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Heat transfer in Newtonian liquids around a circular cylinder

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Abstract—Holographic interferometry is used to study heat transfer in Newtonian liquids flowing around a circular cylinder. Temperature differences are of the order of 0.1°C. Consequently, physical property variations are negligible. Isotherms, and local and mean Nusselt number data are obtained for water, corn syrup and dilute corn syrup, covering a Reynolds number range of 0.002–64 and Gr/Re^2 of 0.001–4.4. The Prandtl number range is from about 5.5 to 27 000. The isotherms and local Nusselt numbers clearly show evidence for the twin vortices at Reynolds numbers greater than about 5. The mean Nusselt number in this entire Reynolds number range can be correlated by a single equation. This is in contrast with previous investigators who employed two equations for Reynolds numbers greater than and less than one. The magnitude of natural convection effects for water is in agreement with previous investigators. However, the value of Gr/Re^2 at which natural convection starts to become important is found to increase with Prandtl number.

1. INTRODUCTION

MORGAN [1], Zukauskas [2] and Gnielinski [3] reviewed the many studies of heat transfer from circular cylinders. Most of these studies were carried out with techniques which required large temperature differences, used air as the heat transfer medium and yielded mean Nusselt numbers. In the low Reynolds number ($Re < 40$) regime, Davis [4] measured mean heat transfer coefficients for a variety of fluids. Piret *et al.* [5] reported mean Nusselt numbers for water. Collis and Williams [6] examined air. With fluids having the viscosity of water or smaller, buoyancy effects might be expected to be significant at these Reynolds numbers, especially when the temperature difference is large. Fand and Keswani [7], Sharma and Sukhatme [8], Sunden [9], and Juma and Richardson [10] examined the influence of natural convection. Again, only mean heat transfer coefficients were measured. Natural convection is considered to be important when the measured Nusselt number deviates from the correlation obtained at low Grashof numbers.

With one exception, none of the work reported to date provides a picture of the isotherms and local Nusselt numbers at low Reynolds numbers. Eckert and Soehngen [11] used interferometry to determine these quantities in air. Reynolds numbers were inferred from the measured mean Nusselt numbers and an available correlation. Such detailed profiles are valuable when developing theoretical models for the heat transfer. In addition, the profiles give qualitative information on the magnitude of any natural convection effects which may be present.

Methods such as interferometry permit direct measurement of the isotherms and have the added benefit of requiring only small temperature differences. Physical property variations are therefore negligible. The use of standard interferometry with liquids has been limited because of the need for a reference cell to equalize the optical path lengths in the two arms of the device. Holographic interferometry eliminates the need for a separate reference cell. In this study, we used the holographic technique to study heat transfer around circular cylinders in crossflow. The fluids used were water, corn syrup and their mixtures. These provided a wide range of Reynolds, Prandtl and Grashof numbers.

2. EXPERIMENTAL SETUP AND PROCEDURES

Water or corn syrup (CS55 ADM, Western Syrup Company, Chicago, Illinois) was forced to flow around a stainless steel cylinder mounted in the middle of a long rectangular flow channel. The corn syrup was a clear liquid with viscosity varying from batch to batch (2–5 Pa s). The viscosity was measured with a rotational viscometer, and found to be Newtonian. The fluids' thermal diffusivities were determined experimentally by a method described elsewhere [12]. The flow channel was a rectangular, plexiglass duct 610 mm long, 200 mm wide and 27 mm high. The cylinder was 5.95 mm in diameter and was mounted in the center of optical quality glass windows located sufficiently downstream in the flow channel to insure that the flow was fully developed by the time it reached

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NOMENCLATURE

c_p	heat capacity
d	cylinder diameter
D	diffusion coefficient
Gr	Grashof number, $\rho^2 d^3 g \beta \Delta T/\mu^2$
g	acceleration of gravity
h	heat transfer coefficient
h'	mass transfer coefficient
k	thermal conductivity
l_h	channel height
Nu	Nusselt number, hd/k
Pr	Prandtl number, $c_p\mu/k$
Re	Reynolds number, $\rho Vd/\mu$
Sc	Schmidt number, $\rho D/\mu$
Sh	Sherwood number, $h'd/D$
ΔT	temperature difference
V	average velocity in the channel without the cylinder
V_c	average velocity corrected for channel blockage.

Greek symbols

β	thermal expansion coefficient
μ	viscosity
ρ	density.

Subscripts

m	mean value around the cylinder.

the cylinder. The cylinder was heated electrically with embedded nichrome wires.

Water flow was by gravity from a feed tank into the experimental flow channel and then into a receiving tank, from which it was pumped back into the feed tank. The corn syrup was pumped into the channel by a gear pump. Flexible hoses were used to reduce vibration. Mass flow rates were measured by weighing the fluid collected in a specified time. Reynolds numbers reported in this work are based on the cylinder diameter and the average velocity of the fluid in the empty (unblocked) channel.

The local Nusselt numbers were obtained by photographing and analyzing the infinite interference fringe patterns formed by heating the cylinder. At the start of an experiment, the fluid was circulated through the system for a sufficiently long time for thermal equilibrium to be established. A hologram of the flowing fluid around the cylinder was then made. Electrical energy was supplied to the cylinder, and the system allowed to come to steady state, as indicated by the constant number of fringes in the heated test region. A photograph of the interference pattern was then taken. The flow rate was changed, and when the system reached the new steady state, a photograph of the new interference pattern was taken.

