AVERAGE HEAT TRANSFER COEFFICIENT IN RECTANGULAR DUCTS WITH BAFFLE BLOCKAGES

M. Molki and A. R. Mostoufizadeh

Department of Mechanical Engineering
Esfahan University of Technology, Esfahan, Iran

Received July 1988

Abstract An experimental investigation was conducted to study the average and the fully–developed heat (mass) transfer coefficients in a rectangular smooth duct and a duct with repeated–baffle blockages. The focus of attention in this work is the conventional correlation $Nu/Nu_{fd} = 1 + C/(X/D)$ for the average heat transfer coefficient. It was shown that for relatively short ducts, the coefficient C is not constant but, in general, it depends on the length of the duct. The experiments were carried out via a mass transfer technique and the analogy between heat and mass transfer was employed to predict the heat transfer coefficients. The flow Reynolds number ranged from 3000 to 50,000 with the height of the baffles equal to h/H = 0, 0.125, 0.25, and 0.50.

چکیده ضریب انتقال حرارت (جرم) در یک مجرای مستطیل شکل صاف و مجرائی با موانع پره‌ای متعدد بصورت تجربی مورد بررسی قرار گرفت. تاکید این بررسی بر رابطهٔ رایج ضریب انتقال حرارت متوسط $\overline{Nu}/Nu_{fd} = 1 + C/(X/D)$ قرار دارد. نشان داده شده که برای مجاری نسبتا" کوتاه، ضریب C ثابت نمی‌باشد، بلکه در حالت کلی به طول مجرا بستگی دارد. آزمایشات بکمک یک روش انتقال جرم انجام گردید و با استفاده از تشابه میان انتقال جرم و حرارت، ضرایب انتقال حرارت پیش‌بینی شدند. در این پژوهش ارتفاع پره‌ها برابر با ۰، ۰/۱۲۵، ۰/۲۵، ۰/۵۰ بوده و عدد رینولدز جریان بین ۳۰۰۰ تا ۵۰۰۰۰ متغیر بوده است.

Heat transfer characteristics of rectangular ducts are employed in the thermal design of many devices where heat transfer occurs. For example, the internal flow passages in a modern air-cooled turbine blade are sometimes modeled as a single straight or a multipass rectangular duct [1]. In this situation, the flow passages are usually short and, therefore, the heat transfer in the thermally developing region of the passage becomes important.

The present investigation is an experimental study of the average heat transfer coefficient in the combined thermal-hydraulic entrance region of a rectangular duct with repeated-baffle blockages. It should be noted that although a knowledge of the local heat transfer coefficient might be of great value in a detailed study of a heat exchange device, it is the average values that are important to the design engineer.

The average heat transfer coefficient for turbulent flow in circular tubes has been discussed in many heat transfer textbooks [2, 3], and it has been correlated as

$$Nu/Nu_{fd} = 1 + C/(X/D) \qquad (1)$$

where Nu and Nu_{fd}, respectively, denote the average and fully developed Nusselt numbers, X/D is the normalized axial coordinate, and C is a constant which depends on the inlet geometry (e.g., C = 6 for sharp-edged inlet [2, 3]). Note that Equation 1 is valid when the length of the tube is about 20 diameters or more.

In a paper by Al-Arabi [4], a large body of heat transfer data were employed to obtain a new correlation for the average heat transfer coefficient in tubes, which is purported to be valid for $X/D > 3$. In fact, he showed that Equation 1 still applies if the constant C is considered to be a function of X/D, namely, $C = 1.683\ (X/D)^{0.423}$. Later, other investigators [5] indicated that for relatively short tubes ($X/D > 2$), C is not only a function of the axial corrdinate X/D, but also depends on on the flow Reynolds number.

In the present investigation an attempt has been made to apply Equation 1 to rectangular ducts. As it will be seen later, again C, in general, is not a constant and it varies with distance from the duct inlet.

THE EXPERIMENTAL APPARATUS

The experiments were carried out via a mass transfer method known as the naphthalene sublimation technique. The mass transfer results obtained by this technique can subsequently be converted to the heat transfer results through the well-known analogy between the heat and mass transfer processes. The theoretical basis for the analogy is discussed by Eckert in Reference 6. In this regard, the terms heat and mass transfer are used interchangeably throughout this paper.

