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Measurement of Heat Transfer and Friction Coefficients for Flow of Air in Noncircular Ducts At High Surface Temperatures

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공기유동에 대한 고온상태의 비원형 도관내에서의 열전달 및 압력강하의 측정

이   동   렬*

Abstract

Measurements of average of heat transfer and friction coefficients were obtained with air flowing through electrically heated ducts having square, rectangular (aspect ratio,5),and triangular cross sections for a range of surface temperature from 540° to 1780° R and Reynolds number from 1000 to 330,000.

The results indicates that the effect of heat flux on correlations of the average heat transfer and friction coefficients is similar to that obtained for circular tubes in a previous investigation and was nearly eliminated by evaluating the physical properties and density of the air at a film temperature halfway between the average surface and fluid bulk temperatures. With the Nusselt and Reynolds numbers based on the hydraulic diameter of the ducts, the data for the noncircular ducts could be represented by the same equations obtained in the previous investigation for circular tubes.

Correlation of the average difference between the surface corner and midwall temperatures for the square duct was in agreement with predicted values from a previous analysis. However, for the rectangular and triangular ducts, the measured corner temperature was greater by approximately 20 and 35 percent, respectively, than the values predicted by analysis.

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1. INTRODUCTION

An experimental investigation was instituted to obtain heat transfer and related pressure drop information for air flowing in tubes at high surface and fluid temperatures. The effects of such variables as surface temperature, inlet air temperature, length-to-diameter ratio, and tubeentrance configuration on heat transfer and pressure drop in smooth round tubes are summarized in Kays and Crawford[1], Bejan[2], and Shah and London[3].

The scope of the general investigation is extended herein to include the effect of flow-passage shape on heat transfer and friction coefficients for air flowing through electrically heated square, triangular, and rectangular tubes at high heat flux condition. Data were obtained for a range of Reynolds number from 1000 to 330,000 and surface temperature from 540° to 1780° R, and the results are compared with those of Kays and Crawford[1], Bejan[2], and Shah and London[3] for circular tubes.

2. APPARATUS AND PROCEDURE

2.1 Arrangement of Apparatus
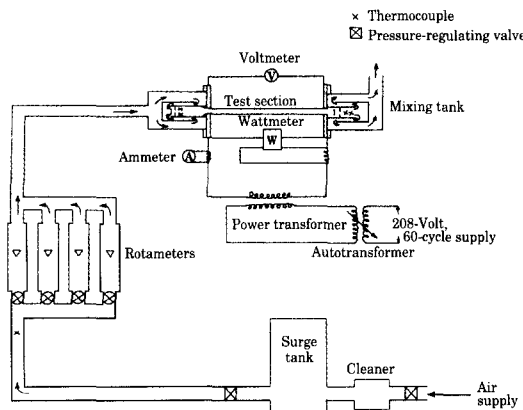
Fig. 1 Schematic diagram showing arrangement of apparatus.

A schematic diagram of the heater tubes and associated equipment is shown in Fig. 1. Compressed air is supplied through a pressure-regulating value, a cleaner, and a surge tank to a second pressureregulating value where the flow rate is controlled. From this valve, the air flows through a bank of rotameters into a threepass mixing tank, through the test section, and into a second mixing tank from which it is discharged to the atmosphere.

Electric power is supplied to the heater tube from a 208 [V], 60 [Hz] supply line through an autotransformer and a 14 : 1 power transformer. The low voltage leads of the power transformer are connected to the heater tube flanges by copper cables. The capacity of the electric equipment is 15 [KVA].

3. TEST SECTIONS

Three difference 24-inch-long cross-sectional shapes, as shown in Fig. 2-square, equilateral triangle, and rectangle-were investigated, the

Shape	Perimeter, [in.]	Hydraulic diameter, [in.]	Length to Hydraulic diameter Ratio
Square Equilateral	1.80	0.45	53
triangle	1.31	0.45	53
Rectangle	3.0	.42	57
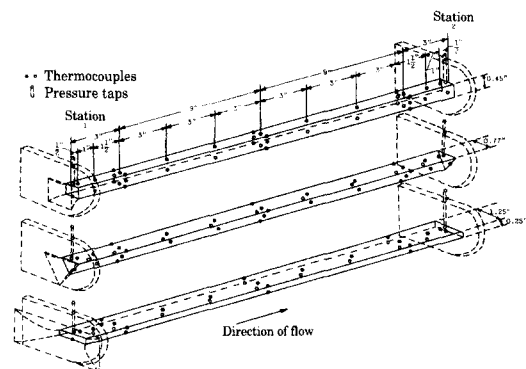
Fig. 2 Test sections showing thermocouple and pressure tap locations.