The photographic negatives were digitized with an EIKONIXSCAN 78/99 digitizing camera system. Since Nusselt numbers were required at fixed angular positions, protractor masks were superimposed on each of the negatives. (See, for example, Fig. 1(a).) The digital images were stored in a DEC VAX 11/780 minicomputer. Subsequently, an image was recalled from memory and displayed and manipulated to obtain the local Nusselt numbers on a LEXIDATA high resolution monitor. An algorithm similar to that of Eckert and Soehngen [11] was used to obtain the local Nusselt numbers. The temperature along each fringe is a constant and is directly related to the number of fringes between the point of interest and a reference point. Three or four fringes closest to the surface were chosen at 5° increments around the entire cylinder. The temperature of each fringe was calculated and fit to a straight line. With the present setup, the temperature difference per fringe was approximately 0.04°C. The slope of this line gave the temperature gradient at the surface, which is directly related to the local Nusselt number.

The local data were integrated to find the mean Nusselt number. To assist in the integration routine, the raw data were smoothed using a cubic spline fit. (Figure 1(b) compares the raw and smoothed data for one experiment.) For all cases, no attempt was made to force the data to be symmetric.

3. RESULTS AND DISCUSSION

Local Nusselt numbers are strongly influenced by the flow pattern around the cylinder. There are two laminar flow regimes covered in this study—creeping flow ($0.002 < Re < 5$) and recirculating vortex flow ($5 < Re < 64$). The first flow regime is characterized by smooth symmetrical streamlines upstream and downstream of the cylinder. The characteristic of the second regime is the formation of a recirculating vortex near the rear stagnation point of the cylinder. Vortex shedding was observed at $Re > \sim 64$.

3.1. *Creeping flow regime*

A representative sample of the isotherms and local Nusselt number pattern for the creeping flow regime is illustrated in Fig. 1. The boundary layer is formed at the front stagnation point (0°) and grows in thickness toward the rear stagnation point (180°). A maximum in the local Nusselt number occurs at the front stagnation point where the boundary layer is thinnest and consequently presents the least resistance to heat transfer. Since this resistance increases with the boundary layer thickness, the Nusselt number

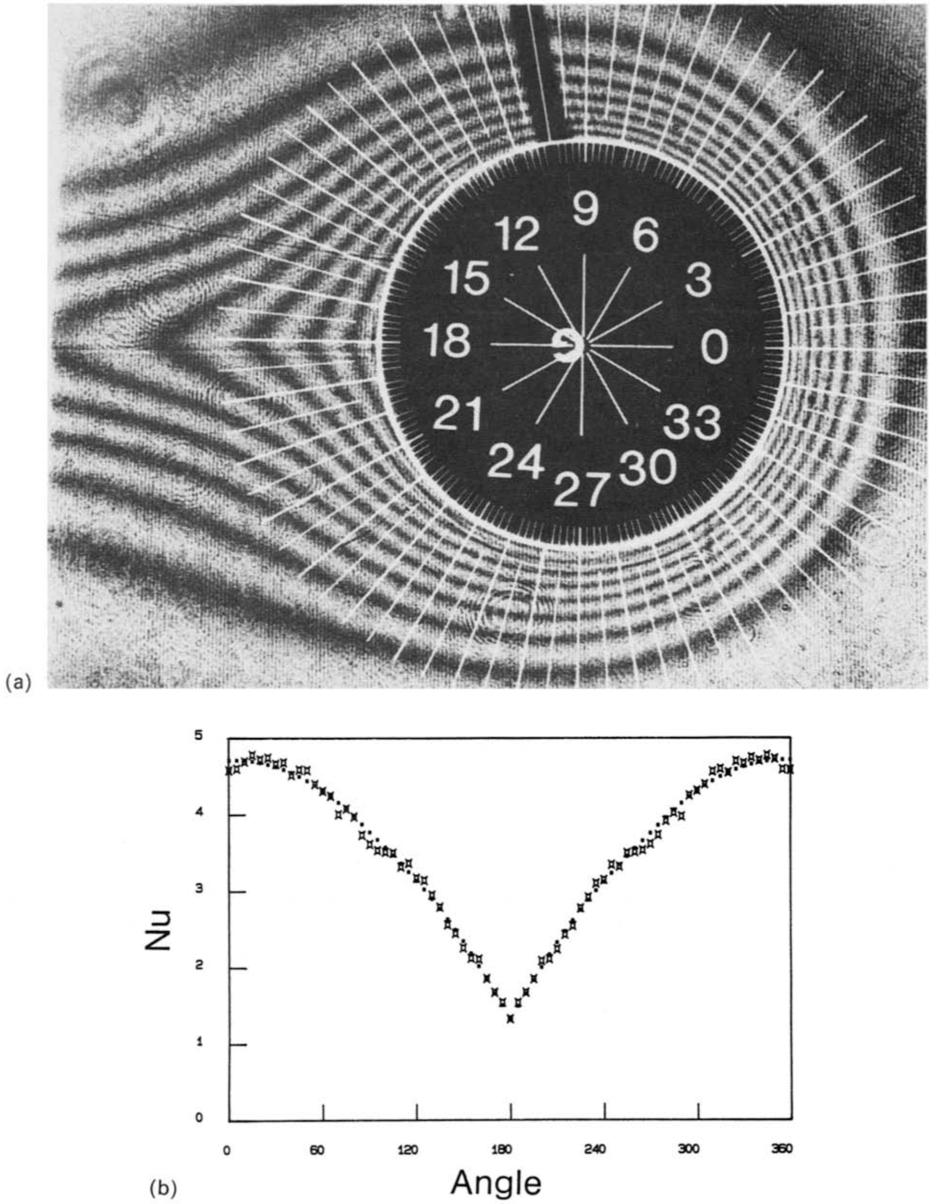

FIG. 1. Isotherms (a) and local Nusselt numbers (b) for flow of dilute corn syrup around a heated cylinder. The raw (□) and smoothed (■) data are illustrated in (b). Flow is from right to left ($Re = 0.036$, $Pr = 5100$, $Gr/Re^2 = 0.065$).

decreases until it reaches a minimum at the rear stagnation point.