The experimental apparatus is shown schematically in Figure 1. As seen, the main components of the apparatus are the test section, the orifice plate, the control valve, and the centrifugal fan. The fan operated in the suction mode to draw the laboratory air into the test section, where mass transfer occurred, and to discharge it to the outdoors.

The test section was a rectangular duct with a number of baffle blockages installed on the upper and lower walls of it in a staggered manner. As pictured in Figure 2a, the test section had a modular design, and a typical assembly of modules, which forms the wall of the duct, is seen in this Figure. Each module consists of a metallic base and layer of solid naphthalene coating (3 mm thick) which was formed through a casting process (Figure 2b). The dimensions of each cast module were W = 6.0 cm (same as the width of the duct) and L = 1.5 cm.

The baffles were fabricated from 0.75 mm galvanized sheet metal. Each baffle was placed between a pair of adjacent modules to form the internal blockages of the test section duct. In this study, three different baffle sizes were

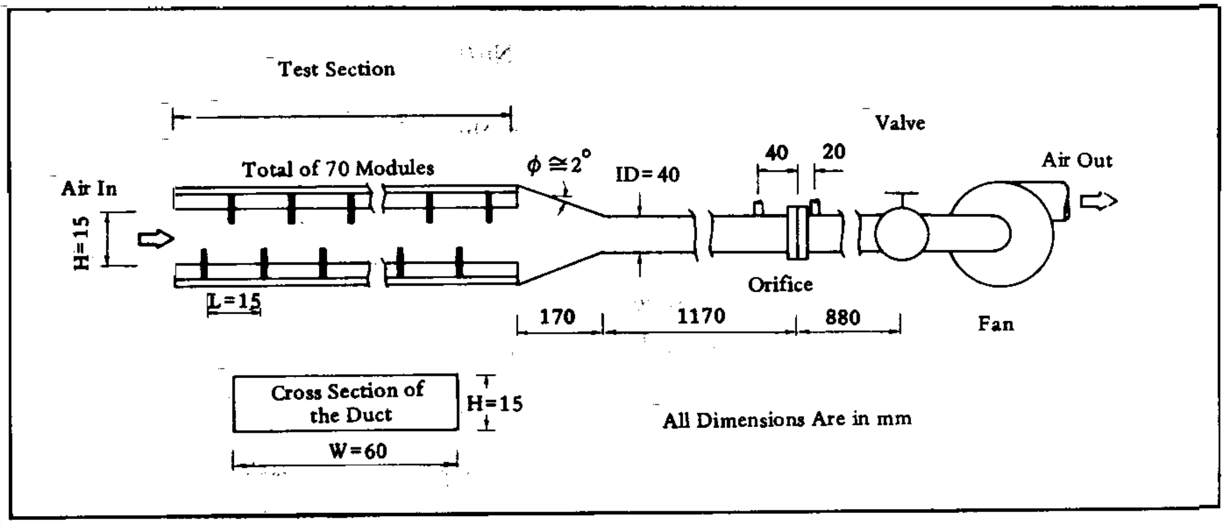

Figure 1. Schematic view of the experimental apparatus

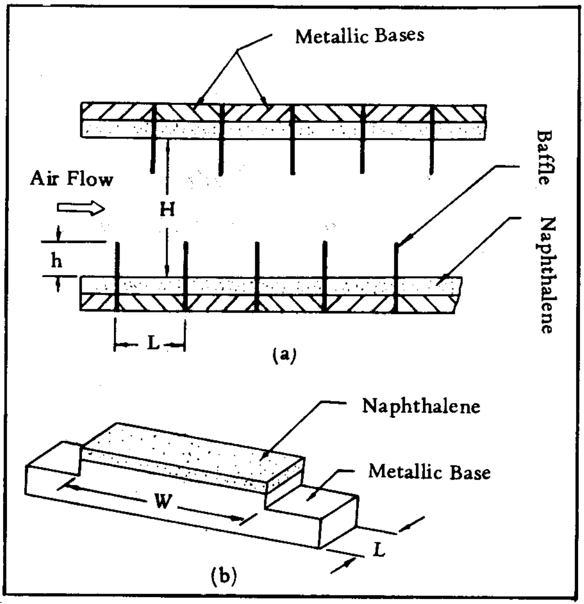

Figure 2. Schematic view of (a) test section, (b) a typical module cast with naphthalene.

employed, namely, $h/H = 0.125$, 0.25, and 0.50, with a fixed vertical interwall distance of $H = 1.5$ cm.