inner dimensions of which were as follows :

The test sections were fabricated from Inconel sheet stick with a thickness of 0.031 inch. The test section sides were cut to the desired dimensions and clamped on ground-steel forms : each corner was welded with a heliarc welder. The excess weld material was ground off so that the corner wall thickness did not exceed 1/32 inch. The inner corners of the test sections were sharp and even along the entire length of the test section. Steel flanges welded to the test sections at each end provided electric contact with the transformer leads from the power supply. Channels were milled in the outer faces of the flanges to minimize end heat losses, and the test sections were thermally insulated.

Outer wall temperatures were measured at 13 stations along the length of the test sections (Fig. 2) with chromel-alumel thermocouples and a self-balancing indicating potentiometer. At each station, thermocouples were located at the center of each side, except for the rectangular test section, where the thermocouples were omitted on the short sides. Thermocouples were also located at each corner at the three stations located 3, 12, and 21 inches from the entrance.

Staticpressure taps were located 1/8 inch from the entrance and exit of each test section, and each section was fitted with a long radius nozzle the throat dimensions of which matched the cross-sectional dimensions of the test section.
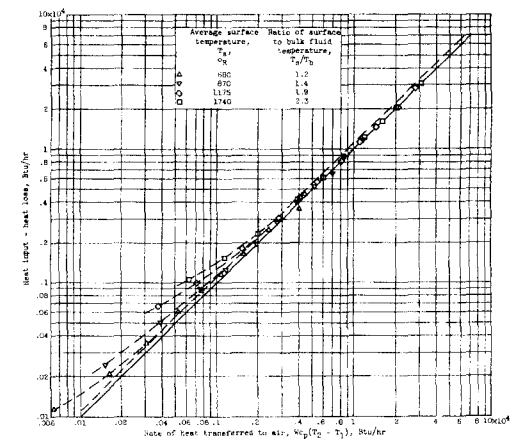
4. RANGE OF CONDITIONS

Heat transfer and associated pressure drop data were obtained with the square, the equilateral triangular, and the rectangular test sections with rounded entrances over a range of Reynolds number from 1000 to 330,000, average outer wall temperatures from 540° to 1780° R,
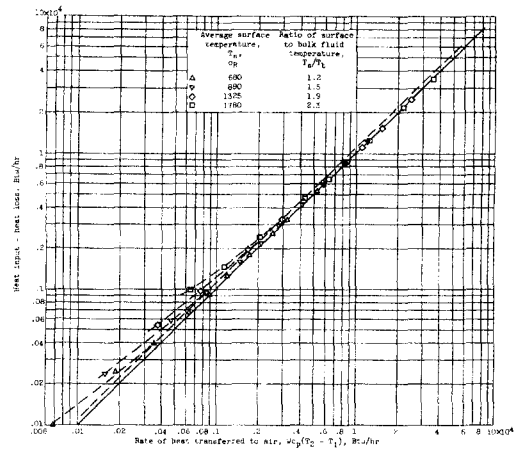
(a) Square duct
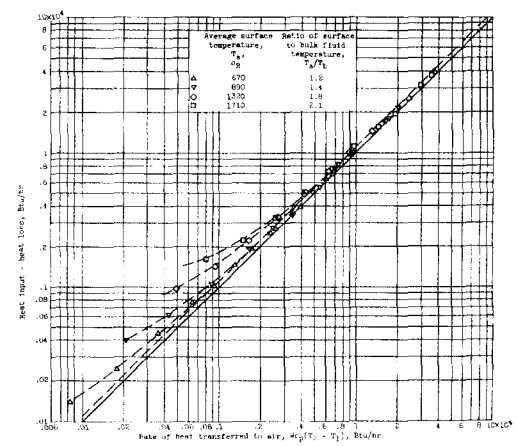
(b) Rectangular duct

(c) Triangular duct

Fig. 3 Heat balance.

and heat flux densities up to 120,000 Btu per hour per square foot of heat transfer area.

