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Experimental investigation of near-wall effects on hot-wire measurements

F. Durst, E. S. Zanoun

Abstract Hot-wire anemometry is a well-established measuring technique in modern fluid mechanics and is widely used to study laminar and turbulent flows. However, unsolved problems still exist when measurements are carried out close to heat-conducting or heat-insulating walls. Additional heat losses occur because of the presence of the wall that are usually not accounted for in the calibration of the wire. Because of this, erroneous fluid velocity measurements result with hot wires if the presence of the wall is not taken into account. The present paper investigates the effect that the wall material has on these additional heat losses from hot wires for walls of different heat conductivities. Similarity analysis of various aspects of the problem, verified by experimental and numerical results, is presented for wall materials of different heat conductivities, and the results are compared with available data in the literature. The data confirm the expected increase in heat losses with increasing wall heat conductivity. For heat-insulating materials the authors' results show that a wall-thickness influence exists. Additional data are provided to show that the heat loss from hot wires increases with increasing wire overheating, and the influence of the wire diameter is also clarified.

List of symbols

a	overheat ratio
c_p	specific heat at constant pressure
d	cylinder/wire diameter
Ec	Eckert number
Gr	Grashof number
g	gravitational acceleration
h	heat transfer coefficient/wall thickness
l	cylinder/wire length
Ma	Mach number
Nu	Nusselt number
Pr	Prandtl number
$\dot{q}$	rate of convective heat transfer
R, r	cylinder radius
r	cylindrical coordinate
Re	Reynolds number
T	temperature
ΔT	temperature difference ($T_w - T_\infty$)
t	time
U	streamwise mean velocity
U_i	Cartesian velocity components
u_τ	wall friction velocity
x_i	Cartesian coordinates
y	normal wall distance

Subscripts

c	for characteristic quantities
m	arithmetic mean $(T_w + T_\infty)/2$
M	wall material
w	at the surface of the cylinder/wire/wall surface
∞	at infinity, free stream

Superscripts

*	nondimensional quantity
+	in wall coordinates

Greek letters

α	thermal diffusivity
λ	thermal conductivity
κ	specific heat ratio
β	thermal expansion coefficient
ρ	fluid density
μ	dynamic viscosity
ν	kinematic viscosity
ϕ	viscous dissipation function
τ	mean wall shear stress
φ	cylindrical coordinate

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1 Introduction

Measurements of heat transfer from long cylinders, located in free flows of different fluids, have been extensively carried out by various researchers, e.g. see Collis (1956), Wood (1968), Piercy et al. (1956), and Wills (1962), since the very early work of King (1914). King presented his results in the form of heat transfer coefficients just one year before Nusselt's suggestions in 1915 to have a more general presentation of heat transfer by employing a dimensionless number, nowadays referred to as the Nusselt number. Subsequent to Nusselt (1915), investigators of heat transfer from cylinders in free flows presented the results of their measurements in *Nu–Re* diagrams, e.g. see Collis and Williams (1959), Wills (1962). From this work, generally applicable information arose (Fig. 1), showing the data of Collis and Williams (1959) in the form

$$Nu_{\mathrm{m}}\left(\frac{T_{\mathrm{m}}}{T_{\infty}}\right)^{-0.17} = f(Re).$$

Figure 1 also contains the corresponding results of the numerical predictions (J.M. Shi et al., personal communication, 2001), which are in close agreement with the experimental data of Collis and Williams (1959). Analytical results are also included in Fig. 1 that agree well in the low Reynolds number range, e.g. see Hieber and Gebhart (1968), but start to deviate for $Re \geq 0.25$. In contrast, the agreement of the experimental data with the analytical results of Cole and Roshko (1954) and Nakai and Okazaki (1975) is less satisfactory, as Fig. 1 shows. Nevertheless, as far as the overall heat transfer from a cylinder in free flows is concerned, there is consistent information available for low Reynolds numbers in free flows that can readily be applied to hot-wire anemometers. To extend the information for higher Reynolds numbers, computational tools are readily available, e.g. see Lange et al. (1998), to yield reliable heat transfer results for heated cylinders over the entire Reynolds number range of interest to flow measurements with hot-wire anemometry.

The existing data on heat transfer from hot wires are less consistent for measurements in the proximity of the wall. This is indicated in Sects. 2 and 3, where summaries are given for the existing data on hot-wire measurements close to walls. It is shown that inherent wall effects are present, rendering the free flow calibration, i.e. the heat transfer information of Fig. 1, invalid. Hence calibration corrections that account for the additional heat loss to the wall are needed. This has been stressed in a number of publications, and a summary of the literature has been provided by Bhatia et al. (1982). A general description of the problem is provided by Bruun (1995), and more recent publications are discussed by Lange et al. (1998, 1999). In Sect. 2 we summarize the information that we obtained about heat transfer from wires close to metal walls, and some references to the existing literature are given. Section 3 provides similar data for heat losses from wires close to walls of low thermal conductivity and it is shown that a different behaviour exists. Our own results are extended by corresponding data from the literature.

