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Convective heat transfer in foams under laminar flow in pipes and tube bundles

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ABSTRACT

The present study reports experimental data and scaling analysis for forced convection of foams and microfoams in laminar flow in circular and rectangular tubes as well as in tube bundles. Foams and microfoams are pseudoplastic (shear thinning) two-phase fluids consisting of tightly packed bubbles with diameters ranging from tens of microns to a few millimeters. They have found applications in separation processes, soil remediation, oil recovery, water treatment, food processes, as well as in fire fighting and in heat exchangers. First, aqueous solutions of surfactant Tween 20 with different concentrations were used to generate microfoams with various porosity, bubble size distribution, and rheological behavior. These different microfoams were flowed in uniformly heated circular tubes of different diameter instrumented with thermocouples. A wide range of heat fluxes and flow rates were explored. Experimental data were compared with analytical and semi-empirical expressions derived and validated for single-phase power-law fluids. These correlations were extended to two-phase foams by defining the Reynolds number based on the effective viscosity and density of microfoams. However, the local Nusselt and Prandtl numbers were defined based on the specific heat and thermal conductivity of water. Indeed, the heated wall was continuously in contact with a film of water controlling convective heat transfer to the microfoams. Overall, good agreement between experimental results and model predictions was obtained for all experimental conditions considered. Finally, the same approach was shown to be also valid for experimental data reported in the literature for laminar forced convection of microfoams in rectangular minichannels and of macrofoams across aligned and staggered tube bundles with constant wall heat flux.

1. Introduction

Microfoams consist of tightly packed spherical bubbles between 10 and 100 μm in diameter with a porosity of up to 70% [1]. They can be produced by spinning a disk at 5000–10,000 rpm in an aqueous surfactant solution contained in a baffled beaker at room temperature [1]. Such microfoams have also been termed colloidal gas aphrons (CGA) [1]. However, the multiple surfactant-shell structure forming around individual bubbles proposed by Sebba [1] has not been directly and unequivocally observed [2]. Thus, we prefer to call this two-phase fluid "microfoams" instead of "CGA". Microfoams have found numerous applications including separation processes [3–5], soil remediation [5–7], water treatment [8,9], and biotechnology [10]. These applications take advantage of (i) their large interfacial area, (ii) the adsorption of particles at the microbubble interfaces, and (iii) their stability for enhanced mass transfer [11]. Microfoam made from mixtures of anionic and cationic surfactants have also been shown to spread over a pool of burning gasoline and to extinguish fire [1].

Moreover, traditional macrofoams are commonly used as fire suppressant [12]. They have also been used as a fracturing fluid for improved oil recovery. Then, convective heat transfer takes place between the rock formation and the foams [13]. Finally, macrofoams have also been considered as a working fluid in heat exchangers to take advantage of the fact that the associated heat transfer coefficient is significantly larger than that achieved using air under the same conditions [14,15]. This could reduce the size and mass of air-based heat exchangers.

In these various applications, it is important to understand and predict transport phenomena in foams including convective heat transfer. The present study investigates experimentally forced convection in microfoams flowing in circular tubes under laminar flow conditions and subject to constant wall heat flux. It also presents a scaling analysis for convective heat transfer in microfoams in rectangular minichannels and macrofoams in tube bundles under laminar flow.

2. Background

2.1. Microfoam rheology

The rheological behavior of foams and microfoams can be described by the pseudoplastic power-law model expressed as [16],

$$\tau_w = K_p \dot{\gamma}_w^n = K_p' \dot{\gamma}_a^n = \mu_f \dot{\gamma}_a \quad (1)$$

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Nomenclature

A	cross-sectional area of test section (m^2)
Ca^*	volume equalized capillary number, Eq. (3)
$C(\chi)$	empirical function, Eq. (4)
c_p	specific heat (J/kg K)
D_h	hydraulic diameter (m)
h_x	local heat transfer coefficient (W/m^2 K)
I	current (A)
k	thermal conductivity (W/m K)
K_p	flow consistency constant
L	length of the test section (m)
l_L	dimensionless longitudinal pitch, Eq. (15)
l_T	dimensionless traverse pitch, Eq. (15)
$\dot{m}$	mass flow rate (kg/s)
M_f	mass of microfoam (Kg)
n	flow behavior index
Nu_x	local Nusselt number, $Nu_x = h_x D_h/k$
$Nu_{D,exp}$	experimental local Nusselt number for tube bundles, Eq. (27)
Nu_∞	Nusselt number for thermally fully developed conditions
Pr	Prandtl number, $Pr = c_p\mu/k$
$\dot{Q}$	volumetric flow rate (m^3/s)
q_f	actual heat input to the microfoam (W)
q_{loss}	heat loss to the surrounding (W)
q_{total}	total power input (W)
q''_w	wall heat flux (W/m^2)
Re_D	Reynolds number, $Re = \rho\dot{Q}/\frac{1}{4}\pi D_h\mu$
$Re_{D,max}$	Reynolds number for flows across tube bundles, Eq. (13)
r_i	inner pipe radius (m)
r_o	outer pipe radius (m)
r_{32}	Sauter mean bubble radius (m)
$T_f(x_i)$	average local temperature of microfoams (K)
T_{in}	microfoam temperature at test section inlet (K)
T_{out}	microfoam temperature at test section outlet (K)
$T_i(x_i)$	inner wall temperature at location x_i (K)
$T_{wall}(x_i)$	outer wall temperature at location x_i (K)
t	time (s)
U_{max}	maximum velocity in minimum flow area (m/s)
ϑ	voltage (V)
V_f	volume of microfoams (mL)
V_p	volume of tube bundles (m^3)
x	dimensional axial length (m)
x_i	thermocouple location from pipe entrance (m)
x^+	dimensionless axial length for cylindrical pipe, $x^+ = 2x/D_h Re_D Pr$

