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Heat transfer in turbulent decaying swirl flow in a circular pipe

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Abstract—Heat transfer coefficients for air are measured along a heated pipe for decaying swirl flow, generated by radial blade cascade. The results are compared with an expression proposed for predicting the heat transfer coefficients in swirling flow. The theoretical predictions are in good agreement with the experimental data with average and maximum deviations of 7 and 11%, respectively. The application of the theoretical approach to the experimental results obtained by other investigators for heat transfer in a decaying swirl flow generated by short-twisted tapes and tangential slots at inlet also give rise to encouraging agreement.

1. INTRODUCTION

PROBLEMS of heat transfer of decaying swirl pipe flow are of practical importance in designing equipment like combustion chambers and heat transfer promoters in different kinds of heat exchangers.

In spite of the present day significance of such problems and a number of experimental and a few theoretical studies reported in the literature, there is still no generalized method to predict heat transfer for decaying swirl flow that can be applied to different swirl generators based on the known inlet flow conditions.

Many investigations have been conducted to determine the heat transfer characteristics in decaying swirl flow experimentally. The effect of swirl intensity on heat transfer was studied experimentally by a number of investigators [1–5] who also present empirical correlations for heat transfer coefficients expressed as a function of the swirl number, in addition to some other parameters.

Reference [6] presented an analytical study of the heat transfer characteristics in decaying turbulent swirl flow in a pipe. However, these equations were developed to predict the heat transfer assuming the flow to be a rotating slug which is the case for the swirl flow generated by a short-twisted tape. Such an assumption was used to determine the magnitude of the tangential velocity at the wall of the tube.

The aim of this paper is to establish an approach to compute the heat transfer in decaying swirl flow generated in a pipe with the help of any swirl generator.

2. EXPERIMENTAL SETUP

Figure 1 shows the schematic diagram of the experimental setup used in this investigation. Air was supplied to the setup by a centrifugal blower, 1, driven by an induction motor. The passage of the blower was provided with a bell-mouth, 2, and a throttling disc, 3, by means of which the flow rate was regulated.

A calibrated orificemeter, 4, designed and fabricated in accordance with ISO specifications, was used to measure the flow rate. A conical diffuser, 5, led the air stream from the orificemeter into a flow stabilizer, 6, at a considerably reduced velocity. The stabilizer was provided with one honeycomb, 7, and two wire-mesh screens, 8, and the fluid stream was directed into a radially inward-flow passage, 9, fitted with adjustable blades to produce different swirl intensities. The radial passage guided the air to the test pipe through a bell-mouth.

The test pipe, 10, was prepared from a commercial smooth seamless steel pipe of 71 mm i.d. and 90 mm o.d. having a heated length of 60 pipe diameters. Heating was achieved by winding continuously a flexible electrically insulated heating cord around the pipe which provided a constant heat flux boundary condition. The heating element was connected to a 230 VAC supply through a voltage stabilizer which maintained the supply voltage constant within $\pm 1\%$. An autotransformer was used to provide a constant input power for all the runs. The test pipe was insulated with glasswool of 50 mm radial thickness. A mixing chamber was used at the exit of the test section to measure the outlet bulk temperature of air.

The wall temperatures were measured by means of Degussa copper–constantan thermocouples at seven axial stations, located at distances of 1.5, 3, 5, 9, 16, 30 and 51.5 pipe diameters from the inlet of the test section. Each axial station had three thermocouples located at equispaced circumferential positions for obtaining average wall temperature corresponding to the axial station. The thermocouple junctions were spring loaded and embedded in radial holes in the pipe wall.

