Two phase spray cooling for specific local quenching of workpieces in flexible flow fields

S. Schuettenberg* and U. Fritsching
Foundation Institute for Materials Science (IWT)
Collaborative Research Centre (SFB 570) "Distortion Engineering"
Badgasteiner Str.3, 28359 Bremen, Germany

Abstract

By quenching for hardening in association with heat treatment processes of workpieces, a distortion compensation can be realized by impressing controlled asymmetric heat transfer conditions on workpiece surfaces by the use of liquid jet or spray arrangements. The controlled quenching in and with liquid media, especially the use of multiphase atomizers for spray cooling processes, increases the possibility for generating specific local heat transfer conditions and therefore achieving results of asymmetric quenching on workpieces.

Introduction

The quenching for hardening of workpieces is often associated with a workpiece distortion after the final heat treatment process. This distortion is practically compensated by material allowance in the manufacturing and finishing rework after the final heat treatment or hardening process, respectively. The avoidance of workpiece distortion in heat treatment processes can also be realized by impressing asymmetric cooling conditions by the use of flexible flow fields based on fluid jets or sprays. The controlled liquid quenching (jet cooling), especially spray cooling, enables the possibility to generate specific local heat transfer conditions on workpiece surfaces, which builds the basis of asymmetric quenching for reducing distortion.

For the analysis of workpiece distortion activated by heat treatment, the hardening process by quenching in adapted flexible flow fields is modelled in the framework of the Collaborative Research Centre (SFB570) "Distortion Engineering" at the University of Bremen [1]. Here, process conditions and the resulting heat transfer from workpiece surfaces to ambient and quenching medium are crucial conditions for the quenching process and possible appearance of distortion. As part of the project work, a special quenching process in gaseous and liquid media is applied for analysis and modelwise description of representative, simple shaped workpieces as rings and shafts. Here, a locally asymmetric quenching process is enabled by the use of special equipment based on flexible jet arrays. By these analyses, the potential for avoidance or reduction of distortion within the heat treatment process is appraised on the basis of simulation calculations and experimental investigations.

The flexible jet quenching has originally been developed for gaseous flow processes [2]. But experimental investigations on the quenching of project relevant shafts of SAE 5120 (EN 20MnCr5) result in insufficient experiences because of the limited heat transfer by this quenching method, which is too low for successful asymmetric quenching for distortion compensation. However, by controlled quenching in liquid media (like water or hardening oil) and by means of jet or spray cooling with these media, the heat transfer process can be heavily intensified and should be able to generate much more potential for heat transfer [3].

The examinations on liquid jet quenching processes are based on the acceptance, that the generated flow velocity on the impingement area outside a liquid jet should be strong enough to suppress any boiling phenomena (vapour layer, nucleate boiling) on the heated surface. Based on this assumption, a pure convective heat transfer from the workpiece surface to the incident flow will realize extremely high cooling rates and for this reason high heat transfer rates. For the numerical simulation of this kind of heat transfer processes, Computational Fluid Dynamics (CFD) simulations are done for solving the one-phase flow and heat transfer problem for liquid media. For achieving accurate results within the simulation of heat transfer based on impinge jet flow, the complete flow and heat transfer problem of the jet quenching process on cylindrical shafts could be rebuilt in 3D-CFD-simulations. The chart of Figure 1 shows the results of Heat Transfer Coefficient (HTC) allocations along a vertical scanline through the centres of impinging jets on a heated workpiece surface. It can be seen clearly that the dimension of liquid jet impingement causes significant higher HTC-profiles ($\alpha_M \approx 25$ kW/(m²K)) in comparison to results of the gas quenching process ($\alpha_M \approx 1$ kW/(m²K)). The high level of the local HTC shows the high potential of the liquid quenching process by impinging jet flows.

However, the application of liquid jets in practical quenching experiments showed that the impressed local HTC are too high for a useful control of asymmetric quenching conditions. On the other hand, the use of multiphase atomizers instead of one phase liquid jets should allow the adjustment of impressed heat transfer conditions with the choice of spray parameters (liquid mass flow and gaseous pressure). For this reason, it is possible

* Corresponding author: svenschu@iwt.uni-bremen.de

to set up local heat transfer conditions for generating HTC results in useful ranges between pure gas and liquid quenching processes, as shown in Figure 1.

