

Numerical investigation of transcritical liquid film cooling in a methane/oxygen rocket engine

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Abstract: Transcritical film cooling was investigated by numerical study in a methane cooled methane/oxygen rocket engine. The respective time-averaged Navier-Stokes equations have been solved for the compressible steady three-dimensional (3-D) flow. The flow field computations were performed using the semi-implicit method for pressure linked equation (SIMPLE) algorithm on several blocks of nonuniform collocated grid. The calculation was conducted over a pressure range of 202 650.0 Pa to 1.2×10^7 Pa and a temperature range of 120.0 K to 3 568.0 K. Twenty-nine different cases were simulated to calculate the impact of different factors. The results show that mass flow rate, length, diameter, number and diffused or convergence of film jet channel, injection angle and jet array arrangements have great impact on transcritical film cooling effectiveness. Furthermore, shape of the jet holes and jet and crossflow turbulence also affect the wall temperature distribution. Two rows of film arranged in different axial angles and staggered arrangement were proposed as new liquid film arrangement. Different radial angles have impact on the film cooling effectiveness in two row-jets cooled cases. The case of in-line and staggered arrangement are almost the same in the region before the second row of jets, but a staggered arrangement has a higher film cooling effectiveness from the second row of jets.

Key words: liquid film cooling; numerical study; rocket; transcritical;
semi-implicit method for pressure linked equation (SIMPLE);
film cooling effectiveness

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Nomenclature

c	concentration	α	axial angle of film nozzle
C	empirical constants in turbulence dissipation rate equation	β	radial angle of film nozzle
D	diameter of the jet holes	ϵ	turbulence energy dissipation rate
D_{f}	diffusivity	μ	viscosity coefficient
h	enthalpy	μ_{t}	eddy-viscosity coefficient
I	grid number in the axial direction	ρ	density
J	grid number in the radial direction	σ_{k}	empirical constants in turbulence kinetic energy equation
K	enthalpy	σ_{e}	empirical constants in turbulence dissipation rate equation
k	turbulence kinetic energy	τ	stress tensor
L	length of film nozzle	Γ	mass flow rate
n	number of the jet holes	Θ	film cooling effectiveness
p	pressure		
Pr	Prandtl number		
Sc	Schmidt number		
T	temperature		
u	velocity		

Subscript

aw	adiabatic wall
c	central of the main flow

cool	coolant film	l0	liquid film inlet
cr	critical point	n	every slice
g0	main flow inlet	out	outlet
i	coordinate direction		

Introduction

In rocket engines, film cooling is widely used to protect the combustion chamber walls from hot gases. Either a liquid or gaseous cooling film can protect chamber walls not only in the vicinity of the injection, but also farther downstream. Especially the liquid film, whose adiabatic film cooling effectiveness is up to 90% when it keeps liquid, still has a good effect on protecting chamber walls after it has been transformed into steam. Many factors affect film cooling efficiency, such as mass flux ratio quantities, injection angle, hole shape, length to diameter of the jet channel, jet array arrangements (single or double rows), jet holes streamwise or spanwise spacing, jet and crossflow Reynolds numbers, density ratio, jet and crossflow turbulence intensity, compressibility effects and shock/boundary layer/jet interactions and so on. Javadi K H, et al^[1] made a conclusion of many studies on numerical study of film cooling.

A series of numerical simulation of film cooling has been studied^[2-9], and parts of that used computational fluid dynamics (CFD) software such as rocket combustion flow analysis module (ROCFLAM)^[10-11]. Goldstein and his partners^[12] found that a common range of blowing ratio for industrial applications is 0.5 ~ 1.0. In Dellimore K H and his friends' study^[3], the model shows that compressibility influences film cooling performance by altering the growth rate of the shear layer between the main and cooling streams. Walters D K and Leylek J H^[13] studied on streamwise injection with cylindrical holes in film cooling physics.

However, most of these studies focused on gas film cooling or subcritical liquid film cooling. Actually, in many cases, liquid film is in a supercritical state because of the high pressure in

rocket combustors. ZHANG Wei and his partners^[14] studied on intersect diffusion of liquid film cooling under supercritical. CHEN Jianhua, et al^[15] analyzed the effect of the film local injection structure on the local flow and cooling effect near the outlet of film cooling slot for the thrust chamber of liquid oxygen LOX/kerosene staged combustion cycle engine. Currently, the pressure of advanced large-scale liquid rocket engine is more than 10 MPa, which has the trend of further increase, and the combustion chamber heat is greater. Multi-band technology of film cooling ring has been used successfully in high-pressure staged combustion engine^[16].

Transcritical liquid film cooling combines the characteristics of liquid film cooling and gas film cooling, but is significantly different from anyone of them. If the film is injected below the critical temperature and then heated by the hot mainstream gas, it passes through the critical temperature at high pressure, which is why transcritical liquid film has a very special performance. When the pressure is more than the critical value, both fluid latent heat of vaporization and surface tension are zero. So the divergence between hot gas and cooling liquid is different from the one under subcritical. Film thickness and length cannot be defined like the definition under subcritical state because there is no vaporization and there is also difference on heat and mass transfer mechanism between subcritical and transcritical. Transcritical rocket engine film cooling technology was studied in this paper. Therefore, by doing three dimensional numerical simulation of transcritical liquid film on different situations, this paper compared the influence of different factors.

1 Material properties

Fig. 1 is a typical phase diagram of pure flu-

ids. Liquid film is gradually evaporated and entrainment in a rocket under subcritical state. But in many cases, liquid film is injected at high pressure but low temperature. The temperature of the film is gradually increased and the pressure of the film is gradually decreased in the nozzle. Because of its different characteristics, liquid film under transcritical state also shows different heat transfer characteristics.

