The Heat Transfer Mechanism in Aqueous Foam Flow in a Channel

IRENA GABRIELAITHIENÉ
Institute of Energy Technology, Kaunas University of Technology
Address: K. Donelaičio g. 20–212, LT–44239, Kaunas, LITHUANIA
Gabriel@geobaltus.com, Irena.Gabrielaitiene@vgtu.lt, www.ktu.lt

JONAS GYLYS
Institute of Energy Technology, Kaunas University of Technology
Address: K. Donelaičio g. 20–212, LT–44239, Kaunas, LITHUANIA
Jonas.Gylys@ktu.lt, www.ktu.lt

ROLANDAS JONYNAS
Department of Thermal and Nuclear Energy, Kaunas University of Technology
Address: K. Donelaičio g. 20, LT–44239, Kaunas, LITHUANIA
Rolandas.Jonynas@ktu.lt, www.ktu.lt

TADAS ŽDANKUS
Institute of Energy Technology, Kaunas University of Technology
Address: K. Donelaičio g. 20–212, LT–44239, Kaunas, LITHUANIA
Tadas.Zdankus@ktu.lt, www.ktu.lt

Abstract: The heat transfer mechanism in two-phase aqueous foam flow was investigated for developing energy-efficient heat exchangers. Such heat exchangers can provide low consumption of energy resources due to enhanced heat transfer rates. An enhanced heat transfer rates are achieved in aqueous foam due to an especially large inter-phase contact surface and reduced surface tension (when compared to pure liquids). A typical element of a plate heat exchanger was considered, which consists of a vertical rectangular channel with a plane heated surface. For such geometry, the heat transfer rates were investigated experimentally for the different foam inlet conditions. Gas Reynolds numbers \( R_e_g \) has ranged from 190 to 380, corresponding foam velocity \( 0.14 - 0.3 \) m/s. Volumetric void fraction (\( \beta \)), defining the wetness of the foam, had the values from 0.996 to 0.999.

Key-Words: aqueous foam flow, heat transfer, heat exchangers

1 Introduction
Heat transfer mechanism in two-phase aqueous foam flow is important, as it can be applied for developing energy-efficient heat exchangers. In such heat exchangers low consumption of primary energy resources can be achieved by enhancing the heat transfer rates. An increased amount of heat can be transferred between the working mediums of a heat exchanger. For this, two main strategies have been applied in the past: first, complex channel geometry of heat exchangers has been introduced; and second, application of advanced heat transfer carrier as working medium of heat exchangers.

Good examples of a first strategy are presented in [1, 2], where corrugated ducts and channels with ribs of different geometry (i.e., V-shape) were applied. Such ribs tend to increase turbulence rates and invoke a secondary flow appearance, and therefore increase heat transfer rates. In other studies, advanced heat transfer carriers were applied including multiphase flow, boiling or condensation [3, 4]. This work considers application of aqueous foam flow, which represents a two-phase flow regime. It consists of gas bubbles separated by a thin liquid film. It has an especially large inter-phase contact surface and reduced surface tension when compared to pure liquids. As a result of this, the heat transfer coefficient of the foam is more pronounced than a heat transfer coefficient for convectional liquids, for example for air [5, 6].

Previous studies of aqueous foam flow were focused on investigation of foam generation and deterioration problems [7]. Physical properties of foam flow have been examined in [8]. Complex rheological behaviour of foam has been studied in [9]. Foam dynamic behaviour in viscous flows and
foam-drop formation were studied in [10]. Different applications of aqueous foam flow have been considered in the past. It includes firefighting and food applications, oil recovery, textile and cosmetics (for example, soap or shaving foam [8]). In this work we study the heat transfer mechanism in aqueous foam flow in geometry relevant for heat exchangers applications.

2 Problem Statement

2.1 Heat Transfer Coefficient for Different Fluids in a Channel with a Single Heated Pipe

The heat transfer coefficient of aqueous foam flow is more pronounced than a heat transfer coefficient for convectional liquids, for example for air. Foam flow heat transfer coefficient varies from 350 to 682 W/m²K depending on the volumetric void fraction. For air this coefficient is 16 W/m²K [11] under the same conditions. This data was obtained from the heat transfer from a single pipe (0.014 m diameter) under the flow velocity of 0.4 m/s. A comparison with another heat transfer carrier — water — shows that aqueous foam has a significantly smaller density. The foam density is 3.2÷5.2 kg/m³ [11], while water density is 998.2 kg/m³ (T=293.15 K, P=101325 Pa).

2.2 Geometry of a Heated Surface in a Heat Exchanger Channel

The foam flow was investigated for geometry of a typical heat exchanger element. It includes a straight vertical channel provided with a heated surface, followed by a U-bend (see Fig. 1). The geometry of heated surface in such channel was a plane vertical surface.

