Tube banks, single-phase heat transfer in
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TUBE BANKS, SINGLE-PHASE HEAT TRANSFER IN.

Following from: Crossflow Heat Transfer; Tube banks, cross-flow over

Tube banks are employed in a wide range of heat exchangers. The average heat flux, $\dot{q}$, is expressed as a linear function of the mean temperature difference between the bulk of the fluid in the tube bank and the wall, according to a *rate equation*,

$$\dot{q} = \frac{\dot{Q}}{A} = \bar{\alpha}\Delta T_M \qquad \textbf{(1)}$$

where $\dot{Q}$ is the rate of heat transfer, $\bar{\alpha}$ is an overall heat transfer coefficient, and ΔT_M is some representative temperature difference between the tube wall and the bulk of the fluid. A is the total heat transfer area, which for plain tubes is given by,

$$A = N\pi DL \qquad \textbf{(2)}$$

where N is the number of tubes of outer diameter D, and length L. The overall heat transfer coefficient, $\bar{\alpha}$ is non-dimensionalized in terms of an average **Nusselt Number**, Nu, according to,

$$\overline{Nu} \equiv \frac{\bar{\alpha}D}{k} \qquad \textbf{(3)}$$

or alternatively as an average **Stanton Number**, S*t*,

$$\overline{St} equiv \frac{\bar{\alpha}}{\rho c_p u_m} \qquad \textbf{(4)}$$

In tube banks, Nu is frequently correlated according to,

$$\overline{Nu} = c\,Re^m \cdot Pr^n \left(\frac{Pr_w}{Pr}\right)^p \qquad \textbf{(5)}$$

where **Reynolds Number** Re, is defined by,

$$Re = \frac{\rho u_{max} D}{\eta} \qquad \textbf{(6)}$$

u_{max} is the bulk velocity in the minimum cross-section, ρ and η are the fluid density and viscosity respectively. The Pr is just,

$$Pr = \frac{\eta c_p}{\lambda} \qquad \textbf{(7)}$$

where c_p is the specific heat, and λ is thermal conductivity. The rate of heat transfer may also be expressed in terms of an energy balance, which for single-phase heat transfer may be written,

$$\dot{Q} = \dot{C}\left(T_{out} - T_{in}\right) \tag{8}$$

where T_{in} and T_{out} are the bulk temperatures at the inlet and outlet of the bank, and $\dot{C}$ is the thermal capacitance of the fluid, presumed constant over the range of interest.

$$\dot{C} = \dot{M} c_p \tag{9}$$

$\dot{M}$ is the total mass flow rate through all of the passages in the tube bank, and c_p is averaged over the range of interest.

A local heat transfer coefficient may be defined in a similar fashion to Eq. 1 The local Nu is a function of a number of parameters such bank type and geometry, flow Re, pressure gradient, location within bank etc. Figure 1 shows local Nu distributions in the interior of tube banks. It can be seen that the staggered-geometry Nu-distribution is similar to the single cylinder case, with a maximum occurring at $\phi = 0°$. For the in-line case, Nu rises to a maximum at the reattachment point around 45°. It can also be seen that heat transfer is somewhat higher for both inline and staggered tube banks, than for single cylinders. This is due to increased free-stream turbulence (as a result of preceeding rows) and shear, due to constriction of the flow passages. For most applications the engineer is not concerned with the details of local heat transfer in tube banks and the reader is referred to the reader is referred to _ Žukauskas and Ulinskas (1988) for further information on the subject .

Experimental data and empirical correlations

Two major sources of data, on both fluid flow and heat transfer, are the Delaware group in the USA, and the group in the Republic of Lithuania. The work at the University of Delaware on overall heat transfer in tube banks has formed the basis for heat exchangers calculations for many years while the recent book by Žukauskas et al. (1988) is a comprehensive reference on heat transfer in tube banks based on many years' research, containing extensive information on local and overall heat transfer. It appears that the Delaware data was gathered at nominally constant wall temperature, T_w, conditions and that of the Lithuanian group at constant wall heat flux, $\dot{q}_w$. Numerous other data also exist, in addition to these two main sources.

