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INTERMITTENT SPRAY COOLING OF METALS AT SURFACE TEMPERATURES ABOVE THE LEIDENFROST POINT

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ABSTRACT
An experimental study is reported on the spray cooling of high temperature metal surfaces using intermittent sprays. Sprays with low mass fluxes (dilute sprays) are produced by varying injection conditions (pulse duration and injection frequency). Using the variable "duty cycle", defined as percentage of cycle time during which the liquid is injected, the injection conditions are combined. The analysis addresses the effects of injection conditions on local decreases in surface temperature and local heat transfer coefficients in the film boiling range. It is observed that at low duty cycles, smaller than 10 %, the heat transfer coefficient depends on pulse duration and injection frequency. For higher duty cycles, the heat transfer coefficient is mainly influenced by injection frequency.

1. INTRODUCTION

Intermittent sprays are used in fuel injection systems of spark-ignition and diesel engines and can also be applied in cooling processes, such as dermatologic surgery application. These cooling processes are characterised by injection frequency f_{inj} and pulse duration Δt_{inj}. Both parameters are linked with the variable duty cycle DC, which is defined as a percentage of the cycle time ($1/f_{inj}$) during which the cooling liquid is injected: $DC = (\Delta t_{inj}\, f_{inj}) \times 100\ \%$ [1]. A duty cycle of 100 % corresponds to a continuous spray.

Generally, transient surface temperature distributions during spray cooling processes above the Leidenfrost point using dilute sprays feature two ranges: a period of direct liquid-solid contact during the initial phase of spray impact and a period, where a vapour layer is formed. The heat transferred from the wall in this first phase is in order of magnitude higher than the amount of heat through the subsequent formed vapour layer [2]. During intermittent spray cooling each injection cycle offers these two ranges, whereby the length of the second period depends on the injections frequency (Figure 1). The surface temperature drop depends on the surface temperature and on the corresponding boiling regime as well as the amount of injected cooling liquid. As shown in Figure 1, pulse duration of 15 ms leads to about two times higher decrease in surface temperature than pulse duration of 5 ms. Furthermore, a suitable choose of injection frequency allows also fast cooling. Thereby, intermittent sprays offer precision control of the cooling process by proper matching of injection frequency and pulse duration, whereby, a definite chilling can be obtained.

The time-dependence of the heat transfer coefficient is experimentally analysed [3][4][5] and is not object of this study. Dividing the heat transfer during intermittent spray cooling in accordance with the spray dynamic behaviour, three time-dependent ranges are obtained [3]. The first period occurs during the leading front of the spray when the initial droplets impact on the surface and the number flux of droplets is the mainly influence parameter for the heat transfer [4]. During the second regime, called steady spray period, the heat transfer is governed by the variation of mean droplet size [3]. The final period is known as spray tail period when heat flux and droplet velocity decreases simultaneously [5].

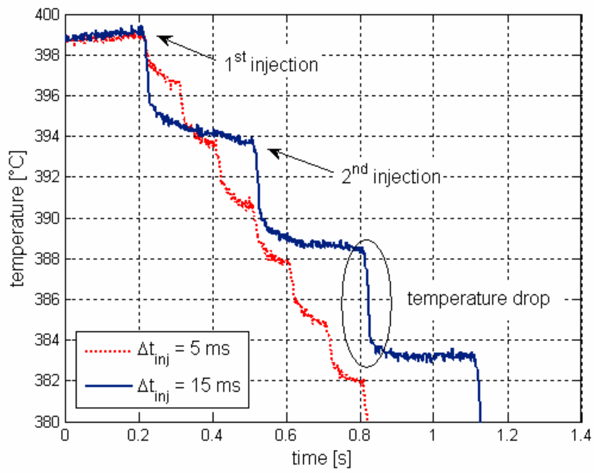

Figure 1: Surface temperature envelopment during an injection with a pulse duration of 5 ms and 15 ms, respectively

Defined cooling conditions require knowledge of the heat transfer coefficient, which depends on mass flux, spray characteristics (droplet diameter and velocity) as well as wall and liquid properties. Panão and Moreira [4] published that neither the injection frequency nor pulse duration produces significant

changes in droplet size and axial velocity distribution of impinging sprays in wide range of duty cycles. This prediction is confirmed by own measurements. Consequently, intermittent spray cooling experiments allow a study of influence on mass flux on heat transfer mechanism without essential changing of spray characteristics.
The objective of this research is quantifying the effects of intermittent sprays on heat transfer in the range of film boiling, whereby the mass flux is varied changing injection frequency and pulse duration.