With regard to the boundary conditions, it should be mentioned that the upper and the lower walls of the duct were covered with solid naphthalene and were kept at uniform temperature. Therefore, the mass transfer model considered here is analogous to a heat transfer problem in which the walls are ideally isothermal. However, the surface of the baffles was not covered with naphthalene, and the two components that contribute to the heat transfer enhancement simultaneously, i.e., additional surface area and improved mixing (due to altered flow field), were intentionally separated and only the latter component was considered in this study. Transfer characteristics of the wall-attached baffles can be found in literature (e.g., see reference 17).

During each data run, the air temperature was recorded several times by a mercury thermometer with a resolution of 0.25 C. The naphthalene vapor pressure and other properties were then evaluated at the arithmetic mean of the recorded temperature. To determine the rate of mass sublimation at each axial station along the duct, the modules were individually weighed on a Sartorius balance before and after each data run, and the mass changes were determined to within 0.1 mg. Mass changes were typically around 15 mg.

Next, we shall discuss the data reduction and will show how the average heat (mass) transfer coefficients are determined.

DATA REDUCTION

The average mass transfer coefficient K_x was evaluated from

$$K_x = (M_x/A_x)/(\text{LMDD}) \tag{2}$$

where M_x is the rate of mass sublimation for the length X, which is directly obtainable from the experimental data, A_x is the mass transfer surface area, and LMDD is the logarithmic mean density difference defined as

$$\text{LMDD} = [(\rho_{nw} - \rho_{nb,0}) - (\rho_{nw} - \rho_{nb,x})] / \ln[(\rho_{nw} - \rho_{nb,0})/(\rho_{nw} - \rho_{nb,x})] \tag{3}$$

Therefore, the coefficient K_x evaluated from Equation 2 applies to the length of the test section duct from the inlet X = 0 to X = X.

In Equation 3, the quantities ρ_{nw} and ρ_{nb} are the naphthalene vapor densities at the duct wall and in the bulk of the fluid, respectively. The subscript 0 refers to the duct inlet and x refers to quantities evaluated at a distance X from the inlet. Since the laboratory air entering the duct was free of naphthalene vapor, $\rho_{nb,0}$ is zero and Equation 3 simplified to

$$\text{LMDD} = \rho_{nb,x} / \ln[\rho_{nw}/(\rho_{nw} - \rho_{nb,x})] \tag{4}$$

To determine ρ_{nw}, we used the vapor pressure-temperature correlation suggested by Sogin [8]

$$\log_{10}P_{nw} = 13.564 - \frac{3729.4}{T} \qquad (5)$$

together with the ideal gas equation

$$\rho_{nw} = \frac{P_{nw}}{RT} \qquad (6)$$

In Equation 5, the temperature T is in degree kelvin and the vapor pressure P_{nw} is in Pascal.

The naphthalene vapor density in the bulk of the fluid was evaluated from the following expression which was developed from a mass balance performed on a control volume that encompassed the portion of the duct from the inlet up to the xth module

$$\rho_{nb,x} = (1.5m_1 + 2\sum_{i=2}^{x-1} m_i + 1.5m_x)/Q_x \qquad (7)$$

where m_1, m_i, and m_x are the rates of mass sublimation for the 1st, the ith, and the xth modules, respectively.

Equation 7 is valid for the subscript x being greater or equal to 3. For the 1st and 2nd modules we have

$$\rho_{nb,1} = m_1/Q_1 \qquad (8)$$

$$\rho_{nb,2} = 1.5\,(m_1 + m_2)/Q_2 \qquad (9)$$

In Equations 7 to 9, the quantities Q_1, Q_2, ..., and Q_x represent the volumetric air flow rates at the 1st, 2nd, ..., and xth module of the test section. It should be noted that when the baffles were installed in the duct, the pressure drop along the flow was large and it was necessary to perform the calculations using the local volumetric air flow rates.