2
Similarity considerations and experimental verification

This short review of the existing knowledge about heat transfer from circular cylinders in free flows, as presented in Fig. 1, starts by describing the heat transfer per unit time and unit length as

$$\dot{q} = h(\pi d)(T_{\mathrm{w}} - T_{\infty}) = \int_{0}^{2\pi} -\lambda_{\mathrm{w}}\left(\frac{\partial T}{\partial r}\right)_{r=R} R \cdot \mathrm{d}\varphi. \qquad (1)$$

Normalizing Eq. (1) in the following way:

$$T = (T_{\mathrm{w}} - T_{\infty})T^{*}, \quad r = Rr^{*}, \quad \varphi = 2\pi\varphi^{*}\,, \qquad (2)$$

yields the following suggestion for a Nusselt number presentation of heat transfer data from circular cylinders in free flows:

$$Nu_{\mathrm{w}} = \frac{hd}{\lambda_{\mathrm{w}}} = -\int_{0}^{1}\left(\frac{\partial T^{*}}{\partial r^{*}}\right)_{r^{*}=1} \mathrm{d}\varphi^{*}. \qquad (3)$$

Introducing the following characteristic quantities:

$$\rho = \rho_{\mathrm{c}}\rho^{*}, \quad U_i = U_{\mathrm{c}}U_i^{*}, \quad x_i = l_{\mathrm{c}}x_i^{*}, \quad \mu = \mu_{\mathrm{c}}\mu^{*}, \quad \lambda = \lambda_{\mathrm{c}}\lambda^{*},$$

yields the corresponding dimensionless forms of the continuity, Navier–Stokes and energy equations

$$\frac{\partial}{\partial x_i^{*}}\left(\rho^{*}U_i^{*}\right) = 0, \qquad (4)$$

$$U_i^{*}\frac{\partial U_j^{*}}{\partial x_i^{*}} = \left[\frac{1}{Re}\right]\frac{\partial}{\partial x_i^{*}}\left[\mu^{*}\left(\frac{\partial U_j^{*}}{\partial x_i^{*}} + \frac{\partial U_i^{*}}{\partial x_j^{*}}\right)\right] + \left[\frac{Gr}{Re^{2}}\right]T^{*}, \qquad (5)$$

$$U_i^{*}\frac{\partial T^{*}}{\partial x_i^{*}} = \left[\frac{1}{RePr}\right]\lambda^{*}\frac{\partial^{2}T^{*}}{\partial x_i^{*2}} + \left[\frac{Ec}{Re}\right]\phi^{*}. \qquad (6)$$

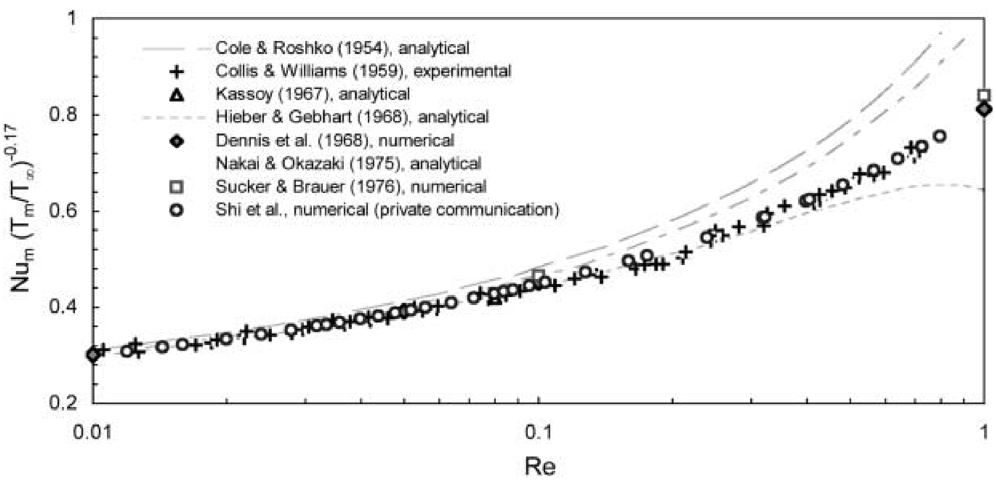

Fig. 1. Comparison of free stream heat transfer from a circularcylinder among analytical, experimental and numerical results

This indicates that the heat transfer from a circular cylinder should show the following general dependence:

$$Nu_{\mathrm{w}} = f\left(Re, Pr, Gr, Ec, \frac{\Delta T}{T_\infty}, \frac{l}{d}\right), \tag{7}$$

on the dimensionless numbers

$$Re = \frac{Ud}{\nu},\ Pr = \frac{c_{\mathrm{p}}\mu}{\lambda},\ Gr = \frac{g\rho^2 d^3 \beta \Delta T}{\mu},\ Ec = \frac{U^2}{c_{\mathrm{p}}\Delta T}. \tag{8}$$

The overheat ratio $\Delta T/T_\infty$ takes the dependence of the heat thermal conductivity $\lambda(T)$ into account and, via the length-to-diameter ratio l/d, the geometry of the cylinder. According to Andrews et al. (1972) and Chew et al. (1998), no length-to-diameter ratio dependence exists for $l/d \geq 200$. The influence of the Eckert number Ec can also be neglected for flows of small Mach numbers Ma, since $Ec = Ma^2(\kappa - 1)(T_\infty/\Delta T)$, e.g. see Hinze (1975). In addition, the influence of the Grashof number Gr is negligible if $Gr < Re^3$ holds, e.g. see Collis and Williams (1959). All these conditions are usually satisfied for hot-wire anemometry, and therefore, for the heat transfer from hot-wires, Eq. (7) reads

$$Nu_{\mathrm{w}} = f\left(Re, Pr, \frac{\Delta T}{T_\infty}\right). \tag{9}$$

It is common practice when hot-wires are used for flow measurements to calibrate them in the same fluid in which they are later applied. Hence, the above given Pr-dependence in Eq. (9) can be eliminated to yield

$$Nu_{\mathrm{w}} = f\left(Re, \frac{\Delta T}{T_\infty}\right), \text{or when mean properties are employed} \Rightarrow$$

$$Nu_{\mathrm{m}} = f(Re)\left(\frac{T_{\mathrm{m}}}{T_\infty}\right)^{0.17}. \tag{10}$$

