Pergamon

Int. J. Heat Mass Transfer. Vol. 41, No. 12, pp. 1787–1794, 1998
© 1998 Elsevier Science Ltd. All rights reserved
Printed in Great Britain
0017–9310/98 $19.00+0.00

PII: S0017–9310(97)00293–7

The effective Reynolds number of a heated cylinder

F. DUMOUCHEL,† J. C. LECORDIER and P. PARANTHOËN
U.M.R. 6614 C.N.R.S., Université de Rouen, 76821 Mont Saint Aignan Cedex, France

(*Received* 8 *April* 1997 *and in final form* 30 *July* 1997)

Abstract—The structure of the near wake behind a heated cylinder in air is investigated experimentally in the regime corresponding to the transition from a 2-D steady to a 2-D periodic wake. In this non-isothermal case an effective Reynolds number, Re_{eff}, is used to characterize the flow regime. The corresponding effective temperature is found to be $T_{\infty}+0.24(T_{\text{w}}-T_{\infty})$ and is lower than the usual film temperature $T_{\text{film}} = T_{\infty}+0.5(T_{\text{w}}-T_{\infty})$. The structure of the velocity fields in the wake downstream the heated cylinder is found to be correctly characterized by this effective Reynolds number. Over this flow regime, Nusselt numbers are found to be dependent on a Reynolds number intermediate between Re_{eff} and Re_{film}. © 1998 Elsevier Science Ltd. All rights reserved.

1. INTRODUCTION

The structure of the near wake behind a circular cylinder, in the regime corresponding to the transition from a 2-D steady to a 2-D periodic wake ($30 < Re < 80$), has always received significant attention of researchers because of its practical importance as well as theoretical interest. Here the Reynolds number is defined by $Re = (U_{\infty}d/\nu_{\text{g}})$, where U_{∞} is the free stream velocity, d is the cylinder diameter and ν_{g} is the kinematic viscosity calculated at the free stream temperature. When this Reynolds number Re is larger than some critical value Re_{c} a time periodic oscillation is observed in the wake, leading to the development of a vortex street downstream of the circular cylinder. The interest in this flow has recently increased when this configuration has been widely used to test new development in the theory of hydrodynamic instability, Provansal *et al.* [1], Huerre and Monkewitz [2], Oertel [3].

Although many aspects of the velocity field have been studied in detail over this low Reynolds numbers range, corresponding investigations of the heat transfer aspects of this problem in the forced convection mode have not yet been achieved up to now. In spite of the importance of this heat transfer problem, detailed experimental results only exist for higher Reynolds numbers, Freymuth and Uberoi [4], LaRue and Libby [5], Matsamura and Antonia [6], Xenopoulos *et al.* [7]. At low Reynolds numbers experimental results are sparse and the information is especially limited to the average heat transport or the determination of the local value of the Nusselt number of the cylinder, Eckert and Soehngen [8], Vilimpoc *et al.* [9] or some characteristics of the temperature distribution within vortices, Mi and Antonia [10].

One of the main reasons of this situation could be linked to the experimental difficulties to carry out such experiments at low velocities in a wake of small dimensions which is very sensitive to small external perturbations. Furthermore, preliminary works carried out by the authors have shown that, over this Reynolds numbers range, the structure of the wake behind a heated cylinder was very sensitive to the heat input fed in the cylinder and that heat was never a passive contaminant, Lecordier *et al.* [11].

When the level of heating is not high enough to generate buoyancy effects, the heated cylinder operates in a forced convection regime. However temperature differences within the fluid lead to variations of its properties: viscosity, density, thermal conductivity. It results that due to the heat input to the cylinder, the characteristics of vortex shedding can be significantly altered and, in air, total suppression can be achieved by increasing the heat input sufficiently, Lecordier *et al.* [11]. Up to now, in this forced convection regime, this effect has been mainly related to the increase of the kinematic viscosity of the gas and to the corresponding decrease of the effective Reynolds number, Sreenivasan *et al.* [12], Lecordier *et al.* [11], Ezerski [13]. Other authors reported similar results but related this effect only to the density change in the near wake, Yu and Monkewitz [14], Schumm *et al.* [15]. However the opposite behaviour found in water experimentally by Paranthoën *et al.* [16] and numerically by Socolescu *et al.* [17] suggests that over this forced convection regime the control is due to both changes of the near wake dynamic viscosity and density with temperature.

When the level of heating is increased and is high enough to induce buoyancy effects mixed convection

† Author to whom correspondence should be addressed.

NOMENCLATURE

d	diameter of the cylinder [m]
g	gravitational acceleration [m s^{-2}]
L	length of the cylinder [m]
Nu	Nusselt number
P	electric power [W]
Re	Reynolds number
Re_c	critical Reynolds number
T	temperature [°C]
U	longitudinal velocity [m s^{-1}]
x	coordinate for horizontal axis [m]
y	coordinate for vertical axis [m].