5. RESULTS AND DISCUSSION

5.1 Heat Balances

Heat balances for each test section are shown in Fig. 3, where the electric heat input minus the heat loss determined for condition of no air flow through the test section is plotted against the rate of heat transferred to the air as determined by air flow rate and temperature measurements. The heat balances obtained at low heat inputs and correspondingly low flow rates were very poor. In this region the air temperature at the exit of the test section could not be measured accurately, because the very low velocities in the outlet temperature mixing tank prevented the attainment of equilibrium conditions in the mixing tank.

The heat balances improved rapidly with increase in flow rate, and for values corresponding to turbulent flow in the test sections the data are in agreement with the match line (solid). The heat transfer coefficients presented in reference 1 for round tubes, with which the present data are compared, were calculated from the flow rate and temperature rise of the air measurements : hence, for consistency the present data are calculated in a similar manner for flow rates corresponding to Reynolds numbers of 10,000 or greater. For lower flow rates, the electric heat input and heat loss measurements were believed to be more accurate than the measured outlet air temperature. Therefore, the heat transfer coefficients for Reynolds numbers less than 10,000 are calculated from the electric heat input and heat loss measurements.

5.2 Correlation of Heat Transfer Coefficients

The average heat transfer coefficient h was computed from the experimental data by the relation

$$h = \frac{Wc_{p,b}(T_2 - T_1)}{S(T_s - T_b)} \qquad (1)$$

where

$$T_2 = T_1 + \frac{3.415EI - Q_l}{c_{p,b}}$$

for Reynolds numbers less than 10,000. The bulk temperature of the air T_b was taken as the arithmetic mean of the temperatures at the entrance T_1 and the exit T_2 of the test sections. The average surface temperature T_s was taken as the arithmetic mean of the average outside corner temperature drop through the wall was neglected.

The physical properties of air used in calculating the Nusselt, Reynolds, and Prandtl numbers are the same as those used in Kays and Crawford[1], Bejan[2], and Shah and London[3], wherein the viscosity and specific heat were based

On values reported in Keenan and Kaye [4], and the thermal conductivity was assumed to vary as the square root of temperature.

The results presented in Kays and Crawford[1], Bejan[2], and Shah and London[3] for turbulent flow in circular tubes indicate that the average Nusselt number decreases progressively as the ratio of surface to fluid bulk temperature increases when the fluid properties are evaluated at the fluid bulk temperature. The effect of the ratio of surface to bulk temperature was eliminated by evaluating the properties of the air, including the density term in the Reynolds number at the film temperature, defined as the arithmetic average of the surface and bulk

temperatures. The data for Reynolds numbers greater than 10,000 were well represented by the following relation :

$$\frac{hD}{k_f}=0.034\left(\frac{\rho_f V_b D}{\mu_f}\right)^{0.8}\left(\frac{c_{p,f}\mu_f}{k_f}\right)^{0.4}\left(\frac{L}{D}\right)^{-0.1}$$

which, for values of L/D between 53 and 57, becomes

$$\frac{hD}{k_f}=0.023\left(\frac{\rho_f V_b D}{\mu_f}\right)^{0.8}\left(\frac{c_{p,f}\mu_f}{k_f}\right)^{0.4} \tag{2}$$