Having determined the average mass transfer coefficient K_x, we subsequently nondimensionalized this coefficient through the conventional definition of Sherwood number.

$$Sh = K_x D/\mathcal{D} \qquad (10)$$

In this equation, D is the hydraulic diameter of the duct ($D = 2WH/(W + H)$), and $\mathcal{D}$ is the mass diffusion coefficient given as $\mathcal{D} = \nu/2.5$ for diffusion of naphthalene in air [8].

In the next Section, the fully developed Sherwood numbers Sh_{fd} will be discussed first. The evaluation of Sh_{fd} is based on the local mass transfer coefficient K_∞ very far from the inlet, defined as

$$K_\infty = m_\infty/A_\infty \Delta\rho_{n,\infty}$$

where

$$\Delta\rho_{n,\infty} = \rho_{nw} - \rho_{nb,\infty}$$

In Equations 11 and 12, the subscript ∞ refers to the modules located very far from the inlet where the fully developed condition has been established.

For more details regarding the calculation of Sh_{fd} and the actual data recorded druing each run, the interested reader is referred to reference [9].

RESULTS AND DISCUSSION

The fully-developed Sherwood numbers Sh_{fd} for the smooth duct, i.e. with no baffles corresponding to $h/H = 0$, are presented in Figure 3. The abscissa is the Reynolds number Re based on the hydraulic diameter of the duct. Also shown in this figure are the well-known Petukhov-Popov correlation [2] and its modified version by Gnielinski [10]. It should be mentioned that these two correlations are heat transfer expression (i.e., in terms of Nusselt number) which have been converted to mass transfer via the application of the analogy between the two processes [6]. Clearly, the close agreement seen in this Figure is well supportive of the present experimental approach.

The periodic fully-developed results for the duct with repeated-baffle blockages are shown on the upper portion of Figure 3, and h/H appears as a parameter. These results show that as soon as the smallest blockage is introduced into the duct (i.e., h/H = 0.125), the Sherwood number is increased. The reason is that in turbulent flow, a large portion of the thermal (mass transfer) resistance lies in the laminar sublayer (wall layer), and the introduction of any solid protuberance at the wall would disturb and break this layer, resulting in higher transfer coefficients. As the size of the baffles is increased, the laminar sublayer is completely removed and a recirculating zone appears behind the baffles which has an additional enhancing effect on the transfer coefficients. The presence of these recirculating zones has been shown by Berner et al. [11] and Kelkar & Patankar [12]. They have also shown that at higher Reynolds number, the size of the recirculating zones is affected and that vortices are shed from the tip of the baffles. All these activities tend to increase the turbulence in the air stream and thus increasing the heat (mass) transfer coefficients.

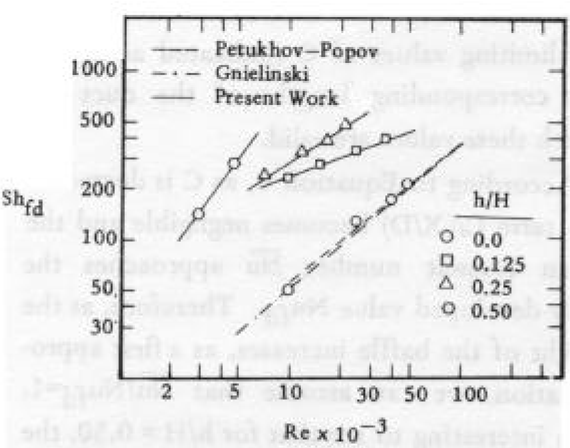

Figure 3. The fully-developed coefficients

Attention is next turned to Figure 4 where the average transfer coefficients are plotted for the smooth duct. In the lower diagram of the Figure, the ordinate is the average Sherwood number $\overline{Sh}$ for the length X of the duct divided by the fully developed Sherwood number Sh_{fd}. According to the analogy between heat and mass transfer, the ratios $\overline{Sh}/Sh_{fd}$ and $\overline{Nu}/Nu_{fd}$ are identical.

As seen, for shorter ducts, namely, for smaller values of X/D, the average transfer coefficients are smaller. These observations are readily rationalized by noting that the local transfer coefficients are larger near the inlet, and they are smaller farther downstream.

