Numerical Simulation of Transpiration Cooling within Variable Property of Water and Steam

Meng Wang¹, Fei He¹,*, Nan Wu¹, Jianhua Wang¹ and Guangqi Dong²

¹ Department of Thermal Science and Energy Engineering, University of Science and Technology of China, Jinhai Road 96, Hefei 230027, PR China
² Beijing Power Machinery Research Institute, Beijing 100074, PR China

ehefeihe@ustc.edu.cn

Abstract. Transpiration cooling with coolant phase change provides a wide range of applications in effective thermal protection. Water is usually employed as the coolant and experiences huge temperature and pressure changes during the transpiration cooling process. Mathematical model and numerical approach of transpiration cooling with phase change in consideration of the real variation of water thermodynamic properties are established in this paper, and compared with the constant property model. The IWAPS Industrial Formulation 1997 is used to calculate the thermodynamic properties of water in wide ranges of temperature and pressure. The numerical results obtained indicate some thermodynamic properties vary significantly with temperature and pressure, leading to higher phase change temperature and coolant driving force. With the increase of mass flow rate, the two property models exhibit the similar changing trend in temperature and pressure and there is a lag in the start and end positions of phase change.

1. Introduction

During hypersonic flight in the near space, aerodynamic heat is one of the severest problems, which leads to extremely high temperature of the vehicle surface, especially for the nose cone and leading edge [1]. For instance, when a hypersonic vehicle cruises with Mach 20 at 47km, the surface temperature may exceed 4000K [2]. The extremely high temperature causes degradation in performance of material, and the structural strength will be affected. Therefore, it is very important to develop effective thermal protection systems to ensure the normal flight of hypersonic vehicles.

Transpiration cooling has been recognized as one of the most effective thermal protection methods for hypersonic vehicles exposed to extremely high heat flux. The coolant absorbs a lot of heat due to the large specific surface area of the porous matrix, and at the same time, the coolant flowing out of the porous matrix forms a continuous, even and stable film on the surface to separate hot flow and structure [3]. And it has been validated that the transpiration cooling with liquid coolant is more effective than gaseous coolant because of the huge latent heat of vaporization [4].

There has been a lot of researches on the transpiration cooling with liquid coolant, including experimental investigation and numerical simulation. In the experimental aspect, a series of investigations on the transpiration cooling with liquid coolant have been carried out by Wang et al. and Zhao et al [5,6]. under subsonic and supersonic conditions. These experiments show the high cooling efficiency of transpiration cooling with liquid coolant. In the numerical simulation aspect, two-phase mixture model (TPMM) and separated phase model(SPM) under local thermal equilibrium(LTE) assumption and non-equilibrium(LTNE) condition have been used to study transpiration cooling with
liquid coolant [7–10]. In these researches, purified water is used as the coolant because of the huge latent heat of vaporization, high density and low price. But almost in all the existing numerical simulations, thermodynamic properties of water and steam are simplified to constant. Actually in the cooling process, when coolant flows from the tank at ordinary temperature to the surface of the vehicle at high temperature, coolant temperature increases and pressure decreases, which will lead to the change of thermodynamic properties. Therefore, it is meaningful to do further researches on the transpiration cooling with liquid water considering the real properties of water.

To calculate the thermodynamic properties of water and steam, IAPWS-IF97 that published by the International Association for the Properties of Water and Steam (IAPWS) is applied. It provides internationally accepted formulations for the properties of light and heavy steam, water and selected aqueous solutions for scientific and industrial applications in 1997 [11].

In this paper, numerical simulations on the two-phase flow of transpiration cooling with liquid water are carried out in real property model and constant property model. The two-phase flow characteristics and the effect of coolant mass flow rate are also discussed and compared.

2. Model

2.1. Physical Model

The physical model considered in this study is an infinite porous plate with a thickness of $L$, which is sketched in Figure 1. The upper side of the plate is exposed in hot main flow, where the heat flux ($Q$ in the figure) boundary condition is exerted. Liquid coolant is injected into the pores from the bottom of the plate at temperature $T_c$ with mass flow rate $m_c$. The coolant flows through the plate and absorbs heat from porous matrix. With enough heat exchange, the plate can be classified into three regions in the vertical direction, i.e. liquid region, two-phase region and vapor region. If the heat flux is low, the vapor region and even two-phase region will disappear.

