A study of heat transfer from cylinders in turbulent flows by using thermochromic liquid crystals

by

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Roland Wiberg 2004 A study of heat transfer from cylinders in turbulent flows by using thermochromic liquid crystals
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Abstract
In gas quenching, metal parts are rapidly cooled from high temperatures, and the convective heat transfer coefficient distributions are of importance for the hardness and the deformation (the shape nonuniformities) of the quenched parts. Thermochromic liquid crystals (TLC) and a thin foil technique, were investigated and used for studies of a circular cylinder in axial flows, affected and not affected by upstream flow modifying inserts. Quadratic prisms in cross flows were also studied, a single prism, two prisms arranged in-line, and for four prisms arranged in a square pattern. In this study, particle image velocimetry (PIV) was used for visualization of the flow, giving physical insight to the convection heat transfer data. Further, relations of the type $Nu = CRe^k$ were established. The TLC and thin foil techniques were also used to indicate the dimensions of separated flow regions.

Descriptors: Fluid mechanics, wind-tunnel, turbulence, gas quenching, convection heat transfer, thermochromic liquid crystals, calibration, temperature measurement errors, thin foils, particle image velocimetry, cylinder in axial flow, flow modifying inserts, quadratic prisms in cross flow
Preface

This thesis is a study of 1) experimental methods for surface temperature and convection heat transfer coefficient measurements, and 2) the distributions of Nusselt numbers on a cylinder in axial flows and on prisms in cross flows, at high Reynolds numbers.

**Paper 1.** Wiberg R. and Lior N., Errors in thermochromic liquid crystal thermometry.

**Paper 2.** Wiberg R. and Lior N., Heat transfer from a cylinder in axial turbulent flows.

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CHAPTER 1

Introduction

This thesis is a part of a larger FaxénLaboratoriet project about gas quenching of steels. This project is ongoing for a few years, with participation of several Ph.D. and Master students. Some have worked with numerical simulations using different turbulence models etc, and some have done experimental investigations at the Department on Mechanics/ FaxénLaboratoriet KTH, and on-site, using facilities owned by the sponsors.

The present thesis is based on experimental investigations performed in the lab of the Department of Mechanics KTH. The focus of the work is on forced convection heat transfer, and methods on how to measure such convective heat transfer coefficient distributions for bodies and multi-body geometries and configurations common in gas quenching applications. The results from these measurements may be directly applicable for gas quenching purposes, or indirectly for validation of numerical methods for quenching improvements.

Quenching is the rapid cooling of metal parts, such as gears, cylinders, engine axles, bearing rings, etc., which makes the metal hard, by changing its phase structure. Not only must the cooling be rapid, but it is also very important to apply it in a way that will minimize distortions due to internal thermal stresses, a bearing-ring, for example, should remain round also after the quenching process.

In quenching, steel is uniformly heated to ca 800-900 °C, and thereafter rapidly cooled, typically in oils or salts baths. The use of gas as the coolant has several important advantages: the cooled parts do not need to be washed afterwards reducing the use of solvents, the need for ventilation and the risk for fire, the liquid coolants and the solvents pose an environmental problem, the quenched parts retain a very clean surface, and have less distortion Troell & Segerberg (1995) and Minarski et al. (2000).

The use of oil and salt solution coolants provides in general higher cooling rates than gas cooling because the cooling is based on boiling, but gas cooling can be improved by using higher gas pressures and velocities, by choosing flows, such as impingement, which provide high heat transfer coefficients, and by the use of gases with better cooling performance such as helium. Another advantage of gas cooling is the possibility of controlling the gas flow during the
1. INTRODUCTION

cooling process, as well as properly directing it to part locations according to cooling and uniformity need.

A typical facility for gas quenching uses a closed-loop system, where an electric motor-driven fan moves the gas, which flows a compartment where the hot parts being quenched are placed, then through a water-cooled heat exchanger, and back to the fan inlet. Single chamber vacuum furnaces and double chamber vacuum furnaces exist, the latter one has initially cold walls which provides higher cooling rates.