2. EXPERIMENTAL SETUP

The experimental configuration consists of a BOSCH port fuel injector directed perpendicular to a sheet made of Inconel 600 (100 x 60 x 0.3 mm), which is directly electrical heated. A schematic of the experimental setup is shown in Figure 2. Using a self-build triggering system, the injection frequency, the pulse duration and the number of injections are controlled. The cooling fluid is distilled water.

An infrared camera (Flir ThermaCam Sc 3000) records the surface temperature with a sampling rate up to 750 Hz. As infrared measurements depend strongly on the emissivity of the surface, the camera facing sheet side is coated with a black painting called Senotherm UHT 600 with a thickness of about 20 µm.

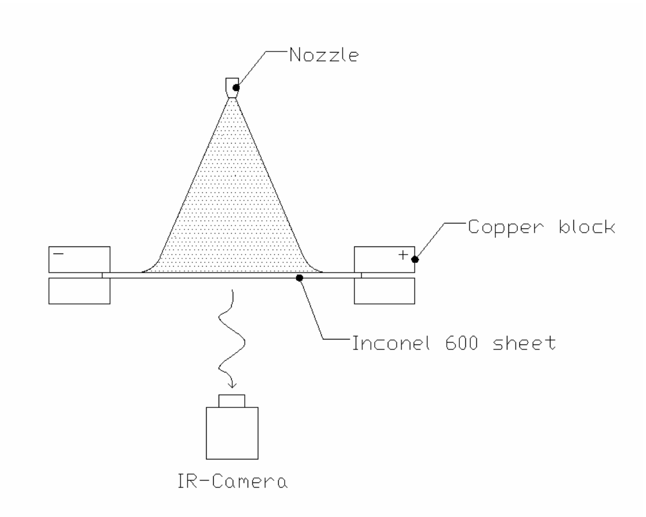

Figure 2: Schematic of the experimental apparatus

The pulse duration varies from 5 ms to 20 ms and the injection frequency is adapted to realise duty cycles from 1 % to 60 %. During these experiments, the impinging distance is kept constant and equal to 150 mm and the injection pressure is set to 7 bar. The produced spray features under these working condition a mean diameter of about 50 µm as well as a mean velocity of about 5 m/s and the total spray angle is about 25 deg.

Mass flux measurements are done with a Patternator, which consists of several small tubes, arranged in one line. The water quantity is collected by all tubes over a certain period of time.

3. CALIBRATION MEASUREMENTS

The cooling process is recorded in a sequence of single images using an infrared camera, whereas the acquisition is restricted to a constant emissivity, which is in fact a function of temperature and needs to be determined in dependence on the surface temperature. Therefore, two coated Inconel 600 sheets are spot welded together and both of them feature a recess for mounting the thermocouple (Figure 3). The test section is directly and uniformly heated. By varying the amperage, different surface temperatures are realised and simultaneously recorded via thermocouple and infrared camera. The difference between outer surface temperature and measured temperature of the thermocouple is calculated using the FOURIER differential equation and is found to be negligible due to the minor thickness of the sheet. Using the measured temperature information, the emissivity can be expressed as a function of surface temperature

$$\varepsilon = 2.273 \cdot 10^{-2} \ln \vartheta_S + 4.412 \cdot 10^{-5} \vartheta_S + 7.406 \cdot 10^{-1}. \quad (1)$$

with $\varepsilon [-]$ and ϑ_S [°C]