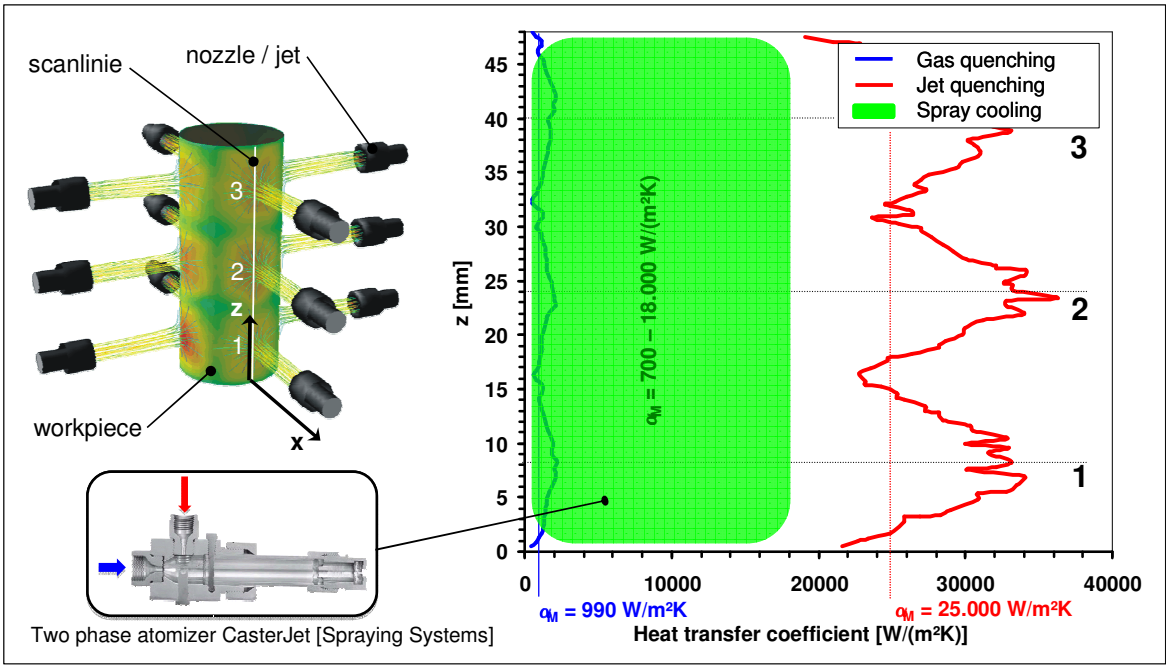

Figure 1: Heat Transfer Coefficients for different jet and spray cooling techniques

Through the use of multiphase atomizers for impressing intensive but controlled local heat transfer on surfaces, it is also possible to avoid vapour layer formation on heated workpiece surfaces [4]. For that, spray of fine liquid and an overlaid gaseous flow is impinged on the heated surface. The liquid fraction of that spray is restricted so that no closed liquid film with a vapour layer underneath is built. On that process, the efficiency of a complete evaporative cooling can be ideally reached.

In this contribution the spray cooling process by controlled two phase sprays of air and water is described. The use of this cooling method for quenching of workpieces in a spray field offers the potential to generate local cooling conditions in the heat treatment process. It builds the process technological basis for a specific distortion compensation during quenching processes on workpieces.

Methods

For analysis of the spray cooling process, a twin fluid atomizer with water and air is used. This nozzle (type CasterJet from the manufacturer Spraying Systems Co.) is typically used for cooling in steel casting processes and is assigned by very high (up to 1600 kg/h) mass fluxes for the liquid phase. The spray pattern of this atomizer type is a flat-spray, which is the optimal geometry for a homogenious cooling over the height of the given cylindrical workpiece, seperated in a field of four atomizers over 360°. A spray pattern with typical operation parameters (water mass flow 100 kg/h, air pressure 0.3 Mpa) and position related to the workpiece geomety is shown in Figure 2. The given operation parameters (air pressure and water mass flow) and especially the operation boundaries are examined and defined in experimental investigations. The analysed workpieces are steel shafts of AISI 5120 (lengh 200 mm, diameter 20 mm) from the process chain of the SFB 570.