Different types of gases may be used for gas quenching. Argon, nitrogen, helium, and hydrogen were compared for that purpose (cf. Minareski et al. (2000)). Hydrogen gives the most effective cooling, followed by helium and then nitrogen. Hydrogen becomes flammable and potentially explosive when mixed with oxygen, and helium is expensive if not recycled (Holm & Segerberg (2000)). Commercial vacuum furnaces for gas quenching often use nitrogen or Helium at 10-20 bar and 50-150°C, at a quench zone Inlet velocity of 15-30 m/s 1. Other parameters which are of importance for gas quench chambers are the uniformity in the inlet gas velocity, and the pressure losses through the quench charge and the system pipes and ducts.

Forced convection heat transfer is usually characterized by the convective heat transfer coefficient, $h$, defined as,

$$h = \frac{q''}{T - T_\infty}$$

where $q''$ is the convective heat flux, $T$ the surface temperature of the solid which the fluid is exchanging heat, and $T_\infty$ the upstream fluid temperature. This relation is expected to be valid for fluid velocities well below the speed of sound, when $T = T_\infty$ at $q'' = 0$.

$h$ is often made dimensionless by dividing it with the reference parameter $k/L$, which can be seen as a heat transfer coefficient valid for steady state heat conduction through a layer of fluid, with the heat conductivity $k$, and a thickness equal to the body characteristic dimension $L$, which thus becomes the Nusselt number $Nu$,

$$Nu = \frac{hL}{k}$$

A large number of metal parts (a charge), may be simultaneously cooled in a gas quench furnace. As an example, the charge may consist of many cylinders in axial or cross flow. In the latter case, each tube-row except the most upstream one is in the wake of other tubes or objects which are upstream of it. To provide the typical magnitudes of some parameters in gas quenching, an example is given below.

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Nitrogen is filled into a furnace up to $P = 10$ bar at $T_{\infty} = 80 \, ^\circ \text{C}$, and during the cooling the fan gives the gas an inlet velocity of $U_{\infty} = 15 \, \text{m/s}$. 10 rows of hot solid cylinders at $850 \, ^\circ \text{C}$ made of a low alloy steel with the diameter $D = 30 \, \text{mm}$ and arranged in a staggered configuration (at the distances $2 \times 2 D$), are introduced into the flow. In the gaps between the cylinders the velocity is higher $U_{\text{m}} = 50 \, \text{m/s}$, which gives a Mach number of $M = 0.1$. For $M < 1$ the flow acts like it is incompressible, the variation in density and temperature etc in the flow are then small.

The Reynolds number $Re$ for the flow over a cylinder is then $4.4 \times 10^7$, for which the surface averaged Nusselt number is $1040$ and $h = \frac{1000 \, \text{W/(m}^2\text{K})}{(\text{from } Zulauskas \text{ and Zinggela (1985))}}$. Initially, the temperature rise of the gas as it cools this the tube bank can be calculated, using the heat capacitances and the flow rate, to be $80 \, ^\circ \text{C}$, and the effects of the natural convection and the radiation together are calculated to be only a few per cent compared to the forced convection heat flux, and decreasing during the subsequent cooling.

For a cooling time $t$ equal to $L^2/\alpha$ (Fourier number $Fo = \alpha t/L^2 = 1$), where $L = \text{volume/area}$ for the cylinder, it is estimated that heat has diffused from the core out to the surface, which for this cylinder occur after 7 seconds of cooling. The average cylinder temperature is, at this time, and using a lumped capacitance method, Incropera & De Witt (1990) ch. 5, estimated to be $700 \, ^\circ \text{C}$.

