Method of high heat flux removal by usage of liquid spray cooling

PRZEMYSŁAW SMAKULSKI[*]

Wroclaw University of Technology, Department of Thermodynamics,
Wybrzeże Wyspiańskiego 27, 50-370 Wrocław, Poland

Abstract   High heat flux removal are important issue in many perspective applications such as computer chips, laser diode arrays, or boilers working on supercritical parameters. Electronic microchips constructed nowadays are model example of high heat flux removal, where the cooling system have to maintain the temperature below 358 K and take heat flux up to 300 W/cm². One of the most efficient methods of microchips cooling turns out to be the spray cooling method. Review of installations has been accomplished for removal at high heat flux with liquid sprays. In the article are shown high flux removal characteristic and dependences, boiling critical parameters, as also the numerical method of spray cooling analysis.

Keywords: Critical heat flux; Spray cooling; Two-phase heat transfer; Numerical simulation

Nomenclature

c_{pl}	–	specific heat for liquid, J/kg K
c_{pv}	–	specific heat for vapor, J/kg K
d	–	liquid particle diameter, m
d_0	–	outlet spray nozzle diameter, m
d_{32}	–	Sauter mean diameter (SMD), m
h_{fg}	–	latent heat of vaporization, J/kg
h_{spray}	–	heat transfer coefficient, W/m² K
H	–	spray-to-surface distance, m
L	–	heat surface specific length, m

[*]E-mail: przemyslaw.smakulski@pwr.wroc.pl

$\dot{m}$	–	liquid mass stream, kg/s
p_{in}	–	liquid inlet pressure, Pa
Δp	–	pressure drop across spray nozzle, Pa
Re_{d0}	–	Reynolds number for cross-section d_0
$\dot{q}$	–	heat flux, W/cm^2
q_*	–	computational critical heat flux, W/m^2
$\bar{Q}''^2$	–	computational flowrate, m^3/m^{2}s
T_f	–	liquid temperature, K
T_{sat}	–	liquid saturation temperature, K
T_{surf}	–	surface temperature, K
ΔT	–	temperature difference between liquid and heating surface, K
We_{d0}	–	Weber number for cross-section d_0

Greek symbols

ρ_f	–	liquid density, m^3/kg
σ	–	surface tension, N/m
Θ	–	spray angle, rad
η	–	spray efficiency, %

1   Introduction

A recent high-power electronic devices and industry applications drive for very large thermal energy removal needs which are still growing parallel to increasing power consumption. The applications, where heat fluxes growing to enormous level are fuel rods in nuclear reactors, heating surfaces in boilers working on supercritical parameters, as also fragile electronics such as diode laser arrays and microchips [2]. In the near future the local heat fluxes generated by electronic circuits can reach a value of about 300 W/cm^2 [1], where permissible working temperature should be below 358K [1].

High heat fluxes is possible to remove by use of new cooling techniques. The best known includes: microchannel heat exchanger [3], coolant flow through porous material [4] or jet cooling [1,5] as also the spray cooling. Liquid spray cooling is characterized by the best balance between ability to remove high heat fluxes and uniformity distribution of the refrigerant [4]. Moreover, is possible to remove the heat by the cooling medium directly from the heating surface without any additional body between fluid and surface. This could cause in reduction of unnecessary heat transfer resistance of intermediate material.

Application of the new technology of heat flux removal is a consequence of research for enhancement of the heat transfer by the forced convection (for gases), and heat reception during the pool boiling. The main parameter determining boiling is its characteristic curve, which shows a relation

of absorbing heat flux to temperature difference (overheating degree). This overheating is defined as a temperature difference between heating surface (T_{surf}), and saturation temperature for the liquid (T_{sat}). Regime of boiling, where heat flux from temperature difference is desired from the heat transport point of view, is determined by nucleate boiling and it goes to its limited value called the critical heat flux (CHF). The CHF point on the boiling curve divides the process between nucleate boiling and film boiling. Heat transfer coefficient for the film boiling are significantly lower than that in nucleate boiling. Decreased heat transfer coefficient has a consequence in the previously mentioned temperature difference, which is much higher in the film boiling region. Therefore it could cause a damage of the heating surface material or even its melting.

The spray cooling improves the boiling characteristic and reaching CHF higher value, which will fulfill the needs of high heat flux removal and protect the heating surface before unexpected failure.

