the cooling effect of high temperature wall by disturbing the cooling gas. From the point of view of the turbulence model, some scholars need to simulate the process. According to the different forms of Reynolds stress terms, the turbulence models can be divided into different types [5]. Among the commercial software (such as fluent), k-ε model has been widely used.

The k-ε model is used to solve the two variables of length and velocity. The equation is about turbulent kinetic energy, k, and turbulent energy dissipation rate, ε, and it is defined:

$$\varepsilon = \frac{u}{\rho}\overline{\left(\frac{\partial u_i^{'}}{\partial x_k}\right)\left(\frac{\partial u_i^{'}}{\partial x_k}\right)} \tag{1}$$

where u_i is the turbulent viscosity, x_k – the high frequency pulsation, ∂ – the boundary disturbance value, and ρ – the near-wall turbulent pulsating curl.

At that time, when the deformation with large average strain rate exists, the standard k-ε model is likely to have negative positive stress. In order to make the flow conform to the laws of turbulent physics, it is necessary to use mathematical methods to restrict the positive stress [6]. Based on this consideration, the realizable k-ε model is proposed. The turbulent kinetic energy equation and the loss transport equation are expressed:

$$\rho \frac{D\varepsilon}{Dt} = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\partial x_j}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + C_1 S \rho\varepsilon - C_2 \frac{\rho\varepsilon^2}{k + \sqrt{ve}} \tag{2}$$

where C_1 = max[0.43, $\eta/(\eta + 1)$], C_2 = 1.0, $D\varepsilon$ – the turbulence intensity in circumferential direction, Dt – the large curvature, x_j – the spatial grid length, μ_t – the time step, S – the flow scale, e – the transport variable, v – the gradient near the wall, and k – the uniform decay value. This model uses the wall function method to predict the heat transfer and flow field structure of the inner wall cooling surface.

The turbulent kinetic energy is determined by solving a semi empirical transport equation [7]. In a standard turbulence model, the length scale is assumed to be the length scale of kinetic energy dissipation:

$$q_w = \frac{k^3}{I_t} \tag{3}$$

where k is the model constant and I_t – the dissipation rate of turbulent kinetic energy, that is, the rate at which the mechanical energy of an isotropic small-scale vortex is converted into heat energy.

In the turbulent core region, there are vortices and turbulence. In addition to the heat conduction perpendicular to the flow direction of the fluid, heat exchange due to the strong mixing movement of fluid particles plays a major role, thus greatly enhancing the heat transfer process [8-10]. The viscous bottom layer near the solid wall is laminar flow, and the heat transfer mode is heat conduction. Because the thermal conductivity of the fluid is very small, the thermal resistance of the viscous bottom layer is the largest in the three-layer flow region. The dimensionless surface heat transfer coefficient, Nu, called Nusselt number, is often introduced in engineering:

$$\mathrm{Nu} = \frac{q_w L}{\lambda\left(T_w - T_0\right)} \tag{4}$$

where λ, T_0, and L are characteristic quantities. Nusselt number can be regarded as the ratio of convective heat transfer between fluid and solid wall and heat conduction in fluid.

Thermal conductivity simulation

In order to analyze the details of the flow, heat transfer and combustion process in the combustion chamber, the distribution of gas-liquid two-phase in the whole combustion flow field and the product concentration distribution of combustion reaction can be obtained, the theories of CFD, computational heat transfer and computational combustion can be used to solve the non-linear simultaneous conservation differential equations of mass, momentum, energy, and components.

For the actual viscous gas, the energy equation including heat transfer without considering potential energy is:

$$-\int_1^2 \frac{\mathrm{d}p}{\rho} = \int_1^2 V\mathrm{d}V + \omega + \varpi_f + q \tag{5}$$

where p is the pressure, ρ – the density, V – the speed, ω – the shaft work, ϖ_f – the friction loss, and q – the heat transfer, ignoring the influence of heat transfer on inlet and outlet velocity, $q < 0$ when the system is exothermic, so the expansion work of gas in turbine is reduced. In other words, the forced heat transfer between the blade and the blade surface will be reduced. The heat transfer reduces the load of the rotor blade.

The effect of heat transfer on the efficiency of wheel rim is analyzed from the First laws of thermodynamics and the Second laws of thermodynamics.

The First law of Thermodynamics:

$$h_1^* + q = h_2^* + \omega \tag{6}$$

The Second law of Thermodynamics:

$$\sum \Delta s = \sum \frac{\Delta q}{T} \tag{7}$$

The entropy change caused by wall heat conduction is only considered, and the entropy change caused by irreversible flow process is not considered. Then, considering the heat conduction, the baking drop under the ideal flow condition is:

$$\eta_u = \frac{L_u}{h_1^* - h_{2s'}} \tag{8}$$

where the superscript * is the stagnation parameter, the subscript s is the isoentropy condition, the subscript 1 is the parameter before the rotor blade, and the subscript 2 is the parameter after the rotor blade.

