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# Transient spray cooling: Similarity of dynamic heat flux for different cryogens, nozzles and substrates

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ABSTRACT

Spray cooling is widely used in industries. Understanding of the complex surface heat transfer characteristics is essential to enhance cooling efficiency. In the present study, surface heat transfer of transient spray cooling with different cryogens, nozzles, and substrates were investigated. Minimum surface temperature reached −46.1 °C, −55.9 °C, and −57.9 °C and maximum surface heat flux values were 294.9, 364.1, and 377.4 kW/m$^2$ for R134a, R407C, and R404A, respectively, on an epoxy resin block. Results proved that R404A has the best cooling capacity. Compared with that of the straight nozzle, $q_{\max}$ of the expansion-chamber nozzle increased from 266.9, 364.1, and 403.9 kW/m$^2$ to 339.3, 442.8, and 445.1 kW/m$^2$ or 16.6%, 18.6%, and 16.7% for R134a, R407C, and R404A, respectively. Transient cooling could be divided into two stages, namely, fast boiling cooling and film evaporation cooling. The similarity of dynamic heat flux with different cryogens, nozzles, and substrates was observed, and the dimensionless correlation was proposed by the spray Biot number, which is defined as the ratio between the internal thermal resistance of the substrate and the surface convective heat transfer resistance. The dimensionless Reynolds number $Re_l$ and Fourier number $Fo_l$ were proposed to represent the maximum heat flux ($q_{\max}$) and the corresponding time ($t_{\max}$), respectively. By coupling the Jakob number with $Re_l$ and the droplet Weber number ($We$), dimensionless correlations of maximum heat flux and the corresponding time were proposed, through which it is indicated that $We$ number was the key factor in spray cooling.

## 1. Introduction

Compared with traditional surface cooling, spray cooling atomizes working media into small droplets sprayed onto a surface and induces effective cooling because of large phase change in latent heat. Spray cooling has been widely applied in many fields, including microelectronics, metallurgy, machining, aerospace engineering, and biomedical engineering, to name a few [1–3] because of its advantages, such as high heat flux, uniform cooling, and resource efficiency.

Steady spray cooling (SSC) of long spurt durations in seconds is usually conducted in microelectronics, metallurgy, and machining areas that require homogeneous and deep cooling. Compared with SSC, research on transient spray cooling (TSC) is relatively scarce. In aerospace engineering, the power of electronic equipment could vary at any time. TSC can match the power change of equipment with the optimal frequency of injection [4]. In addition, intermittent TSC is widely used because it can accelerate film evaporation, thereby enhancing heat transfer. In biomedical engineering, cryogen spray cooling (CSC) has been used to minimize the risk of epidermis heat injury during laser treatment of port wine stain (PWS) birthmarks [5,6]. The spurt duration of CSC is shortened (by approximately tens of milliseconds) to prevent epidermis cold injury. Therefore, the transient effect must be considered. In contrast to traditional spray cooling, TSC combines strong atomization, droplet evaporation, and surface boiling and is therefore a complex transient phase-change heat transfer process. Moreover, research on TSC can facilitate the transient change of surface temperature and heat flux, which is also important to reveal the heat transfer mechanism of SSC.

Given the complex heat transfer of spray cooling, many different surface heat correlations have been proposed by researchers. However, most correlations can only be applied to specific spray media, cooling substrates, and working conditions for SSC; examples of these correlations include surface heat flux ($q$), maximum heat flux ($q_{\max}$), and convective heat transfer coefficient ($h$). With regard to surface heat flux, Rybicki and Mudawar [7] employed PF-5052 as a working medium to investigate the influence of spray orientation on spray cooling. They determined that the orientation

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## Nomenclature

| | |
|---|---|
| $a$ | thermal diffusion coefficient ($m^2/s$) |
| $c$ | specific heat capacity (kJ/kg K) |
| $d_{32}$ | Sauter mean diameter (μm) |
| $D$ | nozzle diameter, inner diameter of the expansion chamber (mm) |
| $g$ | acceleration of gravity ($m/s^2$) |
| $h$ | convective heat transfer coefficient ($W/m^2$ K) |
| $h_{fg}$ | latent heat of evaporation (J/kg) |
| $L$ | spray distance, length of expansion chamber (mm) |
| $m''$ | liquid mass flux per unit area (m/s) |
| $u_0$ | droplet velocity (m/s) |
| $U$ | liquid outlet velocity (m/s) |
| $V$ | volume of expansion chamber ($mm^3$) |
| $t$ | time (ms) |
| $T_0$ | initial temperature of substrate (°C) |
| $q$ | surface heat flux ($kW/m^2$) |
| $r_s$ | spray diameter (mm) |
| $Q''$ | averaged volumetric flux per unit area (m/s) |
| $\theta_c(t)$ | temperature allowance $T_c(t) - T_0$ |
| $L^{-1}$ $[1/H(s)]$ | laplace inverse transform of the transfer function, $f(t)$ |
| $S_q(r)$, $S_T(r)$ | residual error of heat flux and surface temperature |
| Sqrt($\lambda\rho c$) | heat absorption or dissipation coefficient |
| $Pr_l$ | Prandtl number, $c_l\eta_l/\lambda_l$ |
| $h^*$ | spray convective heat transfer coefficient, $q/(T_0 - T_l)$ |
| $Nu_{d32}$ | Nusselt number, $qd_{32}/((T - T_l)\lambda_l)$ |
| $Nu_D$ | Nusselt number, $hD/\lambda_l$ |
| $Re_{d32}$ | Reynolds number, $m''\ d_{32}/\eta_l$ |
| $Re_D$ | Reynolds number, $\rho_l UD/\eta_l$ |
| $Re_l$ | Reynolds number, $\frac{q_{\max}}{\eta_l h_{fg}}\sqrt{\frac{\sigma}{g(\rho_l-\rho_v)}}$ |
| $We$ | Weber number, $\rho_l u_0^2 d_{32}/\sigma$ |
| $We_{Q''}$ | Weber number, $\rho_l Q''^2 d_{32}/\sigma$ |
| $Ja$ | Jakob number, $c_l\Delta T_{sub}/h_{fg}$ |
| $Bo_m$ | Bond number, $q_{\max}L/\eta_l h_{fg}$ |
| $Bi^*$ | Spray Biot number, $h^*\delta/\lambda_s$ |
| $Fo_s$ | Fourier number, $a_s t/\delta^2$ |
| $Fo_l$ | Fourier number, $a_l t_{\max}/d_{32}^2$ |

