EXPERIMENTAL AND NUMERICAL STUDY OF GAS JET COOLING AT HIGH REYNOLDS NUMBER

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ABSTRACT

In order to characterise the maximum possible cooling rate for AHSS with non-oxidizing gas a new test bench has been constructed at VKI, combining 1 central round jet surrounded by 6 jets. The facility allows jet Reynolds number up to 200,000, value for which very few experimental data are available in the literature. Results are obtained for different normalised standoff distances Z/D (ratio between the outlet of the jet and the plate divided by the diameter of the round jet) and for perpendicular and 10° angle of the plate with respect to the array of jets. The heat transfer coefficient is obtained by application of the active quantitative infrared thermography. A plate at constant and uniform heat flux is used. A mean heat transfer coefficient is defined on a characteristic cell. The evolution of the Nu-number and h (W/m²/K) obtained by increasing the Reynolds number to this high value are presented.

The experimental data are compared with numerical simulation for the validation of turbulent heat transfer. The numerical simulations are afterwards performed taking into account the industrial temperature (higher temperature difference between the jets and the metal sheet). A comparison of the heat transfer coefficient and the Nusselt number is made between the laboratory and industrial context.

Keywords: gas cooling, round nozzle, convection, AHSS

INTRODUCTION

Due to the demand for multi-phase high strength steels by the automotive industry while keeping a lean composition for cost reasons, the cooling rate after annealing has increased dramatically along the years. In the 80ies, the standard cooling rate was in the range of 50°C/sec for 1 mm strip thickness. This was to promote the precipitation of carbon in order to make DDQ grades. Nowadays, values as high as 110 to 120°C/sec are demanded for a temperature drop typically between 800 and 250°C. This requires to remove in average 330 to 360KW/m² per side of strip.

Such high cooling rates are quite easy with water thanks to boiling as well as high specific heat capacity. However, this method is not possible for hot dip galvanizing due to the fact that the surface cannot be oxidized before dipping in order to ensure good coating adhesion and surface quality. Therefore, the present technology consists in blowing very strongly a non oxidizing gas on the hot surface located very close to the nozzle. The exit velocity from the nozzles is typically ranging from 100 to 150m/sec and nozzle to strip distance as low as 40mm.

Cooling by Gas convection is well known. Correlations exist to predict the heat transfer coefficient in relation with the Reynolds number and the gas physical properties through Prandtl and gas conductivities formulas. A simple analysis shows that in order to reach the 350KW/m² the average
heat transfer coefficient need to be in the range of 750 W/m².K when the blown gas temperature is only 35°C.

The available published work for high heat transfer coefficient is quite rare. Well known and used for both slot and round nozzles are the Martin correlations [1]. They have however some limitations for round nozzles Reynolds number is limited to 100000

- Distance to diameter ratio (Z/D) between 2 and 12. It is however worth mentioning that Attalla [2] shows that the coefficient is maximum for a ratio of 6.
- The correlations from Martin have been identified by mass transport analysis and so not real heat transfer
- the case of a high temperature difference (higher than 400°C) between the strip and the gas is not considered

The work done and described has the objective to determine the heat transfer coefficient as well as Nusselt number that are reasonable to consider for Reynolds number up to 150000 with a configuration of array of round nozzles. The investigated range of Z/D is from 5 to 6 which corresponds typically to an industrial design with 14mm nozzle diameter and a nozzle to strip distance from 70 to 85mm

**ADRESSING THE PROBLEM**

The methodology followed consists in a simultaneous experimental approach as well as CFD. The reason of that choice is based on the fact that it is impossible to characterize and measure at the lab the industrial situation where the cooling section may be as big as 20m long for a strip width as wide as 2 meters. An example of a typical industrial blowing plenum is in fig 1. The Laboratory experimental design as described in detail below, has the objective to measure the average heat transfer coefficient on a “cell” and compare the results with published data as well as numerical results done on a similar design (fig 2).

![Figure 1 - Typical section of an industrial cooling box, 3m high, 2 m wide](image)

![Figure 2 - Geometry of a typical cell (7 jets) and flow structure of the impinging jets](image)

The numerical approach is conducted to compare the computed heat transfer coefficient and then estimate the reliability of the results obtained with the classical published model [1]. This is a key because CFD is the only way to estimate the industrial heat transfer on a full strip width when hydrogen is used as well as high strip temperature both of which being impossible to test at lab scale.
MEASUREMENT OF HEAT TRANSFER IN THE LAB

Measurements of the heat transfer coefficient have been performed using quantitative infrared thermography. The investigation was focused on an ARN configuration, consisting of 1 central jet surrounded by six additional as sketched and visualized in Figure 3. The arrangement has the following geometrical characteristics:

- Diameter of the nozzle: $D=14$ mm,
- Standoff distance: $Z=50$, 70 & 85 mm
- Nozzle spacing: $W=70$ mm.

