HEFAT2011
8th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
11 July – 13 July 2011
Pointe Aux Piments, Mauritius

MODEL OF HEAT TRANSFER IN THE STAGNATION POINT OF RAPIDLY EVAPORATING MICROJET

Dariusz Mikielewicz[1,*], Tomasz Muszyński[1], Jan Wajs[1], Jarosław Mikielewicz[2], Eugeniusz Ihnatowicz[2]
*Author for correspondence
[1] Gdansk University of Technology, Faculty of Mechanical Engineering,
Department of Energy and Industrial Apparatus,
ul. Narutowicza 11/12, 80-233 Gdansk, Poland
Email: Dariusz.Mikielewicz@pg.gda.pl
[2] The Szewalski Institute of Fluid-Flow Machinery PAS, ul. Fiszera 14, 80-952 Gdansk, Poland

ABSTRACT

The paper presents investigation into the single water microjet surface cooling producing a film of liquid water with phase change. Tests were conducted under steady state conditions. Theoretical model of surface cooling by evaporating microjet impingement in the stagnation area is developed. Experiments were conducted using nozzle size of 70 and 100μm. Results of experiments were compared with predictions of the model showing a good consistency.

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[1-6]. 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. 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

NOMENCLATURE

c_p	J/kg K	specific heat
D	m	jet diameter
G	kg/m^{2}s	mass flowrate
H	m	jets suspension over impinged surface
Ja	[-]	Jacob number
$\dot{m}$	kg/s	mass flux
Nu	[-]	Nusselt number
p	Pa	pressure
Pr	[-]	Prandtl number
r	J/kg	heat of evaporation
R	m	liquid film radius spreading
T	K	temperature
V	m/s	velocity
g	m/s^2	gravity
v,u	m/s	velocity components
α	W/m^{2}K	heat transfer coefficient
ρ	kg/m^3	density
δ	m	liquid film thickness
λ	W/mK	thermal conductivity
τ	s	time
ζ	[-]	contraction factor

Subscripts

l	liquid
v	vapour
w	wall parameter
s	saturation point parameter
0	inlet to nozzle

These correlations do not provide much insight into the underlying physical mechanisms involved in the boiling heat transfer problem. The alternative approach is to use mechanistic models. Generally speaking, a mechanistic model is usually based on the concept of surface heat flux partitioning. That means that the assumption that the surface heat flux comprises of multiple components. These components usually are the amount of heat used for direct evaporation to generate the bubbles, the amount of heat transferred through transient conduction to the liquid replacing the departing bubbles; and the amount of heat transferred through the enhanced convection due to the wakes generated by the emerging bubbles into the liquid. All mechanistic models of boiling heat transfer rely on experimentally developed relations or sub-models. These relations are used to correlate between the bubble departure diameter and release frequency and the other flow and surface parameters. 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.

None of the existing mechanistic models considered the case of boiling heat transfer under impinging jets, where the flow field is quite different from the case of parallel flow. Under an impinging jet, the velocity is normal to the surface at the stagnation zone. Downstream of the stagnation zone there is a parallel flow region where the hydrodynamic and thermal boundary layers are developing. One of the rare examples of modelling the heat transfer in stagnation point is the study due to Omar et al. (2009) [8]. The model predicts the total wall heat flux in the stagnation region of a free planar jet impinging on the flat surface. The model utilizes the concept of additional diffusion due to bubble-induced mixing. Authors accomplished a set of experiments in order to develop the required correlation between the additional diffusivity and the jet velocity, the degree of sub-cooling, and the degree of surface superheat. Model predictions give reasonable accuracy with experiments.

Main objective of this paper was to investigate the physical phenomena occurring on solid surfaces upon impingement of the single water microjet. Intense heat transfer in the impact zone of microjet has been examined and described with precise measurements of thermal and flow conditions of microjets. 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.

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 100 μm in width. The paper presents the investigation of a single water microjet cooling forming an evaporating liquid film on impingement surface. Tests were conducted under steady state conditions. Developed also has been a theoretical model of surface cooling by evaporating microjet impinging in the stagnation point, where the highest heat transfer coefficient occurs.

