Analysis of Influence of Structural Parameters of Regenerative Cooling Channel of Scramjet

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Abstract. In this paper, a three-dimensional modeling analysis of regenerative cooling channels of scramjet engines is carried out, and the cooling channel structure parameters such as inner wall thickness, fin thickness, channel aspect ratio, cross-sectional area and number of channels are used to cool the regenerative cooling. The conclusion is as follows: the thinner inner wall thickness corresponds to a lower wall temperature, while the heat flux density is larger, and the kerosene temperature and pressure change little; the thin fins correspond to the higher wall temperature and oil temperature, but the kerosene is in the cooling channel. The pressure loss is smaller, and the thicker fins have a lower maximum wall temperature and maximum oil temperature, but a larger pressure loss; the greater the aspect ratio, the better the cooling effect of the channel, and the wall temperature and oil temperature are relatively low, but also It is not possible to choose a cooling channel with an aspect ratio that is too large. This is because after the aspect ratio is greater than 1, the pressure loss in the channel increases as the aspect ratio increases; reducing the channel cross-sectional area can increase the flow rate, thereby increasing convection The heat transfer coefficient makes the maximum wall temperature and oil temperature lower, but the kerosene with a too small channel area has a larger flow resistance in the channel, and the effective heat exchange area also becomes smaller; an increase in the number of channels can make the transfer Heat is strengthened, and cooling channels can be arranged more when the temperature of the material permits to give full play to the kerosene heat sink and fin effect to enhance the cooling effect.

1. Introduction

For scramjet engines, the temperature of the combustion chamber wall is too high, the material is difficult to bear, and the mechanical performance drops sharply. If hydrocarbon fuel is used, it may also cause carbon deposition and block the active cooling channel; if the temperature is too low, it will cause the combustion chamber The loss of energy increases the coolant load and reduces engine efficiency. Therefore, the purpose of regenerative active cooling of the scram engine combustion chamber is to ensure that the solid wall temperature of the engine combustion chamber is within the allowable range of the material when the coolant mass flow rate is constant and the pressure drop and temperature rise meet certain conditions.

The structural parameters of the cooling channel have a great influence on the regenerative active cooling effect of the scramjet engine. There are many factors that affect the cooling structure: the thickness of the inner wall, the thickness of the fins, the aspect ratio of the channel, and other factors. This article will analyze the mechanism of these factors affecting heat transfer, compare the cooling effect of different channel parameters, and use the combination of theoretical analysis and numerical
simulation to calculate and analyze the characteristics of the heat transfer performance of the cooling channel under each parameter [1].

2. Calculation model and numerical method

2.1 Geometric model and basic assumptions

The scramjet with regenerative cooling channel is welded by different heat exchange panels. The simple structure is shown in Figure 1. The schematic diagram of the heat exchange plate with regenerative cooling channels is shown in Figure 2.

Due to the symmetry of the regenerative cooling channel, in order to simplify the calculation, only one half of the flow unit of one cooling channel is analyzed. FIG. 3 is a cross-sectional view of the half of the regenerative active cooling channel. Where a is half the rib width, b is the height of the cooling channel, c is the half width of the channel, d is the thickness of the outer wall, and e is the thickness of the inner wall. The channel material is alloy steel, and the coolant is domestic aviation kerosene. Its physical properties are given in [2].

Due to the complexity of heat transfer in the regeneration active cooling channel, the following assumptions are made about the theoretical model:

1) The fluid in the channel has no phase change latent heat release and absorption, the heat exchange process has the characteristics of unidirectional flow, and the fluid is in a supercritical state in the channel;

2) The cross-sectional shape of the channel is regarded as a rectangle;

3) The thermal resistance between the connectors is negligible;

4) The symmetrical surface of the side ribs is the thermal insulation surface;

5) The effect of radiative heat transfer is not considered;

6) Excluding heat dissipation from the channel to the environment;

7) Does not consider the effect of carbon deposition of aviation kerosene.
2.2 Numerical calculation method