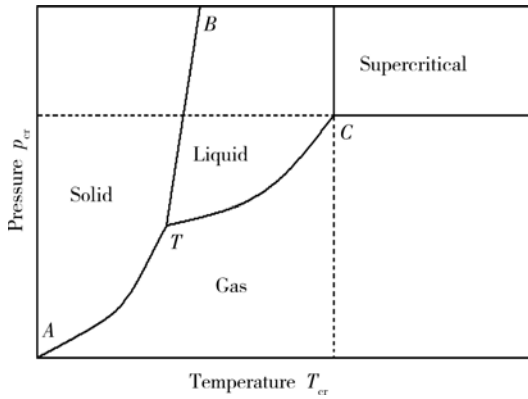


Fig. 1 Phase diagram of pure fluids

In this paper, the thermodynamic and transport properties for pure fluid were calculated by a C program with different methods. As a representative of transcritical fluid, methane is used here. The critical temperature of methane T_{cr} is 190.56 K, and the critical pressure p_{cr} is 4.599 MPa. The results have been compared with NIST (National Institute of Standards and Technology) data. The simulation pressure is between 0.1 MPa and 12.0 MPa, and the temperature range is from 120 K to 2500 K.

According to the comparison of Virial equation^[17], Cubic equation^[18] and a semi-empirical formula^[19], Lee-Kesler^[20] method has been used in supercritical region, virial equation of state has been used in subcritical gas zone, and interpolation of NIST data has been used in subcritical liquid zone for density of methane in this paper. TRAPP (Transport Properties Prediction)^[21] method has been used for conductivity because the error of it is less than that of Chung^[22] value. Furthermore, TRAPP method is better than Lucas^[23] method in calculating the viscosity of methane whether in subcritical area

or supercritical area.

2 Simulation model

The thin liquid film is fed with inlet liquid temperature T_{l0} , inlet pressure p_{l0} , and inlet coolant flow rate Γ_{l0} . The mixture gases flow enters the chamber with an inlet main flow rate Γ_{g0} , uniform temperature T_{g0} and concentration c_{g0} . The following assumptions are made.

- 1) The transport processes are steady and the flow is compressible.
- 2) The main stream gas consists of the combustion products of methane/oxygen, and the combustion process is completed instantaneously and sufficiently.
- 3) Radiation and gravity are neglected.
- 4) The wall of the chamber is adiabatic.

2.1 Flow governing equations

The time-averaged Navier-Stokes equations for the compressible steady three-dimensional (3-D) flow can be written as follows:

- 1) Continuity equation

$$\frac{\partial}{\partial x_j}(\rho u_j) = 0 \quad (j = 1, 2, 3) \quad (1)$$

- 2) Momentum equation

$$\frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} \quad (i, j = 1, 2, 3) \quad (2)$$

Where stress tensor:

$$\tau_{ij} = \mu \left[\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right] - \frac{2}{3} \mu \frac{\partial u_k}{\partial x_k} \delta_{ij},$$

$$\delta_{ij} = \begin{cases} 1 & (j = i) \\ 0 & (j \neq i) \end{cases}$$

- 3) Energy equation

$$\frac{\partial}{\partial x_j}(\rho u_j h) = \frac{\partial}{\partial x_j} \left[\frac{\mu}{Pr} \frac{\partial h}{\partial x_j} + \mu \left(\frac{1}{Sc} - \frac{1}{Pr} \right) \sum_{i=1}^N h_i \frac{\partial c_i}{\partial x_j} \right] \quad (3)$$

- 4) Species transport equation

$$\frac{\partial}{\partial x_j}(\rho u_j c_i) = \frac{\partial}{\partial x_j} \left(\rho D_{if} \frac{\partial c_i}{\partial x_j} \right) \quad (4)$$

2.2 Turbulent model

The standard $k-\epsilon$ model is a semi-empirical model which is widely used in flow field calculation^[24].

$$\bar{\rho} \bar{u}_j \frac{\partial \tilde{k}}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\left(\frac{\mu_t}{\sigma_k} + \mu \right) \frac{\partial \tilde{k}}{\partial x_j} \right] - \frac{\overline{\rho u_i'' u_j''}}{\rho} \frac{\partial \tilde{u}_i}{\partial x_j} - \frac{\mu_t}{\rho} \frac{\partial \bar{\rho}}{\partial x_j} \frac{\partial \bar{p}}{\partial x_j} - \bar{p}_\epsilon \quad (5)$$

$$\bar{\rho} \bar{u}_j \frac{\partial \tilde{\epsilon}}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\left(\frac{\mu_t}{\sigma_\epsilon} + \mu \right) \frac{\partial \tilde{\epsilon}}{\partial x_j} \right] - C_{\epsilon 1} \frac{\tilde{\epsilon}}{\bar{k}} \left[\frac{\overline{\rho u_i'' u_j''}}{\rho} \frac{\partial \tilde{u}_i}{\partial x_j} - \frac{\mu_t}{\rho} \frac{\partial \bar{\rho}}{\partial x_j} \frac{\partial \bar{p}}{\partial x_j} \right] - C_{\epsilon 2} \bar{p} \frac{\tilde{\epsilon}^2}{\bar{k}} \quad (6)$$

Generally, $\frac{\mu_t}{\rho} \frac{\partial \bar{\rho}}{\partial x_j} \frac{\partial \bar{p}}{\partial x_j}$ is not considered in turbulent combustion model in many current literatures. When density remains unchanged, the situation is inconsistent with the real combustion. Especially at transcritical state involved in this calculation, the density is from more than 400 kg/m^3 to less than 10 kg/m^3 , and it has great impact on flow field calculation of edge area.

3 Physical domain and boundary conditions

Geometry and the location of film are shown in Fig. 2 and Fig. 3. Here, the three-dimensional, compressible, and stationary flow field computations were performed with the semi-implicit method for pressure linked equation (SIMPLE) algorithm on a nonuniform collocated grid. The first row of cooling film is at 0.01 m from the entrance, and the second row of cooling film is at 0.08 m or 0.1 m from the throat. A small angle of engine flow field is calculated, which is $\frac{360^\circ}{n}$ of the whole flow field as shown in Fig. 4, where n is the number of jet holes.

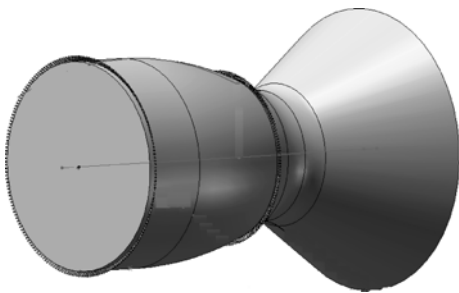


Fig. 2 Physical domain

Fig. 4 shows the meshing. The main flow field is divided into six fields. The first field is

entrance field, where the grid is relatively dense, and the first row of cooling film enters the main flow field from here. The second field is central area of the combustion chamber. The third field is convergence zone of the combustion chamber, and the second row of cooling film enters from the tail of this field. The fourth and the fifth fields are throat convergent field and expansion field, where the grids are more intensive at the throat. The last field is expansion field of nozzle. There are 20676 elements for the main flow field, and 410 to 826 elements for any film inject.