The flat plate on which the jets are impacting is uniformly heated by Joule effect. The heat input is constant and fixed at 4000W/m². The strip temperature is measured by an infrared camera. The thermograms are analysed by an in-house DIP program allowing the determination of the heat transfer coefficient knowing the imposed heat flux and the incoming jet temperature, measured inside the Plenum. The infrared radiometer is a FLIR 6555sc thermocam with an announced NETD of less than 30 mK. The camera provides images of 640*480 pixels. The strip emissivity has been checked by comparing the measured strip temperature with thermocouple and IR Camera.

![Figure 3 - ARN](image)

![Figure 4 - laboratory test bench](image)
Test have been performed for a Reynolds number ranging from 40000 to 160000. The Reynolds number is based on the mass flow passing through the tubes. This mass flow is measured using a calibrated diaphragm located upstream of the stagnation chamber (Plenum). The test bench with the different elements is illustrated in Figure 4.

Two different configurations have been tested: A standard configuration perpendicular to the target with a normalised standoff distance Z/D varying from 3.5 to 6. The second configuration consists of a 10° inclined plate keeping a constant Z/D for the different jets in which the nozzle to strip distance is constant for all nozzles. The two configurations are shown in Figure 5 with a typical heat transfer map obtained for a Reynolds number of approximately 80000. Corresponding temperature profiles along the centreline are also plotted. In the perpendicular configuration, we obtained a symmetric map of heat transfer coefficient.

The mean heat transfer value is extracted from a hexagon geometry centred on the geometrical impact point of the central jet, as illustrated in Figure 3. Values of mean heat transfer coefficient are obtained as function of the Reynolds number and are used as quantitative data to compare with correlations and CFD results.

In the second configuration (right part of Figure 5), all the seven different jets produce similar maximum heat transfer coefficients, but we observe a dissymmetry in the shape of the heat transfer map, linked to the inclination. The effect of the plate inclination can clearly be seen on the imprint of the central jet.
When blowing at high Reynolds number, the heat transfer coefficients are becoming very high and therefore the temperature differences between the incoming jet and the heated plate are decreasing a lot especially at the impact point. To guarantee enough accuracy for the determination of the heat transfer coefficient, it has been verified that the minimum $\Delta T$ was always higher than 10°C. An estimation of the error has been done by repeating the same experiment. It turns out that the results have an error bar of +/- 5% all measurements included: mass flow, gas temperature, heat flux and strip temperature. An example of the results obtained are in Table 1.

Table 1 Results from repeatability Test. Perpendicular situation

<table>
<thead>
<tr>
<th>Z=70mm Z/D=5 Re 156359</th>
<th>h hexagone</th>
<th>Deviation $h_{\text{mean}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>W/m²K</td>
<td>%</td>
</tr>
<tr>
<td>Test 0703</td>
<td>458</td>
<td>-4.4</td>
</tr>
<tr>
<td>Test 0708</td>
<td>523</td>
<td>+8.9</td>
</tr>
<tr>
<td>Test 0709</td>
<td>465</td>
<td>-3.2</td>
</tr>
<tr>
<td>Test 0710</td>
<td>454</td>
<td>-5.2</td>
</tr>
<tr>
<td>Test 0711</td>
<td>499</td>
<td>+3.9</td>
</tr>
<tr>
<td>MEAN</td>
<td>480</td>
<td></td>
</tr>
</tbody>
</table>

THE CFD APPROACH

CFD simulations have been performed in parallel to the laboratory experiments. The geometry of the computation domain is following exactly the geometry of the experimental set-up. Figure 6 and Figure 7 show the computation domain of the two configurations, with also the flow structure of the central jet illustrated by coloured pathlines. The inclination of the plate has an effect on the flow structure.

The mesh used in the CFD approach combines hexahedral elements and tetrahedral elements with a fine refinement close to the walls. The design has a total of 4480000 cells with a cell wall distance as low as 10µm on the area where the heat transfer is computed.

Simulations have been performed for different Reynolds numbers by changing the inlet mass flow of each tube and for different stand-off distance between the outlet of the tubes and the heated plate. A RANS approach is used with the k-ω SST turbulence model.

