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Heat transfer correlation for intermittent spray impingement: A dynamic approach

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ABSTRACT

The work presented here investigates a new approach in the development of heat transfer empirical correlations for intermittent spray impingement, based on simultaneous measurements of the spray droplets characteristics and the surface thermal behavior. Conventionally, heat transfer correlations for spray impingement do not consider the temporal variations of droplets characteristics. However, in applications using intermittent sprays (internal combustion engines, cryogen spray cooling or micro-processor thermal management), the spray transient behavior suggests that heat transfer predictions may be improved using a dynamic approach. Additionally, the impact of multiple consecutive injections on a heated surface implies a certain degree of interaction, depending on the frequency of their intermittency. If the time between consecutive injections is shorten, the result is the formation of a liquid film which mitigates phase-change and privileges a single-phase heat transfer over a two-phase. This suggests that heat transfer correlations for spray impingement should take the spray unsteadiness and the multiple injections interaction degree into account. The dynamic approach here suggested presupposes the identification of systematic periods characterizing the spray dynamic behavior and, once identified, the development of a heat transfer correlation for each period. The analysis ends with a comparison between the dynamic heat transfer correlation with a correlation obtained using the conventional approach and a significant improvement in heat transfer predictions is achieved if the spray dynamic nature is considered.

1. Introduction

Spray cooling is recognized as a well known method for high heat flux removal. Recently, the application of an intermittent spray in thermal management systems has been proposed as a new technological concept for the enhancement of heat transfer, providing the system with an improved performance, as well as an active control over heat transfer mechanisms [1]. From another point of view, further enhancements in spray cooling technology have been suggested implying the active control of the spray characteristics using synthetic-jets [2]. This emphasizes a current trend in the development of efficient spray cooling systems by introducing a 'control' component in the system's design. This opens the research question of what is the relation between the ability to control the heat transfer process and the mechanisms which actually govern the process.

From the 'control' point of view, in intermittent spray cooling, Panão and Moreira [3] identified the 'duty cycle' as a parameter which expresses the interface between controlling the surface temperature and the physical mechanisms associated with spray impaction. The 'duty cycle' (DC) is defined as the percentage of the entire cycle time where fluid is injected, corresponding to the pulse duration (Δt_{inj}), and expressed as $DC = \Delta t_{inj} \times f_{inj}$. From the 'mechanisms governing heat transfer' point of view, some apparent contradictions can be found in the literature about what parameters actually govern heat transfer. For example, while Arcoumanis and Chang [4], Bernardin et al. [5] and Chen et al. [6] argued that droplet axial velocity plays a dominant role in governing local, time-resolved heat transfer, in Estes and Mudawar [7], and Rybicki and Mudawar [8] it is argued that volumetric flux is of much greater significance in characterizing spray heat transfer than drop velocity. In Sawyer et al. [9], Yao and Cox [10] and Cabrera and González [11] arguments are presented for the spray mass flux, and in Rini et al. [12] for the droplet number flux as the main parameters governing heat transfer. In Pikkula et al. [13] it is the Weber number ($\rho U_d^2 D_d/\sigma$), and in Chen and Hsu [14] it is the initial wall superheating degree ($T_w - T_b$) that is considered to be of primary importance for the heat flux removal in spray cooling. Therefore, there is still much uncertainty as to what are the actual parameters which mainly affect spray/wall heat transfer in general. Eventually,

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Nomenclature

c_p	specific heat ($\mathrm{J\,kg^{-1}\,K^{-1}}$)
Ca	capillary number
D	droplet diameter (μm)
DC	duty cycle $= \Delta t_{\mathrm{inj}} f_{\mathrm{inj}}^{-1} \times 100\%$ (%)
f	frequency (Hz)
h_c	heat transfer coefficient ($\mathrm{W\,m^{-2}\,K^{-1}}$)
h_{fg}	latent heat of evaporation ($\mathrm{J\,kg^{-1}}$)
Ja	Jakob number
k	thermal conductivity ($\mathrm{W\,m^{-1}\,K^{-1}}$)
L_w	plate thickness (mm)
La	Laplace number
Nu	Nusselt number
p	pressure (bar)
Pr	Prandtl number
$\dot{q}''$	heat flux ($\mathrm{W\,m^{-2}}$)
$\dot{q}''_{d,n}$	number flux of droplets ($\mathrm{\#\,m^{-2}\,s^{-1}}$)
r_{disc}	disc radius (mm)
r_{tc}	thermocouple radius (mm)
Re	Reynolds number
T	temperature (°C)
t	time (s)
U	droplet axial velocity ($\mathrm{m\,s^{-1}}$)
We	Weber number
Z	axial distance (mm)

Greek letters

β	thermal effusivity ($\mathrm{J\,m^{-2}\,K^{-1}\,s^{-1/2}}$)
χ	evaporated mass fraction
δr	interaction radius of multiple drop impacts
Δt	time interval (ms)
ΔT	temperature difference (°C)
λ	Dimensionless number flux
Λ	average liquid film thickness (μm)
μ	dynamic viscosity ($\mathrm{kg\,m^{-1}\,s^{-1}}$)
θ	top to bottom surface temperature difference
ρ	specific mass ($\mathrm{kg\,m^{-3}}$)
σ	surface tension (N m)
ξ	$= L_w \beta k_w^{-1}$ ($\mathrm{s^{1/2}}$)

Subscripts

b	boiling point
c	convection
er	electric resistance
f	fluid
imp	impingement
inj	injection
w	wall
wb	wall to boiling

Abbreviations

LFS	leading front of the spray
PDA	Phase Doppler Interferometry
SS	steady spray
ST	spray tail

all these parameters are simultaneously present and interrelated, and depending on the experimental or operating conditions, one of them governs heat transfer, while others may govern less. Nevertheless, this usually ends in the lack of universality of any derived empirical correlation, which can only be resolved through incremental improvements. Moreover, to accurately model the energy exchanges in spray cooling, it is important to understand the main effects underlying the interaction between the impinging spray characteristics and the heat flux removed from the impinging heated surface, which is the context of the work presented here.

In cases where the heat flux removal is transient, such as in intermittent spray cooling, the interaction between the impinging spray and the heat removal is affected by the spray dynamic behavior along each injection cycle, and by the interaction between consecutive injections. Therefore, it is also worth questioning about the role of the spray dynamics in governing heat transfer phenomena, which requires simultaneous measurements of the spray characteristics and the heat transferred in the cooling process [15].

This work uses such simultaneous measurements to investigate a novel approach in the development of heat transfer correlations for spray impingement, based on the spray dynamic behavior. The paper is structured as follows. After briefly reviewing what has been the conventional approach in the development of heat transfer empirical correlations in Section 2, an experimental setup and the diagnostic techniques used in the characterization of spray impingement heat transfer are described in Section 3. Section 4 analyzes the simultaneous measurements of the spray droplets characteristics and the resulting heat transfer upon their impaction. The results in this section suggest that devising heat transfer correlations with intermittent sprays should be approached from a dynamic point of view. The section ends with a physical interpretation and comparison with a heat transfer correlation derived from the same data, however, using a conventional approach. Finally, Section 5 contains some concluding remarks.