2 Dispersed phase heat transport mechanism

2.1 Liquid spraying

Due to the type of used energy to spray liquids it could be highlighted: pressure spray nozzles use the energy of the liquid itself; pneumatic – use kinetic energy of the gas to dispersed liquid phase; rotational – use centrifugal force of the liquid on the surface of rotary element [6]. In the described examples of heat transfer it is mainly applied spray nozzles, which use the energy of liquid itself – pressure stream and rotary-stream sprays. The use of these specific spray nozzles is characterized by relatively small energy consumption needs to the dispersed the kilogram of liquid (from 2–4 W/kg [6]), and simplified construction. Less used type of spray nozzle, in the light of energy consumption (approx. 50–60 W/kg [6]), are pneumatic sprays, mainly used for denser liquids, where it is necessary to implement additional gaseous phase to the dispersed fluid. Additional gaseous phase can also supports liquid evaporation thanks to refrigerant vapor partial pressure reduction [4].

2.2 Application of the spray cooling

The most beneficial from the economical point of view, are supposed to be, the use of liquid cooling with phase change. Moreover, to enhancement the
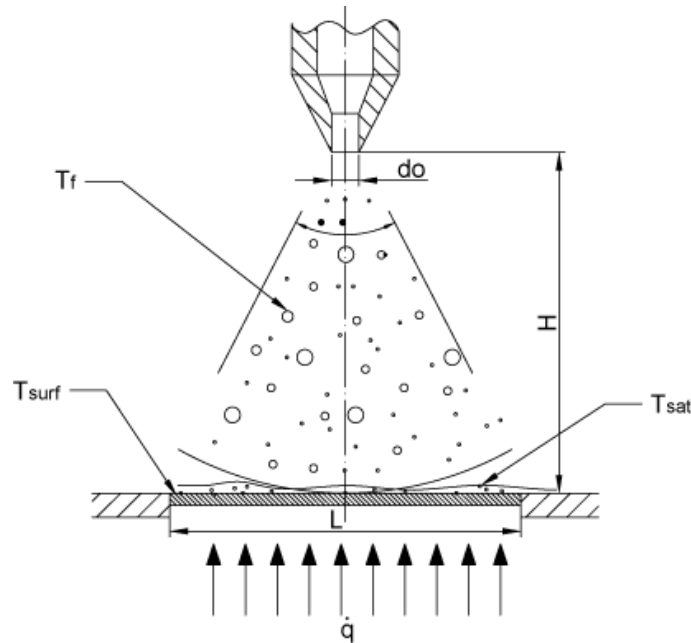
Figure 1. Scheme of the pressure, stream liquid spray nozzle with the characteristic values
(description of the symbols in nomenclature section).

heat transport is begun to be used the solid-vapor phase change by micro-
particles sprays [7]. Thanks to the phase change is possible to getting
additional cooling potential, which supports absorption of heat form the
heating element. Two-phase spray cooling mechanism is complicated and
not fully understood. Closest to real process of spray cooling was described
in paper [8], where two-phase spray cooling was divided into nucleate boil-
ing (in both surface and second nucleation), natural convection and direct
evaporation from the gas-liquid interface. The concept of second nucleation
was introduced to simplify description of heat transfer improvement thanks
to the liquid spray nozzles.

Analytical heat transfer equations for different regions were published
in [9]. For the research purposes, the heat transfer coefficient is defined on
the basis of tests, as a ratio of a heat flux, $\dot{q}$, generating on heating surface
to temperature difference, $\Delta T = T_{surf} - T_{sat}$, between surface temperature
and temperature of liquid

$$h_{spray} = \frac{\dot{q}}{T_{surf} - T_{sat}} \; .$$

(1)

2.3   Efficiency of spray cooling with two-phase heat transport phenomena

In the case of spray cooling is possible to define the method of heat transfer efficiency. Efficiency is defined as a relation of generated heat flux to quantity of heat accumulated in the refrigerant mass stream

$$\eta = \frac{\dot{q}}{\dot{m}\left[c_{pl}(T_{sat} - T_f) + h_{fg} + c_{pv}(T_{surf} - T_{sat})\right]} \; . \tag{2}$$

Absorption of the heat is described by specific heat of the liquid, latent heat of evaporation, and vapor specific heat [4].