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NOMENCLATURE

A	coefficient, equation (16)
A_m	maximum moment of velocity at inlet, YW
a	coefficient, equation (3)
B	coefficient, equation (16)
b	coefficient, equation (3)
C_p	specific heat at constant pressure
D	pipe diameter
$F(Y)$	initial condition function, equation (13)
Gr	Grashof number, $2W_w^2\,Re^2\,\beta\Delta T$
H	tape twist ratio—the number of tube diameters per 180° twist
h	heat transfer coefficient
J_0, J_1	Bessel functions of the first kind of orders zero and one, respectively
k	thermal conductivity
m	constant, equation (15)
Pr	Prandtl number, $\mu C_p/k$
Nu	Nusselt number, hD/k
n	coefficient, equation (3)
R	pipe radius
Re	Reynolds number, $u_{av}D/\nu$
r	radial coordinate
S_n	swirl number
T	temperature
ΔT	difference between wall and fluid temperatures, T_w–T_b
U	non-dimensional axial velocity, u/u_{av}
u	axial velocity
u_{av}	average axial velocity
v_s	resultant swirl velocity, $(u_{av}^2+w^2)^{1/2}$
W	non-dimensional tangential velocity, w/u_{av}
X	non-dimensional axial coordinate, x/D
x	axial coordinate
Y	non-dimensional radial coordinate, r/R
Z	non-dimensional parameter, $4(1+\varepsilon/\nu)\,(x/D)/Re$.

Greek symbols

α	velocity ratio factor, v_s/u_{av}
β	volumetric coefficient of thermal expansion
ε	kinematic eddy viscosity
λ_n	eigenvalues, $n = 1, 2, 3, \ldots$
μ	dynamic viscosity
ν	kinematic viscosity
ρ	density.

Subscripts

a	axial
b	bulk
cc	centrifugal convection
d	decaying swirl
in	inlet
sc	spiral convection
x	local
w	wall.

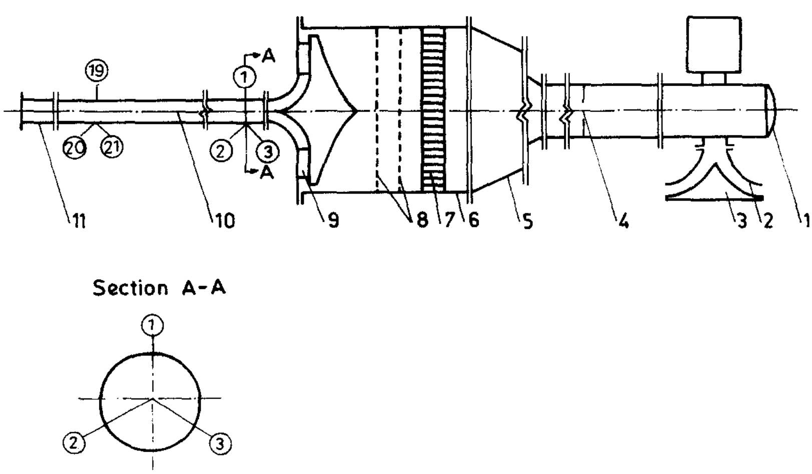

FIG. 1. Experimental setup.

The bulk temperature of the fluid at the inlet was measured with the help of a thermocouple positioned at the flow stabilizer. The exit bulk temperature was measured at the centre line of the mixing chamber. The couple voltage was measured with the help of a $6\frac{1}{2}$ digit Solartron digital multimeter through a multipoint selection switch. The meter has a filter facility which provides a $6\frac{1}{2}$ digit scale with a walking window average. The bulk temperature of the air was calculated from the linear relationship between the inlet and the exit bulk temperatures. This linearity results from a constant heat flux condition with negligible heat transfer along the pipe length.

Prior to collecting the heating run data, the system was run for about 3 h to attain the steady-state condition.

3. PREDICTION EQUATIONS

In ref. [6], the heat transfer coefficients in decaying swirl flow were calculated, assuming a rotating slug flow. While such an approach led to satisfactory prediction, it can only be applied to certain types of swirl generators.

The present work endeavours to establish a calculation procedure for the heat transfer coefficients which can be applied to devices with different types of swirl generators by using the heat transfer model proposed in ref. [6]. On the basis of the model, the elevated heat transfer in decaying swirl flow can be defined as

$$h_{d,x} = h_{sc,x} + h_{cc,x}. \tag{1}$$

The spiral flow heat transfer coefficient h_{sc} can be predicted from straight flow correlations by considering the increased velocity at the tube wall due to the swirl. This increase can be found by the vector summation of the axial and tangential velocity components at the tube wall and can be expressed [6] as

$$\alpha = \frac{v_s}{u_{av}} = (1+W_w^2)^{1/2}. \tag{2}$$

Sleicher and Rouse [7] correlated the local heat transfer coefficients in pipes for both liquids and gases the physical properties of which are temperature dependent. Their equation given below is also recommended for use by Kreith and Bohn [8]