EXPERIMENTAL FACILITY

Present study shows results of steady state heat transfer experiments, conducted for nucleate boiling regime to obtain boiling curves. Studies enable also determination of critical boiling heat fluxes. Fig. 1 shows the schematic diagram of the test section. It consisted of the probe, water supplying system, the measuring devices and DC power supply. Distilled water was fed to the nozzle from a water tank, which also serves as the pressure accumulator. The water pressure in the test section was raised by air compressor. Desired water flow rate was obtained by sustaining the constant pressure of water 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. Volumetric water flow rate was measured at inlet and outlet from the cooling chamber with a graduated flask.

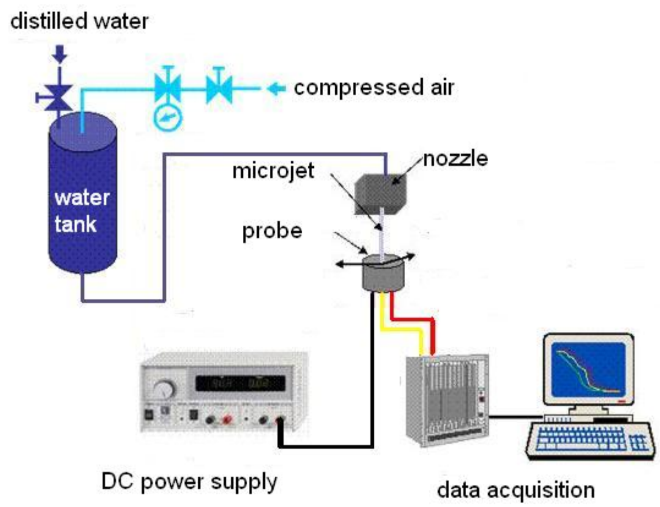

Figure 1 The schematic diagram of test section

The water tank was also fitted with preliminary water heater, which allows to adjust water jet temperature for cooling, i.e. control of liquid subcooling. The distance between the nozzle exit and the cooled surface was fixed at 25 mm. Figure 2 shows the cross-section of the probe.

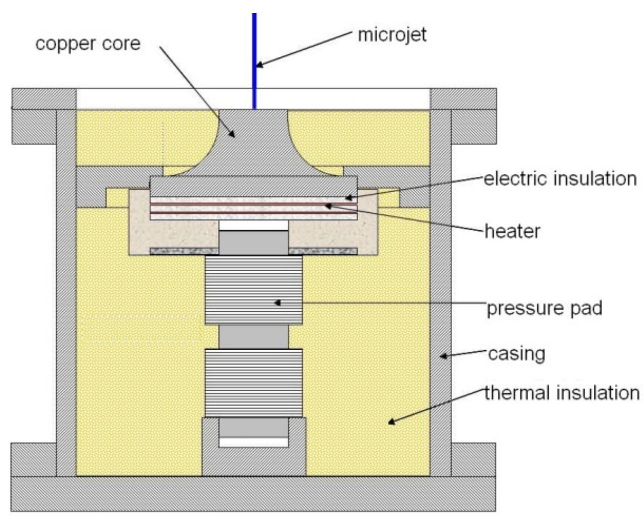

Figure 2 The cross-section of the probe

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 mounted at the bottom of the core. The whole set is thermally insulated 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.

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.

Nozzle construction allows to modify its dimensions. Two nozzle hydraulic diameters were used namely 0.101mm and 0.71mm. Experiments were conducted for the spacing of 25mm between the nozzle exit and impinging surface. Because of limited power supply, low water mass flux were used in order to obtain fully developed boiling and critical heat flux. Measurements were performed in three series with varying water subcooling.

THEORETICAL MODEL

In the analysis of accomplished experiments the simple theoretical model of impinging and evaporating liquid microjet 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, $\Delta \mathrm{p}_{\mathrm{d}} = \frac{\rho u_d^{\,2}}{2}$. In the analysis considered also could be the capillary effect $\Delta \mathrm{p}_{\mathrm{kap}} = \frac{\sigma}{D}$ (with D being the nozzle diameter) and the hydrostatic pressure drop $\Delta \mathrm{p}_{\mathrm{g}} = (\rho_l - \rho_v) g H$ (where H denotes the jet suspension over the surface). The latter two have however been omitted in the present study as they have been regarded as of secondary importance. Tthe total pressure acting on the surface yields:

$$\Delta \mathrm{p}_{\mathrm{c}} = \Delta \mathrm{p}_{\mathrm{d}} + \Delta \mathrm{p}_{\mathrm{kap}} + \Delta \mathrm{p}_{\mathrm{g}} \qquad (1)$$