![Figure 1. Physical model of transpiration cooling with liquid coolant](image)

2.2. Mathematical Model

Based on TPMM-LTNE model, the coolant flow and heat exchange are described by the same conservation equations and constitutive relationships in all regions, which are listed in Table 1 [7,8].

In mass and momentum equations, the two-phase flow is described by a mass-average mixture velocity. Fluid and solid energy equations are solved separately, and heat exchange between fluid and solid within the porous matrix is considered. And in constant property model, Fluid energy equation is simplified.

| Table 1. Model equations |
|--------------------------|
| **Variable property (IAPWS-IF97) model** | **Constant property model** |
| Mass | $\nabla \cdot (\rho \vec{u}) = 0$ |
| Momentum | $\vec{u} = -\frac{1}{\mu} (\nabla p - \rho_k \vec{g})$ |
\[ \nabla \cdot (\gamma \vec{u} H_2) + 2\rho \vec{u} \nabla H_{\text{sat}} = \nabla \left( k_{\text{eff}} \nabla T \right) + \nabla \left[ f(s) \frac{K \Delta p h_{fg}}{v_v} g \right] \]

Fluid energy

\[ \dot{J} = -D(s) \nabla s + \frac{v_v}{f(s) K h_{fg} j} \]

Solid energy

\[ \nabla \cdot \left( \frac{(\rho_v / \rho_l)(1 - s)}{2 h_v \text{sat} - h_{l \text{sat}} s + (\rho_v h_{v \text{sat}} / \rho_l)(1 - s)} \right) = Q_{sf} \]

Constitutive relationships

\[ \nu(s) = \frac{k_{\text{eff}}(s)}{\nu_l} + \frac{k_{\text{rv}}(s)}{\nu_v} \]

\[ \lambda_l(s) = \frac{\nu(s) k_{\text{rl}}}{\nu_l} \]

\[ \lambda_v(s) = \frac{\nu(s) k_{\text{rv}}}{\nu_v} \]

\[ f(s) = k_{\text{rv}}(s) \lambda_l(s) \]

\[ p_c = \left( \frac{\epsilon}{K} \right)^{1/2} \sigma(s) \]

Capillary pressure

\[ f(s) = 1.417(1 - s) - 2.120(1 - s)^2 + 1.263(1 - s)^3 \]

2.3. Thermodynamic Properties Formulation

In the numerical simulations of transpiration cooling with phase change, thermophysical properties of coolant are obtained by two different models, i.e. the variable property model by IAPWS-IF97 formulation, and the constant property model to set properties as those of saturated water and saturated steam at 101325Pa. Different from the constant property model, properties in the variable property model are the functions of temperature and pressure and calculated by IAPWS-IF97. The IAPWS-IF97 consists of a set of equations which cover the following valid ranges of temperature and pressure:

\[ 273.15 \text{ K} \leq T \leq 1073.15 \text{ K} \quad p \leq 100 \text{ MPa} \]

\[ 1073.15 \text{ K} \leq T \leq 2273.15 \text{ K} \quad p \leq 10 \text{ MPa} \]

Specifically, the variations of some main parameters with temperature and pressure are displayed in Figure 2.

![Figure 2. Properties of water and steam at pressure between 0.1 to 0.3MPa, temperature between 293.15 to 800K](a) (b) (c)

3. Numerical Solution

The above mathematical models are solved by limited volume method. The convergence of the solving process is achieved when the residual error drops below \( 1 \times 10^{-8} \). For one-dimensional problem, the computational domain is divided into uniform control volumes. The grid independency is verified by using three meshes with grid numbers of 800, 2500 and 10000. The difference between the results is not significant. Finally, the mesh with a grid number of 2500 is chosen. The detailed solution procedure and algorithmic process are given as follows:

1) Calculate initial corrected enthalpy distribution from temperature and pressure.
2) Solve mass and momentum equation.
3) Update properties dependent pressure.
4) Solve solid and fluid energy equation.
5) Update properties.
6) Repeat from step 2 until convergence.

