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COMPARISON OF HEAT TRANSFER CHARACTERISTICS IN SURFACE COOLING WITH BOILING MICROJETS OF WATER, ETHANOL AND HFE7100

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Abstract The basis of microjet technology is to produce laminar jets which when impinging the surface have a very high kinetic energy at the stagnation point. Boundary layer is not formed in those conditions, while the area of film cooling has a very high turbulence resulting from a very high heat transfer coefficient. Applied technology of jet production can result with the size of jets ranging from 20 to 500µm in breadth and 20 to 100µm in width. Presented data are used in order to validate authors own semi-empirical model of surface cooling by evaporating microjet impingement in the stagnation point. Main objective of this paper was to investigate the physical phenomena occurring on solid surfaces upon impingement of the single microjet in case of three fluids. Intense heat transfer in the impact zone of microjet has been examined and described with precise measurements of thermal and flow conditions of microjets. Reported tests were conducted under steady state conditions for surface cooling by single microjet producing an evaporating film. Obtained database of experimental data with analytical solutions and numerical computer simulation allows the rational design and calculation of microjet modules and optimum performance of these modules for various industrial applications.

Keywords: microjets, heat transfer intensification, boiling,

1. Introduction

Accurate control of cooling parameters is required in ever wider range of technical applications. It is known that reducing the dimensions of the size of nozzle leads to an increase in the economy of cooling and improves its quality. Present study describes research related to the design and construction of the nozzles and microjet study, which may be applied in many technical applications such as in metallurgy, electronics, etc.

Using liquids such as water, boiling is likely to occur when the surface temperature exceeds the coolant saturation temperature. Boiling is associated with large rates of heat transfer because of the latent heat of evaporation and because of the enhancement of the level of turbulence between the liquid and the solid surface, Garimella and Rice (1995). This enhancement is due to the mixing action associated with the cyclic nucleation, growth, and departure or collapse of vapour bubbles on the surface. In the case of flow boiling, such as boiling under impinging jets, the interaction between the bubble dynamics and the jet hydrodynamics has significant effect on the rate of heat transfer. The common approach used to determine the rate of boiling heat transfer is by using a set of empirical equations that correlate the value of the surface heat flux or the heat transfer coefficient with the fluid properties, surface conditions, and flow conditions.

These correlations do not provide much insight into the underlying physical mechanisms involved in the boiling heat transfer problem, Liu and Zhu (2002). The alternative approach is to use mechanistic models. There have been a number of mechanistic models developed for the case of pool boiling and for the case of parallel flow boiling. In the latter case, the boiling heat transfer phenomenon is more complicated due to the strong coupling between the flow, the thermal field, and bubble dynamics.

The basis of microjet technology is to produce laminar jets which when impinging the surface have a very high kinetic energy at the stagnation point, Mikielewicz and Muszynski (2009), Mikielewicz et al (2013). Boundary layer is not formed in those conditions, while the area of film cooling has a very high turbulence resulting from a very high heat transfer coefficient. Applied technology of jet production can result with the size of jets ranging from 20 to 500μm in breadth and 20 to 100μm in width.

Main objective of this paper was to investigate the physical phenomena occurring on solid surfaces upon impingement of the single microjet in case of three fluids. Intense heat transfer in the impact zone of microjet has been examined and described with precise measurements of thermal and flow conditions of microjets. Reported tests were conducted under steady state conditions for surface cooling by single microjet producing an evaporating film. Obtained database of experimental data with analytical solutions and numerical computer simulation allows the rational design and calculation of microjet modules and optimum performance of these modules for various industrial applications. Presented data are used to validate authors own semi-empirical model of surface cooling by evaporating microjet impingement in the stagnation point.

2.Experimental setup

Present study shows results of steady state heat transfer experiments, conducted for single phase cooling in order to obtain wall temperature and heat fluxes. Fig. 1 shows the schematic diagram of the test section. It consisted of the probe, fluid supplying system, the measuring devices and DC power supply. Working fluid was fed to the nozzle from a supply tank, which also serves as the pressure accumulator. The water pressure in the test section was raised by an air compressor. Desired fluid flow rate was obtained by sustaining the constant pressure of fluid with a proper use of flow control valve. In order to reduce pressure drop necessary to create a steady laminar jet, nozzle was 2mm long. Due to low volumetric flow rate of coolant it was measured at inlet and outlet from the cooling chamber with a graduated flask.

