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Modulated flow pattern in a condenser tube with two-phase flow interacting with mesh screen surface at micro-gravity

Dongliang Sun [a,b], Jinling Xu [b,*], Qicheng Chen [b], Yuying Yan [c,*]

[a] School of Mechanical Engineering, Beijing Institute of Petrochemical Technology, Beijing 102617, China
[b] Beijing Key Laboratory of Multiphase Flow and Heat Transfer, North China Electric Power University, Beijing, China
[c] Energy & Sustainability Research Division, Faculty of Engineering, University of Nottingham, UK

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ABSTRACT

Two-phase loop systems are important to maintain low temperature in a spacecraft at micro-gravity. It is known that the condenser size at micro-gravity can be one order magnitude larger than that at the earth gravity. This paper explores the effectiveness of the modulated flow technique at micro-gravity for the condensation heat transfer enhancement. A mesh cylinder is suspended in a tube, dividing the tube cross section into annular region and core region. When an intermittent two-phase flow interacts with the mesh screen structure, gas bubbles are prevented from entering the mesh cylinder thus they flow in the annular region to form the thin liquid film on the wall. Liquid can be separated to flow in the core region. Here the bubble dynamics in the bare tube section and modulated flow section are investigated for three different slug flow cases. It is found that the mesh cylinder modulates the slug flows in the bare tube section to form the elongated-ring-slug bubble train in the modulated flow section. Liquid plugs can disappear due to the continuous liquid flow from the annular region to the core region. Thus the two-phases are thoroughly separated with gas flowing in the annular region, forming ultra-thin liquid film thicknesses on the wall at micro-gravity. The modulated flow pattern forms the liquid film thicknesses to be about 1/30–1/10 of those in the bare tube section for the three cases. Based on the modulated bubble dynamics and parameters, the condensation rates per unit wall area ($\dot{m}$) can be about 10–100 times of those in the bare tube section for the three cases. The modulated flow pattern and enhanced heat transfer are more obvious when the gas volume flow rates are decreased. This study provides a new approach to enhance the condensation heat transfer for space applications under micro-gravity environment.

Introduction

Due to the requirements of decreased weight and compact size for large spacecraft such as the International Space Station, two-phase loop systems (Delil et al., 2002; Haobo and Reinhard, 2002; Kaya et al., 2008) have been developed to maintain low temperatures within the spacecraft depot. Two-phase loop systems have the advantages of large heat transfer capability, uniform temperature distribution and small pumping power, etc., thus they are more attractive for space applications. The two-phase loop systems involve an evaporation process to extract heat from the spacecraft depot and a condensation process to reject the heat to the outer environment.

Condensers are usually too large to meet the size and weight constraints at the normal gravity (Kim and Mudawar, 2012). It is known that the condensation heat transfer coefficients are lower at micro-gravity. Condensation at micro-gravity is a challenging issue. Fig. 1 shows the liquid film distribution over the tube cross section at the earth gravity (horizontal flow) and at the micro-gravity. Micro-gravity omits the buoyancy force between liquid and gas phases, thus liquid films are uniformly distributed on the wall and they are thick to worsen the condensation heat transfer. Da Riva and Sanz (1991) numerically investigated the condensation heat transfer in horizontal tubes with different working fluids. They found that the condenser tube lengths at micro-gravity are 28, 19, 15 and 18 times of those at the earth gravity for vapors of ammonia, water, R1 and R22, respectively (see Fig. 1). Delil (1998) computed the ammonia vapor condensation heat transfer in a 16.1 mm diameter tube at the earth gravity and micro-gravity. The flow direction was downward. It was found that the condenser tube length at micro-gravity is eleven times of that at the earth

* Corresponding authors. Tel./fax: +86 10 61772268 (J. Xu). Tel./fax: +44 0115 951 3168 (Y. Yan).
E-mail addresses: xjl@ncepu.edu.cn (J. Xu), yuying.yan@nottingham.ac.uk (Y. Yan).

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gravity. In summary, the tube length required to achieve complete condensation at the micro gravity is an order of magnitude larger than that at the normal gravity. New approach to make compact condensation tube at the micro-gravity is necessary.

Conventionally, microstructures on the tube wall enhance the condensation heat transfer. The micro groove tubes (Graham et al., 1998), micro-fin tubes (Cavallini et al., 2000), herringbone tubes (Miyara and Otsubo, 2002) and helically corrugated tubes (Suriyan and Somchai, 2010) belong to the enhanced condensation tubes. The heat transfer enhancement mechanism is attributed by mixing the fluid boundary layers and limiting the growth of fluid boundary layers close to the wall surfaces.

Recently, a phase separation condenser tube was proposed by our group to modulate flow patterns for enhancing condensation heat transfer at the normal gravity environment (Chen et al., 2012, 2013; Sun et al., 2013). A mesh cylinder is suspended in a tube, dividing the tube cross section into an annular region and a core region. When an intermittent two-phase flow interacts with the mesh screen structure, gas bubbles are prevented from entering the mesh cylinder inside due to the surface tension effect, if the mesh pores are small enough. This ensures the gas phase flowing in the annular region close to the wall, creating thin liquid films on the wall to enhance the condensation heat transfer. The pressure driven flow pushes liquids towards the core region of the tube and liquids flow there. The phase separation condenser tube is called due to the separated flow paths of gas and liquid phases in two different regions. The objective of this study is to explore the possibility that the phase separation condenser tube can be used at the micro-gravity environment. We investigate how the negligible buoyancy force between two-phases affects the modulated flow structure at micro-gravity. In order to do so, the forced convective adiabatic air–water slug flow is analyzed. The modulated flow pattern and related dynamic and stable parameters are discussed at the micro-gravity.

