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TURBULENT HEAT TRANSFER COEFFICIENTS IN AN ISOTHERMAL-WALLED TUBE FOR EITHER A BUILT-IN OR FREE INLET

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Abstract—Local heat transfer coefficients were determined for turbulent flow in an isothermal-walled circular tube whose inlet was either built into a large wall or was supported so as to be in free space. In either case, the sharp-edged nature of the inlet induced flow separation and subsequent reattachment just downstream of the inlet, and these processes were found to have a dominant influence on the thermal entrance region. At the point of reattachment of the flow, the heat transfer coefficients for the case of the built-in inlet ranged from 4.15 to 2.4 times the corresponding fully developed values over the Reynolds number range from 5000 to 88 000. The length of the thermal entrance region was also increased by the presence of the inlet-induced separation. With the tube inlet built into a large wall or baffle plate, the heat transfer coefficients in the immediate neighborhood of the inlet were found to be lower than those for a tube with a free inlet, the differences being in the range of 10%.

NOMENCLATURE

A_i	mass transfer area per module
D	inside diameter of test section tube
D_o	outside diameter of test section tube
D_B	diameter of baffle plate
$\mathscr{D}$	diffusion coefficient
K_i	mass transfer coefficient per module
L_{mod}	axial length of a module
ΔM_i	sublimed mass per module
$\dot{m}_i$	per-module rate of mass transfer per unit area
Pr	Prandtl number
$\dot{Q}$	rate of volume flow
Re	Reynolds number
Sc	Schmidt number
Sh	local Sherwood number (value of Sh_i assigned to mid-point of module)
Sh_i	Sherwood number per module
Sh_{fd}	fully developed Sherwood number
Sh_{max}	peak value in the Sh distribution
T_w	inside wall temperature of test section tube
X	axial coordinate
$\dot{w}$	rate of mass flow.

Greek symbols

μ	viscosity
ν	kinematic viscosity
ρ_{nw}	naphthalene vapor density at wall
ρ_{nb}	naphthalene vapor density in bulk
τ	duration of run.

INTRODUCTION

IN THIS paper, experiments are reported for turbulent heat transfer in a circular tube with a sharp-edged inlet. A particular feature associated with this type of inlet is the presence of a region of separated, recirculating fluid situated adjacent to the tube wall, just downstream of the entrance cross section. The separated region exerts a dominant influence on the initial development of the velocity and temperature distributions, which continue to develop simultaneously along the length of the tube. Analytical treatments of simultaneously developing velocity and temperature distributions in a tube do not take account of the presence of the flow separation and, thereby, provide predictions for the local heat transfer coefficient that are at variance with reality.

It may be questioned whether the description of an inlet as being sharp-edged is sufficient to fix the velocity distribution at the entrance cross section or in the initial portion of the tube. In this regard, it is relevant to consider the role played by the surface which frames the inlet aperture, particularly, the radial extent of the framing surface. This issue, which has not heretofore been explored in the published literature, is one of the foci of this paper.

Two extreme sizes of framing surface will be investigated. One case corresponds to the inlet being built into the downstream wall of a plenum chamber of large diameter. With such an arrangement, the region from which the tube draws its fluid is limited to the space upstream of the inlet. Furthermore, a portion of the fluid flow that enters the tube passes along the wall which frames the inlet and, in doing so, experiences some degree of hydrodynamic development.

In the other investigated case, the upstream portion of the tube was altogether free of involvement with any wall or baffle, so that the inlet cross section is framed only by the tube wall thickness. For this situation, when the tube is operated in the suction mode, fluid may be drawn into the inlet from all directions in space, so that certain streamlines experience large turns (>90°) as they enter the tube. On the other hand, there is less pre-entry hydrodynamic development than for the case of the built-in inlet.

For both of the aforementioned inlet configurations, local entrance-region and fully developed heat transfer coefficients were measured for Reynolds numbers spanning the range between 5000 and 88 000.

The other objective of the work was to measure and report highly accurate axial distributions of the local heat transfer coefficient corresponding to the uniform temperature boundary condition at the tube wall and to a sharp-edged inlet. In the modern heat transfer literature, the uniform heat flux boundary condition has been used almost exclusively in turbulent pipe-flow experiments because of the convenience afforded by ohmic heating. In the older literature, uniform wall temperature experiments were performed by condensing steam on the outer surface of the test section tube, with individual steam compartments used to get quasi-local heat transfer coefficients. The excessive scatter in the data obtained by this technique [1] raises doubts about their accuracy.