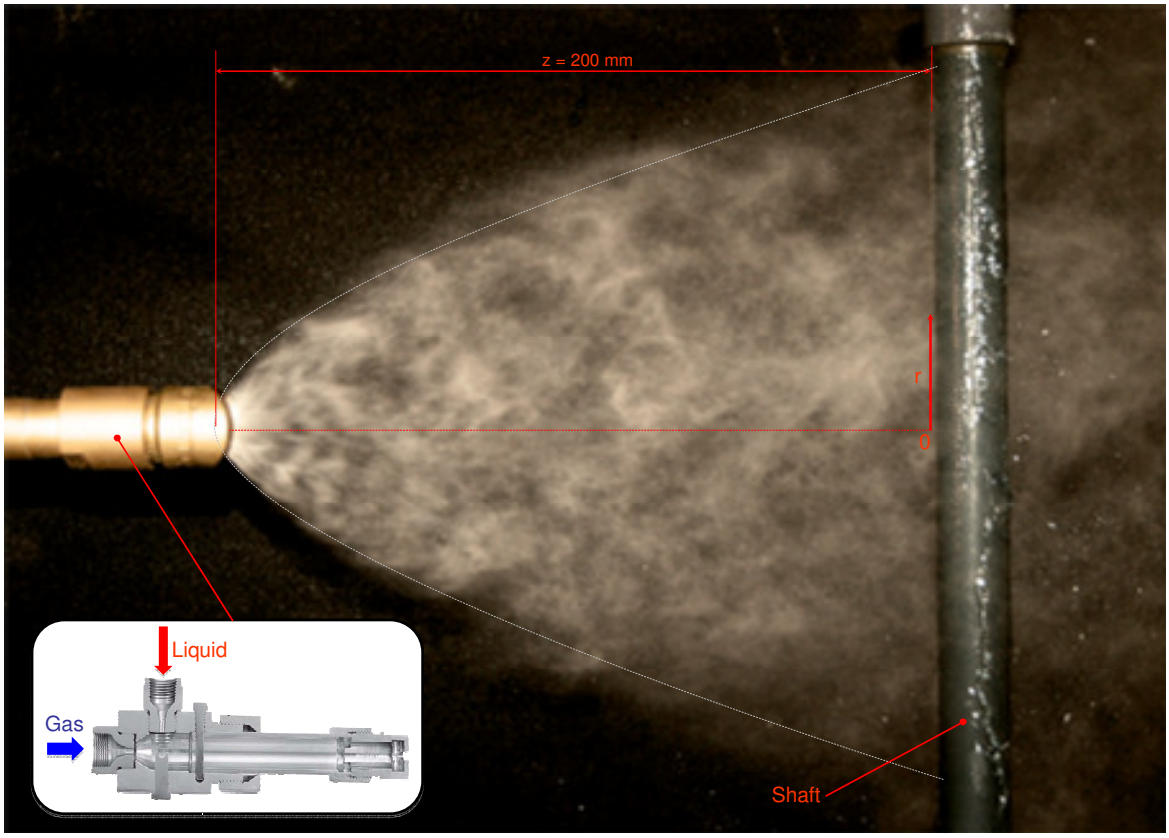

Figure 2: Two phase atomizer arrangement for spray cooling

For efficient estimation of the spray process a spray characterization in combination with evaluation and planned measurement of Heat Transfer Coefficients, according to [5], are performed. The spray characterization consists of drop diameter measurements by using Laser Diffraction Techniques and droplet velocity examinations done by Particle Image Velocimetry (PIV). The measurement of liquid mass flux distributions is done by patternators.

Operating performance of the nozzle

The analysis for spray characterization was done with the inner mixing, two phase atomizer for selected working conditions. The carried water mass flow varies between 40 and 400 kg/h by impressed air pressure of 0.1 to 0.3 Mpa. With these working conditions it is possible to adjust a variety of spray types with different dimensions of drop diameters, velocities and impingement densities. It is assumed that the nozzle builds a symmetric spray profile related to the spray axis. With that, the drop attributes can be represented as functions depending on the distance from the nozzle (z) and distance from the center line (r).

Figure 3 shows the adjacent air mass flow of the used nozzle, depending on the water mass flow for different adjusted air pressures. It can clearly be seen that the air mass flow depends on the water mass flow, which is reduced by increased air pressures. The ratio between air and water mass flow (Gas-Liquid-Ratio) averages from 0.04 to 0.5 for adjacent air pressures.

