Numerical Study on Flow and Heat Transfer of Supercritical Hydrocarbon Fuel in Curved Cooling Channel

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Abstract: The fluid flow and heat transfer of hydrocarbon fuel play a significant role in developing regenerative cooling technology for advanced aeroengines. Numerical simulations have been conducted to investigate the flow and heat transfer characteristics of China RP-3 aviation kerosene with pyrolysis in a 3D, 90° bend, square cooling channel around the cavity flame-holder of a scramjet. A chemical kinetic model, composed of 18 species and 24 reactions, was adopted to simulate the fuel pyrolysis process. Results indicate that the secondary flow enhances the mixing of the fluid, thus, the transports of heat and components are improved between the near-wall region and main flow field in the curved channel. Compared with a straight cooling channel, fuel conversion and heat-absorbing capacity are higher, and the heat transfer is effectively enhanced in a curved cooling channel. In addition, with the increasing inlet mass flow rate and the decreasing radius of curvature, the velocity of the secondary flow increases. The heat and components are easily transferred from the near-wall region to the main flow. The non-uniformities of fuel temperature and conversion at the cross section decreases, which is helpful for improving the utilization of the level of fuel heat-absorbing capacity, and beneficial for enhancing the heat transfer.

Keywords: secondary flow; heat transfer; pyrolysis; hydrocarbon fuel; regenerative cooling

1. Introduction

Scramjets are the most promising power devices in air-breathing hypersonic flight, which has received increasing attention. As the flight Mach number increases, the thermal protection of combustion chambers becomes a critical issue due to its extremely thermal environment [1–5]. Active regenerative cooling technology is proposed as the most feasible thermal protection method, in which the hydrocarbon fuel of a scramjet is used as a coolant [2,6,7]. During the cooling process, hydrocarbon fuel provides not only physical heat sink but also additional chemical heat sink through endothermic reactions (also known as thermal cracking or pyrolysis [8]).

Much research has been conducted on the flow and heat transfer characteristics of hydrocarbon fuels with the consideration of fuel pyrolysis under supercritical conditions in straight cooling channels. Hou et al. [9] and Meng et al. [10] investigated the heat transfer and thermal cracking behaviors of aviation kerosene. Zhao [11] and Ward [12] studied the pressure effect on heat transfer and fuel pyrolysis. Feng et al. [13,14] quantitatively analyzed the coupling relationship between the flow and pyrolysis reactions of hydrocarbon fuel in a mini channel. Results reveal that, in a regenerative cooling system, the flow field and heat transfer are significantly influenced by fuel pyrolysis.
A curved channel is an important component and widely applied in the design of regenerative cooling technology, such as the cavity flame-holder, near the nozzle throat region and the connection area of a combustion chamber [15]. In curved cooling channels, the secondary flow is induced by the centrifugal force in the cross section [16,17]. Some literature [15,18–22] has been published on the influence of secondary flow on the flow field and heat transfer of hydrocarbon fuels without fuel pyrolysis. Results revealed that, compared to straight channels, the secondary flow enhanced flow mixing and promoted heat transfer in curved channels. However, these investigations ignore the pyrolysis process of hydrocarbon fuels.

Until recently, there was a lack of knowledge about the effect of curved configurations on flow field, heat transfer, and fuel pyrolysis. In a relevant paper, Jing et al. [23] investigated the flow and heat transfer characteristics of aviation kerosene in sharply curved cooling channels at supercritical pressures using a simplified pyrolytic reaction mechanism. Sun et al. [24] numerically studied the effect of centrifugal force on the flow field, heat, and mass transfers of hydrocarbon fuel in a helical channel, using the PPD model [25]. However, the results carried out by the PPD model were limited to situations with low fuel conversion. The reason is that many important secondary chemical reactions were ignored. In a curved channel (e.g., a surrounding cavity flame-holder or nozzle throat), the wall heat flux is very high, and a secondary reaction occurs, which makes the conversion of hydrocarbon fuel exceed the mildly cracked regime [26]. The secondary chemical reaction significantly influences the flow and heat transfer of supercritical hydrocarbon fuel in cooling channels [27]. However, there is little information available in the literature about the flow and heat transfer characteristics of hydrocarbon fuel with secondary reactions in curved cooling channels.