Mean heat transfer coefficient on the same hexagon defined from the laboratory investigation has been extracted from the numerical simulation.
The flow structure of Figure 6 has been validated using simple wool visualization as illustrated in Figure 8. The wool is emphasized by the white arrow.

RESULTS AND DISCUSSION

The measurement obtained on the lab stand for both perpendicular and incline strip are shown respectively on fig 9 and 10. Fig 11 compare the CFD results with some of the measures as well as the predictions proposed by the Martin correlations for the perpendicular situation. The color of the correlation formula fits with the color of the points (ex: Fig 9, 85mm). Fig 13 compares the results obtained by CFD, Martin and experiment for a perpendicular blowing and various Reynolds values. Finally, fig 14 show the computed h values for perpendicular and inclined plate for a 85mm distance.

The following observations can be made

- The measured heat transfer is higher for inclined plate than for the perpendicular situation. This could be explained by the totally different structure of the flow as shown on fig 6 and 7
- For all cases the exponent in the power correlation is much lower than that predicted by Martin: 0.35 to 0.5 for measures versus 0.68 for Martin
- Measurement and CFD gives similar values in case of Reynolds of 80000 and 70mm distance.
- The dependent with distance obtained by CFD flows the same rule than that obtained by Martin (exponent ~ -0.45) but Martin values are lower than the CFD ones
Figure 14 shows the different results obtained for a stand-off distance of 70 mm (Z/d=5). Martins’ correlation applied to this ARN configuration gives the lowest value of heat transfer coefficient and the exponent in the correlation is equal to 0.66. The different experimental points show that at low Reynolds number, higher heat transfer coefficient have been measured compared to Martins’ correlation while, when the Reynolds number is above 80000, the heat transfer coefficient is getting closer to the values of the correlation. The exponent in the experimental data correlation is reduced to a value of 0.41. Heat transfer coefficients obtained by CFD simulations are higher than Martins’ correlation with an exponent of 0.57 for the inclined configuration and of 0.69 for the perpendicular configuration. Therefore, great care should be taken when using CFD for the design of a system working at Reynolds number above 80000. CFD simulation overpredicts the heat transfer coefficient.
THE REYNOLDS QUESTION FOR HIGH STRIP TEMPERATURE

Literature as well as our experiments investigate the effect of Reynolds on the heat transfer coefficient. This dimensionless number is well known and describe the ratio of the fluid dynamic versus the viscous resistance. It can be calculated for a round nozzle by:

\[
Re = \frac{\text{gas density} \times \text{Velocity} \times \text{Nozzle diameter}}{\text{gas viscosity}} = \frac{\text{gas massflow} \times 4}{\text{Nozzle Diameter} \times \pi \times \text{Viscosity}}
\]

Fig. 12 - Comparison of Heat Transfer Coefficient as prediction by [1], CFD and measured. Correlation function

Fig. 13 - Heat Transfer Coefficient obtained by CFD for Perpendicular and 10°

Fig 14 – Results for Z/D=5
Formula also use to correlate Reynolds with Nusselt to solve the problem a gas conductivity but the heat transfer coefficient is found by solving for $h$:

$$ Nu = h \frac{\text{Nozzler Diameter}}{\text{gas conductivity}} $$

On a phenomenological standpoint, the convection coefficient depends on the heat conduction through the boundary layer that is defined by the gas impact. It is well known that the gas density as well viscosity and conductivity varies with temperature. However in industrial situation of fast cooling after annealing the substrate to cool is at high temperature. It is then expected that for a defined gas flow and distance the boundary layer of the gas should be thicker than on a cold substrate but also that the conductivity is improved. It becomes not obvious to predict how will be affected the heat transfer coefficient by the substrate temperature. Lab experiments have never been done in that sense due to the extreme difficulties to solve. In addition, using other gas than air like for example a mixture of H2 and N2 is almost impossible.

So the question comes about what is the temperature to consider for the estimation of the heat transfer coefficient in the correlations $Nu=f(Re,Pr)$.

The numeric approach has been conducted using the geometry of the experimental design. The nozzles are perpendicular to the sheet and located 70mm away. The gas flow per tube is fixed to 0.021 kg/sec. The substrate temperature is fixed at 350 and 650°C. 2 types of gas are simulated: air at 295°C and a mixture of N2+30%H2. The published formula are used to predict the variations of the gas properties with temperature. The results are in Table 2 in the column named “h CFD”. The CFD values are averaged on about 600 steps but it must be mentioned that even after 15000 steps, the $h$ value oscillates in a range of +/- 5W/m²K.