2. Review on the development of heat transfer correlations for spray cooling

This section briefly reviews the empirical correlations found in the literature for spray impingement heat transfer and advances a physical interpretation of their outcome. The elementary way of establishing a spray/wall heat transfer correlation is through dimensional analysis. The simplest form of an arbitrary function in dimensional analysis, for developing a spray/wall heat transfer correlation, was used in the modeling scheme of Eckhause and Reitz [16] as

$$f(h_c, \rho, k, \mu, c_p, \Lambda) = 0 \tag{1}$$

The assumptions behind equation (1) depart from a boundary layer flow analogy applied to a single drop impact which floods the wall, forms a thin liquid film of average thickness Λ, thus, the heat transfer is correlated with this liquid film. Considering the 6 parameters in (1) and 4 independent dimensions (kg, m, s, K), two dimensionless groups are determined: i) the Nusselt number, $\mathrm{Nu} = h_c\Lambda/k$; ii) and the Prandtl number, $\mathrm{Pr} = \mu c_p/k$. The final correlation is written as

$$Nu = aPr^b \tag{2}$$

which in Eckhause and Reitz [16] assume the values $a = 3.32$ and $b = 0.333$, for a wetting regime. The fact that $b > 0$ denotes the positive influence of drop impact velocity – determinant to the velocity boundary layer – over the thermal boundary layer developed in the liquid film. In the non-wetting case, particularly in the

Leidenfrost regime, according to the classical boiling theory, the $Nu = 2.0$ [16]. However, this approach does not take into account the influence of the sprays droplets characteristics (velocity and size) in the heat transfer process, even if these, ultimately, determine the liquid film thickness Λ. Therefore, if the characteristic length Λ is replaced by the size of droplets (D_d) and the velocity is included in the arbitrary function (1),

$$f(h_c, \rho, k, \mu, c_p, D_d, U_d) = 0 \tag{3}$$

it implies that a new dimensionless group is introduced in correlation (2), known as the Reynolds number ($Re = \rho U_d D_d/\mu$)

$$Nu = aPr^b Re^c \tag{4}$$

Contrary to the heat conduction assumption in the boundary layer analogy expressed by equation (2), in the correlation above, spray impingement heat transfer is governed by the single-phase convection of the liquid film and the inertial impact exerted by the spray droplets on the heated surface. Rybicki and Mudawar [8] derived such correlation for spray cooling with $a = 4.7$, $b = 0.32$ and $c = 0.61$. The hydrodynamics implicit in correlation (4) is the build-up of a liquid film, through which occurs a single-phase heat transfer. The positive exponent associated with 'Re' ($c > 0$) confirms the importance of the spray inertial impact force for the heat transfer enhancement.

In Arcoumanis and Chang [4], the heat transfer upon a Diesel spray impact is considered to depend on the amount of liquid deposited and spreading over the heated surface, determined by the surface tension σ, therefore, this effect should be considered. When the surface tension is introduced in the arbitrary function (3), another dimensionless group is included in the correlation (4), the Weber ($=\rho U_d^2 D_d/\sigma$) number, such that

$$Nu = aPr^b Re^c We^d \tag{5}$$

The correlation which fitted the experiments in [4] had $a = 0.34$, $b = -0.33$, $c = -0.53$ and $d = 0.94$. The physical interpretation of the exponents in (5) suggests that, similarly to correlation (4), the 'Pr' and 'Re' numbers are associated with a convective heat transfer through the liquid film formed upon spray impaction and, this time, have a negative impact on heat transfer ($b,c < 0$). In fact, Arcoumanis and Chang [4] stated that during/after wall impingement, a liquid film has deposited on the surface, but due to its thinness, it is more likely to evaporate than to nucleate and boil. Since heat transfer enhancements are more favorable through boiling, it could explain the interpretation given for the negative exponents of 'Pr' and 'Re'. Considering this, since $d > 0$, it means the impact energy enhances heat transfer compared with convective heat transfer. The fact that more impact energy implies a greater piercing of cooler liquid droplets into the film could explain the enhancement of heat transfer rates [17].

None of the correlations above explores that different heat transfer regimes may occur in spray/wall heat transfer [1], namely through two additional parameters: the superheating degree ΔT_{wb} and; the latent heat of vaporization h_{fg}. If we include these parameters in the arbitrary function $f(h_c, \rho, k, \mu, c_p, D_d, U_d, \sigma, \Delta T_{wb}, h_{fg}) = 0$, and apply an elementary dimensional analysis, considering mass, length, time, temperature and energy as the independent dimensions, four dimensionless groups are expected. The three of correlation (5) and the fourth is the Jakob number ($Ja = c_p \Delta T_{wb}/h_{fg}$), such that

$$Nu = aPr^b Re^c We^d Ja^e \tag{6}$$

The Jakob number expresses the relative importance between the maximum sensible heat absorbed by the liquid, associated with single-phase heat transfer, and the latent heat absorbed to accomplish phase-change, which is associated with a two-phase heat transfer [18]. This dimensional analysis has been used in Panão and Moreira [19] to derive an empirical correlation for the impact of single injections on a heated surface and the results were $a = 3.4 \times 10^{-5}$, $c = 1.51$ and $e = -0.254$. The analysis on these exponents confirms the importance of the spray inertia upon impact ($c > 0$) and evidences the relevancy of a two-phase heat transfer through the phase-change mechanism ($e < 0$), since a higher the latent heat of vaporization (h_{fg}) implies a lower 'Ja'. The 'We' number is absent from the correlation ($d = 0$) because in a dense spray, such as in port-fuel injection, Choi and Yao [20] showed its negligible importance and; finally, due to the multiple drop impacts, the importance of the boundary layer flow is mitigated, and the 'Pr' number becomes 3–4 orders of magnitude lower than 'Ja', remaining rather constant due to negligible changes in the thermophysical properties during a single injection event, thus, producing a worse correlation if considered and expressing its negligible influence on the heat transfer process ($b = 0$).

One element which has been absent in the work reviewed above is the relevancy, or not, of the spray dynamic behavior for the heat transfer behavioral pattern. Therefore, in order to investigate the influence of the spray intermittency, Panão and Moreira [21] have presented experimental evidence supporting a relation between the spray dynamic characteristics, within the injection cycle, and the surface thermal behavior. Moreover, in Moreira and Panão [1], the degree of interaction between consecutive injections has been shown to significantly influence heat transfer. However, these two issues have not yet been considered in the development of heat transfer correlations for spray cooling. This is the objective of the dynamic approach proposed in Section 4.

The following section describes the experimental setup and the diagnostic techniques used to produce the simultaneous measurements of droplets characteristics and wall heat flux.