In this paper, numerical studies are carried out to analyze the flow and heat transfer characteristics of supercritical China RP-3 aviation kerosene under a wide range of fuel pyrolysis conditions in a 3D rectangular curved cooling channel. The detailed chemical reaction mechanism of RP-3 in Ref. [28] is adopted to simulate the fuel pyrolysis process. In addition, the influence of mass flow rate and curvature on flow and heat transfer in curved cooling channels is also investigated. The numerical model and its validation are described in Section 2. The model setup for the simulation is presented in Section 3. The characteristics of flow and heat transfer are displayed in Section 4. Then, heat transfer and fuel pyrolysis in curved channels at different mass flow rates and curvatures are investigated. Finally, the conclusions of this article are presented in Section 5.

2. Materials and Methods

2.1. Governing Equations and Solution Method

This numerical study was based on finite volume analysis software and user-defined functions. In the fluid region, the conservation equations of mass, momentum, energy, and species mass fractions were numerically solved to study the heat transfer and endothermic fuel pyrolysis of the China RP-3 aviation kerosene at a supercritical pressure.

\[ \nabla \cdot (\rho \vec{U}) = 0, \]

\[ \nabla \cdot (\rho \vec{U} \vec{U}) - \nabla P - \nabla \cdot \tau = S_M, \]

\[ \nabla \cdot (\rho \vec{U} H) - \nabla \cdot \lambda_{\text{eff}} \nabla T - \nabla \cdot \left( \tau_{\text{eff}} \cdot \vec{U} \right) = 0, \]

\[ \nabla \cdot \left( \rho \vec{U} Y_i \right) - \nabla \cdot \rho D_{i,m,\text{eff}} \nabla Y_i = R_i, \]

where \( \rho, \vec{U}, p, \tau, H, T \) and \( D \) are the density, velocity, pressure, viscous stress, enthalpy, temperature, and diffusion coefficient of the fluid, respectively. \( S_M \) is the source term of the
chemical reaction, $\lambda$ is the thermal conductivity, $Y_i$ is the mass fraction of specie $i$, and $R$ is the molar gas constant. The subscripts $m$ and $eff$ represent the mixture and effective parameter.

In the solid region, the heat conduction equation is numerically solved:

$$\nabla \cdot (\lambda \nabla T) = 0,$$

(5)

where $\lambda$ is the thermal conductivity of the solid wall. In addition, the temperature and heat flow boundary conditions at the interface between fluid domain and solid domain were set as follows to meet the continuity conditions:

$$T_{w,s} = T_{w,f},$$

$$q_{w,s} = q_{w,f},$$

(6)

For the mass diffusion coefficient, Fuller’s empirical formula was used at first to obtain the binary mass diffusion coefficient $D_{ij}$ at low pressure:

$$D_{ij} = \frac{0.00143 T^{1.75}}{PM_{ij}^{1/2} \left[ (\sum v_i)^{1/3} + (\sum v_j)^{1/3} \right]},$$

(7)

where $M_{ij}$ is the mixed molar mass of species $i$, and species $j$; $v$ is the diffusion volume.

Then, the effect of pressure was considered through the corresponding-states method suggested by Takahashi, as shown below:

$$\frac{D_{ij,p}}{(D_{ij})^+} = f(T_r, p_r),$$

(8)

where superscript $+$ indicates the value under low pressure; $T_r$ and $p_r$ are the comparison temperature and pressure, respectively.

Finally, based on the binary diffusion coefficient, the effective diffusion coefficient was calculated by the following formula:

$$D_{ij,m} = \frac{1 - x_i}{\sum_{j \neq i} x_i / D_{ij}},$$

(9)

where $x_i$ is the mole fraction of species $i$.

In this paper, $\kappa$-SST turbulent model was adopted to calculate the internal turbulent flow and heat transfer for its high accuracy and good numerical stability to the boundary [29,30]. The fuel density was calculated with the Peng–Robinson equation of state because the physical properties of fuel change dramatically with temperature under supercritical conditions [31,32]:

$$P = \frac{RT}{V - b} - \frac{a}{V + 2bV - b^2},$$

(10)

where $V$ is the volume, and $a$, $b$ are parameters.

The empirical method proposed by Chung was selected to estimate viscosity and thermal conductivity of hydrocarbon fuel [33]. The calculation method of the above thermophysical properties was entered into the commercial CFD software FLUENT through the user-defined function UDF. In the numerical calculation, the solver was based on the finite-volume scheme, and the SIMPLEC algorithm.