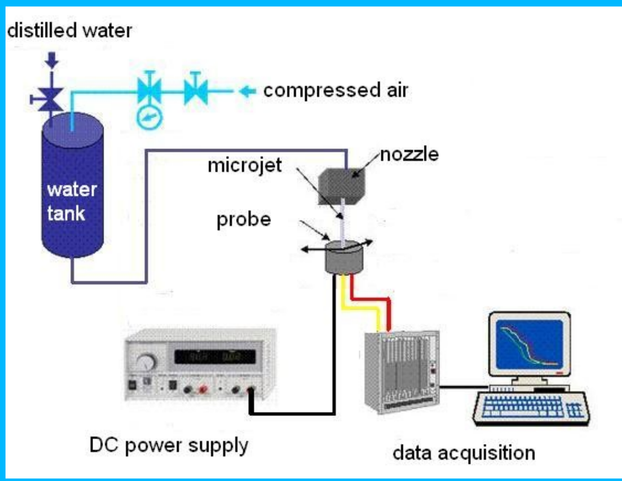

Fig.1 The schematic diagram of test section

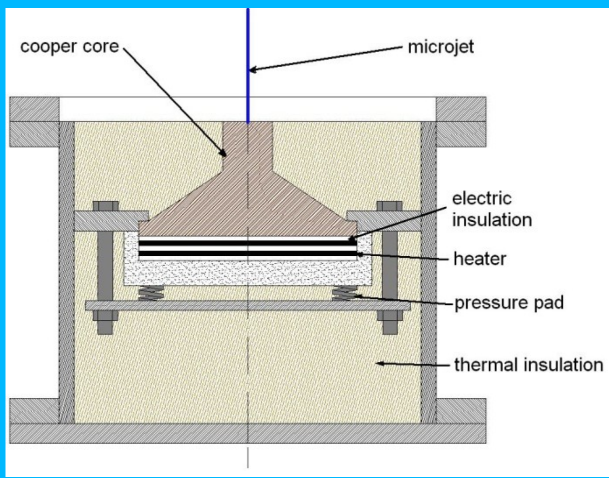

Fig.2 The cross-section of the probe

The cooled surface was the copper truncated cone with top diameter 10mm and 20mm height. Water impingement surface was silver-plated, in order to prevent high temperature erosion. The radial distribution of surface temperature was determined with the aid of five T-type thermocouples, created from embedding 50μm thick constantan wire to the copper core. Heat is supplied by a kanthal heater, electrically isolated with sapphire glass and mounted at the bottom of the core. The whole set is thermally insulated by glass wool and placed in the casing. Heater is powered by a DC power supply and the total power input is determined by measuring current intensity and voltage. During test s heater was capable of dissipating up to 240W.

Additional four K-type thermocouples are attached in the copper rod axis. These thermocouples measure axial temperature gradient at the core of a heating block and control temperature of the heater. They are connected to the National Instruments data acquisition set. The signal from thermocouples was processed with the aid of the LabVIEW application. Heater operating power values are precisely controlled and measured. The applied power losses through conduction into the insulation and radiation to the surroundings are accurately calculated and accounted for in all tests. Data are taken from a steady state measuring points in order to exclude heat capacity of the installation.

The nozzle construction allows modification of its dimensions. In case of these studies nozzle with hydraulic diameter of 75 μm was used. Experiments were conducted for the spacing of 50mm between the nozzle exit and impinging surface. Because of limited power supply, low water mass flux were used in order to obtain wide range of surface temperature.