In practical engineering application, different mathematical models need to be established for different problems. In some problems, the coefficient is non-linear or even discontinuous. If the material is different or other objective conditions change, these factors will lead to the discontinuity of its parameters, such as heat conduction equation:

$$-\nabla k \nabla T_m + \frac{\rho c}{\Delta t} T_m = Q + \frac{\rho c}{\Delta t} T_{m-1} \tag{9}$$

where ρ, c, and k represent the density, specific heat, and thermal conductivity of the material, respectively. These parameters will change according to the actual situation, that is, the non-linear or discontinuous material parameters appear in the solution domain.

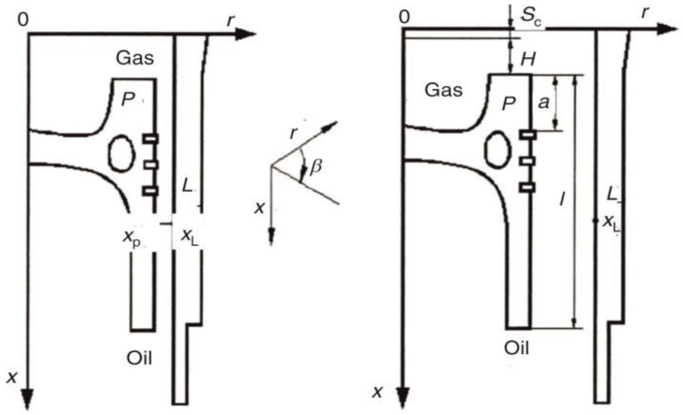

Figure 1. Schematic diagram of relative position between piston group and cylinder liner

For the 3-D heat transfer problem between the piston group and the gas driving sleeve, the cylindrical co-ordinate system is usually used to determine the heat transfer relationship As shown in fig. 1, x is the axial direction of the cylinder liner, downward is positive, r – the radial direction of the cylinder liner. The origin of the co-ordinate system is located in the center of the top plane of the contact between the cylinder gasket and the cylinder liner, P represents the piston, L – the cylinder liner, and g – the gas. In order to make the figure more intuitive, a cycle transient heat transfer model of dynamic contact parts was established by taking β piston ring in a certain direction and comparing with the 2-D view of β on the cylinder liner.

The parts above the first piston ring and the top surface of the piston are in direct contact with high temperature and high pressure gas, in this region, the heat transfer relationship between gas and piston group can be written:

$$q_{g-p} = h_g(\tau)\left[T_g(\tau) - T_p(\beta, x_p, \tau)\right] \tag{10}$$

where τ is the time and T_p – the temperature of the piston group on the moving boundary.

On the dynamic contact boundary of the piston group, the heat transfer coefficient, h_g, is directly determined by the thickness of the lubricating oil film between the piston group and the air siphon sleeve, and its distribution is $h_g(\beta, x_p, \tau)$. The thickness of the lubricating oil film at the piston ring group changes transiently with time, so the heat transfer coefficient here also changes with time. If the thickness of lubricating oil film outside the piston ring group is assumed to be constant, the heat transfer coefficient will not change with time at these positions. Assuming that the temperature distribution at a certain position on the dynamic contact boundary of the cylinder liner is $T_L(\beta, x_L, \tau)$, and the displacement of the piston is H at a certain moment, then the co-ordinates of any point x_p on the piston group will become $x_p + H$. If the ambient temperature of x_p points on the piston group is to be determined, the co-ordinates on the cylinder liner must be $x_L = x_p + H$. At this time, the ambient temperature corresponding to x_p points on the piston bank is the temperature $T_L(\beta, x_L, \tau)$ of x_L point on the cylinder liner.

After finding $h_g(\beta, x_p, \tau)$ and $T_L(\beta, x_L, \tau)$, the heat transfer relationship between cylinder liner and point x_p on the piston group can be written:

$$q_{L-P} = h_p(\beta, x_p, \tau)\left[T_L(\beta, x_L, \tau) - T_p(\beta, x_p, \tau)\right] \tag{11}$$

The inner surface of the piston crown and bottom has direct heat transfer with spray cooling oil mist. The piston skirt and the lower parts have direct heat transfer with lubricating oil. The heat transfer relationship between lubricating oil and piston group can be written:

$$q_{\mathrm{oil}\cdot P} = h_{\mathrm{oil}}(\tau)\left[T_{\mathrm{oil}}(\tau) - T_p(\beta, x_p, \tau)\right] \tag{12}$$

where h_{oil} is the heat transfer coefficient of spray cooling oil mist or lubricating oil and T_{oil} – the temperature of lubricating oil.