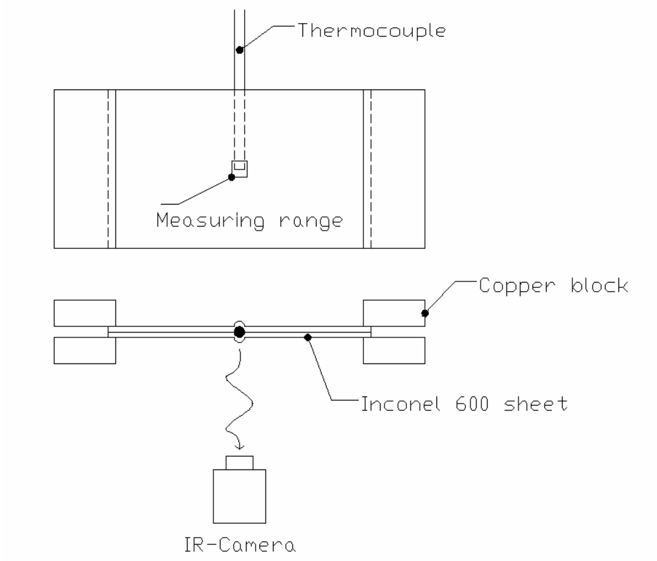

Figure 3: Schematic measurement principle for determining the emissivity

The recorded temperatures via infrared camera are subsequent corrected using the temperature-dependent emissivity expressed in equation (1). An exemplarily temperature profile is shown in Figure 4.

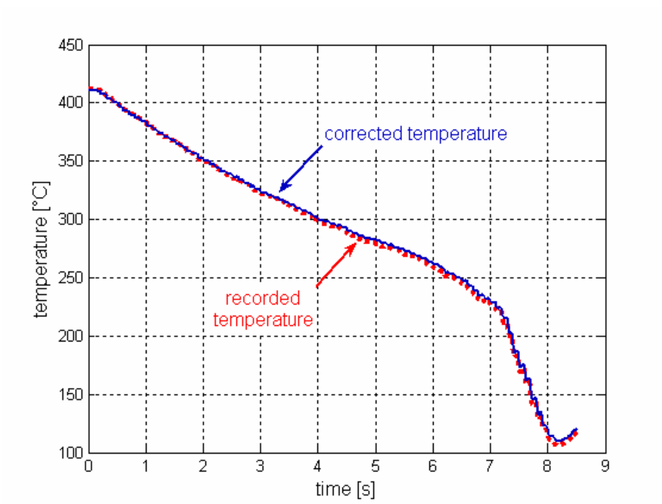

Figure 4: Temperature correction

The heat losses at the plate backside consist of convection and radiation and are expressed as semi-empirical correlations in dependence on the surface temperature. Radiation losses are calculated using equation (1), whereas, the condition small object in a large environmental area is assumed.

$$\alpha_R = 1.0273 \cdot 10^{-4} \vartheta_S^2 + 1.4987 \cdot 10^{-2} \vartheta_S + 4.7435 \qquad (2)$$

with $\alpha_R \; [W/(m^2K)]$ and $\vartheta_S \; [°C]$.

Using an empirical Nusselt-equation according Michejew [6], the convection losses (due to natural convection) at a horizontal heated sheet are determined. The heat transfer coefficient, which describes the convection losses, can be written as

$$\alpha_{Con} = 1.2983 \cdot \ln \vartheta_S - 7.9106 \cdot 10^{-1} \qquad (3)$$

with $\alpha_{Con} \; [W/(m^2K)]$ and $\vartheta_S \; [°C]$.

Finally, the overall heat losses are well described as polynomial function of third order

$$\begin{aligned}\alpha_{Diss} = {} & 9.88347 \cdot 10^{-3} \vartheta_S^3 + 6.1443 \cdot 10^{-6} \vartheta_S^2 \\ & + 4.7587 \cdot 10^{-2} \vartheta_S + 7.5658\end{aligned} \qquad (4)$$

with $\alpha_{Diss} \; [W/(m^2K)]$ and $\vartheta_S \; [°C]$.

The experimentally determined heat transfer coefficient of dissipated heat agrees well with equation (4).

4. DATA PROCESSING

The cooling process is recorded using an infrared camera with a frame rate up to 750 Hz. Hereby, one camera image correspond to a certain time and surface temperatures are obtained for the whole impact area of the spray. Figure 5 shows exemplarily temperature information for a certain time during the cooling process.

For each time, a 2-dimensional matrix is obtained which includes temperature information of the total sheet. The matrix size corresponds to the number pixel for the length and the width of the Inconel 600 sheet The data are further processed in Matlab to a three-dimensional matrix, whereas the third component is the time. This data processing reduces calculation time, since only one file needs be loaded instead of several thousands.

All temperature data of the 3-dimensioanl matrix needs to be subsequently corrected according equation (1) considering the temperature-dependent emissivity. The corrected data are finally used for heat transfer coefficient calculations.

The heat transfer coefficient depends on surface temperature, which is also a function of time, as well as spatial position and is determined in the range of stable film boiling at surface temperatures between 420 °C and 300 °C. For an Inconel 600 plate with thickness of 0.3 mm, the calculated Biot numbers are smaller than 0.1. Therefore, it is reasonable to assume a lumped condition to evaluate the surface heat flux, and the solution of the inverse heat conduction problem is not necessary.