2.4   Research of the critical heat flux

Most research of the spray cooling are based on determining maximal heat removal capabilities by defining CHF. The CHF is a point on the boiling curve, and is individual for each boiling process. Therefore, many investigations on CHF in spray cooling are based on experiment.

In papers [10,11] for the refrigerants FC-72, FC-87, PF-5052 and water there was proposed a correlation for determining the CHF analytically,

$$q* = 2.3 \left[\frac{\rho_f \bar{Q}''^2 d_{32}}{\sigma}\right]^{-0.35} \; , \tag{3}$$

where

$$\bar{Q}''^2 = 2\Delta P/\rho_f \; . \tag{4}$$

This equation is designated with the absolute mean error equal to 14% and includes in its structure the Weber number, which determines ratio of inertia forces to surface tension forces for liquid

$$\mathrm{We}_{d32} = \frac{\rho_f \bar{Q}''^2 d_{32}}{\sigma} \; . \tag{5}$$

In other words, Weber number shows directly the liquid spraying intensity on the heating element surface. It was proved in [10–12], that the higher value of Weber number leads most probably to higher CHF. On the other hand, the efficiency of spray cooling (Eq.(2)) is inversely proportional to this number.

Spray cooling depends on many factors, which determine quality of the heat removal and its CHF value. Parameters, which, except cooling liquid type, influence the heat transfer coefficient are among others:

- liquid spraying quality – Sauter mean diameter (SMD) of drops [8,10,13],

- distance between spray nozzle outlet and heating surface [14],

- subcooling of liquid [2,8,13],

- mass stream of refrigerant [8,10,13,15,16],

- surface roughness [4,15].

Factors influencing the liquid spraying quality except types and geometrical sizes of spray nozzle, are properties of the spraying liquid. These includes among others: a density, kinematic viscosity and surface tension. Value of liquid spraying quality in two-phase flow for the case of interchange of mass and heat is SMD determined as [17]

$$d_{32} = \frac{\Sigma \Delta n d^3}{\Sigma \Delta n d^2} \ , \tag{6}$$

where diameter d_{32} is depended from geometrical spray nozzle outlet diameter (full cone spray) [10]

$$\frac{d_{32}}{d_0} = 3.67 \left[\mathrm{We}_{d0}^{\frac{1}{2}} \mathrm{Re}_{d0} \right]^{-0.259} \ . \tag{7}$$

It was shown in [8,10,13], that the average droplets size, established on the basis of SMD, influenced on CHF. The smaller SMD could cause higher CHF, which is favorable due to protection of the heating surface before overheating or even melting. Mudawar and Estes [14] have been researched the dependence of dielectric cooling mediums (FC-72 and FC-87) from distance of spray nozzle outlet to the heating surface. They have found optimal height, for which spray nozzle should be installed in the aim to reach the maximal value of the heat flux. The relationship connects spray angle Θ and specific length on the acting surface, L

$$H \tan(\Theta/2) = L/2 \ . \tag{8}$$

In the case of subcooling before application to the nozzle, temperature of the fluid is cooled down to the level under its saturation state. Therefore, it gives additional cooling power by liquid fraction increasing after throttling process on the discharge nozzle. Nevertheless, it could turn out that extra portion of latent heat of evaporation, applied with the same mass stream (in comparison of not subcooled fluid), could cause unfavorable increasing of the

surface temperature. It was shown in [2], that overheating degree exceeds its optimum level in the relations of heating surface temperature. For the presented refrigerant R-600a, and parameters: inlet pressure 3.9×10^5 Pa, saturation pressure 1.9×10^5 Pa, and constant heat flux equal to 72 W/cm^2, optimum subcooling for this fluid is 5.8 K. Authors of this article point out that optimal level of subcooling could be different for different boundary conditions and types of spray nozzles.

Increasing of mass stream during cooling process is connected indirectly with pressure drop on spray nozzle, changing of the droplet flow velocity as well as spraying ratio of liquid [8,16]. Considering the influence of different mass stream ratio, intermediate range of phenomena should be determined in the heat acquisition. In overall case, increased mass stream preferably affects the value of CHF, increasing its scope.

