Experimental Thermal and Fluid Science 100 (2019) 1–10

Contents lists available at ScienceDirect

Experimental Thermal and Fluid Science

journal homepage: www.elsevier.com/locate/etfs

Low pressure boiling instabilities in a T-type thermosyphon

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ARTICLE INFO

Keywords:
Water thermosyphons
Low temperature
Boiling instability

ABSTRACT

Experiments were performed to characterize the boiling instabilities in a copper–water T-type thermosyphon consisting of a horizontal square cross-section evaporator embedded into a heat spreading plate with an extended condenser tube. Steady heating was applied from heaters in the plate and removed using a water jacket on the condenser tube. The unsteady heat transfer out of the condenser was compared to temperature fluctuations measured in the heated plate and on the thermosyphon for different condenser inclination angles. There was a delay in the onset of boiling and intermittent boiling with prolonged quiescent periods that caused very large variations in heat transfer rate and evaporator temperature for nominal heat fluxes of 0.43–4.3 W/cm^2 (for a saturation pressure of 1.44–10.6 kPa) when the condenser angle was 22.5° or greater from the horizontal. The effect of the delayed onset of boiling and the intermittent boiling increased with inclination angle up to 60°. Neither delayed onset of boiling or prolonged quiescent periods were observed when the condenser angle was 8° or 15°, except at very low heat transfer rates; however, there was substantial variability in the evaporator performance.

1. Introduction

Thermosyphons are effective two-phase heat transfer devices that can cool devices with large local heat fluxes without external pumping or an internal wick structure. The maximum heat transfer rates for thermosyphons are typically determined by flooding, entrainment, unsteady dry out, or boiling limits [1,2]. The performance of thermosyphons can be affected at lower heat fluxes and low saturation pressures due to boiling instabilities in the evaporator. Low saturation pressures increase the superheat required to initiate boiling, while decreasing the bubble departure frequency and for water increasing the bubble departure size [3]. These factors can result in few active boiling sites and unsteady heat transfer in pool boiling [4,5], geyser or intermittent boiling in thermosyphons [6–13], and evaporator temperature overshoot in thermosyphons at start-up even when steady boiling is later achieved [14–16].

Geyser or intermittent boiling in the thermosyphon evaporators is typically characterized by cyclic variations in the evaporator wall temperature. The evaporator temperature increases during quiescent periods. A large bubble or slug forms in the evaporator and the temperature decreases with the bubble departure and subsequent boiling until another quiescent period is initiated [7–9]. Temperature variations in the evaporator ranging from a few degrees [7,9,13] to tens of degrees [8,12] have been observed. The time period and magnitude of the fluctuations associated with the intermittent boiling typically decrease with an increase in the operating pressure or heat flux to the evaporator [7–9]. For very low operating pressures in water, the temperature variations were not regular with small, moderate, and large amplitude variations [8]. The large and moderate fluctuations appear to become less frequent with increasing heat flux rather than their magnitude or time scale changing [8].

Intermittent boiling can be mitigated by using working fluids with lower saturation temperatures. The use of water as the working fluid, however, is desirable in many applications to meet environmental requirements. The effect of intermittent boiling can be reduced in many thermosyphons by using lower fluid loadings or shorter evaporator sections [9–11]. The inclination angle of the thermosyphon also has a significant effect [17,18]. Visualizations suggest that the slug formed in the evaporator can break for moderate angles before reaching the condenser [7]. This lead to wavy flows at low angles [7] that may mitigate intermittent boiling [10,12].

The stability and performance of T-type thermosyphons at low operating pressures are considered here. T-type thermosyphons consist of a horizontal evaporator section that can be embedded into a heat spreader plate and an extended tube from the center that acts as the condenser section. The angle of the condenser section relative to the

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https://doi.org/10.1016/j.expthermflusci.2018.08.020
Received 12 February 2018; Received in revised form 10 August 2018; Accepted 15 August 2018
Available online 24 August 2018
0894-1777/ © 2018 Published by Elsevier Inc.

Nomenclature

l	Distance from bottom heater block, (mm)
$\dot{m}_w$	Mass flow rate of cooling water, (kg/s)
mC_p	Thermal capacitance of the aluminum heat spreader plate (J/K)
q	The heat flux to the evaporator, (W/m^2)
$\dot{Q}_o$	Instantaneous heat rate to the water jacket, (W)
$\dot{Q}_{in}$	Instantaneous heat rate into the evaporator, (W)
$\dot{Q}_{o,av}$	Accumulated average heat rate to the water jacket during the test time, (W)
$\dot{Q}_{in,av}$	Accumulated average heat rate into the evaporator, (W)
$\dot{Q}_h$	Nominal heat input of the heaters, (W)
$T_{p,i}$	Temperature readings at l = 26 mm, (K)
$T_{e,i}$	Temperature readings at l = 31 mm, (K)
$T_{e,m}$	Temperature readings at l = 42 mm, (K)
$T_{e,o}$	Temperature readings at l = 51 mm, (K)
$T_{ad.}$	Temperature readings at l = 58 mm, (K)
T_{sat}	Saturation temperature (K)
T_{tip}	External wall temperature readings at the condenser tip l = 326 mm, (K) = T_{sat}
$T_{w,i.}$	Cooling water inlet temperature, (K)
$T_{w,o}$	Cooling water outlet temperature, (K)
$T_{w,m}$	Cooling water mean temperature, ($T_{w,i}$ + $T_{w,o}$)/2, (K)
$\Delta T_.$	Superheat temperature difference, (K)
V_e	Evaporator volume, (m^3)
V/V_e	Fluid loading
c_{pl}	Specific heat of water, (J/kg K)
k_l	Thermal conductivity of water, (W/m K)
μ_l	Dynamic viscosity of liquid water, (Pa s)
ϑ_l	Kinematic viscosity of liquid water, (m^2/s)
ρ_l	Density of liquid water, (kg/m^3)
ρ_v	Density of saturated vapor water, (kg/m^3)
σ	Surface tension of water, (N/m)
g	Gravitational acceleration, (m/s^2)
h_{fg}	Latent heat of water, (J/kg)
P_v	Saturated internal pressure, (Pa)
P_{crit}	Critical vapor pressure of water, (Pa)
P_{atm}	Atmospheric pressure, (Pa)
θ	Condenser inclination angle from horizontal plane, (degrees)