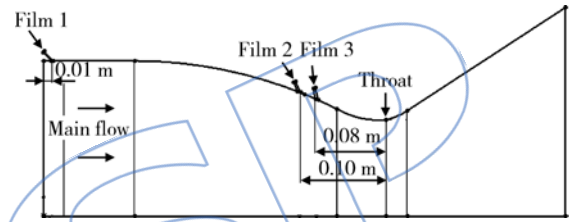


Fig. 3 Front view of the film cooled rocket

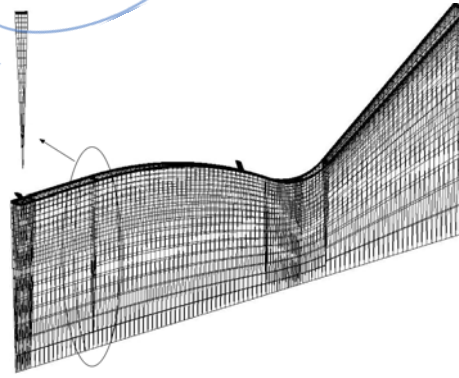


Fig. 4 Meshing of the calculation fields

There are twelve layers of the boundary layer mesh for better calculation of the boundary flow fields. Different from gas film cooling, transcritical film cooling requests more on the accuracy of boundary calculation. In this field, the calculation of methane tends to be instable since its great change of density and other physical properties between 120 K and 300 K .

In all the cases, the entrance pressure of the main flow p_{g0} is $1.01 \times 10^7 \text{ Pa}$, and the entrance temperature T_{g0} is 3568.0 K . The whole mass flow rate of the main flow Γ_{g0} is 116.075 kg/s , so

the mass flow rate of every slice of the main flow Γ_{g0n} is $\frac{116.075}{n}$ (kg/s). The entrance pressure of the film p_0 is 1.2×10^7 Pa, and the entrance temperature T_0 is 120.0 K. There are eight components in the main flow at the entrance, i. e., CO, H₂, H₂O, CO₂, O₂, H, OH, and O. The mass fractions of them are 0.2598, 0.0093, 0.4167, 0.2464, 0.0194, 0.0011, 0.0423, and 0.0050. The outlet pressure p_{out} is 202650.0 Pa.

Twenty-nine different cases were simulated in the present study, including one case without

film cooling and 28 cases with film cooling. Cases in Table 1 and Table 2 are cases of single-row holes of film cooling. Cases 1~3 are only

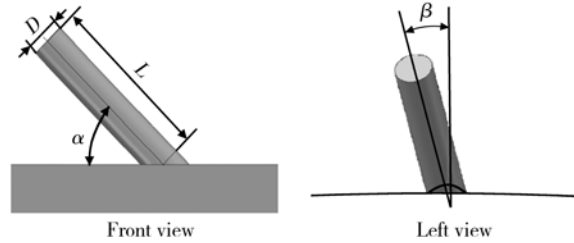


Fig. 5 Geometry dimensions of the film injects

Table 1 Parameters of cases 1~13 of single-row holes

Case No.	Mass flow rate/%	Length of film nozzle/mm	Number of the holes	Diameter/m	Axial angle/(°)	Radial angle/(°)
1	5	10	200	0.002	45	0
2	7	10	200	0.002	45	0
3	10	10	200	0.002	45	0
4	7	15	200	0.002	45	0
5	7	10	180	0.002	45	0
6	7	10	160	0.002	45	0
7	7	10	200	0.0016	45	0
8	7	10	200	0.0024	45	0
9	7	10	200	0.002	30	0
10	7	10	200	0.002	60	0
11	7	10	200	0.002	75	0
12	7	10	200	0.002	45	15
13	7	10	200	0.002	45	30

Table 2 Parameters of cases 14~19 of single-row holes

Case No.	Shape of the holes	Diffused or convergent	Turbulent intensity of main flow input /(°)	Turbulent intensity of film input /(°)
2	round	straight	2	2
14	square	straight	2	2
15	trapezoidal	straight	2	2
16	round	diffused	2	2
17	round	convergent	2	2
18	round	straight	2	5
19	round	straight	5	2

different in mass flow rate of the film inlet. Cases 2 and 4 are only different in the length of film nozzle. Cases 2, 5 and 6 are different in the number of the holes, but the whole mass flow rates of film are equal. Cases 2, 7 and 8 are not the same in diameter of the film nozzle. Cases 2, 9, 10 and 11 are not alike in axial angle of film nozzle, and cases 2, 12 and 13 are different in radial angle. In Fig. 5, D and L are the diameter and length of film nozzle. α and β are axial angle and radial angle of film nozzle. Cases 14 ~ 17 give the results of impact of different shapes of film holes. Cases 2, 18 and 19 are inconsistent

in turbulent degree of flow inlet. In addition, the parameters which are not shown in tables are the same with case 2.

Table 3 shows parameters of cases of double-row holes. The second row of holes is at position film 2 or film 3 in Fig. 6. Cases 20 ~ 23 are different in the distribution of mass flow. The second row of holes in cases 24 and 25 is at position film 3. Case 26 is different because of its staggered arrangement of jet arrays. Cases 27 and 28 are special in the radial angle of nozzles.

Wall function method and finite difference method were used in solving Eqs. (1) ~ (6).

