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Predicting two-phase flow distribution and stability in systems with many parallel heated channels

Tijs Van Oevelen, Justin A. Weibel, Suresh V. Garimella *

School of Mechanical Engineering, Purdue University, 585 Purdue Mall, West Lafayette, IN 47907, USA

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A B S T R A C T

Two-phase heat exchangers are used in a variety of industrial processes in which the boiling fluid flows through a network of parallel channels. In some situations, the fluid may not be uniformly distributed through all the channels, causing a degradation in the thermal performance of the system. A methodology for modeling two-phase flow distributions in parallel-channel systems is developed. The methodology combines a pressure-drop model for individual parallel channels with a pump curve into a system flow network. Due to the non-monotonicity of the pressure drop as a function of flow rate for boiling channels, many steady-state solutions exist for the system flow equations. A new numerical approach is proposed to analyze the stability of these solutions, based on a generalized eigenvalue problem. The method is specifically designed for analyzing systems with hundreds of identical parallel channels.

The method is first applied to analyze the flow distribution and stability behavior in two-channel and five-channel systems. The asymptotic behavior of the flow stability is then analyzed for increasing numbers of channels, and it is shown that the stability behavior of a system with a constant flow-rate pump curve simplifies to the stability behavior for a constant pressure-drop pump curve. A parametric study is conducted to assess the influence of inlet temperature, heat flux, and flow rate on the stability of the uniform flow distribution solution as well as on the severity of flow maldistribution. Below some critical inlet subcooling, uniform flow distribution is always stable and maldistribution does not occur, regardless of heat flux and flow rate. Above this critical inlet subcooling, there is a range of operating parameters for which uniform flow distribution is unstable. With increasing inlet subcooling, this range broadens and the severity of the associated maldistribution increases.

1. Introduction

Two-phase heat exchangers are used in a variety of industrial processes such as steam generation, air conditioning, and nuclear reactor cooling. Increased attention is being targeted at microscale two-phase heat sinks for cooling of advanced microelectronics devices used in high-performance computing clusters, power conversion systems, and radar technologies. Such two-phase flow cooling strategies allow for increased heat transfer coefficients with reduced temperature gradients as they exploit the latent heat of evaporation. However, two-phase flow instabilities may reduce heat sink performance and limit predictability and reliability. These instabilities can pose a severe impediment to industrial-scale implementation of such cooling strategies.

Two-phase flow instabilities are commonly categorized into static and dynamic instabilities [1,2]. Static instabilities occur when a disturbance causes a steady-state operating point to jump to a different operating point. Examples are the Ledinegg (excursive) instability, boiling crisis, and flow pattern transition instabilities. Dynamic instabilities occur when several physical mechanisms interact through feedback, influenced by inertia and delay. Pressure-wave (acoustic) oscillations, density-wave oscillations, and pressure-drop oscillations are the most common dynamic instabilities. Two-phase heat sinks usually comprise a large number of parallel channels to maximize the heat transfer area density. Additional instability mechanisms that may occur in these parallel channels include flow maldistribution instability and parallel-channel instability. Flow maldistribution occurs when the distribution of flow rate across parallel channels becomes non-uniform. Parallel channel instabilities constitute sustained out-of-phase channel-to-channel oscillations.

Two-phase flow instabilities have been reviewed in the literature [1–6]. A comprehensive literature review on flow maldistribution in systems with two-phase inlet mixtures, as often encountered in air conditioning systems, can be found in Ref. [7]. In those systems, the uniformity of the phase distribution in the

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* Corresponding author.

E-mail addresses: tvanoeve@purdue.edu (T. Van Oevelen), jaweibel@purdue.edu (J.A. Weibel), sureshg@purdue.edu (S.V. Garimella).

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Nomenclature

A_c	channel cross-section area (H_cW_c)
A	linearized system matrix
C	Chisholm constant
c_p	specific heat capacity
D_h	hydraulic diameter ($2H_cW_c/(H_c + W_c)$)
e	natural eigenvalue, Eq. (12)
$F_p\,(W, \Delta p)$	pump curve
F_w	volumetric wall shear force
f	friction factor
$f\,(W)$	channel load curve
G	mass flux (W/A_c)
$g\,(\lambda)$	characteristic function
H_c	channel height
h	specific enthalpy
J	relative average flow rate starvation, Eq. (21)
L_c	channel length
M	mass matrix
m	channel inertial coefficient (L_c/A_c)
N	number of parallel channels
N_{boil}	boiling number ($Q'L_c/(W_{avg}h_{fg})$)
N_{sub}	subcooling number ($(h_f - h_{in})/h_{fg}$)
N_z	number of streamwise grid cells
n	channel fraction
$P_{[0,1]}$	projection on the interval $[0, 1]$
p	pressure
Δp	pressure drop ($p_{in} - p_{out}$)
Q'	heat input per unit length
Re	Reynolds number, Eq. (39)
R_i	flow rate fraction (W_i/W)
S	slip ratio (u_g/u_f)
T	temperature
t	time coordinate