With the aid of Equation 1 and using the experimental data of the present study, the quantity C was evaluated at various lengths of the duct X/D, and is plotted in the upper diagram of Figure 4. It is clearly seen that for relatively short ducts, C is not a constant and increases with X/D. However, as X/D is increased, the value of C appears to asymptotically approach the value of $\overline{C}$ = 4.29 (Table 1). In this regard, it might be said that if the length of the smooth duct X/D is greater than 6.31, Equation 1 could be used to predict the average transfer coefficients with the constant value of C = $\overline{C}$ = 4.29. It should be noted that the fully-developed values of Nu_{fd}

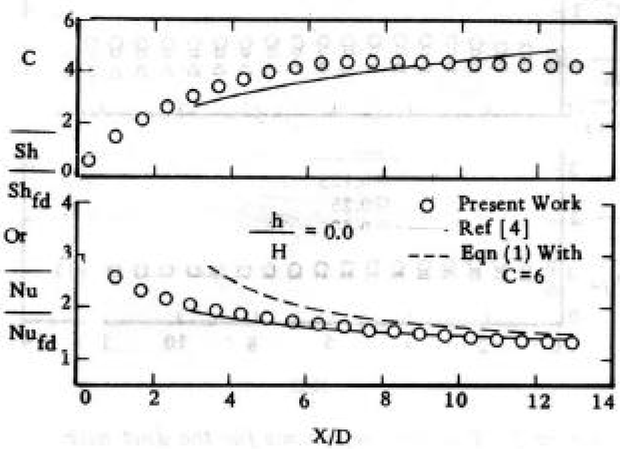

Figure 4. Transfer coefficients for the smooth duct

(or Sh_{fd}) in Equation 1 are obtained from Figure 3.

Also shown in Figure 4 are the results reported by al-Arabi [4] and Equation 1 whith C = 6, both of which are for circular tubes with a sharp-edged inlet. It is seen that the Al-Arabi's correlation is nearly valid for rectangualr ducts provided that the diameter of the tube is replaced by the hydraulic diameter of the duct. However, this correlation is limited to $X/D \geqslant 3$, while the results of the present work extend to $X/D \geqslant 3$, while the results of the present work extend to $X/D \geqslant 0.31$. Moreover, equation (1) with C=6 is not applicable to rectangular ducts.

Before moving on to the next figure, it is noteworthy that in the experiments performed on the smooth duct, the Reynolds number ranged from 10,000 to 50,000 (Table 1). However, since in this range the effect of Re on $\overline{Sh}/Sh_{fd}$ was found to be small, the values of $\overline{Sh}/Sh_{fd}$ at the respective locations (i.e., X/D) were averaged, and thus the data points seen in Figure 4 were obtained.

The results of the experiments performed on the rectangular duct with repeated-baffle blockages are shown in Figure 5, where the height of the baffles h/H appears as a parameter. The lower diagram in this figure clearly

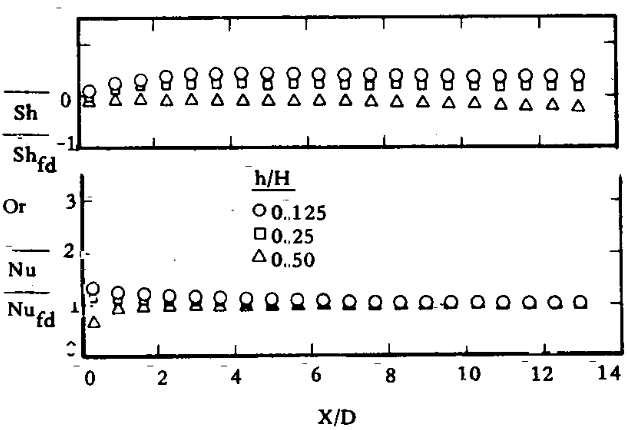

Figure 5. Transfer coefficients for the duct with repeated-baffle blockages.

Table 1. The ranges of Reynolds number and the limiting values of C

h/H	Re	$\overline{C}$	
0	10,000-50,000	4.29 for X/D ⩾6.31	
0.125	10,000-37,600	0.39	2.31
0.25	7000-21,600	0.20	1.65
0.50	3000-4800	−0.13	0.31

shows that the length of the duct required for the flow to reach a fully-developed condition is much shorter than that for the smooth duct. In fact, the presence of the baffles throughout the duct promotes turbulence, and the resulting additional mixing effect enhances the exchange of momentum and mass (or heat in the equivalent heat transfer problem) between fluid particles, thus reducing the entrance length.