Table 3 Parameters of cases of double-row holes

Case No.	Mass flow rate of film 1/%	Mass flow rate of film 2/%	Mass flow rate of film 3/%	Staggered or in-line arrangement	Radial angle of film 1/(°)	Radial angle of film 2/(°)	Radial angle of film 3/(°)
20	5	5	0	in-line	0	0	
21	4	6	0	in-line	0	0	
22	6	4	0	in-line	0	0	
23	7	3	0	in-line	0	0	
24	5	0	5	in-line	0		0
25	4	0	6	in-line	0		0
26	6	4	0	staggered	0	0	
27	6	4	0	in-line	15	15	
28	6	4	0	in-line	-15	15	

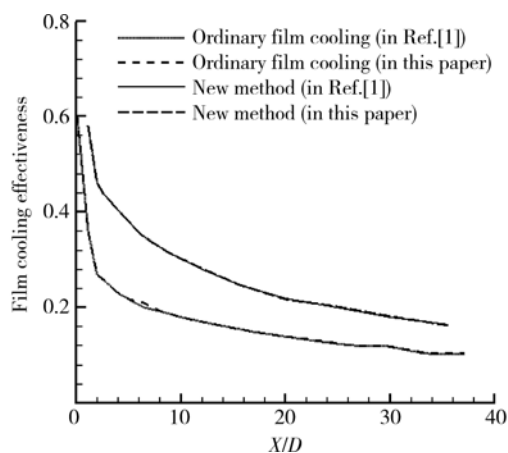


Fig. 6 Comparison of results in Ref. [1] and in this paper

4 References certification and grid independent test

The method has been used in many references. Cases in Refs. [1-3] have been tested by the calculation method in this paper and the results are basically consistent with the references. Fig. 6 shows the comparisons of our results and results in Ref. [1]. As it is obviously from these numbers, the results in this paper are effective.

For case 1, four types of grids have been used here to validate the grid independence. Table 4 shows different grids arrangements. It can be seen from Fig. 7 that it is not too much differ-

Table 4 Various grid arrangements

No.	Area 1 $I \times J \times K$	Area 2 $I \times J \times K$	Area 3 $I \times J \times K$	Area 4 $I \times J \times K$	Area 5 $I \times J \times K$	Area 6 $I \times J \times K$	Boundary layer $J \times K$
Grid 1	$10 \times 28 \times 4$	$14 \times 28 \times 4$	$10 \times 28 \times 4$	$16 \times 28 \times 4$	$8 \times 28 \times 4$	$28 \times 28 \times 4$	12×4
Grid 2	$10 \times 45 \times 4$	$14 \times 45 \times 4$	$10 \times 45 \times 4$	$16 \times 45 \times 4$	$8 \times 45 \times 4$	$28 \times 45 \times 4$	15×4
Grid 3	$20 \times 28 \times 4$	$28 \times 28 \times 4$	$20 \times 28 \times 4$	$32 \times 28 \times 4$	$16 \times 28 \times 4$	$56 \times 28 \times 4$	12×4
Grid 4	$20 \times 45 \times 4$	$28 \times 45 \times 4$	$20 \times 45 \times 4$	$32 \times 45 \times 4$	$16 \times 45 \times 4$	$56 \times 45 \times 4$	15×4

ence between the results of grid 1 and grid 4, but the latter one needs more compute time than the former one. So grid 1 has been chose here.

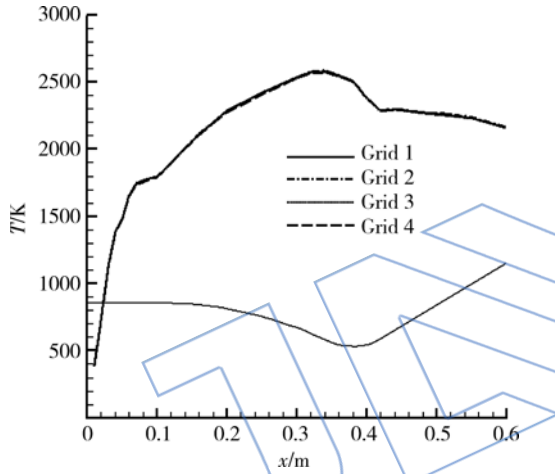


Fig. 7 Comparisons of T for various grid arrangements for a typical case

5 Results and discussion

Arnold, et al^[25] and Kirchberger, et al^[26] used the film cooling effectiveness to measure the efficiency of film cooling:

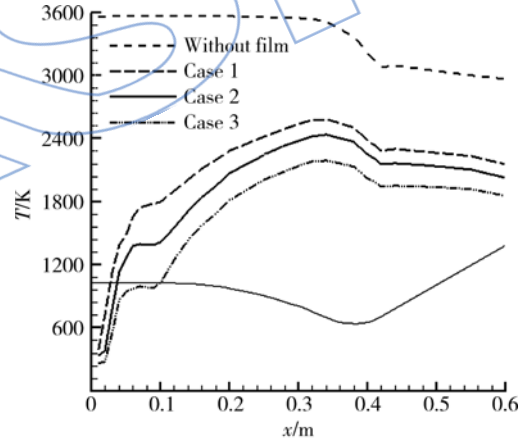
$$\Theta(x) =$$

$$\frac{T_{\text{wall, withoutfilm}}(x) - T_{\text{wall, withfilm}}(x)}{T_{\text{wall, withoutfilm}}(x) - T_{\text{cool}}} \quad (8)$$

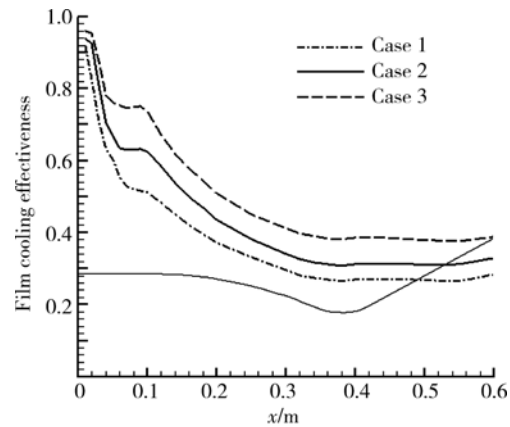
5.1 Influence of mass flow rate on film cooling effectiveness

Generally, no more than 10% of fuel or oxidizer is used for film cooling, 5% of the fuel in case 1, 7% in case 2 and 10% in case 3 has been used to analyze the film cooling effectiveness.