$$Nu = (5+0.015Re^a Pr^b)\left(\frac{T_w}{T_b}\right)^n \tag{3}$$

where

$$a = 0.88 - \frac{0.24}{(4+Pr)} \tag{4}$$

$$b = 1/3 + 0.5\exp(-0.6Pr) \tag{5}$$

and

$$n = -\log\left(\frac{T_w}{T_b}\right)^{1/4} + 0.3. \tag{6}$$

Since in the present investigation, the experiment has been carried out using air as the fluid, the simplified version of equation (3) applicable to gases [7] will be used which is given as

$$Nu = 5+0.012Re^{0.83}(Pr+0.29)\left(\frac{T_w}{T_b}\right)^n \tag{7}$$

for $0.6 < Pr < 0.9$ and the fluid properties are evaluated at T_b.

The factor α can now be incorporated in this equation which yields

$$h_{sc} = \left[5+0.012(\alpha Re)^{0.83}(Pr+0.29)\left(\frac{T_w}{T_b}\right)^n\right]\frac{k}{D}. \tag{8}$$

The centrifugal convection is similar to the natural convection circulation, established over a heated horizontal plate facing up under the effect of body forces. The magnitude of the centrifugal convection (h_{cc}) should be predictable in a like manner. The applicable Grashof number is based on centrifugal acceleration and can be shown to be [6]

$$Gr = 2W_w^2 Re^2 \beta \Delta T. \tag{9}$$

For a horizontal plate with the heated surface facing upward, a number of correlations have been reported [9–11]. Al-Arabi and Reidy [10] studied the natural connection for horizontal plates of different shapes and a circular plate of a geometry similar to the pipe and recommended the relation given below

$$Nu = 0.155\,(Gr\,Pr)^{1/3}. \tag{10}$$

The centrifugal convection may then be predicted by

$$h_{cc} = 0.155(2W_w^2 Re^2 \beta \Delta T Pr)^{1/3}\frac{k}{D}. \tag{11}$$

Combining this equation with equations (1) and (8) yields the final prediction equation for heating in decaying swirl flow

$$h_{d,x} = \frac{k}{D}\left[5+0.012(\alpha Re)^{0.83}(Pr+0.29)\left(\frac{T_w}{T_b}\right)^n + 0.155(2W_w^2 Re^2 \beta\Delta T Pr)^{1/3}\right]. \tag{12}$$

4. SWIRL DECAY

In ref. [12] a calculation procedure was proposed which determines the swirl intensity, axial and tangential velocity distributions in the core region of flow at any section along the pipe by defining only the flow field at the inlet of the test pipe. The procedure utilized an equation for the eddy viscosity [13] as a function of the swirl number as well as the Reynolds number and found it in good agreement with experimental results. Such an approach can be used to determine the tangential velocity W_w at the pipe wall by a process of numerical extrapolation of the computed values of $W(Y,Z)$ for $Y \leqslant 0.9$ [14, 15]. The computation procedure [12] makes use of the following equations for the tangential velocity at inlet, swirl number, axial velocity and eddy viscosity.

Tangential velocity at inlet

$$F(Y) = W(Y,0) = \frac{A_m}{Y}(1-e^{-BY^2}) \tag{13}$$

where A_m is the maximum moment of the tangential velocity at the inlet section and constant B can be taken as 6.5.

Swirl number

$$S_n = \frac{2\pi\rho \int_0^R uwr^2 \, dr}{2\pi R\rho \int_0^R u^2 r \, dr}. \qquad (14)$$

Axial velocity

$$U = \left[\frac{(m+1)(2m+1)}{2m^2}\right](1-Y)^{1/m} + S_n[7Y - 3.1(1.2-Y)^{-0.51}] \quad (15)$$

where m is a parameter which is dependent on the Reynolds number and has the same value as in the case of the power law for the velocity distribution in swirl free turbulent flows. For the range of Reynolds numbers considered in the experiment its value can be taken as 7.

