7-6-2016

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Proceedings of the 6th International Conference on Porous Media and its Applications in Science and Engineering
ICPM6
July 3-8, 2016, Waikoloa, Hawaii

NUMERICAL INVESTIGATION OF THE PHASE CHANGE IN TRANSPIRATION COOLING WITH THE VOF METHOD

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ABSTRACT
Transpiration cooling with phase change is numerically investigated in the present work. A model coupling the heat transfer and flows in porous media and the hot-gas main flow zone is developed to simulate the above phenomenon. With the assumptions of local thermal equilibrium and the clear interface between the liquid and gas phases, the volume-of-fluid (VOF) method can be used in this model. This clear interface is tracked by the VOF method and divides the computed zone into the liquid phase region and the gas phase region. On this liquid-gas interface, a phase change model should be applied in this modeling. Thus, the modified Lee's phase change model is proposed and used to insure the phase change flow in porous media can be converged. A two-dimensional case study is introduced to show the characteristics of transpiration cooling with phase change. The effects of preferred path and solid matrix thermal conductivity are analyzed. The effect of preferred path contributes to heat transfer deterioration in order that the porous media temperatures become very high at the points far from the preferred paths

INTRODUCTION
Transpiration cooling is one of the most advanced cooling technique potentially applied in the high heat flux removal for gas turbine blades, combustors, rocket nozzles and etc. If a solid component is exposed to a hot gas or thermal radiation without any protection, it is possibly heated at a very high temperature and burnt out. In this case, transpiration cooling can thermally protect the solid component with applying a porous medium as its outer wall and injecting a coolant from the inner side of this permeable wall. The coolant flows through the porous medium and forms a film covering the porous surface. As a result, the coolant absorbs heat when passing through the porous medium and reduces the heat flux received by the porous surface as a fluid film.

The coolant for transpiration cooling can be gaseous or liquid. Transpiration cooling with a liquid coolant is more efficient than that with a gaseous coolant because the liquid coolant can absorb a lot of latent heat if phase change occurs. In that case, transpiration cooling with a liquid coolant can also been called transpiration cooling with phase change, as shown in Figure 1.

In current literatures, many experimental, numerical and analytical investigations have been reported on transpiration cooling with a gaseous coolant [1-4], while the investigations of transpiration cooling with phase change focused more on experiments [5, 6] and one dimensional analyses [7]. The state of the art of numerical investigations of transpiration cooling with phase change are limited due to the complexity of phase change in porous media. Shi and Wang [8] proposed a local thermal non-equilibrium model for the heat transfer in porous media and solved the one-dimensional numerical result for the transpiration cooling with phase change. He et al. [6] solved a local thermal equilibrium model based on Shi and Wang [8] and compared the result to experiment data. Their numerical result was also solved for the one-dimensional porous media. However, no investigation has focused on the coupling of heat and mass transfer between the hot-gas main flow and the phase change of coolant in porous media for transpiration cooling with phase change.

The present work introduces a new model coupling the heat transfer and fluid flow for the hot-gas main flow and the phase-change coolant flow in porous media. A two-dimensional case study is introduced to show the characteristics of transpiration cooling with phase change.

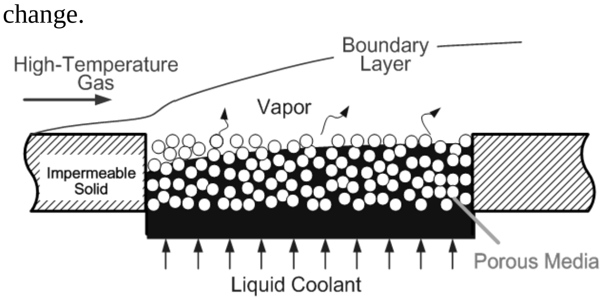

Figure 1: Schematic diagram for transpiration cooling with phase change

NOMENCLATURE

c_p	=	specific heat capacity
c_f	=	skin friction coefficient
D	=	diffusivity
F	=	Forchheimer term factor
h_{fg}	=	latent heat
$\mathbf{I}$	=	unit tensor
k	=	turbulence kinetic energy
K	=	permeability
$\dot{m}$	=	phase change rate per unit volume
L	=	length of the porous medium
M	=	molecular weight
$\mathbf{n}$	=	unit normal vector of porous media surface
p	=	pressure
$\mathrm{Pr_t}$	=	turbulence Prantle number
$\mathbf{S}$	=	strain tensor
$\mathrm{Sc_t}$	=	turbulence Schmidt number
T	=	temperature
$\mathbf{u}$	=	velocity vector
x	=	x coordinate
y	=	y coordinate
Y	=	species mass fraction