2.2.1 Control equation

Control equations include mass continuity equation, energy equation, momentum equation, as shown in equations (1), (2), (3) respectively.

\[
\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i} (\rho u_i) = 0
\]  
(1)

\[
\frac{\partial}{\partial t} (\rho E) + \frac{\partial}{\partial x_j} (\rho(u_j (\rho E + p))) = \frac{\partial}{\partial x_i} (k_{\text{eff}} \frac{\partial T}{\partial x_i}) - \sum_{j'} h_{j'} J_{j'} + u_j (\tau_{ij})_{\text{eff}} + S_i
\]  
(2)

\[
\frac{\partial}{\partial t} (\rho u_i) + \frac{\partial}{\partial x_j} (\rho u_i u_j) = - \frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_i} + \rho g_i + F_i
\]  
(3)

In equation (2), \(k_{\text{eff}}\) is the effective thermal conductivity coefficient: \(k_{\text{eff}} = k_i + k_t\), \(k_t\) is the turbulent heat conduction coefficient, which is determined according to the turbulence model. \(J_{j'}\) is the diffusion flow rate of component \(\dot{j}\), \(S_i\) contains the heat of chemical reaction and other user-defined volumetric heat source terms.

\[
E = h - \frac{p}{\rho} + \frac{u_i^2}{2}
\]

In equation (3), \(\rho g_i\) and \(F_i\) are the gravity volume force and external volume force in the i direction, respectively, \(\tau_{ij}\) is the stress tensor:

\[
\tau_{ij} = [\mu (\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i})] - \frac{2}{3} \mu \frac{\partial u_i}{\partial x_j} \delta_{ij}
\]
where \(\mu\) is the viscosity coefficient, \(\mu = \mu_t + \mu_l\), \(\mu_t\) is the laminar viscosity coefficient, and \(\mu_l\) is the turbulence viscosity coefficient, defined according to the turbulence model.

2.2.2 Meshing and boundary conditions

Simulation conditions: the inlet mass flow rate of coolant aviation kerosene is 0.01 kg / s, the calculation area is long 0.5m, wide 1 = 5mm, inner wall heat flux density 1MW / m², and inlet pressure is 4 MPa. Channel parameters: half rib thickness 1mm, channel height 4mm, width 4mm, outer wall thickness 3mm, inner wall thickness 2mm.

To simplify the problem, the lateral heat transfer between adjacent channels is not considered, symmetry is used to reduce the number of grids, and the calculation time is shortened. The fluid-solid coupling interface is encrypted when dividing the grid, comparing the number of 200,000 grids and 600,000 grids The calculation result of the quantity, the temperature difference is within the range of 15K, so the calculation uses a grid of 200,000, and the grid division is shown in Figure 4.
In numerical calculation, proper and correct boundary conditions are of great significance to the accuracy of the calculation. The selection of different boundary conditions results in different results.

The inlet AO is the mass inflow inlet boundary conditions, mass flow rate \( \dot{m} = \dot{m}_0 \) inlet pressure \( p = p_0 \), inlet temperature \( T = T_0 \), turbulent kinetic energy \( k = k_0 \), turbulent dissipation rate \( \varepsilon = \varepsilon_0 \), where \( k_0 \) and \( \varepsilon_0 \) are shown in equation (4) and Equation (5), respectively.

\[
k_0 = \frac{3}{2} \left[ \bar{u} (0.16 \Re_{DH})^{-1/8} \right]^2
\]

\[
\varepsilon_0 = C_u \frac{k^{3/2}}{l}
\]

Where \( \bar{u} \) is the average fluid velocity, DH is the hydraulic diameter, \( C_u \) is the empirical constant taken as 0.09, and \( l \) is the turbulence length scale, approximately 0.07D.

The upper wall surface, the outlet wall and the inlet wall surface are insulated wall surfaces, and the wall heat flux density is zero. The lower wall surface is the wall surface of the heating section, and the heat flux density is the heat flux density applied in the physical model. Both sides are set as symmetrical boundary conditions, and the axial velocity, temperature and pressure are all zero. The outlet cross section is a pressure outlet, and the outlet back pressure is set to zero.

