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# Experimental investigation of heat transfer from vertical flat surface to upward aqueous foam flow

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ABSTRACT

Paper presents the results of the experimental investigation of the local and average heat transfer process between the vertical flat surface and upward macro foam flow. Macro foam was generated by bubbling gas (air) into the liquid made from the 0.5% concentration of the washing powder (Tide Absolute) in water. Macro foam flow parameters: velocity $0.10 \div 0.25$ m/s; volumetric void fraction $0.996 \div 0.998$; bubbles size $0.005 \div 0.015$ m. Surface temperature varied from 25 to 40 °C; macro foam flow temperature $12.5 \div 18.0$ °C.

Investigation showed that the heat transfer rate depends on the foam flow parameters: velocity and volumetric void fraction. It was stated that the heat transfer rate increased with the increase of the foam flow velocity and with the decrease of the volumetric void fraction. The local heat transfer rate is lower at the sides and is higher at the middle part of the surface. The maximum of the local heat transfer rate (up to 1137 W/(m$^2$K)) is identified at the inlet region. Average heat transfer rate for the whole surface varied from 152 to 331 W/(m$^2$K).

The results of the experimental investigation were generalized using the criterion equations.



## 1. Introduction

Besides a wide spectrum of the possible application [1–3], aqueous macro foam (foam) flow can be used as a coolant in the various heat exchangers [4,5]. Possibility to change heat transfer rate by changing foam flow velocity and volumetric void fraction are very important features allowing applying the foam flow for the hot surfaces cooling [4]. Another important advantage of the foam flow as a coolant is a possibility to reach high values of the heat transfer coefficient using relatively small mass flow rates [4]. Density of the macro foam ($3.2 \div 5.2$ kg/m$^3$) [4,5] is $2.6 \div 4.3$ times higher than that of the air but cooling rate is higher by $30 \div 50$ times [4,6]. From the other side, foam density is approximately $200 \div 300$ times lower than that of the water but cooling rate is lower by $5 \div 6$ times only [4,6]. Therefore the ratio of the heat transfer rate (W/(m$^2$K)) to the coolant density (kg/m$^3$) for the various coolants can be put into the following rank: water flow – about 4, air flow – 13, macro foam flow – 120. It is clear that the usage of the foam flow as a coolant allows reaching higher heat transfer rate with the lower mass flow rate.

Apart those well-known advantages, the practical application of the foam flow is quite complicated due to the some specific self-destructive processes: liquid drainage from the foam [7,8]; diffusive gas transfer [5,9]; division, junction and destruction of the foam bubbles [5,10]; Gibbs–Marangoni effect [11,12] and etc. Therefore the aqueous foam which is used as a coolant must keep its structure for the sufficient time period necessary for passing through the heat transfer zone [4].

There are many different works [6,13,14] which are devoted to the investigation of the heat transfer process between the various types of the tube bundles or flat surfaces and the single-phase coolant. At the same time there are not enough results related to the investigation of the possibility to use the macro foam flow as a coolant in the heat exchangers. Early investigations [4] of the heat transfer from the single tubes (0.004 and 0.014 m diameter) and from the vertical line of the tubes (0.014 m diameter) to upward macro foam flow showed that the heat transfer rate varied from 350 to 682 W/(m$^2$K) depending on the foam flow velocity (up to 0.4 m/s) and on the volumetric void fraction (up to 0.999). Investigation of the heat transfer from the staggered [15], in-line [16] and non-standard [17] tube bundles to the macro foam flow showed that the peculiarities of the foam flow (velocity, void fraction, shadowing effect, drainage, flow direction end etc.) had

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**Nomenclature**

| | |
|---|---|
| $A$ | area, m$^2$ |
| $c$ | coefficient, dimensionless |
| $d$ | external diameter of the tube, m |
| $G$ | flow rate, m$^3$/s |
| $h$ | heat transfer coefficient, W/(m$^2$K) |
| $I$ | amperage, A |
| $L$ | heated surface length, m |
| $m$ | coefficient, dimensionless |
| $n$ | coefficient, dimensionless |
| Nu | Nusselt number, dimensionless |
| $P$ | cross-sectional perimeter of the channel, m |
| Pr | Prandtl number, dimensionless |
| $q$ | heat flux density, W/m$^2$ |
| Re | Reynolds number, dimensionless |
| $s$ | spacing between the centers of the tubes, m |
| $T$ | 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*

| | |
|---|---|
| $\beta$ | volumetric void fraction, dimensionless |
| $\lambda$ | thermal conductivity, W/(m K) |
| $\nu$ | kinematic viscosity, m$^2$/s |