The upper diagram in this figure shows the C values which have been evaluated from the present experimental data through equation (1). It is clearly seen that C is almost constant for various lengths of the duct X/D. However, comparing the three sets of data points shown in this diagram, it is observed that the small variations of C decreases as h/H is increased. Moreover, the values of C are much smaller than those of the smooth duct. Table 1 shows the limiting values of C (indicated as $\overline{C}$) and the corresponding lengths of the duct for which these values are valid.

According to Equation 1, as C is decreased the term C/(X/D) becomes negligible and the mean Nusselt number $\overline{Nu}$ approaches the fully-developed value Nu_{fd}. Therefore, as the height of the baffle increases, as a first approximation, we can assume that $\overline{Nu}/Nu_{fd}=1$. It is interesting to see that for h/H = 0.50, the C's are even negative, implying that the enhanced turbulence caused by the presence of the baffles has increased the fully-developed transfer coefficients to such a large extent that $\overline{Nu}/Nu_{fd}$ is less than unity. These conclusions are restricted to the range of Re shown in Table 1.

CONCLUSIONS

The present experimental study has revealed the average and the fully-developed transfer characteristics of a rectangular duct with repeated-baffle blockages. The height of the baffles ranged from h/H = 0, corresponding to the smooth duct, to 0.50, and the Reynolds number was varied between 3000 to 50,000.

In the case of the smooth duct, it was found that the conventional circular tube correlation $\overline{Nu}/Nu_{fd} = 1 + C/(X/D)$ with C = 6 does not predict the correct average transfer coefficients of the rectangualr duct. In general, C depends on the length of the duct X/D, and for a given value of X/D, the corresponding value of C should be used (Figure 4) in order to make a correct prediction. Moreover, as X/D was increased, C approached the limiting value of 4.29 and not to the circular tube value of 6.

The presence of baffles in the duct reduced the length of the entrance region, resulting in nearly constant C's. In addition, as h/H was increased, the values of C decreased. For the special case of h/H = 0.50, the values of C decreased. For the special case of h/H = 0.50, the C's appeared to be negative (Table 1).

In order to use the results of the present investigation to predict the average coefficient $\overline{Nu}$ (or $\overline{Sh}$) for a relatively short rectangular duct (smooth or with baffles), if the length of the duct X/D is greater tahn 6.31, 2.31, 1.65, and 0.31 corresponding, respectively, to h/H = 0, 0.125, 0.25, and 0.50, the constat values of $C = \overline{C}$ as given in Table 1 together with Equation 1 should be used. If the length of the respective ducts are shorter, the local C's as given in Figure 1 and 2 are recommended. In any case, the total length of the duct should not be less than 0.31 hydraulic diameter.

NOMENCLATURE

A_x	mass transfer surface area for the the length X of the duct
C	coefficient in Equation 1
$\overline{C}$	asymptotic value of C at large X
D	hydraulic diameter
	diffusion coefficient
h	height of baffles
H	vertical interwall distance
$\overline{K}_x$	average mass transfer coefficient for the length X of the duct
K_∞	mass transfer coefficient very far from the inlet
L	axial length of a test section module
LMDD	logarithmic mean density difference, Equation 3
m	rate of mass sublimation from a module
M_x	rate of mass sublimation for the length X
$\overline{Nu}$	average Nusselt number over the length X of the duct
Nu_{fd}	fully-developed Nusselt number
P_{nw}	naphthalene vapor pressure at the wall
Q	volumetric air flow rate
R	gas constant for naphthalene vapor
Re	Reynolds number
$\overline{Sh}$	average Sherwood number over the length X of the duct
Sh_{fd}	fully-developed Sherwood number
T	temperature
W	duct width
X	axial coordiante (X=0 corresponds to the duct inlet)

Greek Symbols

$\Delta\rho_{n,\infty}$ wall-to-bulk difference in naphthalene vapor density far from the inlet, Equation 12.

ν kinematic viscosity

$\rho_{nb,\infty}$ naphthalene vapor density in bulk far from the inlet

$\rho_{nb,0}$ naphthalene vapor density in bulk at the duct inlet

$\rho_{nb,x}$ naphthalene vapor density in bulk at station X

ρ_{nw} naphthalene vapor density at wall

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