The Biot number for the cylinder is $\text{Bi} = hL/k = 0.2$, where $k$ is the heat conductivity of the steel. $\text{Bi}$ is an estimate of the temperature differences in the solid, compared to the differences between the cylinder and the gas. Based on this and for $Fo = 1$ the temperature differences in the solid can be estimated to $120 \, ^\circ \text{C}$, decreasing during the cooling period. The time to reach a desired average temperature of $150 \, ^\circ \text{C}$, is estimated using the lumped capacitance method to be 83 seconds. This is then an estimate of the total cooling time for this specific case.

Another issue which is worth mentioning, is the change of the surface temperatures in time during the cooling process, and what possible effect that may have on $h$. This question may be answered by a simple estimate of the characteristic times involved in the process. The time it takes for the gas to flow from one side to the other of the quenched body above can be estimated to 0.004 s, which is very small compared to the total quenching time estimated to 83 s. The local $h$ for the body, is therefore expected to be equal to $h$ for a body with a temperature constant in time.

Experimental data, for such a cylinder in cross flow, shows that $h$ also varies on the surface $\pm 50 \, \%$ and sometimes more around the cylinder perimeter, for tube bundles Zulauskas & Zinggela (1985), and for a single cylinder at high $Re$ Achenbach (1975). Such local heat transfer coefficients nonuniformities may produce non-uniform hardness and out-of-roundness, and are thus one of the main motivators for the thesis. The geometries studied in this thesis are a
cylinder in axial flows (in paper 2), and single and multiple quadratic cylinders (prisms) in cross flows (in paper 3).

1.1. Some investigations related to gas quenching

Investigations of the gas flows and the resulting convection heat transfer distributions connected to gas hardening problems, have been an on-going issue for some years for the heat treatment group at FaxénLaboratoriet, involving a few M.S. and Ph.D. students. A review is given below, showing the main contributions from FaxénLaboratoriet and some other publications in this direction so far.

An experimental study was performed using a real gas quenching furnace, in which steel cylinders (400 mm long, 30 mm in diameter) were placed in a staggered charge configuration. The cylinders were initially heated to 850°C and then cooled in a flow of nitrogen at 6 bars. During the cooling, the temperatures of the test cylinder were measured using thermocouples, Lind (2001). Noise in the measured data was a problem, but local heat transfer coefficients could be calculated for different temperature levels during the cooling.

The main conclusion from the experiment was that $h$ on the cylinder did not change with the surface temperature, besides for the initial transient fill up of gas into the system, $h$ was in practice constant and independent of $T - T_{\infty}$. Constant $h$ was also seen from measurements in Nitrogen and Helium and using a cylindrical heat flux sensor, Edenhofner (1996). These results encourage heat transfer coefficient measurements (and simulations), to be performed at lower near ambient temperatures, using wind tunnels etc., and which simplifies and increase the number of methods available for the measurements.

Other investigations also indicates this independence of $h$ for gases. Primarily related to heat exchanger design, numerous measurements of the flow of and the heat transfer coefficients were performed for single cylinders in cross flow, cylinders in staggered configurations, and cylinders in in-line configurations, and at various distances in between, and for convection flows of oil, water and air, Zakauskas (1989) ch. 11 and 13. For all configurations and fluids, and for $Re > 2 \times 10^5$, the surface averaged $Nu$ was successfully expressed as,

$$Nu_f = C Re_f^m Pr_f^n \left( \frac{Pr_f}{Pr_s} \right)^p$$

(1.3)

where $m = 0.8$, $n = 0.4$ and $p = 0.25$, the subscript $f$ indicates properties evaluated at the fluid temperature, and $s$ properties evaluated at the wall surface temperature. The characteristic length for $Re_f$ and $Nu_f$ was the diameter of the cylinder. The constant $C$ is 0.023 for a single cylinder, 0.038 for a tube in a inner row of a staggered tube bundle, and a function of the distances between the cylinders for an in-line tube bundle. The exponent $n$ may vary around the
Perimeter of the cylinder Zukauskas (1989), and for other flows and geometries vary in the range \( n = 0.33 \text{--} 0.43 \) Incropera & De Witt (1990) ch. 7.