Greek symbols

α	non-dimensional parameter
β	temperature coefficient for volume expansion [°C^{-1}]
ΔT	temperature excess [°C]
ε	reduced Reynolds number
ν_g	kinematic viscosity [m^2 s^{-1}].

Subscripts

crit	critical
eff	refers to conditions at effective temperature
film	refers to conditions at arithmetic mean film temperature, $T_{film} = (T_w + T_\infty)/2$
w	wall
∞	free stream.

or free convection regimes have to be considered. Under these circumstances the buoyancy effects which appear in the near wake become predominant in comparison with variable fluid properties effects. A large number of experimental or numerical studies have reported the influence of such buoyancy effects on the development of the vortex shedding phenomenon in the vertical configuration, Noto *et al.* [18], Chang and Sa [19], Vilimpoc *et al.* [9], Hatanaka and Kawahara [20], Nakabe *et al.* [21], Michaux-Leblond and Belorgey [22], Mori *et al.* [23]. In the vertical configuration when the Richardson number Ri increases, the vortex shedding is found at first to increase owing to the positive buoyancy force and then to break down.

The limits between the forced convection regime, the mixed convection regime and the free convection regime depend on the geometry of the flow and can be characterized by critical values of the Richardson number Ri. Ri is a number which describes the relative importance of free convection to forced convection. Here $Ri = [(g\beta(T_w - T_\infty)d^3)/Re^2\nu_g^2]$ where g is the acceleration due to the gravity, β is the temperature coefficient for volume expansion and $(T_w - T_\infty)$ is the temperature difference between the cylinder and the upstream flow. For small $|Ri|$ forced convection effects dominate while for large $|Ri|$ it is the free convection which is important.

Based on change of the effective Nusselt number with mixed convection to the Nusselt number with forced convection alone the limit value of the Richardson number is about $5 \cdot 10^{-1}$ in the cross flow situation, Morgan [24], Fand and Keswani [25]. For the breakdown of the Bénard–Karman vortex street when the buoyancy effect works in the opposite direction to the gravitational force, the critical Richardson number is found between 0.15 and 0.5, Chang and Sa [19], Nakabe *et al.* [21].

In the cross flow situation that we investigate here, the above limit of heating for the forced convection regime leads for the laminar vortex street range ($Re \sim 50$) to consider temperature differences between the cylinder and the flow such that: $(T_w - T_\infty) < (12.5\nu_g^2/\beta d^3)$.

It results that conditions where buoyancy contamination can be neglected strongly depend on the nature of the fluid. For an example, with $d = 4 \cdot 10^{-3}$ m the values of $(T_w - T_\infty)_{limit}$ are 12 K and 1 K in air and in water, respectively. In order to have significant change of thermal properties of the fluid in the wake, without buoyancy effects, higher cylinder temperature can be chosen by selecting smaller diameter cylinders.

When free convection effects can be ignored this thermal control may reveal some of the keys to the onset of vortex shedding. From a practical point of view, it is also essential to relate the influence of the cylinder heating to the existence of an effective temperature used to calculate an effective Reynolds number. This would then allow one to know the effective Reynolds number for a heated bluff body and determine the flow regime downwind of this obstacle.

Up to now the notion of effective temperature has especially been used in hot-wire anemometry in the Nusselt–Reynolds relationships. Depending on authors, it is customary to evaluate in these relationships fluid properties either at the mean film temperature, defined as the arithmetic mean of cylinder and fluid temperatures, Hilpert [26], Collis and Williams [27] or the dynamic viscosity at the mean film temperature and the density at the free stream temperature, McAdams [28].

In the present communication the structure of the near wake behind a heated or unheated cylinder is investigated experimentally in air, in the flow regime corresponding to the transition from a 2-D steady to a 2-D periodic wake. When the cylinder is heated the cylinder always operates in the forced convection regime. We present some results obtained in air concerning some aspects of the velocity fields over the

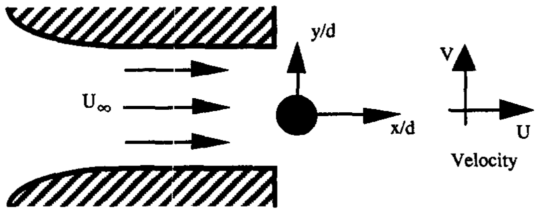

Fig. 1. Experimental set-up.

Reynolds numbers range ($45 < Re < 75$). Particular attention is paid to the influence of the heating on the structure of the wake in relation with the change of physical properties of the fluid with temperature. The notion of effective Reynolds number is discussed and the physical soundness of this approach is sustained by velocity measurements of the near wake. Measurements of the Nusselt number are also presented and analyzed in relation with the effective and film temperatures.