Eddy viscosity

$$\frac{\varepsilon}{\nu} = A\,Re^B \qquad (16)$$

where

$$A = 7.65 \times 10^{-4} - 2.25 \times 10^{-3} S_n^{0.5} \exp(-2.35 S_n) \qquad (17)$$

$$B = 0.89 + 0.75 S_n^{0.5} \exp(-2.4 S_n). \qquad (18)$$

The equation for $W(Y, Z)$ used in the procedure is that reported in ref. [6] and is given below

$$W(Y,Z) = \sum_{n=1}^{\infty} C_n J_1(\lambda_n Y) \exp(-\lambda_n^2 Z) \qquad (19)$$

where

$$C_n = \frac{2}{J_0^2(\lambda_n)} \int_0^1 YF(Y) J_1(\lambda_n Y) \, dY \qquad (20)$$

and

$$Z = \frac{4(1+\varepsilon/\nu)}{Re}\left(\frac{x}{D}\right). \qquad (21)$$

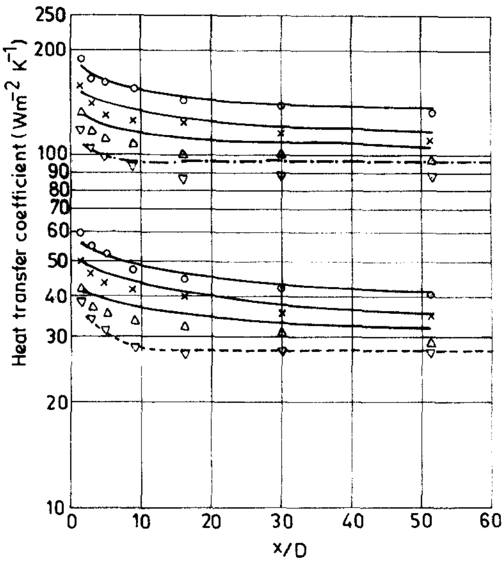

FIG. 2. Heat transfer coefficient as a function of tube length. $Re = 1.9 \times 10^5$: ——, equation (12); -·-·-, Sleicher–Rouse equation (7). $Re = 3.8 \times 10^4$: ——, equation (12); ---, Sleicher–Rouse equation (7). ○, $(S_n)_{in} = 0.52$; ×, $(S_n)_{in} = 0.31$; Δ, $(S_n)_{in} = 0.14$; ∇, zero swirl.

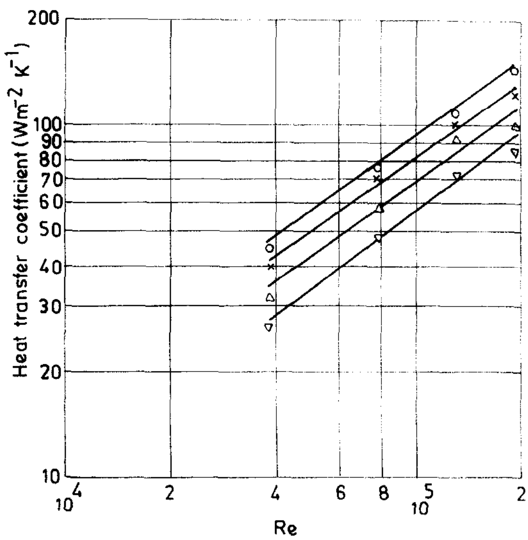

FIG. 3. Heat transfer coefficient as a function of Reynolds number, $x/D = 16$: ○, $(S_n)_{in} = 0.52$; ×, $(S_n)_{in} = 0.30$; Δ, $(S_n)_{in} = 0.14$; ∇, zero swirl; ——, equation (12).

5. DISCUSSION OF RESULTS

It is well known that the swirl flow augments the heat transfer mainly due to: (1) the increased resultant velocity in the swirl channel, and (2) the circulation of the fluid on account of centrifugal convection as the low density warmer fluid at the pipe wall be displaced into the cooler stream.

The experimental heat transfer coefficients for air obtained in this study are shown in Figs. 2–4. In all the cases, the swirling flow gives higher values of local heat transfer coefficients than those for fully developed non-swirling turbulent flow. The variation of experimental non-swirling flow heat transfer coefficients are also shown in Figs. 2 and 3 which were obtained in the same pipe, over the same range of Reynolds numbers and using the same temperature measurement system.

