International Journal of Multiphase Flow 77 (2015) 58–64

Contents lists available at ScienceDirect

International Journal of Multiphase Flow

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

Transient data for flow of evaporating fluid in parallel mini pipes and comparison with theoretical simulations

Dvora Barnea*, Michael Simkhis, Yehuda Taitel

School of Mechanical Engineering, Tel Aviv University, Tel-Aviv 69978, Israel

ARTICLE INFO

Article history:
Received 25 January 2015
Revised 4 August 2015
Accepted 9 August 2015
Available online 21 August 2015

Keywords:
Two-phase flow
Evaporation
Parallel pipes

ABSTRACT

Flow rate distribution and pressure drop of an evaporating fluid in small diameter parallel pipes were investigated at steady state and transient conditions.

New quantitative experimental results on the transient response of the system to various changes in the inlet flow rate and the heating input are presented. The experimental results are compared with a time dependent model based on the temporal/local flow patterns in the parallel pipes. The experimental results compare fairly well to the theoretical ones.

Introduction

Evaporating two phase flow in parallel pipes occurs widely in power plants and cooling systems heat exchangers. In the new generation of the solar power plants, based on the parabolic trough technology, steam is generated directly in an array of parallel pipes instead of using oil as a secondary heating medium (Zarza et al., 2006; Eck and Steinmann, 2005; Eck and Hirsch, 2007). Evaporating fluid in parallel mini and micro channels is a promising tool for cooling high heat flux electronic devices (Kandlikar et al., 2006; Thome, 2006). In the above processes sub cooled liquid is fed into a parallel pipe array from a common manifold, however due to heating and evaporation, instability and uneven flow rate distributions may occur even for identical heating of all pipes.

Steady state solutions for the flow rate distribution and pressure drop in two parallel pipes were calculated by Natan et al. (2003) by using flow pattern and drift flux models. Minzer et al. (2004) added a stability analysis to demarcate among stable and unstable solutions. Minzer et al. (2006) suggested a simplified model to account for the transient behavior of the system. Baikin et al. (2011) use this model with some modifications and performed various transient simulations. They also performed steady state experiments and compared the theoretical flow rate distribution with their data. Zhang et al. (2011) presented a stability analysis and active flow and temperature control of a parallel channel evaporator. Taitel and Barnea (2011) presented a transient model for flow rate distribution in evaporating two parallel pipes based on the instantaneous local flow pattern in each pipe.

In the present work an experimental system of two heated transparent mini channels was constructed and new experimental transient data on the temporal flow rate distribution and pressure drop are presented and compared to the prediction of the transient model.

Analysis

For the sake of clarity we briefly describe the flow pattern based model (Taitel and Barnea, 2011) which is capable to predict steady state and transient behavior of the parallel pipe evaporating system.

The first step in modeling the flow in parallel pipes is to analyze a single heated pipe. The momentum equation for a flow of gas/liquid mixture in a pipe reads:

$$\frac{\partial}{\partial t}(\rho_L A\alpha_L U_L + \rho_G A\alpha_G U_G) + \frac{\partial}{\partial z}\left(\rho_L A\alpha_L U_L^2 + \rho_G A\alpha_G U_G^2\right) = -\tau s - \rho g A \sin\beta - A\frac{\partial P}{\partial z} \quad (1)$$

where ρ_L and ρ_G are the liquid and gas densities respectively, α_L and α_G are the liquid and gas void fractions, U_L and U_G are the liquid and gas velocities, A is the pipe cross sectional area, τ is the wall shear stress, s is the pipe periphery, ρ is the two-phase mixture density, β is the pipe inclination angle, and $\partial P/\partial z$ is the local pressure drop.