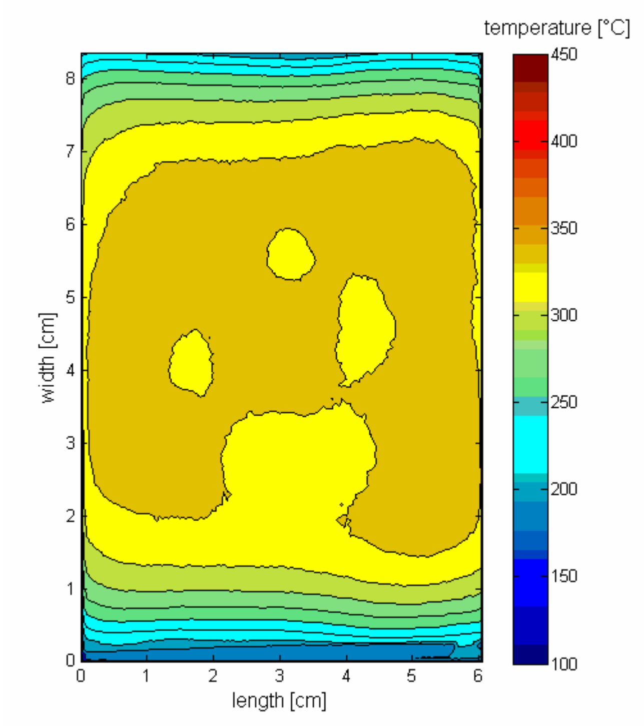

Figure 5: Temperature profile for a certain time

Thus, the temperature measured at the backside of the plate is used to represent the temperature at the front side. Furthermore, it is assumed that the surrounding temperature has the same value than the fluid temperature. Considering the negligible radial conduction over the thin plate a lumped system can be applied to any specific region during cooling. The heat transfer coefficient is calculated by

$$\frac{dT}{dt} = \underbrace{\frac{\dot{q}_V}{\rho c_p} + \frac{\alpha_{Spray} + \alpha_{Diss}}{\rho c_p s} T_{fl}}_{M} - \underbrace{\frac{\alpha_{Spray} + \alpha_{Diss}}{\rho c_p s}}_{N} T \,. \qquad (5)$$

Specific heat capacity as well as the density of Inconel 600 feature only weak temperature sensitivity and are assumed to be constant. The electrical heat source can be written using the specific resistance, which is a function of temperature and is experimentally obtained

$$\dot{q}_V = \frac{I^2 \rho_{elec}}{b^2 s^2} \,. \qquad (6)$$

The analytical solution for discrete time steps of equation (5) reads as

$$T_i = \frac{M}{N} + \left(T_{i-1} - \frac{M}{N}\right) e^{-M(t_i - t_{i-1})} \,. \qquad (7)$$

For estimation the heat transfer coefficient, a numerical solver is used which solves nonlinear data-fitting problems in the least squares sense.

The analytical solution implies that the electrical resistance ρ_{elk} as well as the overall heat transfer coefficient of dissipated heat flux α_{Diss} are constant during a time step. Thereby, mean temperatures of each time step are used calculating both.

5. RESULTS AND DISCUSSION

Intermittent sprays are characterised by injection frequency and pulse duration. Both parameters are related with the variable duty cycle which is often used as influence parameter on heat transfer. There are two possibilities changing DC, modifying the frequency and keeping the pulse duration constant, or vice-versa. It is known that an increase of DC (if the pulse duration is fixed) leads to faster cooling. In generally, the number flux or mass flux of an impinging spray features a distribution over the impact surface. Hence, the variable duty cycle is not sufficient to describe all effects during a cooling process.

Figure 6 represents a temperature distribution and the corresponding mass flux distribution versus the width of the Inconel 600 plate. The data indicate that for the same parameter of pulse duration and DC different decrease of surface temperatures are obtained in dependence on the local mass flux.

A further important parameter for evaluate the intermittent cooling process is the amount of mass M which is total atomised. This parameter is in general described by the number of injections N.

$$M \sim N\,\Delta t_{inj} \qquad (8)$$

Studying the effect of pulse duration on heat transfer is based on a constant mass. Doubling the pulse duration, approximately, the double amount of liquid is atomised. Hence, changing of pulse duration requires an altering of injection numbers according equation (8). The table below summarised all studied cases and their conditions.