For the corn syrup and corn syrup solutions, the Grashof numbers are small, and therefore natural convection is unimportant. In these cases, the local Nusselt numbers and isotherm patterns are symmetrical around the rear stagnation point. A plot of the local Nusselt numbers for different Reynolds numbers is shown in Fig. 2. Since different fluids were used for the different experiments shown in this figure, the Nusselt numbers are normalized with respect to the Prandtl number to the 0.31 power. As discussed below, this is the observed Prandtl number dependence in all our experiments. Hence, Fig. 2 shows how the Nusselt number depends only on the Reynolds number; fluid property effects are eliminated. When normalized with respect to the maximum local Nusselt number, the shape of all these curves remains unchanged with increasing Reynolds number up to $Re \sim 5$. This suggests that there is no change in the flow pattern around the cylinder until the onset of the recirculating vortices.

3.2. *Recirculating vortex flow regime*

Taneda [13] found that in the Reynolds number range of about 5–45, flow in the rear portion of the cylinder forms twin recirculating vortices. He found vortex shedding at $Re > 45$. As expected, the effect of this vortex pattern is apparent in the isotherms and local Nusselt numbers. An example of these patterns is shown in Fig. 3. Downstream from the cylinder the higher temperature of the fluid along the wake boundary quickly cools due to energy exchange with the outer flow. When this cooler fluid recirculates back

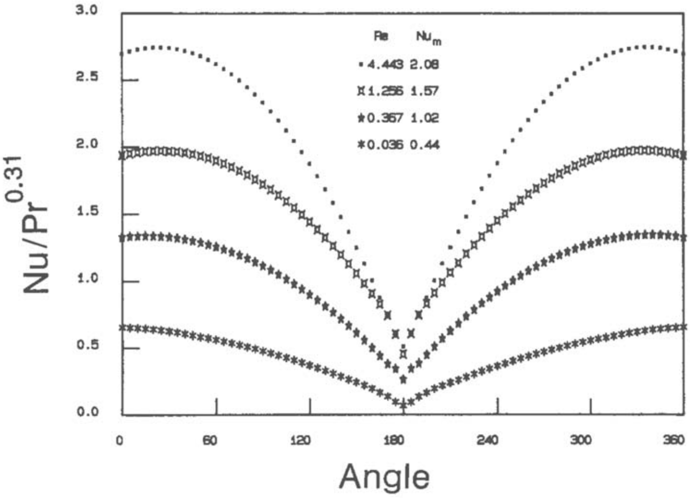

FIG. 2. Normalized local Nusselt numbers for different Reynolds numbers for corn syrup and dilute syrup in the creeping flow regime. Natural convection is unimportant in these cases. The data are normalized by $Pr^{0.31}$ to eliminate differences based on fluid properties.

to the cylinder it increases the rate of heat transfer at the rear stagnation point (180°), resulting in a local maximum at this point.

The local Nusselt number at different Reynolds numbers in this flow regime is shown in Fig. 4. All of these experiments were performed with water. As the Reynolds number increases, the Nusselt numbers on the upstream side of the cylinder increase, while the Nusselt numbers near the rear stagnation point (150°–210°) vary by only 20%. In Fig. 5, the Nusselt numbers for these experiments are normalized with respect to the maximum value for the particular Reynolds number. The curves for the two highest Reynolds numbers have the same shape, indicating that the flow patterns are similar for Reynolds numbers in this regime. At the lower Reynolds numbers, natural convection is seen to be important. This is true for all the low Reynolds number water experiments. Indeed, the exact Reynolds number where the recirculating vortex appears cannot be determined in these experiments because of natural convection. The normalized local Nusselt numbers for the two flow regimes (creeping flow and recirculating vortex) are shown for comparison in Fig. 6. Natural convection is addressed in somewhat more detail below.

An additional feature of Figs. 5 and 6 should be pointed out. The normalized Nusselt numbers in all experiments are not unity at both 0° and 360° as expected, and the slopes of the Nusselt number curves are not the same when approached from the top and bottom of the cylinder. This is an indication that the cylinder was not perfectly aligned with the camera for some of these experiments. Hence, there is a slight difference in the observed fringe pattern on the top and bottom of the cylinder.

Yonemoto and Tadaki [17] examined the problem of mass transfer from a cylinder in a flowfield, both experimentally and theoretically. Their calculations are valid up to the Reynolds number at which shedding occurs. Their predictions for the local Nusselt number around a cylinder in water at a Reynolds number of 33.6 is shown in Fig. 7. As shown in the figure, their predictions are in good agreement with our experiments at approximately the same Reynolds number. The average Nusselt number computed by Yonemoto and Tadaki is 8.05. This is approximately 11% different from the value predicted by the correlation discussed below.

While Taneda [13] found vortex shedding for Reynolds numbers greater than 45, we did not observe this phenomenon until $Re > 64$. This is probably due to the proximity of the channel wall to the cylinder. Grove *et al.* [14] reported that this proximity can retard the shedding. In those cases where vortex shedding took place, isotherms similar to Fig. 8 were observed. The downstream tails of the isotherms periodically elongated and broke away from the cylinder, and new ones formed in their place. This observation agreed qualitatively with the numerical analysis of Jain and Goel [15] for unsteady laminar forced convection. Their analysis was for Reynolds numbers of 100 and 200. Comparison of Fig. 8 with their computed isotherms show very little difference. We did not obtain any data in the vortex shedding regime because of the time dependence of the isotherms. However, visual observation of the expanding, shedding and forming of the isotherms led us to conclude that the local Nusselt numbers on the downstream side of the cylinder vary with time while those on the upstream side remain relatively constant. This, too, is in agreement with the results obtained by Jain and Goel. The magnitudes of the local maximum at the rear stagnation point also seem to increase significantly with the Reynolds number in contrast to the relatively constant minimum for the creeping flow regime.

