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Convective heat transfer during forced crossflow of fluids over a circular cylinder, including free convection effects

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CONVECTIVE HEAT TRANSFER DURING FORCED CROSSFLOW OF FLUIDS OVER A CIRCULAR CYLINDER, INCLUDING FREE CONVECTION EFFECTS

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CONVECTIVE HEAT TRANSFER DURING FORCED CROSSFLOW OF FLUIDS OVER A CIRCULAR CYLINDER, INCLUDING FREE CONVECTION EFFECTS

1. NOTATION AND UNITS

The sets of units given below do not require the use of conversion factors in forming the non-dimensional parameters used in this Item. The SI set is part of a completely coherent system of units but the British set is not.

		Units	
		SI	British
c_p	specific heat at constant pressure	J/kg K	Btu/lb°F
D	outside diameter of cylinder	m	ft
Gr	Grashof number, $\frac{\beta_g D^3 \rho^2 T_{w, m} - T_b }{\mu^2}$		
g	gravitational acceleration	m/s ²	ft/h ²
h	heat transfer coefficient	W/m ² K	Btu/ft ² h°F
L	length of cylinder	m	ft
Nu	Nusselt number, $\frac{hD}{\lambda}$		
n	constant defined by Equation (3.5)		
Pr	Prandtl number, $\frac{c_p \mu}{\lambda}$		
p	constant defined by Equation (3.6)		
$\dot{Q}$	total rate of heat flow between fluid and surface of cylinder	W	Btu/h
Re	Reynolds number, $\frac{\rho V D}{\mu}$		
St	Stanton number, $\frac{h}{\rho V c_p}$		
T	absolute temperature	K	°R(°F + 460)
V	mean free stream fluid velocity	m/s	ft/h
β	coefficient of volumetric expansion of fluid	1/K	1/°F
θ	inclination of cylinder axis to flow	degrees	degrees

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λ	thermal conductivity*	W/m ² (K/m)	Btu/ft ² h(°F/ft)
μ	dynamic viscosity	[†] N s/m ²	lb/ft h
ρ	density	kg/m ³	lb/ft ³

For the purposes of this Item the fluid properties are to be evaluated at the fluid bulk temperature, T_b

Suffixes

b	appropriate to conditions in the bulk of the fluid
c	appropriate to constant fluid property conditions
m	mean value
w	appropriate to conditions at the wall of the cylinder.

* The symbol k is often used for thermal conductivity.

[†] 1 N s/m² = 1 kg/m s.

2. INTRODUCTION

This Item presents convective heat transfer data in non-dimensional form for the crossflow of a single-phase Newtonian fluid over a circular cylinder under conditions of negligible thermal radiation and zero heat generation within the fluid. The cases of horizontal and vertical cylinders are covered and the data apply for gases and for liquids with the exception of liquid metals. Figure 1 shows, in terms of the values of Re and $GrPr$, the regions in which the data may be applied and also indicates where in the Item the appropriate information may be found.

Section 3 introduces the principal factors influencing convective heat transfer. Sections 4 and 5 give numerical data for the average Nusselt number in forced and free convection respectively. Insufficient information was available to allow provision of data for the mixed convection regime.

Section 6 considers briefly a number of other factors which may influence the convective heat transfer and Section 7 discusses the variation of the local heat transfer coefficient over the surface of the cylinder.

Appendix A shows the ranges of the various parameters covered by the experimental data used in the correlations; Appendix B gives the distribution of errors for these correlations.

3. PRINCIPAL FACTORS INFLUENCING CONVECTIVE HEAT TRANSFER FOR A FLUID FLOWING OVER A CYLINDER

3.1 General Considerations

The convective heat transfer between the external surface of a cylinder and a fluid moving relative to it can be expressed in terms of a mean heat transfer coefficient defined by

$$h = \frac{\dot{Q}}{\pi DL(T_{w,m} - T_b)} \quad (3.1)$$

In Equation (3.1), T_b is the bulk temperature of the fluid (*i.e.* the temperature of the fluid remote from the cylinder) and $T_{w,m}$ is the mean temperature of the outside surface of the cylinder. A number of methods were used in the Derivations to arrive at an average of the measured local values of wall temperatures and it was not possible to reduce the data to a common form. However, the differences in these methods contribute only a small amount to the scatter of the experimental data on Nusselt number.

