Int. J. Heat Mass Transfer. Vol. 36, No. 13, pp. 3343–3349, 1993
Printed in Great Britain

0017-9310/93 $6.00+0.00

Forced-convection heat transfer in the entrance region of pipes

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(*Received* 22 *September* 1992)

Abstract—The steady-state local heat transfer characteristics for air flowing turbulently inside an electrically heated pipe have been determined experimentally over a range of Reynolds numbers from 12 000 to 56 000. A surface-boundary heating condition of uniform heat flux was employed. The entrance configuration of the heat transfer pipe was either a similar pipe of $L/D = 50$, 30, 15; or a sharp edge (i.e. $L/D = 0$). Based on the obtained experimental data, a correlation for predicting the rate of heat transfer was developed.

1. INTRODUCTION

THE TRANSFER of heat to or from fluids flowing turbulently inside pipes is one of the most important modes of industrial heat transfer. For a typical heat exchanger, the fluid inside the circular tube is subjected to an abrupt contraction at the entrance which may cause turbulence in the fluid and alter the developing velocity and temperature profiles at the entrance of the test section. Ideally, both profiles are flat at the entrance. Both velocity and temperature profiles start to develop along the tube simultaneously until they attain a limit at a critical X/D after which they are constant and equal to the fully developed profiles (i.e. from a pipe of infinite length). Usually, the velocity profile develops faster than the temperature profile for liquids (i.e. $Pr \geqslant 1$) [1]. In general, a fully developed flow can be achieved via using calming sections of $L/D \geqslant 50$ [2].

Two main sub-divisions of pipe entrance shapes can be distinguished: (i) those which create abnormal turbulence at the entrance to the heat transfer pipe (e.g. bends, orifices, sharp edges); and (ii) shapes which do not cause abnormal turbulence. The latter include those in which, at the entrance of the heat transfer pipe: (a) the flow is already developed (e.g. a long calming section); and (b) the velocity distribution is uniform (e.g. a bell-mouth).

The average heat transfer coefficient (h_a), from the inner surface of a pipe to a fluid flowing turbulently inside it, can be obtained experimentally using different lengths of the same pipe. This method was employed by early investigators, such as Nusselt [3], to study the effect of pipe length on the average heat transfer coefficient. However, Latzko [4] and Boelter *et al.* [5] introduced an extensive and more accurate technique. This relies on measuring the local heat transfer coefficient (h_x) at any length of the heat transfer pipe (X/D), and then calculating the average coefficient from it.

2. LITERATURE SURVEY

Several experimental, numerical and analytical studies are available, for air in turbulent forced convection inside pipes, to explore the effect of the length and shape of the entrance section. The experimental studies were performed in either the 'uniform wall temperature' condition or the 'uniform heat flux' condition. However, confusion appears to exist between the two conditions.

As far back as 1913, Nusselt [3] drew attention to the effect of length on the average heat transfer coefficient and suggested the following equation:

$$Nu = C\,Re^m\,Pr^n(D/X)^{0.055}. \qquad (1)$$

However, Latzko [4], investigated theoretically the effect of the 'entry length' for a 'uniform wall temperature' condition. This 'thermal entry length' is traditionally defined as the pipe length required to achieve a local value of h_x equal to 1.05-times the asymptotic value of the corresponding h_∞, for a certain thermal boundary condition specified at the surface [6]. Latzko [4] predicted that for X/D greater than 5, the average heat transfer coefficient from zero to L is

$$h_a = h_\infty[1+S(D/X)] \qquad (2)$$

where the dimensionless factor S is a weak function of Re

$$S = 0.067\,Re^{0.25}. \qquad (3)$$

On the other hand, Boelter *et al.* [5] and Grass [7] showed experimentally that the factor S is a constant which differs according to the shape of the entrance section, whereas Al-Arabi [8] correlated several previous results [5, 9–11] and found that S is not constant, but varies with both X/D and Re

$$S/[(X/D)^{0.3}] = (6000/Re)+0.55. \qquad (4)$$

The above equation is valid for air and water at

NOMENCLATURE

C	constant
D	diameter of the pipe [m]
f	dimensionless factor
h	heat transfer coefficient [W m^{-2} K^{-1}]
k	thermal conductivity of air [W m^{-1} K^{-1}]
L	axial length of the pipe [m]
m	constant
n	constant
Nu	Nusselt number, $(hD)/k$
Pr	Prandtl number
Q	power input [W]
Re	Reynolds number, $(UD)/\nu$
S	dimensionless factor
T	steady-state temperature [°C]
U	velocity of the air flowing in the pipe [m s^{-1}]
X	distance from the entrance of the pipe [m]
Z	constant.