Experimental study

According to the characteristics of gas turbine transition section structure working in high temperature environment for a long time, the distribution law of temperature field is analyzed. Effective cooling methods are being studied. In order to perform simulation analysis, first of all, the corresponding simplified model must be established scientifically and reasonably.

Figure 2 shows the temperature contour map inside the blade. The lowest temperature occurs in 2 and 3 cooling holes, the number of cooling holes is less than the leading edge, and the flow rate of cold gas is small, so the temperature reaches the highest in this region. Figure 2(b) shows the blade pressure surface. The temperature of suction surface is higher than that of pressure surface, which indicates that the heat transfer process of suction surface is stronger than that of pressure surface.

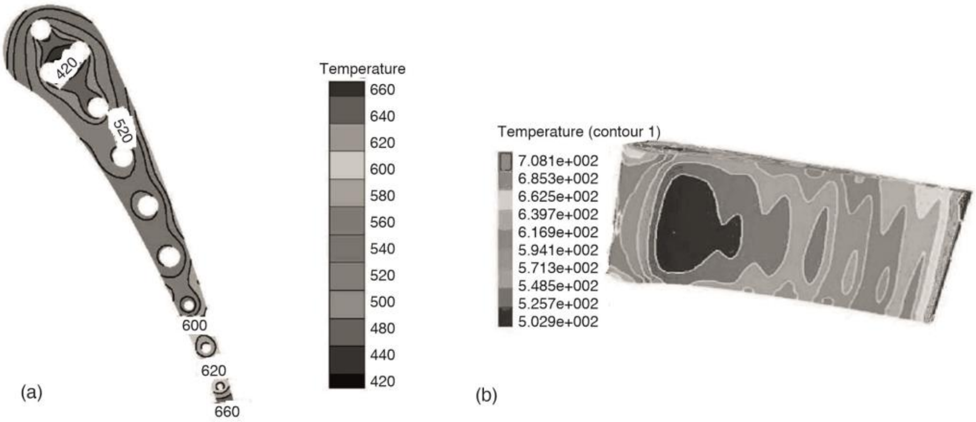

Figure 2. Temperature contour map inside blade; (a) temperature distribution in high section of 50% blade and (b) temperature distribution nephogram of blade pressure surface

The heat transfer coefficient of blade surface is affected by the main stream turbulence, Reynolds number and boundary-layer development. Laminar flow, transition state and turbulence often exist in the blade boundary-layer at the same time. Figure 3 shows the distribution of heat conduction coefficient on the profile at the middle section of the blade.

It can be seen from fig. 3 that at the pressure side of the blade, the heat transfer coefficient gradually decreases from the leading edge point. After a sudden change, it rises to the original value, then decreases to the lowest value along the original trend, and then gradually increases. On the suction side, the heat transfer coefficient keeps rising, reaching the local maximum value at about $X/L = 0.65$.

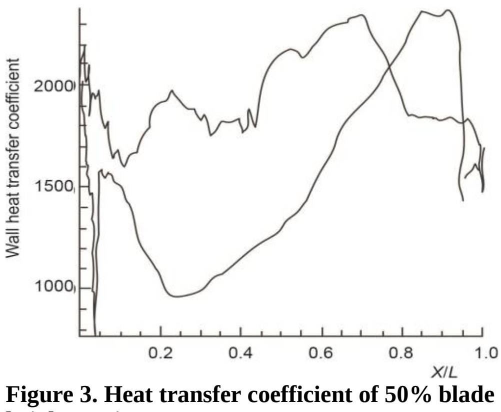

Figure 3. Heat transfer coefficient of 50% blade height section

Conclusion

In this paper, real-time numerical simulation of thermal conductivity of marine gas turbine lubricating oil under complex sea conditions is presented. The turbulent kinetic energy equation and the loss transport equation are obtained by introducing the turbulent column and combining with the k-ε model. The Nusselt number can be regarded as the ratio of the convective heat transfer between the fluid and the solid wall and the heat conduction in the fluid. Based on the theories of CFD, computational heat transfer and computational combustion, the non-linear simultaneous conservation differential equations of mass, momentum, energy, and components are solved by numerical method. According to the equations, the transient heat transfer model of vertical moving contact parts is established, and the heat transfer relationship between lubricating oil and plug group is obtained. The area of low temperature (yellow region in fig. 2) in the scheme is larger than that in the traditional scheme, which means that the efficiency of impingement cooling is improved after the introduction of the pin structure in the impingement cooling design. The heat transfer process of suction surface is stronger than that of pressure surface, and the heat transfer coefficient of blade is rising continuously, reaching the local maximum value at about $X/L = 0.65$.

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