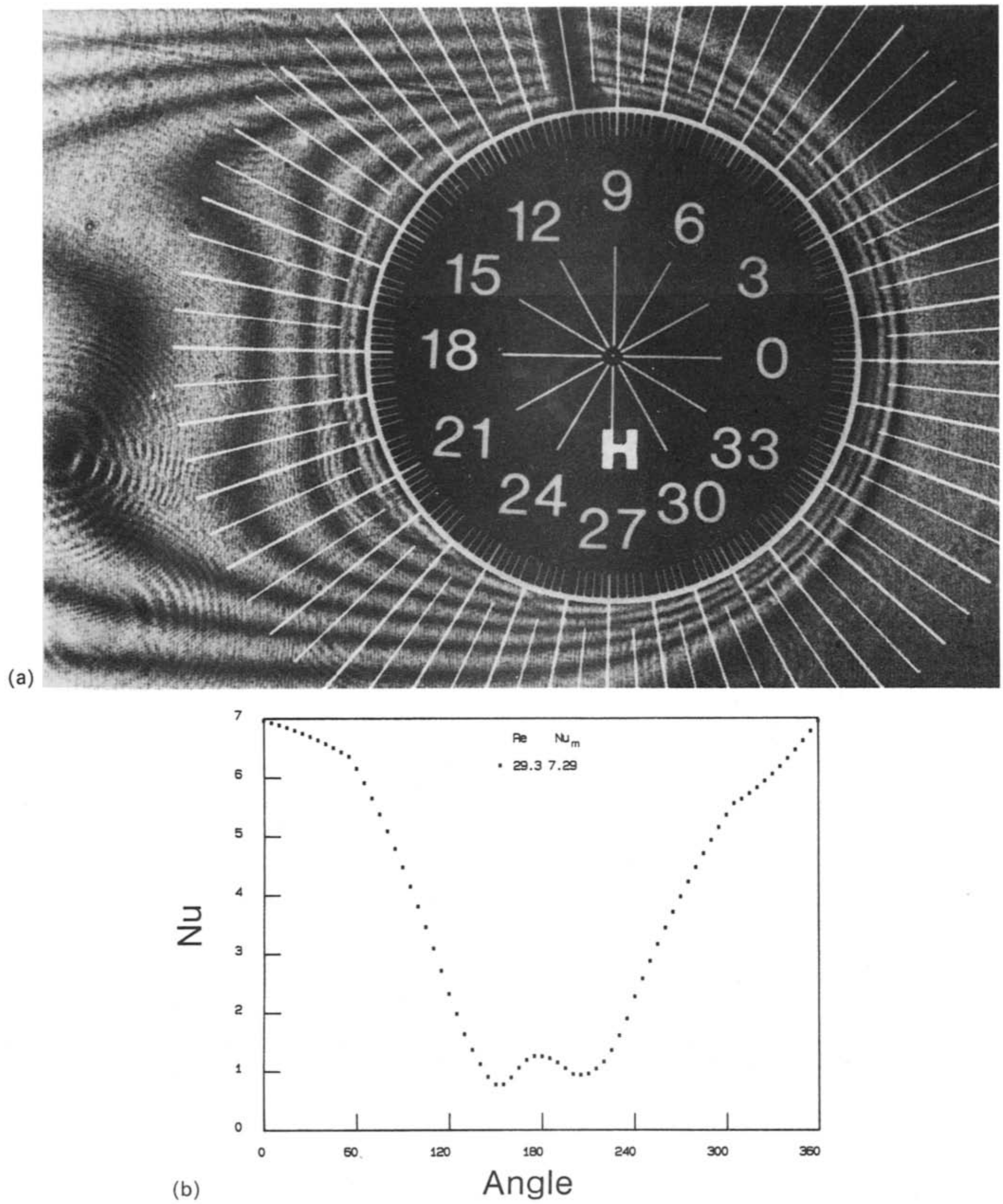

FIG. 3. Isotherms (a) and local Nusselt numbers (b) for water flow in recirculating vortex regime. Flow is from right to left ($Re = 29.3$, $Pr = 6.0$, $Gr/Re^2 = 0.022$).

3.3. *Natural convection effects*

Based on their experiments in air, Sharma and Sukhatme [8] proposed a transition from free convection to combined convection when $Gr/Re^{3.25} = 0.185 \pm 0.10$, and combined convection to forced convection when $Gr/Re^{1.8} = 0.58 \pm 0.13$. Similarly, Fand and Keswani [7] proposed that for water when $Gr/Re^2 < 0.5$ forced convection heat transfer dominates; when $0.5 < Gr/Re^2 < 2$, the predominant heat transfer mechanism is still forced convection, but natural convection affects the overall heat transfer coefficient by as much as 10%. When $2 < Gr/Re^2 < 40$, natural and forced convection effects are of the same order of magnitude, and when $Gr/Re^2 > 40$ natural convection effects predominate.

