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Review

Review of databases and predictive methods for pressure drop in adiabatic, condensing and boiling mini/micro-channel flows

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ABSTRACT

Two-phase flow in mini/micro-channels has been the flow configuration of choice for many cooling applications demanding very high rates of heat dissipation per unit volume. Past research aimed at predicting the frictional pressure drop in mini/micro-channels includes a large number of studies that rely on either the Homogeneous Equilibrium Model (HEM) or semi-empirical correlations. But as the number of published studies continues to rise, thermal design engineers are confronted with tremendous confusion when selecting a suitable model or correlation. The primary reason behind this confusion is limited validity of most published methods to a few working fluids and narrow ranges of operating conditions. The present study addresses this limitation by discussing the development of two consolidated mini/micro-channel databases. The first is for adiabatic and condensing flows, and consists of 7115 frictional pressure gradient data points from 36 sources, and the second for boiling flow, and consists of 2378 data points from 16 sources. These consolidated databases are used to assess the accuracy of previous models and correlations as well as to develop 'universal' correlations that are applicable to a large number of fluids and very broad ranges of operating conditions.

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Nomenclature

Bd	Bond number
Bd^*	modified Bond number
Bo	Boiling number, q''_H/Gh_{fg}
C	parameter in Lockhart–Martinelli [94] correlation
Ca	Capillary number, $Ca = (\mu_f G)/(\rho_f \sigma)$
D	tube diameter
D_h	hydraulic diameter
f	Fanning friction factor
Fr	Froude number
G	mass velocity
g	gravitational acceleration
h	enthalpy
H_{ch}	height of rectangular channel
h_{fg}	latent heat of vaporization
J_f	superficial liquid velocity, $J_f = G(1-x)/\rho_f$
J_g	superficial vapor velocity, $J_g = Gx/\rho_g$
L	length
M	two-phase Mach number
MAE	mean absolute error
N	number of data points
N_{conf}	Confinement number
p	pressure
p_{atm}	atmospheric pressure
p_{crit}	critical pressure
P_F	wetted perimeter of channel
P_H	heated perimeter of channel
P_R	reduced pressure, $P_R = p/p_{crit}$
Δp	pressure drop
q''_{base}	heat flux averaged over base of heat sink
q''_H	effective heat flux averaged over heated perimeter of channel
Re	Reynolds number
Re_{eq}	equivalent Reynolds number
Re_f	superficial liquid Reynolds number, $Re_f = G(1-x)D_h/\mu_f$
Re_{fo}	liquid-only Reynolds number, $Re_{fo} = GD_h/\mu_f$
Re_g	superficial vapor Reynolds number, $Re_g = GxD_h/\mu_g$
Re_{go}	vapor-only Reynolds number, $Re_{go} = GD_h/\mu_g$
Re_{tp}	two-phase Reynolds number, $Re_{tp} = GD_h/\mu_{tp}$
Su	Suratman number
T	temperature
t	time
u	mean velocity
v	specific volume
$\bar{v}$	mixture specific volume
v_{fg}	specific volume difference between saturated vapor and saturated liquid
W	width of heat sink
W_{ch}	width of rectangular channel
We	Weber number
X	Lockhart–Martinelli parameter
x	quality
z	stream-wise coordinate

Greek symbols

α	void fraction
β	channel aspect ratio ($\beta < 1$)
θ	percentage predicted within ±30%
μ	dynamic viscosity
ξ	percentage predicted within ±50%
ρ	density
$\bar{\rho}$	mixture density
σ	surface tension
ϕ	two-phase multiplier; channel inclination angle
Ω	parameter in Chen et al. [103] correlation

Subscripts

A	accelerational
di	dryout incipience
exp	experimental (measured)
F	frictional
f	saturated liquid
fo	liquid only
G	gravitational
g	saturated vapor
go	vapor only
in	channel inlet
k	liquid (f) or vapor (g)
$pred$	predicted
sat	saturation
sub	subcooling
tp	two-phase
tt	turbulent liquid–turbulent vapor
tv	turbulent liquid–laminar vapor
vt	laminar liquid–turbulent vapor
vv	laminar liquid–laminar vapor

1. Introduction

Developers of many modern devices are faced with two conflicting trends: the need to dissipate increasing amounts of heat, and the quest for more compact and lightweight designs. These trends have spurred unprecedented increases in heat dissipation per volume and per surface area, rendering most present air cooling and single-phase liquid cooling solutions virtually obsolete [1,2]. High performance computers, electric vehicle power electronics, avionics, and defense laser and microwave systems are but a few examples of modern applications that are confronted with these trends. Cooling demands in these and many other applications have resulted in a paradigm shift from single-phase to two-phase cooling strategies to capitalize upon the coolant's sensible and latent heat rather the sensible heat alone. And phase change cooling solutions come in a variety of configurations that could meet the system requirements of the application in question. These include pool boiling [3,4], channel flow boiling [5,6], mini/micro-channels [7,8], jet [9–12], spray [13–15], boiling on enhanced surfaces [16–19], and hybrid cooling techniques that combine the benefits of two or more two-phase cooling schemes [20,21].

While most two-phase cooling systems employ air-cooled condensers to reject the heat to the ambient, the quest for more compact and light weight system designs has shifted interest to the use of miniature condensers to reject the heat by condensing a primary coolant in a compact primary cooling loop. The heat is transferred to a secondary liquid coolant and transported to a remote heat exchanger where it is ultimately rejected to the ambient. Employing boiling and condensation in a cooling system has netted orders of magnitude enhancement in heat transfer coefficients and therefore large reductions in cooling system's weight and volume compared to single-phase liquid cooling counterparts.

Among the different two-phase cooling schemes, two-phase flow in mini/micro-channels has been especially favored for its ability to deliver very high heat dissipation per unit area and unit volume, in addition to offering a number of other practical advantages such as ease of fabrication and small coolant inventory. Fig. 1 shows several examples of applications that have benefited greatly