



Convective heat transfer during crossflow of fluids over plain tube banks

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CONVECTIVE HEAT TRANSFER DURING CROSSFLOW OF FLUIDS OVER PLAIN TUBE BANKS

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1. NOTATION AND UNITS

Two sets of units are given below. The SI set is part of a completely coherent system of units. The British units are not.

		Units	
		SI	British
A_{min}	minimum cross-sectional area between tubes normal to local flow direction (see Figures 2 and 3)	m ²	ft ²
a	parameter in Equation (3.2)		
c_p	specific heat capacity at constant pressure	J/kg K	Btu/lb°F
D	outside diameter of tube	m	ft
F_1	factor accounting for variation in fluid properties (see Section 3.3)		
F_2	factor accounting for effect on $\overline{Nu}$ of $N \neq 10$ (see Section 3.4)		
F_3	factor accounting for effect on $\overline{Nu}$ of tube inclination (see Section 3.5)		
F_4	factor giving row to row variation in Nu_n (see Section 4)		
G	mass flow per unit cross-sectional area, $\rho_b V$	kg/s m ²	lb/h ft ²
$\bar{h}$	average external heat transfer coefficient for the bank	W/m ² K	Btu/ft ² h°F
L	effective length of tube	m	ft
m	power law index of Re_b in Equation (3.2)		
N	number of tube rows in bank in direction of flow		
$\overline{Nu}$	average Nusselt number for the bank, $\bar{h}D/\lambda_b$		
Nu_n	mean Nusselt number for separate tube rows		
Nu_r	value of $\overline{Nu}$ for the reference conditions (see Section 3.1)		
n	tube row number in direction of flow		

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Pr	Prandtl number, $c_p \mu / \lambda$		
$\dot{Q}$	rate of heat transfer	W	Btu/h
Re_b	Reynolds number, $\rho_b V_{max} D / \mu_b$, $G_{max} (D / \mu_b)$		
s_l	longitudinal pitch (see Sketches on Figures 2 and 3)	m	ft
s_t	transverse pitch (see Sketches on Figures 2 and 3)	m	ft
T	thermodynamic temperature	K	°R
ΔT	temperature difference between the tube wall and crossflow fluid (see Section 3.1)	K	°R
V	velocity of crossflow fluid	m/s	ft/h
W	mass flow	kg/s	lb/h
X_l	longitudinal pitch to diameter ratio, s_l / D		
X_t	transverse pitch to diameter ratio, s_t / D		
θ	inclination of tube axis to duct centre line (see Figure 7)	degree	degree
λ	thermal conductivity	W/m ² (K/m)	Btu/ft ² h (°F/ft)
μ	dynamic viscosity	kg/m s	lb/ft h
ρ	density	kg/m ³	lb/ft ³
<i>Suffixes</i>			
b	appropriate to conditions in the bulk of crossflow fluid		
c	appropriate to constant fluid property conditions		
m	mean value		
max	relating to value calculated for minimum flow area between tubes normal to local flow direction (see Figures 2 and 3)		
N	relating to value for bank with N rows		
w	appropriate to conditions at wall of a tube		

2. INTRODUCTION

For the calculation of heat transfer in a tube bank, data are required for the internal heat transfer between the tube walls and the fluid flowing through the tubes and the external heat transfer due to the crossflow of fluid over the tubes. Item Nos. 67016, 68006 and 68007 (References 2-4) deal with the internal heat transfer; this Item covers the case of convective heat transfer for the crossflow of a fluid over tube banks of circular cylinders of negligible thermal radiation and zero heat generation within the fluid.

The case of forced convection heat transfer into or out of the tubes is covered in detail but no data are available from which to assess the effects of combined forced and free convection (see Section 5.1).

The data are presented in non-dimensional form and are valid for the crossflow of a single-phase Newtonian fluid within the bounds of Table 8.2. This includes most normal liquids and gases. In applying the data to the heating of liquids, the temperature of the tube outside surface at all points in contact with the liquid must be less than the saturation temperature for the liquid concerned.