Previously, a heated surface of a circular pipe was considered. In a first study, a single pipe and a column of pipes were investigated in a foam flow of a vertical channel [11]. The pipe bundles were then introduced into the experimental channel, because each pipe exerts exposure on flow of neighbouring pipes. As a result, heat exchange in bundle differs from heat transfer of individual pipe. Moreover, distances between heated pipes influence the heat transfer rates greatly. Thus, the impact of different bundles configurations was investigated.

Staggered pipe bundle configurations were compared with an inline pipe bundle configurations (i.e., pipes arranged in corridor patterns) [5, 12]. The latter was found to produce higher heat transfer rates, but only in downward flow. In upward flow, it depends on values of gas Reynolds numbers, Re_g. Thus, the inline pipe bundle configuration was investigated further with different horizontal and vertical steps between the pipes. Horizontal steps of 1.5 and 3 were considered with a constant vertical step of 1.5 (representing a ratio of a distance between pipe rows or columns and the pipe diameter). It was demonstrated that the intensity of heat transfer bundles with horizontal and vertical steps 1.5x1.5 is above that found for 3x1.5 steps in upward flow [6]. Another study [13] indicated that an increased vertical spacing between the pipes (i.e., vertical step greater than 1.5) might be beneficial for heat transfer intensity, which highlights than other configurations should also be considered.

However, it was difficult to find a configuration of pipe bundles that could provide enhanced heat transfer rates for wide range of considered gas Reynolds numbers, Re_g, and volumetric void fraction in upward and downward flow. Therefore, a plane vertical surface was considered as a heated surface, which is relevant to plate heat exchangers. The above-mentioned geometry was investigated for foam flow with following inlet conditions: gas Reynolds numbers Re_g varied from 190 to 380; volumetric void fraction (β) had the values 0.996, 0.997, 0.998 and 0.999.

3 Experimental Set-up and Procedure

Experimental work was performed by changing the values of volumetric void fraction and foam velocity. Foam velocity was modified by changing the flow rate of the gas (see eq.3). The efficiency of the heat transfer process was evaluated by determining the heat transfer coefficient for each situation.

Foam was made of gas (air) and a liquid solutions, where water was mixed with detergent of 0.5% concentration. A surfactant, (phosphoric compound and stabilizers) was used to create a disjoining force that prevents the interfaces between bubbles from rupturing and the bubbles from merging.

3.1 Experimental Set-up

The experimental set-up included a supply system of a gas and liquid solution, and a foam generation riddle (Fig. 1). The supply system was build of a liquid reservoir (1, in Fig.1), which stored the liquid solution of water and detergent additive; a liquid level control reservoir (3); liquid (2,4) and gas
control valves (7) and flow meters (5, 6). The foam generation section (9) consisted of a perforated metal plate with holes of diameter 1mm staggered at 5 mm distance. The liquid solution was supplied from four walls of the rectangular experimental channel (13) to ensure equal distribution of the liquid. The vertical heated plate (11) involved a stainless steel foil with a thickness of 0.1 mm. On this plate the copper-constanatan thermocouples (0.1mm) were placed, as shown in Fig. 2. The same thermocouples were applied for inlet and outlet foam flow temperature measurements (10). The other equipment was electric current transformer and stabilizer (16).

![Experimental setup diagram](image)

**Fig.1 Experimental set-up.**

In Fig. 1: 1,3 – liquid reservoir and receiver; 2, 4, 7 – liquid and gas control valves; 5, 6 – flow meter; 8 – compressor; 9 – foam generation riddle; 10, 12 – thermocouples; 11 – vertical heated plate; 13 – experimental channel; 16 – transformer.

Experimental channel was made of transparent material that enables examining foam flow visually. It had a square cross section, where side dimension is 0.14 m. Experimental channel height was 1.8 m with U-turn radius R=0.17 m. Turbulent flow was created at the inlet of experimental channel, which stabilizes approximately after 20 cm from the inlet. Change of inlet conditions was stabilized after 5 min.

**3.1 Experimental Procedure**

From the record of heated surface and foam flow temperatures, the difference between averaged surface and foam flow values (ΔT) was determined, which was used for the estimation of the heat transfer coefficient (eq. 1). From registration of electric current and voltage of a vertical plane plate, the heat flux density on the heated surface \( q''_w \) is determined. The average heat transfer coefficient was estimated from equation (1):

\[
\frac{q''_w}{\Delta T} = h
\]

(1)

where \( q''_w \) – is heat flux, estimated from measurements of electric current and voltage of heated surface [W/m²]; \( \Delta T \) – is difference between average temperature of a heated surface and average temperature of the foam flow. The later is estimated from the temperature of foam flow before and after the heated surface.