Fig. 8 shows the relationship between wall temperature or film cooling effectiveness and mass flow rate. As can be seen from Fig. 8(b), when the mass flow rate increases, the cooling effectiveness also obviously enhances. In addition, the difference downstream is greater than upstream. This is because the mixing of transcritical film and main flow is weaker in the upstream, so the cooling effect hasn't been brought



(a) Temperature at different x



(b) Film cooling effectiveness at different x

Fig. 8 Different mass flow rates of the film

into full play within a short distance in the upstream. But with continuously mixing of film and main flow, the cooling efficiency was reduced. However, the increase of mass flow rate can make the reduction of film cooling effectiveness slower.

It should be noted that the temperature of the film rises to above the critical point quickly after it is injected into the combustor chamber. The change of physical properties is very large in the section where the film just been injected, and also the film cooling effectiveness is very good at this section. Theoretically, the film cooling efficiency will be higher if the mass flow is greater. Actually, incomplete combustion in the border region appears because of the use of film cooling, which results in that the loss of impulse will become greater with the increasing of mass flow rate. When the film is just injected into the chamber, the temperature of the film increases rapidly to the critical point as the absorption of heat which is transferred from the hot gas. In Fig. 8(a), the temperature gradient forefront is the smallest, because the physical properties of transcritical liquid film below the critical temperature are similar to liquid.

Fig. 8(a) also shows that the temperature gradient becomes smaller before the convergence section in all the cases. The location of the convergence section causes velocity of the liquid near the wall become smaller, but the temperature of the coolant has increased. The temperature difference between the film and main flow and the velocity difference between them become smaller, so the heat flux is smaller. That's why the temperature elevation here is smaller than it at the first section of the chamber, and it is more obvious when the flow rate of the liquid film is larger because the decrease of the heat transfer coefficient and heat flux is more obvious. Wall heat flux increases in the convergence section, so the temperature gradient is bigger from $x=0.1$ m.

It is also can be seen from Fig. 8 that the film not only influence the entrance region but

also the back of the engine.

5.2 Influence of length, diameter and number of film jet channel on film cooling effectiveness

Generally, the longer the length of film jet channels, the more loss of momentum of the coolant in the cooling channels. So, it's not a big difference in the front of the cooling section but the cooling effect is slightly worse in the latter part if the length of film jet channel is longer, which is clearly shown in Fig. 9 that about from $x=0.04$ m the film cooling effectiveness of case 2 ($L=0.01$ m) is better than that of case 4 ($L=0.015$ m). However, their effectiveness is almost the same at the beginning because the effect of small loss of momentum has not yet been clear. At the end of the expansion area of the nozzle, this effect becomes smaller, and length has little effect on the film cooling effectiveness.

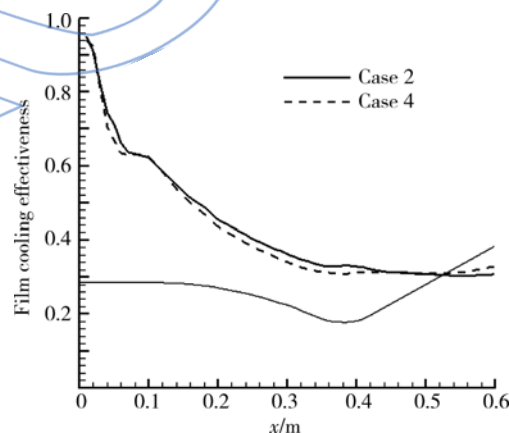


Fig. 9 Different lengths of the film jet channel

If the diameter of the coolant entrance is larger, cover area of coolant is larger at the beginning, so coolant distribution is more uniform at this section. In Fig. 10, case 7 ($D=0.0016$ m) is worse than case 2 ($D=0.002$ m) and case 8 ($D=0.0024$ m) at the beginning. But because of the same flow rate, flow velocity of the film will be slower at the condition of larger diameter and temperature increases very fast at the beginning, so the effectiveness of large diameter is not better than small diameter from about $x=0.17$ m.

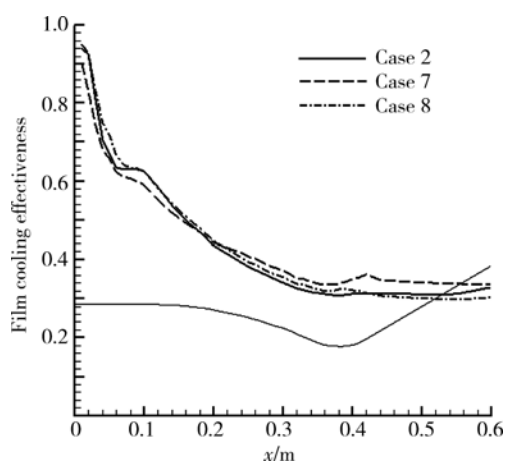


Fig. 10 Different diameters of the jet holes

Similarly, if the number of the jet holes is larger, coolant distribution is more uniform at the beginning section, where the film cooling effectiveness is better. But in case 2 ($n=200$), case 5 ($n=180$) and case 6 ($n=160$), the impact of the number of the jet holes is not so apparent because the cooling holes are close enough in all the cases. Yet we can still find that at the front and central part case 2 is the best but it is the worst at the last section by Fig. 11.

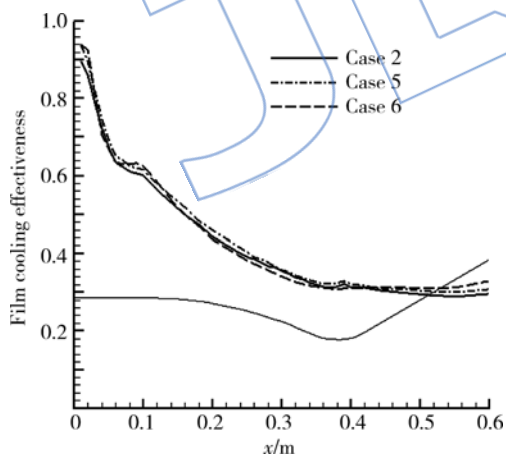


Fig. 11 Different numbers of the jet holes

5.3 Influence of injection angle on film cooling effectiveness

The injection angle influences the penetration of coolant jets into a hot crossflow boundary layer, the mixing process, the flow separation, and wake formation after the coolant jet. In Fig. 12, case 2 ($\alpha=45^\circ$), case 9 ($\alpha=30^\circ$), case

10 ($\alpha=60^\circ$), case 11 ($\alpha=75^\circ$) are different in axial angle. Incident angle change the coolant flow near the wall. With decreasing of the incident angle α , momentum vertical to the main flow of the cooling film decreases, so the coolant is more close to the wall, which thereby improves the cooling effect. α is the smallest in case 9, the cooling effectiveness of which is the best in the whole flow field. Therefore, if we want to take advantage of transcritical film cooling, small angle incident or even tangential incident should be chosen.