horizontal depends on the application but is typically at low angles from the horizontal for vertical plates and vertical for horizontal plates. Experiments are performed here to examine the performance of the evaporator for a range of condenser angles. The performance is characterized following an approach similar to that of Giraud et al. [5] for pool boiling where the instantaneous heat transfer rate was compared to the instantaneous superheat. The experimental facility used in the investigations is presented in the next section followed by the results of the experiments. Conclusions are then outlined.

2. Experimental facility

The experiments to characterize the evaporator section in the T-type thermosyphon were performed using the test facility shown in Fig. 1. The evaporator was an 18 mm × 18 mm copper square cross section with a wall thickness of 1 mm. The outer length was 67 mm with approximately 3 mm thick end walls. The condenser section was a 15.87 mm O.D. internally grooved copper tube with a wall thickness of 0.58 mm with an overall length of 290 mm. The end of the tube had a semi-spherical cap with a filling tube. The condenser tube was mounted in the middle of the evaporator to form the T-shape, with an inclination angle of 8° relative to the evaporator section face. The effect of condenser orientation was examined by rotating the thermosyphon around the horizontal evaporator axis. Measurements were performed for a thermosyphon filled with 20 cm^3 of deionized water that was approximately 130% of the evaporator volume. Thermosyphons with less working fluid were found to be more unstable.

The evaporator was embedded into a 100 mm long aluminum heat spreading plate that had a nominal 50.8 mm by 50.8 mm cross section with flanges on the end away from the condenser tube. The thermosyphon was brazed into the plate. The plate was heated using two embedded 9.5 mm diameter, 100 mm long cartridge heaters. The power input into the heaters was controlled by a variable voltage controller and measured using a power transducer with an uncertainty of ± 0.5%. The thermosyphon was cooled using water from a dedicated chiller passing through a 152.4 mm long cooling jacket mounted around the extended condenser tube. The flow rate was measured using a turbine flow meter with an uncertainty of ± 0.1%, while the water temperature at the inlet and outlet was measured using T-type thermocouples.

The temperature in the heat spreading plate was measured using sets of three 0.5 mm diameter sheathed T-type thermocouples located at four positions (l) relative to the bottom of the block as shown in Fig. 1. The positions were between the heater and bottom wall of the evaporator (l of 26 mm), the evaporator surface closest to the heater (31 mm); the side surface of the evaporator (42 mm); and the outer surface (50.8 mm). The temperature on the extended tube was also measured using thermocouples at positions between the heat spreading plate and the water jacket (58 mm) and near the tip beyond the water jacket (326 mm) where the distance is relative to the bottom plate along the centerline. All thermocouples were calibrated before the experiment against a RTD (Omega Model DP251) with an uncertainty of ± 0.01 °C in a chilled water bath. The temperature of the water bath was varied from 4 °C to 80 °C in steps of 2 °C up to 30 °C and then in steps of 5 °C till 80 °C. The voltages from the thermocouples were recorded using a thermocouple A/D. The uncertainty in the temperature measurements was ± 0.1 °C. A separate A/D was used to record the output from the power meter and turbine flow meter to minimize the noise in the thermocouple measurements. The data in both cases were sampled at a rate of 2 samples per second.

The entire system, including the thermosyphon, heat spreader plate and water jacket was fully insulated during the experiments to prevent extraneous heat losses. The instantaneous heat transfer out of the condenser was determined from an energy balance to the flow through the cooling water jacket as

$$\dot{Q}_o = \dot{m}_w c_{pl}(T_{w,out} - T_{w,in})$$

The effect of bias errors in the water inlet and outlet thermocouples was reduced by subtracting the temperature difference measured before the experiments from the balance. The instantaneous heat transfer rate into the evaporator was estimated by subtracting a lumped estimate of the rate of change in energy storage in the heat spreader plate from the nominal heater input power as

$$\dot{Q}_{in} = \dot{Q}_h - mC_p\frac{dT_p}{dt}$$

where $\dot{Q}_h$ is the nominal input power to the heater, mC_p is the thermal capacitance of the aluminum heat spreader plate and T_p is the heat spreader plate temperature. The readings from the twelve thermocouples embedded in the heat spreader plate were found to be temporally well aligned and in phase, with a maximum temperature difference of about 2 °C. The time derivative of the heat spreader plate temperature (T_p) was determined using the average of the temperatures at the 3 central locations at l = 31 mm. The time response of the thermocouples was on the order of 0.5 s, and the instantaneous heat

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