Table 1: Working Conditions

Cases	Δt_{inj} [ms]	DC [%]	N [-]
1-11	5	1, 2.5, 5, 10, 12.5, 15, 20, 25, 30, 40, 60	40
12-22	10		20
23-33	15		13
34-44	20		10

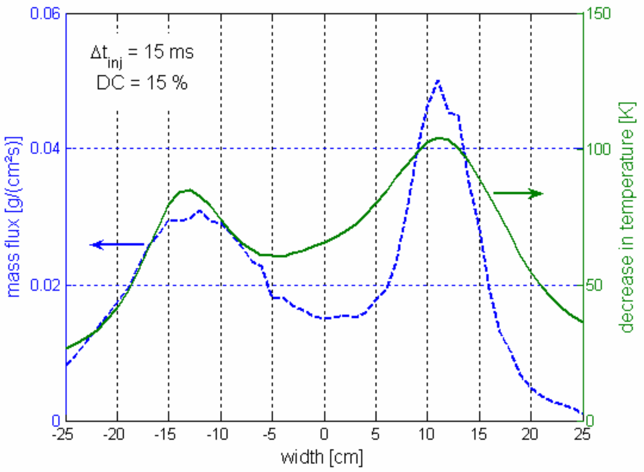

Figure 6: Comparison of decrease in surface temperature and mass flux for DC of 15 % and pulse duration of 15 ms

Since the mass flux seems to be the primary factor affecting the decrease in surface temperature, the temperature drop is replotted versus the liquid mass flux in Figure 7 for different duty cycles and constant pulse duration of 15 ms. The lines are curve fits to the experimental data.

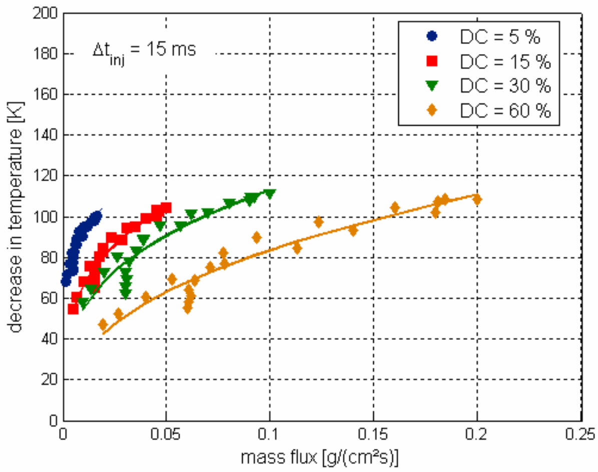

Figure 7: Temperature drop versus mass flux for a pulse duration of 15 ms

The same mass flux can be realised with different duty cycles. Whereby, it is interesting to identify that with the same mass flux at lower DC a higher cooling is obtained. Considering the shape of the curve fit, it can be described by an exponential fit

$$\Delta T = B\,\dot{m}^{C}, \qquad (9)$$

where ΔT is the decrease in surface temperature and B as well C are the parameters for each fit and their values are functions of pulse duration and duty cycle. The units of the coefficient A and B depend on the mass flux as well as the decrease in surface temperature. Therefore, a normalised mass flux should be used for comparative studying of injection condition effects on heat transfer. The mass flux is normalised by a homogenised mass flux which can be understood as uniform mass flux over the total impact area

$$\dot{m}_{uni} = \frac{\dot{M}}{A_{Imp}} \qquad (10)$$

whereby, the mass flow can be written as followed

$$\dot{M} = \int_{A_{Imp}} \dot{m}(A)\,dA\,. \qquad (11)$$

The heat transfer coefficient is the common parameter for evaluating cooling processes. The overall heat transfer coefficient in the range of stable film boiling is calculated according equation (7). Figure 8 shows exemplarily obtained heat transfer coefficients, whereas the film boiling regime is divided in one, two and four ranges of constant heat transfer coefficients, respectively. With diminishing surface temperature, the heat transfer coefficient decreases lightly. The overall constant heat transfer coefficient in the film boiling regime can be considered as a mean value and is used for comparing and evaluating several combinations of pulse duration and duty cycles.