3. Theoretical model

In the analysis of carried out experiments the simple theoretical model of impinging liquid microjet with phase change is presented. Proposed is the model of heat transfer in the stagnation zone, where liquid rapidly evaporates due to contact with the hot surface. The model is developed on the basis of known pressure difference between the nozzle exit and the stagnation point. As a result of evaporation of impinging jet, there is formed a vapour blanket on the surface. The dynamic pressure is the way in which the nozzle interacts with the surface, $\Delta p_d = \rho u_d^2/2$. In the analysis considered also could be the capillary effect $\Delta p_{kap} = \sigma/D$ (with D being the nozzle diameter) and the hydrostatic pressure drop $\Delta p_g = (\rho_l - \rho_v) gH$ (where H denotes the jet suspension over the surface). The latter two have however been omitted in the present analysis as they have been regarded as of secondary importance. The total pressure acting on the surface yields:

$$\Delta p_c = \Delta p_d + \Delta p_{kap} + \Delta p_g \quad (1)$$

The schematic of rapidly evaporating microjet is presented in fig. 3. The radius on which the spreading of liquid film is taking place can be determined from the energy balance on the element of the cooled plate, which reads:

$$q \pi R^2 = \dot{m}_l \left[h_{lv} + c_p \left(T_{SAT} - T_0 \right) \right] \quad (2)$$

From (1) the impingement radius reads:

$$R = \sqrt{\frac{\dot{m}_l (h_{lv} + c_p \Delta T)}{\pi q}} \quad (3)$$

The applicability for further calculations of the radius obtained from equation (3) has also been confirmed experimentally in the course of authors own experiment (Mikielewicz et al. 2009). In that study values or the range of cooling of the surface, obtained from (3) showed a very good consistency with experimental findings. Substituting into that equation the values of properties at atmospheric pressure and ambient temperature of 20°C (c_p=4184.3 J/kg K, h_{lv}=2256.4 kJ/kg, and ΔT=80K, C=908.2) the obtained radius of cooling was equal to 2.87mm, which agreed very well with experimental finding for that case when mass flow rate was 3×10^{-5} kg/s and heat flux q=3.0MW/m^2.

In the analysis of the evaporating jet the following assumptions were made:

- only a dynamic part of pressure difference in (1) is acting on the jet,
- liquid temperature on the radius of evaporation R is higher than saturation temperature.

The mass balance on evaporation surface yields:

$$dm_l = \pi R^2 d\delta \rho_l = dm_v = 2\pi R \delta_v \rho_v V_v d\tau \quad (4)$$

Equation (4) enables to determine the expression for estimation of the film thickness

$$\dot{m}_l = \frac{dm_l}{d\tau} = 2\pi R \delta_v \rho_v V_v \quad (5)$$

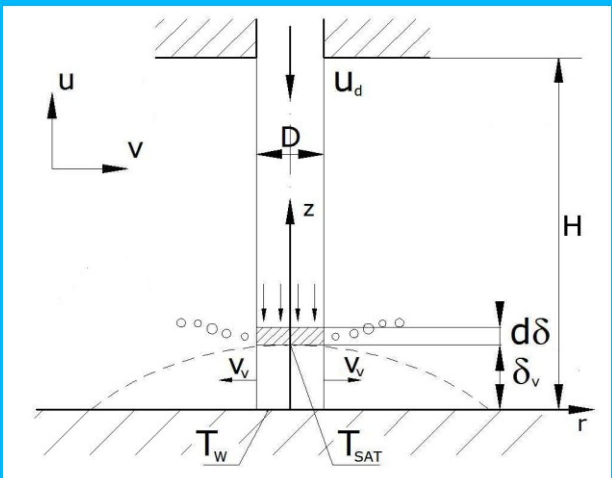

Fig. 3. Scheme of rapidly evaporating microjet

In order to solve equation (2) we must provide the means for calculation of vapour velocity V_v. That can be done by the reference to the concept by Kutateladze who solved a similar problem for the outflow from a large tank through a small hole. In our approach we assume that the radial vapour motion from the stagnation point can be modeled in a similar way. It is assumed in our approach that vapour which is formed as a result of impingement forms a cylinder with the base corresponding to the nozzle diameter, as seen from fig. 3. Similarly to Kutateladze's approach, such motion can be modeled as flow of vapour from the side wall of the cylinder of the size $\pi D\delta$, by analogy to the outflow from the tank through a small hole. Such problem, as sketched in fig. 2 can be described by Bernouli equation:

$$p_1 + \frac{\rho_v {u_1}^2}{2} + gh_1 = p_2 + \frac{\rho_v {u_2}^2}{2} + gh_2 \tag{6}$$