2.2. Chemical Kinetics Model

The typical hydrocarbon fuel, HF-I, a commercial China RP-3 aviation kerosene, was selected as the coolant. This paper used the detailed chemical kinetic model of China RP-3 aviation kerosene [27], which contains 18 species and 24 reactions, and is suitable for a
wide range of operating conditions. The numerical model used to simulate the fluid flow, heat transfer, and fuel pyrolysis of RP-3 under supercritical conditions was verified in our previous work [34]. The results show a good agreement with experiment data of the fuel bulk temperature and conversion of RP-3.

2.3. Validation of Numerical Model

Model validations were further conducted in a curved channel based on the experimental data of supercritical nitrogen heat transfer obtained by Michael L. Meyer at NASA Lewis Research Center [35]. The curved channel with a cross-sectional aspect ratio of 1 and a 45° bend with 137 mm mean radius in the heated portion was adopted for model validation. The inside and outside dimensions of the square channel were 4.14 mm × 4.14 mm and 6.17 mm × 6.17 mm, respectively. The inlet velocity was 21.94 m/s, the inlet temperature of fuel was 280 K, the back pressure was 8.3 MPa, and the heat flux was 0.28 MW/m².

The experimental heat transfer data and numerical results are described in Figure 1. Here, the Nusselt number, \( N_u \), is the experimental Nusselt number, which is calculated by its definition, and the Nusselt number calculation, \( N_{u,\text{calc}} \), is the reference Nusselt number, which was obtained by a semi-empirically developed equation [36]:

\[
N_{u,\text{calc}} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} \left( \frac{T_w}{T_h} \right)^{-0.3},
\]

where \( T_h \) is the bulk fluid temperature, and \( T_w \) is the wall temperature. \( \text{Re} \) and \( \text{Pr} \) are the Reynolds number and the Prandtl number, respectively.

![Figure 1. Validation of heat transfer in curved channel.](image)

In the straight heat section (0 < \( x/D_h < 28 \)), the maximum discrepancy of the ratio was less than 20%, which was within the uncertainty of the semi-empirical model. In the curved heat section (28 < \( x/D_h < 53.77 \)), the discrepancy of the experimental data was greater than the numerical data, which is because the experimental data considered the effect of the electrical power input and the cooling enthalpy gain [36]. Very good agreements between experimental data and numerical results can be observed in Figure 1. Therefore, the numerical model is suitable to anticipate the heat transfer characteristics in a curved channel under supercritical conditions.
3. Model Setup

The cavity is an efficient flame-holding structure in the scramjet engine combustor [37] and is illustrated in Figure 2. In this study, a typical cooling channel around the leading edge of the cavity was selected as the study object. The geometry of the cooling channel for numerical simulation is shown in Figure 3 [38]. The inside and outside dimensions of the rectangular channel were $2\,\text{mm} \times 2\,\text{mm}$ and $4\,\text{mm} \times 4\,\text{mm}$, respectively. Therefore, its hydraulic diameter $D_h$ was $2\,\text{mm}$. The rectangular cooling channel consisted of four parts. The first part (Part I) was a straight section with a length of $30D_h$ to provide fully developed turbulent flow. The second part (Part II) was a straight section with a length of $10D_h$ to provide a developed thermal boundary layer. The third part (Part III) was a $90^\circ$ curved channel with a radius of $25D_h$. The last part (Part IV) was a straight outlet section with a length of $10D_h$, which was used to eliminate the curvature effect. The heating section (Part II, Part III, and Part IV) was heated from the concave side, with a constant heat flux of $5\,\text{MW/m}^2$ imposed on the external wall. The inlet mass flow rate of hydrocarbon fuel was $1\,\text{g/s}$, and the inlet temperature was $700\,\text{K}$. The back pressure was set to be $3\,\text{MPa}$.

![Figure 2. The flow-path configuration of engine [37].](image1)

![Figure 3. Schematic and dimensions of simulation model: (a) cross section of channel (b) schematic of channel along axial direction.](image2)

4. Results and Discussion

4.1. Effect of Secondary Flow on Pyrolysis and Heat Transfer

According to the previous analysis, the secondary flow in a curved channel enhances the fluid mixing and thermal exchange. In this paper, a case was also calculated in a straight channel for the purpose of comparison. The fuel conversion and heat transfer coefficient

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were numerically conducted in a straight channel and a curved channel to analyze the effect of the secondary flow on pyrolysis and heat transfer. The sectional geometry and boundary conditions of the straight channel were consistent with the curved channel, except that the bending radius is infinite.

The conversion of RP-3, $Z_{\text{RP-3}}$, is defined as [39]:

$$Z_{\text{RP-3}} = \left(1 - \frac{\int_A \rho u Y_{\text{RP-3}} dA}{\int_A \rho u dA}\right) \times 100\%,$$

where $u$ is the velocity of fluid, and $A$ is the surface area of the channel.