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:

$$q\pi R^2 = \dot m_l \left[r + c_p \left(T_w - T_0 \right) \right] \qquad (2)$$

From (1) the impingement radius can be found:

$$R = \sqrt{\frac{\dot m (r + c_p \Delta T)}{\pi q}} \qquad (3)$$

The applicability for further calculations of the radius obtained from equation (3) has also been confirmed experimentally in the course of authors own experiment. 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 values of properties at atmospheric pressure and temperature 20^{o}C (cp=4184.3 J/kg K, hlv=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.

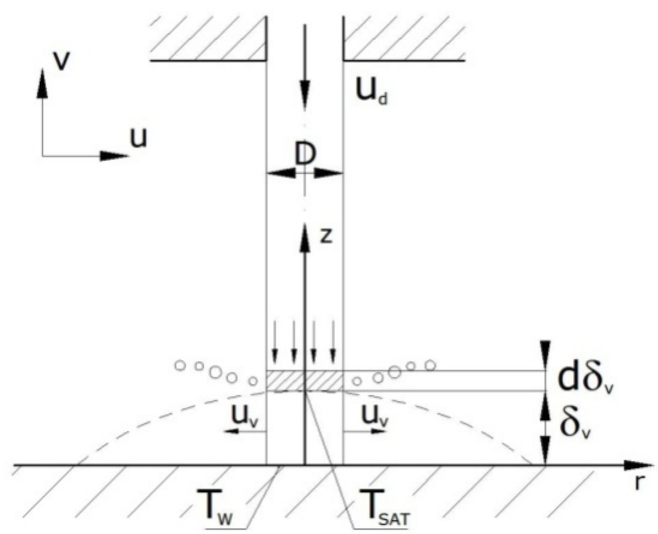

Figure 3 Scheme of rapidly evaporating microjet

The mass balance on evaporation surface yields:

$$d\dot m_l = \pi R^2 \frac{d\delta}{d\tau} \rho_l = d\dot m_v = 2\pi R \delta \rho_v V_v \qquad (4)$$

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

$$\frac{d\delta}{d\tau} = \frac{2}{R}\delta\frac{\rho_v}{\rho_l}V_v \tag{5}$$

In order to solve equation (5) we must provide the means for calculation of vapour velocity V_v. That can be done relying of the concept of Kutateladze who solved a similar problem in the large tank outflow 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 figure 3. According to Kutateladze, 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. Kutateladze solved such problem, as sketched in fig. 2 by consideration of Bernouli equation:

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

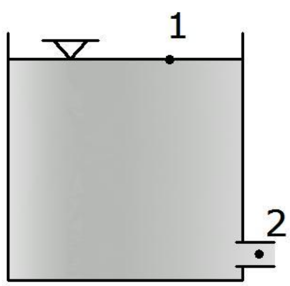

Figure 4 Outflow from the large tank through a small hole

Kutateladze solution is obtained assuming that u_1=0 and velocity at outlet is determined from expression:

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

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 equation:

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

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

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

In equation (9) $\Delta p_c = \frac{1}{2}\rho_l V_0^2$ is the microjet dynamic pressure and ζ - the contraction number. Substituting vapour velocity to equation (5):

$$\frac{d\delta}{d\tau} = \frac{2}{R}\delta\varsigma\sqrt{\frac{\rho_v}{\rho_l}}V_0 \tag{10}$$

The film thickness can be determined after incorporation of the radius of microjet influence available from (3):

$$\delta_v = \sqrt{\frac{\lambda_v(T_w - T_s)D}{4\varsigma\sqrt{\rho_l\rho_v}V_0\left[r + c_p(T_w - T_0)\right]}} \tag{11}$$