However, to transfer the authors' data to oil and water flows requires the inclusion of the Prandtl number. This was not part of the present work and therefore the authors refer readers to Alfredsson et al. (1988). The relationship in Eq. (10) was found experimentally by Collis and Williams (1959) and confirmed numerically by Lange et al. (1998). Figure 1 summarizes the existing results presented, as suggested by Collis and Williams (1959), in the form

$$Nu_{\mathrm{m}}\left(\frac{T_{\mathrm{m}}}{T_\infty}\right)^{-0.17} = f(Re). \tag{11}$$

In addition, when a wire is placed in the wall region of a velocity field with a velocity gradient, the general relation, Eq. (10), needs to be extended to take into account the velocity gradient across the hot wire, yielding a velocity difference between the top and bottom of the wire. This might seem small in absolute terms, but it is rather large when seen in relative terms, that is, normalized with viscous length and velocity scales to give

$$Nu_{\mathrm{m}} = f\left[Re, \frac{\Delta T}{T_\infty}, \frac{d}{U}\left(\frac{\mathrm{d}U}{\mathrm{d}y}\right)\right]. \tag{12}$$

In dimensionless form, the dependence on the velocity gradient close to a wall can be rewritten as

$$\frac{d}{U}\left(\frac{\mathrm{d}U}{\mathrm{d}y}\right) = \frac{d}{U}\frac{\tau_{\mathrm{w}}}{\mu}\frac{\rho}{\rho} = \frac{d^+}{U^+}, \tag{13}$$

and the local Reynolds number, computed with the velocity at the center of the wire, reads as

$$Re = \frac{Ud}{\nu} = U^+ d^+ = y^+ d^+, \tag{14}$$

where $U^+ = U/u_\tau$, $d^+ = du_\tau/\nu$ and $y^+ = yu_\tau/\nu$. As a result, the following general relationship holds for the heat transfer from hot wires close to walls:

$$Nu_{\mathrm{m}} = f\left(y^+ d^+, \frac{d^+}{U^+}, \frac{\Delta T}{T_\infty}\right) = f\left(y^+, d^+, \frac{\Delta T}{T_\infty}\right). \tag{15}$$

The resulting data are presented in Fig. 2.

Therefore, to verify that the influences of the normalized parameters expressed in Eq. (15) exist, we performed experimental investigations using the experimental setup described by Durst et al. (2001). The results are presented in Figs. 2 and 3 for hot wires close to heat-conducting and insulating walls.

Figure 2 represents the heat transfer data from hot wires in form of the Nusselt number versus the Reynolds number, defined here as $Re = d^+ y^+$. This figure clearly shows that the heat loss from hot wires is affected by the presence of the wall. Approaching the wall causes this effect to increase with decreasing wall distance and increasing overheat ratio $\Delta T/T_\infty$. The dependence on overheat ratio implicitly takes the dependence of the heat thermal conductivity $\lambda(T)$ into account.

Equivalent to the Nusselt number presentation of heat transfer in Fig. 2, results can also be represented in the form of U^+, showing the dependence of heat loss from hot wires on the normalized wall distance, wire diameter and overheat ratio, as given below (Fig. 3). This is equivalent to employing the data presented in Fig. 2 of heat transfer in both free flows and wall region to calibrate the hot wires

$$U^+ = f\left(y^+ d^+, \frac{d^+}{U^+}, \frac{\Delta T}{T_\infty}\right) = f\left(y^+, d^+, \frac{\Delta T}{T_\infty}\right). \tag{16}$$

Figure 3 shows the dependence equivalent to Fig. 2.

The heat loss from hot wires close to heat-conducting walls has been the subject of a number of experimental, numerical and analytical investigations. Most of these studies agree well with the fact that for $y^+ < 5$ the hot wire cannot be used without the need for corrections, see e.g. Collis and Williams (1959), Oka and Kostic (1971), Hebbar (1978), Bahtia et al. (1982), Krishnamoorthy (1985) and Lange et al. (1999).

Figure 3 shows that the authors' results confirm the dependence of the hot-wire readings on the normalized wall distance y^+ and the wire diameter under a constant overheat ratio. The results show typical deviations from the linear velocity distribution $U^+ = y^+$ for wall distances of $y^+ < 3.5$ as a result of the increased heat transfer from the presence of the metal wall (aluminum), i.e. a wall of highly heat-conducting material. Higher apparent velocities with

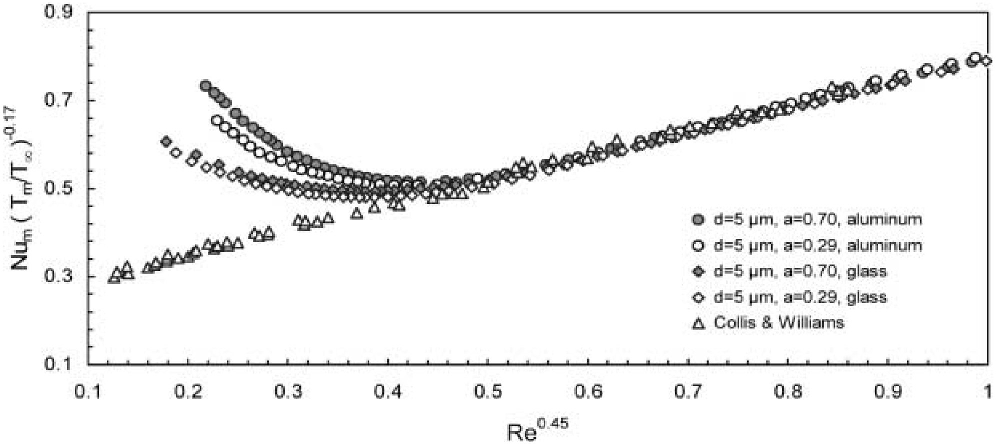

Fig. 2. Heat transfer results from hot wires expressed by the Nusselt number as a function of the Reynolds number, compared with experimental data (Collis and Williams 1959)