Section 3 presents data averaged for a complete bank of tubes and Section 4 considers the variations in heat transfer from row to row. Section 5 considers briefly a number of other factors that may influence convective heat transfer. The methods presented here are expected to yield heat transfer coefficients accurate to within about ± 15 per cent over most of the Reynolds number range but Appendix A and Table 8.2 discuss the derivation and accuracy of the data in greater detail. A summary of the calculation procedure is given in Section 9.

3. AVERAGE HEAT TRANSFER DATA FOR A BANK WITH N TUBE ROWS

3.1 Calculation of Heat Transfer

In general terms, when free convection effects can be ignored, the Nusselt number can be related to the flow parameters, fluid properties and tube bank geometry by the expression

$$\overline{Nu} = f(Re_b, Pr_b, Pr_w, X_t, X_p, N, L/D, \theta).$$

This general equation can be expressed in the simplified form

$$\overline{Nu} = Nu_r \times F_1 \times F_2 \times F_3, \quad (3.1)$$

where the term Nu_r (see Section 3.2 and Figures 2 and 3) is defined as the average Nusselt number based on the following reference conditions,

- (i) constant fluid properties corresponding to bulk fluid conditions,
- (ii) tube bank having 10 tube rows with at least 6 tubes in each row,
- (iii) tube bank normal to oncoming flow,
- (iv) $L/D > 5$

The terms F_1 , F_2 and F_3 are factors to account for the effect on $\overline{Nu}$ of departures from the reference conditions given in the following Table. (No information is available for $L/D < 5$).

Factor	Description	Section	Figure
F_1	Correction factor for the variation of fluid properties due to difference between T_w and T_b .	3.3	5
F_2	Correction factor for a bank having $N \neq 10$.	3.4	6a, 6b
F_3	Correction factor for banks having inclined tubes.	3.5	7

Thus the mean Nusselt number for a particular tube bank is found by evaluating the various terms on the right-hand side of Equation (3.1) as described in the following Sections. The mean heat transfer coefficient is then given by

$$\bar{h} = \bar{Nu} \lambda_b / D$$

and the heat transfer rate is

$$\dot{Q} = \bar{h} \Delta T \pi D L \text{ (number of tubes)}$$

where ΔT is the appropriate mean temperature difference between the tube wall and the bulk of the crossflow fluid, evaluated by such methods as given in Reference 1 for example.

When large temperature changes occur in the fluid flowing over the bank it is more accurate to assess the heat transfer by splitting the bank into sections containing a few tube rows where the use of mean temperatures does not lead to significant errors in the temperature driving force or physical properties. Information relating to the heat transfer for individual rows is discussed in Section 4.

3.2 Calculation of Nu_r (Figures 1 to 4)

The variations Nu_r of with Re_b and Pr_b covering gases and liquids are given by Figures 2a–2c and Figure 4 for various in-line tube banks and Figures 3 and 4 for staggered tube banks. Figure 1 gives a guide to enable the appropriate curve in Figure 2 to be chosen for in-line tube arrangements. Values of Nu_r given by Figures 2–4 are represented by the equation

$$Nu_r = a Re_b^m Pr_b^{0.34} \quad (3.2)$$

where Re_b and Pr_b are based on bulk fluid conditions. Values of the parameters a and m are given in Table 8.1 and represent a best fit to the available data over the quoted ranges of Reynolds number.

Because Re_b is based on the maximum velocity between tubes, V_{max} , which in turn is dependent on the transverse tube spacing, the effect of X_t on the variation of $\bar{Nu}$ with Re_b for a given Pr_b is minimised. However, an effect of tube row spacing on the parameters in Equation (3.2) remains for in-line tube banks with small longitudinal pitching. The grouping of the data in Table 8.1 for various tube row spacings, together with the dependency of the parameters on Re_b , is discussed in Appendix A in relation to the uncertainty of the data.