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

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

where $\Pr_{lv} = \dfrac{\nu_l}{a_{vl}} = \dfrac{\nu_l}{\dfrac{\lambda_v}{c_{pv}\sqrt{\rho_l\rho_v}}}$ $Ja_v = \dfrac{(T_w - T_s)c_{pv}}{r}$ and

$$Ja_l = \frac{(T_s - T_0)c_{pl}}{r}$$

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

$$\varsigma = (30 - 2.5V_0)\left(\frac{q}{q_{cr}}\right)^{2.2} \tag{13}$$

RESULTS OF CALCULATIONS

Figure 5 shows the experimental results of fully developed nucleate boiling heat transfer at the stagnation zone for impinging saturated water jet using the nozzle with D=0.101mm. Heat transfer data are plotted in the form of boiling curves for different liquid subcoolings. The Kutatieladze's empirical correlation is also presented in the figures 5 and 6 as a solid 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 can be simplified to [7]:

$$q_{cr} = 15\cdot(T_W - T_S)^{\frac{10}{3}}\left[\frac{W}{m^2}\right] \tag{14}$$

A clear trend can be observed that increasing subcooling of liquid leads to the increasing 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. The shift towards smaller wall superheats can also be observed, which increases with reduction of the nozzle size.

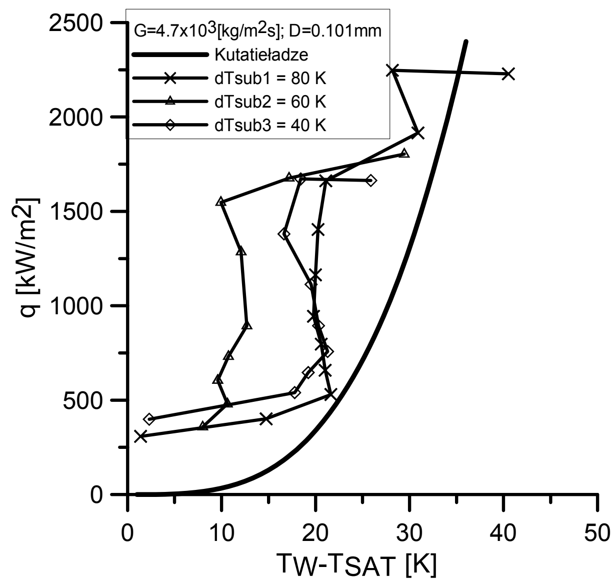

Figure 5 Nucleate boiling curves of evaporating water jet for the nozzle hydraulic diameter D=0.101mm

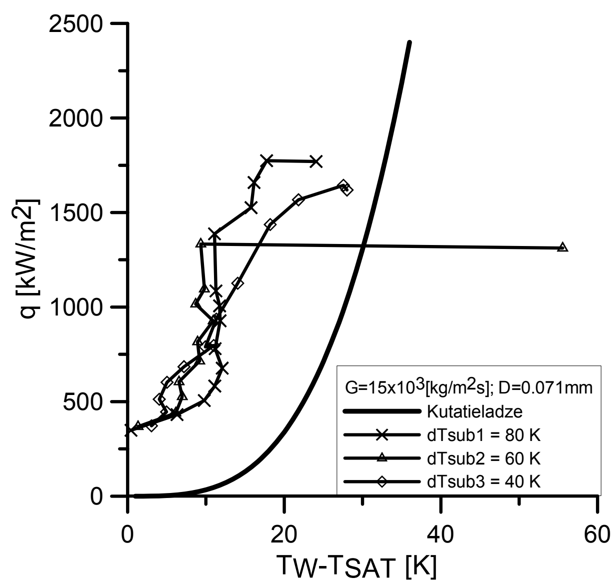

Figure 6 Nucleate boiling curves of evaporating water jet for the nozzle hydraulic diameter D=0.071mm

Fig. 7 presents the stagnation point Nusselt numbers in function of predicted values of Nusselt number resulting from equation (12). Application of (13) enables calculation of heat transfer coefficients with a very reasonable accuracy. In Fig. 8 presented are distributions of experimental to theoretical Nusselt number ratio in function of wall superheating, which shows that the model (12) is capable of appropriate capturing the trends in boiling heat transfer in impinging in the stagnation point.