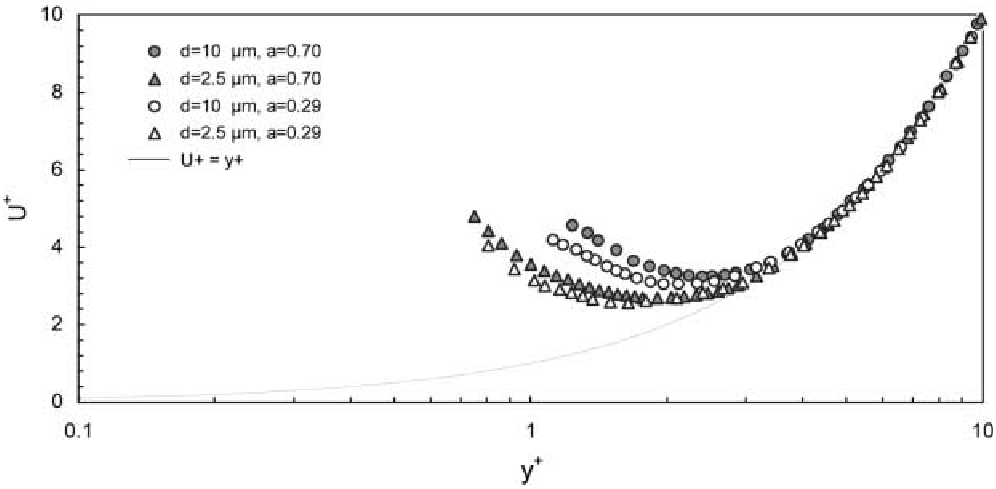

Fig. 3. Normalized velocity from hot-wire readings as a function of normalized wall distance, wire diameter and overheat ratio in proximity of aluminum wall

larger wire diameters were observed. The heat transfer area of the wire subjected to the wall increases with increasing wire diameter, resulting in a higher heat loss to the wall. As a consequence, this reveals a dependence on the wire diameter, and the correction needed for hot-wire readings increases, therefore, with increasing wire diameter for a constant overheat ratio. Hence the effect of the wire diameter cannot be neglected, since it exerts a significant influence on the velocity and the temperature fields near the wall. This was quantified with more detail in Durst et al. (2001) and was in good agreement with the corresponding information from the literature (Oka and Kostic 1971; Hebbar 1978). This dependence was also quantified by Collis and Williams (1959) and Chew and Shengxi (1993). On the other hand, Bhatia et al. (1982) neglected wire diameter effects, claiming in their analysis that a wire of very small diameter has no influence on the flow field, i.e. its velocity wake is negligible, and therefore its influence diminishes.

In addition to the effect of the wire diameter, a dependence on the overheat ratio close to the wall was observed (Fig. 3), as predicted from Eq. (16). The figure also indicates, as expected, that the additional heat loss from the wire, due to the presence of the wall increases with increasing overheat ratio $\Delta T/T_\infty$. However, this dependence gradually decreases with decreasing wire diameter, as Fig. 3 shows; see Durst et al. (2001) and Zanoun et al. (2000). Chew and Shengxi (1993), Krishnamoorthy et al. (1985) and Zemskaya et al. (1979) also reported the effect of overheat ratio and wire diameter on the heat loss from hot wires in the vicinity of the wall. However, Chew et al. (1998) concluded that no apparent influence was observed when the overheat ratio changed.

To demonstrate that our measurements are in close agreement with other experimental data for the additional heat losses from hot wires, Fig. 4 shows existing data for hot-wire readings close to highly heat-conducting walls compared with the present findings. The comparisons in Fig. 4 for data with 5-μm diameter wires and for almost the same overheat ratio show satisfactory agreement among all the experimental results. The differences in the results shown in the figure can be attributed to differences in the wall thermal conductivities ($\lambda_{\mathrm{aluminum}}$=204–237 W/m K and λ_{steel}=15–55 W/m K, Eckert et al. 1987), in the quality of experimental test facilities, in the accuracy of hot-wire calibration in the free stream, in the wire geometry and in the wall distance determination.