As an alternative to the direct measurement of heat transfer coefficients, mass transfer results may be transformed to heat transfer results by applying the analogy between the two processes. According to the analogy, the Prandtl numbers of the deduced heat transfer coefficients are equal to the Schmidt numbers of the measured mass transfer coefficients. Furthermore, if the mass transfer experiments are performed for uniform concentration of the transferred species at the tube wall, the deduced heat transfer coefficients correspond to the uniform wall temperature boundary condition.

In light of the foregoing, the available mass transfer coefficients measured for turbulent pipe flows by electrochemical techniques can be transformed to uniform-wall-temperature heat transfer coefficients. Since these mass transfer coefficients correspond to high Schmidt numbers (> 1000), so do the deduced heat transfer coefficients correspond to the same range of Prandtl number. In this regard, it is well established that the thermal development in a turbulent pipe flow is highly sensitive to Prandtl number, with the thermal entrance length decreasing as the Prandtl number increases. Therefore, the entrance region heat transfer coefficients deduced from electrochemical mass transfer measurements are not applicable to moderate Prandtl numbers.

In the present experiments, a mass transfer technique is also employed—the naphthalene sublimation technique. However, the Schmidt number for naphthalene sublimation in air is 2.5, which corresponds to a Prandtl number of 2.5 for the deduced heat transfer coefficients. This Prandtl number is intermediate between those for air and water. As noted earlier, entrance region and fully developed mass (heat) transfer coefficients are presented here for Reynolds numbers ranging from 5000 to 88 000.

EXPERIMENTAL APPARATUS

The test section tube was made up of an array of interlocking modules of the type shown in Fig. 1. Each module consists of a cylindrical metallic (aluminum) shell whose inner surface is coated with a layer of solid naphthalene. The coating is applied by a casting procedure. Precise mating of successive modules is assured by interlocking recesses that are provided at the respective ends of each module.

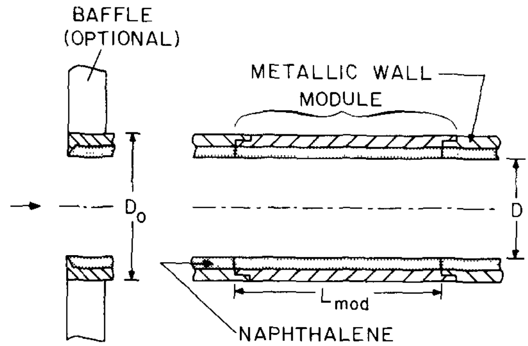

FIG. 1. Details of the experimental apparatus.

The inner surface of the naphthalene coating, which served as the bounding surface for the airflow, had a diameter $D = 3.272$ cm. Modules of three different axial lengths L_{mod} were employed, respectively, $L_{mod}/D = 0.388$, 0.776, and 1.553, with the shortest modules used adjacent to the inlet in order to resolve the more rapid transfer coefficient variations which occur there. All told, the test section tube included 21 modules. Once the modules had been assembled at the start of a data run, the joints at the module interfaces were sealed by pressure-sensitive tape applied to the outer surface of the metallic shell.

The forwardmost module, that containing the tube inlet, was of somewhat different design from the others. It is shown on the LHS of Fig. 1 (the baffle plate is optional and will be discussed shortly). The main feature of this module is the internally beveled metal cap at its upstream end, which ensures that the front face of the tube does not participate in the mass transfer process. In the analogous heat transfer problem, it would be virtually impossible to achieve perfectly adiabatic conditions at the front face. The value of L_{mod}/D for the forwardmost module was 0.466.

As noted in the Introduction, two types of inlet framing arrangements were employed. In one case, the upstream end of the tube was supported so as to be free of involvement with any walls or baffles. In that case, the inlet is framed by the wall thickness of the tube which, from Fig. 1, can be characterized by the diameter ratio D_o/D, the value of which was 1.475. For the other framing arrangement, the baffle plate shown on the LHS of Fig. 1 was employed. The upstream face of the baffle was positioned flush with the front face of the tube. A better perspective on the size of the baffle plate is provided by the inset of Fig. 2, which is drawn to scale. The ratio of the baffle diameter to the tube bore diameter, D_B/D, was equal to 28, which is believed large enough to approximate infinity.