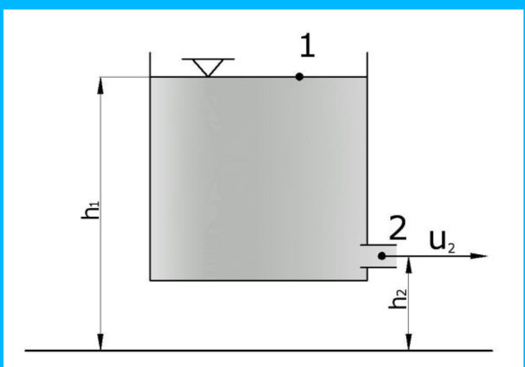

Fig. 4. Outflow from the large tank through a small hole

Standard solution is obtained assuming that u_1=0 and velocity at outlet is found from:

$$u_2 = \zeta\sqrt{\frac{2\Delta p}{\rho}} \tag{7}$$

In our case we are dealing with the flow of vapour coming out of the nozzle. Hence we assume that $\rho=\rho_v$. We are aware that although in the model due to Kutateladze velocity at state 1 is assumed zero, in our case it has a finite value which can be determined from:

$$u_1 = \frac{q}{\rho_v h_{lv}} \tag{8}$$

In subsequent works on that topic eqn (8) will be attempted to be included into modeling but for the time being the vapour velocity is:

$$V_v = \varsigma\sqrt{\frac{2\Delta p_c}{\rho_v}} \tag{9}$$

In (9) $\Delta p_c = \frac{1}{2}\rho_l u_d^2$ is the microjet dynamic pressure and ζ - the contraction number. Substituting vapour velocity to (5) yields:

$$\dot{m}_l = 2\pi R\delta_v\varsigma\sqrt{\rho_v\rho_l}u_d \tag{10}$$

After describing the heat transfer on cooled surface as heat conduction in vapour layer, the film thickness can be found:

$$\delta_v = \sqrt{\frac{\lambda_v(T_w - T_{SAT})D}{4\varsigma\sqrt{\rho_l\rho_v}u_d\left[h_{lv} + c_p(T_{SAT} - T_0)\right]}} \tag{11}$$

Knowledge of film thickness allows to determine the heat transfer coefficient:

$$Nu = \frac{\alpha D}{\lambda_v} = \frac{2}{\sqrt{\frac{Ja_v}{\varsigma \Pr_{lv}(1 + Ja_l)}}} \tag{12}$$

where

$$\Pr_{lv} = \frac{\nu_l}{a_{vl}} = \frac{\nu_l}{\frac{\lambda_v}{c_{pv}\sqrt{\rho_l\rho_v}}}, \quad Ja_v = \frac{(T_W - T_{SAT})c_{pv}}{h_{lv}} \quad \text{and}$$

$$Ja_l = \frac{(T_{SAT} - T_0)c_{pl}}{h_{lv}}$$

Some attention should be devoted to the calculation of the contraction factor present in equation (9). A good consistency of results is partially devoted to the fact that the contraction factor present in (9) is calculated from the empirically adjusted formula:

$$\varsigma = 62.34 \times 10^{18} Bo^{3.804} \quad (13)$$

4 Experimental validation

Fig. 7 shows the experimental results of fully developed nucleate boiling heat transfer at the stagnation zone for impinging saturated water jet using the nozzle dimensions of 0.1x0.05mm. Heat transfer data are plotted in the form of boiling curves for different liquid mass flux. The Kutatieladze's empirical correlation is also presented as a solid line, and Mostinski's correlation as dashed line to point out the similarity of the considered jet cooling to the pool boiling case of liquid. For nucleate pool boiling of water at atmospheric pressure, correlation due to Kutateladze (1952) can be simplified to:

$$q_{cr} = 15.58 \cdot (T_W - T_S)^{\frac{10}{3}} \left[\frac{W}{m^2} \right] \quad (14)$$

The Mostinski model (1963) takes into account only the critical parameters of working fluid:

$$q'' = \left(0.1011 \cdot P_{crit}^{0.69}\right)^{3.33} \left(T_w - T_{sat}\right)^{3.33} P*^{3.33} \quad (15)$$

where:

$$P* = \left(1.8 \cdot \left(\frac{p}{P_{crit}} \right)^{0.17} + 4 \cdot \left(\frac{p}{P_{crit}} \right)^{1.2} + 10 \cdot \left(\frac{p}{P_{crit}} \right)^{10} \right)$$

A clear trend can be observed that increasing mass flux of liquid leads to the higher value of critical heat flux. The curves follow in a qualitative way the distribution of the boiling curve predicted by the equation due to Kutateladze. Higher mass fluxes also result in smaller wall superheats, however are shifted left.