3. Experimental setup and diagnostic techniques

This section describes the experimental setup and the diagnostics techniques used for acquiring simultaneously measurements of the spray droplets characteristics and the surface thermal behavior, including a brief description of the method for calculating the instantaneous wall heat flux calculation with an imposed heat source.

The flow configuration is that of a spray striking perpendicular onto a flat aluminum disc with a 10 mm radius (r_{disc}), 12 mm thick (L_w), and an electric resistance with a copper plate which uniformly distributes heat to the disc. The spray is generated by a pintle-type port-fuel injector with 0.79 mm of pintle diameter inserted in a cylindrical hole of 0.9 mm diameter.

The injection frequency, pulse duration and number of injections are also software controlled by the NI5411 arbitrary function generator from National Instruments. The surface temperature is measured by three Medtherm eroding-K-type thermocouples assembled in the disc, with a response time of 10 μs and spaced by 4 mm (r_{tc}) with the first thermocouple located at the disc center as depicted in Fig. 1.

Thermocouples' signals are sampled at 50 kHz with a NI6024E National Instruments DAQ board plus a BNC2120, and the electrical signal is amplified with a gain of 300 before processing. The sample rate of 50 kHz is chosen to fully capture the surface temperature variations occurring after spray impaction. Inaccuracies in temperature due to electronic noise increase as the surface temperature decreases and were found to be smaller than 1%.

Droplet size, velocity and number flux are measured with a two-component Particle Dynamic Analyzer (PDA) system from

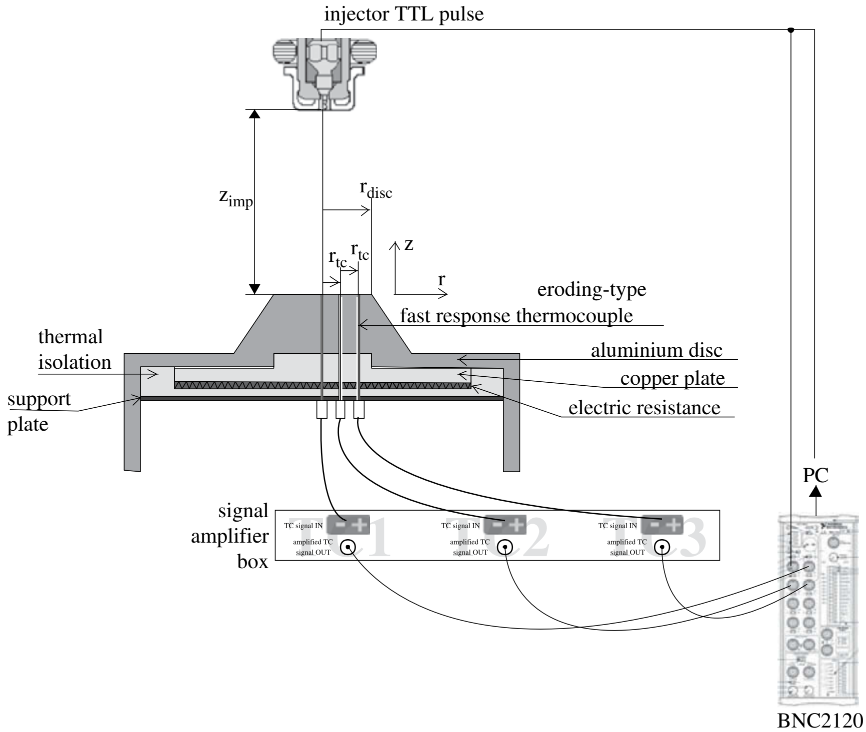

Fig. 1. Schematic of the heated disc setup and the temperature measurement system.

DANTEC Dynamics (Denmark) consisting of a 55× transmitting optics, a 57×10 PDA receiving optics, and a 58N10 Covariance processor. More details about the phase-Doppler optical system configuration can be found in [21]. Phase-Doppler measurements are made at a normal distance of 2 mm above the target, which is the smallest distance the probe volume is able to approach the impinging surface without blocking the intersected laser beams used to measure the axial velocity component. The droplets impinging on the wall are distinguished from those produced by secondary atomization mechanisms through the axial velocity component (U). A positive axial velocity indicates impinging droplets, otherwise, secondary droplets. The number flux of impinging droplets depends on the effective cross-section area of the PDA measurement volume, which is calculated according to Roisman and Tropea [22] and Panão and Moreira [21]. Error propagation analysis resulted in uncertainties of less than 10% for mass flux quantities (further details in [21]). Finally, the synchronization is achieved through the electronic signal sent to open the injector, which is simultaneously used in the Phase-Doppler processor unit and in the temperature data acquisition system.

The uncertainty in the measured drop size and velocity distributions is based on an information theory approach developed in Panão and Moreira [23] which assesses the accuracy obtained in the discrete size distribution relatively to is 'actual' function, assuming it exists. The maximum mean errors in this accuracy are 6.75% for the velocity distributions *per* time-bin and 6.2% for the size distribution.

The liquid used in the experiments is HFE-7100 and its thermophysical properties, at the reference temperature of 22 °C, are listed in Table 1: specific mass (ρ); dynamic viscosity (μ); surface tension (σ); boiling temperature (T_b); liquid specific heat (c_p); and latent heat of vaporization (h_{fg}). The variation of HFE-7100 properties with temperature is provided by the manufacturer (3 M).

Table 2 contains all the case studies used in this work. In each case, 100 series of 40 injection cycles are used in a phase-average analysis. In this analysis, the data acquired is restricted to the interaction radius of multiple drop impacts δr (500 µm) and each time-bin δt_{bin} has 0.5 ms.

A methodology has been formulated to calculate the instantaneous wall heat flux from surface temperature measurements and considering an imposed heat flux as an important boundary condition. It has been previously shown that lateral heat conduction is less than 0.08% of the axial heat flux (for more details see [1]), therefore, the problem is considered unidirectional. The experiments performed in this work consider the heat transfer across a finite slab heated by a constant heat source, $\dot{q}''_{er}$, for which the equation of energy is expressed by

$$\frac{\partial \theta}{\partial t} = \left(\frac{k_w}{\beta}\right)^2 \frac{\partial^2 \theta}{\partial z^2} + \frac{k_w}{L_w \beta^2}\dot{q}''_{er}, \quad \begin{cases} \theta(z,t) = [T_w(z,t) - T_w(L_w, 0)] \\ \beta = \sqrt{\rho k c_p} \end{cases}, \tag{7}$$

where T_w is the wall temperature, z is the spatial coordinate perpendicular to the wall, β is the thermocouple thermal effusivity, k_w is the wall thermal conductivity and L_w is the disc thickness. The following initial and boundary conditions are considered in the solution of equation (7):

Table 1
Fluids thermophysical properties.

Fluid (22 °C)	HFE-7100
ρ (kg/m^3)	1488
μ (kg/m/s)	5.7×10^{-4}
σ (mN · m)	13.6
c_p (J/kg/K)	1177
T_b (°C)	61
h_{fg} (kJ/kg)	111.6

Table 2
Working conditions used in the experiments performed with HFE-7100.