To further investigate the physical cause of the additional heat losses from hot wires close to metal walls, we carried out experiments using hot wires with and without flows. The results are shown in Fig. 5 for wires of 5 and 10 μm and for a common overheat ratio of 0.70. The data clearly indicate the dominance of heat conductivity close to metal walls. The influence of convection on the heat

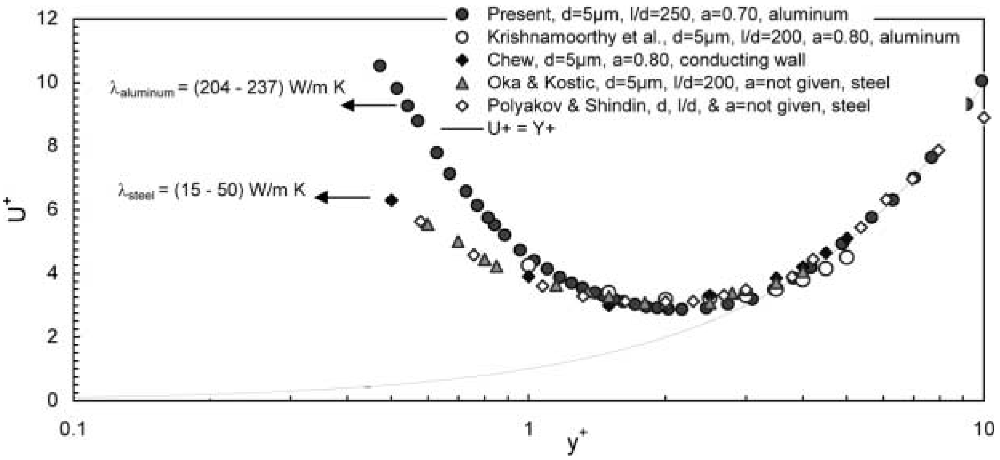

Fig. 4. Comparison of the present results of the effect of heat-conducting materials on the measured normalized velocity distribution with other studies

transfer from a 5-μm hot wire is negligible for wall distances of $y \leq 100$ μm. This is readily understood by looking at the normalized energy equation for steady flow conditions

$$\underbrace{U_i^* \frac{\partial T^*}{\partial x_i^*}}_{I} = \underbrace{\left[\frac{1}{RePr}\right] \lambda^* \frac{\partial^2 T^*}{\partial x_i^{*2}}}_{II} + \underbrace{\left[\frac{Ec}{Re}\right] \phi^*}_{III} . \tag{17}$$

In the wall region where small velocities exist, heat loss by free convection may be neglected if $Gr<Re^3$ (Collis and Williams 1959). In the present study, in common with Collis and Williams (1959) using hot wires in the wall region, the Grashof number was on the order of 10^{-7} and the Reynolds number on the order of 10^{-2}; therefore the condition under which free convection becomes insignificant is satisfied. Thus, most of the heat went into the metal wall because of heat diffusion, term (II) in Eq. (17) for constant Pr. Hence, diffusivity is dominant in the wall region, and consequently, the main role is played by heat diffusion rather than by forced and free convection.

In the absence of flow, the data in Fig. 5 show the influence of heat conduction and free convection over a horizontal flat plate standing under the hot wire. In addition, Fig. 5 also shows the heat loss by forced convection under flow conditions. The experiments without flow clearly indicate the importance of heat conductivity in the wall region. Moreover, for the current experimental work with a flow of 6-m/s free stream velocity, the dominance of heat conductivity in the wall region remains. This is clearly shown by the fact that the two sets of data, with and without flow, in Fig. 5 are close to each other for the last 100 μm of the 3-mm boundary layer thickness. These findings correct the claims made by Collis and Williams (1959) that the increasing buoyancy effect is the cause of the additional heat losses from hot wires at small values of the Reynolds numbers. To support these findings further, our results for the 10-μm diameter wire and an overheat ratio of 0.70 are also shown in Fig. 5, indicating a noticeable influence of flow. This is explained by

$$\frac{\text{Diffusivity}}{\text{Convection}} = \frac{(1/RePr)}{(Gr/Re^2)} = \frac{Re}{GrPr} . \tag{18}$$

By introducing the definitions of the Grashof and Reynolds numbers into Eq. (18) it is clear that the ratio of diffusivity to convection varies as d^{-2}. Hence an increase in both the Grashof and Reynolds numbers from the wire diameter resulted in an increase in the convection effect in comparison to the corresponding diffusion effect. Therefore, for larger wire diameters and also because of the velocity gradient, there is still a small influence of convective heat transfer from the wire in the wall region as a consequence of increasing both the Grashof and Reynolds numbers.

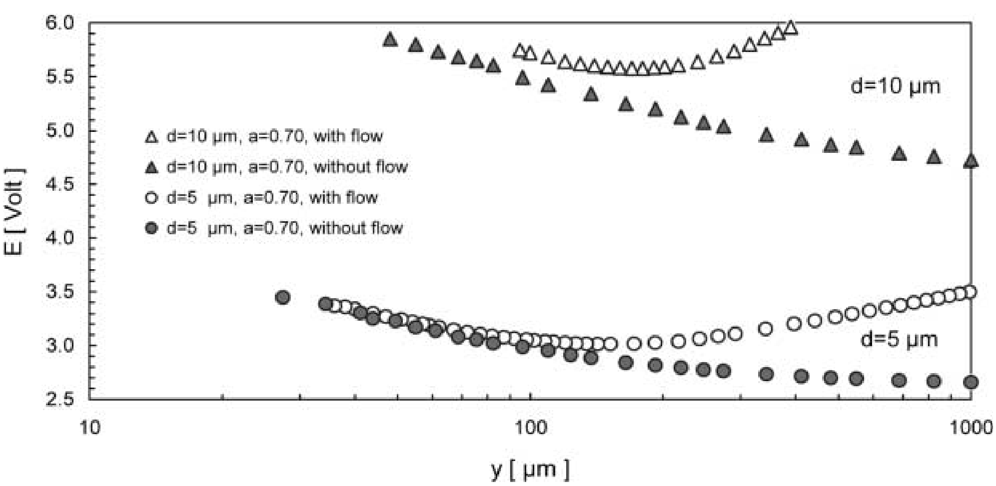

Fig. 5. Hot-wire output voltage with and without flow close to metal (aluminum) wall for 0.70 overheat ratio

3 Heat losses to walls of low heat conductivity

The summary of the existing knowledge provided in Sect. 2 clearly shows that there is common agreement on the heat transfer from hot wires close to metal walls. On the other hand, this agreement is less satisfactory for hot wires located close to walls of low heat conductivity, such as ceramics, glass, Perspex and Textolite. For such wall materials a number of experimental investigations have been carried out and are described in the literature. Regarding the results of hot-wire measurements in the proximity of such kinds of walls, the data provide a less consistent picture. This is indicated in Fig. 6, which provides information extracted from the literature on heat transfer measurements from hot wires close to heat-insulating walls.