3   Spray cooling review

In the following review are presented laboratory solutions for the spray cooling technology which used different refrigerants and types of spray nozzles. In the publications [2,9,11,13,15,16] single liquid nozzles were used, and in the works [8,18,19] the multinozzle spray cooling were used. Due to enhanced heat removal assessment, the CHF value was taken into consideration for comparative purposes. That value gives an information about maximal possible heat flux taken from the surface.

Table 1 shows the physical parameters of the spray cooling with respect to the CHF value. The review includes different refrigerant types, from the industrial refrigerants to the natural coolants, i.e., water. Except CHF value for liquid, the surface temperature, the heat transfer coefficients values, as also the average droplets diameter SMD, or pressure drop through spray nozzle were shown.

Direct comparison of the given results is troublesome a bit, as a reason of applied research methods variety, or even performer types of liquid sprays. Nevertheless, in the above list it could be noticed that biggest CHF occurs for the case of water and methanol. In the same time, it could be observed, that for the higher CHF, the temperature of the heating surface is also higher. Surface temperature value below desired microchips working temperature (lower than 358 K), are achievable by FC-87, R-600a, R-134a, PF-5052, R-22. Unfortunately, in the present study none of these refrigerants did reach the heat flux of 300 W/cm^2, or even closer to the critical

its value. Closest in performance refrigerant to the required parameters seems to be refrigerant R-22, which maintains the lowest surface temperature among others liquids, at relatively high CHF value. Unfortunately, under the Montreal Protocol published in 1987, and also on the basis of the very harmful influence on the natural environment, R-22 was withdrawn from the official distribution.

Table 1. Spray cooling review.

No.	Author	fluid	CHF	h_{spray}	T_{surf}	ΔT	p_{in}	Δp	Flowrate	SMD	η
			W/cm^2	$W/cm^2 K$	K	K	$\cdot 10^5$ Pa	$\cdot 10^5$ Pa	$\cdot 10^{-6}$ m^3/s	μm	%
1	[8]	FC-72	83.5	2.23	362.0	309.0	-	3.1	4.44	38.7	27.1
2	[8]	FC-87	90.0	2.06	355.0	313.0	-	2.4	4.08	36.5	30.6
3	[2]	R-600a	110.0	3.00	304.5	NA	2.9	1.1	NA	60-90	NA
4	[19]	FC-72	150.0	1.60 - 2.40	NA	NA	NA	-	16.6	NA	NA
5	[18]	R134a	165.0	3.9	321.0	NA	-	4.7	NA	182 - 145	25.0
6	[11]	PF-5052	236.7	NA	345.4	321.4	-	1.2	10.10	174.0	NA
7	[20]	R-22	276.1	7.00	299.8	310.0	8.0	4.6	NA	NA	NA
8	[13]	water	320.0	NA	420.0	394.0	NA	NA	0.81	85-100	NA
9	[13]	water	322.0	NA	418.0	394.0	NA	NA	0.53	85-100	NA
10	[9]	FC-77	349.0	NA	402.4	375.4	-	1.7	23.90	183.0	NA
11	[8]	methanol	490.0	6.45	386.0	333.0	-	2.4	7.26	62.3	13.3
12	[8]	water	500.0	9.78	393.0	323.0	-	2.4	4.98	88.8	11.6
13	[15]	water	638.0	NA	NA	NA	-	2.6	NA	583.0	NA
14	[16]	water	945.7	8.38	400.8	385.8	5.5	-	NA	68.7	NA

NA – not available

4 Numerical simulation

Spray cooling heat transfer process was described in Section 2.2. For detailed knowledge of the phenomena, the numerical simulation was applied. The preliminary simulation of heat transfer in spray cooling was performed using ANSYS CFX 14.5 software [21] for two-phase fluid flow. For the cooling medium was chosen water at temperature equal to 300 K.

The simple computational 3D model of spray cooling is presented in Figs. 2 and 3. The numerical calculation are focused on temperature field on the heating surface. Droplets of water fall down through the nozzle to the hot surface, where particles hit the wall and become a part of the wall film. Water particles have been based on Lagrangian particle tracking model for multiphase flow [21]. Water as an homogenous mixture of liquid and vapour, and air defined as continuous fluid were implemented in this

code. Boundary conditions, as also the major geometrical values, are estimated on the basis of the information included in papers related to the water spray cooling [8,15]. Outlet nozzle diameter, d_0, is equals to 1 mm, and spray-to-surface distance, H, equal to 30 mm. Heat flux on the heating surface is set to 100 W/cm^2. This value is well below CHF value for water sprays (see Tab. 1) and it was chosen only to visualize the temperature propagation on the surface as well as its level in stagnation point and other parts of the surface.