Case	f_{inj} (Hz)	Δt_{inj} (ms)	p_{inj} (bar)	ΔT_{wb} (°C)	Z_{imp} (mm)
1–16	10, 20, 30, 60	5, 7.5, 10, 15	3	43.7	50
17–18	10	5, 10	4	43.7	50
19–22	10, 30	5	3	43.7	30, 40
23–26		10, 30 5	3	20, 70	30

$$\theta(z,0) = \underbrace{\theta(L_W,0)}_{=0} - \frac{\dot{q}''_{er}}{k_W}(L_W - z), \tag{8}$$

$$\dot{q}''(0,t) = -k_W \frac{\partial\theta(0,t)}{\partial z}, \tag{9}$$

$$\frac{\partial\theta(L_W,t)}{\partial z} = -\frac{\dot{q}''_{er}}{k_W}, \tag{10}$$

The term $\theta(L_W,0)$ on the initial condition (8) is null, and the boundary condition (10) for the imposed heat flux on the bottom surface assumes at $t=0$ a linear the temperature profile along the z-axis. To solve the differential equation (7), Laplace transformations are applied and the final general solution is given by

$$\theta(z,s) = A\,\mathrm{e}^{z\sqrt{s/(k_w/\beta)^2}} + B\,\mathrm{e}^{-z\sqrt{s/(k_w/\beta)^2}} + \frac{\dot{q}''_{er}}{s^2}\left(\frac{L_W - z}{k_W} + \frac{k_W}{L_W\beta^2}\right). \tag{11}$$

From the Laplace transform of the initial condition (8) and boundary conditions (9) and (10), the solution of (11) at the wall ($z=0$) is given by

$$\theta(0,s) = \frac{\dot{q}''(0,s)}{\beta\sqrt{s}}\frac{\mathrm{e}^{2\xi\sqrt{s}}+1}{\mathrm{e}^{2\xi\sqrt{s}}-1} + \frac{\dot{q}''_{er}}{\beta s^{3/2}}\left(\frac{1}{\sqrt{s}}\left(\frac{1}{\xi}-\xi\right) + \frac{1}{s}\frac{\left(\mathrm{e}^{\xi\sqrt{s}}-1\right)^2}{\mathrm{e}^{2\xi\sqrt{s}}-1} - \frac{2\mathrm{e}^{\xi\sqrt{s}}}{\mathrm{e}^{2\xi\sqrt{s}}-1}\right), \quad \xi = \frac{L_W\beta}{k_W} \tag{12}$$

The equation above should now be explicitly expressed for $\dot{q}''(0,s)$, however, some of the terms in (12) include ratios with exponential functions elevated to an expression containing the Laplacian variable $\sqrt{s}$. However, because $\sqrt{s}$ directly reflects the frequency of temperature variations at the top surface according to Chen and Nguang [24], for values of the order of 0.5–1 kHz, such as in the present work, one can approximate the solution for $\dot{q}''(0,s)$ as being bounded by $\sqrt{s}\to\infty$. In this case, the limits in the ratios with exponential functions are determined and the explicit form for the wall heat flux from (11) can be simplified into

$$\dot{q}''(0,s) = \beta s\theta(0,s)\frac{1}{\sqrt{s}} - \dot{q}''_{er}\left(\frac{1}{s^{3/2}}\left(\frac{1}{\xi}-\xi\right) + \frac{1}{s^2}\right). \tag{13}$$

When the frequency of temperature variations is small, for example, in the absence of spray impact, such that $\sqrt{s}\to 0$, the limits of the ratios with exponential functions lead to $\dot{q}''(0,s) = \dot{q}''_{er}$, thus showing the consistency of the derivation above.

The convolution theorem is used to find the analytical solution of (13), which, after some mathematical manipulation, becomes

$$\begin{cases} \dot{q}''(0,t) = \dfrac{\beta}{\sqrt{\pi}}\displaystyle\int_0^t \frac{\partial\theta(0,\tau)}{\partial\tau}\frac{1}{\sqrt{t-\tau}}\mathrm{d}\tau - \dot{q}''_{er}\left(t + \frac{2}{\xi}\sqrt{\frac{t}{\pi}}\right) \\ \xi = \dfrac{L_W\beta}{k_W} \end{cases}. \tag{14}$$

It is noteworthy that the solution above for the case $\dot{q}''_{er} = 0$, reduces to that already derived by Reichelt et al. [25] for semi-infinite slab in the absence of heat sources. Equation (14) shows that the effect of $\dot{q}''_{er}$ over $\dot{q}''(0,t)$ increases for thin wall thicknesses ($L_W \to 0$) and as heat transfer unfolds in time. Furthermore, the numerical implementation of equation (14), in order to obtain the instantaneous surface heat transfer from the measured temperature data, follows the approach developed in Schultz and Jones [26] allow obtaining the following final expression,

$$\begin{cases} \dot{q}''(0,t_n) = \dfrac{2\beta_{t_n}}{\sqrt{\pi\delta t}}\displaystyle\sum_{i=1}^{n}\frac{\theta(0,t_i)-\theta(0,t_{i-1})}{\sqrt{n-i}+\sqrt{n-i+1}} - \dot{q}''_{er}\left(t_n + \frac{2}{\xi_{t_n}}\sqrt{\frac{t_n}{\pi}}\right) \\ \xi_{t_n} = \dfrac{L_W\beta_{t_n}}{k_W} \end{cases} \tag{15}$$

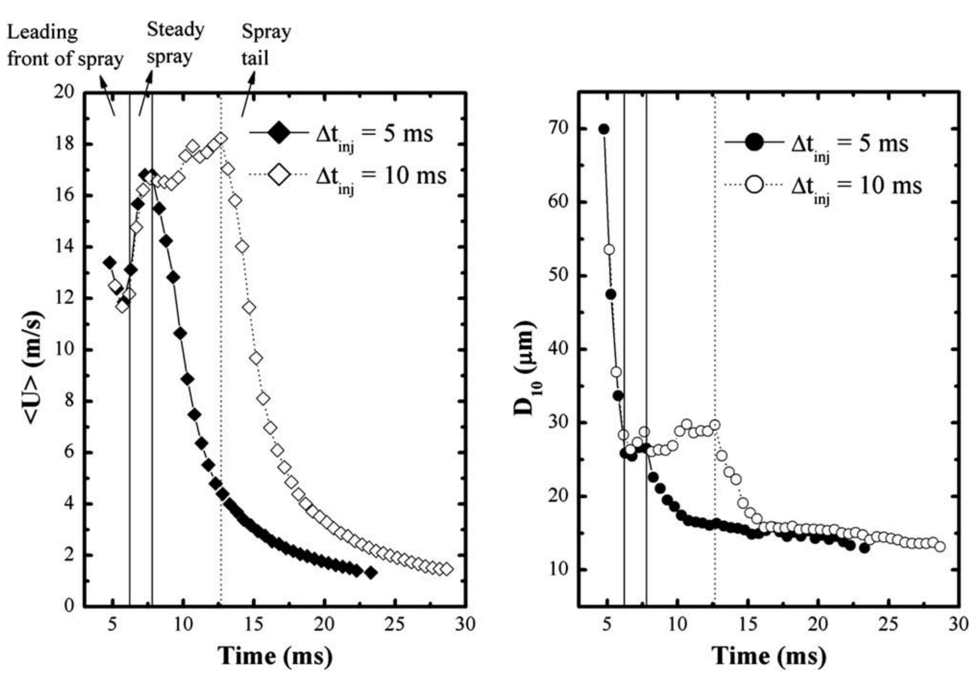

Fig. 2. Identification of characteristic periods in the spray dynamic behavior.

where $t_i = i\delta t$ and δt is the sampling time, or else the inverse of the sampling rate.