Greek Symbols

ε	=	void fraction
λ	=	thermal conductivity
μ	=	viscosity
ρ	=	density
$\boldsymbol{\tau}$	=	shear stress
φ	=	volume fraction of fluid
ω	=	turbulence specific dissipation rate

Subscripts

a	=	air
eff	=	effective
g	=	gas phase
l	=	liquid phase
m	=	mixture of liquid and gas phases
s	=	solid phase
sat	=	saturation
t	=	turbulence
v	=	water vapor
w	=	wall
τ	=	shear
$\perp$	=	normal
∞	=	at infinity

1 Modeling and formulation

Figure 2 shows the geometry and the boundary conditions for a case of transpiration cooling with phase change. Three zones are considered: main flow zone, porous media zone and the reservoir zone. A hot air main flow blew the surface of a porous medium and the liquid coolant was injected into the reservoir.

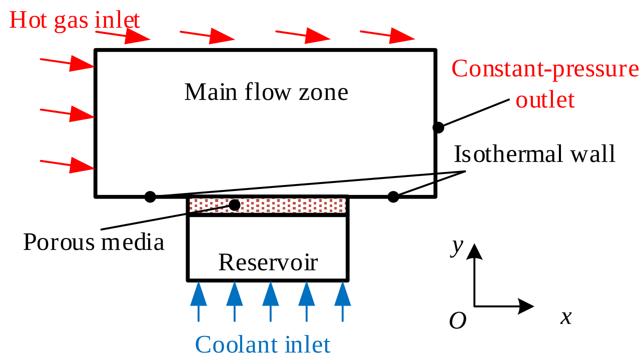

Figure 2: Schematic diagram of the case

In the present work, the VOF method is used to capture the phase-change interface between the liquid and gas phases. The heat transfer and fluid flow equations are solved coupling the main flow and the flow in porous media. The following assumptions are adopted to simplify this problem:

1) Capillary can be neglected in porous media.
2) The thickness of the phase change region is small enough, which assumes that the liquid coolant evaporates in a very short distance. The VOF method is reasonable only under this assumption.
3) The temperature of the solid phase locally equals to that of the fluid phase in porous media. This is also called local thermal equilibrium assumption.
4) The saturation temperature of coolant is constant.

Based on these assumptions, the following governing equations are obtained simultaneously for the three zones in Figure 2:

Volume fraction equations:

$$\nabla \cdot (\varphi_l \rho_l \mathbf{u}) = -\dot{m}_{l-v} \tag{1}$$

$$\varphi_l + \varphi_g = 1 \tag{2}$$

where the phase change rate per unit volume can be modeled by a modified Lee's model [9]:

$$\dot{m}_{l-v} = \begin{cases} C\lambda_{eff}\varphi_l\rho_l \dfrac{T - T_{sat}}{T_{sat}}, T > T_{sat} \\ -C\lambda_{eff}\varphi_v Y_v \rho_v \dfrac{T - T_{sat}}{T_{sat}}, T < T_{sat} \end{cases} \tag{3}$$

Continuity equation:

$$\nabla \cdot (\rho_m \mathbf{u}) = 0 \tag{4}$$

Momentum equation:

$$\nabla \cdot \left(\frac{\rho_m \mathbf{uu}}{\varepsilon^2} \right) = -\nabla p + \nabla \cdot \boldsymbol{\tau}_{eff} + \rho_m \mathbf{g} - \left(\frac{\mu_m}{K} + \frac{\rho_m F}{\sqrt{K}} |\mathbf{u}| \right) \mathbf{u} \tag{5}$$

where

$$\boldsymbol{\tau}_{eff} = (\mu_m + \mu_t)\left(2\mathbf{S} - \frac{2}{3}\nabla\cdot\mathbf{uI}\right) - \frac{2}{3}\rho_m k\mathbf{I} \tag{6}$$

$$\mu_t = \alpha^* \frac{\rho_m k}{\omega},\ \mathbf{S} = \frac{\nabla\mathbf{u} + \mathbf{u}\nabla}{2} \tag{7}$$

k-ω turbulence equations:

In the main flow zone, the following equation is occupied for k and ω:

$$\nabla\cdot(\rho_m\mathbf{u}k) = \nabla\cdot\left(\left(\mu_m + \frac{\mu_t}{\sigma_k}\right)\nabla k\right) + 2\mu_t\mathbf{S}:\mathbf{S} - \rho_m\beta_k f_{\beta k} k\omega \tag{8}$$

$$\nabla\cdot(\rho_m\mathbf{u}\omega) = \nabla\cdot\left(\left(\mu_m + \frac{\mu_t}{\sigma_\omega}\right)\nabla\omega\right) + \frac{2\alpha\omega\mu_t}{k}\mathbf{S}:\mathbf{S} - \rho_m\beta_\omega f_{\beta\omega}\omega^2 \tag{9}$$

while $k \equiv 0$ inside the porous media zone and the reservoir zone.

The boundary condition for ω on the porous surface [10]:

$$\omega = \frac{\rho_m u_\tau^2}{\mu_m} \frac{25}{\dfrac{u_\perp}{u_\tau}\left(1 + 5\dfrac{u_\perp}{u_\tau}\right)} \tag{10}$$

where

$$u_\tau = \sqrt{\frac{\boldsymbol{\tau}_{eff}\cdot\mathbf{n}}{\rho}},\ u_\perp = \mathbf{u}\cdot\mathbf{n} \tag{11}$$

Energy equation:

$$\nabla\cdot\left(\mathbf{u}\left(\rho_m c_{pm} T + \rho_m\frac{\mathbf{u}^2}{2} + p\right)\right) = \nabla\cdot(\lambda_{eff}\nabla T + \boldsymbol{\tau}_{eff}\cdot\mathbf{u}) - \dot{m}_{l-v} h_{fg} \tag{12}$$

where

$$\lambda_{eff} = \varepsilon\lambda_m + (1-\varepsilon)\lambda_s + \frac{c_{pm}\mu_t}{\mathrm{Pr}_t} \tag{13}$$

Species transport equations in the gas phase:

$$\nabla\cdot(\rho_g\varphi_g\mathbf{u}Y_v) = \nabla\cdot\left(\varphi_g\left(\varepsilon\rho_g D_{a-v} + \frac{\mu_t}{\mathrm{Sc}_t}\right)\nabla Y_v\right) + \dot{m}_{l-v} \tag{14}$$

$$Y_a + Y_v = 1 \tag{15}$$

Material properties:

$$\begin{aligned} \rho_m &= \varphi_l\rho_l + \varphi_g\frac{M_g p}{RT} \\ \mu_m &= \varphi_l\mu_l + \varphi_g(Y_v\mu_v + Y_a\mu_a) \\ \lambda_m &= \varphi_l\lambda_l + \varphi_g(Y_v\lambda_v + Y_a\lambda_a) \\ c_{pm} &= \varphi_l c_{pl} + \varphi_g(Y_v c_{pv} + Y_a c_{pa}) \\ M_g &= \frac{M_v M_a}{M_a Y_v + M_v(1-Y_v)} \end{aligned} \tag{16}$$

The equations are discretized on a two-dimension mesh using the finite volume method and solved with SIMPLE algorithm.

To validate the numerical model and the code, the numerical result based on the present code was compared to the experimental data of Dahmen et al. [3]. Dahmen et al. investigated the transpiration cooling with a gaseous coolant and measured the skin friction on the porous surface. The comparison is shown in Figure 3. The porous media was distributed from 0 to L on the axial direction. Thus, the skin friction coefficient decreases within this region with a secondary blow. The numerical result using the present model has a good agreement with the experimental data.