Greek symbol

ν	kinematic viscosity [m^2 s^{-1}].

Subscripts

a	average
b	bulk
c	corrected
m	calming section
s	surface
x	local
∞	infinite length.

X/D values between 3 and 13. Values of S at $X/D \geqslant 14$ are the same as those for $X/D = 13$.

It is clear that equation (2) is of a better form. At X/D equal to infinity, equation (1) gives a zero heat transfer coefficient, which is physically wrong, whereas equation (2) gives a coefficient equal to h_∞.

Local heat transfers for fluids flowing turbulently in the entrance region of pipes continued to be of interest, as extensive experimental and theoretical expressions were obtained and predicted, respectively [12–19]. The results were presented in the dimensionless form Nu_x/Nu_∞. A correction for X/D, a function of Re, Pr and the boundary conditions, was proposed [20]. Mikheyev [21] presented his results in the dimensionless form Nu_a/Nu_∞ and Gnielinski [22] recommended the following correlation for the entry length:

$$Nu_a = Nu_\infty[1+(D/L)^{0.667}]. \tag{5}$$

Recently, Ghajar *et al.* [1, 23] studied experimentally forced-convection heat transfer for a wide range of Re and Pr in the entrance and fully developed regions of a circular horizontal electrically heated straight tube fitted with two different configurations (re-entrant and square-edged). An interactive computer program has been developed to calculate the local inside wall temperatures and local peripheral heat transfer coefficients along the tube [24]. The influence of the inlet configuration on the heat transfer coefficient was negligible and the proposed correlation predicted the experimental data with an average absolute deviation of about 4%.

3. SCOPE OF THE PRESENT INVESTIGATION

It is now an established fact that the heat transfer from a pipe to a fluid in turbulent flow inside it depends on the pipe length. A reduction of this length results in an increase of the average heat transfer from the pipe and enables more efficient use of the heat transfer surface. However, no general correlation of the different data is available.

The aim of this study is to (i) explore, via a carefully designed forced-convection heat transfer rig, the effect of employing different 'short-length' calming sections; and (ii) provide a simple general equation for engineers and designers to use. The influence of Prandtl number is not considered in this investigation.

4. THE HEAT TRANSFER RIG

The apparatus used in the present investigation consisted of a horizontal heat transfer copper pipe ($D = 29.1$ mm, $L/D = 61.5$) preceded by either a copper pipe of the same diameter with $L/D = 50$, 30, 15 or a sharp-edge (i.e. $L/D = 0$). Air was driven into the heat transfer pipe through the entrance section, then it was exhausted to the atmosphere via a butterfly control valve. Orifice meters made according to BS 1042 were used to achieve air velocities ranging from 8 to 32 m s^{-1}. A precision micromanometer and a differential manometer were employed to measure the pressure drop across the orifice meter.

In order to estimate the roughness of the heat transfer pipe, the pressure drop along its length was measured and the friction coefficient calculated for several Re. The results were in good agreement with those from the Blasius equation ($\pm 1\%$). Thus, the pipe can be considered smooth.

The heat transfer pipe was electrically heated via main as well as guard heaters. The necessary power inputs were taken from the mains and measured by calibrated voltmeters and ammeters. Both circuits were connected in parallel to a voltage stabiliser. The temperatures at the surface of the pipe were measured by 39 glass-covered copper–constantan thermocouples placed along it. The junctions were soft-soldered to the grooves milled in the surface parallel to the pipe axis. The thermojunctions were arranged to be flush with the inner surface of the pipe, thereby avoiding tripping the boundary layer airflows within the pipe. A layer of insulation of 5 mm thick was employed to cover the pipe. The wire of the main heater was then wound uniformally along the pipe (i.e. to ensure a uniform heat flux). It was electrically insulated by means of ceramic beads, and thermally insulated by a 20 mm thick layer of insulation. In this layer, four pairs of thermocouples were fitted: each pair opposite to each other. Similarly, an electrically insulated guard heater was wound uniformly on the thick layer of insulation and then covered with another layer of insulation of 10 mm thick (see Fig. 1). In addition, to make it possible to establish the heat transmitted to the pipe between any two cross sections, the main heater was tapped at various points so that a voltmeter could be connected across any two points.

For a certain input to the main heater, the input to the guard heater was adjusted until there was no temperature gradient across the insulation layer, as shown by the readings of the pairs of thermocouples fitted along the pipe (see Fig. 1). The power to the guard heaters was controlled separately at various points along the length of the test section, thus, all the heat generated by the main heater was then going to the surface of the heat transfer pipe.