The physical structure

The phase separation condenser tube

Fig. 2a shows the phase separation condenser tube. Fig. 2b shows the metallic mesh screen structure, suspending in the tube. The mesh cylinder includes a flat bottom mesh pore surface and a circular side mesh pore surface. The mesh cylinder exit is open to discharge the separated liquid. The mesh cylinder inside is empty. The new condenser tube consists of a bare tube section 1 and a modulated flow section. In the modulated flow section, the tube cross section is divided into an annular region 3 and a core region 2, interfaced by the mesh pore surface. Table 1 shows the geometric parameters for the condenser tube.

During the condensation process, the flow evolves several flow patterns as annular flow and intermittent (slug, plug and bubble) flow consecutively along the flow direction. The annular flow maintains higher heat transfer coefficients in the tube upstream due to the thin liquid film. However, the thick liquid film thickness for the intermittent (slug, plug and bubble) flow deteriorates the heat transfer, yielding the decreased heat transfer coefficients along the flow length. The modulated flow section tackles the thick liquid film issue for intermittent flow patterns, thus it is arranged from a specific distance away from the tube inlet. In this study, the bare tube section arrangement is to compare the bubble dynamics between the modulated flow section and bare tube section.

The 3D to 2D-axisymmetric conversion of mesh screen

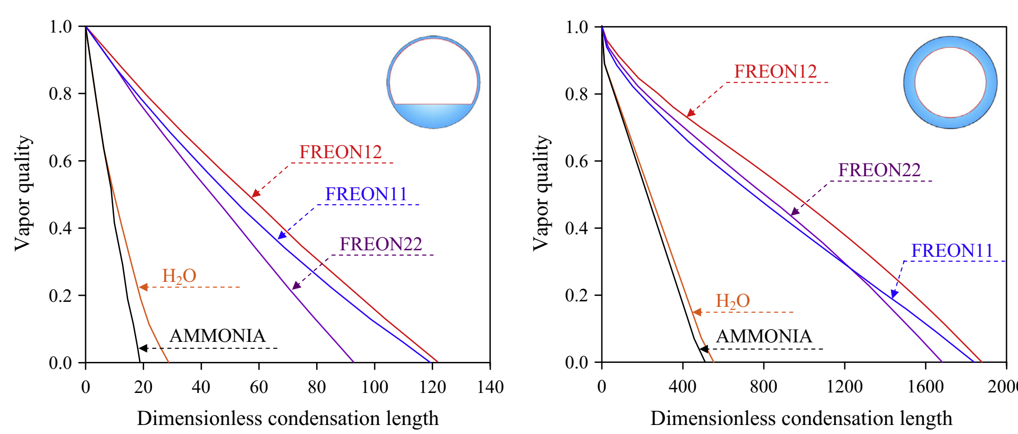
Fig. 2b shows a practical 3D mesh screen used in our experiment. The square mesh wire has a thickness of $\delta = 70$ μm. The square mesh pore has a width of $w = 150$ μm. The number of mesh pores attains about 240,000 for a mesh cylinder with its diameter of 10.36 mm and length of 350 mm. The grid number required is in the order of 100-million, making impossible to fulfill the numerical simulation. It is necessary to perform the 3D to 2D-axisymmetric conversion for simplification purpose.

The 3D mesh screen can be converted into a 2D-axisymmetric stripe-type mesh screen, by replacing δ and w (see Fig. 2b) with the stripe-type mesh wire thickness of δ' and stripe-type mesh pore gap of w' (see Fig. 2c). The following criteria should be satisfied:

The equal equivalent diameter criterion: the square mesh pore has an equivalent diameter of w, and the stripe-type mesh pore has an equivalent diameter of $2w'$. The equal equivalent diameter criterion for both types of mesh pores yields $w' = 0.5w$.

The equal flow area criterion: the projective area of the total mesh pores (see Fig. 2b) should be equal to that shown in Fig. 2c. The equal flow area criterion gives $[w/(w+\delta)]^2 = w'/(w'+\delta'')$. Thus we have $\delta' = \delta + \delta^2/2w$.

The equal capillary pressure criterion: the capillary pressure influences the gas–liquid interface near mesh pores. Once the above equal equivalent diameter and flow area criteria are satisfied, the equal capillary pressure criterion is automatically satisfied. This is because the capillary pressure created by the square mesh pores is $\Delta p = 4\sigma\cos\theta/w$, which equals to $\Delta p = 2\sigma\cos\theta/w'$ for the stripe-type mesh pores, where σ is the surface tension coefficient and θ is the contact angle.



(a) Earth gravity (horizontal position) (b) Micro-gravity

Fig. 1. Vapor quantity versus diemensionless condensation length (Da Riva and Sanz, 1991).