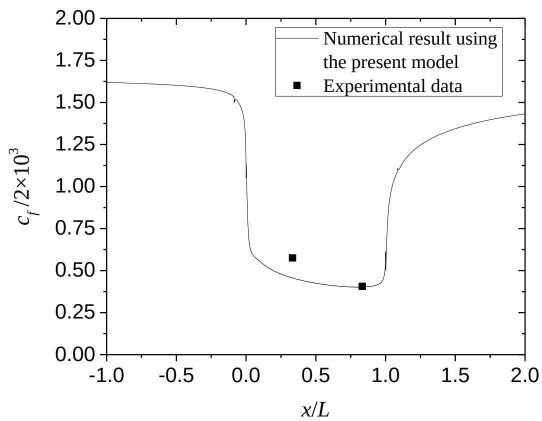

Figure 3: Comparison between the numerical result and experimental data [3]

2 Case study and discussion

In the present case, a hot air main flow blew the surface of a porous medium and two isothermal walls, as shown in Figure 2. The flow conditions for the hot gas were as follow: static temperature T_∞ = 2400 K, density ρ_∞= 0.767 kg/m^3, Mach number = 3, and the angle of attack = 2°. The porous medium was 150 mm long and 10 mm thick which was sintered with the porosity = 0.1. Liquid water at 300 K was injected from the bottom of the reservoir as the coolant. The blow ratio of the injected coolant to the main flow was 1.65%.

Two cases are solved for different conductivities of porous media solid matrix, i.e., k_s=8 and 18 W/m/K. The saturation temperature of water is set to be constant at 373 K. The heat, momentum and mass transfer with turbulence and multiphase flow are solved using the present model for the simultaneously for the main flow, porous media and reservoir zones. We mainly focus on the heat transfer and fluid flow inside the porous medium in the following discussion.

Figure 4 shows the typical temperature distribution of transpiration cooling with phase change under supersonic conditions. A shock wave was formed by the leading edge of the isothermal wall, the static temperature increased behind the shock wave.

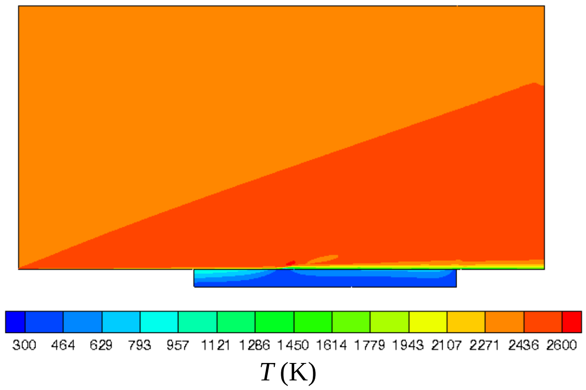

Figure 4: Static temperature for k_s= 18 W/m/K

2.1 Preferred path effect

Figures 5-8 show the gas volume fraction, mass flux magnitude, pressure and temperature inside the porous medium and reflect the non-linear effects of transpiration cooling with phase change in porous media.

As shown in Figure 5, the thickness of the vapor region is not monotone decreasing along the x axis but first decreasing to zero, then increasing and finally decreasing to zero. Thus, two preferred paths are formed for the injected liquid coolant: one is near x/L=0.3~0.4, and the other is near x/L=1. As shown in Figure 6, the mass flux is much larger at the preferred paths which the coolant prefers to go through. That can be explained with Darcy's law:

$$\mathbf{u} = -\frac{K}{\mu}\nabla p \tag{17}$$

Since the mass flux magnitude is constant along an arbitrary stream tube for a fully evaporated flow, the pressure gradients are significantly different between the vapor region and the liquid region for the same mass flux:

$$\frac{|\nabla p|_v}{|\nabla p|_l} = \frac{\mu_v \rho_l}{\mu_l \rho_v} = \frac{\nu_v}{\nu_l} \tag{18}$$

The ratio of ν_v/ν_l can be up to 30~200 for water. That means the flow resistance in the vapor region is much larger than that in the liquid region for the coolant to flow through a porous medium. Thus, the vapor region blocks the porous media and the coolant prefers to sweep around the thick vapor region. As a result, less coolant can evaporate to cool down the thick vapor region, the vapor region becomes thicker. This kind of positive feedback contributes to heat transfer deterioration in order that the porous media temperatures become very high at the points far from the preferred paths, as shown in Figure 7.