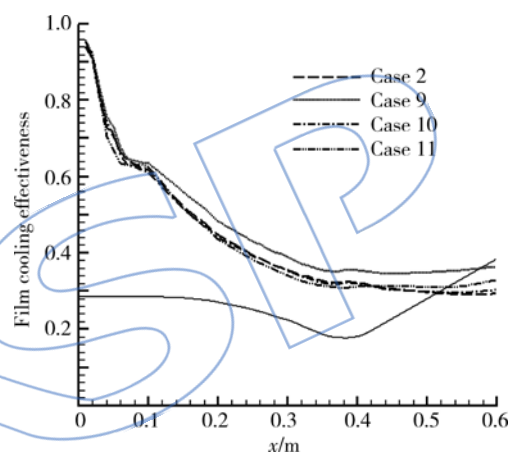


Fig. 12 Different axial angles of the jet holes

Case 2 ($\beta=0^\circ$), case 12 ($\beta=15^\circ$) and case 13 ($\beta=30^\circ$) are different in radial angles. Fig. 13 shows that the primary role of the introduction of lateral inclination is to make the coolant adherent to the wall in a certain incident angle. On the one hand, it makes more uniform distribution of the coolant on the wall, and avoids the situation that centerline cooling effect is better than the external. On the other hand, a certain radial angle increases the distance of the coolant flow which attached to the wall, thus the cooling area has been increased. With the increasing of β , wall temperature drops near injects region, but the attenuation of the film is speeded up, and the area of low temperature region reduces. That is because the distance from upstream to downstream of the coolant is much more if the radial angle is bigger, which results in the reduction of cooling effect. Therefore, suitable ra-

dial angle should be chosen by analyzing the specific circumstances.

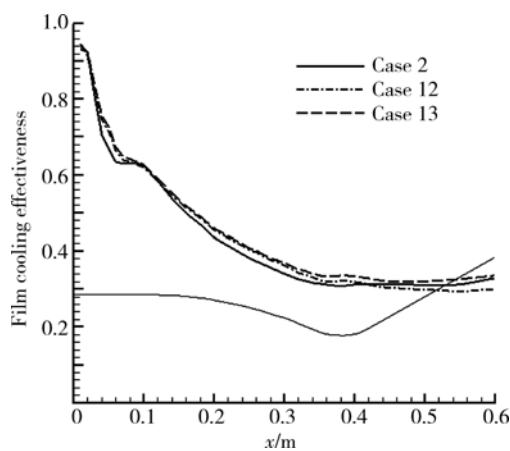


Fig. 13 Different radial angles of the jet holes

5.4 Influence of shape and diffused or convergence of the jet holes on film cooling effectiveness

Fig. 14 shows different shapes of the jet holes. Jet hole shapes affect the vortical flow generated downstream of the jet and the interactions between the coolant jet and the hot cross-flow severely. Case 2 (round), case 14 (square) and case 15 (trapezoidal) are compared because they are only different in shape of the jet holes. The areas of trapezoidal and square are the same with round. Fig. 15 shows that under the condition of the same mass flow rate, film cooling effectiveness of these cases is extremely close. Trapezoidal makes the initial distribution area of the coolant larger. Hence its cooling effectiveness is slightly better at the beginning. But with the diffusion of the coolant, the film cooling effectiveness of the three cases is close.

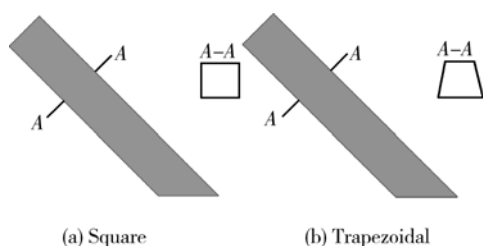


Fig. 14 Shape of the jet holes

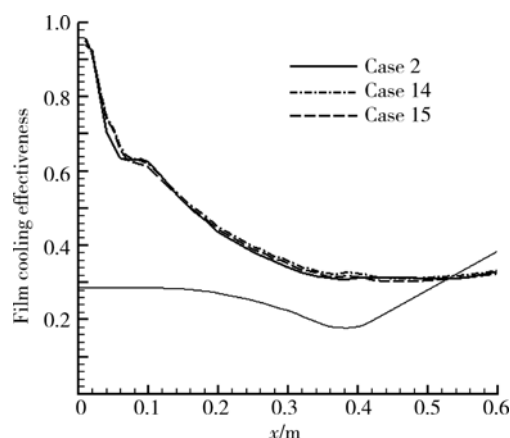


Fig. 15 Different shapes of the jet holes

Diffused and convergence of a jet hole is shown in Fig. 16. Diffused and convergence of a jet hole is shown in Fig. 16. Cooling effect of the diffused jet holes is better than convergent ones upstream. And low temperature area of diffused condition is larger. But owing to the increase of velocity and decreases of density and pressure, the mixing of coolant of diffused jet holes and main flow is faster than the mixing of coolant of convergent jet holes. In case 2 the jet hole is straight, in case 16 it is diffused and in case 17 it is convergent. So, it can be seen from Fig. 17 that curves intersect with each other in the downstream area. The cooling effectiveness is worse after the initial rapid decay in the early stage. Similarly, the cooling effectiveness of convergent holes is better upstream and worse downstream. However, in the calculations of this paper, since the whole calculation area is not very long, the effect of attenuation is not fully demonstrated even at the end of the nozzle. The cooling effectiveness of diffused holes is still the best in this area.