\( Pr \) varies strongly with the temperature for water and typically by several orders of magnitude for oils in the quenching temperature range of interest. For gases (air, nitrogen, helium) at pressures and temperatures relevant for gas quenching, (say 0-1000°C and 1-50 bar), \( Pr \) is constant within a few percent, which indicates an independence of \( Nu \) on the temperature differences between the surface and the fluid. Equation 1.3 can thus be simplified for gases to

\[
Nu_f = CRe_f^m
\]

(1.4)

\( Nu_f \) numbers measured for a specific \( Re_f \) for one of these gases are therefore expected to be valid with fair accuracy also for the other gases at the same \( Re_f \).

Another investigation of convective heat transfer from a cylinder in cross flow was performed by the author, using a test cylinder of stainless steel in a wind tunnel, at flows with different velocities and turbulence levels (Wiberg (1999)). In this M.S. thesis a review was given of the different parameters that affect convection heat transfer. Nusselt numbers (\( Nu \)), surface pressures, the positions of flow separation, and the surface shear stress, were measured around the perimeter of the cylinder for \( Re \) up to \( 3 \times 10^5 \) and TLC was used for the measurement of the surface temperatures. Parts of these results were further published, see Wiberg et al. (2000b) and in a shorter version, Wiberg et al. (2000a).

Numerical simulations for a circular cylinder in cross flow were performed, in the Faxén Laboratory heat treatment project Lind et al. (1998). Three \( k-\epsilon \) turbulence models were tested, and one of them the \( k-\epsilon \) LSY-CG, produced the best heat transfer coefficients in comparison to available experimental data. Still differences up to 40% was seen in the separated region.

This study was extended by cooperation with the Swedish Institute for Metals Research (SIMR). In this work the \( h \) values obtained from the numerical simulation were applied to the surface of a long tube made of bearing ring steel, 32 mm in diameter. The distributions of the relative amounts of the Perlite, Bainite and Martensite phases, and of the hardness, as well as the distortion, were computed using a SIMR code, and were found to be affected by the non-uniform cooling, Thuander et al. (1999).

Further numerical work were performed. A finite length cylinder in axial flows and cylindrical rings were investigated, using different turbulence models. For the cylinder in axial flow which include a large separated and reattached flow region, agreement were found within 30% in comparison to measured heat transfer coefficients (performed by the author, paper 2 in this thesis), using an omega-Reynolds stress model, M. S. thesis by Brandt & Torquist (2003).
1. INTRODUCTION

Another approach in gas cooling is the use of gas jets, single or multiple, for cooling many parts distributed in a single layer or individual parts. The cooling rate may be controlled both in space over the surface, and in time during the cooling and the cooling rates may be much higher than for uniform flow gas quenching, and higher than for oil quenching, Wunning (1993) and Edenhofer (1999). Different correlations for convective cooling with jets were studied, Ferrari et al. (2003). Based on these empirical heat transfer coefficients, simulations of the heat flow and the material structures inside a ring of steel was performed, and an optimal configuration of jet nozzles was suggested.

A parameter of interest in gas quenching is the pressure drop in pipes and ducts and the uniformity in inlet velocities etc. Improvements in these parameters may reduce the power consumption and increase the quenching performance. Such studies have been performed at FaxénLaboratoriet, but are outside the subject of this thesis and are therefore not explained in detail, see paper 1 in the licentiate thesis by Ferrari (2002) and Maclion et al. (2003).

Experimental work was performed on convective heat transfer from single and multiple quadratic prisms. This specific geometry was chosen because the flow pattern around them is less sensitive to Re levels, compared for example to that around a circular cylinder, as for example discussed in Zdravkovic (1997). Such data may also serve as a test case in comparisons with numerical simulations.