4. A dynamic approach to the development of a heat transfer correlation

The first part of the results presented in this section emphasize what is meant by the 'spray dynamic behavior', its dimensionless form and addresses the influence of the degree of interaction between consecutive injection events to the heat transfer process. The second part determines a heat transfer correlation based on a conventional approach before proposing a dynamic approach, and both are physically interpreted and their accuracy in heat transfer predictions is compared.

4.1. *Analysis on the spray dynamic behavior*

An important feature in multiple-intermittent spray impaction is the systematic variations of the axial velocity (U_{d}), size (D_{d}) and number flux ($\dot{q}''_{\mathrm{d},n}$) of droplets within an injection cycle, induced by the opening and closing of the injector pintle, and their influence on heat transfer processes. These variations follow a pattern which is systematically identified by three characteristic periods, as illustrated in Fig. 2 through the mean axial velocity and size of droplets for 10 Hz intermittent sprays with pulse durations of 5 ms and 10 ms:

(i) the first period, $t_{\mathrm{impact}} \leq t \leq t_{\mathrm{impact}} + 1$, or the *leading front of the spray* (LFS), has been observed to systematically last 1 ms in every experiment and is characterized by a sudden expansion of the liquid after pintle opening, resulting in a local minimum of the mean axial velocity at the end of the period and an intense decrease of the mean drop size;
(ii) the second period, $t_{\mathrm{impact}} + 1 \leq t \leq t_{\mathrm{impact}} + \Delta t_{\mathrm{inj}}$, corresponds to the *steady spray* (SS) and starts when the mean drop size attains a plateau value and lasts until the end-of-injection;
(iii) the third and final period, $t_{\mathrm{impact}} + \Delta t_{\mathrm{inj}} > t$, is the *spray tail* (ST) beginning after the end-of-injection, and is characterized by an asymptotic decrease of the mean axial velocity of droplets toward 0 m/s and of the mean size of droplets toward that of those which remain suspended in the air between consecutive injection cycles.

These periods are common to intermittent sprays [27], which is important for developing heat transfer correlations using a dynamic approach. Therefore, after having identified these three periods associated with the dynamics of intermittent sprays, instead of performing a dimensional analysis and establish the dimensionless groups, the following step is to express the characteristic parameters describing this behavior (U_{d}, D_{d}, $\dot{q}''_{\mathrm{d},n}$) in a dimensionless form. From the spray impingement literature, both Laplace ($\mathrm{La} = \rho\sigma D_{\mathrm{d}}/\mu^2$) and Capillary ($\mathrm{Ca} = \rho U_{\mathrm{d}}\nu/\sigma$) dimensionless numbers are used to describe drop impact mechanisms [28,29] and allow the independent investigation of the effects of size and axial velocity of impinging droplets on heat transfer. In fact, it is noteworthy that each of these numbers can be expressed by a relation between 'We' and 'Re' ($\mathrm{La} = \mathrm{Re}^2/\mathrm{We}$ and $\mathrm{Ca} = \mathrm{We}/\mathrm{Re}$). In correlations derived by dimensional analysis, this means that 'La' and 'Ca' could be used as a combination of 'Re' and 'We'. Moreover, if the number flux ($\dot{q}''_{\mathrm{d},n}$) is introduced in the arbitrary function as a parameter important for heat transfer [30], this implies an additional dimensionless number. In the work presented here, instead of deriving this number through a dimensional analysis, we followed the work of Roisman and Tropea [31] which uses the number flux to express the average number of droplets impinging in the vicinity of each other as $\lambda = \pi\delta r^2 \delta t_{\mathrm{bin}} \dot{q}''_{\mathrm{d},n}$, with δr as the interaction radius of multiple drop impacts and δt_{bin} as a time-bin in the phase-average analysis. This has the advantage of considering the effect of multiple drop impacts in the heat transfer correlation. The Nusselt number in the phase-average analysis is given by $\mathrm{Nu} = \overline{h}_{\mathrm{w,bin}} D_{\mathrm{d}}/k$ with $\overline{h}_{\mathrm{w,bin}} = \overline{\dot{q}''_{\mathrm{wbin}}}(\overline{T}_{\mathrm{w}} - T_{\mathrm{f}})$, where T_{f} is the fluid temperature and $\overline{\dot{q}''_{\mathrm{wbin}}}$ corresponds to the time-average wall heat flux calculated by the integration of its instantaneous value obtained in equation (15) from the surface temperature measurements.