Note that in the evaluation of Nu_r for banks of tubes inclined to the axis of the duct, values of V_{max} or G_{max} used in the calculation of Re_b should be estimated assuming that the tubes are not inclined. The factor F_3 (Section 3.5) makes allowance for the effect on $\overline{Nu}$ of inclined tubes.

3.3 Calculation on Fluid Property Variation Factor F_1 (Figure 5)

The values of F_1 to be used in Equation (3.1) for both staggered and in-line tube banks are given in Figure 5. This factor takes into account the effect on Nu_r of variable fluid properties due to the difference in temperature between the bulk fluid and the tube wall. It is given by the empirical equation

$$F_1 = \left[\frac{Pr_b}{Pr_w} \right]^{0.26} \quad (3.3)$$

which was derived from an analysis of data for both staggered and in-line tube banks, for values of Pr_b up to about 600, and can be applied in practice for $Re_b > 10$.

Equation (3.3) is applicable to both gases and liquids and within the scatter of the available data appears to be valid for both heating and cooling cases.

3.4 Calculation of Bank Depth Factor F_2 (Figure 6)

Values of the factor $F_2 = \overline{Nu}_N / \overline{Nu}_{N=10}$ to be used in Equation (3.1) are given in Figures 6a and 6b for in-line and staggered tube banks respectively. This factor accounts for the effect on the average Nusselt number of the number of tube rows (in the direction of the flow) being greater or less than 10.

Values of F_2 were derived by summing and averaging the mean data from measurements carried out on individual tubes in a bank (see Section A1.3 of Appendix A). The method by which the row to row variation of Nu_n can be obtained is discussed in Section 4.

Equations relating F_2 and N in the form

$$F_2 = C_0 + C_1/N - C_2/N^2 + C_3/N^3 \quad (3.4)$$

are valid for $4 \leq N \leq 20$ with the following values of C_0, C_1, C_2 and C_3 . (For $N > 20$, the value of F_2 for $N = 20$ can be assumed.)

Arrangement	$(X_l - 1)/(X_t - 1)$	Re_b range	C_0	C_1	C_2	C_3
In-line	> 0.2	$> 2 \times 10^3$	0.990	0.873	9.60	18.6
In-line	0.5 to 2	10^2 to 10^3	1.055	0.548	14.7	37.3
Staggered	$X_l, X_t > 1.0$	$> 10^2$	1.025	0.093	4.06	6.60

3.5 Calculation of Tube Inclination Factor F_3 (Figure 7)

When the tube bank is inclined to the oncoming flow (see Figure 7) the heat transfer is decreased by a factor F_3 where

$$F_3 = \frac{\overline{Nu}_\theta}{\overline{Nu}_{\theta = 90^\circ}}$$

Values of the factor F_3 are given in Figure 7 and have been derived as described in Section A1.2 of Appendix A. For values of θ between 90° and 20° Figure 7 is represented by the equation

$$F_3 = (\sin \theta)^{0.6} \quad (3.5)$$

and this can be assumed to apply to all tube banks for values of Re_b between about 10^2 and 10^6 .

For values of θ less than 10° it is recommended that the problem be regarded as one of internal flow (in the space along and between tubes) using data from References 2-4 and an equivalent hydraulic diameter derived from the cross-sectional area between the tubes in the axial direction.

4. MEAN HEAT TRANSFER DATA FOR SEPARATE ROWS

The row to row variation in heat transfer depends in general on the spacing between tube rows and the Reynolds number. In most cases the heat transfer from tubes in the first row is considerably less than that from succeeding rows. Turbulence created by the first few rows of tubes causes an increase in the heat transfer coefficient on inner rows although the heat transfer becomes more or less constant after the first 3 or 4 rows. An exception to this is for in-line banks at low level Reynolds numbers. In this case the first row contributes a greater proportion to the overall heat transfer because of a shielding effect on subsequent rows, particularly where the longitudinal spacing between tubes is small compared with the lateral spacing.