In the general case, the heat transfer coefficient may be related to the fluid properties, cylinder geometry and flow parameters in the following way:

$$\begin{aligned} Nu &= \frac{hD}{\lambda} = f\left(\frac{\rho VD}{\mu}, \frac{c_p \mu}{\lambda}, \frac{\beta g D^3 \rho^2 |T_{w,m} - T_b|}{\mu^2}, \frac{L}{D}, \theta\right) \\ &= f(Re, Pr, Gr, L/D, \theta). \end{aligned} \quad (3.2)$$

In some applications it may be more convenient to use the Stanton number which is related to the Nusselt number by

$$St = Nu/(RePr).$$

3.2 Heat Transfer Regimes

Free or natural convection conditions are due to the buoyancy forces which exist when temperature differences occur in a fluid.

In the case of free convection alone, Equation (3.2) reduces to

$$Nu = f_1(Gr, Pr, L/D, \theta), \quad (3.3)$$

which is usually considered in the alternative but equivalent form

$$Nu = f_2(GrPr, Pr, L/D, \theta). \quad (3.4)$$

The product $GrPr$ is often called the Rayleigh number, denoted by Ra .

For horizontal cylinders, provided end effects are insignificant (typically for L/D greater than about 3) Nu is independent of L/D .

Forced convection conditions are set up by the imposition of exterior forces on a fluid. Under forced convection conditions free convection is always present, but is of varying significance.

When forced convection is the dominant effect, provided that end effects are insignificant (typically for L/D greater than about 3), Equation (3.2) may be reduced to

$$Nu = f_3(Re, Pr, \theta).$$

Between the conditions where either free convection or forced convection is the dominating feature, there is a region of mixed convection.

Thus free convection effects are characterised by the product $GrPr$ and forced convection effects by the Reynolds number Re . It is, in principle, possible to define on a plot of Re against $GrPr$ the regions in which forced convection or free convection dominates and the region of mixed convection. In practice, the available experimental data permit the regime boundaries to be established in part for gases only. It was found possible to correlate most of the published gas data and all of the liquid data by means of pure forced or pure free convection correlation equations. As a broad working guide the following rule applies. In practical situations where there is forced convection, this component dominates any free convection effects, except at very low Reynolds numbers (typically less than unity).

Figure 1 shows the distribution of experimental data and indicates suggested limits for the application of the correlations.

3.3 Variable Fluid Properties

Under conditions where the difference between the wall and fluid bulk temperatures is large, there are significant variations of the fluid properties affecting heat transfer, namely, viscosity, thermal conductivity, specific heat and density. As a result of these variations, the value of the heat transfer coefficient is modified.

In the case of gases, the effect of variation of the fluid properties with absolute temperature can be represented approximately by simple power laws. In order to account for variable property effects, an empirical relation for Nusselt number, written in the form

$$\frac{Nu}{Nu_c} = \left(\frac{T_{w,m}}{T_b} \right)^n, \quad (3.5)$$

is used.

In the case of liquids, where the property that varies most significantly with temperature is viscosity, the Nusselt number can be expressed in the empirical form

$$\frac{Nu}{Nu_c} = \left(\frac{\mu_{w,m}}{\mu_b} \right)^p. \quad (3.6)$$

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4. AVERAGE HEAT TRANSFER DATA FOR FORCED CONVECTION

When plotted in the form Nu against Re the experimental data for average heat transfer due to forced convection in crossflow over single cylinders lie on smooth curves showing no distinct indication of the various flow regimes which are found for the drag coefficient data.

4.1 Liquids, Cylinder Normal to Flow

The variation of Nusselt number with Reynolds number for fluids of various Prandtl numbers from 5 to 1500 is shown in Figure 2 for the case of constant fluid properties. The curves of Figure 2 are based on the equation

$$Nu_c = \exp[-0.186 + 0.338(\log_e Re) + 0.362(\log_e Pr) + 0.0131(\log_e Re)^2 - 0.00926(\log_e Pr)^2]. \quad (4.1)$$

The effect of variable fluid properties is shown on Figure 3 which presents a factor based on the empirical equation

$$\frac{Nu}{Nu_c} = \left(\frac{\mu_{w,m}}{\mu_b} \right)^{-0.137} \quad \text{for } 0.061 < \frac{\mu_{w,m}}{\mu_b} < 0.936. \quad (4.2)$$

Equations (4.1) and (4.2) together correlate experimental data from Derivations 3, 10 and 15 with an overall r.m.s. error of 6.8 per cent*.