*Greek symbols*

| | |
|---|---|
| $\lambda$ | heat conductivity (kW/m K) |
| $\rho$ | density ($kg/m^3$) |
| $\tau$ | liquid film resistance time (ms) |
| $\delta$ | thickness of the substrate (m) |
| $\sigma$ | surface tension (N/m) |
| $\eta_l$ | kinetic viscosity (Pa/s) |
| $\Delta T_{sub}$ | droplet subcooling degree (°C) |

*Subscripts*

| | |
|---|---|
| $b$ | boiling point |
| $sat$ | saturation state |
| $v$ | vapor |
| s | surface |
| $r$ | real value |
| $k$ | calculated value |
| $n$ | expansion-chamber nozzle |
| max | maximum value |
| min | minimum value |

showed no measurable influence on spray cooling regimes. A heat flux correlation with an overall mean absolute error of 13.1% was proposed by creating a database to facilitate a generalized correlation for single-phase heat transfer:

$$Nu_{d_{32}} = 4.70 Re_{d_{32}}^{0.61} Pr_l^{0.32}, \tag{1}$$

where $Nu_{d32}$ and $Re_{d32}$ are the Nusselt and Reynolds numbers, respectively, characterized by the droplet Sauter diameter $d_{32}$, $Nu_{d32} = qd_{32}/((T - T_l)\lambda_l)$, $Re_{d32} = m''d_{32}/\eta_l$. $Pr_l$ is the Prandtl number defined by the thermal properties of liquid (spray medium, here PF-5052), $Pr_l = c_l\eta_l/\lambda_l$. $T$ and $T_l$ are the surface and liquid inlet temperatures, respectively, and $m''$ is the liquid mass flux per unit area, which is based on circular impact area of spray. $\eta$, $\rho$, $c$, and $\lambda$ are the kinetic viscosity, density, specific heat, and heat conductivity, respectively. Eq. (1) illustrates that $Nu_{d32}$ has functional dependence on $Re_{d32}$ and $Pr_l$; the larger the impingement force of the droplet ($Re_{d32}$, $Pr$), the temperature difference between surface and droplet ($T - T_l$), and the heat transfer coefficient ($\lambda_l$), the larger the surface heat flux ($q$). A large droplet ($d_{32}$) will decrease heat flux and inhibit surface heat transfer because of the small specific surface area. Eq. (1) was proposed for SSC; thus, it cannot be used directly for TSC. Moreover, this correlation does not consider phase change.

Mudawar and Valentine [8] used water as a working medium and a hot metal surface as a cooling substrate to investigate the boiling heat transfer of spray cooling in alloy quenching. They observed that heat flux was affected by the surface and water temperatures and was irrelevant to the droplet diameter and spray flow rate. A correlation for nucleate boiling was proposed:

$$q = 1.87 \times 10^{-5}(T - T_l)^{5.55}. \tag{2}$$

Eq. (2) indicates that the surface heat flux in nucleate boiling is only affected by the temperature difference between the surface wall and the droplet. This correlation is quite simple, but is only appropriate for nucleate boiling ($T_l$ = 22.5–23.5 °C).

The peak of surface heat flux, $q_{\max}$, also called critical heat flux (CHF), has been regarded as the criterion for the cooling ability of spray and boiling crises. Many researchers have proposed numerous correlations for CHF. Visaria and Mudawar [9] investigated the dominant factors of CHF for FC-77 spray cooling. He proposed a correlation that considered the effect of different nozzles, spray flow rate, spray orientation, and subcooling degree of the working medium. Direct use of this correlation may cause large error for the multivariable nature. Cabrera et al. [10] investigated the nuclear boiling of water and the effect of surface roughness on CHF. He obtained a correlation comprising maximum heat flux, droplet Weber number, and surface roughness. This correlation had a confidence level of 95% with a maximum error of ±15% in the experimental data. Estes and Mudawar [11] employed a copper disk as the substrate for spray cooling and proposed a correlation applicable for FC-72, FC-78, and water with an average error of ±25%:

$$q_{\max} = \rho_g Q'' h_{fg}\left(\frac{\pi}{4}\right)2.3\left(\frac{\rho_l}{\rho_g}\right)^{0.3} We_{Q''}^{-0.35}\left(1 + 0.0019\left(\frac{\rho_l}{\rho_g}\right)Ja\right), \tag{3}$$

where $We_{Q''}$ is the droplet Weber number characterized by $d_{32}$, $We_{Q''} = \rho_l Q''^2 d_{32}/\sigma$. $Ja$ is the Jakob number, $Ja = c_l\Delta T_{sub}/h_{fg}$, the ratio between the cooling capacity provided by subcooling droplets and the latent heat. $\rho$ is the density, $Q''$ is the volumetric flux averaged