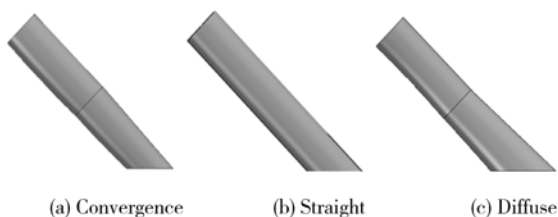


Fig. 16 Diffused and convergence of a jet hole

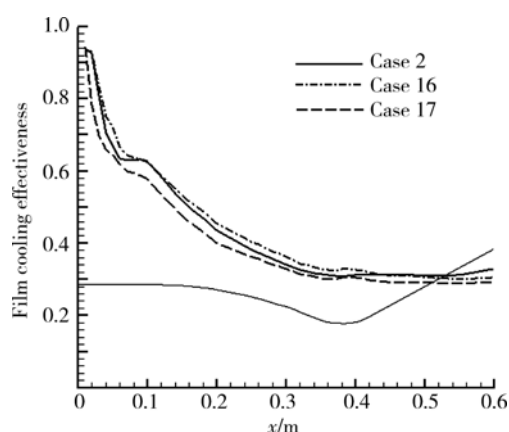


Fig. 17 Diffused or convergence

5.5 Influence of jet and crossflow turbulence on film cooling effectiveness

The jet and crossflow turbulence intensity considerably enhances the mixing of the coolant jet with the main hot crossflow. In gas film cooling, high mainstream turbulence level has little effect at low-blowing ratios, for which the coolant jets remain attached to the surface. At high-blowing ratios, however, for which the coolant jets are somewhat detached, high mainstream turbulence level causes a decrease in film cooling effectiveness. In this paper, the jet and main flow turbulent intensity of case 2 are 2 percent and 2 percent. These two parameters of case 18 and case 19 are 5 percent and 2 percent, 2 percent and 5 percent. Similarly, the increase of jet turbulence intensity has little impact on cooling effectiveness for the injection speed of transcritical film is limited. Fig. 18 also shows that the

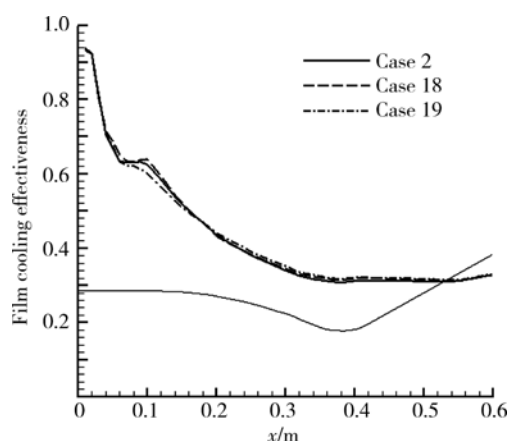


Fig. 18 Different jets and crossflow turbulences

increase of main flow turbulence intensity makes the film cooling effectiveness lower but the impact is not so much, for mixing and diffusion between the film and main flow is intensified.

5.6 Influence of jet array arrangements on film cooling effectiveness

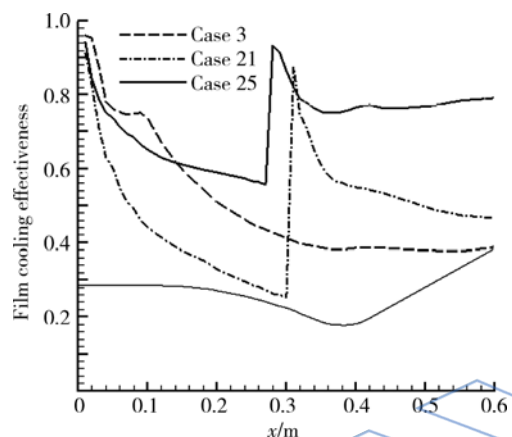
Both of the suitable location of coolant jets and their arrangements over the surfaces are the most important factors in film cooling. In case 3, all of the 10 percent of flow rate was injected into combustor chamber from location film 1 (Fig. 6). In case 21 and case 25, the mass flow is divided into two parts: one is 4 percent injected from location film 1, the other is 6 percent injected from location film 2 or film 3. Fig. 19 (a) shows that, when the location of coolant jets is suitable, the cooling effectiveness of double-row holes is much better than the effectiveness of single-row holes. Especially at the back of the combustor chamber, when the cooling efficiency of the first row of jets has been down, the effect of addition of some other coolant is apparent. So, the cooling effectiveness of case 25 is much better than case 3.

Furthermore, suitable locations of coolant jets are very important for several row of film cooling in different engine shapes. In case 20 and case 24, coolant is divided into two equal parts. Fig. 19 shows that when the location of the second row of coolant is at film 2 is better than that when it is at film 3 for the current engine shape. Because of the same coolant flow rate of the jets at film 1, the film cooling effectiveness of both cases is good in the first section. Liquid film was transferred to high temperature gas completely after the first low temperature section, and the velocity of it is large, so coolant added at a suitable location will make the velocity of the gas before it near wall smaller and affect the heat transfer between the main flow and coolant.

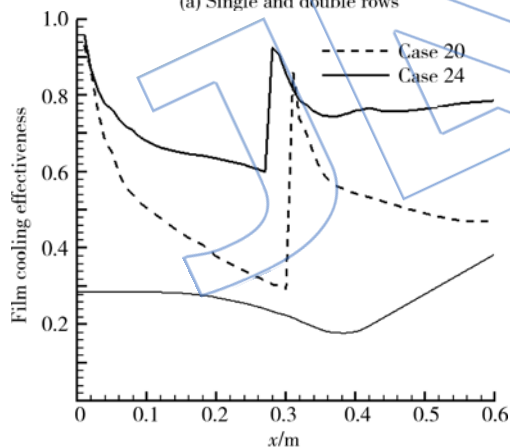
In Fig. 19(c), the flow distribution of location film 1 and film 2 are: case 20 (5%, 5%), case 21 (4%, 6%), case 22 (6%, 4%), case 23 (7%, 3%). The cooling effectiveness before the second row of jets is decided by mass flow rate of

the first row of film jets. The larger the flow rate of it is, the higher the cooling effectiveness is. Yet the impact of mass flow rate of the second row of jets is not so much. The film cooling effectiveness rises rapidly at the point where the second row of coolant is injected. Although it is not so obvious, if the mass flow rate of the second row of jets is more, the cooling effectiveness is higher from there.