2.2.3 Solving governing equations
Solve the control equation according to the incompressible and variable physical conditions. Fluent is used for numerical simulation, the turbulence model uses the RNG \( k-\varepsilon \) model, the second-order upwind style is used to discretize the equation, and the pressure and velocity coupling solution is solved using the SIMPLEC algorithm. The time and space dispersion of all variables in the control equation have second-order accuracy.

3. Calculation structure and analysis

3.1 Influence of inner wall thickness on cooling
In order to study the effect of the inner wall thickness on the heat transfer performance of the cooling channel, the channel configuration of the inner wall surface of 1mm, 2mm, 3mm, and 4mm were studied without changing the height and width of the channel and the fin thickness. Its configuration parameters, maximum wall temperature, maximum oil temperature and pressure loss are shown in Table 1.
Table 1. Simulation calculation results of four different inner wall thickness configurations

| Structure | Inner wall thickness (mm) | Channel width (mm) | Channel height (mm) | Rib thickness (mm) | Wall temperature (K) | Oil temperature (K) | Pressure loss (MPa) |
|-----------|--------------------------|--------------------|---------------------|-------------------|----------------------|---------------------|---------------------|
| 1         | 1                        | 4                  | 4                   | 1                 | 842                  | 783                 | 1.807               |
| 2         | 2                        | 4                  | 4                   | 1                 | 886                  | 771                 | 1.801               |
| 3         | 3                        | 4                  | 4                   | 1                 | 940                  | 737                 | 1.797               |
| 4         | 4                        | 4                  | 4                   | 1                 | 1006                 | 775                 | 1.781               |

The calculation results show that the thicker the inner wall thickness of the cooling channel, the higher the maximum wall temperature, but the maximum temperature of kerosene does not change much. This is because when the heat transfer is stable, the heat flow absorbed through the wall surface is almost unchanged regardless of the inner wall thickness. All are taken away by the kerosene in the channel, so the maximum temperature of the kerosene is almost unchanged.

Figures 5 and 6 show the maximum wall temperature, maximum oil temperature and the pressure loss in the channel under different inner wall thickness configurations. It can be seen from the figure that as the thickness of the inner wall increases, the maximum wall temperature gradually increases, but the maximum oil temperature is not affected. The pressure loss in the channel changes very little, but it slightly decreases as the wall thickness increases. The distribution of wall temperature, centerline convective heat transfer coefficient, centerline oil pressure, and oil temperature along the channel at the centerline of the inner wall surface calculated with different inner wall thicknesses is shown in Figure 5.7-Figure 5.10.
It can be seen from the figure that the thinner inner wall corresponds to a lower wall temperature and a higher heat flux density. However, different inner wall thickness has little effect on the kerosene central flow temperature and pressure, but the central flow temperature and pressure increase as the inner wall thickness increases slightly decreased, it can be seen that the thickness of the inner wall has little effect on the temperature and oil pressure of kerosene in the channel, but it has a certain effect on the temperature of the inner wall and the convection heat transfer coefficient at the centerline.

### 3.2 Effect of fin thickness on cooling

To increase the amount of convection heat exchange in the cooling channel and improve the cooling efficiency, it can be achieved by increasing the heat exchange area. The use of fins in the cooling channel is an effective way to increase the heat exchange area. The so-called rib refers to an expanded surface attached to the base surface. In order to characterize the effectiveness of fin heat transfer, a parameter called rib efficiency is introduced $\eta_f$. For straight ribs of equal section, the rib efficiency:

$$\eta_f = \frac{h_P \theta_0 \delta (mH')}{m h_P H' \delta} = \frac{th(mH')}{mH'}$$

(6)

In the formula,

$$m = \frac{2h}{\lambda \delta}, \quad H' = H + \frac{\delta}{2}$$

Where $\delta$ is the fin thickness, $H$ is the channel height, $\lambda$ is the thermal conductivity, and $h$ is the convection heat transfer coefficient.