The distributions of fuel temperature and conversion at the cross sections of 0°, 15°, 45°, and 75° in the curved cooling channel, and at the same positions in the straight cooling channel, are presented in Figures 4–7. The cooling channel was made of alloy and had high thermal conductivity. Although the channel was one-side-heated, the temperature of the hydrocarbon fuel in the near-wall regions increased first. Due to the higher fluid temperature, the fluid in the near-wall region had a higher chemical reaction rate than that in the main flow in both kinds of channels. Moreover, under the action of molecular viscous force, the hydrocarbon fuel in the near-wall region had a smaller axial velocity and longer residence time than that in the main flow. Therefore, in both curved and straight channels, the fuel conversion was high in the near-wall region and low in the main flow.

![Figure 4](image)

**Figure 4.** Temperature distribution at different cross sections: (a) curved channel and (b) straight channel.

![Figure 5](image)

**Figure 5.** Fuel temperature in the centerline along y axis at different cross sections.
However, in the curved channel, the secondary flow pattern appeared in cross sections because of the centrifugal force, as shown in Figure 8. The secondary flow drove the fluid with low velocity near the wall inward, while the fluid with high velocity in the main flow swept outward [40]. The intensity of the secondary flow can be characterized by the secondary flow velocity [40], which is calculated as $V_{sec} = \sqrt{(u_y)^2 + (u_z)^2}$.

In the curved cooling channel, the high-temperature fluid in the near-wall region moved to the main flow due to the secondary flow. The fuel in the central region flowed towards the wall, as shown in Figure 8. The secondary flow transferred energy and components from the near-wall region to the main flow, and the radial heat and mass transfers were promoted. Therefore, compared with the straight cooling channel, the fuel in the curved cooling channel absorbed more heat. Meanwhile, due to the secondary flow, the main flow temperature increased faster in the curved cooling channel (as shown in Figure 5), and the chemical reaction rate was larger. Fuel conversion in the main flow of the curved channel was higher, as shown in Figure 7. Moreover, as the secondary flow enhanced mass transfer, the non-cracked hydrocarbon fuel in the main flow was pushed to the near-wall region, and the pyrolytic products in the near-wall region were transported to the main flow, and more hydrocarbon fuel underwent pyrolysis. For these two reasons, in the curved cooling channel, fuel conversion was higher, and the chemical heat absorption was larger, as shown in Figures 7 and 9. At the outlet, the average fuel conversion of the curved channel was 34.25%, and that of the straight channel was 27.17%.
The above discussions show different distributions of fuel temperature and conversion between curved and straight cooling channels. Fuel temperature and conversion distributions are closely related to heat transfer. Thus, a comparative analysis of the heat transfer coefficient of curved and straight cooling channels is carried out. The heat transfer coefficient $h$ is defined as [38]:

$$h = \frac{q''}{T_w - T_b'},$$

where $q''$ is the wall heat flux. The bulk fuel temperature, $T_b$, is defined as follows [38]:

$$T_b = \frac{\int_A \rho u C_p T dA}{\int_A \rho u C_p dA},$$

where $C_p$ is the specific heat capacity.

The secondary flow promoted the heat and mass transfers of the hydrocarbon fuel. The fuel conversion and heat-absorbing capacity were higher in the curved cooling channel than those of the straight cooling channel. Therefore, as shown in Figure 10, the heat transfer coefficient in the curved cooling channel was higher. At the outlet of the curved section, the heat transfer coefficient increased by around 90%. Based on the above analysis,
the secondary flow can effectively promote the heat and mass transfers of fuel and improve the cooling efficiency of a cooling channel.

Figure 10. Variation of heat transfer coefficient along the flow direction.

4.2. Effect of Mass Flow Rate on Heat and Mass Transfers in the Curved Cooling Channel

In the curved channel, the centrifugal force and secondary flow were directly proportional to the inlet mass flow rate of fuel. The change of the inlet mass flow rate may induce different heat and mass transfer characteristics. Therefore, in this section, the variations of fuel conversion and heat transfer coefficient distributions with the inlet mass flow rate, \( m \), are analyzed to study the secondary flow effect on the heat and mass transfers of hydrocarbon fuels. The three different mass flow rates were set to be 1.341 g/s, 1.788 g/s and 2.235 g/s, respectively. The relevant parameters are listed in Table 1.