The system was operated in the suction mode, with air drawn into the tube inlet from the temperature-controlled laboratory room. Once the air passed through the test section, it was ducted successively to a

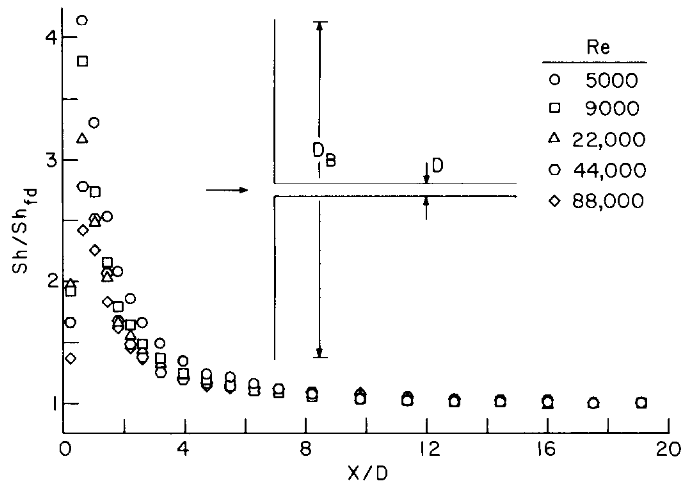

FIG. 2. Axial distributions of the local Sherwood number for the case in which the tube inlet is built into a large wall or baffle plate.

flow metering station (calibrated rotameters), a control valve, and a blower. The latter was situated in a service corridor adjacent to the laboratory room so that its compression-heated, naphthalene-laden discharge could be vented away from the laboratory.

With regard to instrumentation, a calibrated thermocouple was installed in each of four selected modules during the casting process, flush with the exposed surface of the naphthalene. The instrumented modules were distributed along the length of the tube, and axial temperature uniformity (typically to 0.05°C) was found to prevail. Periodic readings of the thermocouple e.m.f.'s were made during the course of a data run by a programmable datalogger having a resolution of 1 μV. The mass of each module was measured both before and after each data run. The mass measurements were made with a Sartorius ultra-precision, electronic analytical balance with a resolving power of 10^{-5} g and a capacity of 166 g. Typical changes of mass during a data run were in the range of 0.05 g.

DATA REDUCTION

The mass transfer coefficients to be evaluated here are quasi-local in that they are average values with respect to the lengths of the respective modules. These mass transfer coefficients correspond to the boundary condition of uniform naphthalene vapor density ρ_{nw} at the tube wall. This follows from the fact that there is a one-to-one correspondence between ρ_{nw} and the wall temperature T_w, and the temperature was found by measurement to be axially uniform.

The per-module mass transfer coefficient K_i for a typical module i was evaluated from the defining equation

$$K_i = \dot{m}_i/(\rho_{nw} - \rho_{nb,i}). \quad (1)$$

In this equation, $\dot{m}_i$ is the rate of mass transfer averaged over the surface area of module i, and $\rho_{nb,i}$ is the bulk density of naphthalene vapor in the fluid passing through i. If ΔM_i represents the measured mass of naphthalene sublimed during the duration τ of the data run and A_i is the exposed surface area, then $\dot{m}_i = \Delta M_i/\tau A_i$. The vapor density ρ_{nw} is obtained from the Sogin vapor pressure–temperature relation [2] in conjunction with the perfect gas law.

To find $\rho_{nb,i}$, it may be noted from mass conservation that the increase in ρ_{nb} sustained by the fluid passing through any module j is

$$(\Delta\rho_{nb})_j = (\Delta M_j/\tau)/\dot{Q}, \quad (2)$$

where $\dot{Q}$ is the volume flow. Then if ρ_{nb} is zero at the inlet of the test section

$$\rho_{nb,i} = \sum_{j=1}^{i-1} (\Delta\rho_{nb})_j + \tfrac{1}{2}(\Delta\rho_{nb})_i. \quad (3)$$

The variation of the volume flow along the test section was negligible in these experiments.