The non-swirling flow heat transfer coefficients, plotted as a function of pipe length in Fig. 2, clearly indicate the entry length effect. The values over the first few pipe diameters are higher compared to the remaining portion, diminishing with increasing x/D and retain the fully-developed nature after about 15 pipe diameters which is in agreement with other reported data [16]. The experimentally determined axial heat transfer coefficients agree with an average deviation of 5% with those predicted by the Sleicher–

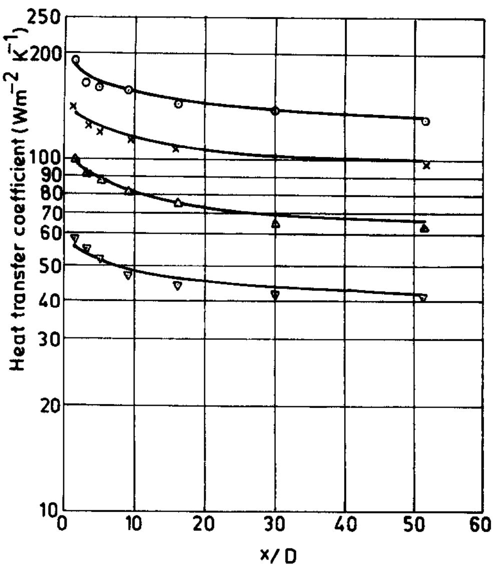

FIG. 4. Heat transfer coefficient as a function of tube length, $(S_n)_{in} = 0.52$: O, $Re = 1.90 \times 10^5$; ×, $Re = 1.31 \times 10^5$; Δ, $Re = 7.75 \times 10^4$; ∇, $Re = 3.80 \times 10^4$; ——, equation (12).

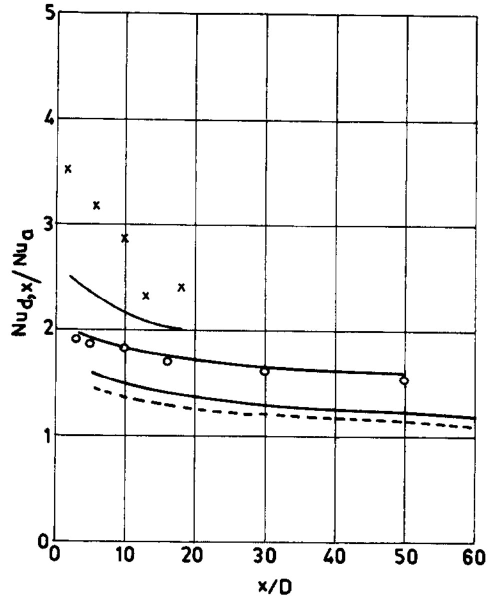

FIG. 5. Heat transfer in decaying swirl pipe flow: ----, Klepper [17], $Re = 1.5 \times 10^5$, $(S_n)_{in} = 0.45$; O, present work, $Re = 1.9 \times 10^5$, $(S_n)_{in} = 0.52$; ×, Hay and West [2], $Re = 2.46 \times 10^4$, $(S_n)_{in} = 1.41$; ——, equation (12).

Rouse equation (7) with a maximum deviation of 10% for higher Reynolds numbers.

The swirling flow heat transfer coefficients are also shown plotted in Fig. 2 against the axial distance x/D for two different Reynolds numbers. As seen from the plot for the highest swirl intensity there is an augmentation in the values of swirling flow heat transfer coefficient at inlet to the extent of 90% compared to that of fully-developed non-swirling flow. However, this enhancement in the heat transfer comes down to 40% at 50 diameters from the inlet. The intensification in heat transfer decreases as the swirl intensity gets reduced as is evident from the plot. As mentioned earlier the entrance effect exists in the axial flow heat transfer, but this sort of effect does not appear in the results of swirling flow heat transfer because the swirl enhances the turbulent mixing.

Figure 3 shows the variation of local heat transfer coefficient at $x/D = 16$ as a function of Reynolds number for three different inlet swirl intensities along with a corresponding plot for axial flow. The local heat transfer coefficients for the same swirl intensity for four Reynolds numbers are given in Fig. 4. This figure clearly indicates that the higher the Reynolds number, the higher is the heat transfer for the same swirl intensity. A good agreement of the experimental results was found with the theoretical prediction with average and maximum deviations of 7 and 11%, respectively.