Table 2. Numerical simulation conditions.

Analysis type	Steady-state
Ambient pressure	101.300 kPa
Ambient temperature	297 K
Inlet fluid velocity, v_{in}	2 m/s
Fluid mass stream, $\dot{m}$	3.7×10^{-6} kg/s
Sauter mean diameter (SMD)	111 μm
Fluid temperature, T_f	300 K
Heat flux, $\dot{q}$	100 W/cm^2

Presented simulation shows a heat flux propagation structure, by usage of spray cooling – radially spreading temperatures on heating surface. The visualization of water two-phase spray cooling shows the temperature field on the surface (Fig. 3), as also the velocity field in the cross-sectional area (Fig. 2).

5   Summary

High heat fluxes removal task is interesting area of the research, founding the use of many applications, mainly in the field of electronics or power industry. Presented method of usage the spray nozzle to cooling is profitable for the economic point of view, but unfortunately very difficult to unequivocal description. In the presented paper the agents influence the quality of absorbing heat were characterized as also a example of installation working on different cooling agents together with its critical values of heat removal. Application of numerical simulation will help to better understand the heat removal phenomena and to knowledge developement in the topic of spray cooling.
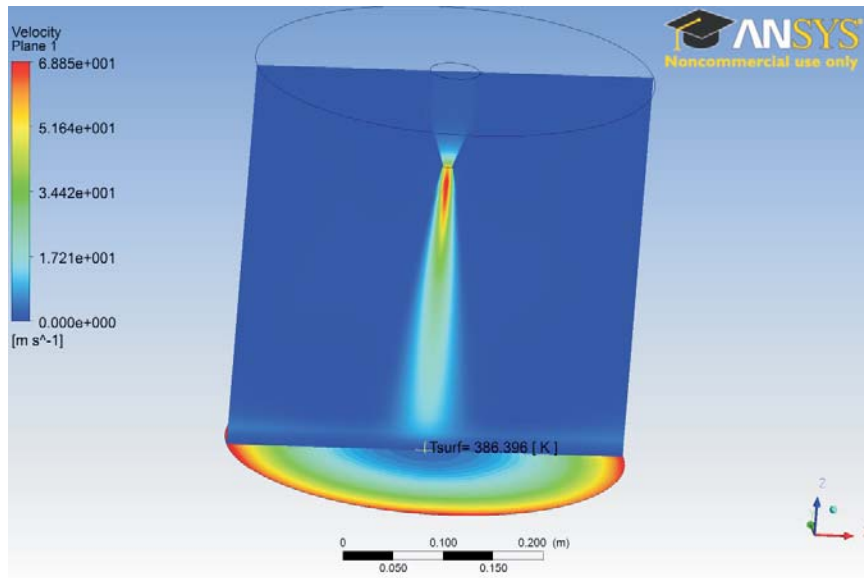
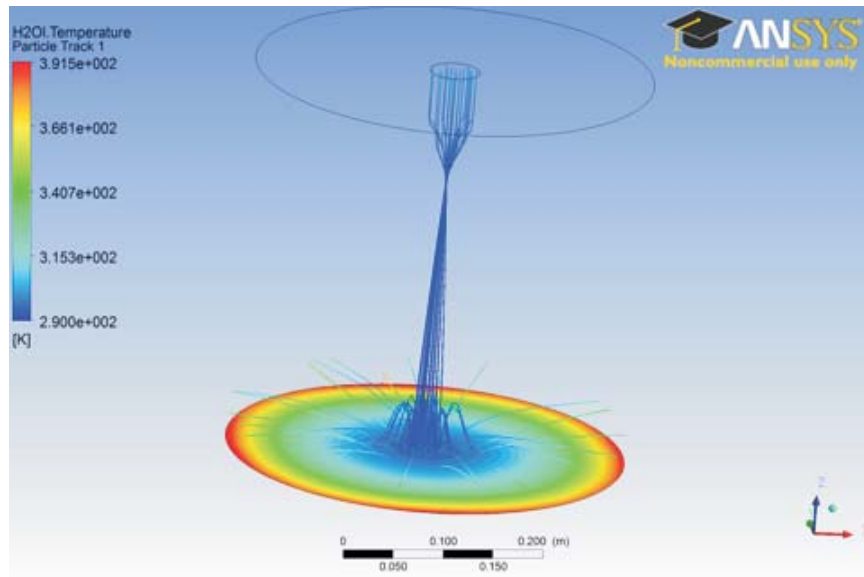Figure 2. Numerical simulation of the sprayed fluid velocity vector for water.