Hot-wire measurements close to heat-insulating walls were initiated by Wills (1962). He concluded that the heat loss from a hot wire is related to the Reynolds number and to the wall distance normalized by the wire diameter and recommended corrections for mean velocity in turbulent flow to be half of the corresponding laminar flow correction, without physical interpretation for this partial correction. In contrast, it was found by Janke (1987) that velocity corrections needed in turbulent flow are the same as those in a laminar flow under the same wall shear stress. Thereafter, Polyakov and Shindin (1978) introduced a satisfactory correction for thermal and hydrodynamic effects of the wall on hot wires. Following Polyakov and Shindin (1978), Bhatia, et al. (1982) concluded that no corrections for nonconducting walls are needed since the heat losses are small and negligible. Extending the numerical work of Bhatia et al. (1982), Chew and Shengxi (1993) concluded that corrections are necessary for adiabatic walls and also for conducting walls for $y^+<5$. Further investigations by Ligrani and Bradshaw (1987) showed that the geometry of wires can affect turbulence measurements over a mirror glass wall. They obtained the smallest deviation starting at $y^+ \leq 1.5$ from the linear velocity distribution $U^+=y^+$. More studies were carried out by Khoo et al. (1996), who concluded that their hot-wire performance was independent of both the flow conditions and the wall material. Chew et al. (1998) continued the work of Chew and Shengxi (1993), deducing larger wall effects as the wall material becomes more thermally conducting. Recently, Lange et al. (1999) obtained a positive correction for hot-wire readings, i.e. negative wall effect, close to adiabatic walls. A summary of almost all previous studies near heat-insulating walls is depicted in Fig. 6, which shows that the available data are less consistent, but still provide the following general features:

1. Additional heat losses occur for all insulating materials very close to the wall, but a wide spread of data exists.
2. Some sets of data start to deviate at $y^+ \leq 1.5$ with lower heat transfer in the presence of walls, whereas others start at $y^+ \leq 5$ and show higher deviations.
3. There is no difference between the heat transfer data obtained in laminar and in turbulent flows near the wall.

The above unsatisfactory situation regarding the existing knowledge on the heat transfer from hot wires close to walls made of materials of lower thermal conductivity encouraged us to extend our work and also to carry out some measurements for glass and Plexiglas walls. For this purpose, a laminar boundary layer was set up in a small wind tunnel using flat plates of different wall materials and configurations. The aluminum plate employed so far in our experiments (Durst et al. 1995, 2001; Zanoun et al. 2000) was replaced with glass plates of 8- and 3-mm thicknesses. Hot-wire measurements were performed across the entire boundary layer, with particular attention being given to the measurements in the proximity of the wall, i.e. in the region where $y^+ \leq 10$. Parallel to the hot-wire measurements, laser-Doppler anemometry (LDA) measurements were also performed to describe the wall boundary layer measurements through laser-Doppler data (Fig. 7). For this measuring technique no wall influence on the mean velocity measurements exists (Durst et al. 2001; Zanoun et al. 2000), and the measured velocity profile was therefore well represented by the Blasius laminar profile. Hence the measured LDA data can also be given by the analytical equations for the investigated flat-plate boundary layer.

The LDA data in the region very close to the wall fitted the $U^+=y^+$ relationship as shown in Fig. 7. Figure 7 also contains our hot-wire results close to the wall for the aluminum and the glass plates for the two different over-

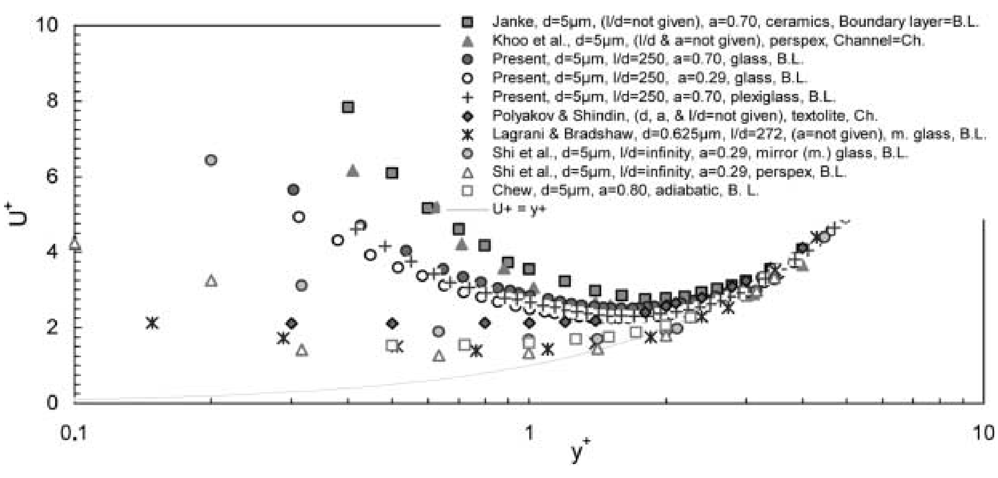

Fig. 6. Comparison of the present results with some experimental findings on the effect of heat-insulating materials on normalized velocity distribution

heat ratios and the 5-μm diameter wire employed. The data confirm a reduced wall effect for the glass plate. The data also show that the wall effect on the hot-wire heat transfer is negligible for normalized wall distances $y^{+}\geq 3.5$, regardless of the wall conductivity.