The average heat transfer coefficients obtained herein for square, rectangular, and triangular ducts for a range of Reynolds number from 1000 to 200,000 and rations of surface to bulk temperature from 1.2 to 2.3 are correlated accordingly in Fig. 4. A solid line representing data obtained in reference 1 with a circular test section having a length-to-diameter ratio of 60 for similar conditions is included for comparison. For Reynolds numbers greater than 10,000, the reference line represents equation (2) ; for small Reynolds numbers the reference line represents the data of Kays and Crawford[1], Bejan[2], and Shah and London[3] recomputed on the basis of the electric heat input and heat-loss measurements for the purpose of comparison. The data for the square tube (Fig. 4(a)) agree well with the reference line for all surface to bulk temperature ratios and Reynolds numbers. For the rectangular duct having an aspect ratio of 5, the data (Fig. 4(b)) are considerably higher than the reference line for Reynolds numbers from 1000 to 10,000 and are represented by equation (2) for Reynolds numbers above 2500. The higher values are in agreement with data obtained for noncircular ducts at lower heat fluxes by other investigators. These data indicate that use of the
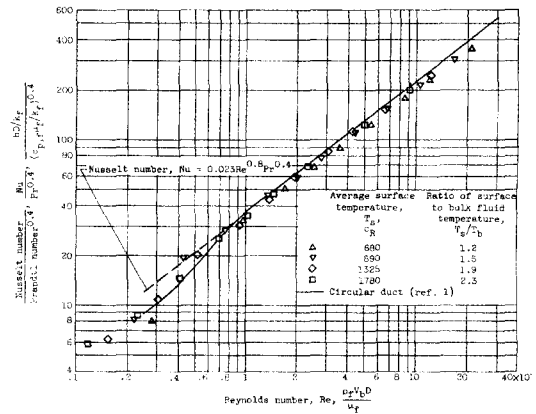
(a) Square duct
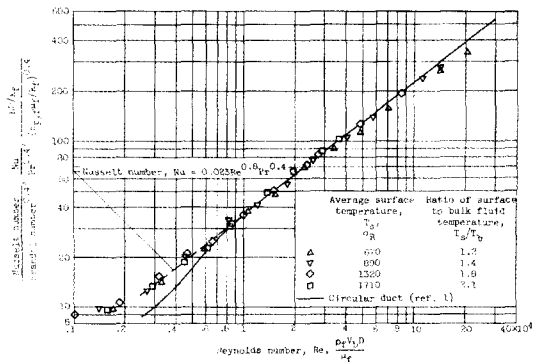
(b) Rectangular duct.
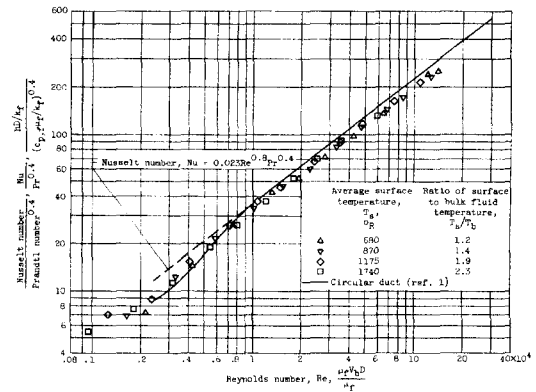
(c) Triangular duct

Fig. 4 Correlation of heat transfer coefficients for air flow in ducts with variable heat flux. Bellmouth entrance ; inlet temperature, 535R ; properties of air evaluated at film temperature.

hydraulic diameter does not result in cor-
relation of data for various passage shapes in
the laminar and transition flow regions. For
Reynolds numbers above 40,000 the data fall
slightly below the reference line.

In Fig. 4(c), the average heat transfer
coefficients for the triangular duct vary simi-
larly to those obtained for the square duct. For
Reynolds numbers above 10,000 the data were
lower than the reference line by 5 to 15 percent
and could best be represented by a line having
a slope of 0.78 rather than 0.8. This difference
in slope is also noticeable in Fig. 4(b) for the
data of the rectangular duct for high Reynolds
numbers. This variation in slope could be elimi-
nated by defining the average surface tempera-
ture as the midwall temperature instead of as
the arithmetic average of the corner and
midwall temperature. For example (as is shown
in Fig. 7), the difference between the average
surface and the average midwall temperatures
for the triangular tube is 3 percent of the
difference between the average midwall
temperature and the fluid bulk temperature for
a Reynolds number of 10,000, and 11 percent for
a Reynolds number of 100,000. Evaluating the
average heat transfer coefficient on the basis of
the difference between the midwall and fluid
bulk temperatures would result in a corre-
sponding increase in the average heat transfer
coefficient and, hence, would bring the data into
agreement with the reference line for circular
tubes. Measurements of the variation in rate of
heat transfer around the periphery of non-
circular ducts are required in order to define the
average surface temperature for evaluating the
average heat-transfer coefficient.