6. COMPARISON WITH OTHER INVESTIGATIONS

The experimental data from the paper by Klepper [17] in decaying swirl flow where the swirl was generated by short-twisted tapes, has been compared with the values predicted by equation (12) in Fig. 5. These results are in good agreement and the theoretical predictions appear to be about 11% higher.

Hay and West [2] reported experimental data for cooling of air flowing through a pipe with a swirling motion produced by a single tangential slot, and the data has been plotted in Fig. 5 for comparison. The authors indicated that the temperature measurement system used by them recorded a lower temperature than that of the boundary wall inner surface which resulted in a heat transfer coefficient higher by about 40% than the actual values. Thorsen and Landis [18] found an approach to correlate their cooling air data by an equation with the centrifugal convection contribution having a negative sign. Later on Lopina and Bergles [19] suggested that the best correlation was achieved by simply deleting the centrifugal convection term whereupon they were able to predict the cooling data with ±10%. In the present case, the cooling data from ref. [2] could be predicted better with equation (12) by following the latter approach with a maximum and average deviations of 28 and 22%, respectively.

7. CONCLUSIONS

The present study has led to formulation of a theoretical approach which enables the determination of heat transfer coefficients in decaying swirl flow generated by different types of swirl generators located at the inlet, namely radial blades, short-twisted tapes and tangential slots.

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TRANSFERT THERMIQUE DANS UN ECOULEMENT TOURBILLONNAIRE ET TURBULENT DECROISSANT, DANS UN TUBE CIRCULAIRE

Résumé—Les coefficients de transfert de chaleur pour l'air sont mesurés le long d'un tube chauffé et un écoulement tourbillonaire décroissant, généré par une cascade radiale d'aubes. Les résultats sont comparés avec une expression proposée pour prédire les coefficients de transfert dans un écoulement tourbillonnaire. Les prédictions théoriques sont en bon accord avec les données expérimentales avec un écart moyen de 7% et maximal de 11%. L'application de l'approche théorique aux résultats théoriques obtenus par d'autres expérimentateurs pour le transfert de chaleur dans un tourbillon décroissant créé par des rubans torsadés et des fentes tangentielles à l'entrée donne aussi un accord encourageant.

WÄRMEÜBERGANG IN TURBULENTER ABKLINGENDER DRALLSTRÖMUNG IN EINEM RUNDEN ROHR

Zusammenfassung—Es wurden Wärmeübergangskoeffizienten für Luft entlang eines beheizten Rohrs bei abklingender Drallströmung gemessen, die von einem radialen Schaufelgitter erzeugt worden waren. Die Ergebnisse wurden mit einer Gleichung für den Wärmeübergangskoeffizienten bei Drallströmung verglichen. Die Gleichung stimmte gut mit den Meßdaten überein; es ergaben sich Abweichungen von 7% im Mitel und 11% in Maximum. Die Anwendung der Gleichung auf die Meßergebnisse anderer Autoren für den Wärmeübergang bei abklingender Drallströmung, die von verdrehten Bändern bzw. tangentialen Schlitzen am Einlaß erzeugt wurde, zeigte ebenfalls eine gute Übereinstimmung.

ТЕПЛООБМЕН В ТУРБУЛЕНТНОМ ЗАТУХАЮЩЕМ ЗАКРУЧЕННОМ ПОТОКЕ В КРУГЛОЙ ТРУБЕ

Аннотация—Измерены коэффициенты теплообмена вдоль нагреваемой трубы для затухающего закрученного потока воздуха, создаваемого каскадом радиальных лопастей. Результаты сравнивались с выражением, предложенным для расчета коэффициента теплообмена в закрученном потоке. Теоретические расчеты хорошо согласуются с данными эксперимента со средним и максимальным отклонением 7 и 11%, соответственно. Применение теоретического подхода к экспериментальным результатам, полученным другими исследователями для теплообмена в затухающем закрученном потоке, созданном короткими закрученными полосами и тангенциальными щелями на входе, также показывает удовлетворительное соответствие.