Heat transfer coefficients around the prisms were measured, using a method in which a solid prism of stainless steel was electrically heated at the core, and surface temperatures were measured using thermochromic liquid crystals (TLC). Based on these data the temperature field inside the prism, and the surface heat flux, were numerically computed, and expressed as local Nu values. Flow velocities around the prisms were also measured using particle image velocimetry (PIV), Wiberg (2000). It was found that this method incurred errors in the measured surface temperatures, which were magnified to larger errors in Nu. A small temperature difference between two closely spaced positions on the surface, is by the heat conduction in the solid connected to large differences in heat flux between these positions. The errors in the temperatures were typically ± 5 % of the TLC effective range, while the errors in Nu were typically ± 50 %, somewhat reduced after smoothing of the data.

Additional measurements were performed on the quadratic prisms, using the TLC and foil technique and using PIV velocity measurements. Specifically, relations between Nu and different turbulence parameters above the surfaces of separated regions were investigated, in the M.S. thesis by Stroppiana (2001). A close correlation was found between the surface averages of Nu and a turbulence intensity parameter for all geometries studied: a single prism, two prisms in tandem at varying distances, and a prism with a downstream splitter.
1.2. Some methods for the measurement of $h$

The above-mentioned method for measurement of $h$, using a body of steel, produced large errors, and a better method was therefore needed. For one class of methods, surface temperatures are measured on a thermally insulated body, and the fluid (or the surface) temperature is subjected to a step or periodic change. The measured transient response in the temperature is introduced into a semi-infinite and one-dimensional solution of the transient heat conduction equation, from which $h$ can be calculated. Camci et al. (1993) Yan & Owen (2002) Ireland et al. (1999) Von Wiffersdorf et al. (1993) Baughn et al. (1998).

The transient temperature change may be introduced by switching valves, insertion of a preheated object, removal of a shield blocking and a upstream thin-wire mesh heater. Advantages of such methods may be the possibility to use them on complex shaped surfaces (with the one-dimensional heat conduction assumption still valid). Disadvantages may be the less controlled surface temperature condition and difficulties to achieve the necessary change in the fluid temperature.

In another method, $h$ is measured in a thermal steady state, while a uniform surface heat flux is applied on a thermally insulated test body. The local $h$ is then directly related to the local surface temperature, which may be measured using TLC, Baughn et al. (1985) Hippensteele et al. (1985) Gao & Sundén (2001) Prašner et al. (2001). The constant heat flux may be delivered from an electrically heated foil, which is thin enough to make the heat conduction parallel in the foil small in comparison to the heat flux to the fluid. Different foils have been tested, gold foils on a plastic sheet, carbon impregnated papers, stainless steel. The heat flux may also be applied using radiation from a lamp, Critoph et al. (1999).

Advantages of the constant heat flux methods are: 1) large surfaces temperature differences arise, which can thus be measured with a good accuracy, 2) the thermal steady state, which simplifies the measurements, 3) the well defined constant heat flux surface condition, 4) the straightforward calculation of $h$, from its definition. Disadvantages may be: 1) difficulties to produce a real constant sufficiently uniform heat flux over the entire surface 2) the surfaces cannot have complex shapes when using foils.

For the measurements in this thesis, the method using a thin electrically heated foil with TLC was chosen, which for the investigations shown in this thesis gave errors in $h$ typically within 7-8%. For this purpose a new foil
was designed and manufactured \(^2\), which consisted of many closely-spaced thin Inconel strips embedded in a thin plastic sheet.

Inconel has a relatively low thermal conductivity, and its electrical resistance has a weak dependence on the temperature. The foil is described in more detail in papers 2 and 3 in this thesis. Other foils may or may be able to produce uniform heat flux, in part caused by the longitudinal Hall effect, which in some cases can make the heat flux 30% higher at the center of the foil than at the edges, Tarasuk (1983). The uniformity in the delivered heat flux over the surfaces for the foil used here was typically within 1-2% from its average.

This foil can also be used on surfaces with curved boundaries, such as a cylinder end surface, which is a part of an investigation of a two-diameter long cylinder in axial flows, paper 2 in this thesis.