A number of empirical correlations may also be found: The Engineering Science and Data Unit correlate Nu vs. Re according to Eq. 5 (ESDU, 1978). Values of the coefficients c and m are given in Table I. Re is based on u_{max}. The correlation of Žukauskas and Ulinskas (1988) agrees well with experimental data, but exhibits poor continuity across certain Re ranges. Several other correlations may be found in the literature. Agreement among heat transfer correlations is, generally-speaking better than for pressure drop correlations. In most correlations, Nu is treated as being primarily a function of Re and configuration; ie. the pitch-to-diameter ratios are often treated as being relatively unimportant over a wide range. Similarly the effect of thermal boundary conditions (constant T_W vs. constant $\dot{q}_w$), though often significant, is usually simply ignored.

Tradition is to plot Nu or St in the form of log-log plots with both the Pr-dependence and the property-variation being removed by defining,

$$j' = j \left(\frac{Pr_w}{Pr}\right)^{-p} = St Pr^{1-n} \left(\frac{Pr_w}{Pr}\right)^{-p} \qquad \textbf{(10)}$$

$$k' = k \left(\frac{Pr_w}{Pr}\right)^{-p} = Nu Pr^{-n} \left(\frac{Pr_w}{Pr}\right)^{-p} \qquad \textbf{(11)}$$

The heat-transfer factors j, k, (un-primed) are Pr-independent. Primed values j', k' indicate that the influence of *property variations* across the boundary layer has also been taken into consideration. The Delaware group chose $n = 1/3$, ie. j is the so-called *Colburn heat transfer factor*. ESDU (1973) and Žukauskas and Ulinskas (1988) propose similar but slightly higher values n = 0.34 and 0.36, respectively (over most of the range). Figure 2 shows j and k as a function of Re for Pr = 1. It can be seen that heat transfer is a little higher for staggered than in-line geometries at lower Re.

The influence of fluid properties is accounted for in Eq. 5 by the term $(Pr_w/Pr)^p$, where Pr is evaluated at the mean bulk temperature, and Pr_w at T_w. The Delaware group use $(\eta_w/\eta)^p$, the matter is largely one of preference (but their value of p = 0.14 was based on the *Seider-Tate correlation* for flow inside tubes). ESDU (1973) suggest p = 0.26 regardless of whether the fluid is being heated or cooled.

The choice of reference temperature is a matter for concern; since in most situations, the engineer must use a mean value for thermal design/analysis. For constant T_w, the appropriate temperature to use in Eq. 1 is the *log-mean temperature difference*,

$$\Delta T_M = \Delta T_{LM} \equiv \frac{\Delta T_{out} - \Delta T_{in}}{\ln\left(\frac{\Delta T_{in}}{\Delta T_{out}}\right)} \quad \textbf{(12)}$$

where, $\Delta T_{in} = T_{in} - T_w$ and $\Delta T_{out} = T_{out} - T_w$ are the differences between the fluid and wall temperatures at the inlet and outlet, respectively. (See also **Mean Temperature Difference)** The reference temperature, T_M, for ennumerating property values should be $T_w + \Delta T_{LM}$. For constant $\dot{q}_w$, the arithmetic-mean temperature should be used,

For most applications it is immaterial whether properties, ρ, η, etc. are evaluated at the log-mean or the arithmetic-mean temperature, and the latter is often employed for convenience.

$$T_M = \overline{T} \equiv \frac{T_{out} + T_{in}}{2} \quad \textbf{(13)}$$

For large numbers of banks, and low values of the product Re.Pr the choice of T_M is quite important.

Empirical correlations are obtained from data for idealised tube banks. In practice Nu is a function of a number of other parameters: These are often accounted for by writing,

$$\overline{Nu} = k_1 k_2 \overline{N'u} \qquad \textbf{(14)}$$

where Nu' is the idealised-correlation-based Nusselt number, and the k-coefficients account for deviations from this situation. One example, already mentioned, is the factor $(Pr_w/Pr)^p$. Other factors could include the effects of (a) finite numbers of rows, (b) angle of attack, or other considerations. These are discussed below.