In order to investigate the magnitude of the natural convection effect, plots of the local Nusselt numbers divided by $Pr^{0.31}$ for water (indicated by ○) and dilute corn syrup (indicated by ●) at about the same Reynolds numbers are shown in Fig. 9. Natural convection increases the local Nusselt number and displaces the minimum in the Nusselt number from the rear stagnation point toward the top side of the cylinder. The ratio Gr/Re^2 for the two curves is 4.4 and 0.005. The ratio of the areas under the two curves, which is equivalent to the ratio of combined convection to forced convection, is 1.37. This is in agreement with the prediction of Fand and Keswani who indicate equal contributions from both mechanisms at this value of Gr/Re^2.

Also shown in Fig. 9 are data for corn syrup solution at approximately the same value of Gr/Re^2, 4.2, as the data for water. (Since these data are at a much lower value of the Reynolds number, they are not

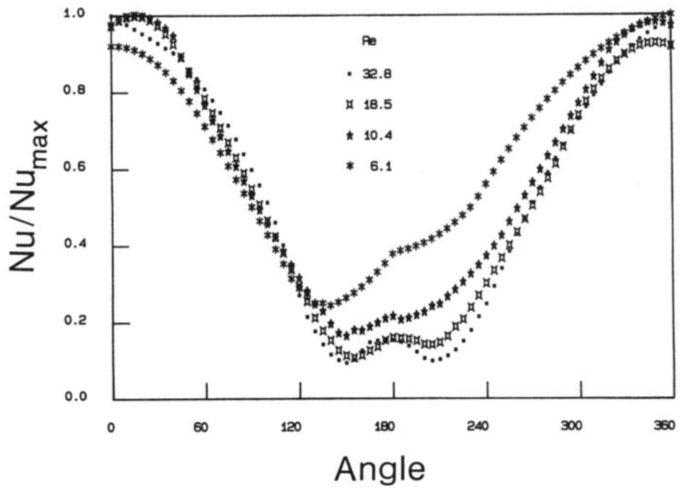

FIG. 4. Local Nusselt numbers for the recirculating vortex regime. All experiments performed with water.

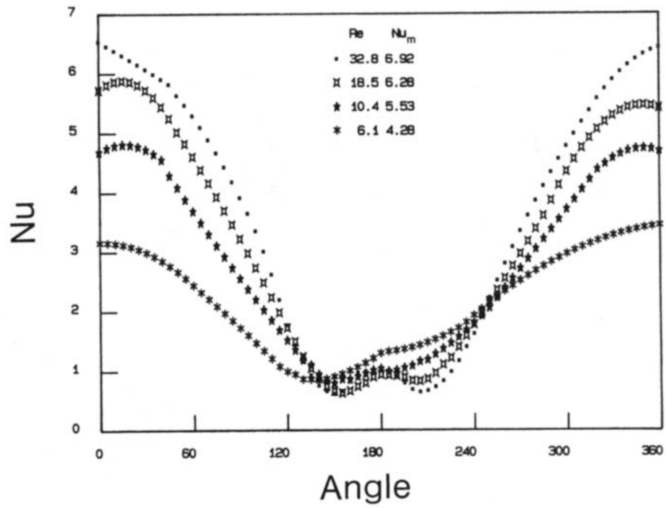

FIG. 5. Local Nusselt numbers for the recirculating flow regime normalized with respect to the maximum local Nusselt number at each Reynolds number.

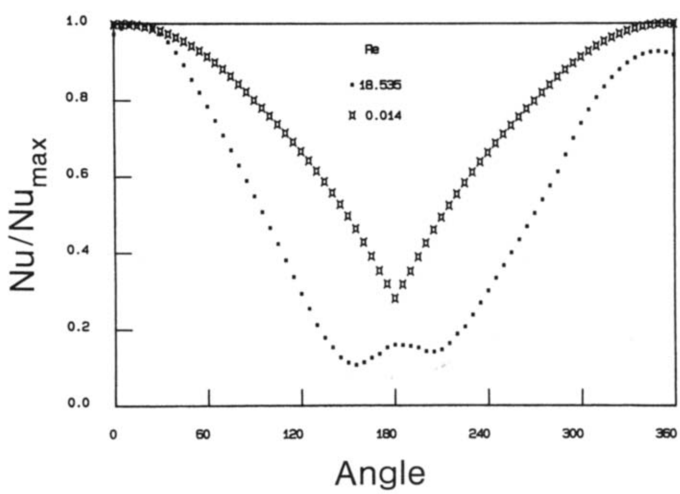

FIG. 6. Normalized local Nusselt numbers for the creeping (□) and recirculating (■) flow regimes.

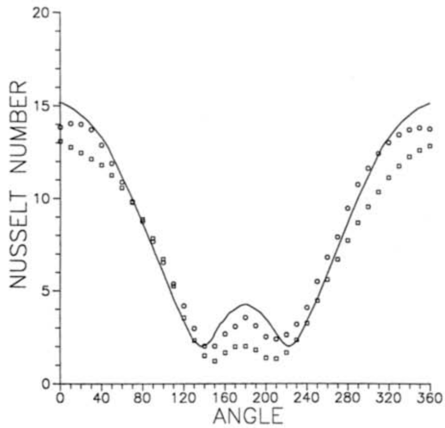

FIG. 7. Comparison of theoretical predictions of Tadaki for water at $Re = 33.6$ (——) with experimental measurements at $Re = 32.8$ (□) and 39.4 (○).