Figures 5 to 7 show the heat flux on cooled surfaces with water , ethanol and HFE7100, for different liquid velocities. In every case increased fluid inlet velocity resulted in higher heat fluxes removal. Fig. 8 presents the stagnation point heat transfer coefficients in function of predicted values of heat transfer coefficient resulting from equation (12), in form of dimensionless Nusselt number. Application of (13) enables calculation of heat transfer coefficients with a very reasonable accuracy. In Fig. 9 presented are distributions of the ratio of experimental to theoretical Nusselt number in function of wall superheating, which shows that the model (12) is capable of capturing the trends in boiling heat transfer in impinging in the stagnation point, within reasonable error limits. A trend to predict lower HTC than experimental values, in higher wall superheat can be diminished by improvement of eq.(13).

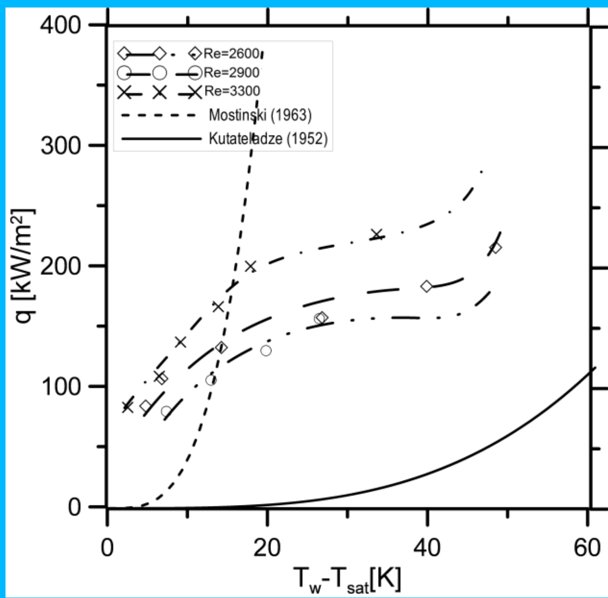

Fig. 5. Heat flux density in function of jet Reynolds number for boiling HFE7100 jet

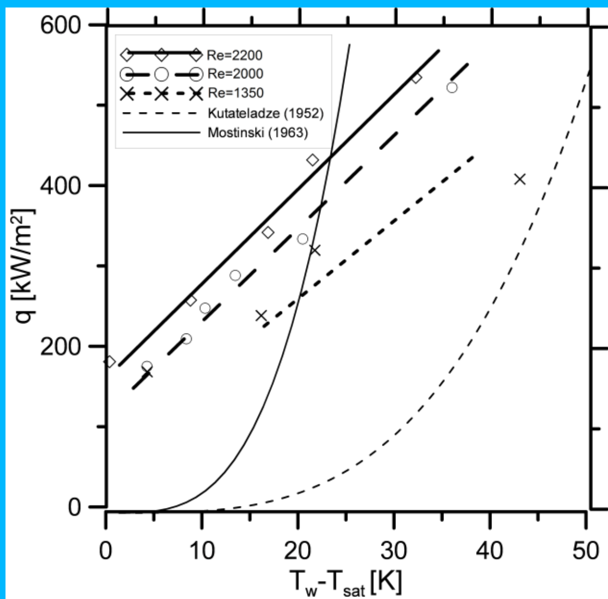

Fig. 6. Heat flux density in function of jet Reynolds number for boiling jet of ethanol