Figure 3. Temperature and fluid droplets distribution visualization.

References

[1] Agostini B. *et al.*: *State of the art of high heat flux cooling technologies.* Heat Transf. Eng. **28**(2007), 4, 258–281.

[2] Chunqiang S. *et al.*: *Development and experimental investigation of a novel spray cooling system integrated in refrigeration circuit.* Appl. Therm. Eng. **33**–**34**(2012), 246–252.

[3] Piasecka M., Maciejewska B.: *Application of the finite element method to the determining of boiling heat transfer coefficient for minichannel flow.* Arch. Thermodyn. **34(**2013), 1, 55–69.

[4] Kim J.: *Spray cooling heat transfer: The state of the art.* Int. J. Heat Fluid Fl. **28** (2007), 753–767.

[5] Mikielewicz D., Muszyński T., Mikielewicz J.: *Model of heat transfer in the stagnation point of rapidly evaporating microjet.* Arch. Thermodyn. **33**(2012), 139–152.

[6] Orzechowski Z., Prywer J.: *Spraying Liquids*, 2nd edn. WNT, Warszawa 1991 (in Polish).

[7] Ishimoto J., Oh U., Tan D.: *Integrated computational study of ultra-high heat flux cooling using cryogenic micro-solid nitrogen spray.* Cryogenics **52(**2012), 505–517.

[8] Lin L., Ponnappan R.: *Heat transfer characteristics of spray cooling in a closed loop.* Int. J. Heat Mass Transf. **46**(2003), 3737–3746.

[9] Mudawar I., Valentine W.S.: *Determination of the local quench curve for spray-cooled metallic surfaces.* J. Heat Treat. **7**(1989), 107–121.

[10] Estes K.A., Mudawar I.: *Correlation of Sauter mean diameter and critical heat flux for spray cooling of small surfaces.* Int. J. Heat Mass Transf. **38**(1995), 16, 2985–2996.

[11] Rybicki J.R., Mudawar I.: *Single-phase and two-phase cooling characteristics of upward-facing and downward-facing sprays.* Int. J. Heat Mass Transf. **49**(2006), 5–16.

[12] Hsieh S. *et al.*: *Spray cooling characteristics of water and R-134a. Part I: nucleate boiling.* Int. J. Heat Mass Transf. **47**(2004), 5703–5712.

[13] Visaria M., Mudawar I.: *Effects of high subcooling on two-phase spray cooling and critical heat flux.* Int. J. Heat Mass Transf. **51**(2008), 5269–5278.

[14] Mudawar I, Estes K.A.: *Optimizing and predicting CHF in spray cooling of a square surface.* J. Heat Transf. **118**(1996), 672–679.

[15] Ortiz L., Gonzalez J.E.: *Experiments on steady-state high heat fluxes using spray cooling.* Exp. Heat Transf. **12**(1999), 215–233.

[16] Chen R., Chow L.C., Navedo J.E.: *Effects of spray characteristics on critical heat flux in subcooled water spray cooling.* Int. J. Heat Mass Transf. **45**(2002), 4033–4043.

[17] ORZECHOWSKI Z.: *Two-Phase Flows, One-Dimensional, Steady Adiabatic.* PWN, Warszawa 1990 (in Polish).

[18] TAN Y.B. *et al.*: *Multinozzle spray cooling for high heat flux applications in a closed loop system.* Appl. Therm. Eng. **54**(2013), 372–379.

[19] SHEDD T.A.: *Next generation spray cooling: High heat flux management in compact spaces.* Heat Transf. Eng. **28**(2007), 2, 87–92.

[20] HOU Y.: *Experimental study on phase change spray cooling.* Exp. Therm. Fluid Sci. **46**(2013), 84–88.

[21] *ANSYS CFX-Solver Modeling Guide*, 2009.