Table 1. Boundary conditions in different mass flow rate cases.

| Case | \( m \) (g/s) | \( T_{in} \) (K) | \( P_{in} \) (MPa) | Re_{in} | \( q'' \) (W/m²) |
|------|---------------|----------------|-------------------|--------|-----------------|
| 1    | 1.341         | 700            | 3                 | \( 6.0 \times 10^4 \) | \( 5.0 \times 10^6 \) |
| 2    | 1.788         | 700            | 3                 | \( 8.0 \times 10^4 \) | \( 6.67 \times 10^6 \) |
| 3    | 2.235         | 700            | 3                 | \( 1.0 \times 10^5 \) | \( 8.33 \times 10^6 \) |

Furthermore, results in Section 4.1 indicate that the distributions of fuel temperature and conversion in the cross section of the cooling channel present non-uniformity, which significantly affects fuel heat-absorbing capacity. Therefore, the non-uniformity coefficient is introduced to study the influence of inlet mass flow on the heat and mass transfer of hydrocarbon fuels in a curved cooling channel. Therefore, the non-uniformity coefficient is introduced to study the influence of inlet mass flow rate. The non-uniformity coefficients of fuel temperature, \( R_T \), and coefficients of fuel conversion, \( R_Z \), are defined as follows [41]:

\[
R_T = \frac{T_{\max} - T_{\min}}{T_{\text{mean}}} \times 100\%, \quad (15)
\]

\[
R_Z = \frac{Z_{\max} - Z_{\min}}{Z_{\text{mean}}} \times 100\%, \quad (16)
\]

where \( T_{\max}, T_{\min} \) and \( T_{\text{mean}} \) are the maximum, minimum, and mass-weighted average temperatures in the cross section of the cooling channels. \( Z_{\max}, Z_{\min} \) and \( Z_{\text{mean}} \) are the maximum, minimum, and mass-weighted average conversions in the cross sections of the cooling channels.
With the increase in inlet mass flow rate, the centrifugal force became larger. The average velocity of secondary flow in the cross-sectional planes increased with the inlet mass flow rate, as shown in Figure 11. The heat and components in the near-wall region were easily transported to the main flow. As shown in Figures 12 and 13, the non-uniformity coefficients of fuel temperature and conversion decreased when the inlet mass flow rate increased. Therefore, with the increasing inlet mass flow rate, the fuel heat-absorbing capacity increased, and the heat transfer coefficient increased, as shown in Figure 14. When the inlet mass flow rate changed from 1.341 g/s to 2.235 g/s, the heat transfer coefficient at $x/D_h = 80$ increased by 30.5%. The increasing inlet mass flow rate enhanced the heat and mass transfers and improved the utilization level of the fuel heat-absorbing capacity in the curved cooling channel.

![Figure 11. Secondary flow velocity with different mass flow rates along the flow direction.](image1)

![Figure 12. Distribution of $R_T$ with different mass flow rates along the flow direction.](image2)
4.3. Radius of Curvature Effect on Heat and Mass Transfers in the Curved Cooling Channel

In the curved channel, the centrifugal force and secondary flow velocity were inversely proportional to the radii of the curvature. In this section, the supercritical hydrocarbon fuel flow field for various radii of curvature, $R$, is numerically investigated to study the secondary flow effect on the heat and mass transfers of hydrocarbon fuels.

The four radii of curvature were 100 mm, 150 mm, 200 mm, and 250 mm, respectively. The total length of the curved cooling channel was 450 mm. The geometric dimensions of the curved channels are shown in Table 2. Parts I, II, and IV are straight channels, and Part III is a 1-rad bending section. The parameters in the numerical simulations are the same as those of case 1.

Table 2. Geometric parameters in simulations with different radii of curvature.

| Case | $R$ (mm) | Part I (mm) | Part II (mm) | Part III (mm) | Part IV (mm) |
|------|----------|-------------|--------------|---------------|--------------|
| 4    | 100      | 50          | 50           | 100           | 250          |
| 5    | 150      | 50          | 50           | 150           | 200          |
| 6    | 200      | 50          | 50           | 200           | 150          |
| 7    | 250      | 50          | 50           | 250           | 100          |
Since the radius of curvature is inversely proportional to the centrifugal force, the smaller the radius of curvature, the greater the secondary flow velocity, as shown in Figure 15. Therefore, with the decreasing radius of curvature, the heat and components exchange between the near-wall region and the main flow is much more severe (shown in Figure 16). The non-uniformity coefficients of fuel temperature and conversion decrease along the mass flow direction, as shown in Figures 17 and 18. Thus, with the decreasing radius of curvature, the fuel heat-absorbing capacity rises, and the heat transfer coefficient increases, as shown in Figure 19. When the radius of curvature changed from 100 mm to 200 mm, the heat transfer coefficient at \( x/D_h = 90 \) decreased by 7.2%. The increasing radius of curvature deteriorated the heat and mass transfers and inhibited the utilization level of fuel heat sink in the curved cooling channel.