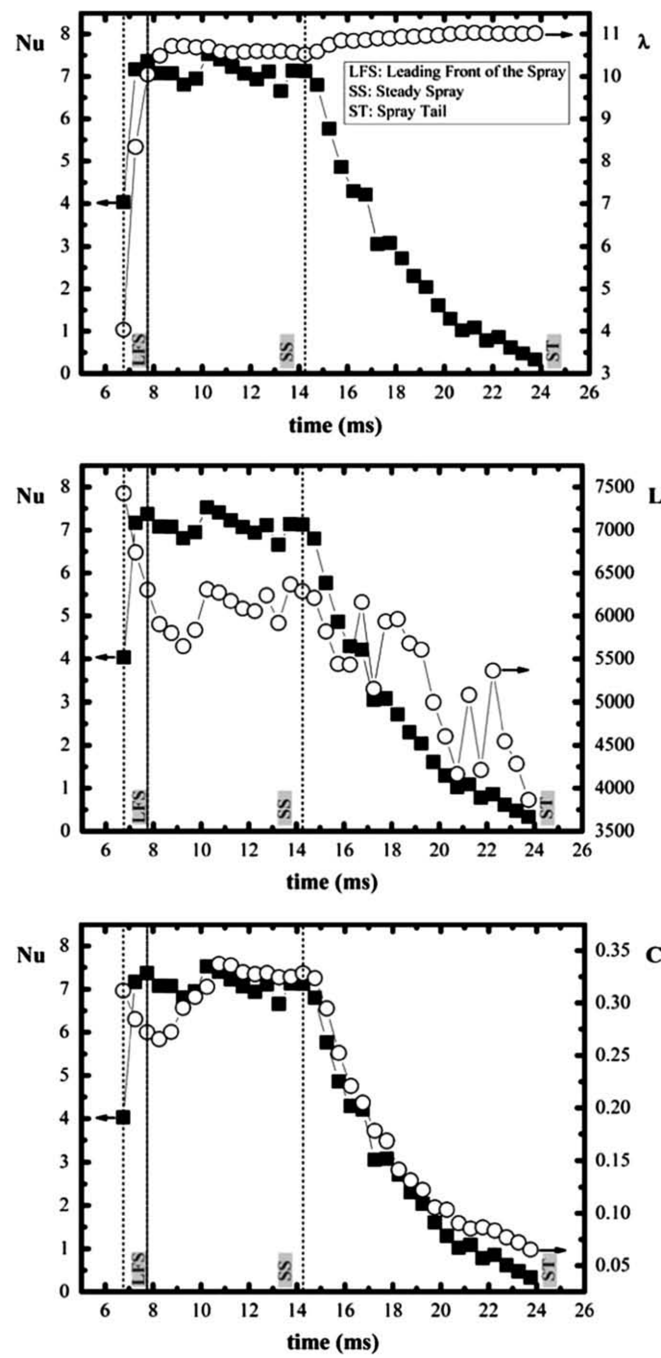

Fig. 3. Correlation between spray dynamic behavior and the heat transfer associated with the cooling process in the three identified periods of the leading front of the spray (LFS), steady spray (SS), and spray tail (ST), for an injection frequency of 30 Hz and a 10 ms pulse.

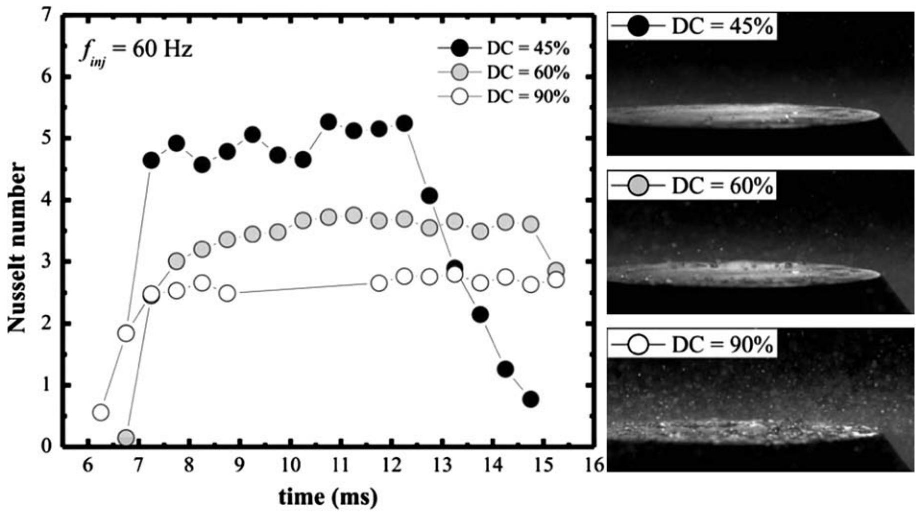

Fig. 4. Effect of duty cycle (DC) on the Nusselt number variation along an injection cycle and, on the right, illustration of the liquid film built-up as a result of an increasingly greater degree of interaction between consecutive injection events.

In Fig. 3 the phase-average variations of ‘Nu’ within an injection cycle are compared with the dimensionless number flux λ, the Laplace ‘La’ and the Capillary ‘Ca’ numbers. The leading front of the spray period (LFS) corresponds to the first contact of cooling liquid with the heated surface and, consequently, is characterized by the rise of ‘Nu’. At this initial stage of the spray cooling event, the behavioral pattern followed by the dimensionless number flux of droplets and that of ‘Nu’ are similar, suggesting a strong correlation between these two parameters in this period. Relatively to the spray droplets size and velocity, in the LFS, the impact momentum of these droplets is relatively high, which means that secondary atomization is promoted instead of liquid deposition for cooling purposes. As we enter into the steady spray (SS) period, the dimensionless number flux of drops stabilizes, the ‘Ca’ has a local minimum and also moves toward stabilization of its value, and it is noteworthy that variations in ‘Nu’, follow the same behavioral pattern as ‘La’, emphasizing the relative importance of the mean drop size in this period. Finally, once the injector closes its pintle, one enters the spray tail (ST) period and here the correlation between ‘Nu’ and ‘Ca’ is clearly evident, therefore, the behavioral pattern appears particularly linked with drop velocity.

Additionally to the influence of the spray dynamic behavior on heat transfer, multiple consecutive injections imply a degree of interaction between them, and this leads to the formation of thin liquid films, which will further influence the phase-change heat transfer mechanism [3]. This interaction degree increases with the duty cycle (DC) and is inversely proportional to the ‘dead-time’ between the end-of-injection and the start of a new one. If the injection frequency is fixed at 60 Hz and the DC increased between 45% and 90%, the effect is shown in the decrease observed for ‘Nu’, as depicted on the left side of Fig. 4. The reason for this is illustrated on the right side with images for each condition, taken 18 ms after the last of a series of 40 injections. The images evidence the presence of a liquid film and an increase of secondary atomization as a result of increasing the interaction degree, explaining why phase-change is mitigated and that less impinging mass is depositing on the wall for cooling purposes.

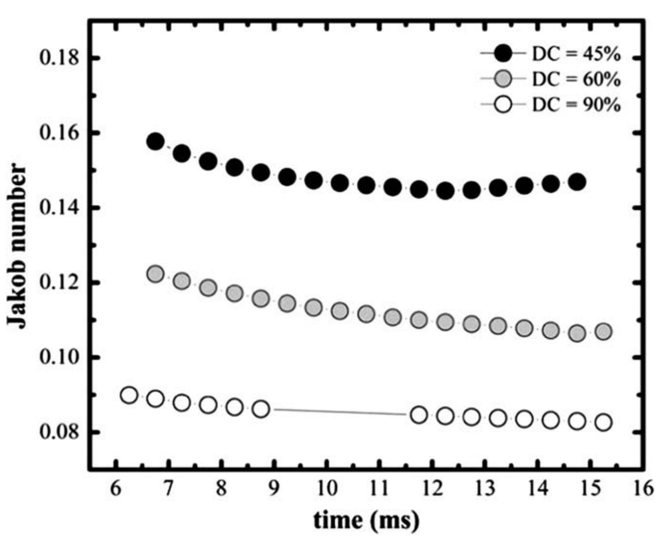

Fig. 5. Effect of duty cycle (DC) on the Jakob number variation along an injection cycle (at 60 Hz) evidencing the increasingly mitigation of phase-change as a result of the formation of a liquid film.