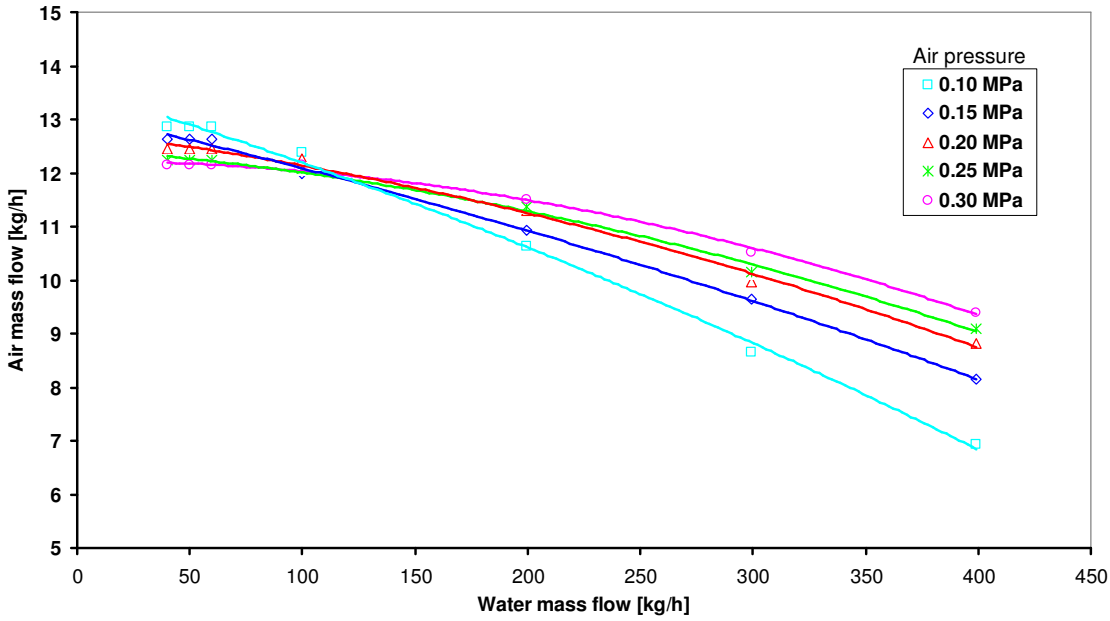

Figure 3: Mass flow ratio for CasterJet nozzle

Drop diameter

The drop diameter measurements were executed with a particle size analyser by laser diffraction techniques. The result of a drop size distribution measurement with the CasterJet nozzle on one point inside the spray pattern is shown in Figure 4 as a volume density allocation and volume cumulative distribution. The represented measurement were done on a distance of z = 200 mm in the centre of the spray (r = 0) by water mass flow of 300 kg/h and air pressure of 0.3 MPa. The median diameter $d_{50,3}$ of that monomodal allocation was determined to be 43 µm.

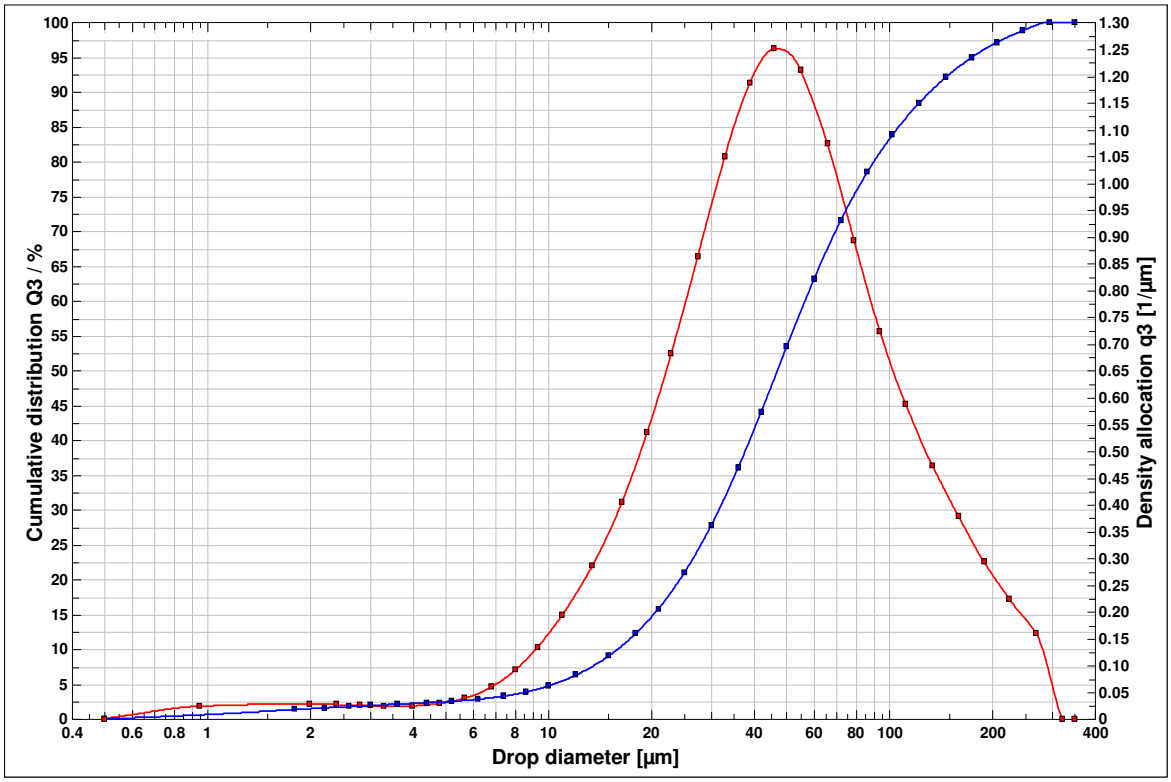

Figure 4: Volume density allocation and volume cumulative distribution

For further estimation of the measured drop diameters, the mean volumetric drop diameter d_{30} is used. The d_{30} is the characteristic dimension for examinations of water drop evaporation processes. Here, it is used for the calculation of the drop dependent Weber number and the evaluation of Heat Transfer Coefficients.