4.2 Gases, Cylinder Normal to Flow

The variation of Nusselt number with Reynolds number for gases (air and nitrogen) of Prandtl number equal to 0.7 is shown on Figure 2 for the case of constant gas properties. The curve on Figure 2 is a mean line through the experimental data from Derivations 1, 2, 4, 5, 6, 9, 11, 12 and 16 and the theoretical data from Derivations 14 and 16. For Reynolds numbers greater than 0.5 the curve on Figure 2 is approximated by the equation

$$Nu_c = \exp[-0.179 + 0.252(\log_e Re) + 0.0528(\log_e Re)^2 - 0.00666(\log_e Re)^3 + 0.000335(\log_e Re)^4]. \quad (4.3)$$

At Reynolds numbers higher than 10^4 the deviations from the line given by Equation (4.3) are appreciable and systematic. They are probably related to the intensity of turbulence in the gas flow but insufficient information is available to quantify this effect. References 1 and 2 discuss the effect of varying turbulence on heat transfer.

The effect of variable gas properties is shown on Figure 4, based on the expression

$$\frac{Nu}{Nu_c} = \left(\frac{T_{w,m}}{T_b} \right)^n, \quad (4.4)$$

* A more detailed breakdown of the distribution of errors is given in Appendix B.

where $n = 0.175$ for $1.12 < T_{w,m}/T_b < 4.47$ and $n = -0.120$ for $0.246 < T_{w,m}/T_b < 0.505$.

Equations (4.3) and (4.4) together correlate the experimental data from Derivations 1, 2, 4, 5, 6, 9, 11 and 12 with an overall r.m.s. error of 12 per cent* for $3 \times 10^{-1} < Re < 3 \times 10^5$.

4.3 Effect of Inclination of Cylinder to Flow

Few data are available on this effect. In the cases quoted in Derivations 7 and 8, for a single cylinder whose axis was inclined at an angle of 45° to the direction of flow of the fluid, and where $Pr = 0.7$, and $5 \times 10^3 < Re < 4 \times 10^4$, the value of the heat transfer coefficient was about 75 per cent of the value obtained on the equivalent 90° crossflow arrangement. These Derivations also give results for flow at various angles to banks of cylinders. It is suggested that, in the absence of more specific data, those results (shown in Figure 5) should be used as an approximate indication of the effect of cylinder inclination.

5. AVERAGE HEAT TRANSFER DATA FOR FREE CONVECTION

5.1 Vertical Cylinders

The variation of Nusselt number with the product of Grashof number and Prandtl number for vertical cylinders is shown on Figure 6. For $GrPr > 5 \times 10^3$, this Figure may be represented by the equation

$$Nu_c = \exp[-2.95 + 1.02(\log_e GrPr) - 0.0829(\log_e GrPr)^2 + 0.00267(\log_e GrPr)^3] \quad (5.1)$$

The equation is based on data for air only and applies only for cylinders with $L/D > 7.5$. For shorter cylinders, end effects are significant and depend on L/D and on the end configuration. The data used to derive Equation (5.1) were all for turbulent flow, the criterion being that $GrPr(L/D)^3$ is greater than about 10^9 .

Figure 8 shows a factor which may be allowed for the effect of variable gas properties and is based on the expression

$$\frac{Nu}{Nu_c} = \left(\frac{T_{w,m}}{T_b} \right)^{0.175} \quad \text{for } 1.04 < \frac{T_{w,m}}{T_b} < 1.57 \quad (5.2)$$

Equations (5.1) and (5.2) together correlate experimental data from Derivations 22, 25 and 28 with an overall r.m.s. error of 14.2 per cent*.

5.2 Horizontal Cylinders

The variation of Nusselt number with the product of Grashof and Prandtl number for horizontal cylinders is shown in Figure 7. Figure 7 may be represented by the equation

$$Nu_c = \exp[0.0545 + 0.0922 \log_e Pr - 0.0147(\log_e Pr)^2 + 0.118 \log_e GrPr + 0.00485(\log_e GrPr)^2] \quad (5.3)$$

* A more detailed breakdown of the distribution of errors is given in Appendix B.