4. Results and Discussions
In all the simulations, the pressure and heat flux applied on the hot side of the plate is $101325 \text{Pa}$ and $2 \times 10^6 \text{W/m}^2$, respectively. The porous plate has a thickness of 10cm and a thermal conductivity of $20 \text{W/(m} \cdot \text{K)}$. The permeability of the porous matrix is $1 \times 10^{-12} \text{m}^2$ when the porosity is 0.3 and average sphere diameter is $5 \times 10^{-4} \text{m}$.

4.1. Distributions of Temperature and Pressure
When the coolant mass flow rate is set as $0.5 \text{ kg}/(\text{m}^2 \cdot \text{s})$, the transpiration cooling performances are investigated by the IAPWS-IF97 model and constant property model respectively. Figure 3 shows the fluid temperature and pressure distributions in the porous plate.

**Figure 3.** Temperature (a) and pressure (b) distributions in two property models

Two models show the same change trend of temperature in liquid and vapor regions, i.e. temperature increases slightly in liquid region and increases rapidly in vapor region. But in two-phase region, the fluid temperature distributions obtained by the models are different, including the following interesting phenomena: 1) compared to the constant property model, the fluid temperature in the variable property model is much higher and the phase change location is closer to the hot side; 2) a decrease of fluid temperature along the flow direction can be observed in the two-phase region, as shown in the subfigure (a) of Figure 3. These differences can be explained by the pressure distribution in 3(b). Due to the kinematic viscosity varying with temperature and pressure, the pressure drops obtained by the variable property model is higher than that obtained by constant property model, which indicates that the driving force of coolant required is much underestimated using the traditional constant property model. A higher pressure corresponds to a higher phase change temperature. In IAPWS-IF97 model, the variations of coolant thermal properties with pressure are considered, so the phase change temperature predicted decreases along the coolant flow direction. Furthermore, higher phase change temperature means more latent heat is absorbed when phase change happens, therefore a thicker liquid region and thinner two-phase region are obtained using the IAPWS-IF97 model.

4.2. Influence of Coolant Mass Flow Rate on Temperature and Pressure Distributions
In this section, the temperature and pressure distributions are calculated using the two models when coolant flow rate varies from $0.5 \text{ kg}/(\text{m}^2 \cdot \text{s})$ to $1.0 \text{ kg}/(\text{m}^2 \cdot \text{s})$. And the results are shown in Figure 4.
For the same heat flux, the specific enthalpy becomes smaller as the coolant injected increases, and thereby the coolant temperature at the hot side drops. When the coolant injected further increases, the specific enthalpy is lower than saturated enthalpy, and the temperature keeps at phase change temperature. The difference between the temperatures obtained by two models is small because the change of specific heat is tiny in the range of temperature and pressure considered here. In both models, the driving pressure decreases firstly then increases with the increase of coolant mass flow rate due to the reduction of vapor region and two-phase region. The driving pressure obtained by IAPWS-IF97 model is always higher than that by constant property model. The maximum difference occurs at the mass flow rate of 0.5 kg/(m²·s), where driving pressure by IAPWS-IF97 model is 57% higher.

**Figure 4.** The variation of temperature at hot side (a) and driving pressure (b) with coolant mass flow rate

### 4.3. Influence of Coolant Mass Flow Rate on Phase Change Location
Pressure loss of coolant occurs mainly in two-phase and vapor regions, which greatly affects the coolant distribution. The phase change locations at different mass flow rates are investigated here. Figure 5 shows the starting and ending positions of two-phase region. As coolant mass flow rate increases, the starting location of two-phase region moves obviously towards the hot side of the plate. It is noted that the starting position of two-phase region obtained by IAPWS-IF97 model is always behind that by constant property model. That’s because the phase change temperature is higher in IAPWS-IF97 model due to a higher pressure. When mass flow rate increases up to 0.77 kg/(m²·s), the vapor region disappears, leading to a step change of starting location of two-phase region there. The ending locations predicted by the two models both move towards the hot side with the coolant mass flow rate and maintain at the hot side when vapor region disappears. Thickness of two-phase region means the ending location of phase change minuses starting location. It can be concluded that the two-phase region predicted by IAPWS-IF97 model is smaller than that by constant property model.