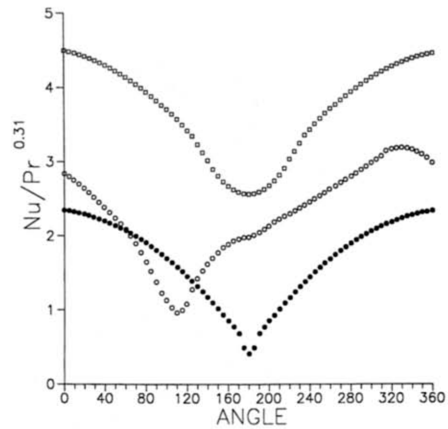

FIG. 9. Local Nusselt numbers divided by $Pr^{0.31}$ with (○) and without (●) natural convection at approximately the same Reynolds number (○, water, $Re = 1.93$, $Pr = 6.0$, $Gr/Re^2 = 4.4$; ●, dilute corn syrup, $Re = 1.82$, $Pr = 120$, $Gr/Re^2 = 0.005$). Also shown are Nusselt numbers for dilute corn syrup (□) at $Re = 0.002$, $Pr = 11700$, $Gr/Re^2 = 4.2$. These data show no evidence of natural convection.

scaled with the Prandtl number.) The Prandtl number for this fluid was 1.17×10^4. Unlike the water data, these show no evidence of natural convection. Consequently, the Reynolds number for the transition from forced to combined convection must increase with the Prandtl number.

An experimental isotherm at a ratio of $Gr/Re^2 = 4$ is shown in Fig. 10. This pattern may be compared with the numerically calculated isotherm pattern of Badr [16] for air. Our calculated increase in mean Nusselt number due to the contribution from natural convection in this figure is 25%, with the mean Nusselt number due to forced convection calculated from the correlation discussed below. This is substantially larger than the approximately 10% increase at a Reynolds number of 5 and a Prandtl number of 0.7 calculated by Badr.

3.4. *Mean Nusselt number correlation*

When the mean Nusselt number data (excluding those with $Gr/Re^2 > 0.5$) were correlated, it was found that the data could be fit with the single equation

$$Nu_m = 1.34 Pr^{0.31} Re^{0.32}. \tag{1}$$

The data and the correlating curve are illustrated in Fig. 11. As an estimate of the mean error, 90% of all the data lie within $\pm 10\%$ of this curve. Also plotted

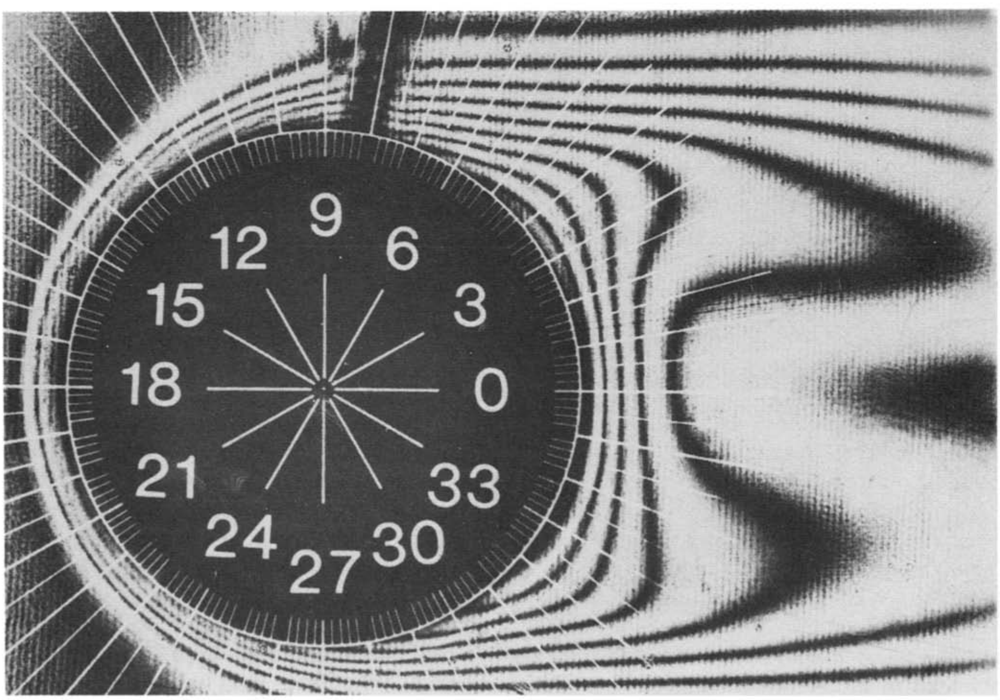

FIG. 8. Isotherms for water at conditions close to the initiation of shedding. Flow is from left to right ($Re = 51$, $Pr = 6$, $Gr/Re^2 = 0.063$).

