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Heat transfer in the evaporator section of moderate-speed rotating heat pipes

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Abstract

Experiments were performed to investigate the heat transfer mechanism in the evaporator section of non-stepped rotating heat pipes at moderate rotational speeds of 2000–4000 rpm or accelerations of 40*g*–180*g*, and evaporator heat fluxes up to 100 kW/m^2. The thermal resistance of the evaporator section as well as that of the condenser section was examined by measuring the axial temperature distributions of the flow in the core region of the heat pipe and along the wall of the heat pipe. The experimental results indicated that natural convection heat transfer occurred in the liquid layer of the evaporator section under these conditions. The heat transfer measurements were in reasonable agreement with the predictions from an existing rotating heat pipe model that took into account the effect of natural convection in the evaporator section.

Keywords: Rotating heat pipe; Evaporator heat transfer; Natural convection

1. Introduction

Wickless rotating heat pipes are highly effective two-phase heat transfer devices that have been proposed and used for thermal management in rotating machinery [1–3]. The liquid flow and heat transfer in rotating heat pipes depend on a number of factors, including the heat pipe design, the working fluid and fluid loading, the rotational speed or centrifugal acceleration, and the heat flux. The liquid condensate in rotating heat pipes is usually fully annular along the inner surface when the heat pipes operate at moderate rotational speeds with accelerations greater than approximately 40*g* of interest here [4]. Models have been developed for the overall heat transfer performance of rotating heat pipes by solving the governing equations for the flow and heat transfer of the liquid annulus throughout the heat pipe [5–8]. A modified Nusselt-type laminar film condensation model has been found to be applicable for predicting the heat transfer in the condenser section [9,10]. The heat transfer in the evaporator section, however, remains less well understood.

The heat transfer mechanism in the evaporator section of stepped rotating heat pipes, where the liquid film thickness is usually more than 1 mm, appears to be similar to pool boiling under acceleration [5,10]. Natural convection and nucleate boiling occurred in sequence as the heat flux increased, with the onset of nucleate boiling being suppressed to higher heat fluxes with rotational speed. Once nucleate boiling was initiated, the thermal resistance of the liquid film in the evaporator was normally small compared to that in the condenser [11], and thus could be neglected while considering the overall heat transfer performance of rotating heat pipes. Somewhat surprisingly, these measurements suggested the evaporator heat transfer in the natural convection region was independent of rotational speed [5], which was not observed for natural convection in liquid pools under acceleration.

The liquid film thickness in the evaporator section of non-stepped rotating heat pipes is usually much smaller than in stepped heat pipes and is often comparable to or

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Nomenclature

a	centrifugal acceleration [m s^{-2}]
A	cross-section area [m^2]
c_p	specific heat at constant pressure [J kg^{-1} K^{-1}]
g	gravitational acceleration [m s^{-2}]
h_{fg}	latent heat of phase transformation [J kg^{-1}]
L	length [m]
$\dot{m}_{\mathrm{c}}$	mass flow rate of the cooling water in the condenser water jacket [kg s^{-1}]
m_{g}	mass of non-condensable gas [kg]
Nu_δ	liquid film Nusselt number, $\frac{h\delta}{k_l}$
p	vapour pressure corresponding to saturation temperature [Pa]
Q	heat transfer rate [W]
r	local radius [m]
R	thermal resistance [K/W]
Ra_δ	liquid film Rayleigh number, $\frac{\rho_l\omega^2 r\cos\alpha\beta\Delta T\delta^3}{\mu_l\alpha_l}$
Ra_δ^*	modified liquid film Rayleigh number, $\frac{\rho_l\omega^2 r\cos\alpha\beta\Delta T\delta^3}{\mu_l\alpha_l}(1+\frac{h_{\mathrm{hg}}\dot{V}\delta}{c_{\mathrm{p}_l}\alpha_l\Delta T})$
R_{g}	gas constant [J kg^{-1} K^{-1}]
T	temperature [°C]
$\dot{V}$	volumetric evaporation rate per unit area [m s^{-1}]
α	taper angle [°], or thermal diffusivity [m^2 s^{-1}]
β	thermal expansion coefficient [K^{-1}]
δ	liquid film thickness [m]
μ	dynamic viscosity [kg m $^{-1}$ s^{-1}]
ρ	density [kg m^{-3}]
ω	Angular velocity [rad s^{-1}]

Subscripts

a	adiabatic section
ac	active portion of the condenser
c	condenser section
e	evaporator section
g	non-condensable gases
i	inner wall or inlet of water jacket
in	inactive portion of the condenser
l	liquid
o	outer wall or outlet of water jacket
v	vapour core
w	wall

smaller than the diameter of typical vapour bubbles (0.1–1 mm) [12]. In this case, it was thought film evaporation would occur rather than nucleate boiling [11,13], and the heat transfer has been modeled as such in previous models for non-stepped heat pipes [6–8]. The initial models assumed conduction across the liquid film in the evaporator [6,7], while a recent model [8] considered the effect of acceleration-induced natural convection within the thin liquid film using a correlation developed for natural convection in liquid pools in a rotating boiler [14]. It was found that predictions from the model that considered natural convection were in much better agreement with measurements of the overall performance of non-stepped heat pipes at moderate rotational speeds ($40 < a/g < 180$) than those that only considered conduction [15]. Heretofore, however, there does not appear to have been measurements on non-stepped heat pipes that could be used to directly determine the heat transfer coefficient, or the thermal resistance of the evaporator section at moderate rotational speeds.

The objective of this study was to examine the heat transfer mechanism in the evaporator section of non-stepped rotating heat pipes at moderate rotational speeds. The experiments were performed for two heat pipes, one with an inner tapered condenser and the other with a cylindrical inner surface. The temperature distributions along the wall and the core regions of the heat pipes were measured using thermistors embedded in the heat pipes. The experimental facility and the heat pipes tested are described in the next section, followed by the experimental results and comparison to predictions from the models for the performance of rotating heat pipes.

2. Experimental methodology

The rotating heat pipes were tested using the facility shown in Fig. 1, also used in Song et al. [15]. The heat pipe was fit inside a Teflon sleeve that was then pressed into a stainless steel tube supported by two self-aligning bearings, and driven by a motor capable of rotational speeds up to 4775 rpm. One end of the heat pipe was heated using a dedicated induction heating unit that had a 125 mm long heating coil. The total length of the hollow working section in the heat pipes encased by the induction coil (the evaporator section) was approximately 114 mm. The end cap at the evaporator end was a 3 mm thick copper assembly that was also positioned within the induction coil during the experiments. The induction heating unit could supply heat fluxes up to approximately 100 kW/m^2 into the evaporator section. The heat transfer through the heat pipe was removed from the other end by flowing cooling water through a plastic jacket that encased the condenser. The length of the condenser section that was cooled by the flowing water in the jacket was 98 mm. The water flow to the jacket was provided using a closed loop system. The inlet water temperature was maintained constant during the tests using a PID controlled tape heater wrapped on the pipe leading to the water jacket. The volumetric flow rate of the water was measured using a rotameter with an uncertainty of ±1 mL/s, while the inlet and outlet water temperatures were measured using T-type thermocouples at the entrance and exit of the water jacket. The rotameter was calibrated by measuring the total flow over a given period of time. The thermocouples were calibrated using a