*Subscripts*

| | |
|---|---|
| $av$ | average |
| $b$ | bubble |
| $ch$ | channel |
| $f$ | foam flow |
| $fl$ | fluid (air or water) flow |
| $g$ | gas |
| $i$ | location of thermocouple along the surface ($i$=1, 2, ..., 6) |
| $in$ | input |
| $j$ | location of thermocouple across the surface ($j$=1, 2, 3) |
| $l$ | liquid |
| $out$ | output |
| $w$ | wall of the heated surface |

a significant influence on the heat transfer rate. Apart that it was noticed that, differently from the single coolant case [6], the heat transfer rate of the frontal tubes was higher than that of the furthered tubes [15–17]. Heat transfer from the flat inclined surface to the longitudinal upward foam flow was investigated as well [18,19]. It was stated that the liquid drained from the foam flow and formed a thin film on the inclined flat surface. Heat transfer rate of the inclined surface was influenced not only by the foam flow velocity and volumetric void fraction, but by the drained liquid film thickness, film flow direction and velocity also [18].

The current work presents the results of the experimental investigation of the heat transfer process between the heated vertical flat surface and the upward macro foam flow. The local and average heat transfer coefficients were determined and compared with the experimental results obtained for the single-phase flow, inclined flat surface, single tube, staggered, in-line and non-standard tube bundles.

## 2. Experimental set-up and methodology

The experimental investigation was performed on the set-up (Fig. 1) which was made from the vertically oriented transparent experimental channel (height 1.8 m; cross-section area $0.14 \times 0.14$ m$^2$) with one electrically heated stainless steel surface ($L \times W = 0.5 \times 0.12$ m$^2$; thickness 0.0001 m) (Fig. 2). Set-up also included: macro foam flow generation and control system, heating system, data measuring and monitoring systems. The macro foam was generated by bubbling gas (air) into the solution made from the 0.5% concentration of the washing powder (Tide Absolute) in water. Solution parameters 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$^3$. The perforated stainless steel plate (thickness 0.002 m) was used for the foam generation. 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 $2.5 \cdot 10^{-3}$ to $6.3 \cdot 10^{-3}$ m$^3$/s. Accuracy of gas flow measurements was $\pm 0.1 \times 10^{-3}$ m$^3$/s. Liquid flow rate varied from $5.0 \cdot 10^{-6}$ to $25.3 \cdot 10^{-6}$ m$^3$/s. Accuracy of liquid flow measurements was $\pm 0.25 \times 10^{-6}$ m$^3$/s.

Macro foam flow parameters (velocity and volumetric void fraction) were controlled using gas and liquid valves and flow meters. Foam flow velocity range was $0.10 \div 0.25$ m/s; homogeneous flow of the macro foam was reached at the volumetric void fraction: 0.996, 0.997 and 0.998 (mass void fraction was 0.235, 0.290 and 0.381 correspondingly). Approximate diameter of the foam bubbles: $d_b = 0.005$ m for $\beta = 0.996$, $d_b = 0.010$ m for $\beta = 0.997$ and $d_b = 0.015$ m for $\beta = 0.998$; macro foam flow temperature varied in the range $12.5 \div 18.0$ °C. Gas flow control valve (Fig. 1) was used for the control of the foam flow velocity; liquid flow rate was controlled by the liquid flow regulation valve (Fig. 1). Velocity and volumetric void fraction of foam flow was controlled by changing gas flow rate.

Surface was heated electrically (current: $I = 0 \div 80 \pm 0.5$ A and voltage: $U = 0 \div 2 \pm 0.02$ V). Voltage was measured by multimeter "ESCORT 3136A". The second channel of the input of this multimeter was connected to the current shunt and was used for the electric current measurement. Temperature of the surface was measured by eighteen calibrated copper – constantan (type T) thermocouples, which were allocated in three rows along the surface with six thermocouples in each row (Fig. 2). Distance between two thermocouples in the row 80 mm, distance between two columns 40 mm. Four additional thermocouples were used for the measurement of the foam flow temperature: two thermocouples were installed before and two after the heated surface. Surface temperature varied from 25 to 40 °C. All thermocouples were connected to three data loggers "PICO TC–08". Measured values of the temperatures were stored on the hard disk of the laptop.

The following equations were used to obtain the main parameters:

– volumetric void fraction

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

– mean velocity of the foam flow

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