In the following, without considering the influence of the number of cooling channels, the heat exchange conditions in the channels with the thickness of the fins of 0.7 mm, 1.5 mm, 2 mm, and 3 mm are studied. Its configuration parameters, maximum wall temperature, maximum oil temperature and pressure loss are shown in Table 2.

| Structure | Inner wall thickness (mm) | Channel width (mm) | Channel height (mm) | Rib thickness (mm) | Wall temperature (K) | Oil temperature (K) | Pressure loss (MPa) |
|-----------|--------------------------|--------------------|--------------------|-------------------|---------------------|---------------------|-------------------|
| 5         | 0.7                      | 5.3                | 4                  | 2                 | 935                 | 824                 | 0.842             |
| 6         | 1.5                      | 4.5                | 4                  | 2                 | 917                 | 799                 | 1.434             |
| 7         | 2                        | 4                  | 4                  | 2                 | 886                 | 771                 | 1.800             |
| 8         | 3                        | 3                  | 4                  | 2                 | 825                 | 741                 | 3.861             |
Figures 11 and 12 show that thin fins correspond to higher wall and oil temperatures, but kerosene has a smaller pressure loss in the cooling channel, while thicker fins have lower wall thicknesses and oil temperatures, but pressure losses big. Therefore, the choice of rib thickness needs to consider the influence of various factors.

Figure 11. Comparison of temperature at different rib thickness.

Figure 12. Pressure loss under different rib thickness.

3.3 Effect of aspect ratio on cooling

The aspect ratio of the regenerative cooling channel has an important effect on the flow and heat transfer characteristics of kerosene in the channel. In order to explore how the aspect ratio affects the flow and heat transfer in the channel, this section does not consider the effect of the number of cooling channels and the cross-sectional area. Next, the heat transfer in the channel under the conditions of aspect ratio of 3, 1.56, 1, 0.64, and 0.34 was studied. Its configuration parameters, maximum wall temperature, maximum oil temperature and pressure loss are shown in Table 3.

| Structure | Inner Wall Thickness (mm) | Channel Width (mm) | Channel Height (mm) | Aspect Ratio (H/W) | Channel Area (mm²) | Rib Thickness (mm) | Wall Temperature (K) | Oil Temperature (K) | Pressure Loss (MPa) |
|-----------|--------------------------|--------------------|--------------------|-------------------|-------------------|-------------------|---------------------|---------------------|-----------------|
| 9         | 2                        | 2.3                | 6.9                | 3                 | 1                 | 857.364           | 748.402             | 2.157               |
| 10        | 2                        | 3.2                | 5                  | 1.56              | 1                 | 888.096           | 771                 | 1.925               |
| 11        | 2                        | 4                  | 4                  | 1                 | 16                | 886               | 775.372             | 1.801               |
| 12        | 2                        | 5                  | 3.2                | 0.64              | 1                 | 902.494           | 781.984             | 2.094               |
| 13        | 2                        | 6.9                | 2.3                | 0.34              | 1                 | 901.694           | 785.924             | 2.938               |

As the aspect ratio decreases, the maximum wall temperature of the cooling channel increases and the maximum oil temperature also increases. This is because the larger aspect ratio increases the heat transfer efficiency of the fins and enhances the convection heat. Figures 13 and 14 are comparisons of maximum wall temperature, maximum oil temperature and pressure loss at different aspect ratios $\text{Ar}$. 

Figure 13. Comparison of temperature at different aspect ratios.