2. EXPERIMENTAL SETUP AND PROCEDURES

The experiments were carried out in air in the potential core of a laminar plane jet. As shown in Fig. 1 the jet exits normally to an end plate (17×35 cm) from a slit of width 1.5 cm and span 15 cm centrally located in this plate. On the centreline, at the nozzle exit, the turbulence intensity u'/U_∞ was approximately 0.4%. The vortex shedding bluff body was a smooth stainless steel 1 mm diameter tube mounted horizontally in the middle of the jet close to the exit plane. Its total length was 15 cm ($L/d = 150$). This obstacle could be heated by Joule effect by means of direct current. The heat input per unit length was calculated from the supplied voltage and current. The temperature of this circular cylinder was obtained by inserting a chromel–constantan thermocouple within the tube at the centre position. The e.m.f. was recorded by a digital voltmeter with an accuracy of $\pm 1°$C. In order to avoid vibrations, the circular cylinder was damped with pieces of foam located at its ends. With the selected diameter, $d = 1$ mm, Reynolds numbers from 40 up to 120 could be obtained by varying the upstream velocity between 0.60 m s^{-1} and 1.80 m s^{-1}. The critical Reynolds number Re_c, corresponding to the transition from a 2-D steady to a 2-D periodic wake, is about 46.1. The 1 mm cylinder temperature excess could reach 150 K at the maximum leading to a maximum value of Ri of about 0.012 corresponding to the forced convection regime. It is worth to note that the small value of the selected cylinder diameter, $d = 1$ mm, is a compromise between spatial resolution problems for velocity and temperature measurements in the wake and low value of Ri to avoid free convection contamination problems. Velocity measurements were made using an LDA TSI system incorporating a 1.5 W Spectra-Physics laser system, an integrated transmission unit and a light collecting system for forward scattering mode. The optical measuring volume was $0.08 \times 0.08 \times 1$ mm^3.

As shown in Fig. 1 the origin of the coordinate system was taken at the centre of the cylinder. The x-axis was measured in the direction of the flow, the y-axis was perpendicular to the flow. In our study all the lengths are non-dimensionalized by the diameter of the cylinder d. The velocities are normalized by the upstream velocity U_∞.

Re is the Reynolds number in isothermal conditions, i.e. calculated with viscosity of the upstream flow. Re_{eff} and Re_{film} are the effective and film Reynolds numbers calculated at the effective temperature T_{eff} and film temperature T_{film}, respectively. The effective temperature T_{eff} is defined in the next section.

3. EXPERIMENTAL RESULTS—DISCUSSION

3.1. *Effective Reynolds number*

As shown in previous studies, Lecordier *et al.* [11], in presence of a 2-D periodic wake, heating the cylinder in air is found to stabilize the flow. In all these experiments made in air the critical heating input needed to cancel the vortex shedding increases with increasing value of the Reynolds number Re calculated with the kinematic viscosity of the non-heated upstream flow. The critical heating input P/L_{crit} and critical cylinder temperature excess $\Delta T_{\mathrm{w_{crit}}} = T_{\mathrm{w_{crit}}} - T_\infty$ are found to be approximately a linear function of the relative reduced Reynolds number ε [$= (Re - Re_c)/Re_c$].

In order to characterize the wake when the cylinder is heated it is possible to define simply an effective temperature and an effective Reynolds number from the following manner. When the heating is just sufficient to stabilize the flow, the effective Reynolds number Re_{eff} [$= U_\infty d/\nu_g(T_{\mathrm{eff}})$] is equal to the critical Reynolds number Re_c. By knowing the temperature dependence of ν_g it is then possible to determine T_{eff}. The effective temperature T_{eff} is the temperature needed to increase the air kinematic viscosity in order that $Re_{\mathrm{eff}} = Re_c$. As shown in Table 1 the determination of T_{eff} for $0 < \varepsilon < 0.25$ shows that: $(\Delta T_{\mathrm{eff}}/\Delta T_{\mathrm{w}}) = 0.24 \pm 0.02$ where $\Delta T_{\mathrm{eff}} = T_{\mathrm{eff}} - T_\infty$ and $\Delta T_{\mathrm{w}} = T_{\mathrm{w}} - T_\infty$.

This result confirms the qualitative observation of Sreenivasan *et al.* [12] and the previous quantitative estimates of Lecordier *et al.* [11] and Paranthoën *et al.* [16] where $(\Delta T_{\mathrm{eff}}/\Delta T_{\mathrm{w}})$ was, respectively, 0.3 and 0.27. This value is also close to the value 0.23 found by Ezerski [13] for the case of a vertical heated cylinder located in a horizontal flow.

It is worth noting that this effective temperature T_{eff} is much lower than the film temperature generally used to take into account the influence of temperature on fluid properties in usual Nu–Re relationships, Collis and Williams [27]. It results that the value of the Reynolds number, calculated using fluid properties at

Table 1. Determination of the effective temperature at the transition

Re	P/L (W m^{-1})	ΔT_w (°C)	Re_{eff}	T_{eff} (°C)	$(\Delta T_{eff}/\Delta T_w)$	T_{film} (°C)	Re_{film}
48.1	9.3	28.2	46.1	28.5	0.25	35.6	44.2
50.1	21.8	63.8	46.1	37.7	0.22	55.5	41.8
52.1	31.7	90.2	46.1	45.7	0.24	69.4	40.6
54.1	41.3	114.1	46.1	50.9	0.25	79.6	39.7
55.8	54.0	145.9	46.1	54.7	0.23	93.9	37.9

the mean film temperature, from which a vortex street exists in the wake of a heated cylinder, cannot be a constant value.