5.3 Correlation of Friction Coefficients

The method of calculating the average
friction coefficient is essentially the same as

described in Kays and Crawford[1], Bejan[2], and
Shah and London[3], where in

$$f_r = \frac{\Delta p_{fr}}{4\frac{L}{D}\frac{\rho_f V_b^2}{2g}} \tag{3}$$

where

$$\Delta p_{fr} = (p_1 + p_2) - \frac{G^2 R}{g}\left(\frac{t_2}{p_2} - \frac{t_1}{p_1}\right)$$

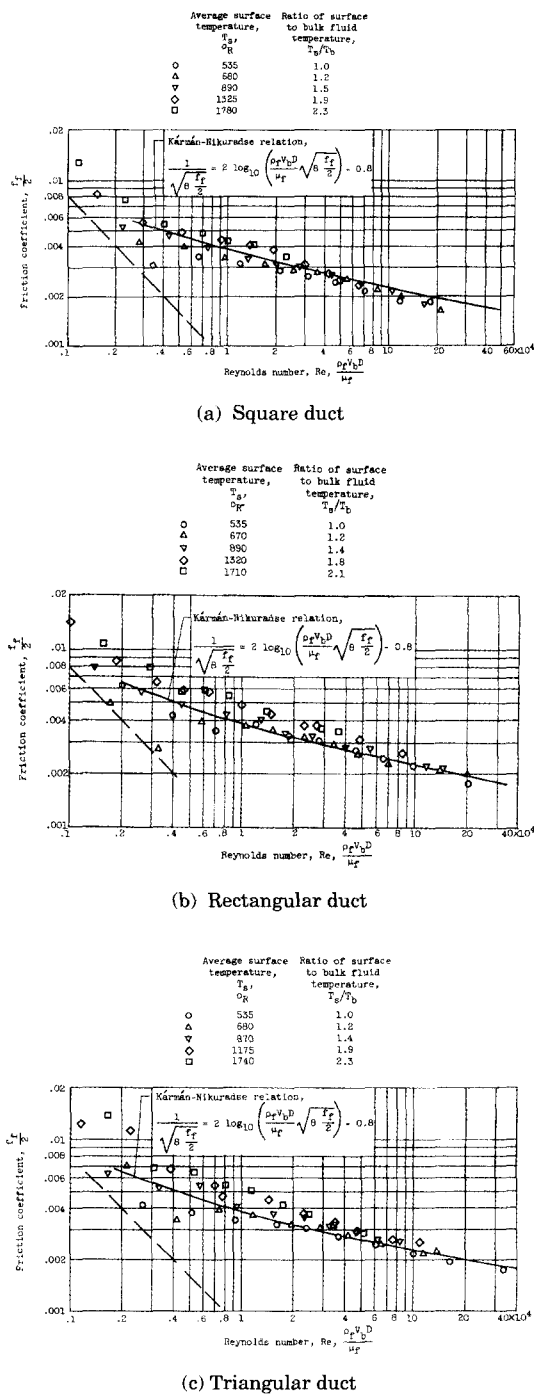$$\rho_1 = \left(\frac{\rho_1 + \rho_2}{2}\right)\left(\frac{t_b}{t_f}\right)$$

and

$$t = \frac{-\gamma g}{(\gamma-1)R}\left(\frac{p}{G}\right)^2 + \sqrt{\left[\frac{\gamma g}{(\gamma-1)R}\left(\frac{p}{G}\right)^2\right]^2 + 2T\frac{\gamma g}{(\gamma-1)R}\left(\frac{p}{G}\right)^2}$$

The subscripts 1 and 2 refer to positions
within the ducts, located 1/8 inch from the
entrance and exit ends of the ducts, respec-
tively. For Reynolds numbers less than 10,000
the exit static temperature t_2 is based on the
value of the exit total temperature T_2 deter-
mined from the electric heat input and heat
loss measurements, as was mentioned in the
preceding section.

