International Journal of Heat and Mass Transfer 69 (2014) 230–236

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

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

Experimental investigation of heat transfer from inclined flat surface to aqueous foam

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ARTICLE INFO

Article history:
Received 7 February 2013
Received in revised form 2 October 2013
Accepted 3 October 2013
Available online 6 November 2013

Keywords:
Experimental investigation
Macrofoam flow
Heat transfer
Inclined flat surface
Drained liquid film

ABSTRACT

Paper presents the results of the experimental investigation of heat transfer process between the inclined flat surface and the longitudinal upward macrofoam (foam) flow. Investigation was performed on the special laboratory stand using macrofoam, which was generated inside the channel on the perforated plate. Foaming solution was made from the 0.5% concentration of the washing powder (TIDE Absolute) solution in water and gas (air). Macrofoam parameters: velocity 0.10 ÷ 0.30 m/s; volumetric void fraction 0.996 ÷ 0.998; bubble dimensions 0.005 ÷ 0.015 m. Inclination angle of the flat surface 45°. Experimental surface was heated using electric current; surface temperature varied from 16.8 to 36.2 °C; foam flow temperature varied from 14.6 to 19.5 °C.

Investigation showed that the heat transfer rate depends on the foam flow characteristics (velocity and volumetric void fraction) and on the drained liquid film parameters (film thickness, film flow velocity and direction). It was stated that the heat transfer rate increases with the increase of the foam flow velocity up to the critical value. When the foam flow velocity exceeds the critical value, the thickness of the drained liquid film begins to grow and heat transfer rate decreases. Further augmentation of the foam flow velocity forms a concurrent drained liquid flow and heat transfer intensity increases again.

The experimental results were compared with the results obtained for the vertical flat surface and for the different types of the tube bundles as well.

1. Introduction

Aqueous two-phase foam has wide application for various technical and technological purposes. For example, nowadays the fire-fighting is inconceivable without usage of the aqueous foam; foam can be used for gas clearing from dust [1], for plant protection against the frost [2], for preparation of concrete with new characteristics [3], etc. Moreover, aqueous two-phase foam can be used as a coolant in various heat exchangers [4,5]. One kind of the foams – microfoams – is widely investigated nowadays in order to use them as a coolant in microchannels and micro equipment [5]. Recent investigations [4,6] show that the macrofoams (foams with polygonal structure and very high content of the gas) can be used as a coolant also. Main advantage of the macrofoam in comparison with the one-phase coolant is relatively high heat transfer rate at a low mass flow. At the macrofoam density 3.2–5.2 kg/m^3 and flow velocity less than 0.5 m/s heat transfer rate reaches 1.0 kW/(m^2 K) and more [4,6]. From the other side, gas (air) density is less than macrofoam density only by 3–4 times, but heat transfer rate under the same flow velocity is less by 100 and more times [7]. Some specific processes, such as liquid drainage [8,9], diffusive gas transfer [10,11], division and junction of the foam bubbles [11] and others take place inside the macrofoam flow. Therefore heat transfer rate depends not only on the foam flow velocity and volumetric void fraction but also on the geometry, shape and orientation of the heat transfer surfaces as well.

Our previous work [12] was devoted to the investigation and analysis of the heat transfer between the vertical flat surface and upward macrofoam flow. It was stated that an irregular cross-sectional distribution of the foam flow velocity influences on the heat transfer rate at the different parts of the surface. Distribution of the volumetric void fraction across and along the surface also has impact on the cooling rate. Heat transfer rate of the vertical flat surface mainly depends on the foam flow velocity and increases according to the increase of the foam flow rate [12].

Recent work presents the results of the experimental investigation of the heat transfer between heated inclined flat surface and the longitudinal upward flow of the macrofoam. In this case liquid drains from the foam and forms a thin film on the inclined flat surface. Heat transfer rate of the inclined surface depends not only on the foam flow velocity and volumetric void fraction, but also on the drained liquid film thickness, film flow direction and velocity [13]. The experimental results of the investigation of the inclined flat

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http://dx.doi.org/10.1016/j.ijheatmasstransfer.2013.10.005