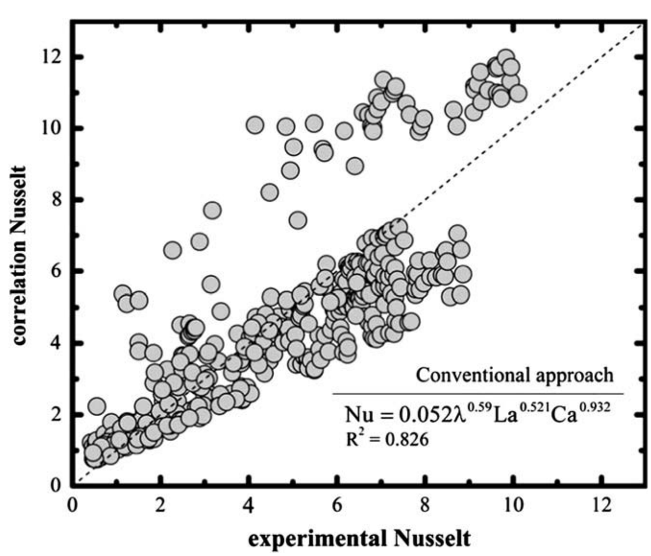

Fig. 6. Overall heat transfer correlation for spray impact using a conventional approach, with all data jointly considered independent of the spray dynamic behavior.

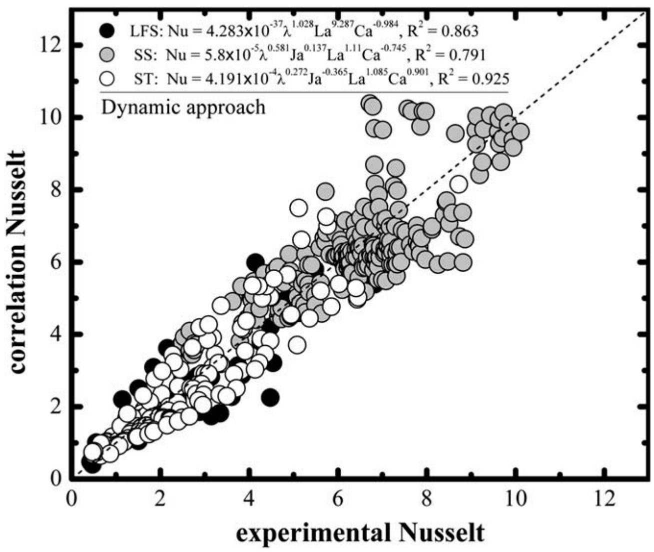

Fig. 7. Dynamic heat transfer correlation for spray impact using the three systematic periods characteristic of intermittent sprays: leading front of the spray (LFS); steady spray (SS) and; spray tail (ST).

As earlier stated, the Jakob number expresses the relative importance between single-phase and two-phase heat transfer by mitigating or promoting, respectively, the phase-change mechanism, which removes high heat fluxes from the heated surface through the latent heat of evaporation. For the same conditions of DC = 45%, 60% and 90%, Fig. 5 shows a general decrease of the 'Ja' number, similarly to the results obtained for 'Nu', and since h_{fg} remains constant, because the experiments are made at atmospheric pressure, this outcome can only be attributed to a decrease of the time-average superheating degree $(\overline{T}_w - T_b)$, which would justify the relatively greater effect of single-phase heat transfer over two-phase, consonant with the fact that phase-change has been mitigated. Based on the considerations above, the 'Ja' number should also be taken into account, with λ, 'La' and 'Ca', in the development of the spray/wall heat transfer correlation.

4.2. Development of a dynamic heat transfer correlation

The general form of an empirical correlation sought for the Nusselt number is basically that of equation (6), however, with the addition of λ, the 'La' and 'Ca' instead of 'Re' and 'We', and 'Ja'. Relatively to the 'Pr' number: its value is negligible compared with 'Ja'; it fails to capture the spray dynamic nature since the thermophysical properties involved do not change significantly within the timescale of each injection cycle, and; ultimately, experimental results showed that considering it produced worse correlations. Therefore, the final form of the correlation used, henceforth, in our analysis, is

$$Nu = a\lambda^{b_1} Ja^{b_2} La^{b_3} Ca^{b_4} \tag{16}$$

It is noteworthy that each exponent, in terms of its sign, may be used to physically interpret what kind of influence is exerted by the associated dimensionless number relatively to heat transfer as expressed by the Nusselt. For example, if $b_2 > 0$, which is associated with 'Ja', it means that single-phase exerts the main influence on heat transfer and eventually mitigates phase-change.

In a conventional approach to the development of a heat transfer correlation, all data points are used regardless the fact of having identified distinct periods in the spray dynamic behavior. This has been the approach followed in the works reviewed in Section 2. Therefore, if this approach is followed, from the simultaneous data acquired for the working conditions summarized in Table 2, a single correlation is determined by linearizing equation (18) and solving the matricial system by the direct method. The correlation obtained is

$$Nu = 0.052\lambda^{0.59} La^{0.521} Ca^{0.932} \tag{17}$$

with $R^2 = 0.826$ and a mean absolute error of 30% (see Fig. 6). The exponent values emphasize the influence of the number flux of droplets and their impact momentum for the enhancement of heat transfer. The exponent associated with 'Ja' was small compared with the remaining and including it produced a worse correlation, which means that convective heat transfer and phase-change have negligible influence in the result provided by the conventional approach. This is quite counterintuitive since convective heat transfer and phase-change are crucial in the spray cooling event, thus supporting the need to develop a new approach.

In fact, the results presented in Section 4.1 suggest that finding a more accurate heat transfer correlation for intermittent sprays may imply the development of a dynamic approach instead of the conventional one. And by a dynamic approach, one simply means a different correlation for each period of injection as

$$Nu = \begin{cases} a_{LFS}\lambda^{b_1^{LFS}} Ja^{b_2^{LFS}} La^{b_3^{LFS}} Ca^{b_4^{LFS}}, & t_{impact} < t \le t_{impact} + 1 \\ a_{SS}\lambda^{b_1^{SS}} Ja^{b_2^{SS}} La^{b_3^{SS}} Ca^{b_4^{SS}}, & t_{impact} + 1 < t \le t_{end\text{-}of\text{-}injection} \\ a_{ST}\lambda^{b_1^{ST}} Ja^{b_2^{ST}} La^{b_3^{ST}} Ca^{b_4^{ST}}, & t > t_{end\text{-}of\text{-}injection} \end{cases} \tag{18}$$

After sorting the data according to their corresponding period of injection, the constants in the empirical correlations were determined. The results are depicted in Fig. 7, the validation domain for each variable, in each period, is listed in Table 3 and the expression derived is

$$Nu = \begin{cases} 4.283 \times 10^{-37}\lambda^{1.028} La^{9.287} Ca^{-0.984}, & t_{impact} < t \le t_{impact} + 1 \\ 5.8 \times 10^{-5}\lambda^{0.581} Ja^{0.137} La^{1.11} Ca^{-0.745}, & t_{impact} + 1 < t \le t_{end\text{-}of\text{-}injection} \\ 4.191 \times 10^{-4}\lambda^{0.272} Ja^{-0.365} La^{1.085} Ca^{0.901}, & t > t_{end\text{-}of\text{-}injection} \end{cases} \tag{19}$$