The average friction coefficients as calculated
above are shown in Fig. 5 correlated by the
method summarized in Kays and Crawford [1]
for high heat flux conditions. Included for com-
parison is the line representing the Karman-
Nikuradse relation for turbulent flow in pipes
modified for the effect of heat flux on the
friction coefficient, which is

$$\frac{1}{\sqrt{\frac{8f_f}{2}}} = 2\log_{10}\left(\frac{\rho_f V_b D}{\mu_f}\sqrt{\frac{8f_f}{2}}\right) - 0.8 \tag{4}$$

In Fig. 5(a), the average friction coefficients
for the square ducts for the square duct for
Reynolds numbers above 10,000 to 20,000
agree reasonably well with the reference line

Average surface temperature, T_s, $^\circ R$	Ratio of surface to bulk fluid temperature, T_s/T_b
○ 535	1.0
△ 680	1.2
▽ 890	1.5
◇ 1525	1.9
□ 1780	2.3

(a) Square duct

Average surface temperature, T_s, $^\circ R$	Ratio of surface to bulk fluid temperature, T_s/T_b
○ 535	1.0
△ 670	1.2
▽ 890	1.4
◇ 1320	1.8
□ 1710	2.1

(b) Rectangular duct

Average surface temperature, T_s, $^\circ R$	Ratio of surface to bulk fluid temperature, T_s/T_b
○ 535	1.0
△ 680	1.2
▽ 870	1.4
◇ 1175	1.9
□ 1740	2.3

(c) Triangular duct

Fig. 5 Correlation of friction coefficients for air flow in ducts with variable heat flux. Viscosity and density evaluated at film temperature.

for circular tubes, although the friction coefficient increases with an increase in heat flux or surface to fluid bulk temperature ratio. In the transition region for low Reynolds numbers, the friction coefficient varies considerably with heat flux. Similar variations were obtained in Kays and Crawford[1] for circular tubes.

The data for the rectangular and triangular ducts (Figs. 5(b) and (c), respectively) indicate that the friction coefficient increases more with increasing heat flux than in the square duct (Fig. 5(a)), which may possibly be caused by the development of secondary flows in the rectangular and triangular ducts.
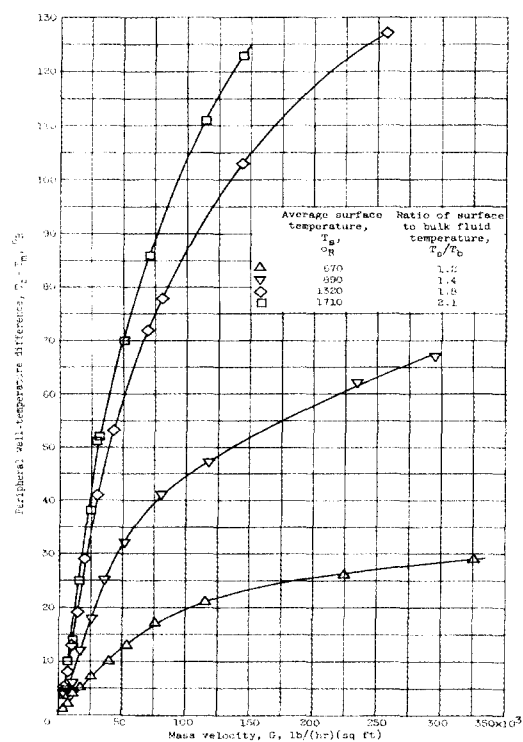
5.4 Correlation of Peripheral Temperature Variations