Once the K_i are determined, they may be represented in dimensionless form in terms of the Sherwood number

$$Sh_i = K_i D/\mathscr{D} = (K_i D/\nu)Sc, \quad (4)$$

in which the Schmidt number $Sc\,(=\nu/\mathscr{D})$ has been used to eliminate the diffusion coefficient. According to ref. [2], $Sc = 2.5$ for naphthalene diffusion in air. The kinematic viscosity which appears on the RHS of the equation was evaluated as that for pure air. The Sherwood number results will be parameterized by the Reynolds number defined in the conventional manner for pipe flows as

$$Re = 4\dot{w}/\mu\pi D, \quad (5)$$

in which $\dot{w}$ is the mass flow rate.

RESULTS AND DISCUSSION

To begin the presentation, a complete set of Sherwood number distributions, encompassing both the entrance and the fully developed regions, are plotted in Fig. 2. In this figure, data are presented for

Reynolds numbers ranging from 5000 to 88 000. Each data point represents the Sherwood number for an individual module, and the point is plotted at the X/D value at the axial midpoint of the module. The local Sherwood numbers for each Reynolds number are normalized by the corresponding fully developed value, so that all the plotted distributions approach unity at sufficiently large downstream distances (the fully developed values will be presented shortly). As indicated by the diagram in the inset, the data correspond to the case in which the tube inlet is built into a large wall.

The distributions for all Reynolds numbers share a common shape which is typical of that for pipe flows with inlet-related flow separation. Immediately downstream of the inlet, the Sherwood number rises steeply, attains a sharp maximum, and then decreases, rapidly at first, more gradually later, and finally levels off to a fully developed value. The maximum corresponds to the reattachment of the separated flow at the tube wall, and the subsequent decrease is associated with the development of the reattached flow.

This type of distribution is at variance with that predicted by analyses of turbulent pipe flow with simultaneously developing velocity and temperature distributions. Those analyses ignore flow separation and predict Sherwood (Nusselt) number distributions which decrease monotonically along the length of the pipe. Furthermore, the experimentally determined Sh/Sh_{fd} values attained in the initial portion of the entrance region are much higher than those predicted by the separation-free analytical models.

The distributions are quite sensitive to the Reynolds number in the initial portion of the entrance region, especially at the peak point. In the range of Re from 5000 to 88 000, the peak value of Sh/Sh_{fd} ranges from 4.15 to 2.4. Indeed, aside from the first measurement station, the lower the Reynolds number, the larger is the value of Sh/Sh_{fd} in the initial portion of the entrance region ($X/D \leqslant 6$). However, beyond the peak, which occurs at $X/D = 0.660$, the effect of Reynolds number steadily wanes, and the distributions tend to draw together. In the downstream portion of the entrance region ($X/D > 6$), the Reynolds number dependence of the results tends to reverse (i.e. higher Re, larger Sh/Sh_{fd}), but the overall spread with Reynolds number is very small.

At the first measurement station, the data point for $Re = 5000$ is out of order relative to the subsequent stations. This may be due to the fact that the first station lies in the separated region, and the low-Reynolds-number recirculating flow may be overly lacking in vigor compared with the higher Reynolds number cases.

The length of the entrance region will now be considered, and for this purpose the criterion $Sh/Sh_{fd} = 1.05$ will be used to define the entrance length $(X/D)_e$. From this it follows that $(X/D)_e = 8$–10 for $Re = 5000$ and 9000, and $(X/D)_e = 10$–12 for $Re = 22\,000, 44\,000$, and 88 000. If account is taken of the fact that $Sc = 2.5$ for these experiments, comparison with the literature (Fig. 4 of ref. [3]) shows that the aforementioned entrance lengths are larger than those for tubes with separation-free inlets.

The measured fully developed Sherwood numbers used in the foregoing Sh/Sh_{fd} distributions are presented as a function of the Reynolds number in Fig. 3. Also shown in Fig. 3 is the Petukhov–Popov correlation [4] and the modification of that correlation by Gnielinski [5]. Whereas the former is applicable only for the fully turbulent range $Re \geqslant 10\,000$, the latter is purported to be valid for both the transition and turbulent regimes (i.e. $Re > 2300$). Both of these correlations are based on heat transfer data, and they have been transformed here to mass transfer terms by replacing Nu with Sh and Pr with Sc.

Whereas the data fall within the 6% confidence limit specified by Petukhov–Popov for their correlation, they are equally supportive of the Gnielinski correlation. The very good agreement between the present mass transfer data and widely accepted heat transfer correlations lends strong support to the present experimental approach as well as to the naphthalene sublimation technique.