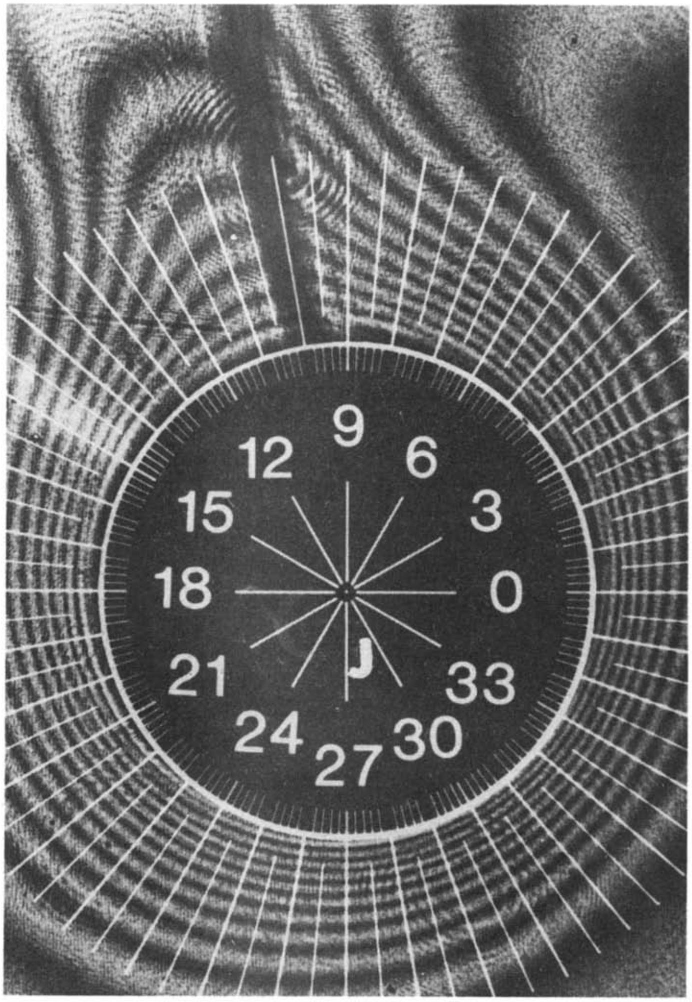

FIG. 10. Isotherms for water at large Gr/Re^2. The contribution of natural convection to the overall Nusselt number is about 25%. Flow is from right to left ($Re = 2.2$, $Pr = 6.0$, $Gr/Re^2 = 4.0$).

in this figure are the correlations suggested by Morgan [1]

$$0.09 < Re < 1, \quad Nu_m = 0.91 Re^{0.28}\, Pr^{0.31} \qquad (2)$$

$$1 < Re < 35, \quad Nu_m = 0.90 Re^{0.384}\, Pr^{0.31} \qquad (3)$$

and that of Gnielinski [3]

$$Re < 1, \quad Nu_m = 0.75 Re^{1/3}\, Pr^{1/3} \qquad (4)$$

$$Re > 1, \quad Nu_m = 0.664 Re^{1/2}\, Pr^{1/3}. \qquad (5)$$

The present data do not show any significant reason for using a different Reynolds number dependence above and below $Re = 1$. In addition, the present data fall significantly above the available correlations. This latter discrepancy has to do with our definition of the Reynolds number. (In Fig. 11, the Reynolds number based on the velocity in the unblocked channel was used for all the curves.)

Previously, most investigators sought to correct the velocity used in the Reynolds number for solid or wake blockage, for non-uniform velocity profiles, and for the finite aspect ratios (diameter of cylinder/channel width) of their test channels. For example, Gnielinski suggests defining the velocity as

$$V_c = V/(1-\pi d/4l_h) \qquad (6)$$

where V_c is the corrected velocity, V the average velocity in the open channel, d the cylinder diameter and l_h the channel height. Corrections such as this are supposed to make correlations generally applicable, even for experiments with small wires in channels with zero aspect ratio. These attempts to correct for channel blockage cannot be considered wholly satisfactory since an order of magnitude difference can be found in the corrected velocity when different methods are used. However, using the blockage formula of equation (6), the coefficients in equations (4) and (5) become 0.799 and 0.73, respectively, for the present experiments. Clearly, this change will not lead to any significant improvement in the agreement.

Similar discrepancies were found by Yonemoto and Tadaki [17]. Their mass transfer experiments gave Sherwood numbers which were higher than those predicted by available correlations such as equations (2)

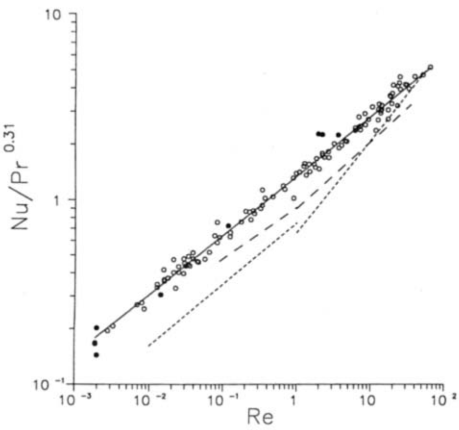

FIG. 11. Mean Nusselt numbers divided by $Pr^{0.31}$ as a function of Reynolds number. $Gr/Re^2 < 0.5$ (○); $Gr/Re^2 > 0.5$ (●). Also shown are the predictions of equations (2) and (3) (——), and equations (4) and (5) (-----).

and (3) or (4) and (5). As with our heat transfer results, their values are about 50–60% greater than those predicted by the available correlations. They developed a new correlation based on their computations. However, their experimental data, which were obtained in the Reynolds number range of about 2–40, appear to fit the following expression:

$$Sh_m = 1.6Sc^{0.31}\,Re^{0.33} \qquad (7)$$

with the Schmidt number dependence assumed to be the same as that found here for the Prandtl number. This expression predicts Sherwood numbers only about 20% greater than ours.

Also plotted in Fig. 11 are the data for $Gr/Re^2 > 0.5$, although these were not included in the correlation for the mean Nusselt number. The water data discussed above with $Gr/Re^2 \approx 4$ are seen at $Re \approx 1$. The data for other fluids at lower Reynolds numbers are all within $\pm 10\%$ of the correlation. It is apparent that there is no significant increase in the heat transfer as a result of the presence of natural convection. Indeed, based on the data in Fig. 9, natural convection effects become important at a value of Gr/Re^2 which depends on the Prandtl number.

4. CONCLUSIONS

Holographic interferometry has been used to measure the temperature distributions around a cylinder in crossflow at low Reynolds numbers. Local and mean Nusselt numbers were found from the temperature profiles. The results indicate that the distribution of the local Nusselt numbers varies in the same manner as the velocity. In the creeping flow regime ($Re < 5$), the Nusselt number is a maximum at the front stagnation point and a minimum at the rear of the cylinder. In the steady recirculating vortex regime ($5 < Re < 65$), there is a local maximum at the rear stagnation point, with minima at $180° \pm 30°$ due to the vortices.