Drop velocity

The distribution of the water spray drop velocities is measured by means of the Particle Image Velocimetry (PIV) method. The principle of the PIV measurement method is sketched in Figure 5a. In Figure 5b one of a pair of recorded process images shows as an example the spray pattern on water mass flow 300 kg/h and air pressure 0.3 MPa.

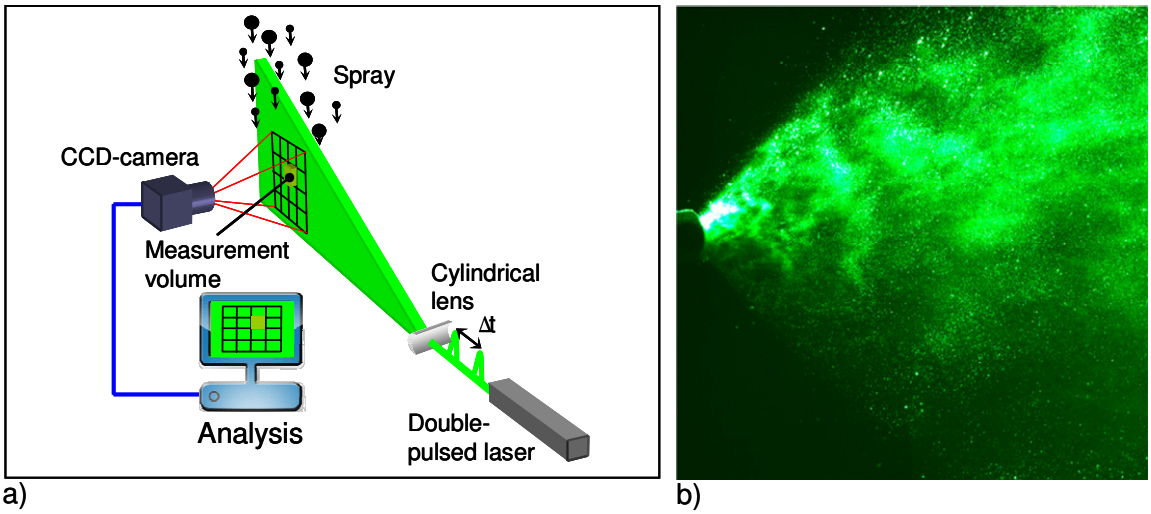

Figure 5: PIV velocity measurement method

The drop velocities were measured on the relevant distance z = 200 mm for a free spray and nearby to the workpiece surface, but no significant changes were found.

Impingement density

For measuring the water impingement density, a patternator as principally sketched in Figure 6 was used. The patternator consists of a number of tubes which are positioned in a parallel line under the spray pattern. The patternator tubes are separately collecting the water amounts for a defined time Δt, which are weighted after the experiment.

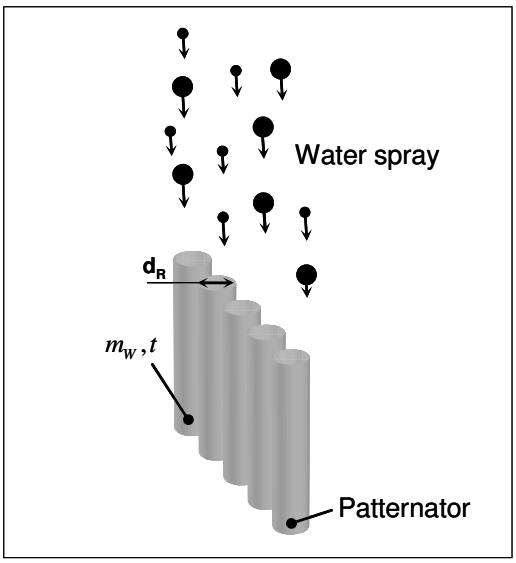

Figure 6: Patternator system

The amount of water (m_W) collected can be used in the equation:

$$\dot{m}_S = \frac{m_W}{\pi/4 \cdot d_R^2 \cdot \Delta t}, \tag{1}$$

with the inner diameter of the tubes (d_R). With (1) the averaged water impingement density for relevant positions under the spray axle can be calculated.