**Figure 5.** Starting (a) and ending (b) locations of two-phase region with coolant mass flow rate
5. Conclusion
In this paper, the real thermal properties of water and steam are considered in the simulation of transpiration cooling with liquid water as coolant. Then the numerical investigations on transpiration cooling using the variable property model based on IAPWS-IF97 formulation and the constant property model were carried out. Through the analysis and comparison, the following conclusions can be drawn:

1. Comparing to the constant property model, the phase-change temperature predicted by variable model is higher and decreases in the two-phase region due to the pressure change there, and the driving force of coolant estimated by IAPWS-IF97 model is much higher and shows more significant variation trend along the flow direction.

2. When the coolant mass flow rate increases, coolant temperature at the hot side drops and the driving force firstly decreases and then increases. Comparing the results from the two models, the temperatures at hot side are always very close, but the driving forces show remarkable difference.

3. With the increase of coolant mass flow rate, the phase change location move toward the hot side and two-phase region decreases, but the starting and ending positions predicted by the variable property model have a lag in comparison with constant property model.

Reference
[1] T. Reimer, M. Kuhn, A. Gülhan, B. Esser, M. Sippel and A. Foreest 2011 Transpiration Cooling Tests of Porous CMC in Hypersonic Flow 17th AIAA International Space Planes and Hypersonic Systems and Technologies Conference (San Francisco, California: American Institute of Aeronautics and Astronautics)
[2] M. Sippel 2007 Introducing the SpaceLiner Vision 7th International Symposium on Launcher Technologies (Barcelona, Spain) p 10
[3] A. Laganelli 1970 A comparison between film cooling and transpiration cooling systems in high speed flow 8th Aerospace Sciences Meeting (West Germany: American Institute of Aeronautics and Astronautics)
[4] J. Bellettre, F. Bataille, A. Lallemand and H. Y. Andoh 2005 Studies of the transpiration cooling through a sintered stainless steel plate Exp. Heat Transf. 18 33–44
[5] J. H. Wang, L. J. Zhao, X. C. Wang, J. Ma and J. Lin 2014 An experimental investigation on transpiration cooling of wedge shaped nose cone with liquid coolant Int. J. Heat Mass Transf. 75 442–9
[6] L. J. Zhao, J. H. Wang, J. Ma, J. Lin, J. L. Peng, D. J. Qu and L. Z. Chen 2014 An experimental investigation on transpiration cooling under supersonic condition using a nose cone model Int. J. Therm. Sci. 84 207–13
[7] C. Y. Wang and C. Beckerma 1993 A two-phase mixture model of liquid-gas flow and heat transfer in capillary porous media-l. Formulation Heat Mass Transf. 36 2747–58
[8] J. X. Shi and J. H. Wang 2011 A Numerical Investigation of Transpiration Cooling with Liquid Coolant Phase Change Transp. Porous Media 87 703–16
[9] F. He, J. H. Wang, L. C. Xu and X. C. Wang 2013 Modeling and simulation of transpiration cooling with phase change Appl. Therm. Eng. 58 173–80
[10] W. J. Dong and J. H. Wang 2018 A New Model and its Application to Investigate Transpiration Cooling with Liquid Coolant Phase Change Transp. Porous Media 122 575–93
[11] W. Wagner, J. R. Cooper, A. Dittmann, J. Kijima, H.-J. Kretzschmar, A. Kruse, R. Mareš, K. Oguchi, H. Sato, I. Stöcker, O. Šifner, Y. Takaishi, I. Tanishita, J. Trübenbach and Th. Willkommen 2000 The IAPWS Industrial Formulation 1997 for the Thermodynamic Properties of Water and Steam J. Eng. Gas Turbines Power 122 150