Finite number of rows

Idealised correlations, described above, are for deep tube banks, where the number of rows, N_{row}, is large. For each of the first few rows in a tube bank, heat transfer may be substantially different (usually less) than occurs deep in the bank. 3 shows the correction factor k_1 as a function of N_{row}.

Inclined Crossflow

In many situations, the flow is not one of pure crossflow, ie. $\beta < 90°$. Heat transfer is reduced from by a factor, k_2. 3 shows k_2 vs. β.

Two commonly-employed methods of enhancing heat transfer in tube banks are the use of rough surfaces and finned tubes. (**See Augmentation of Heat Transfer**)

Rough tubes

Rough tubes are used to increase the ratio of heat transfer to drag by increasing turbulence. The effects appear to be more pronounced for staggered than in-line geometries. Žukauskas and

Ulinskas, (1988) propose the following prescription for staggered geometries,

$$\begin{aligned} Nu &= 0.5\left(\frac{a}{b}\right)^{0.2} Re^{0.65} Pr^{0.36} \left(\frac{Pr_w}{Pr}\right)^{0.25} \left(\frac{k}{d}\right)^{0.1} \quad 10^3 \le Re \le 10^5 \\ Nu &= 0.1\left(\frac{a}{b}\right)^{0.2} Re^{0.8} Pr^{0.4} \left(\frac{Pr_w}{Pr}\right)^{0.25} \left(\frac{k}{d}\right)^{0.15} \quad 10^5 \le Re \le 2x10^6 \end{aligned} \qquad \textbf{(15)}$$

where k is the roughness height.

Finned tubes

External fins may be employed with gas-crossflow, in order to increase the heat-transfer area (internal fins are also used sometimes). The book by Stasiuleviius and Skrinska (1988) is devoted to the subject, as is a chapter in Ž̆ukauskas et al. (1988). When using *finned tubes* in a design, the engineer must use $\overline{Nu}$ correlations appropriate to finned-tubes, and properly account for temperature variations throughout the fin.

Nu correlations for finned-tube banks

A number of general-purpose correlations for finned tube-banks may be found; Gnielinski et al. in the Heat Exchanger Design Handbook (1983), ESDU (1984, 1986), Stasiuleviius and Skrinska (1988), and Ž̆ukauskas and Ulinskas (1988). These typically assume forms similar to Eq. 5, but with additional parameters to account for the fin geometry. The engineer should obtain Nu correlations specific to the particular geometrical configuration under consideration, whenever possible.

Fin conduction

When variation in temperature across the fins is significant, the rate equation, Eq. 1, is re-written as,

$$\dot{Q} = E_f A \overline{\alpha} \Delta T_M \qquad \textbf{(16)}$$

A is the total area for heat transfer, $A = A_f + A_w$, where A_f is the total area of the fins and A_w that of the exposed tube-wall. E_f is a surface effectiveness factor,

$$E_f = \frac{\eta_f A_f + A_w}{A} \qquad \textbf{(17)}$$

and η_f is a fin-efficiency, ie. the ratio of the actual rate of heat transfer through the fins to that which would occur if all fin material were at constant T_w.

Figure 5 is a plot of η_f vs. non-dimensional height h^*, at various diameter ratios, D_f'/D. λ_f is the thermal conductivity of the fin, δ the fin thickness and the average heat transfer coefficient. Primed values, h' and D', indicate that they are adjusted for heat transfer through the fin-tip, as indicated in Figure 5. Charts of η_f vs. h^* have been devised for a variety of geometries; the article by Gnielinski et al. in the Heat Exchanger Design Handbook (1983) contains a selection. These should be used in preference to Figure 5, which was obtained from the one-dimensional fin equation solution,

$$\eta = \frac{\tanh h^*}{h^*} \qquad \textbf{(18)}$$

where h' is calculated according to Schmidt (Stasiuleviius and Skrinska, 1988).