Eq. (1) can be presented in terms of the total mass flow rate, W, the vapor quality, X, and the slip ratio S $(=U_G/U_L)$.

$$\frac{\partial}{\partial t}\left(\frac{W}{A}\right) + \frac{\partial}{\partial z}\left\{\left(\frac{W}{A}\right)^2\left[\frac{X}{\rho_G} + \frac{S(1-X)}{\rho_L}\right]\left(X + \frac{1-X}{S}\right)\right\} = -\frac{\tau s}{A} - \rho g \sin\beta - \frac{\partial P}{\partial z} \quad (2)$$

* Corresponding author.
E-mail address: dbarnea@eng.tau.ac.il (D. Barnea).

http://dx.doi.org/10.1016/j.ijmultiphaseflow.2015.08.005
0301-9322/

Steady state solution

At steady state the first term in Eq. (2) is zero. The pipe is subdivided into n cells and the pressure in each cell is calculated by

$$P_i = P_{i-1} - \frac{\Delta z}{2}\left[\left(\frac{\tau s}{A}\right)_{i-1} + \left(\frac{\tau s}{A}\right)_i\right] - \frac{\Delta z}{2}[(\rho g \sin\beta)_{i-1} + (\rho g \sin\beta)_i] - \frac{1}{2}\left[\left(\frac{W}{A}\right)^2 (B_{i-1} + B_i)\right] \tag{3}$$

where

$$B = \left[\frac{X}{\rho_G} + \frac{S(1-X)}{\rho_L}\right]\left(X + \frac{1-X}{S}\right) \tag{4}$$

The three last terms on the rhs of (3) are the frictional, gravitational and accelerational pressure drops. The inlet pressure is assumed and the pressure in each cell is calculated yielding the outlet pressure, P_n. The procedure is repeated until the outlet pressure (P_n) matches the prescribed outlet pressure. During this iteration the enthalpy, H, and quality, X, of each cell is calculated by:

$$H_i = H_{i-1} + \frac{Q_i \Delta z}{W} \tag{5}$$

$$X_i = \frac{H_i - h_{L,i}}{h_{G,i} - h_{L,i}} \tag{6}$$

where Q_i is the net heat absorbed in each cell (per unit length). $h_{L,i}$ and $h_{G,i}$ are the enthalpies of saturated liquid and saturated vapor respectively at the local pressure. The impinging heat Q_{in}, consists of the net heat absorbed by the fluid, Q_i and the heat lost to the surrounding.

$$Q_{\text{in}} = U(T_{s,i} - T_i) + U_\infty(T_{s,i} - T_\infty) \tag{7}$$

where U and U_∞ are the equivalent heat transfer coefficients to the fluid and to the surrounding, respectively. Their values were estimated based on heat balance test performed in our experimental system (see Baikin et al., 2011). T_s is the heater local temperature. T_∞ is the ambient temperature. Eq. (7) is used to calculate the heater local temperature, T_s, for the local mixture temperature T_i. Then the heat absorbed by the mixture is

$$Q_i = U(T_{s,i} - T_i) \tag{8}$$

For a given flow rate, W, the calculated quality, X_i, in each cell (Eq. 6) along with the local gas and liquid properties enable to predict the specific flow pattern in each cell using the unified model of Barnea (1987). The flow patterns along the heated pipe for three typical mass flow rates are shown in Fig. 1. The experimental flow patterns are determined by visual observation enhanced using enlarged photos. The experimental transitions compare fairly well with the theoretical ones. Note however that due to the chaotic nature of the boiling system the experimental transition boundaries are not sharp and are presented in the figure as a smeared range (shown by the arrows).

Once the local flow pattern is known the hydrodynamic parameters of the flow pattern are calculated yielding among other things U_G, U_L, the void fraction as well as the slip ratio. The frictional and gravitational pressure drop in each cell is calculated based on the specific flow pattern. The additional acceleration term which stems from the evaporation process is calculated using Eq. (4) where the slip ratio $S = U_G/U_L$ is flow pattern based. Note that during the iteration process the properties of each cell are updated.