The calibrated thermocouple wires were insulated by means of ceramic beads. They were connected, through ribbon cables, to a selector switch and then to a data-logger used to measure the temperatures each 30 min until the steady-state condition was attained. The reference junction was kept in melting ice and a potentiometer was employed for the final readings.

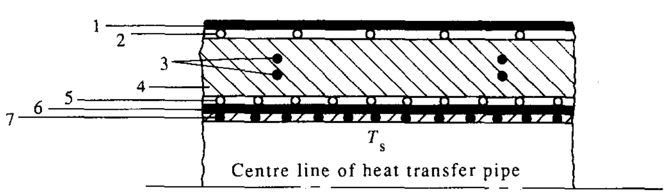
The 'short pipe' fitted directly to the heat transfer pipe will result in the loss of some heat from the latter. This end-conduction was virtually eliminated by thermally insulating the pipe at the ends by teak-wood connection pieces made to cause negligible resistance to the flow. The temperature of air at the inlet of the heat transfer pipe was measured by means of a thermocouple fed through a very small hole in the teak-wood connection piece, opened only at the time of taking the final measurements (i.e. at the steady-state condition). A mixing chamber was fitted at the end of the heat transfer pipe in order to measure the 'mixed mean' exit temperature of air.

5. THE OBSERVATIONS

A 'long calming' section is defined as an inlet pipe of the same diameter as the heat transfer pipe and of length sufficient to ensure that fully developed fluid flow exists at the entrance of the heat transfer pipe. However, if the calming section is shorter than this, a different condition will exist. Such a condition at the entrance of the heat transfer pipe will be related to how short (i.e. the actual length) the calming pipe is.

The variation of the steady-state surface temperature with X/D for the heat transfer pipe had the same characteristics for all the 'short' calming sections tested (see Fig. 2 for a selected run). In addition, local and average heat transfer coefficients were evaluated. They followed the same pattern as those from a heat transfer pipe preceded by a 'long calming' section [25]. The heat transfer coefficients decreased gradually until a limit was attained at the 'thermal entry length', after which they were constant (see Fig. 3 for a selected run). This 'thermal entry length' was equal to $X/D = 12$, and was independent of the magnitude of Re. By employing equation (2), the experimental values of the dimensionless factor S were determined.

The value of the local Re varied along the heat transfer pipe as a result of the change in the physical properties due to the variation in the surface temperature of the pipe. Thus, all local Re were corrected accordingly. At the critical length of $X/D = 12$, the S values were plotted against the corrected local Re for the different calming sections tested as well as the 'long calming' section (see Fig. 4). After a $Re_{x,c}$ of 48 000, the variation of S became very small. For practical purposes, this can be considered to be constant. In addition, the S values decreased gradually as the calming section increased until a critical L/D was



1,4,6 – Insulation layers
2 – Guard heater
3 – Pair of thermocouples
5 – Main heater
7 – Thermojunctions in milled grooves

FIG. 1. Schematic representation of a section through the considered heat transfer pipe.

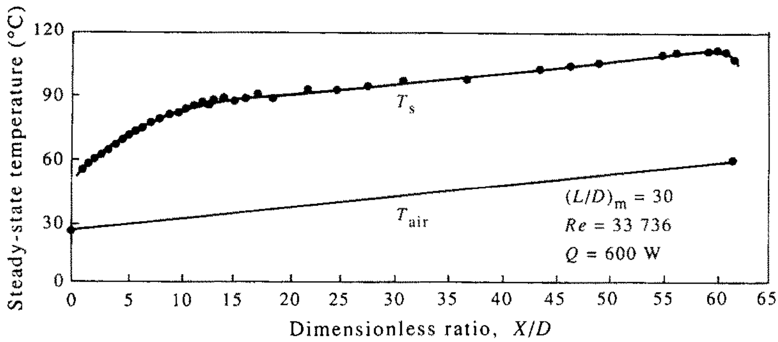


FIG. 2. Steady-state surface and air temperature distributions along the heat transfer pipe.

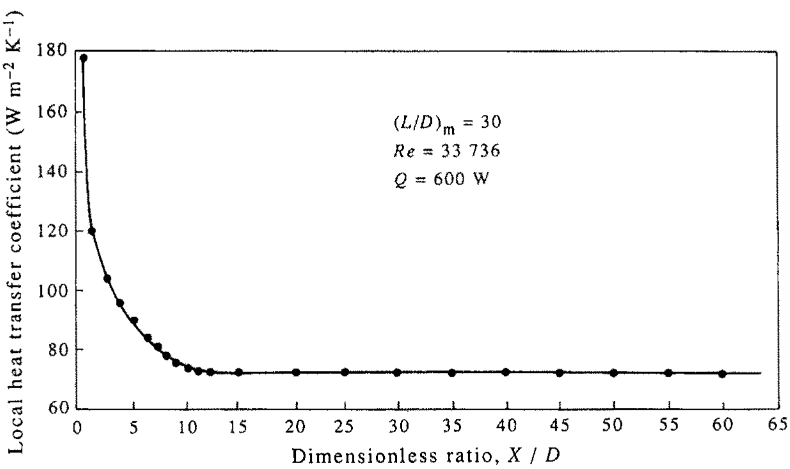


FIG. 3. Variation of the local heat transfer coefficient vs the dimensionless ratio X/D.