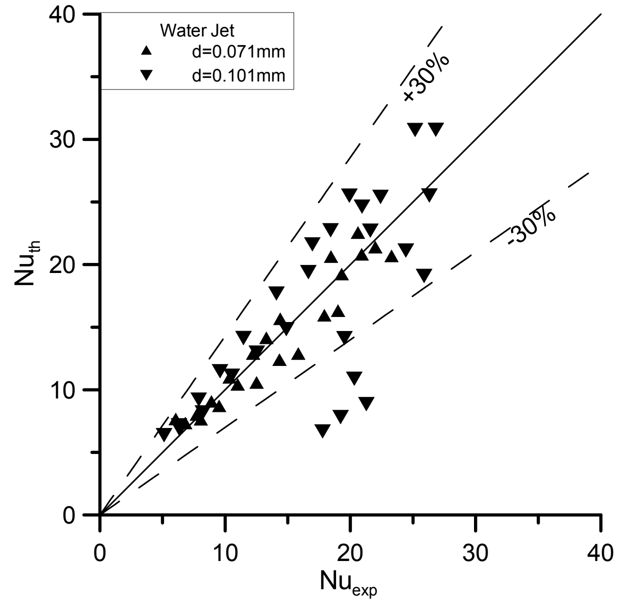

Figure 7 Relation between experimental and theoretical Nusselt numbers calculated using equation (12)

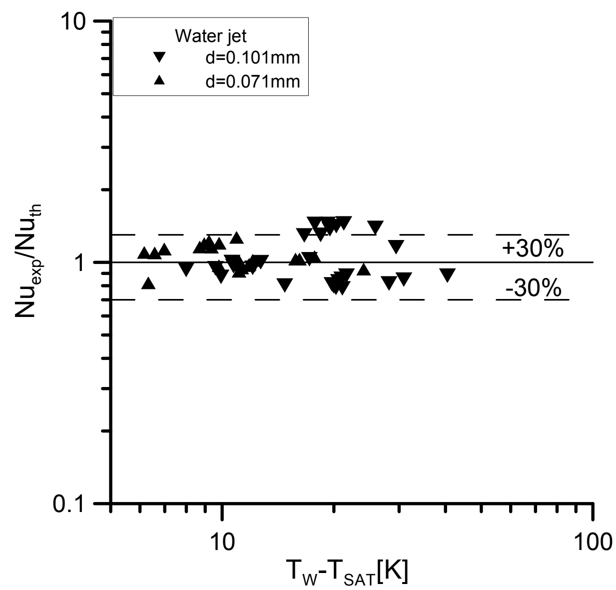

Figure 8 Relation between experimental and theoretical Nusselt number in function of wall superheat.

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.

ACKNOWLEDMENTS

The work presented in the paper was partially funded from the Ministry for Science and Higher Education research grant R06 01103 in years 2007-2010 and from a National Project POIG.01.01.02-00-016/08 *Model agroenergy complexes as an example of distributed cogeneration based on a local renewable energy sources.*

REFERENCES

1. Garimella S.V., Rice R.A., *Confined and submerged liquid jet impingement heat-transfer*, JHT vol.117, pp. 871–877, 1995.
2. Mikielewicz D., Muszynski T., *Experimental study of heat transfer intensification using microjets*, Int. Symp. on Convective Heat and Mass Transfer in Sustainable Energy, Hammamet, Tunisia, 2009.
3. Goldstein R.J., Timmers J.F., *Visualisation of heat transfer from arrays of impinging jets*, Int. J. Heat Mass Transfer, vol.25, pp. 1857-1868
4. Meyer M.T., Mudawar I., *Single-phase and two-phase cooling with an array of rectangular jets*, IJHMT vol.49 (2006) , pp. 17-29
5. Sung M.K., Mudawar I., *Single-phase and two-phase heat transfer characteristics of low temperature hybrid micro-channel/micro-jet impingement cooling module*, IJHMT vol.51 (2008), pp. 3882-3895
6. San J., Lai M., *Optimum jet-to-jet spacing of heat transfer for staggered arrays of impinging air jets*, IJHMT vol.44 (2001), pp. 3997-4007
7. Liu Z-H, Zhu Q-Z, *Prediction of critical heat flux for convective boiling of saturated water jet impinging on the stagnation zone*, Journal of Heat Transfer, vol. 124, pp. 1125-1130, 2002.
8. Omar A.M.T., Hamed M.S., Shoukri M., *Modeling of nucleate boiling heat transfer under an impinging free jet*, Int. Journal of Heat and Mass Transfer, vol. 52, 5557–5566, 2009.