It turns out that the computed heat transfer coefficient is about 10% higher for the low strip temperature than for the hot one and this is true for both gas mixture. Strange also to see that when the heat input of the strip is fixed as it has been done in the simulation of the experimental stand, the computed coefficient is 385W/m²K, a value in between the 2 strip temperatures.

The right part of Table 2 reports the computed heat transfer coefficient as well as Nusselt and Reynolds value in 3 cases using the correlations proposed by Martin [1]

- Gas density and viscosity at 30°C
- Gas density at 30°C but viscosity and conductivity at the average temperature between strip and gas
- Gas density at 30°C but viscosity and conductivity at the strip temperature

The drop in Reynolds value is due to the gas viscosity increase with temperature. This impacts directly the Nusselt value. Finally, the conductivity, better at higher temperature, allows to calculate the heat transfer coefficient. The difference between the various cases is quite high and this rise a real problem for the design of the cooling sections. It is interesting to compare with necked eye the various results in fig 15 for AIR and 16 for N2+30%H2. It seems that the best prediction should include the gas properties below the strip temperature but over the average T gas blown-T strip.

When the strip is inclined, the difference between hot and cold strip still exists and the values of the heat transfer coefficient are a little lower than for the perpendicular situation (table 3)
Table 2 Comparison of $h$ obtained by CFD and various assumption to compute the gas viscosity in case of high strip temperature

<table>
<thead>
<tr>
<th>Nozzle Perpendicular to strip 70mm distance, 14mm diameter</th>
<th>h CFD</th>
<th>h MARTIN W/m²/K</th>
<th>Nusselt Martin</th>
<th>Reynolds</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas mass flow per nozzle kg/sec/nozzle</td>
<td>Strip Temp °C</td>
<td>Rho 30°</td>
<td>μ 30°</td>
<td>Rho 30°</td>
</tr>
<tr>
<td>AIR</td>
<td>0.021</td>
<td>350°C</td>
<td>473 +/-5</td>
<td>325</td>
</tr>
<tr>
<td></td>
<td>650°C</td>
<td>440 +/-5</td>
<td>325</td>
<td>416</td>
</tr>
<tr>
<td>30% H2</td>
<td>0.021</td>
<td>350°C</td>
<td>785 +/-5</td>
<td>594</td>
</tr>
<tr>
<td></td>
<td>650°C</td>
<td>743 +/-5</td>
<td>594</td>
<td>750</td>
</tr>
</tbody>
</table>

Table 3 Heat transfer coefficient for inclined plate. Ratio Perpendicular-Inclined

CONCLUSION

The heat transfer coefficient of round nozzle blowing on a surface perpendicular and with a 10° angle has been investigated experimentally and with CFD for a Reynolds range as high as 150000. The Nozzle to strip distance ratio is 5. The situation of a cold gas cooling a hot surface is also addressed. The results can be summarized as below:

- The heat transfer coefficient measured and computed by CFD are over the correlations proposed by Martin [1]
- The measures as well as CFD results are quite similar for Reynolds number below 100000. Over that value, CFD gives significantly higher values than those measured. Exact cause need to be investigated
• The exponent coefficient $a$ in the correlation $h = K \cdot Re^a$ is found to be only around 0.4 for experiment but 0.7 from CFD results. This implies that the difference between computed and measured value becomes significant at Reynolds higher than 100000. The large difference between the exponent is presently not explained and cannot be attributed to errors in the measuring technique.

• The difference between perpendicular to the sheet and 10° inclined is quite small for both CFD and experimental results. It is however important to note that the general structure of the flow is totally different. This is quite different from what has been obtained by measurement.

• The question of determination of the Reynolds number in case of strip at high temperature (higher than 350°C) is discussed in order to identify the gas viscosity to consider due to its strong variation with temperature. The CFD results show that the heat transfer coefficient is about 10% higher when the substrate is at 350°C versus 650°C. It comes that the temperature to consider to compute the gas viscosity is quite difficult to identify because the recommendation looks in relation with the strip temperature.

• More generally, the predictions obtained by Martin are below the CFD and measured values.

More globally, it turns out that some points at this stage are not so clear and would need deeper analysis. The significant difference between CFD need to be understood and this is true not only for the high Re values but also for the effect of the inclined plate. Some future would should also be needed to better characterize the effect of the distance for both inclined and perpendicular plate.

**REFERENCES**
