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An experimentally verified numerical model of finned heat pipes in crossflow

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ABSTRACT

A numerical model is developed to predict heat rates associated with a heat pipe whose finned condenser section is subjected to external forced convection. Multiphase, conjugate heat transfer inside the heat pipe is predicted using a 2-dimensional model, while fluid flow and convection heat transfer within the fin array is described with a coupled 3-dimensional shear stress transport (SST) model. Predictions of local temperatures and overall heat rates are verified experimentally. The SST model is also validated with 3-dimensional direct simulations that show highly time-dependent, 3-dimensional phenomena in the fin array. A previously unreported phenomenon, localized depression of temperatures in the heat pipe wall, is presented and parametric simulations reveal the sensitivity of system performance to the number of fins and the air velocity in the fin array.

1. Introduction

As is well known, heat pipes (HPs) can pose thermal resistances that are orders of magnitude smaller than those associated with conduction within high thermal conductivity materials of similar dimension [1–3]. Recent reviews [4–13] have thoroughly described both HP applications and their principles of operation. Due to the remarkably small thermal resistances posed by HPs, the heat transfer capability of HP systems is often limited by, for example, convective resistances associated with external fluid flow about the HP condenser and/or evaporation sections. Hence, external fins or fin arrays are often employed to reduce the overall thermal resistances of such systems [14–22].

Numerical modeling has been used to analyze heat transfer processes both (*i*) within HPs [22–32] and (*ii*) external to HPs such as those equipped with exterior fin arrays [19–22]. Both approaches have employed simplifying assumptions. For example, interior modeling often employs accurate descriptions of the evaporation, condensation, and heat transfer processes within the HP itself, but is hampered by the specification of idealized external thermal boundary conditions at the HP evaporator and condensation sections. Alternatively, recently-reported simulations provide an accurate prediction of the external convective heat transfer processes, but typically treat heat transfer within the HP in a simplified manner [19–21]. Recent investigations describe advances that have been made by developing overall HP system models that include detailed descriptions of both internal and external heat transfer processes. However, these models have been limited to 2D systems [22,26–28].

For a finned HP subjected to 3D external forced convection, a unified and full 3D approach to solve both the internal and external heat transfer processes concurrently would be computationally expensive. Moreover, detailed 3D predictions of phenomena within the HP may not be needed in some cases due to the multiphase (evaporation and condensation) processes that are affiliated with relatively small thermal resistances. To the authors' knowledge, full 3D predictions of heat transfer within both the interior and exterior to a HP have not been reported in the literature.

The objective of this study is to develop and demonstrate a novel computational methodology to couple 2D internal and 3D external simulations for a common configuration; a vertical HP with isothermal conditions at its lower evaporator section, and unsteady 3D convective conditions external to its upper finned condenser section. The model is computationally less expensive than a full 3D simulation and, as will become evident, can satisfactorily replicate measurements of various heat transfer quantities.

2. Numerical model

As shown in Fig. 1a, an externally-finned HP of circular cross section and length $L_{hp} = L_e + L_a + L_c$ is oriented vertically, with an

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Nomenclature

A	area (m^2)
D	diameter (m)
F_1, F_2	SST blending functions
g	gravitational acceleration (m/s^2)
Gr	Grashof number
h	convection coefficient (W/m^2 K)
H	height (m)
k	thermal conductivity (W/m K)
L	length (m)
m	fin constant
N	number of fins
p	pressure (Pa)
P_κ	SST production term (kg/m s^3)
q''	heat flux (W/m^2)
Q	heat rate (W)
R	thermal resistance (K/W)
Re	Reynolds number
r, z, θ	coordinate directions
S	fin pitch (m), turbulent strain rate (s^{-1})
St	Strouhal number
T	temperature (°C)
t	time (s), thickness (m)
u	velocity (m/s)
V	average air velocity (m/s)
W	width (m)
x, y, z	coordinate directions

Greek symbols

α	thermal diffusivity (m^2/s)
α_1, β_1, β_2	SST model constants
β	thermal expansion coefficient (K^{-1})
θ	reduced temperature $T - T_\infty$ (°C), angular direction
κ	turbulent kinetic energy (m^2/s^2)
μ	dynamic viscosity (kg/m s)
ν	kinematic viscosity (m^2/s)
ρ	density (kg/m^3)
σ_κ, σ_ω, $\sigma_{\omega 2}$	SST model constants
ω	specific rate of turbulence dissipation (s^{-1})

Superscripts

–	average

Subscripts

0	reference
2, 2c	fin radius, corrected fin radius
2D, 3D	2- or 3-dimensional
a	adiabatic, air
ag	aerogel
Al	aluminum
ar	Airloy
c	condenser
ch	channel
conv	convection
d	downstream
e	evaporator
eff	effective
exp	experimental
f	fin
hp	heat pipe
hs	heat spreader
i	case index
min, max	minimum, maximum
SST	shear stress transport
tot	total
tp	thermal paste
ts	top fin surface
turb	turbulent
u	upstream
v	heat pipe vapor
w	heat pipe wall
wi	heat pipe wick
∞	inlet

evaporator section of length L_e positioned beneath the adiabatic and air-cooled condenser sections of lengths L_a and L_c, respectively. The condenser section is finned, with air flowing through the fin array. The finned HP is shown installed in a flow channel in a manner consistent with an experimental setup described later.

A novel overall model, to be presented, consists of two coupled sub-models. The first describes axisymmetric, multiphase heat transfer and fluid flow within the HP. The second describes conduction processes within the HP and the 3D single phase convection external to the HP condenser section. The two sub-models are coupled as will be described shortly. Predictions of the overall model are verified with experimental data, and the model is used to parametrically investigate the HP system performance and reveal previously-unobserved thermal phenomena within the HP.

2.1. HP sub-model

A HP sub-model is used to solve the 2D, axisymmetric equations that govern the transient evaporation and condensation processes within an internally-wicked HP, as well as the fluid flow and heat transfer (vapor phase advection, and conduction) within the HP. Details of the model, including the descriptive equations, are available in Sharifi et al. [26]. Due to their length, they will not be repeated here.

The computational domain for the HP sub-model (shown in detail in Fig. 1 of [26] and in Fig. 1a) is comprised of the HP wall, wick, and vapor regions. The region ($0 \leqslant z \leqslant L_{hp}$; $0 < r \leqslant r_v$) contains the vapor phase of the HP working fluid. Heat transfer in this region is governed by conservation of mass, r- and z-momentum, and energy, as laid out in Eqs. (1)–(8) of [26]. The wick ($r_v < r \leqslant r_w$) is a porous metal that is assumed to be saturated with the liquid

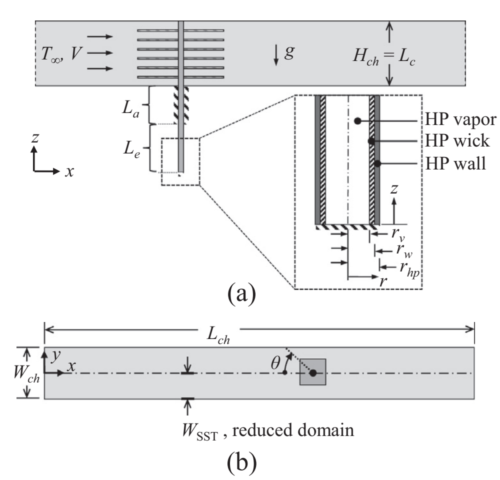

Fig. 1. Schematic of the physical system. (a) Test section, (b) flow channel. Two separate coordinate systems are shown.