Greek symbols

α	rectangular channel aspect ratio, (α = width/height)
χ	surfactant mass fraction (wt.%)
ϵ	specific expansion ratio
ϕ	microfoam porosity
$\dot{\gamma}_a$	apparent shear rate (1/s)
$\dot{\gamma}_w$	wall shear rate (1/s)
μ	dynamic fluid viscosity (Pa s)
σ	surface tension (N/m)
τ_w	wall shear stress (Pa)
τ^*	dimensionless shear stress

Subscripts

f	refers to microfoam or foam property
g	refers to air in microfoam
$pipe$	refers to stainless steel pipe property
w	refers to water property or wall shear
3,4	refers to rectangular channels heated from 3 or 4 walls

Superscript

*	refers to power-law fluid dimensionless numbers and correlations

where τ_w is the wall shear stress, $\dot{\gamma}_w$ is the true wall shear rate, and $\dot{\gamma}_a$ is the apparent shear rate while the effective foam viscosity is denoted by μ_f. The empirical constants K_P and n are the so-called flow consistency and flow behavior index, respectively. The true wall shear rate $\dot{\gamma}_w$ can be derived from $\dot{\gamma}_a$ through the Rabinowitsch–Mooney relationship [17],

$$\dot{\gamma}_w = \left(\frac{3n+1}{4n}\right)\dot{\gamma}_a \quad \text{and} \quad K'_p = K_p\left(\frac{3n+1}{4n}\right)^n. \tag{2}$$

Recently, Larmignat et al. [16] investigated the rheology of microfoams flowing through cylindrical pipes. Experimental data were collected for aqueous solutions of non-ionic surfactant polyoxyethylene (20) sorbitan monolaurate (Tween 20) with mass fraction ranging from 0.03 to 9.96 wt.%. The authors defined the volume equalized dimensionless shear stress and Capillary number as

$$\tau^* = \frac{\tau_w r_{32}}{\sigma\epsilon} \quad \text{and} \quad Ca^* = \frac{\mu_w r_{32}\dot{\gamma}_a}{\epsilon\sigma} \tag{3}$$

where r_{32} is the Sauter mean bubble radius, σ is the surface tension, μ_w is the viscosity of water, and ϵ is the specific expansion ratio defined as the ratio of the densities of the liquid phase and microfoam. Experimental data established that $\tau^* = C(\chi)(Ca^*)^{2/3}$ where $C(\chi)$ is an empirical function dependent on the surfactant mass fraction χ (in wt.%). In practice, microfoams made with aqueous solutions of Tween 20 can be treated as a shear-thinning fluid with an effective viscosity given by [16],

$$\mu_f = \mu_w C(\chi)(Ca^*)^{-1/3} \quad \text{with} \quad C(\chi) = 0.4 + 0.8(1 - e^{-\chi/0.018}) \tag{4}$$

These results were in good agreement with the model developed by Denkov et al. [18,19] for foams with porosity larger than 90% made of fluids with low surface dilatational modulus resulting typically in tangentially mobile bubble surface. They were also confirmed by experimental measurements for anionic and cationic surfactants including hexadecyl–trimethyl–ammonium bromide (CTAB) and sodium lauryl sulfate (SDS) [20].

2.2. Convective heat transfer in power-law fluids in circular pipes

Bird [21] derived an analytical solution predicting the local Nusselt number in the entry region Nu_x^* and in the thermally fully developed region Nu_∞^* for single phase power-law fluids with flow behavior index n flowing through circular pipes subject to constant wall heat flux as,

$$Nu_x^* = \frac{h_x D_h}{k} = 1.41\left(\frac{3n+1}{4n}\right)^{1/3}\left(\frac{2}{x^+}\right)^{1/3} \quad \text{and} \quad Nu_\infty^* = \frac{8(5n+1)(3n+1)}{31n^2+12n+1}. \tag{5}$$

Here, D_h is the hydraulic diameter of the wetted perimeter, k is the thermal conductivity of the power-law fluid, and h_x is the local heat transfer coefficient. The dimensionless axial length x^+ is defined as $x^+ = 2x/D_h Re_D Pr$ where x is the axial location measured from the heated pipe entrance. The Reynolds and Prandtl numbers respectively denoted by Re_D and Pr were defined as,

$$Re_D = \frac{4\rho\dot{Q}}{\pi D_h\mu} \quad \text{and} \quad Pr = \frac{c_p\mu}{k} \tag{6}$$