Figure 15. Secondary flow velocity with different radii of curvature along the flow direction.

Figure 16. Distributions of flow field at the cross section of \( x/D_h = 75 \) with different radii of curvature: (a) Temperature and (b) mass fraction of RP-3.
Figure 17. Distribution of $R_T$ with different radii of curvature along the flow direction.

Figure 18. Distribution of $R_Z$ with different radii of curvature along the flow direction.

Figure 19. Heat transfer coefficient with different radii of curvature radius along the flow direction.
5. Conclusions

In this paper, the flow and heat transfer characteristics of China RP-3 aviation kerosene in the curved rectangular regenerative cooling channel were numerically studied. A detailed pyrolytic chemical reaction mechanism was adopted to describe the hydrocarbon fuel pyrolysis. The secondary flow effect on fluid flows and heat transfer was studied by comparing the distribution of the fuel conversion and heat transfer coefficient in both a curved cooling channel and a straight cooling channel. In addition, the fluid flows and heat transfer in the curved cooling channel at different inlet mass flow rates and radii of the curvature were also investigated. The main results are summarized as follows:

1. The secondary flow enhanced the fluid mixing and thermal exchange in the curved cooling channel. Additionally, the fuel conversion and heat-absorbing capacity was higher, and the heat transfer was effectively enhanced. At the outlet of the curved section, the heat transfer coefficient increased by around 90%.

2. With the increase of inlet mass flow rate, the centrifugal force became larger. The heat and components exchange between the near-wall region and the main flow was much more severe. The non-uniformities of the fuel temperature and conversion at the cross section decreased, which was beneficial for the promotion of fuel pyrolysis, and helpful for improving the utilization level of the fuel heat-absorbing capacity. The increasing inlet mass flow rate can effectively enhance the heat transfer in a curved cooling channel.

3. The radius of curvature was inversely proportional to the centrifugal force, so as the radius of curvature decreased, the secondary flow velocity increased. Therefore, the increasing radius of curvature raised non-uniformities of the fuel temperature and conversion at the cross section, which suppresses the fuel pyrolysis and reduces the utilized efficiency of the fuel heat-absorbing capacity. The increasing radius of curvature deteriorated the heat transfer in the curved cooling channel.

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Nomenclature

| Symbol | Description | Unit |
|--------|-------------|------|
| A      | surface area | m²   |
| a, b   | parameters in PR Eos |       |
| C_p   | specific heat capacity | J/kg/K |
| D     | diffusion coefficient | m²/s |
| D_h   | hydraulic diameter | mm   |
| U, u  | velocity | m/s |
| V     | volume | m³   |
| v     | diffusion volume | m³   |
| V_{sec} | secondary flow velocity | m/s |
| x, y, z | three coordinate directions of | Cartesian coordinate system |
| Symbol | Description |
|--------|-------------|
| $h$   | heat transfer coefficient, W/(m$^2$·K) |
| $x_i$ | mole fraction of species i |
| $H$   | Enthalpy, J/Kg |
| $y$   | mass fraction |
| $M$   | mass flow rate, g/s |
| $Z_{RP-3}$ | conversion of RP-3 |
| $M_{ij}$ | the mixed molar mass of species i and species j, kg/(K·mol) |
| $Nu$  | Nusselt number |
| $\rho$ | density, kg/m$^3$ |
| $Pr$  | Prandtl number |
| $\tau$ | viscous stress, N/m$^2$ |
| $q''$ | curvature radius, mm, or molar gas constant, J/(K·mol), or non-uniformity |
| $R$   | Reynolds number |
| $S_M$ | source term of chemical reaction |
| $T$   | Temperature, K |
| $T_b$ | Bulk fluid temperature |
| $T_w$ | wall temperature |
| $T_r$ | comparison temperature |
| Greek |
| $\lambda$ | thermal conductivity, W/(m·K) |
| $\rho$ | effective parameter |
| $\nu$ | wall parameter |
| $b$   | bulk fluid parameter |
| in    | inlet parameter |
| $S$   | mixture |
| $M$   | value under low pressure |

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