Attention is now turned to the effect of the surface which frames the tube inlet, and Fig. 4 has been prepared for this purpose. Figure 4 consists of three graphs, and in each graph Sh/Sh_{fd} is plotted vs X/D for two Reynolds numbers. For each Reynolds number, the open symbols represent the case in which the tube inlet is built into a large baffle plate or wall, while the closed symbols denote the case in which the inlet is framed only by the tube wall thickness.

From an inspection of Fig. 4, it is seen that in the separated region and at reattachment, the Sh values for the no-baffle case are consistently higher than those in the presence of the baffle. The deviations appear to be greatest at the lowest and highest of the investigated Reynolds numbers, with a lesser spread between the two sets of data at the intermediate Reynolds numbers.

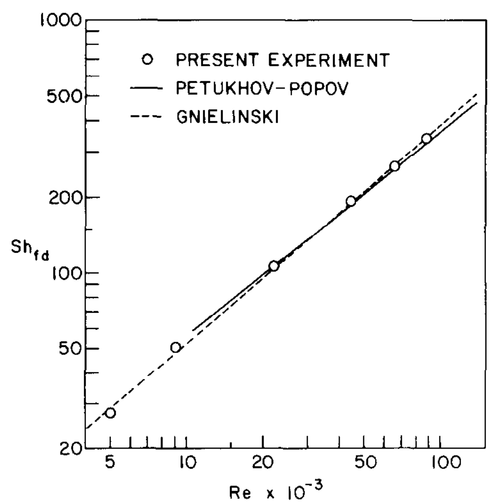

FIG. 3. Fully developed Sherwood numbers.

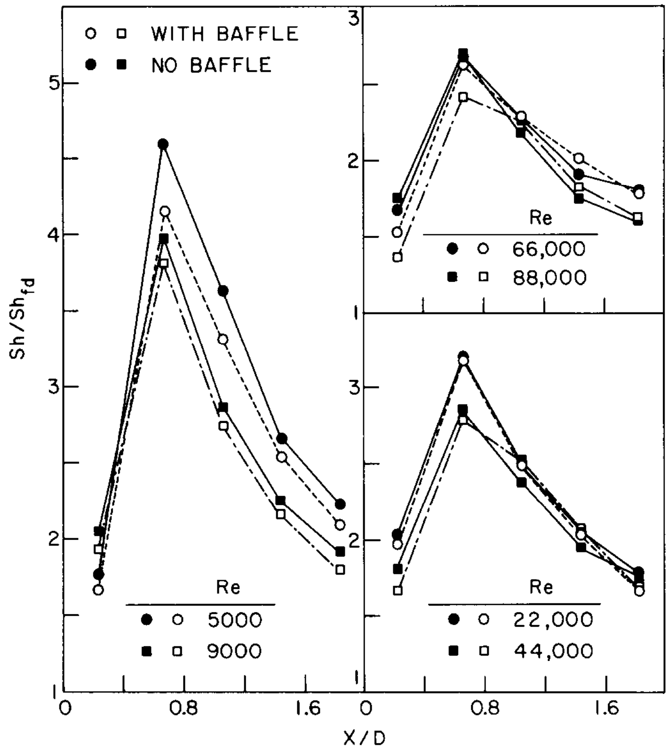

FIG. 4. Comparison of local Sherwood numbers with and without a large wall or baffle plate at the tube inlet.

At $Re = 5000$, the respective peak values of Sh/Sh_{fd} for the no-baffle and with-baffle cases are 4.6 and 4.15, and at $Re = 88\,000$ the corresponding values are 2.7 and 2.42.

Beyond the peak of the curve, the no-baffle data continue to fall somewhat higher than the with-baffle data for $Re = 5000$ and 9000. On the other hand, for the larger Reynolds numbers, there is no clear trend as to which set of data falls higher, suggesting that the influence of the different framing configurations has died away.

Overall, aside from one point at which there is a deviation in excess of 20%, the baffle-related spread of the data is less than about 10%.

The decrease of the Sherwood number due to the presence of the baffle may be attributed, at least in part, to the buildup of a boundary layer on the surface of the baffle plate. The boundary layer will be thicker at low Reynolds numbers and, on this basis, it is reasonable to expect that the magnitude and the downstream persistence of the decrease would be greatest for these Reynolds numbers. This expectation is, in large part, fulfilled by the data. It appears, however, that an inertia-related mechanism comes into play at the highest Reynolds numbers and influences the separated and reattachment Sherwood numbers.