Heat Transfer

The basis for a possibly high heat transfer in spray cooling processes is a high evaporation efficiency. It increases by decreasing drop diameters and is often used in dependency to the Weber number, whereby the drop behaviour during workpiece surface impact can be characterized and evaluated by approaches of Bolle/Moureau and Berg [6, 7]:

$$We = \frac{u^2 \cdot \rho \cdot d}{\sigma}. \tag{2}$$

On Weber numbers above 80, a liquid film is built on the heated surface after drop impact. The film next collapses in a number of small drops which results in a high heat transfer. For setting up a high evaporation efficiency, it is necessary to adjust the spray for getting Weber numbers above 80.

For calculation of Heat Transfer Coefficients, the empiric approach of Puschmann [4] was primary chosen:

$$\alpha_C = \dot{m}_S \cdot 16{,}8 \cdot u^{0{,}12} \cdot d^{-0{,}29}. \tag{3}$$

Puschmann could show in his work, that the results for calculated Heat Transfer Coefficients in spray cooling processes are directly proportional to the water impingement density for heated surfaces above the Leidenfrost temperature. Here the drop diameter and velocity only have a secondary role in comparison to the water impingement density.

Results

Impingement density

In Figure 7 the measured impingement density for a water mass flow of 300 kg/h is displayed for the distance to the spray axis in dependence to the selected air pressures. The measured data was transformed by a curve approximation to reach the typical bell profile. The maximum of the impingement density is now found above the centre of the spray axis (r = 0). It can be seen, that the impingement density decreases from the maximum by a increasing distance to the centre. On increasing air pressure, the qualitative curve trend is kept up and the impingement density increases proportional to the air pressure. From the measuring point r = ±200 mm, the impingement density is independent of the air pressure.

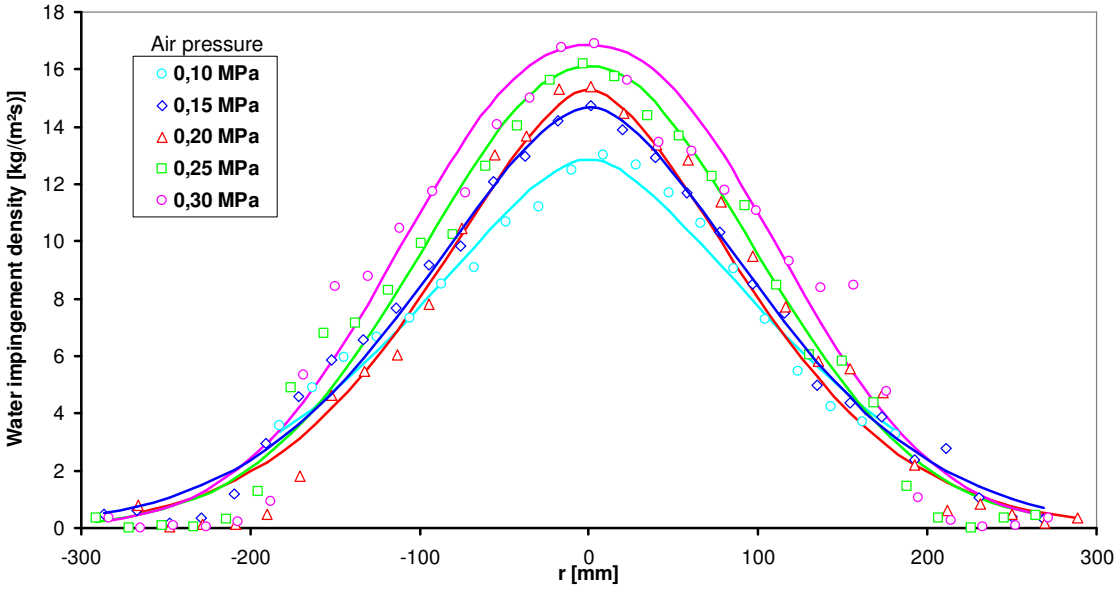

Figure 7: Water impingement density

Drop characteristic

The CasterJet nozzle was used for measurements by different water mass flows and air pressures. The result of the drop diameter measurements on the spray axis for a distance of z = 200 mm is shown in Figure 8. The chart shows the dependence of the mean volumetric diameter d_{30} with the water mass flow and air pressure. It can be seen that the d_{30} increases with increasing water mass flow, but it shrinks in proportion to increased air pressures.