For most cases the heat transfer for a particular row may be obtained in a similar way as for a bank (Equation (3.1)) except that the factor F_2 is replaced by a new factor F_4 , relating the performance of a particular row (the n th) to the average for a 10 row bank. Thus

$$\begin{aligned} Nu_n &= F_4 \overline{Nu}_N = 10 \\ &= Nu_r \times F_1 \times F_3 \times F_4 \end{aligned} \quad (4.1)$$

where Nu_n is the mean Nusselt number for the n th row and F_4 is a function of n . Values of the factor F_4 are given in Figures 8a and 8b for in-line and staggered tube banks respectively.

For in-line tube banks the data of Figure 8a for $Re_b > 2 \times 10^3$ are applicable for arrangements where $(X_l - 1)/(X_t - 1) > 0.2$. The data of Figure 8a for Reynolds numbers between 10^2 and 10^3 may be tentatively applied for values of $(X_l - 1)/(X_t - 1)$ in the range 0.5 to 2. For more detail of the special case when $(X_l - 1)/(X_t - 1)$ is very small reference should be made to Derivation 22.

The data for staggered tube banks were derived for $Re_b > 10^2$ and, within the scatter of data between the various sources, can be assumed independent of Re_b and tube spacing.

The accuracy of the data presented in Figures 8a and 8b is discussed in Section A1.3 of Appendix A.

5. OTHER FACTORS INFLUENCING CALCULATION OF HEAT TRANSFER

5.1 Effect of Free Convection

Two factors should be distinguished. Firstly, that due to the distribution of buoyancy forces arising from temperature variations in the fluid close to the tube surface; no experimental data exist for this effect with turbulent crossflow in tube banks, although it is known that in flow through vertical pipes such forces can significantly affect turbulence production and hence heat transfer. Secondly, in the case of a crossflow heat exchanger in which the tubes are horizontal and the flow vertical, there will of course be a "chimney effect" so that the change in average density of the fluid must be accounted for in, for example, the determination of pressure drop across the heat exchanger.

5.2 Effect of Thermal Radiation

For all cases of liquid heating, and for most cases of gas heating, thermal radiation will be negligible compared with forced convection. In cases where large surface to surface temperature differences occur, or where surface to gas temperature differences are very large and the gas has significant absorption characteristics (e.g. CO₂ and H₂O), a special investigation will be required (see, for example, Reference 1).

5.3 Effect of Vibration

No account has been taken in this Item of the effect on the heat transfer of vibration of the tubes or of sound waves propagated through the fluid. Frequencies in certain narrow band ranges can have a significant effect, usually to increase the heat transfer, so that if this effect is ignored performance will err on the safe side as far as surface area calculations are concerned.

6. EXAMPLE

A rectangular duct, 0.95 m by 5 m is used to carry an airflow of 12 000 kg/h across a simple plain-tube heat exchanger consisting of 5 rows of tubes containing 20 tubes in each row each having an outside diameter of 2.5 cm. Steam at atmospheric pressure is condensed within the tubes which have an effective length of 5 m and are arranged on a 4.5 cm equilateral triangular pitch. What will be the rise in temperature of the air across the tube bank due to forced convection heat transfer assuming the tube wall temperature to be 100°C and the air inlet temperature to be 20°C?

The steps of the calculation procedure (see Section 9) will be followed.

(i) $T_{w,m} = 100^{\circ}\text{C}.$

The mean bulk temperature of the air flow across the bank is not known but a value of $T_{b,m} = 30^{\circ}$ will be assumed as a first approximation.

(ii) For air the following physical properties apply at $T_{b,m}$ and $T_{w,m}$.

$$c_{pb} = c_{pw} = 1.013 \times 10^3 \text{ J/kg K},$$

$$\mu_b = 19 \times 10^{-6} \text{ kg/m s}, \quad \mu_w = 22 \times 10^{-6} \text{ kg/m s},$$

$$\lambda_b = 26.4 \times 10^{-3} \text{ W/m K}, \quad \lambda_w = 31.6 \times 10^{-3} \text{ W/m K}.$$