Fig. 2 shows the calculated pressure drop vs. flow rate at steady state for water-vapor flowing in a single pipe of 3 mm diameter, for various heating powers. The curves show the typical behavior of evaporating liquids in pipes where a range of a negative slope exists resulting in possible multiple steady state solutions for the flow rate at each pressure drop. Depending on the flow rates, water can exit the pipe as hot water, two phase mixture or as superheated vapor. Note

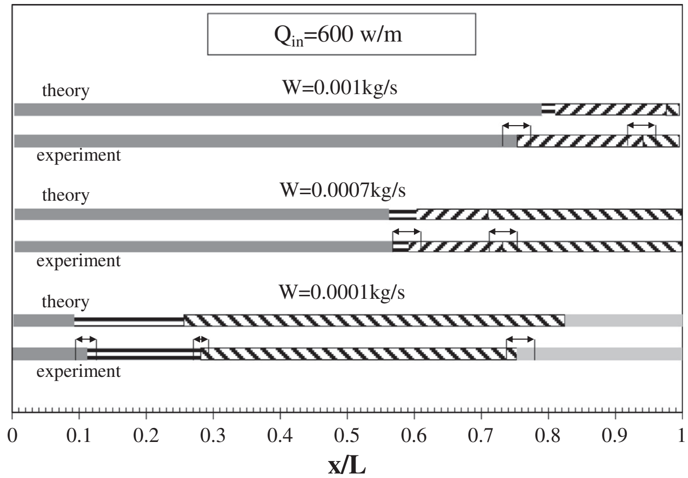

Fig. 1. Flow pattern distribution along the heated pipe for various flow rates. Water system, $D = 3$ mm, $L = 1$ m, $Q_{\text{in}} = 600$ w/m, $P_{\text{out}} = 1$ bar. liquid; stratified; slug; annular; vapor.

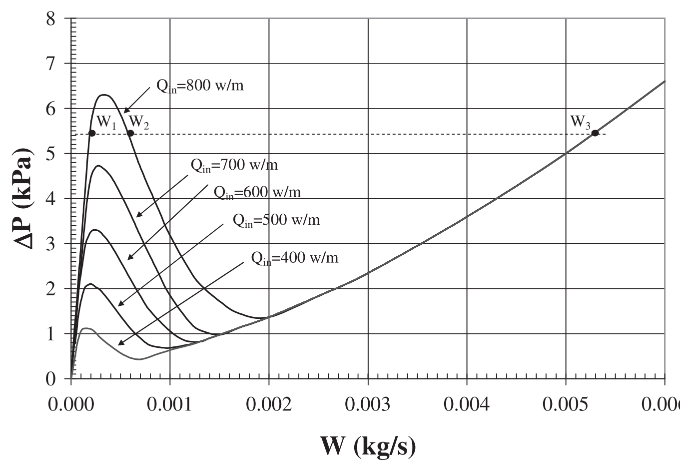

Fig. 2. Steady state solution of pressure drop vs. flow rate, single pipe. Water system, $D = 3$ mm, $L = 1$ m, $P_{\text{out}} = 1$ bar.

that the calculated pressure takes into account the additional pressure drop in the lead line from the manifold to the entrance to the test section. For a given heating power, Q_{in}, and at a specific pressure drop each pipe may have three possible steady state solutions for the flow rates W_1, W_2 and W_3.

For N parallel pipes with common inlet and outlet headers, the pressure drop in all pipes is the same. For a given inlet flow rate, W_{in}, the flow may split among the parallel pipes in various ways satisfying equal pressure drop in all pipes. For a particular inlet pressure there are 3^N possible solutions for the inlet flow rate, W_{in}. If all pipes are identical and disregarding the location of the pipe within the array one may obtain $[3 + N - 1]!/[N!(3-1)!]$ steady state values for W_{in}.

Using Fig. 2 and scanning the range of pressure drops from low to high, yields all possible combinations of W_j/W_{in} where j is the solution number ($j = 1,2,3$) at a specific pressure drop. Eventually the flow rate ratio in each pipe as a function of W_{in} is obtained.

$$R_k = W_k/W_{\text{in}} \tag{9}$$

where k is the pipe number.

Fig. 3 shows the steady state solutions for the flow rate distribution, R, vs. the total inlet flow rate, W_{in}, for two parallel pipes, where both are heated with $Q_{\text{in}} = 600$ w/m. Blue lines are the theoretical stable solutions and the red lines are unstable solutions (Baikin et al., 2011). For high and low inlet flow rates a single stable solution is