The correlation obtained for the LFS is linearly correlated with the number flux of impinging droplets ($\lambda^{b_1}, b_1 \approx 1$) and has a very small constant a because of the significantly high value of the exponent associated with 'La', which by itself has a magnitude 10^4 higher than the remaining dimensionless values. This denotes that heat transfer in this period is largely influenced in magnitude by the size of impinging droplets (La^{b_3}, $b_3 > 0$), even though this is balanced by the negative impact of the axial velocity (Ca^{b_4}, $b_4 < 0$). While bigger droplets contain more mass to cool the heated surface, if these have a larger velocity, it eventually triggers secondary atomization mechanisms, and the impinging liquid,

Table 3
Validation domain for each variable in the dynamic empirical heat transfer correlation for spray impingement.

	LFS	SS	ST
λ	$\in$ [0.341, 20.91]	$\in$ [2.743, 22.1]	$\in$ [2.933, 21.82]
Ja	–	$\in$ [0.017, 0.627]	$\in$ [0.097, 0.599]
La $\times 10^3$	$\in$ [6.25, 7.906]	$\in$ [5.027, 7.212]	$\in$ [3.549, 6.67]
Ca	$\in$ [0.272, 0.363]	$\in$ [0.224, 0.376]	$\in$ [0.053, 0.311]

instead of depositing on the surface, is being removed from the surface in the form of secondary droplets.

In the SS period, the Jakob number begins exerting its influence, although mild compared to the remaining variables, however, the fact that its exponent is positive gives the predominant role to single-phase heat transfer, rather than two-phase through phase-change. This could be explained by the contribution of impinging droplets, during this period, to build-up a thin liquid film (La^{b_3}, $b_3 > 0$), even if, again, this is balanced by larger velocities which imply secondary atomization (Ca^{b_4}, $b_4 < 0$) and, for the reasons already mentioned, this has a negative impact on heat transfer. Finally, the influence attributed to the dimensionless number flux is the same as in the LFS, although less expressive since the magnitude of its validation domain remains (λ^{b_1}, $b_1^{\mathrm{LFS}} > b_1^{\mathrm{SS}}$).

Finally, in the ST period after the end-of-injection, the impact momentum of impinging drops, driven by pressure forces at the injector nozzle exit, loses its source and the outcome is the asymptotic decrease of the impact momentum of droplets governing heat transfer during this period (Ca^{b_4}, $b_4 > 0$). Relatively to the 'Ja', without multiple drop impacts and the supply of cooling liquid to the film, phase-change begins to dominate heat transfer over single-phase (Ja^{b_2}, $b_2 < 0$). The size of droplets impinging during ST also decreases toward the smaller size of those droplets which fall by gravity between consecutive injection cycles (see Fig. 2), and given their lower impact energy, these probably deposit on the surface and positively contribute to the heat removal (La^{b_3}, $b_3 > 0$). The continuing decrease of the importance of the number flux to heat transfer is consistent with the decrease observed between LFS and SS.

When the results between Figs. 6 and 7 are compared, some data points using the conventional approach would be discarded, but this is not the case when a dynamic approach is used instead.

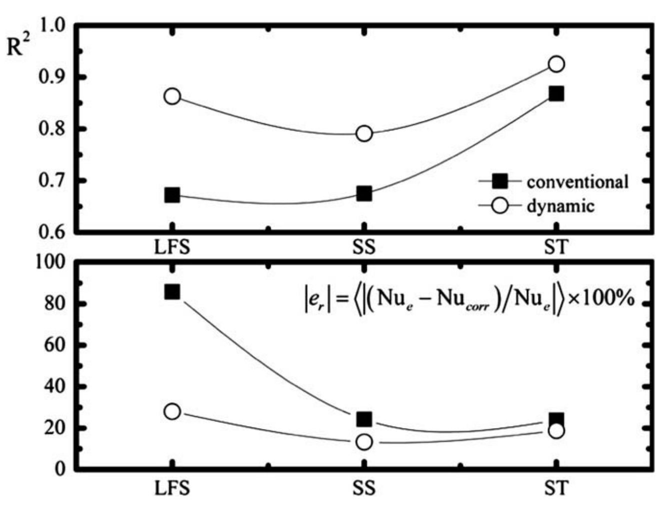

Fig. 8. Comparison between the conventional and dynamic approaches to the development of a heat transfer correlation through the correlation coefficient R^2 and the mean absolute error.

This could be explained by the underlying idea behind the dynamic approach, which emphasizes that heat transfer mechanisms depend of the spray dynamic behavior, *i.e.* the mechanisms vary according to the dynamic characteristics of the impinging spray. If we also compare the correlation coefficient (R^2) and mean absolute error given by the correlation derived in each period, with those obtained if the conventional approach were used instead, it becomes clearer the greater accuracy obtained in heat transfer predictions, as depicted in Fig. 8.

5. Concluding remarks

Intermittent spray cooling is an example of an emergent trend in thermal management techniques seeking the active control of the heat transfer process. The work presented here investigates a new approach in the development of heat transfer correlations for intermittent spray impingement, based on simultaneous measurements of the spray droplets characteristics and the surface thermal behavior. Conventionally, these empirical correlations are derived without considering the effect of time variations of droplets characteristics on heat transfer. However, this work shows that temporal variations of the axial velocity, size and number flux of impinging droplets set the heat transfer behavioral pattern and suggest that a more dynamic approach in developing heat transfer correlation improves their accuracy. On the one hand, this presupposes the identification of systematic periods characterizing the spray dynamic behavior, namely, the leading front of the spray, the steady spray and the spray tail. On the other hand, the degree of interaction on multiple consecutive injections should also be considered. In fact, if the time between consecutive injections is shorten, consequently increasing their degree of interaction, the result is the formation of a liquid film which mitigates phase-change and privileges a single-phase heat transfer over a two-phase.

Considering the spray dynamics and the interaction between multiple consecutive injections, dimensionless parameters have been devised, including the dimensionless number flux (λ), the Laplace (La), the Capillary (Ca) and the Jakob (Ja), which correlate the heat transfer with the spray characteristics. Contrary to the conventional approach for developing a heat transfer correlation, the dynamic approach here proposed consists in deriving a correlation for each period. Although this might seem too simple, the underlying principle of this proposal is that different characteristics of the spray in each period, in fact, imply different heat transfer mechanisms. Therefore, a more accurate correlation *unites* the expressions obtained for each period, and not *uniformizes* all periods into one correlation. The comparison between the dynamic and conventional approaches shows that heat transfer predictions improve if the spray dynamic nature is considered.

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