To clarify the role played by the wall thermal conductivity on hot-wire readings, it is necessary to consider the heat transfer from the hot wire to the wall material as follows:

$$\rho_{M} c_{PM} \frac{\partial T}{\partial t} = \lambda \frac{\partial^2 T}{\partial x_i^2}. \qquad (19)$$

Normalizing Eq. (19) yields the following characteristic time scale t_c for heat conduction through the wall material:

$$t_c = \frac{l_c^2}{\alpha_c}, \text{ where } \alpha_c = \frac{\lambda_c}{\rho_c c_{Pc}}. \qquad (20)$$

Hence for two different wall materials (e.g. aluminum and glass) with the same characteristic length scale l_c, the ratio of the heat conduction time scale for aluminum and glass is equal to the inverse ratio of the two thermal conductivities, i.e.

$$\frac{t_c^* \text{aluminum}}{t_c^* \text{glass}} = \frac{\alpha_c^* \text{glass}}{\alpha_c^* \text{aluminum}}. \qquad (21)$$

This clearly indicates that walls of heat-conducting materials absorb heat faster than heat-insulating materials. This explains why hot-wire readings over heat-conducting walls deviate more than over heat-insulating walls, thus confirming the findings presented in Fig. 7.

To further demonstrate the importance of heat conductivity in comparison with heat convection, we repeated our measurements close to the glass walls. One set of results is shown in Fig. 8. If Fig. 8 is compared with Fig. 5, it becomes quite clear that the heat convection in the proximity of heat-insulating materials cannot be neglected. Nevertheless, in the case of heat-insulating materials the local heat loss from the wire by conduction is much less compared with walls of highly heat-conducting materials for the same Reynolds and Prandtl numbers. Therefore, convective heat transfer plays an increased role in the proximity of heat-insulating wall materials in addition to heat conduction.

Apparent inconsistencies in the available data (Fig. 6) might also be explained by the effect of wall thickness, which the authors found in their investigations. Such a wall-thickness effect existed for glass plates, but was not found for metal walls. More detail is given below.

Consider two flat plates of the same thermal conductivity and two different wall thicknesses h_1 and h_2, where $h_2>h_1$. Following the argument relating to the thermal conductivity effect in Eq. (19), the time scale ratio of heat conduction through the wall is directly proportional to the ratio of the wall thickness squared

$$\frac{t_c^*(h_2)}{t_c^*(h_1)} = \left(\frac{h_2^*}{h_1^*}\right)^2. \qquad (22)$$

In addition to the above argument, from simple laws of heat transfer through wall materials the overall heat transfer coefficient is inversely proportional to the wall thickness. As a result, a thinner wall has a smaller thermal heat resistance and a faster heat loss than a thicker wall. For heat-insulating materials, in particular, heat accumulation within the thicker wall modifies the temperature field around the wire and therefore produces, in general, a smaller temperature gradient and consequently less heat transfer from the hot wire to the wall. Local heat transfer by convection with heat-insulating materials plays a significant role in the wall region, which makes rate of heat loss faster with the thinner wall, see Fig. 9.

4 Conclusions

To provide clearer information than is available in the literature regarding the additional heat loss from hot wires in the proximity of walls of different heat conductivities, we carried out investigations near aluminum, glass and Plexiglas walls. The results obtained were compared with data in the literature, yielding the information described below and provided in Fig. 10.

1. There is no wall material available that permits hot-wire measurements close to walls without corrections for the additional heat loss.

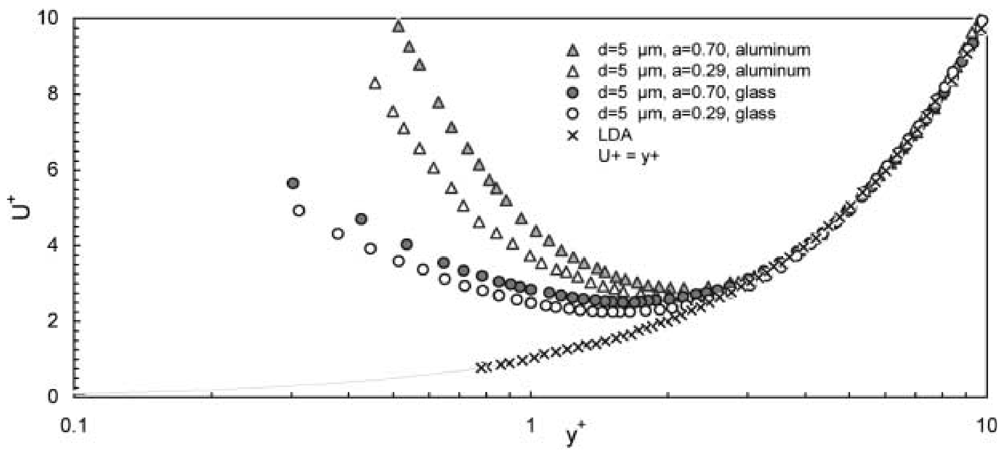

Fig. 7. Effect of wall thermal conductivity and overheat ratio on normalized velocity distribution $U^{+}=f(y^{+})$, measured by hot wire over 8-mm aluminum plate