For low viscosity fluids, the boundary between the creeping flow and recirculating vortex regimes is difficult to identify clearly because of the onset of natural convection effects. Examination of the local Nusselt number patterns indicates that the effects of natural convection in water are in qualitative agreement, at least, with the results of Fand and Keswani [7]. For the other fluids with Prandtl numbers more than two orders of magnitude larger than water, the onset of natural convection effects on the local Nusselt numbers appears to be delayed to higher values of the Gr/Re^2 ratio. While Fand and Keswani give a critical value of 0.5 for this ratio, the high Prandtl number fluids ($Pr \approx 10^3$) show no evidence of natural convection up to a value of 4.

Finally, our mean Nusselt number correlation for Reynolds numbers less than about 65 show a single Reynolds number dependence over the entire range. Nusselt numbers are 30–50% higher than those from previous correlations. This is certainly an aspect ratio effect.

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TRANSFERT THERMIQUE DANS DES LIQUIDES NEWTONIENS AUTOUR D'UN CYLINDRE CIRCULAIRE

Résumé—On utilise l'interférométrie holographique pour étudier le transfert thermique dans des fluides newtoniens qui s'ecoulent autour d'un cylindre circulaire. Les différences de température sont de l'ordre de 0,1°C. Par suite, les variations des propriétés physiques sont négligeables. Des isothermes, des nombres de Nusselt locaux et moyens sont obtenus pour l'eau, du sirop de maïs plus ou moins concentré, pour couvrir des domaines de nombres de Reynolds entre 0,002 et 64 et de Gr/Re^2 entre 0,001 et 4,4. Le domaine de nombre de Prandtl va de 5,5 à 27 000. Les isothermes et les nombres de Nusselt locaux montrent clairement les tourbillons jumeaux à des nombres de Reynolds plus grands que 5 environ. Le nombre de Nusselt moyen dans ce domaine de nombre de Reynolds peut être représenté par une équation simple. Ceci est en contradiction avec d'autres représentations qui utilisent deux équations selon que le nombre de Reynolds est plus grand ou plus petit que l'unité. L'importance des effets de la convection naturelle est comme indiquée par des prédécesseurs. Néanmoins, la valeur de Gr/Re^2 pour laquelle la convection naturelle commence à devenir importante est trouvée augmenter avec le nombre de Prandtl.

WÄRMEÜBERGANG IN EINEM NEWTON'SCHEN FLUID AN EINEM KREISRUNDEN ZYLINDER

Zusammenfassung—Der Wärmeübergang an einem von einem Newton'schen Fluid umströmten Kreiszylinder wird mittels holografischer Interferometrie untersucht. Die auftretenden Temperaturunterschiede liegen in der Größenordnung von 0,1°C; daher kann mit konstanten Stoffwerten gerechnet werden. Bei Reynolds-Zahlen von 0,002 bis 64, Prandtl-Zahlen von ungefähr 5,5 bis 27 000 und einem Verhältnis (Gr/Re^2) von 0,001 bis 4,4 werden sowohl die lokale und mittlere Nusselt-Zahl als auch die Isothermen für Wasser und verdünnten sowie unverdünnten Maissirup bestimmt. Bei Reynolds-Zahlen größer als ungefähr 5 lassen die Isothermen und die lokalen Nusselt-Zahlen klar Doppelwirbel erkennen. Die mittlere Nusselt-Zahl korreliert über den gesamten Bereich mit der Reynolds-Zahl und läßt sich mit einer Gleichung beschreiben. Hierin unterscheiden sich diese Ergebnisse von vorherigen Untersuchungen, die jeweils eine Gleichung für Reynolds-Zahlen größer und kleiner eins angeben. Die Effekte aus natürlicher Konvektion in Wasser liegen in der gleichen Größenordnung wie sie von anderen Forschern festgestellt wurden. Weiterhin konnte gezeigt werden, daß mit ansteigender Prandtl-Zahl auch das Verhältnis (Gr/Re^2) ansteigt. Ob die natürliche Konvektion berücksichtigt werden muß, ist an dem Verhältnis (Gr/Re^2) zu erkennen.

ТЕПЛОПЕРЕНОС В НЬЮТОНОВСКИХ ЖИДКОСТЯХ ПРИ ОБТЕКАНИИ ЦИЛИНДРА КРУГЛОГО СЕЧЕНИЯ

Аннотация—С использованием голографической интерферометрии исследован теплоперенос в ньютоновских жидкостях при обтекании цилиндра круглого сечения. Разность температур была порядка 0,1°C. Следовательно, изменения физических свойств были пренебрежимо малыми. Для воды, кукурузного сиропа и разбавленного кукурузного сиропа получены изотермы, а также локальные и средние числа Нуссельта в диапазоне чисел Рейнольдса 0,002–64 и в диапазоне величин отношения Gr/Re^2, составляющем 0,001–4,4. Значения числа Прандтля изменялись от приблизительно 5,5 до 27 000. Изотермы и локальные числа Нуссельта во всем указанном диапазоне изменений числа Рейнольдса могут быть описаны единым уравнением. Это противоречит результатам предыдущих исследований, где для чисел Рейнольдса больше или меньше единицы использовались два уравнения. Значения, полученные для естественной конвекции воды, согласуются с результатами предыдущих исследований. Однако найдено, что величина отношения Gr/Re^2, при которой естественная конвекция становится важным фактором, увеличивается с ростом числа Прандтля.