u	streamwise velocity
v	specific volume
$\mathbf{v}$	eigenvector
W	flow rate
ΔW_i^-	flow rate starvation, Eq. (19)
W_c	channel width
x	vapor quality
$\mathbf{y}$	vector of state variables
z	streamwise coordinate

Greek symbols

α	void fraction
β	aspect ratio (smallest of W_c/H_c or H_c/W_c)
ϵ	partial derivative of load/pump curve
δ	deviation
γ	relative finite difference step size
λ	eigenvalue
μ	dynamic viscosity

Subscript

avg	average
c	channel
eq	thermodynamic equilibrium
f	liquid
g	vapor
I/II/III	flow rate region, Fig. 1
i	channel index
in	inlet
out	outlet
p	pump
sat	saturation

inlet header to the different channels plays a dominant role. The focus of this work is instead on two-phase flow maldistribution in parallel-channel systems with a subcooled inlet state.

1.1. Flow maldistribution

Flow maldistribution in parallel-channel two-phase heat sinks has been observed experimentally in various studies [8–13]. Maldistribution can have several causes: asymmetrical inlet header designs, differences in channel geometry or surface properties, non-uniform heating, and the non-monotonic nature of channel pressure drop as a function of flow rate. The latter two causes are specific to the boiling flows of interest in the current work. Mechanisms underlying these two causes can be explained using Fig. 1.

Fig. 1 includes a schematic representation of the pressure drop across a boiling channel as a function of flow rate for a fixed uniform heat flux. This kind of curve is referred to as the channel load curve. A pump curve represents the pressure head provided by the pump as a function of flow rate. One general pump curve and two special cases, *viz.*, constant flow rate (vertical line) and constant pressure drop (horizontal line), are shown in the figure. Steady-state system operating points are at the intersections between the channel load curve and the pump curve (*e.g.*, points B, D, and F are all possible operating points when flow is supplied according to the general pump curve). In networks of parallel channels, each channel has its own load curve, but the operating points of each channel are not independent of each other. In particular, the system must satisfy mass conservation, *i.e.*, the sum of all channel flow rates must equal the total pump flow rate, and the pressure drop across each channel must be the same.

The N-shaped load curve of the heated channel is in contrast to the monotonic adiabatic channel load curve (Fig. 1). At high enough flow rates, the heated channel load curve is similar to the adiabatic case because the coolant is in the liquid state throughout the full length of the channel. At lower flow rates, boiling occurs in the heated channel. The vapor generation leads to

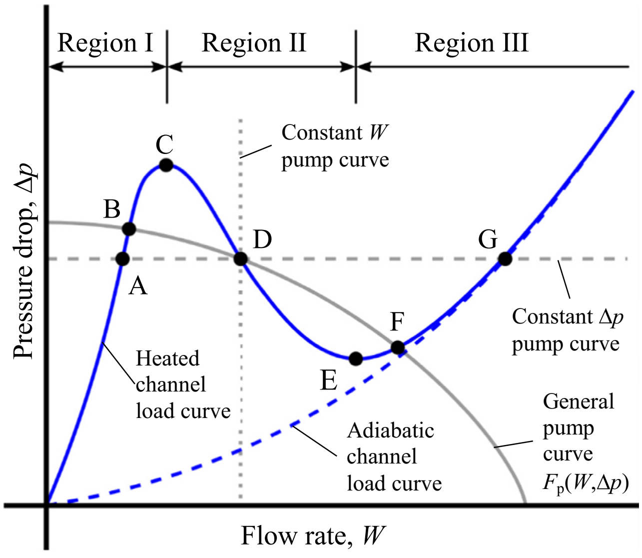

Fig. 1. Diagram of pressure drop Δp versus flow rate W, including schematic pump curves as well as load curves for single adiabatic and uniformly heated channels.