Temperature measurements performed downstream of the heated circular cylinder have shown that this effective temperature is representative of the fluid temperature in the near wake, Paranthoën *et al.* [16].

3.2. *Use of* Re_{eff} *to characterize the flow regime*

As shown in previous studies, Lecordier *et al.* [11], Ezerski [13] in presence of a 2-D periodic wake, heating the cylinder in air is found to stabilize the flow. Suppression of the instability is characterized by a continuous decrease of the amplitude and of the frequency when the heating is increased. It results that the velocity field downstream of the heated cylinder depends on the level of heating. As for an example, the profiles of the longitudinal mean velocity U, rms values of the longitudinal and transverse velocities u' and v', measured at $Re = Re_c + 15$ for various levels of heating ($P/L = 0$ and 44 W m^{-1}) are presented in Figs. 2–4. The large differences observed between the two cases result of differences between the effective Reynolds numbers corresponding to these two experimental situations.

In order to test the validity of our effective Reynolds number approach, some experiments have been performed successively in isothermal conditions and with the heated cylinder. In these experiments the Reynolds numbers Re were different but the same effective Reynolds numbers were obtained by selecting the temperature of the cylinder in order to get the needed effective temperature. The needed cylinder temperature was deduced from the relation $T_{eff} = T_\infty + 0.24(T_w - T_\infty)$ obtained at the transition in Section 3.1.

The profiles of the longitudinal mean velocity and the rms values of the longitudinal and transverse vel-

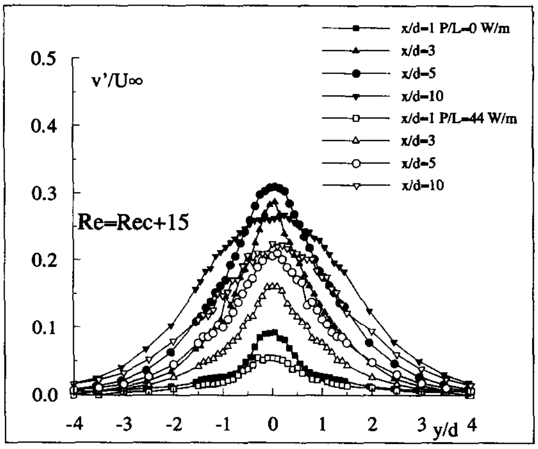

Fig. 3. rms transverse velocity $Re = Re_c + 15$ ($P/L = 0$ W m^{-1}, $P/L = 44$ W m^{-1}).

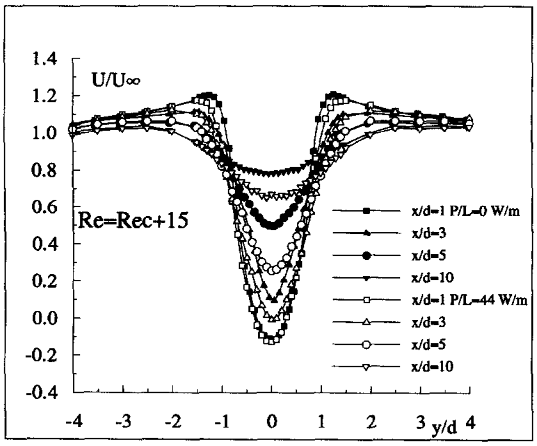

Fig. 2. Mean velocity profiles $Re = Re_c + 15$ ($P/L = 0$ W m^{-1}, $P/L = 44$ W m^{-1}).

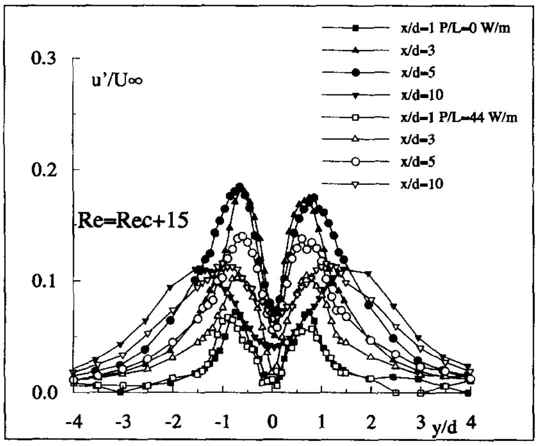

Fig. 4. rms longitudinal velocity $Re = Re_c + 15$ ($P/L = 0$ W m^{-1}, $P/L = 44$ W m^{-1}).

ocities measured for two values of the effective Reynolds number: $Re_{\text{eff}} = Re_c+6$ and $Re_{\text{eff}} = Re_c+15$ are shown in Figs. 5–10. The cases $Re_{\text{eff}}+6$ and $Re_{\text{eff}} = Re_c+15$ have been obtained for the following conditions ($Re = Re_c+10$, $P/L = 20$ W m^{-1}; $Re = Re_c+15$, $P/L = 44$ W m^{-1}) and ($Re = Re_c+15$, $P/L = 0$ W m^{-1}; $Re = Re_c+20$, $P/L = 22$ W m^{-1}), respectively. For $Re_{\text{eff}} = Re_c+6$ results gather very well while for $Re_{\text{eff}} = Re_c+15$ some slight discrepancies appear. These results concerning velocity measurements of the near wake sustain the physical soundness of this approach based on the notion of effective Reynolds number.