In Fig. 5, the Sherwood numbers at the respective peaks of the axial distributions are brought together and plotted as a function of the Reynolds number. Data are shown for both the with-baffle and no-baffle cases and, in accordance with Fig. 4, the data for the latter lie above the former. However, aside from the 10% spread of the data at the highest and lowest Reynolds numbers, the deviations between the two cases are rather small.

Least-squares power laws of the form

$$Sh_{max} = C\ Re^n, \tag{6}$$

have been passed through the data, with values of C, n equal to 0.3586, 0.683 and 0.4058, 0.676, respectively for the with-baffle and no-baffle cases. The fact that the peak Sherwood numbers are well described by a power law with an exponent of approximately 2/3 is not unexpected. A similar finding has been encountered in other turbulent duct flows in which there is a reattachment downstream of a zone of separation—for instance, in the heat transfer experiments of ref. [6]

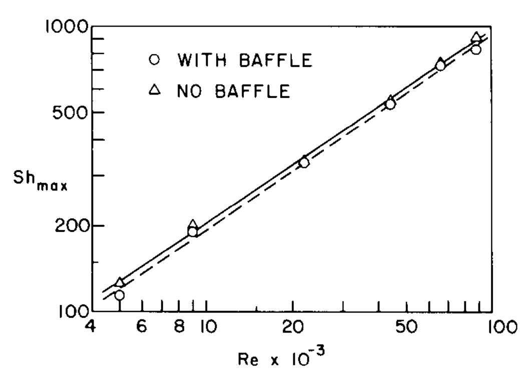

FIG. 5. Sherwood numbers corresponding to the peaks of the axial distributions.

where the separation was caused by an orifice plate with a central aperture.

To generalize the Sh_{max} correlation of equation (6) to apply to heat transfer situations with a Prandtl number other than 2.5, use may be made of the Zukauskas correlation for cylinders in crossflow [7]. In view of the important role played by flow separation in that situation, the Prandtl number dependence, $Pr^{0.37}$, of the Zukauskas correlation will be used here. With this, equation (6) becomes (after substituting Nu for Sh)

$$Nu_{max} = 0.712C\ Re^n\ Pr^{0.37}, \qquad (7)$$

in which C and n are specified in the text following equation (6).

CONCLUDING REMARKS

The local Sherwood numbers determined here can be regarded equally well as Nusselt numbers and, as such, they correspond to the uniform wall temperature boundary condition. These data are believed to be of significantly higher accuracy than those previously available for this boundary condition from direct heat transfer measurements.

The flow separation and subsequent reattachment which occurs because of the sharp-edged inlet has a dominant effect on the thermal entrance region. At the point of reattachment of the flow, very high heat transfer coefficients prevail—ranging, for the built-in case, from 4.15 to 2.4 times the respective fully developed values over the Reynolds number range from 5000 to 88 000. The thermal entrance length is also increased due to the presence of the inlet-induced separation.

When the tube inlet is built into a large wall or baffle plate, the heat transfer coefficients in the immediate neighborhood of the inlet are lower than those for a tube whose inlet is framed only by the tube wall thickness. The differences are in the 10% range.

REFERENCES

1. L. M. K. Boelter, G. Young and H. W. Iversen, An investigation of aircraft heaters XXVII. Distribution of heat transfer rate in the entrance region of a circular tube, NACA TN 1451 (1948).
2. H. H. Sogin, Sublimation from disks to air streams flowing normal to their surfaces, *Trans. Am. Soc. Mech. Engrs* **80**, 61–71 (1958).
3. J. A. Malina and E. M. Sparrow, Variable-property, constant-property, and entrance-region heat transfer results for turbulent flow of water and oil in a circular tube, *Chem. Engng Sci.* **19**, 953–962 (1964).
4. B. V. Karlekar and R. M. Desmond, *Heat Transfer* (2nd edn.), pp. 497–498. West Publishing, St. Paul, Minnesota (1982).
5. V. Gnielinski, New equations for heat and mass transfer in turbulent pipe and channel flow, *Int. Chem. Engng* **16**, 359–368 (1976).
6. K. M. Krall and E. M. Sparrow, Turbulent heat transfer in the separated, reattached, and redevelopment regions of a circular tube, *J. Heat Transfer* **88**, 131–136 (1966).
7. A. A. Zukauskas, Heat transfer from tubes in crossflow, in *Adv. Heat Transfer* **8**, 93–160 (1972).