1.3. A parameter which may affect \( h \)

\( h \) may depend on the upstream surface temperatures. High upstream surface temperatures heat the fluid, which makes \( h \) lower downstream, and low upstream surface temperatures heat the fluid less, which gives a higher \( h \) downstream. As an example, the laminar flow over a flat plate having a constant heat flux surface (producing large temperature differences) gives a local \( h \), which is 36% higher compared to that of a constant temperature surface, Incropera & De Witt (1990) ch. 7. On the other hand, for a turbulent flow over the flat plate, the difference is only 4%.

In another test, a square prism was used in cross flow for \( Re < 5.6 \times 10^4 \), no differences were detected in the average \( Nu \) (and \( h \)) between the case of a constant temperature surface and that of a constant heat flux surface, Igarashi (1985). Here most of the surface was located in separated regions, where the turbulence is high. Therefore, in turbulent flows, the turbulence tends to be more important for \( h \) than the upstream temperature history.

Based on this information, and on the fact that most flows in gas quenching are at \( Re > 10^6 \) and are turbulent, it can be expected that \( h \) for such flows are similar within a few percent, i.e. independent of the surface temperature distribution, whether for a constant heat flux surface (with large temperature differences), for an isotherm surface, or for a cooled steel body. Larger differences are expected locally on smooth surfaces, which are facing a low turbulence flow, producing a laminar boundary layer on the surface.

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CHAPTER 2

Summary of papers

2.1. Paper 1

This paper investigates experimentally and assesses the errors that may be incurred in the hue-based thermochromic liquid crystal TLC method, and their causes. The errors include response time, hysteresis, aging, surrounding illumination disturbance, direct illumination and viewing angle, illumination light intensity, TLC thickness, digital resolution of the image conversion system, and measurement noise. Some of the main conclusions are that (1) the 3×8 bits digital representation of the red, green and blue TLC color values produces a temperature measurement error of typically 1% of the TLC effective temperature range, (2) an 8-fold variation of the light intensity into the camera produced variations, which were not discernable from the digital resolution error, (3) this temperature depends on the TLC film thickness, (4) thicker films are less susceptible to aging and thickness nonuniformities. Parts of this paper were published Wiberg & Lior (2003b).

2.2. Paper 2

Local convection heat transfer coefficients were measured on a two-diameter long cylinder in axial flows of air, at Reynolds numbers of 8.9×10⁴ < Re < 6.17×10⁶ (9 to 63 m s⁻¹), using thermochromic liquid crystals (TLC) and an electrically heated foil consisting of thin metal bands. The flow in front of the cylinder was modified by the use of a turbulence generating grid, and by inserts, circular discs of two sizes, in front of the cylinder. These create a major change in the local convection heat transfer coefficient distribution on the cylinder. Increase of the turbulence intensity from Tu < 0.1% to 6.7% at the same Re, increased the average calculated Nusselt number Nu over the cylinder by 25%, and decreased the Nu non-uniformity (σₙᵢᵢ and σₘₐₓ) by 30% and 12% respectively. One of the flow modification inserts reduced σₙᵢᵢ by 25%. The position of flow reattachment was measured using tufts, and the envelope of the flow separation region was successfully visualized by a specially TLC-based heated flat plate mounted in the flow above the cylinder. Correlation’s between the Nu and Re in the form Nu = CRe⁶ were established and presented for the average Nu on the three different cylinder surfaces, and the variation of
the local exponent $e$ was shown along the cylinder. Parts of this paper were published Wiberg & Lior (2003a).