Heat transfer coefficients evaluate the amount of transferred heat during a certain time step and at a certain impact area. Consequently, cooling processes with higher duty cycles, which correspond to a higher injection frequency and a fast cooling, offer higher overall heat transfer coefficients. This is illustrated in Figure 9 and Figure 10 , where heat transfer coefficient is plotted versus normalised mass flux for a duty cycle of 5 % and 20 %, respectively. Concerning the influence of the heat losses of the dry side of the sheet, it should be mentioned that the duration of a cooling process is equally for experiments with the same duty cycle. The figures show that the heat transfer coefficient increases monotonically with the mass flux.

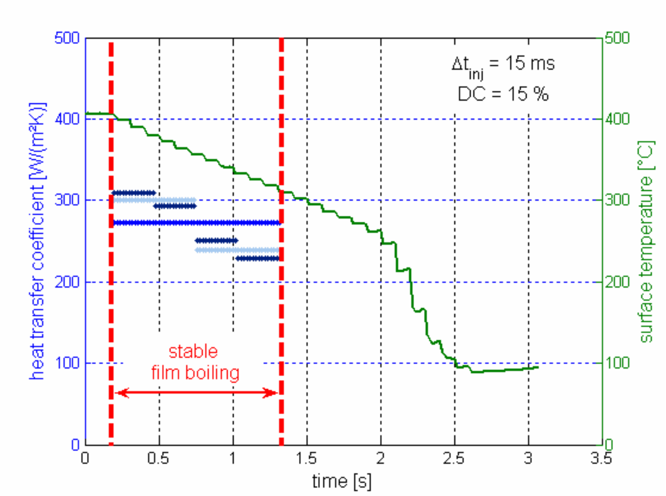

Figure 8: Heat transfer coefficient for a pulse duration of 15 ms and DC of 15 %

Figure 9 identifies that at low duty cycles a pulse duration of 5 ms leads to intensified heat transfer coefficients. Higher pulse durations offer no essential differences in values of heat transfer coefficient, which could likely due to the interference effect of drops on heat transfer. This occurs when the spray density increases and drops do not have independent heat transfer because frequently the local surface temperature does not fully recover before another drop impact at the same point.

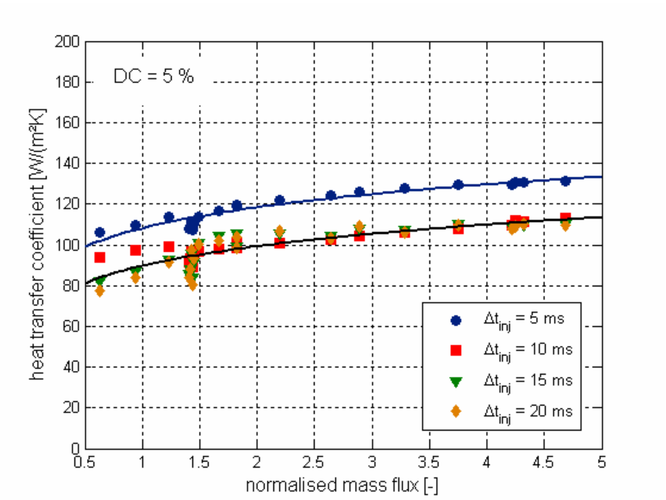

Figure 9: Heat transfer coefficient versus normalised mass flux for a duty cycle of 5 %

Figure 10 shows no essential effect of pulse duration on heat transfer coefficient for a duty cycle of 20 %. The mass flux is the govern parameter for describing the heat transfer coefficient in the film boiling regime.

Summarised, the parameters which influence the heat transfer are pulse duration and injection frequency at low duty cycles. The heat transfer at higher duty cycles is mainly influenced by the mass flux which is proportional to DC.

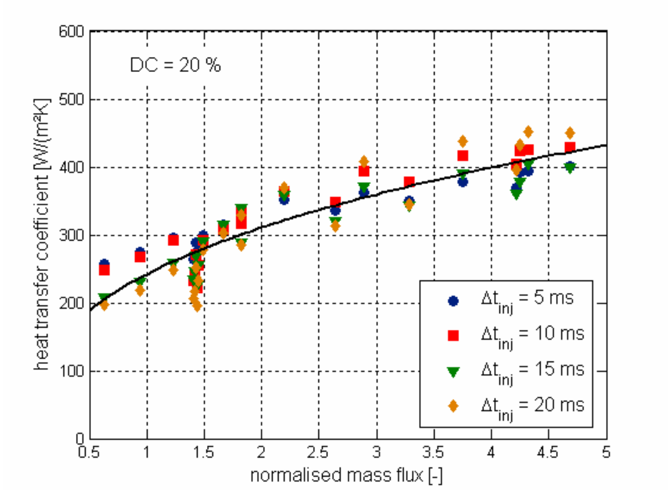

Figure 10: Heat transfer coefficient versus normalised mass flux for a duty cycle of 20 %