The experiments were repeated for different values of volumetric void fraction. Volumetric void fraction was defined by the following equation:

\[
\beta = \frac{G_g}{G_g + G_l}
\]

(2)

The \( G_g \) and \( G_l \) are the volume flow rate of gas and liquid flow before entering a generation section [m³/s].

![Thermocouple layout diagram](image)

**Fig.2 Layout of thermocouples on a plane vertical surface**

Foam velocity was estimated from the following equation:

\[
\frac{Q_f}{F} = \frac{G_g + G_l}{F}
\]

F is a cross-sectional area of the foam flow [m²].

Gas Reynolds number of foam flow [m].

Gas Reynolds number of foam flow was computed by the formula:
where \( l \) – characteristic length \([\text{m}]\); \( A \) – cross-sectional area of a channel \([\text{m}^2]\), \( \nu_g \) – gas viscosity \([\text{m}^2/\text{s}]\).

Accuracy of all thermocouples was ±0.5 K across its operating temperature range of 273.15–373.15 K (0–100 °C). Gas and liquid flow rates were measured by flow meters. Accuracy of flow meter was ±0.1×10^{-3} \text{ m}^3/\text{s} for gas (i.e., air) across all operating range, which varied from 0 to 10 ×10^{-3} \text{ m}^3/\text{s}; and it was ±0.25×10^{-6} \text{ m}^3/\text{s} for detergent solution across all operating range, which varied from 0 to 40×10^{-6} \text{ m}^3/\text{s}. Calorimeters power supply system’s voltage was stabilized by stabilizer and was reduced by transformer; electric current magnitude was measured by ammeter and voltage – by voltmeter. Accuracy of the ammeter measurements were ±0.1 A across all its operating range, which was from 0 to 10 A; accuracy of the voltmeter measurements were ±0.05 V across all its operating range, which was from 0 to 25 V.

The uncertainty analysis was performed by method outlined by Coleman and Steele (1999). It indicated that the values of heat transfer coefficients were estimated with 8.1 % uncertainty.

4 Effect of Volumetric Void Fraction on Heat Transfer Rates

The influence of volumetric void fraction on foam structure is shown on Fig. 3. Pictures on Figs. 3 and 4 were obtained by a camera, as the walls of the experimental channel were made of transparent plexiglas. On Fig. 3, the geometry of liquid foam is depicted for the gas Reynolds numbers, \( Re_g = 190 \).

![Fig.3 Structure of a foam flow (Re\(_g\)=190); a) \( \beta=0.998 \), b) \( \beta=0.997 \), c) \( \beta=0.996 \).](image1)

The structure of the foam consists of bubbles with liquid residing between the bubbles to form a channel network, i.e., Plateau borders. Such highly structured geometry of liquid foam (liquid films bounded by Plateau borders and junctions) and mechanics at the film level influence the heat transfer rates. In the Figures 3 and 4, it is a Plateau borders that are apparent, rather than the liquid films, which forms the bubbles walls (as they are very thin, for such relatively dry foams).

![Fig.4 The foam flow in experimental channel.](image2)

For a wet foam (\( \beta=0.996 \)), an average bubble diameter was relatively small and constitutes 5mm±2 mm (Fig. 3 (c)). Such wet foam provides better heat transfer condition, than dry foam (\( \beta = 0.998 \)), where larger bubbles appears in the main stream and small bubbles in the area close to the walls. The bubble shapes are more polyhedral in the channel centre, where foam is dry, and more spherical close to the channel walls, where the foam is wet, as can be seen in Fig. 4. Similarly, the bubbles are more polyhedral for relatively dry foam (Fig.3 (a)) and more spherical for relatively wet foam (Fig. 3 (c)). The average diameter for dry foam was \( d_b=15±2 \text{ mm} \) (Fig. 3 (a)), while the average diameter for \( \beta=0.997 \) was \( d_b=10±1.5 \text{ mm} \) (Fig. 3 (b)). When increasing gas flow rate further (\( Re_g \) is equal to 380), the average bubble diameter is 1.5-2 time smaller than in the above-mentioned cases, and flow structure becomes homogenous.

In Figure 5 heat transfer rates are depicted for the different values of volumetric void fraction, (\( \beta \)). For example, the average heat transfer coefficient of a wet foam (\( \beta = 0.996 \)) is higher than heat transfer
coefficient of a dried foam ($\beta = 0.9970$) by about 20%. Average heat transfer coefficients are 172.9 W/m\(^2\)K and 144.8 W/m\(^2\)K for foam $\beta = 0.996$ and foam $\beta = 0.997$, respectively. In wet foam, the amount of liquid in Plateau borders and junctions has increased. The gravity and capillary forces facilitate the liquid drainage from the foam and increase the thickness of liquid film, composing on the heated surface and channel walls.