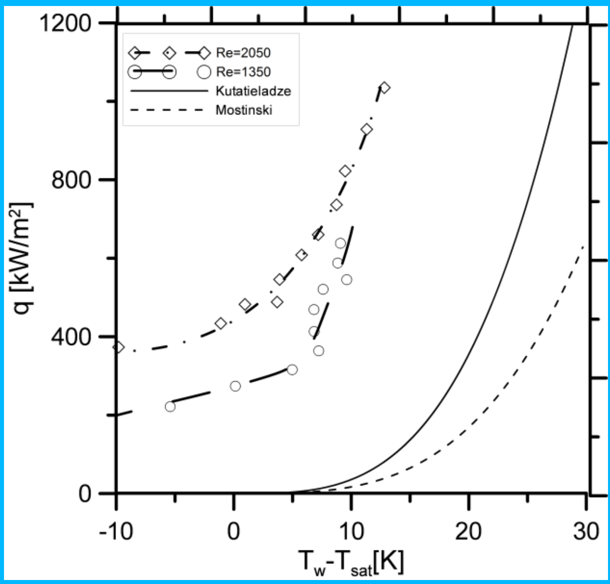

Fig. 7. Heat flux density in function of jet Reynolds number for boiling water jet.

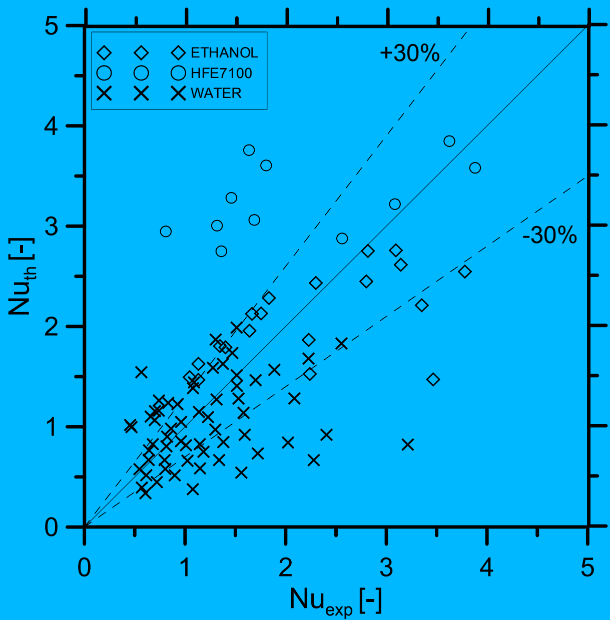

Fig. 8. Experimental and theoretical heat transfer coefficients calculated using (12)

CONCLUSIONS

In the paper presented have been theoretical and experimental studies of evaporating microjet impingement. Analytical model of stagnation Nusselt number was presented. Modifications and further development of this model will take place in course of further work.

Very important issue in calculation of heat transfer in microjet cooling is calculation of friction factors. It has been found that its value varies with liquid subcooling. Future activities should proceed in the direction of better determination of that parameter.

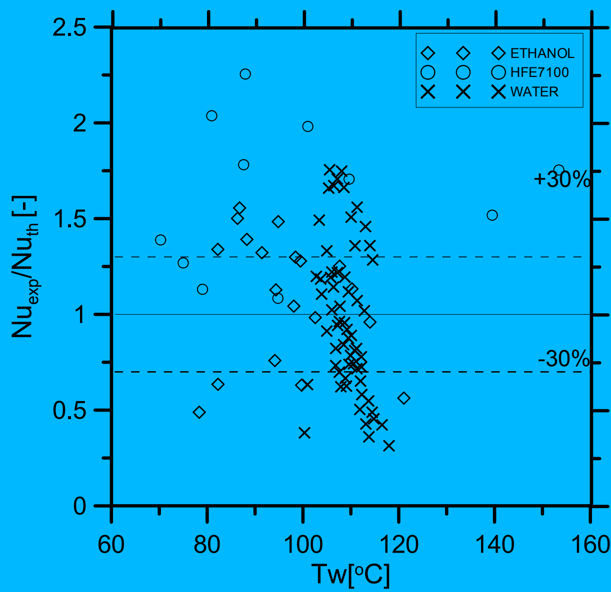

Fig. 9. Relation between experimental and theoretical heat transfer coefficients in function of wall temperature.

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