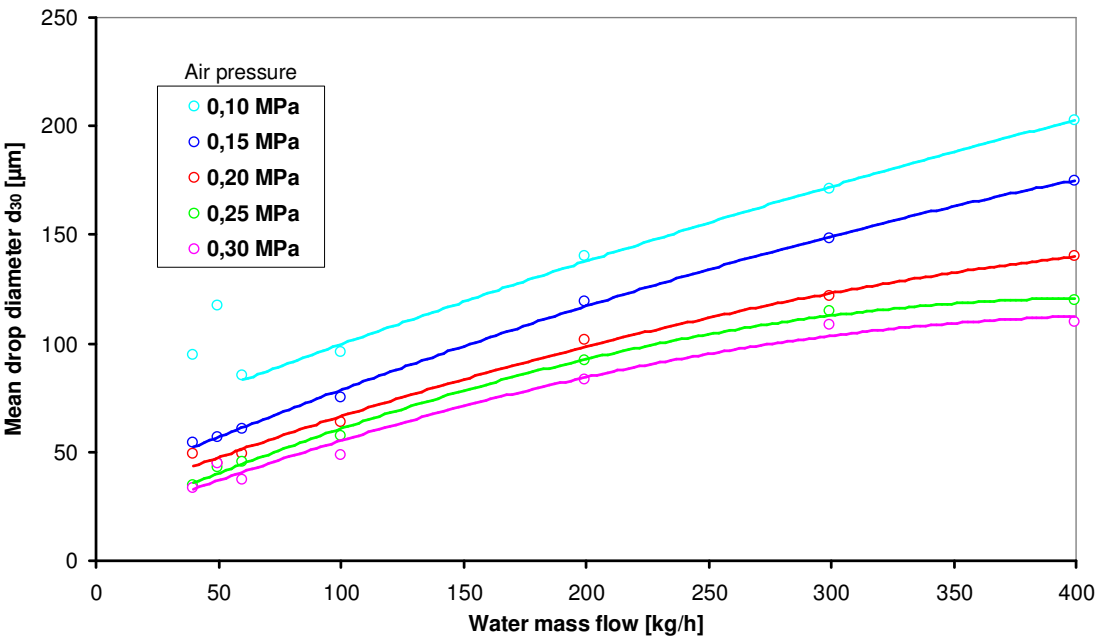

Figure 8: Mean drop diameter

The local drop velocities inside the spray were measured by the use of PIV method. Figure 9 shows exemplarily results of velocity measurements for typical water mass flows by air pressure 0.15 MPa. The trend of the averaged velocities for an increased distance to the spray axis shows a maximum on r = 0 and a light decrease of the velocity for outer ranges.

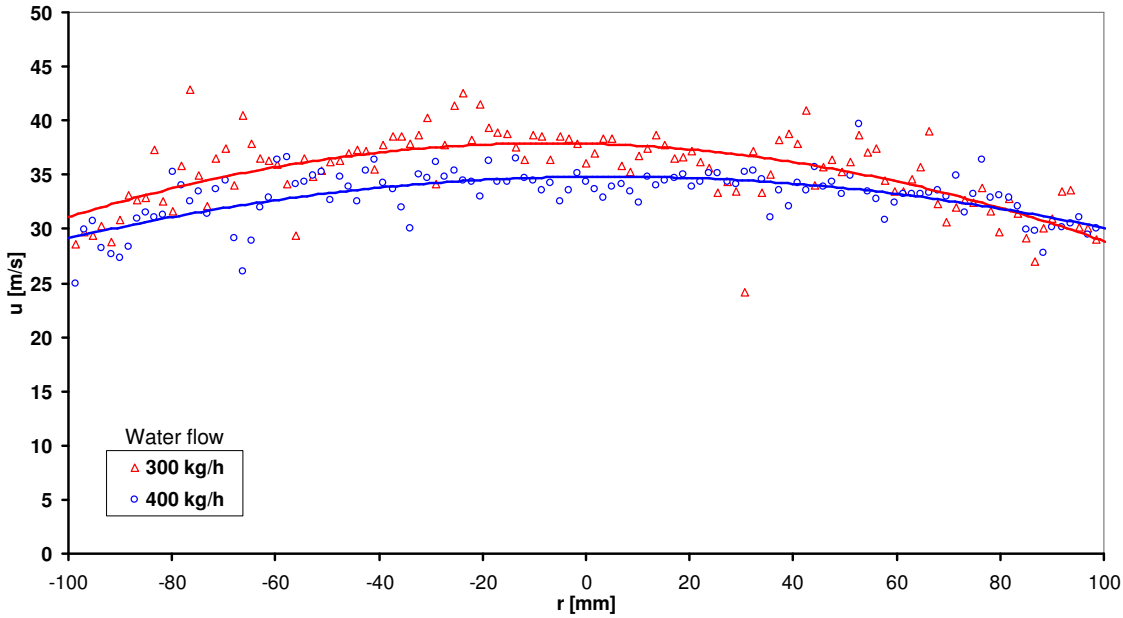

Figure 9: Drop velocity measurement

Heat Transfer Coefficient

With knowledge of the drop diameter and velocities on different points inside the spray it is possible to calculate local Weber numbers by equation (2). The calculated Weber numbers should validate the applicability of the used approach for calculating local Heat Transfer Coefficients by equation (3). The calculations results Weber numbers higher than 85 for every spray parameter combination (water mass flow and air pressure). Based on this, equation (3) is valid.

