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Thermal characteristics of a closed thermosyphon under various filling conditions

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ABSTRACT

A two-dimensional numerical model is developed to simulate the transient operation of a thermosyphon with various working fluid filling ratios. Conservation equations for mass, momentum, and thermal energy are solved using finite volume scheme to determine the hydrodynamic and thermal behavior of the thermosyphon. The heat transfer due to the liquid pool and liquid film are accounted for. The numerical model is validated through comparison with experimental data available in the literature. The model is capable of predicting the optimal filling ratio which corresponds to a condensate film extending from the condenser end cap to the evaporator end cap at steady-state for a given heat input. Overfilled and underfilled conditions for which the working fluid inventories are respectively greater than and less than the optimal case are also investigated. Simulation results show that the evaporator temperature of the underfilled thermosyphon rises dramatically due to dryout. The optimally-filled thermosyphon has the shortest response time and the lowest thermal resistance, however, a slight increase in the input power will cause breakdown of the condensate film. The overfilled thermosyphon poses a slightly slower thermal response and greater thermal resistance compared to the optimal condition. To ensure optimal and stable steady-state operation, an optimally-filled thermosyphon is recommended with a small amount of additional working fluid to prevent breakdown of the liquid film.

1. Introduction

A thermosyphon is a wickless, gravity-assisted heat pipe that contains a certain amount of working fluid for heat transfer. The working fluid is vaporized by the heat input at the evaporator. The vapor then rises through the adiabatic section of the device to the condenser, where it condenses and releases its latent heat of vaporization. The action of gravity then returns the condensate to the evaporator section. Internal phase change circulation of the working fluid serves as the mechanism of heat transfer in the thermosyphon. The thermosyphon is widely used in electronics cooling systems, solar photovoltaic cells, and energy recovery systems due to its simple structure, operation under environmentally-sound conditions, and high heat transfer capacity which results from its operation via phase change [1–4].

A number of experimental investigations pertaining to the thermal–hydraulic mechanism and the improvement of thermosyphon design and performance have been reported in recent years. Kiatsiriroat et al. [5] experimentally investigated the thermal performance enhancement of a thermosyphon using ethanol–water and triethylene glycol (TEG)–water as the working fluid. The boiling correlation of Rohsenow [6] and the condensation equation of Nusselt [7] were modified to predict the heat transfer inside the thermosyphon. It was found that the equation developed by Faghri [1] can be used to predict the critical heat flux due to the flooding limit. Noie [8] experimentally analyzed the effects of the input heat transfer rate, the working fluid filling ratio, and the evaporator length on the heat transfer performance in a two-phase, closed thermosyphon. The experimental boiling heat transfer coefficients were compared with existing correlations and the optimum filling ratio for the operation of a thermosyphon was analyzed. Park et al. [9] tested a two-phase, closed thermosyphon with various filling ratios. The experimental data for the smooth surface generally exhibited agreement with the correlation reported by Rohsenow [6] for the smooth surface. The heat transfer capacity was determined by the dryout limitation for a small filling ratio. It was determined by the flooding limitation for a large filling ratio.

The heat transfer capacity of a thermosyphon is subject to a number of heat transfer limits which are critical to thermosyphon design and operation. Many mathematical models have been developed to analyze the flooding and dryout heat transfer limits in a thermosyphon. Zuo and Gunnerson [10] presented a numerical model to predict the performance of inclined thermosyphons. Liquid–vapor interfacial shear stress and the effects of working fluid inventory at various inclination angles were included in the model. They analyzed the dryout and flooding limiting mechanisms and demonstrated that the model is capable of predicting