Spray cooling process using dilute sprays is described by many researchers using empirical correlations. The approach to formulate the heat transfer coefficient often reads as

$$\alpha_{Sspray} = D\,\dot{m}^{E} \tag{12}$$

with $\alpha_{Spray}\left[W/\left(m^2K\right)\right]$ and $\dot{m}\left[g/\left(cm^2s\right)\right]$.

This approach can be applied to intermittent sprays at high duty cycles where the mass flux is the govern influence parameter. Figure 11 shows several empirical correlations in comparison to own results obtained at duty cycles above 10 %. The own empirical correlation agree well with the values of Yao and Choi [7].

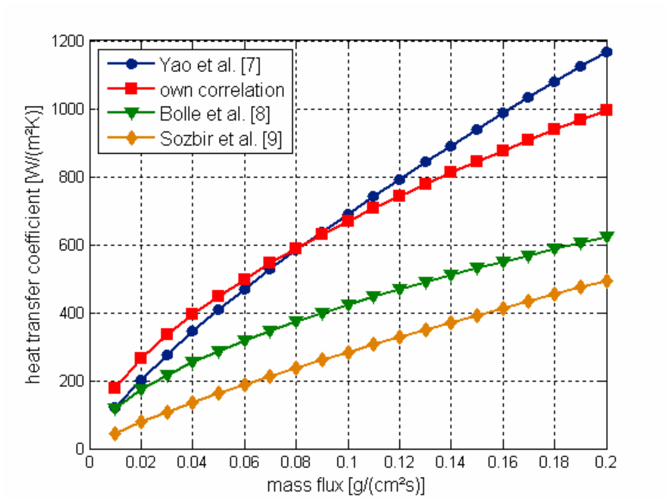

Figure 11: Comparison of own results with empirical correlations for the heat transfer coefficient

6. CONCLUSION

Experimental studies are conducted to obtain the heat transfer of intermittent water sprays on high temperature metal surfaces. Local heat transfer coefficients were determined in film boiling regime at various pulse duration and injection frequencies. The major conclusions from the study are as follows:

1. The heat transfer coefficient depends strongly on the mass flux.
2. Using the same mass flux, a higher decrease in surface temperature is obtained at lower duty cycle.
3. At low duty cycles, the heat transfer coefficient depends on pulse duration and injection frequency. The lowest pulse duration yields highest values of the heat transfer coefficient.
4. At high duty cycles (DC ≥ 10 %), the heat transfer is only influenced by DC. A changing of pulse duration shows no significant effect on heat transfer coefficient.

7. ACKNOWLEDGMENT

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8. NOMENCLATURE

A_{Imp}	Impact area	m²
B	fitting coefficient	$cm^{2C}s^{C}K/g^{C}$
b	width	m
c_p	specific heat capacity	J/(kg K)
C	fitting coefficient	-
D	fitting coefficient	$cm^{2E}s^{E}K/g^{E}$
DC	duty cycle	%
E	fitting coefficient	-
f_{inj}	injection frequency	Hz
I	amperage	A
M	mass	kg
$\dot{M}$	mass flow	kg/s
$\dot{m}$	mass flux	kg/(m²s)
p	pressure	Pa
T	temperature	K
Δt_{inj}	pulse duration	s
ΔT	decrease in surface temperature	K
s	thickness	m

Greek symbols

α_{Con}	convective heat transfer coefficient	W/(m²K)
α_{Diss}	heat transfer coefficient of dissipated heat flux	W/(m² K)
α_R	radiation heat transfer coefficient	W/(m² K)
α_{Spray}	spray heat transfer coefficient	W/(m²K)
ε	emissivity	-
ϑ_S	surface temperature	[°C]
ρ_{elec}	specific resistance	Ωm

Subscripts

Con	convection
Diss	dissipated
e	evaporation
elec	electrical
i	time step i
Imp	impact
inj	injection
p	pressure
R	radiation
S	surface
spray	spray

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