Corner surface temperatures were measured at three stations along the length of each test section (3, 12, and 21 in. from entrance). The average differences between the corner and midwall temperatures are plotted against the mass velocity in Fig. 6. The average difference plotted herein was taken as the arithmetic average of the differences between the local average corner and midwall temperature at each station. In general, the local average of the test section ; however, in several instances the local values near the exit were less than those measured at the center station, which indicates the possibility of locally developed secondary flow near the corners. The average peripheral temperature differences increased at a decreasing rate with increases in mass velocity and heat flux. Maximum values of 35°, 127°, and 110° R ducts, respectively.
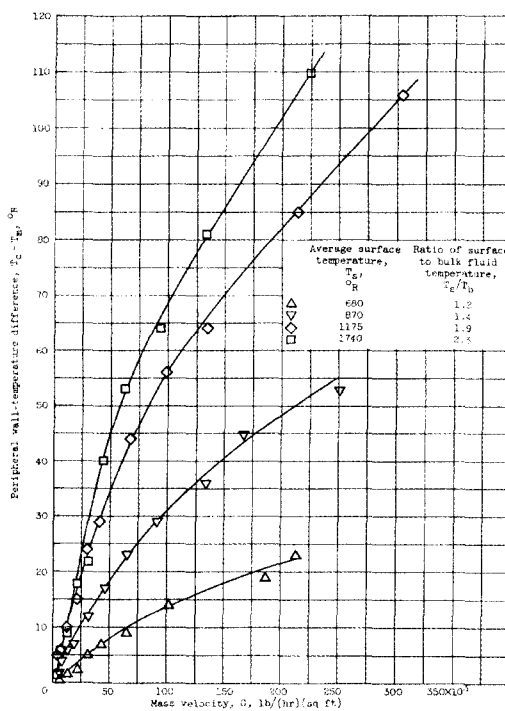
In Eckert and Drake[5], Jones[6], and Novotny et al.[7], a method was developed for predicting peripheral wall temperature variations for flow in noncircular tube heat exchangers with internal heat generation in the tube walls
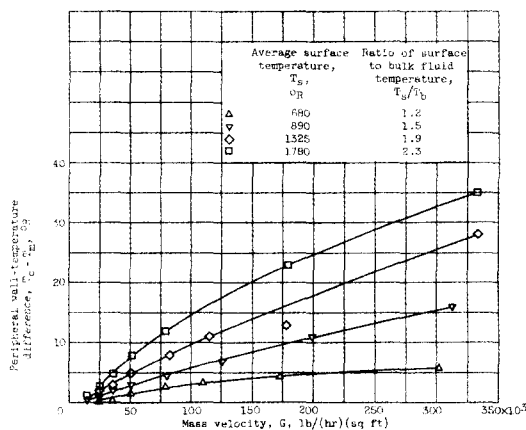
(a) Square duct

(c) Triangular duct

(b) Rectangular duct

Fig. 6 Variation of peripheral wall temperature difference with mass velocity at various average wall temperature.

based on shear stress distributions measured by Nikuradse for flow in noncircular ducts. The average peripheral temperature differences are correlated accordingly in Fig. 7, where the ratio of the difference between the corner and the midwall temperature to the difference between the surface (which was taken as the average midwall temperature) and fluid bulk tem-
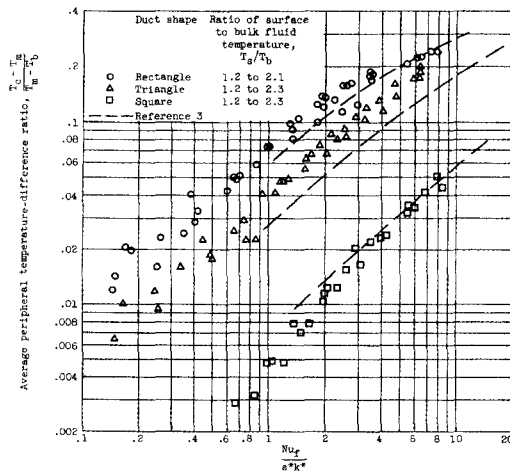
**Fig. 7 Correlation of average peripheral tempera-
ture difference for flow in noncircular
ducts with variable heat flux.**

peratures is plotted against the dimensionless
parameter, Nusselt number divided by the
ratio of wall thickness to hydraulic diameter of
the flow passage and the ratio of thermal
conductivity of the wall material to the fluid
(wherein the average heat transfer coefficient
is based on the difference between the average
midwall and fluid bulk temperature). Included
for comparison are the reference lines
calculated for turbulent flow in the ducts from
Eckert and Drake[5], Jones[6], and Novotny et
al.[7] for the case of no flow over the outer
surface of the duct ; hence for the heattransfer
coefficient for the outer surface equal to zero.
The data for the square duct agree fairly well
with the predicted values in the turbulent flow
range. The measured values for the rectan-
gular and triangular ducts were greater on the
average by about 20 and 35 percent, respec-
tively, than the predicted values. This differ-
ence in measured and predicted values results
in part from the assumption of similarity
between variation of rate of heat transfer and
shear stress distribution around the periphery
of a noncircular duct and in part from the

uncertainty in estimating the shearstress
distribution for a duct with aspect ratio of 5
from measurements obtained for a duct with
aspect ratio of 3.5. Similar results are indicated
for the triangular duct. Maximum values of the
ratio of the average difference between the
average surface and bulk temperatures
obtained were 0.05, 0.25, and 0.20 for the
square, the rectangular, and the triangular
ducts, respectively.