3.3. *Nusselt number measurements*

It is also interesting to know if the use of this effective temperature can ameliorate our knowledge of heat transfer from circular cylinders located normal to a horizontal airstream. In our configuration Nusselt numbers of the cylinder have been calculated over the range $30 < Re < 120$ for various values of heating.

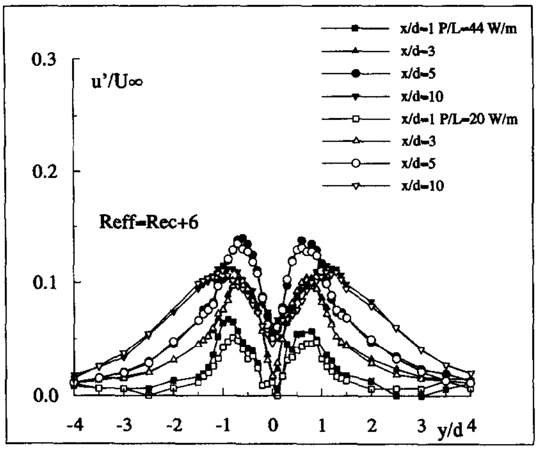

Fig. 7. rms longitudinal velocity $Re_{\text{eff}} = Re_c+6$.

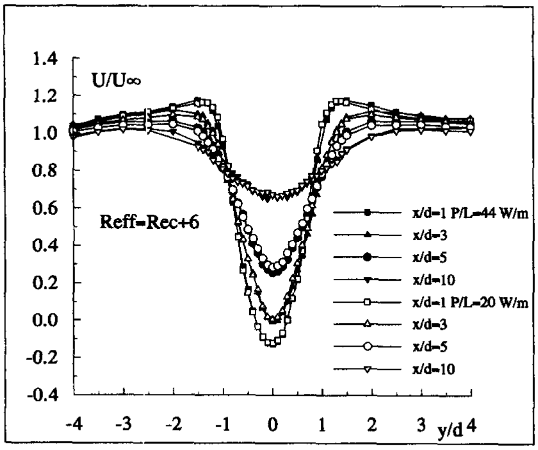

Fig. 5. Mean velocity profiles $Re_{\text{eff}} = Re_c+6$.

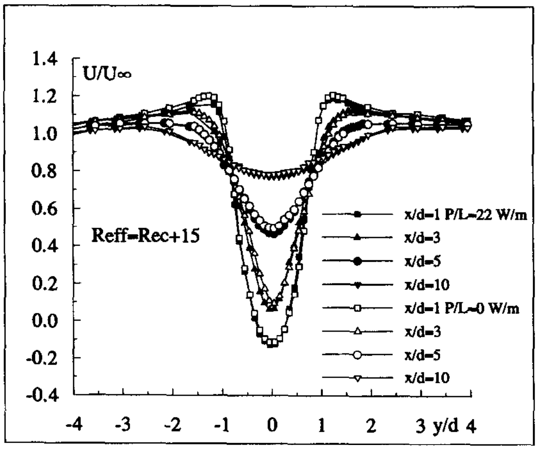

Fig. 8. Mean velocity profiles $Re_{\text{eff}} = Re_c+15$.

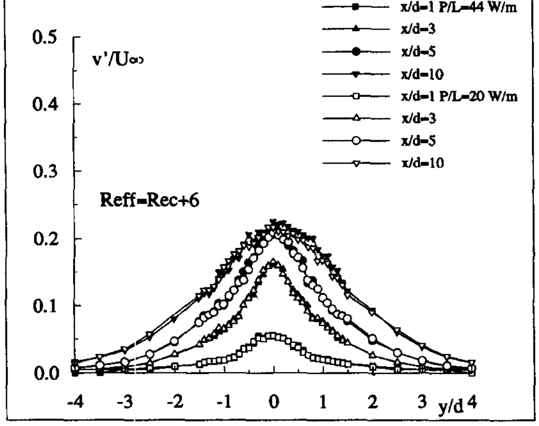

Fig. 6. rms transverse velocity $Re_{\text{eff}} = Re_c+6$.