Nomenclature

A	area, m^2
d	external diameter of the tube, m
G	flow rate, m^3/s
h	heat transfer coefficient, $W/(m^2\,K)$
I	amperage, A
i	location of thermocouple along the surface ($i = 1,2,\ldots,6$)
j	location of thermocouple across the surface ($j = 1,2,3$)
L	length of the surface, m
Nu	Nusselt number
P	cross–sectional perimeter of the channel, m
q	heat flux density, W/m^2
Re	Reynolds number
T	temperature, K
U	voltage, V
w	flow rate, m/s
W	heated surface width, m
x	distance along the channel, m
y	distance across the channel, m

Greek symbols

β	volumetric void fraction, dimensionless
γ	angle of inclination, °
λ	thermal conductivity, W/(m K)
ν	kinematic viscosity, m^2/s

Subscripts

av	average
b	bubble
ch	channel
f	foam flow
g	gas
l	liquid
w	wall

surface were compared with the results obtained for the vertical flat surface [12] and for the different types of the tube bundles [6,14,15].

2. Experimental set-up and methodology

An experimental investigation of the heat transfer from the inclined flat surface to the upward foam flow was performed on the laboratory set-up (Fig. 1), which was composed of the experimental channel, flat surface with electrical heating system, foamable liquid (foaming solution) and gas delivery systems, measuring instruments and auxiliary equipment. Main part of the experimental set-up was made from the transparent square cross-section channel ($0.14 \times 0.14\ m^2$) with the inclination angle of 45° (Fig. 2). Stainless steel flat surface ($0.5 \times 0.12\ m^2$, thickness 0.001 m) was assembled at the bottom wall of the channel. Surface was heated electrically (current 64.5 ± 0.5 A and voltage 1.55 ± 0.02 V). Temperature of the experimental surface was measured by eighteen calibrated type T thermocouples. Four additional four thermocouples were used for the measurement of the foam flow temperature: two of them were installed at the 0.025 m distance before and two – after the experimental surface. Surface temperature varied from 16.8 to 36.2 °C; foam flow temperature mediated from 14.6 to 19.5 °C. Heat transfer rate was estimated from 150 to 1200 W/m^2 K.

Macrofoam flow was generated at the lower part of the set-up by bubbling air into the foaming solution which was prepared from water and surfactant (washing powder TIDE Absolute, concentration 0.5%) [4]. Solution characteristics at the 20 °C temperature were as follows: surface tension 0.0375 N/m; viscosity 0.00995 cm^2/s; density 1003.0 kg/m^3. Air bubbled through the perforated stainless steel plate (thickness 0.002 m): orifices (diameter 0.001 m) were located in a staggered order; spacing between the centers of the orifices 0.005 m. Gas (air) flow rate varied from 1.96×10^{-3} to $5.88 \times 10^{-3}\ m^3/s$; liquid rate – from 3.92×10^{-6} to $19.68 \times 10^{-6}\ m^3/s$.

Foam flow parameters (velocity and volumetric void fraction) were controlled using gas and liquid valves and flow meters. Foam flow velocity range: $0.10 \div 0.30$ m/s; volumetric void fraction: 0.996, 0.997 and 0.998; approximate dimension of foam bubbles was: $0.005 \div 0.015$ m.

Main parameters were obtained according to the following equations:

– volumetric void fraction

$$\beta = \frac{G_g}{G_g + G_l} \tag{1}$$

- mean velocity of the foam flow

$$w = \frac{G_g + G_l}{A_{ch}} \tag{2}$$

- heat flux density

$$q_w = \frac{U \cdot I}{A_w} \tag{3}$$

- local heat transfer coefficient

$$h_{ij} = \frac{q_w}{\Delta T_{ij}} \tag{4}$$

- average cross-sectional heat transfer coefficient

$$h_i = \frac{1}{W}\int_0^W h(x,y)dy \tag{5}$$

- average heat transfer coefficient for the whole surface

$$h_{av} = \frac{1}{A_w}\int_0^{A_w} h(x,y)dA_w \tag{6}$$

- average heat transfer rate generalized by Nusselt criterion

$$Nu_{fav} = \frac{h_{av} \cdot W}{\lambda_{fav}} \tag{7}$$

- thermal conductivity

$$\lambda_{fav} = \beta \cdot \lambda_{gav} + (1-\beta) \cdot \lambda_{lav} \tag{8}$$

- foam flow gas Reynolds number

$$\mathrm{Re}_g = \frac{G_g \cdot d_e}{A_{ch} \cdot \nu_g} \tag{9}$$

- an equivalent diameter d_e of the cross-section of the experimental channel