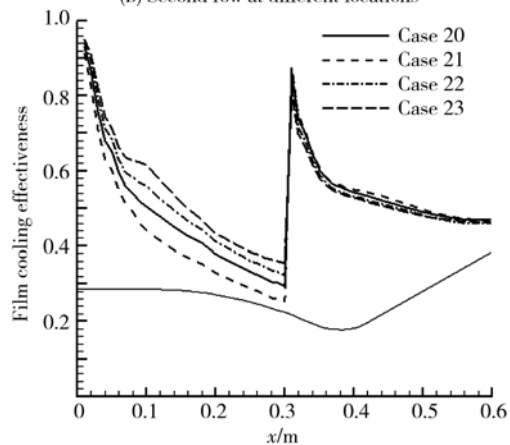
For the same geometry and for the same flow characteristics, a staggered arrangement of



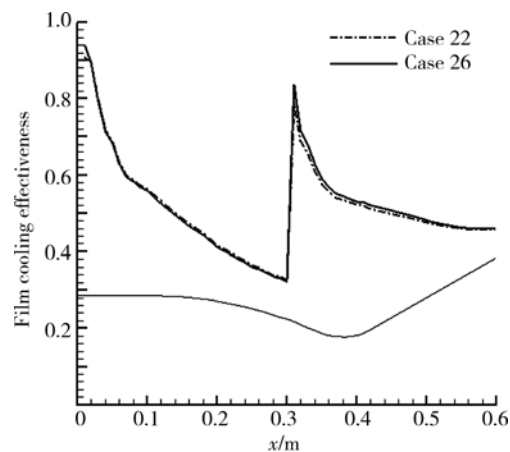
(a) Single and double rows



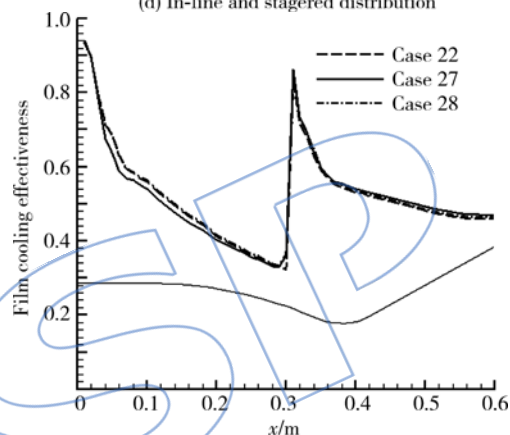
(b) Second row at different locations



(c) Different flow rates distribution at two rows of jets



(d) In-line and staggered distribution



(e) Different radial angles of film jets

Fig. 19 Different jet array arrangements

jet arrays has higher overall film cooling effectiveness compared with the regular (in-line) arrangement. Fig. 20 shows the difference between them. The staggered arrangement helps the jets to recover for each other, leading to a more uniform coolant film distribution. In Fig. 19 (d) case 22 (in-line) and case 26 (staggered) are almost the same in the region before the second row of jets, but a staggered arrangement of jet arrays has a higher film cooling effectiveness from the second row of jets. For the same reason,

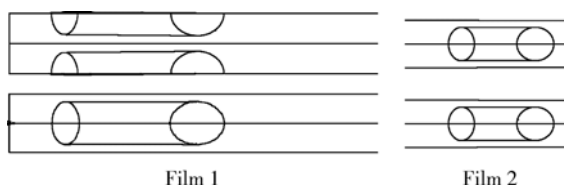


Fig. 20 Staggered arrangement and in-line arrangement

the increasing of radial angle can increase the film cooling effectiveness because it optimizes the distribution of the coolant. Fig. 18(e) shows that the effectiveness of case 27 ($\beta_{\text{film1}} = 15^\circ$, $\beta_{\text{film2}} = 15^\circ$), case 28 ($\beta_{\text{film1}} = -15^\circ$, $\beta_{\text{film2}} = 15^\circ$) is almost the same in the region before the second row of jets, but they are a little better than case 22 ($\beta_{\text{film1}} = 0^\circ$, $\beta_{\text{film2}} = 0^\circ$).

6 Conclusion

In this work, a C program has been used to calculate the thermodynamic and transport properties for pure fluid by some different methods. By comparison of the results, Lee-Kesler method and virial equation of state were used for density of methane, and TRAPP method has been used for the calculation of conductivity and viscosity.

The time-averaged Navier-Stokes equations for the compressible steady 3-D flow have been solved with the standard $k-\epsilon$ model in a transcritical film cooling engine. Twenty-nine different cases were simulated in the present study. The results show the following conclusions of transcritical liquid film cooling:

1) The cooling effectiveness is obviously enhanced with the increasing of the film coolant mass flow. And the difference downstream of them is greater than upstream.

2) When the length of film jet channel is longer, it's not a big difference in the front of the cooling section but the cooling effect is slightly worse in the latter part. Similarly, when the diameter of the coolant entrance or the number of the jet holes is larger, the film cooling effectiveness is better at the beginning.

3) Small angle incident or even tangential incident should be chosen if we want to take advantage of transcritical film cooling. Suitable radial angle should be chose by analyzing the specific circumstances.

4) The cooling effectiveness of trapezoidal inject holes is slightly better at the beginning. However, with the diffusion of the coolant, the cooling effectiveness of the trapezoidal, round

and square cases is close. Cooling effect of the diffused jet holes is better than convergence ones upstream, and low temperature area of it is larger.

5) The increase of jet turbulence intensity has little impact on cooling effectiveness of transcritical film cooling. The increase of main flow turbulence intensity makes the film cooling effectiveness lower but the impact is not so much.

6) The cooling effectiveness of double-row holes is much better than that of single-row holes when the location of coolant jets is suitable. The flow distribution of the two locations is very important for the film cooling effectiveness. The case of in-line arrangement and staggered arrangement are almost the same in the region before the second row of jets, but a staggered arrangement has a higher film cooling effectiveness from the second row of jets. When radial angle $\beta_{\text{film1}} = 15^\circ$, $\beta_{\text{film2}} = 15^\circ$ and $\beta_{\text{film1}} = -15^\circ$, $\beta_{\text{film2}} = 15^\circ$, the film cooling effectiveness is almost the same in the region before the second row of jets, but they are a little better than case of $\beta_{\text{film1}} = 0^\circ$, $\beta_{\text{film2}} = 0^\circ$.

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