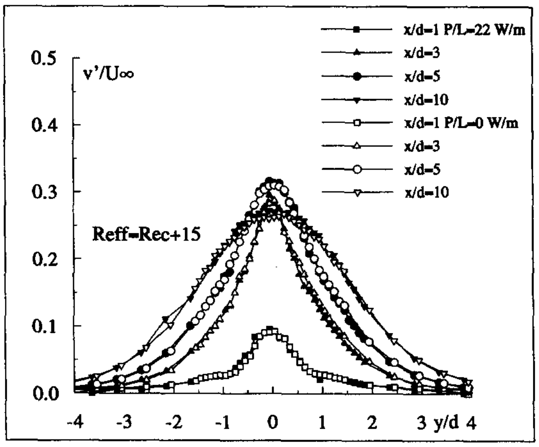

Fig. 9. rms transverse velocity $Re_{\text{eff}} = Re_c+15$.

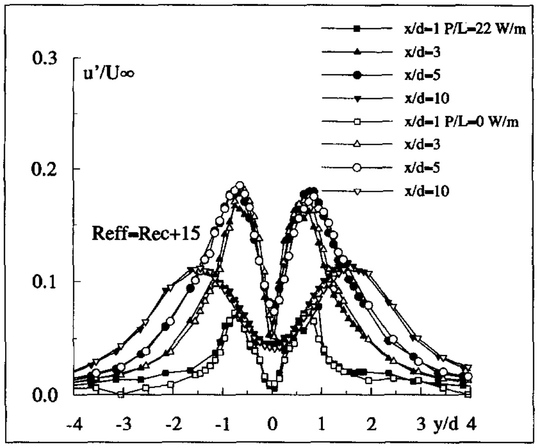

Fig. 10. rms longitudinal velocity $Re_{eff} + Re_c + 15$.

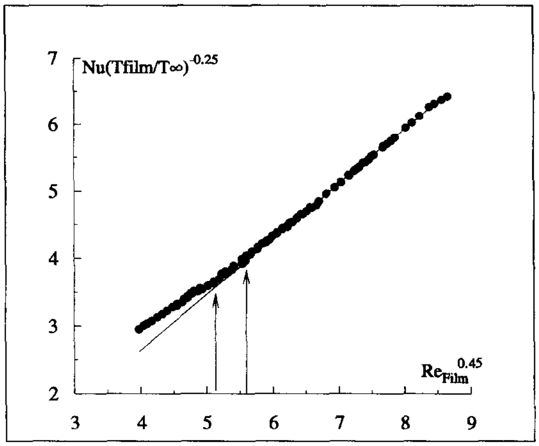

Fig. 12a. Evolution of the Nusselt number with temperature loading vs. $Re_{film}^{0.45}$.

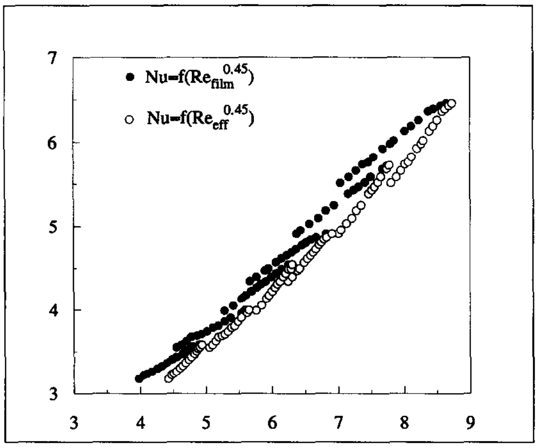

Fig. 11. Evolution of the Nusselt number without temperature loading.

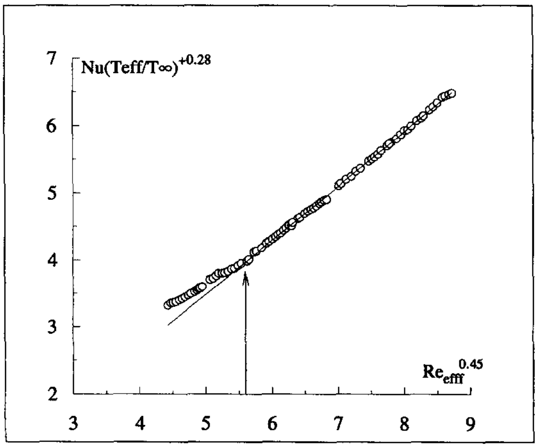

Fig. 12b. Evolution of the Nusselt number with temperature loading vs. $Re_{eff}^{0.45}$.

The value of the global Nusselt number of the cylinder Nu is related to the heating input per unit length P/L, the cylinder temperature excess $(T_w - T_\infty)$ and the thermal conductivity of the fluid λ_g by the following relation:

$$P/L = Nu\pi\lambda_g(T_w - T_\infty).$$

The value of λ_g was calculated at the film temperature which is the characteristic temperature of the heat transfer zone around the cylinder. It was mentioned in Section 1 that previous correlations between Nu and Re realized by Hilpert [26], Collis and Williams [27] or McAdams [28] have been achieved by evaluating fluid properties either at the film temperature or at the free stream temperature. Here the measured values of the Nusselt number are plotted as Nu vs. $Re_{film}^{0.45}$, i.e. by use of the kinematic viscosity evaluated at the film temperature, or as Nu vs. $Re_{eff}^{0.45}$, i.e. by use of the kinematic viscosity evaluated at the effective temperature. As shown in Fig. 11 it is not possible to plot Nu as a continuous function of either $Re_{eff}^{0.45}$ or $Re_{film}^{0.45}$. It turns out that the measured Nusselt number is not only dependent on the effective Reynolds number or on the film Reynolds number. The disagreement and need for a temperature loading factor are evident. As shown in Figs. 12a and b, a continuous function can be obtained by using a temperature loading factor: $(T_{film}/T_\infty)^{-0.25}$ in the Nu–Re_{film} formulation, or a temperature loading factor: $(T_{eff}/T_\infty)^{+0.28}$ in the Nu–Re_{eff} formulation. The temperature loading factor obtained in the Nu–Re_{film} formulation is close to the one proposed by Collis and Williams [27]: $(T_{film}/T_\infty)^{-0.17}$.

The change in the slope of the heat transfer curve already noted by Hilpert [26] and Collis and Williams [27] appears more clearly in the Nu–Re_{eff} formulation. The reason of this result has to be related to the fact that T_{film} is not the temperature to consider to characterize the regime of the wake. We have indicated in Figs. 12a and 12b the location of the critical

Reynolds number when the cylinder temperature excess is between 0 and 150 K. $(Re_{\text{eff}})_{\text{crit}}$ is equal to 46.1 for definition while (Re_{film}) ranges from 36.8 up to 46.1.

As mentioned by Churchill and Brier [29] the use of a temperature factor is arbitrary, and successful representation of the data by this procedure gives no direct information on the fundamental mechanisms. It is worth noting that results in Fig. 11 means that the global Nusselt number is dependent on a Reynolds number intermediate between Re_{eff} and Re_{film}. These results suggest that the front part of the cylinder, from the front stagnation point up to the separation point, could be independent from the wake flow and only dependent on the boundary layer around the cylinder characterized by Re_{film}. Conversely the downstream part of the cylinder, from the separation point up to the rear stagnation point, would depend on the wake by the influence of the recirculation zone and would be characterized by Re_{eff}. A similar analysis has already been made by Douglas and Churchill [30], Van der Heege Zijnen [31], Richardson [32] who proposed a correlation in the following form: $Nu = a(Re_{\text{film}})^{0.5} + b(Re_{\text{film}})^{0.67}$ where the first term represents the heat transfer through the laminar boundary layer on the front portion of the cylinder and the second term represents the heat transfer from the rear portion. It is worth noting that these authors used Re_{film} to characterize the heat transfer in the region where separation occurs.

Due to the limited length of our cylinder we have not tried at this time to propose a general *Nu–Re* relation including Re_{film} and Re_{eff}. This will be made in the future.

4. CONCLUSION

Experimental study of the heated wake downstream of a circular cylinder at low Reynolds numbers has shown the following facts:

—The flow regime downstream of a heated cylinder is strongly dependent on the level of heating and, even in absence of buoyancy effects, heat is never in this situation a passive contaminant.

—The flow regime downstream of the heated circular cylinder can be characterized by an effective Reynolds number calculated at an effective temperature $T_{\text{eff}} = T_\infty + 0.24(T_w - T_\infty)$. Here T_∞ is the temperature of the upsteam flow and T_w is the cylinder temperature.

—This temperature T_{eff}, close to the temperature of the recirculation zone, is lower than the usual film temperature $T_{\text{film}} = T_\infty + 0.5(T_w - T_\infty)$.

—The global Nusselt number is found to be dependent on a Reynolds number intermediate between the effective Reynolds number and the film Reynolds number. This result could be related to the particular influence of the front and rear flow pattern around the cylinder on the local Nusselt number.