By [4] the calculated Heat Transfer Coefficients are direct proportional to the water impingement density. The impingement density was measured on different positions inside the spray for the distance of z = 200 mm. Figure 10 shows results on the dependence of the water mass flow and used air pressures.

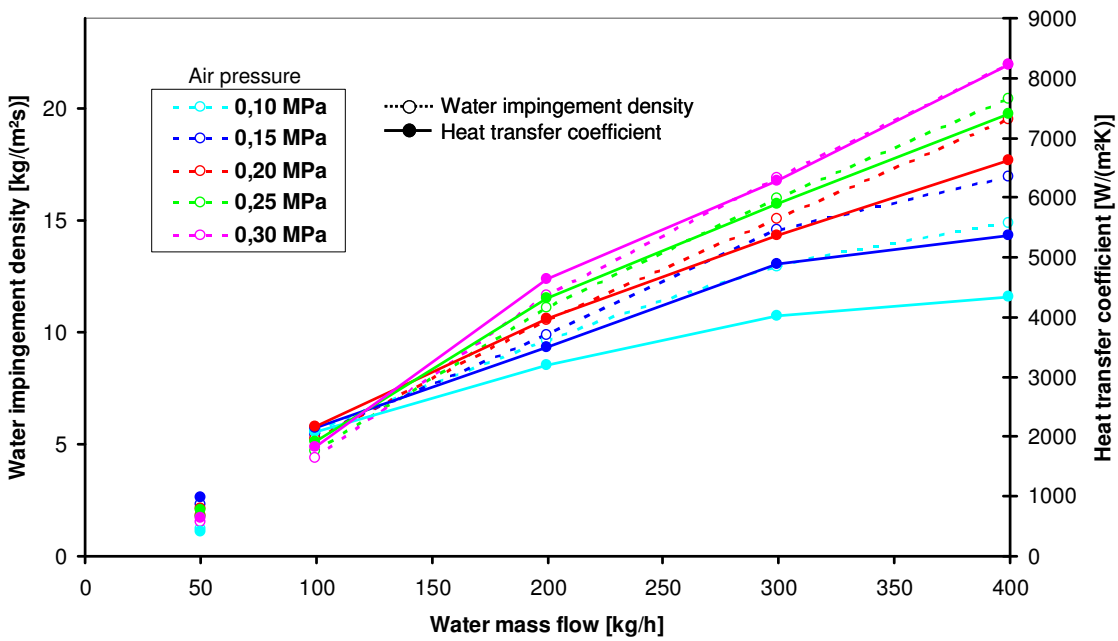

Figure 10: Heat Transfer Coefficients

The determined results from the spray characterization for drop diameter, velocities and impingement densities are used to calculate Heat Transfer Coefficients by equation (3) for the position r = 0. The results of these calculations are also included in Figure 10 for direct comparison. Based on this, a direct proportionality of the water impingement density to the Heat Transfer Coefficients can be validated under consideration of the adjusted air pressures.

Conclusion

The spray cooling process was analyzed by use of a special two-phase atomizer. That atomizer is assigned by very high mass fluxes for the liquid phase. It should be used for generating asymmetric cooling conditions on workpiece quenching processes. For analyzation, a spray characterization consisting of drop diameter, drop velocity and impingement density measurements was performed. Based on the results of the spray characterization, Weber numbers and Heat Transfer Coefficients were calculated for quantification of the spray cooling process.

The knowledge of the calculated Heat Transfer Coefficients for chosen operation parameters builds the basis for continuous analysis and validation of the cooling effect by asymmetric quenching of workpieces by spray cooling. Specimen cooling curves in spray cooling processes will be discussed and different calculation models for Heat Transfer Coefficients in spray cooling processes will be analysed for validation.

Acknowledgement

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Nomenclature

d	Drop diameter [µm]
d_{30}	Mean volumetric drop diameter [µm]
$d_{50,3}$	Median drop diameter [µm]
d_R	Patternator diameter
$\dot{m}_S$	Impingement density [$kg \cdot m^{-2} \cdot s^{-1}$]
m_W	Water mass [kg]
r	Distance to spray axis
t	Time [s]
u	Velocity [$m \cdot s^{-1}$]
We	Weber number
z	Distance [mm]
α_C	Convective Heat Transfer Coefficient [$W \cdot m^{-2} \cdot K^{-1}$]
α_M	Mean Heat Transfer Coefficient [$W \cdot m^{-2} \cdot K^{-1}$]
ρ	Density [$kg \cdot m^{-3}$]
σ	Surface tension [$N \cdot m^{-1}$]

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