Figure 14. Pressure loss at different aspect ratios.
The change of pressure loss with the aspect ratio of the channel cross section is shown in the figure. In the figure, the abscissa represents the aspect ratio of the channel, and the ordinate represents the flow pressure loss of the channel. It can be seen that under the condition that the inlet condition is unchanged, the channel cross-sectional aspect ratio AR is in the range of (0-3), the pressure loss decreases first and then increases with the increase of the channel cross-sectional aspect ratio, showing a nonlinear development trend. When the aspect ratio AR is in the range of (0-1), the pressure loss of the channel flow decreases rapidly with the increase of the cross-sectional aspect ratio. When AR = 1, the pressure loss is minimal, and the heat flux density introduced into the inner wall surface changes more uniform, and when kerosene enters the supercritical state, the convective heat transfer is significantly enhanced. At this time, the corresponding hydraulic diameter is larger, resulting in a smaller dimensionless distance. When the aspect ratio AR = 1, the pressure loss is minimal. When the aspect ratio AR is greater than 1, the pressure loss increases as the aspect ratio increases[10].

Figures 15 to 18 show the distribution of the inner wall surface centerline wall temperature, convection heat transfer coefficient, oil temperature at the center flow and oil pressure along the channel under different aspect ratios. It can be seen from Figure 15 that the greater the aspect ratio of the channel, the lower the inner wall temperature, and the distribution of the inner wall temperature along the channel shows a trend of normal heat transfer, heat transfer deterioration, and heat transfer enhancement. The degree of deterioration is low. Figure 16 shows the distribution of the convective heat transfer coefficients along the channel at the center line. It can be seen that when the wall temperature changes abruptly and the heat transfer deteriorates, the heat transfer coefficient also changes suddenly and the channel has a larger height and width. The convective heat transfer coefficient is larger, while the smaller heat transfer coefficient is smaller. The distribution of oil temperature change at the central flow is shown in Figure 17, the change trend is basically consistent and presents a linear growth trend, but the larger the same aspect ratio corresponds to the lower oil temperature. Figure 18 shows the distribution of oil pressure. The configuration with an aspect ratio of 1 corresponds to the lowest oil pressure, while the remaining configurations are higher than those with an aspect ratio of 1.
Figure 9. Central flow oil temperature with different channel aspect ratios.

Figure 10. Central flow oil pressure with different channel aspect ratios.

Figures 19 to 23 show the temperature distribution of the inlet section, center section, and outlet section of the cooling channel under different aspect ratios. This figure intuitively shows the temperature distribution of different cooling channels under different aspect ratios. It can be seen from the figure that the channel with a larger aspect ratio has a better cooling effect, and the wall temperature and oil temperature are relatively lower. However, in the design process of the regenerative cooling channel, the cooling channel with an excessively high aspect ratio cannot be selected, because the pressure loss in the channel increases as the aspect ratio increases after the aspect ratio is greater than 1.

Figure 19. Ar = 3 channel temperature distribution cloud.

Figure 20. Ar = 1.56 channel temperature distribution cloud.

Figure 21. Ar = 1 channel temperature distribution cloud.

Figure 22. Ar = 0.64 channel temperature distribution cloud.
3.4 Effect of cross-sectional area on cooling

In order to study the influence of the cross-sectional area of the channel on the regenerative cooling channel, without changing the thickness of the inner wall of the channel, the aspect ratio, and the thickness of the rib wall, only the size of the cross-sectional area of the channel was changed, and the cross-sectional area was simulated as 4mm², 9mm², 16mm², 25mm², 36mm². In the case of heat exchange in the channel, the configuration parameters, maximum wall temperature, maximum oil temperature and pressure loss are shown in Table 4.

| structure | Inner wall thickness (mm) | Channel width (mm) | Channel height (mm) | Channel Area (mm²) | Rib thickness (mm) | Wall temperature (K) | Oil temperature (K) | Pressure loss (MPa) |
|-----------|--------------------------|--------------------|--------------------|-------------------|-------------------|----------------------|---------------------|-------------------|
| 14        | 2                        | 2                  | 2                  | 4                 | 1                 | 796                  | 668                 | 3419              |
| 15        | 2                        | 3                  | 2                  | 3                 | 1                 | 840                  | 719                 | 2256              |
| 16        | 2                        | 3                  | 4                  | 9                 | 1                 | 889                  | 775                 | 1800              |
| 17        | 2                        | 2                  | 5                  | 16                | 1                 | 991                  | 799                 | 1109              |
| 18        | 2                        | 2                  | 6                  | 25                | 1                 | 1104                 | 857                 | 763               |