REFERENCES

1. Provansal, M., Mathis, C. and Boyer, L., Bénard–von Karman instability: transient and forced regimes. *Journal of Fluid Mechanics*, 1987, **31**, 1–22.
2. Huerre, P. and Monkewitz, P., Local and global instabilities. *Annual Review of Fluid Mechanics*, 1990, **22**, 473–537.
3. Oertel, H., Wakes behind blunt bodies. *Annual Review of Fluid Mechanics*, 1990, **22**, 539–564.
4. Freymuth, P. and Uberoi, M. S., Structure of temperature fluctuations in the turbulent wake behind a heated cylinder. *Physics of Fluids*, 1971, **14**, 2574–2580.
5. LaRue, J. C. and Libby, P. A., Temperature fluctuations in a plane wake. *Physics of Fluids*, 1974, **17**, 1956–1967.
6. Matsumara, M. and Antonia, R. A., Momentum and heat transport in the turbulent intermediate wake of a circular cylinder. *Journal of Fluid Mechanics*, 1993, **250**, 651–668.
7. Xenopoulos, G., Stapounzis, H., Salpistis, C. and Goulas, A., Flow in the wake of a heated circular cylinder in rotary oscillation. *International Symposium on Turbulence, Heat and Mass, Lisbonne*, 1995, 1.3.11.3.6.
8. Eckert, E. R. G. and Soehngen, E., Distribution of heat transfer coefficients around a circular cylinder in crossflow at Reynolds numbers from 20 to 500. *Transactions of the ASME*, 1952, **74**, 343–347.
9. Vilimpoc, V., Cole, R. and Sukanek, P. C., Heat transfer in newtonian liquids around a circular cylinder. *International Journal of Heat and Mass Transfer*, 1990, **33**, 447–456.
10. Mi, J. and Antonia, R. A., Temperature distribution within vortices in the wake of a cylinder. *International Journal of Heat and Mass Transfer*, 1994, **37**, 1048–1050.
11. Lecordier, J. C., Hamma, L. and Paranthoën, P., The control of vortex shedding behind heated circular cylinders at low Reynolds numbers. *Exp. in Fluids*, 1991, **10**, 224–229.
12. Sreenivasan, K., Tavoularis, S., Henry, R. and Corrsin, S., Temperature fluctuations and scales in grid-generated turbulence. *Journal of Fluid Mechanics*, 1980, **100**, 597–621.
13. Ezerski, A. B., Detached flow around a heated cylinder at small Mach number. *Prikladnaya Mekhanika i Tekhn. Fizika*, 1990, **5**, 56–62.
14. Yu, M. H. and Monkewitz, P. A., The effect of nonuniform density on the absolute instability of two-dimensional inertial jets and wakes. *Physics of Fluids A*, 1990, **2**, 1175–1181.
15. Schumm, M., Berger, E. and Monkewitz, P., Self excited oscillations in the wake of two-dimensional bluff bodies and their control. *Journal of Fluid Mechanics*, 1994, **271**, 17–53.
16. Paranthoën, P., Browne, L., Le Masson, S. and Lecordier, J. C., Control of vortex shedding by thermal effect at low Reynolds number. *Int. Rept. MT*1. Rouen University, 1995.
17. Socolescu, L., Mutabazi, I., Daube, O. and Huberson, S., Etude de l'instabilité du sillage 2D derrière un cylindre faiblement chauffé. *C.R. Acad. Sciences Paris* 1996, t322 Serie IIb, pp. 203–208.
18. Noto, K. and Matsumoto R., Numerical simulation on development of the Karman vortex street due to the natural convection. *Flow Visualization III*, **5**, 769–809, Pineridge, 1987.
19. Chang, S. K. and Sa, J. Y., The effect of buoyancy on vortex shedding in the wake of a circular cylinder. *Journal of Fluid Mechanics*, 1990, **220**, 253–266.
20. Hatanaka, K. and Kawahara, M., A numerical study vortex shedding around a heated-cooled circular cylinder by the three-step Taylor–Galerkin method. *International Journal for Numerical Methods in Fluids*, 1995, **21**, 857–867.
21. Nakabe, K., Hasegawa, H., Matsubara, K. and Suzuki,

K., Heat Transfer from a heated cylinder in a flow between parallel plates in a free-forced combined convection regime. ISTP-9, Singapore, 25–28 June 1996.

22. Michaux-Leblond, N. and Belorgey, M., Near wake behaviour of a heated circular cylinder: viscosity–buoyancy duality. *Exp. Therm. Fluid Sci. J.*, 1997 (to be published).
23. Mori, Y., Nijikata, K., Nobuhara, T., A fundamental study of symmetrical vortex generation behind a cylinder by wake heating or by splitter plate or mesh. *International Journal of Heat and Mass Transfer*, 1986, **29**(8), 1193–1201.
24. Morgan, V. T., The overall convective heat transfer from smooth circular cylinders. *Advances in Heat Transfer*, 1975, **11**, 199–263.
25. Fand, R. M. and Keswani, K. K. Combined natural and forced convection heat transfer from horizontal cylinders to water. *International Journal of Heat and Mass Transfer*, 1973, **16**, 1175–1191.
26. Hilpert, R., Wärmeabgage von geheitzen Drähten und Rohren in Lufstrom. *ForschArb. IngWes.*, 1933, **4**, 305–314.
27. Collis, D. C. and Williams, M. J., Two dimensional convection from heated wires at low Reynolds numbers. *Journal of Fluid Mechanics*, 1959, **6**, 357–384.
28. McAdams, W. H., *Heat Transmission*, 3rd ed. McGraw-Hill, 1954.
29. Churchill, S. W. and Brier, J. C., Convective heat transfer from a gas stream at high temperature to a circular cylinder normal to the flow. *Chem. Engng. Prog. Symp. Ser*, 1953, **17**(51), 57–65.
30. Douglas, W. J. M. and Churchill, S. W., Recorrelation of data for convective heat transfer between gas and single cylinders with large temperature differences. *Chemic. Engng. Prog. Symp. Series*, 1956, **52**, 23–28.
31. Van der Heege Zijnen, B. G., Modified correlation formulae for the heat transfers by natural and forced convection from horizontal cylinders. *Appl. Sci. Res.*, 1957, **A6**, 129–140.
32. Richardson, P. D., Estimation of the heat transfer from the rear of an immersed body to the region of separated flow. *ARL* 62-423, 1962.