2.3. Paper 3

Experimental investigations on the local and average heat transfer from a square prism, with an edge facing the flow, and from two prisms arranged in-line at different distances, and from four prisms arranged in a square, were carried out in cross flows of air at Reynolds numbers $2.9 \times 10^4 < Re < 1.39 \times 10^5$. Thermo-chromic liquid crystals (TLC) and thin foils, were used for these measurements and particle image velocimetry (PIV) was used for flow velocity measurements. A splitter plate behind the single prism killed the vortex shedding and halved the average Nusselt number $Nu$ on the downstream side. For the two prisms arranged in-line, $Nu$ showed three main different distributions appearing in three different intervals of distance, and the flow was found to be unstable and intermittently switching, in the interval 5.83 to 6.67 side lengths between the prisms. The largest values in $Nu$ appeared for distances 1.66 to 2.00 side lengths apart. The experimental work on the convective heat transfer was carried out by Roland Wiberg and Alberto Stroppiana in equal parts, while the PIV measurements were carried out by Alberto Stroppiana and in part evaluated by the author.
CHAPTER 3

Conclusions

This thesis is a part of a project for improving gas quenching, and describes several advances in the state of the art of measurement methods, knowledge of convective heat transfer from single and multiple bodies in cross- and axial-turbulent flows, and the fluid dynamics that dominates this heat transfer. The primary new findings relative to the state of the art are:

Thermochronic liquid crystals (TLC) and thin foil techniques, were used together with digital image computer processing for the measurements of the convective heat transfer. Different parameters, which affects the TLC measurements were found in the literature, and other parameters were found and quantified by experiments, especially the aging of the TLC and its relation to the TLC film thickness.

For the measurements of the convective heat transfer a thin heating foil was designed, which was shown to produce a constant heat flux over the surface investigated within 1-2%. The TLC measured temperatures had errors typically within < ± 5%, of its effective temperature range. The errors in the measured $h$ and $Nu$ were then estimated within ± 7-8 %.

$Nu$ was measured on a two-diameter long cylinder in axial flows of air, at Reynolds numbers of $8.9 \times 10^4 < Re < 6.17 \times 10^5$ (9 to 63 m s$^{-1}$). In quenching the uniformity in $h$ (and $Nu$) is of importance and some attempts to modify the distribution were made by the use of a turbulence generating grid, and by inserts, circular discs of two sizes, in front of the cylinder. In this comparison, the grid generated turbulence gave the smallest non-uniformity and an increased $Nu$.

In purpose to visualize the separated flow region above the cylinder, the TLC and the foil techniques were used on a thin plate inserted parallel with the flow. A maximum convective heat transfer track appeared on the plate, which was found to agree well with the dividing stream line, measured for a similar axial flow.

$Nu$ was measured on a single square prism, with and without a downstream splitter plate eliminating the vortex shedding, and for two prisms arranged in-line at different distances apart, and for four prisms arranged in a square pattern, for $2.9 \times 10^4 < Re < 1.39 \times 10^5$ at low upstream $Tu$. Except for the single square prism in cross flow, such data are not found in the literature,
while data exists for circular cylinders in similar configurations. In addition, PIV measured velocities were shown and were qualitatively related to $Nu$ for these flows.

$Nu$ measured for a specific $Re$ on a bluff body with a constant heat flux surface, as was used here, and in highly turbulent flows, is expected to be valid with a fair accuracy also for a body, in the gas quenching process, which is of the same shape and is located in the same type of upstream flow (turbulence levels, etc), and independent of $(T - T_\infty)$, type of gas used, gas pressure, and the temperature distribution on the body.
CHAPTER 4

Acknowledgments

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References


FERRARI, J. 2002 Studies of flow and heat transfer in gas quenching; quench chambers, impingement cooled bodies, and inverse solutions. Licentiate Thesis ISRN KTH/MEK/TR-02/02-SE. Department of mechanics/ FaxénLaboratorium, KTH.


Wiberg, R. 1999 Experimental study of the heat transfer from a cylinder in a cros

Wiberg, R. 2000 Heat flux and velocity measurements on and around a single quadratic cylinder and using groups of quadratic cylinders in the cross flow of air. Not published https://www2.mech.kth.se/liorweb/roland/. Dept. of mechanics/FaxónLaboratorium, KTH.