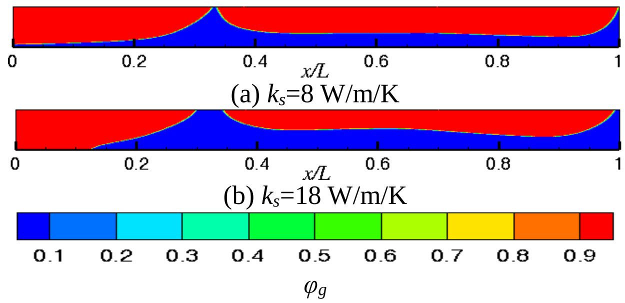

Figure 5: Gas volume fraction in the porous medium

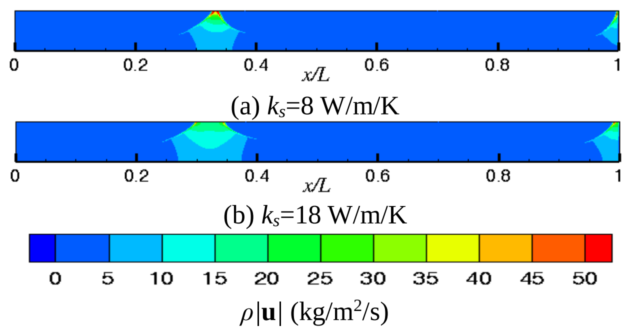

Figure 6: Mass flux magnitude in the porous medium

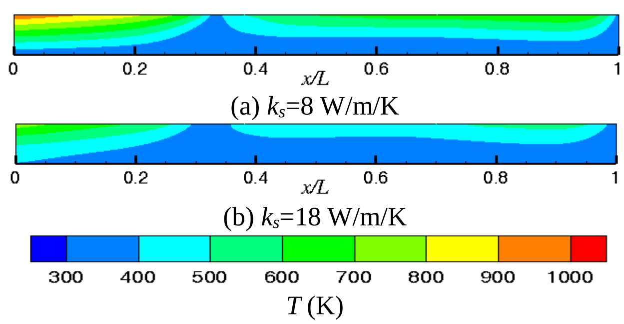

Figure 7: Temperature distribution in the porous medium

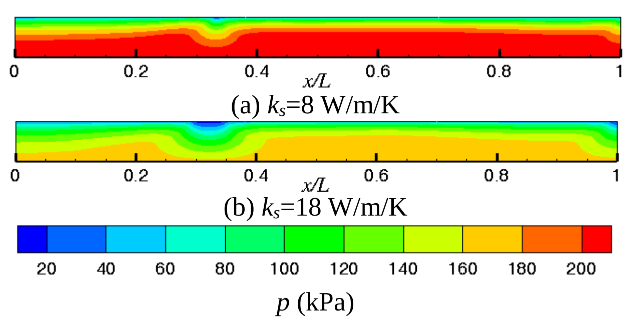

Figure 8: Pressure distribution in the porous medium

2.2 Effects of solid matrix thermal conductivity

As shown in Figures 5-8, the thermal conductivity of porous media solid matrix has large effects on the heat transfer and flow in this case. A lower k_s gives thinner vapor region at x/L<0.3 because heat is more difficult to conduct to the bottom of porous media (Figure 5). The lower k_s also gives more narrow preferred paths. Thus, the mass flux magnitude is larger when the liquid coolant flows through the preferred paths (Figure 6) and the pressure drop is larger (Figure 8).

CONCLUSIONS

In the present work, a model coupling the heat transfer and fluid flow for the hot-gas main flow and the phase-change coolant flow in porous media is developed. A two-dimensional case study is introduced to show the characteristics of transpiration cooling with phase change. The effects of preferred path and solid matrix thermal conductivity are analyzed. The effect of preferred path contributes to heat transfer deterioration in order that the porous media temperatures become very high at the points far from the preferred paths

ACKNOWLEDGEMENT

The authors would like to acknowledge support from the National Natural Science Foundation of China (No. 51506109, No.51321002 and No.51276094).

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