COEFFICIENT DE TRANSFERT THERMIQUE TURBULENT DANS UN TUBE A PAROI ISOTHERME AVEC ENTREE ENCASTREE OU LIBRE

Résumé—Des coefficients locaux de transfert thermique sont déterminés pour un écoulement turbulent dans un tube à section circulaire et à paroi isotherme, dont l'entrée est soit encastrée dans un mur étendu soit supportée de façon à être libre dans l'espace. Dans chaque cas, la nature de l'entrée à bord abrupt induit une séparation de l'écoulement et un réattachement juste en aval de l'entrée, et ceci a une influence dominante sur la région d'entrée thermique. Au point de réattachement les coefficients de convection, dans le cas de l'entrée encastrée, sont entre 4,15 et 2,5 fois les valeurs correspondantes du régime établi à des nombres de Reynolds allant de 5000 à 88 000. La longueur d'établissement thermique est aussi accrue par la présence de la séparation induite à l'entrée. Avec l'entrée du tube prise dans un mur étendu ou un baffle plan, les coefficients de convection dans le voisinage immédiat de l'entrée sont plus faibles que ceux pour un tube avec entrée libre, les différences étant de l'ordre de 10%.

WÄRMEÜBERGANGSKOEFFIZIENTEN BEI TURBULENTER ROHRSTRÖMUNG MIT ISOTHERMER WANDUNG BEI EINGEBAUTEM UND FREIEM EINLASS

Zusammenfassung—Es werden örtliche Wärmeübergangskoeffizienten bei turbulenter Strömung in einem Kreisrohr mit isothermer Wand bestimmt. Der Einlaß war entweder in einer großen Wand eingebaut oder wurde so angebracht, wie wenn er im freien Raum wäre. In beiden Fällen verursachte die scharfkantige Einlaßöffnung ein Ablösen der Strömung und unmittelbar stromab davon ein Wiederanlegen. Es zeigt sich, daß diese Vorgänge einen starken Einfluß auf das thermische Einlaufgebiet haben. Am Punkt des Wiederanlegens der Strömung waren die Wärmeübergangskoeffizienten bei eingebautem Einlaß 4,15 bis 2,4 mal größer als die vergleichbaren Werte bei voll ausgebildeter Strömung über einen Reynoldszahlbereich von 5000 bis 88 000. Die Länge des thermischen Einlaufgebietes wurde durch die vom Einlaß verursachte Strömungsablösung vergrößert. Wird der Rohreinlaß in eine große Wand oder ein Leitblech eingebaut, so werden die Wärmeübergangskoeffizienten in unmittelbarer Nähe des Einlasses geringer als bei einem Rohr mit freiem Einlaß, wobei die Unterschiede im Bereich von 10% liegen.

КОЭФФИЦИЕНТЫ ТЕПЛОПЕРЕНОСА ПРИ ТУРБУЛЕНТНОМ ТЕЧЕНИИ В ТРУБЕ С ИЗОТЕРМИЧЕСКИМИ СТЕНКАМИ ПРИ РАЗЛИЧНОМ РАСПОЛОЖЕНИИ ВХОДНОГО УЧАСТКА

Аннотация—Найдены локальные коэффициенты теплообмена для турбулентного потока в круглой трубе с изотермическими стенками. Входная секция трубы была либо непосредственно соединена с основным участком, либо выдвинута вверх по потоку. Оказалось, что как в том, так и в другом случаях основное влияние на начальный тепловой участок оказывает отрывная область, генерированная острой передней кромкой на входе. В точке присоединения потока значения коэффициента переноса тепла для случаев присоединенной входной секции превышают в 4,15–2,4 раза соответствующие значения для полностью развитого течения в диапазоне чисел Рейнольдса от 5000 до 88 000. Длина начального теплового участка также была увеличена вследствие отрывной зоны на входе. В случае входной секции, присоединенной к трубе, оказалось, что коэффициент теплопереноса в непосредственной близости от входа ниже коэффициента для трубы с выдвинутой входной секцией, причем разница этих коэффициентов находится в пределах 10 %.