Fig. 5 Heat transfer coefficient for the different values of volumetric void fraction, ($\beta$)

As the wetness of the foam decreases, the difference between the average heat transfer coefficients becomes more significant. It constitutes about 54% between foam $\beta = 0.997$ and foam $\beta = 0.998$. Furthermore, the difference is about 74% between foam $\beta = 0.998$ and foam $\beta = 0.999$. Average heat transfer coefficients are 94.2 W/m\(^2\)K and 54.1 W/m\(^2\)K for foam $\beta = 0.998$ and foam $\beta = 0.999$, respectively.

From the comparison of relatively wet foam ($\beta=0.996$) and dry foam ($\beta=0.998$), it was found that for wet foam flow ($\beta=0.996$) an average heat transfer coefficient was 70% higher than for dry foam. The gas Reynolds numbers $Re_g$ ranged from 190 to 380, corresponding foam velocity 0.14 - 0.3 m/s.

5 Effect of Foam Velocity on Heat Transfer Rates

The increase in foam velocity has the opposite effect on heat transfer rates than the increase of values of volumetric void fraction. While heat transfer rates increase with the decrease of beta values ($\beta$), the increase of gas flow rate produce the better heat transfer conditions. When increasing gas flow rate, the (average) bubble diameter becomes smaller especially in the centre of the channel and flow structure transforms into homogenous. Small bubbles in comparison with large bubbles (provided that the wetness of the foam is constant), ensured more effective drainage conditions. The circulation of the liquid through the Plateau borders and junctions is more effective, although the Plateau borders might become thinner for high gas Reynolds numbers, $Re_g$. It affects the thickness of liquid film, which is composed on the heated surface and channel walls.

In Fig. 6, the increase of heat transfer rates has been denoted for the different values of foam velocity: from 0.14 to 0.3 m/s (gas Reynolds numbers $Re_g$ from 190 to 380). Foam velocity was modified by changing the flow rate of the gas (see eq. 3). For example, for dry foam ($\beta = 0.999$), heat transfer rates have increased from 46,2 W/m\(^2\)K to 54,3 W/m\(^2\)K (18% increase). For foam ($\beta = 0.998$), heat transfer rates have increased from 83,9 W/m\(^2\)K to 110 W/m\(^2\)K (30% increase). For foam ($\beta = 0.997$), heat transfer rates have increased from 135 W/m\(^2\)K to 165 W/m\(^2\)K; while for foam ($\beta = 0.996$), heat transfer rates have increased from 150 W/m\(^2\)K to 192 W/m\(^2\)K. This contains 22% and 29% increase for the foam ($\beta = 0.997$) and foam ($\beta = 0.996$), respectively.

Fig. 6 Heat transfer rates for the different values of volumetric void fraction.

From the comparison of heat transfer enhancement with increase of the gas Reynolds numbers, $Re_g$ from 190 to 380 (foam velocity 0.14 - 0.3 m/s), it was noted that an increase in heat transfer rates is similar for relatively wet foam ($\beta=0.996$) and dry foam ($\beta=0.998$). This increase constituted 30% in both cases.

6 Conclusion

The work details an experimental study for investigation of two-phase aqueous foam flow in a heat exchanger channel. The impact of the velocity and volumetric void fraction on heat transfer rates was examined for the plane vertical heated surface. Heat transfer rates were established for foam flow with following inlet conditions: gas Reynolds
numbers $Re_g$ varied from 190 to 380; volumetric void fraction ($\beta$) had the values 0.996, 0.997, 0.998 and 0.999.

- When comparing relatively wet foam ($\beta=0.996$), and dry foam ($\beta=0.998$), it was found that for wet foam flow, average heat transfer coefficient was 70% higher than for dry foam. In wet foam, the amount of liquid residing between the bubbles (i.e., in Plateau borders and junctions) is higher than in dry foam. The gravity and capillary forces facilitate the liquid drainage from the wet foam and increase the thickness of liquid film, composing on the heated surface and channel walls.
- When comparing a heat transfer enhancement with increase of the gas Reynolds numbers $Re_g$ from 190 to 380, it was noted that an increase in heat transfer rates is similar for relatively wet foam ($\beta=0.996$) and dry foam ($\beta=0.998$). This increase constituted 30% in both cases. In a foam flow with high flow rates of gas, the average bubble diameter is much smaller than for a foam flow with low flow rates of gas (wetness of the foam is constant). This occurs especially in the centre of the channel and flow structure becomes nearly homogenous. Small bubbles in comparison with large bubbles (provided that that wetness of the foam is constant), ensured more effective drainage conditions and increase of heat transfer rates.

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