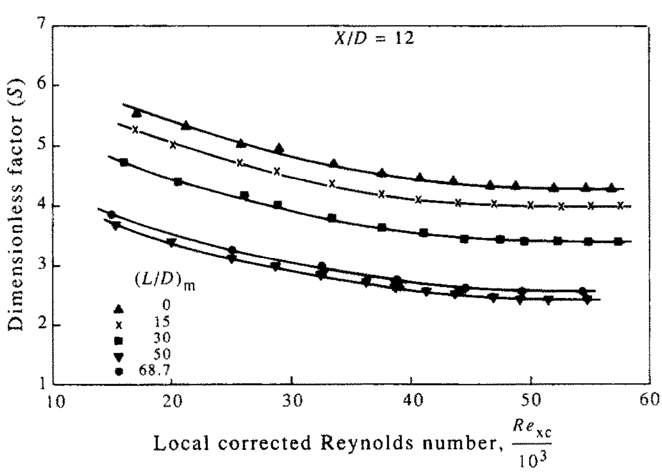


FIG. 4. Variations of the dimensionless factor S vs the corrected local Reynolds number.

reached between 55 and 60, after which the S values became constant. Therefore, a minimum value of $(L/D)_m = 60$ is needed to ensure that fully developed flow exists at the entrance of the heat transfer pipe.

Assuming that [26]

$$Re_{x,c}/Re_x = (T_s/T_b)^f \tag{6}$$

the values of f were evaluated for the 'short calming' sections being tested and a 'long calming' section, i.e. $(L/D)_m = 68.7$ [25]. However, these values were dependent on X/D (see Table 1).

A logarithmic plot for $Nu_x/Pr_x^{0.4}$ vs Re_x, at 'thermal entry lengths' $\geqslant 12$, was drawn (see Fig. 5). The experimental results were represented by straight lines, and

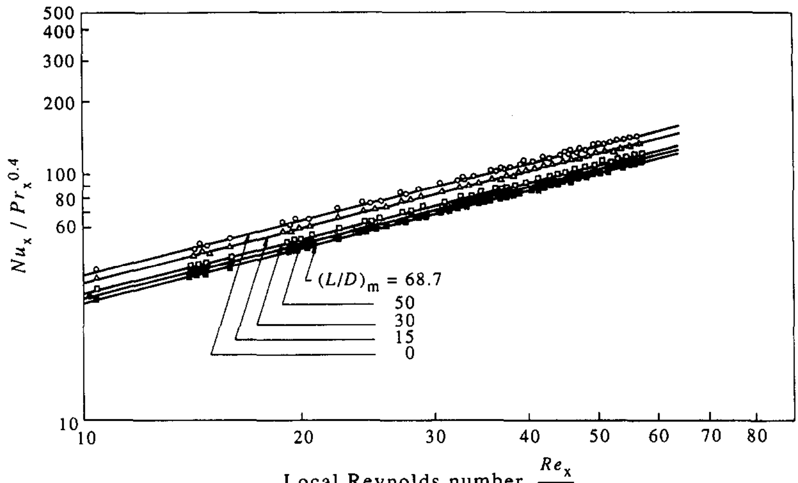


FIG. 5. Variations of the local Nusselt number for the 'short calming' sections under consideration.

Table 1.

f	$(L/D)_m$
$-2.210\times10^{-3}+0.031(X/D)$	0
$4.664\times10^{-3}+0.028(X/D)$	15
$-12.090\times10^{-3}+0.025(X/D)$	30
$-25.824\times10^{-3}+0.025(X/D)$	50
$2.846\times10^{-3}+0.024(X/D)$	68.7

an equation was evolved over a range of Re between 12 000 and 56 000. This agreed quantitatively with that for the 'long calming' section, i.e. $(L/D)_m = 68.7$ [25]

$$Nu_x = Z\,Re_x^{0.81}\,Pr_x^{0.4} \tag{7}$$

where

Table 2.