6. CONCLUSION

The results of this investigation of heat
transfer and pressure drop for air flowing
through noncircular ducts having square,
rectangular (aspect ratio, 5), and equilateral
triangular cross sections for a range of surface
temperature from 540° to 1780° R, correspond-
ing surface to bulk temperature ratio from 1,2 to
2,3, and Reynolds numbers from 1000 to 330,000
may be summarized as follows :

1. The effect of the ratio of surface to bulk
   temperature on correlations of the average
   heat transfer and friction coefficients was
   the same as that obtained in a previous
   investigation for similar ranges of condi-
   tions for flow in circular tubes and was
   nearly eliminated by evaluating the phy-
   sical properties and density of the air at a
   film temperature halfway between the bulk
   and surface temperatures.
2. Correlations of the average heat transfer
   and friction coefficients were in reasonable
   agreement with the results obtained for the
   circular tubes with the Nusselt number and
   Reynolds number based on the hydraulic
   diameter of the duct.
3. Correlation of the average peripheral
   temperature difference for the square duct
   was in agreement with values predicted by

a previous analysis. However, the measured values for the rectangular and triangular ducts were greater by approximately 20 and 35 percent, respectively, than the predicted values.

4. Maximum values of the ratio of the average difference between the corner and midwall temperature to the difference between the average surface and bulk temperatures obtained were 0.05, 0.25, and 0.20 for the square, the rectangular, and the triangular ducts, respectively.

NOMENCLATURE

A : cross-sectional area, (sq ft)

c_p : specific heat, (Btu/(lb)($^\circ$F))

D : inner hydraulic diameter, $\dfrac{4A}{Perimeter}$,(ft)

E : voltage drop across test section, (v)

F : average friction coefficient

f_f : modified film friction coefficient

G : mass velocity, (lb/(hr)(sq ft))

G : acceleration due to gravity, $(4.17 \times 10^8 \text{ft} /\text{hr}^2)$

H : average heat transfer coefficient, (Btu/ (hr)(sq ft)($^\circ$F))

I : current flow through test section, (amp)

k : thermal conductivity, (Btu/(hr)(sq ft) ($^\circ$F/ft))

k^* : ratio of thermal conductivity of wall material to coolant

L : length of test section, (ft)

Nu : Nusselt number, hd/k

Pr : Prandtl number, $(c_p\mu/k)$

P : static pressure, (lb/sq ft abs)

Δ_p : overall static pressure drop across test section, (lb/sq ft)

Q_l : heat loss from test section to surroundings, (Btu/hr)

R : gas constant, 53.35 (ft-lb/(lb)($^\circ$R))

Re : Reynolds number, $\rho VD\mu$

S : heat transfer area of teat section, (sq ft)

S^* : ratio of wall thickness to hydraulic diameter

T : total temperature, ($^\circ$R)

T_b : average fluid bulk temperature, $\dfrac{T_1+T_2}{2}$, ($^\circ$R)

T_f : average fluid film temperature, $\dfrac{T_s+T_B}{2}$, ($^\circ$R)

T_s : average surface temperature, $T_m+\dfrac{T_c-T_m}{2}$,

t : static temperature, ($^\circ$R)

V : velocity, (ft/hr)

W : flow rate, (lb/hr)

γ : ratio of specific heats

μ : absolute viscosity, lb/(hr)(ft)

ρ : density, lb/cu ft

Subscripts

Av : average

B : bulk (when applied to properties, indicates evaluation at average bulk temperature, Tb)

c : peripheral location at corner of test section

f : film (when applied to properties, indicates evaluation at average film temperature, Tf)

fr : friction

m : peripheral location midway between corners of test section

s : surface (when applied to properties, indicates evaluation at average surface temperature

1 : test-section entrance

2 : test-section exit

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