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Nomenclature

Ar	Archimid number, $\mathrm{Ar} = \left(\frac{g D_i^3}{\nu_l^2}\right)^2 \left(1 - \frac{\rho_v}{\rho_l}\right)$
Bo	Bond number, $\mathrm{Bo} = \frac{D_i}{l_m}$
c	specific heat (J/kg K)
C_m	modification coefficient (Eq. (30))
D	diameter (m)
Fr	Froude number, $\mathrm{Fr} = \left(\frac{q''_{e,i}}{\rho_l h_{fg}}\right)^2 \left[\frac{\rho_l}{D_i g (\rho_l - \rho_v)}\right]$
g	gravitational acceleration (m/s^2)
h	heat transfer coefficient (W/m^2 K)
h_{fg}	heat of vaporization (J/kg)
l_m	bubble length scale (m), $l_m = \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}}$
k	thermal conductivity (W/m K)
L	length in z-direction (m)
M	mass (kg)
M_{f+v}	total mass of liquid film and vapor (kg)
$\dot{m}'''$	mass source term (kg/m^3)
P	pressure (Pa)
Pr	Prandtl number, $\mathrm{Pr} = \frac{\nu}{\alpha}$
q	heat transfer rate
q''	heat flux (W/m^2)
q'''	thermal energy source term (W/m^3)
R_{th}	thermal resistance (K/W)
r	radial coordinate (m)
R_f	liquid film thermal resistance (m^2 K/W)
R_g	gas constant (J/kg K)
R_i	inner radius of thermosyphon shell (m)
R_o	outer radius of thermosyphon shell (m)
R_v	radius of vapor space (m)
Ra	Rayleigh number, $\mathrm{Ra} = \frac{g \beta q''_{e,i} L_p^4}{\alpha_l \nu_l k_l}$
Re	Reynolds number
T	temperature (K, °C)
$\overline{T}$	average temperature (K, °C)
t	time (s)
v	radial velocity (m/s)
w	axial velocity (m/s)
$\bar{w}_v$	average axial vapor velocity (m/s)
y	coordinate direction (Fig. 1(b))
z	axial coordinate (m)

Greek symbols

α	diffusion coefficient in heat transfer equation (m^2/s)
β	thermal expansion coefficient (1/K)
Δ	related to computational cell length in a particular direction
δ	film thickness (m)
μ	dynamic viscosity (Pa s)
ν	kinematic viscosity (m^2/s)
ρ	density (kg/m^3)
σ	surface tension (N/m)

Subscripts

a	adiabatic section
atm	atmospheric
ave	average
c	condenser section
cl	centerline
CV	control volume
e	evaporator section
f	liquid film
i	inner
l	liquid
NB	nucleate boiling
o	outer
p	pool
s	saturation
SC	single-phase convection
t	total
TC	two-phase convection
v	vapor
w	wall
δ	related to liquid film-vapor interface
∞	ambient

Superscripts

n	related to previous time step
$n+1$	related to current time step

the performance of an inclined thermosyphon. El-Genk and Saber [11] developed a one-dimensional, steady-state model to determine the operation of closed, two-phase thermosyphons in terms of dimensions, type, vapor temperature of working fluid, and power throughput. The thermosyphon operation-envelope was an enclosure with three critical boundaries related to dryout, boiling, and flooding limits. The calculations showed that an increase in the thermosyphon diameter, evaporator length, or vapor temperature expanded the operation-envelope, whereas an increase in the length of either the condenser or the adiabatic section only slightly changed the envelope's upper and lower boundaries.

The thermal–hydraulic behavior in a thermosyphon is determined by the device's key parameters, such as its input heat, filling ratio of the working fluid, geometry, orientation, and the thermophysical properties of the working fluid. Several numerical studies have analyzed the effect of these design parameters on the performance of thermosyphons. Harley and Faghri [12] presented a transient, two-dimensional model of a thermosyphon which accounted for the conjugate heat transfer through the wall and the falling condensate film. The falling condensate film was modeled with a quasi-steady, Nusselt-type analysis. The vapor flow and the coupling between the vapor and wall were modeled using the SIMPLE scheme [13] as described by Cao and Faghri [14]. Pan [15] presented a condensation model for a two-phase closed thermosyphon by considering the interfacial shear stress due to mass transfer and interfacial velocity. He concluded that the relative velocity ratio and the ratio of the interfacial shear in presence of mass transfer (due to evaporation/condensation) to the interfacial shear without mass transfer greatly affect the condensation heat transfer in the thermosyphon. Jiao et al. [16] developed a model to investigate the effect of the filling ratio on the distribution of the liquid film and liquid pool. The total heat transfer rate of the liquid pool, including natural convection and nucleate boiling, was calculated by combining their effective areas and heat transfer coefficients. The correlation for effective area was obtained based on experimental results. A range was proposed for the filling ratio to ensure the steady and effective operation of the thermosyphon. The effects of heat input, operating pressure, and geometries of the thermosyphon on the proposed range of the filling ratio were also discussed.

It is well known that the filling ratio of the working fluid, defined here as the ratio of the working fluid volume to the evaporator section volume, has a profound effect on the heat transfer performance of a thermosyphon. Usually the input working fluid is overfilled, which refers to the situation when the liquid pool remains during thermosyphon operation. In some high-temperature thermosyphons, since high heat transfer rates occur in the condenser section, burst boiling periodically takes place which is