When the mass flow rate is constant, changing the cross-sectional area of the channel is to change the flow rate of the inlet. As can be seen from the above table, as the cross-sectional area of the channel increases, the maximum wall temperature and oil temperature increase, but the pressure loss Decreasing. As shown in Figure 24 and Figure 25, the relationship between the inlet cross-sectional area (inlet flow rate) and the maximum wall temperature, oil temperature, and channel pressure loss can be analyzed more clearly.

Figure 24. Comparison of temperature at different channel cross-sectional areas.

Figure 25. Pressure loss at different channel cross-sectional areas.
It can be seen from the analysis chart that reducing the cross-sectional area of the channel can increase the flow rate, thereby increasing the convective heat transfer coefficient, so that the maximum wall temperature and the oil temperature are both low, but this brings two disadvantages: First, the reduction in the cross-sectional area of the channel will cause the coolant flow resistance increases, which is more obvious at high heat flux density, because the higher heat flux density will make the flow resistance increase more obviously; the second is that the reduction of the cross-sectional area of the channel will reduce the contact area of kerosene and the channel, the effective cooling area is reduced.

3.5 Influence of the number of cooling channels
In this paper, the effects of different channel parameters on heat transfer are studied separately without considering the number of cooling channels. However, for a given size scramjet engine, changing the parameters of the channel will affect the channel. Changes in quantity, such as increasing the width of the fins, will inevitably reduce the number of channels, and decreasing the width of the channels increases the number of channels.

The research results show that, under the condition that the total kerosene mass flow rate is constant, when the number of channels increases, the flow rate of a single channel will decrease, the flow rate will slow down, the kerosene temperature increase will increase, and the pressure loss will also be smaller, which will be transmitted to the inner wall surface. The heat flux density is also reduced and helps to play the role of fins. Although the increase in the number of channels can strengthen the heat transfer under certain circumstances, the temperature rise of kerosene is relatively large, and it is easy for the material to reach the withstand value. Therefore, when designing the number of regenerative cooling channels, more cooling channels can be arranged when the temperature of the material allows, to give full play to the kerosene heat sink and fin effect, and enhance the cooling effect.

4. Summary
In this chapter, the influence of regenerative cooling channel parameters on the cooling effect is studied, and the effects of the inner wall thickness, fin width, channel aspect ratio, and channel cross-sectional area on the heat transfer effect are analyzed. The results show that:

(1) The thicker the inner wall of the cooling channel, the higher the maximum wall temperature, but the maximum temperature of kerosene does not change much. The thickness of the inner wall has little effect on the temperature and oil pressure of kerosene in the channel, but it has a certain effect on the inner wall temperature and the convection heat transfer coefficient at the centerline.

(2) Thin fins correspond to higher wall temperature and oil temperature, but the pressure loss of kerosene in the cooling channel is smaller, while the thicker fin thickness has the highest wall temperature and maximum oil temperature, but the pressure loss is larger. Therefore, the selection of rib thickness needs to consider the influence of various factors.

(3) The greater the aspect ratio of the channel, the better the cooling effect, and the wall temperature and oil temperature are relatively low. However, in the design process of the regenerative cooling channel, a cooling channel with an excessively high aspect ratio cannot be selected, because the pressure loss in the channel increases as the aspect ratio increases after the aspect ratio is greater than 1.

(4) Reducing the cross-sectional area of the channel can increase the flow rate, thereby increasing the convective heat transfer coefficient, so that the maximum wall temperature and oil temperature are lower, but the kerosene with a too small channel area has a large flow resistance in the channel and the effective heat transfer area also become smaller.

(5) The increase in the number of channels can strengthen the heat transfer under certain circumstances. When the material temperature allows, more cooling channels can be arranged to give full play to the kerosene heat sink and fin effect to enhance the cooling effect.

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