Z	$(L/D)_m$
0.0228	0
0.0212	15
0.0189	30
0.0183	50
0.0175	68.7

In order to formulate a simple general equation for the heat transfer from a heated pipe to air flowing turbulently inside it, the values of Z were plotted against their corresponding calming sections. The following equation was deduced:

$$Nu_x = 1.92\times10^{10}\,Re_x^{0.81}\,Pr_x^{0.4}[(L/D)_m+1000]^{-3.978}. \tag{8}$$

Higher values of Re and higher heat fluxes are needed to extend the range covered by this correlation. Experiments with fluids other than air are also required to investigate the effect of Prandtl number.

A comparison between the current values of the dimensionless factor S, for a 'long calming' section, and the theoretical predictions as well as the experimental measurements obtained by previous investigators is shown in Fig. 6. However, their data did not allow the calculation of the corrected local Re, thus, all S values were plotted vs the mean Re.

Due to (i) the simplified assumptions made in the analysis; and (ii) the unensured fully developed flow at the entrance of the heat transfer pipe, the predicted and measured values of S in Fig. 6 were lower than those of the current study. The predictions of Deissler [12] decreased with Re, whereas those of Sparrow *et al.* [13], Siegel and Sparrow [14] and Reynolds *et al.* [17] were almost independent of Re. On the other hand, the correlation of Al-Arabi [8] was based on data obtained at both the uniform heat flux and uniform wall temperature conditions for air and water. Of previous experimental studies, only those of Mills [15] and Depew [16] were performed at a uniform heat flux condition. However, a fully developed flow was not achieved as they employed a 'long calming' section of $(L/D)_m = 33$ and 30, respectively. The condition at which the experiment of Mikheyev [21] was carried out was not given. However, it was included in the comparison.

For a 'short calming' section, a comparison between the current results and those of previous investigators is not possible, as not enough data are available. For instance, the experimental studies of Boelter *et al.* [5] and Mills [15] employed only one calming section of $L/D = 2.8$ (at a uniform wall temperature condition) and $L/D = 3$, respectively.

6. MEASUREMENT UNCERTAINTIES

Extra care was taken in constructing the heat transfer rig as well as in measuring the temperatures and the electrical power supplied. A potentiometer capable of

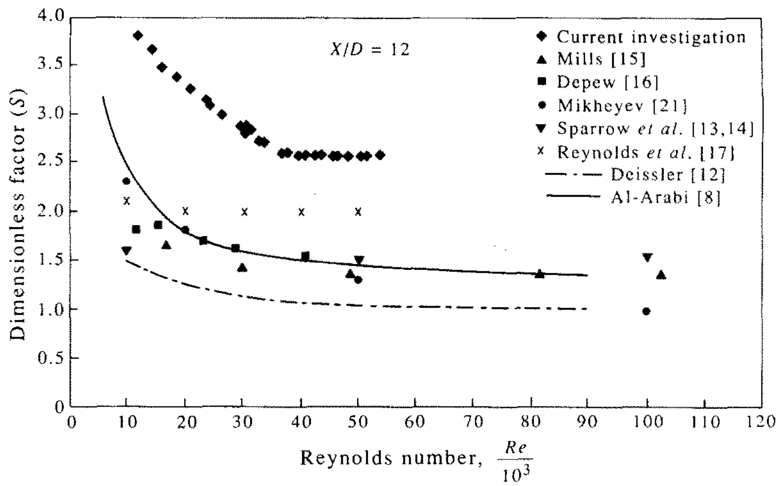


FIG. 6. Comparison between the measured dimensionless factor S and those from previous investigations for a 'long calming' section (i.e. $L/D = 68.7$).

reading to 0.001 mV was used. However, the fitted curve (see Fig. 2) for the temperature distribution, along the surface of the heat transfer pipe, was within $\pm 1.5\%$ of the experimental data. The scatter may be attributed to the relative locations (i.e. the traces) of the heater wires on the copper pipe, which were detectable by the thermocouples. Nevertheless, this was kept to the minimum by winding the wires on an insulation layer as well as insulating all the thermocouple wires with ceramic beads. The experimental uncertainties were estimated according to Coleman and Steele [27]. Typical uncertainties for the Nusselt number were about 3% at $X/D \leqslant 10$ decreasing to 2% at $X/D \geqslant 11$.

7. CONCLUSIONS

The effect of employing 'short calming' sections on the steady-state rate of forced-convection heat transfer from a hot pipe to air flowing turbulently inside it has been investigated experimentally. A simple general correlation has been developed to predict the rate of heat transfer over a range of Reynolds number from 12 000 to 56 000. The results were in good agreement qualitatively with those from previous studies.

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