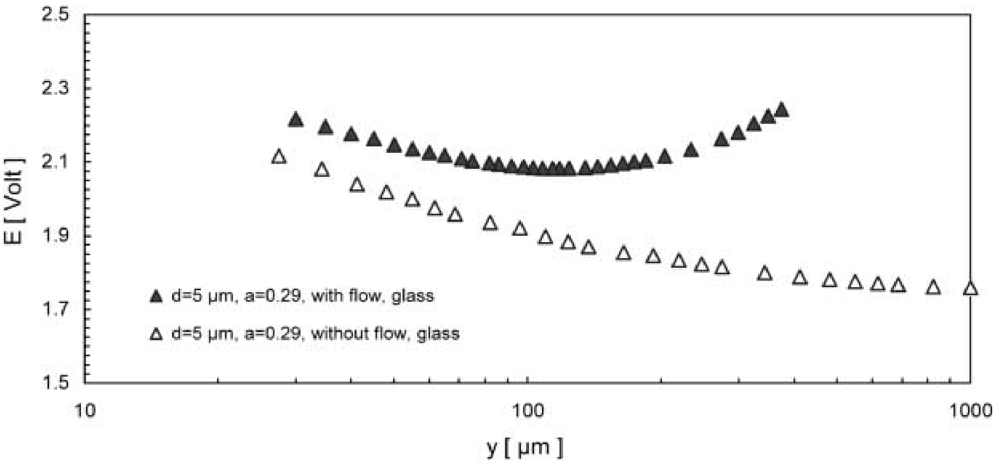

Fig. 8. Conductive and convective effect on hot-wire readings in proximity of the wall

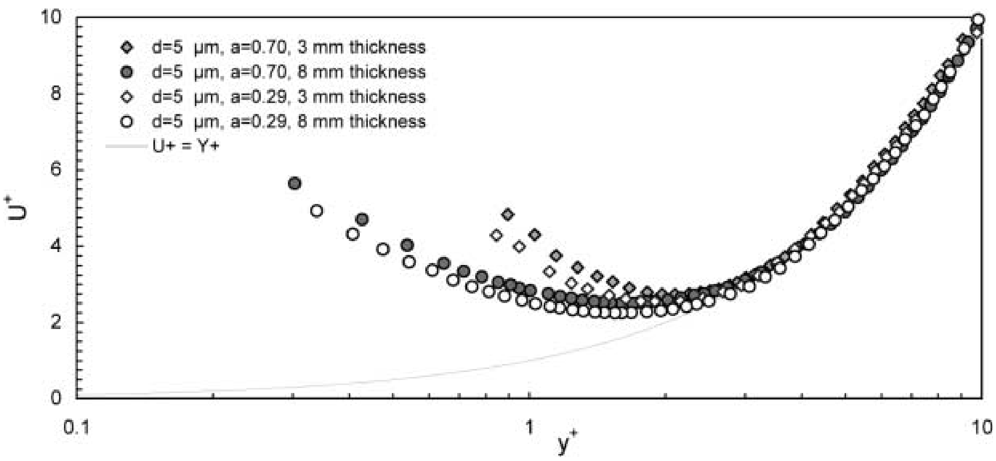

Fig. 9. Wall-thickness effect on heat transfer from hot wire in proximity of glass plate

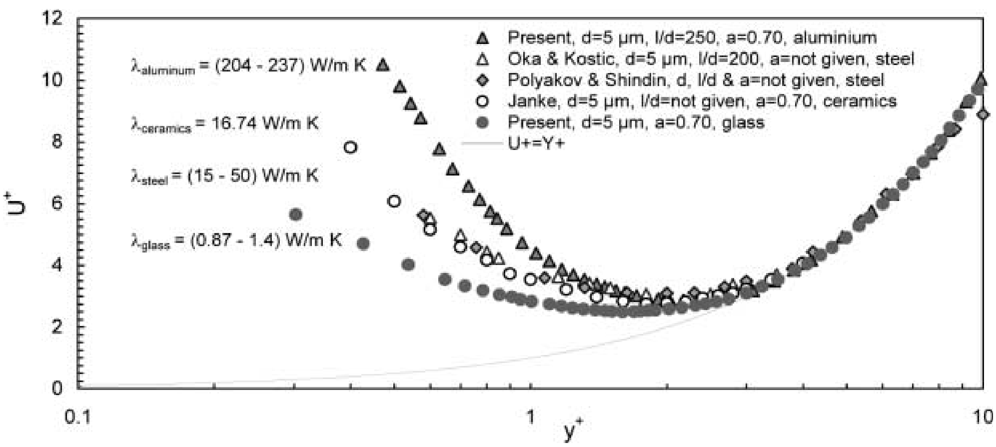

Fig. 10. Summary of wall-material effect on hot-wire readings in the vicinity of the wall

2. The major effect on the additional heat loss from hot wires close to walls is due to the wall itself, causing modifications of the temperature field responsible for heat conduction. The results in Fig. 10 show that the corrections for wall effects on hot-wire readings increase with increasing wall conductivity.
3. Corrections for hot-wire measurements in the near-wall region are needed for all values of $y^+ \leq 3.5$, and the corrections increase with increasing wire diameter.
4. Increasing the overheat ratios of the wires increases the additional heat losses from the wires, but the influence of the overheat ratio decreases with decreasing wire diameter.

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