Bypass Effects

Bypass streams are relatively ineffective in transferring heat, and are usually treated as adiabatic, ie. only the main flow stream is used to compute heat transfer. Since the temperature and hence the viscosity differs from that in the main flow lanes, this must be taken into consideration when computing the bypass and main mass-flow rates.

Methods for calculating the thermal performance of tube banks

How to compute the performance of a tube bank will depend to a larger extent on which of the parameters, $\dot{Q}$, $\overline{\alpha}$, A, T_{in}, T_{out}, etc. are prescribed or required, whether fluid properties are constant, and the precision required. It is common to differentiate between `rating' and `sizing' of heat exchangers. This section is intended to provide some guidelines for the reader, not a single general-purpose procedure. Two cases will be considered in detail: (a) Plain tubes at constant T_w and (b) non-isothermal plain or finned tubes:

Plain tubes, constant T_w:

The log-mean temperature-difference (LMTD) approach is based on the fundamental assumption that the heat flux is directly proportional to the temperature difference between the bulk of the fluid at some interior location and the wall , ie. the rate equation, Equation 1, with ΔT_M prescribed according to Equation 12.

Unless all temperatures are known, a priori, a trial-and-error method based on successive applications of the rate equation combined with a heat balance is required. An alternative approach, due to Nusselt, is to postulate the rate of heat transfer to be proportional to the difference between the inlet and wall temperatures,

$$\dot{Q} = E\dot{C}(T_w - T_{in}) \qquad \textbf{(19)}$$

The *effectiveness*, *E*, represents the ratio of the actual to the maximum possible heat transfer. For constant T_w ,

$$E = 1 - e^{-\frac{A\bar{\alpha}}{\dot{C}}} \qquad \textbf{(20)}$$

where the quantity $A\bar{\alpha}/\dot{C}$ is known as the number of transfer units, NTU. The advantage of T_{in} as reference (if known) is that no iteration is required provided fluid properties are constant and upstream values can be used: Even if these vary, the E-NTU-method is often preferable, as T_M affects the calculation of via property variations only. A typical procedure, is as follows,

1. Given T_{in} guess a value for T_{out} (unless known).

2. Calculate Re and Pr based on property values evaluated at the arithmetic-mean bulk temperature, and Pr_w at T_w.
3. Compute the value of Nu' for an ideal bank using Eq. 5 applying corrections, k_1, k_2 etc. as discussed above, to obtain Nu and hence for the actual bank [Eq. 14].
4. Compute the overall rate of heat transfer, , using one of two methods: Either calculate ΔT_{LM} Eq.12, and use the rate equation Eq. 1 to obtain , or calculate E from using Eq. 20 and obtain using Eq. 19.
5. Compute the exit bulk temperature, T_{out} from , by means of a heat balance, Eq. 8.
6. Re-iterate steps 2-6 based on the new value of T_{out} (unless prescribed) until satisfactory convergence is obtained.

If changes in temperature are large, break the bank up into a individual or groups of rows, and proceed in a sequential fashion from the inlet to the exit.

Plain or finned tubes, general case

In many practical engineering applications, the tube-side fluid also undergoes significant changes in temperature. Under these circumstances, Eq. 1 is no longer appropriate, and the rate of heat transfer is,

$$\dot{Q} = UA\Delta T_M \qquad \textbf{(21)}$$

where U, the **Overall heat Transfer Coefficient** based on the outer surface A (including fins) is given by,

$$\frac{1}{U} = \frac{1}{(E_f\,\alpha)_{bank}} + \left(\frac{R_f}{E_f}\right)_{bank} + R_w + \frac{A}{A_i}\left(\frac{R_f}{E_f}\right)_{tube} + \frac{A}{A_i}\frac{1}{(E_f\,\alpha)_{tube}} \qquad \textbf{(22)}$$

where the subscript `bank' refers to the external bank-side crossflow, and the subscript `tube' refers to the internal tube-side longitudinal flow. The terms on the right-hand side of Eq. 22 may be regarded as resistances to the flow of heat in the presence of a temperature difference. A_i is the total area for heat transfer on the inner tube surface, $E_{f\,bank}$ and $E_{f\,tube}$ are fin-surface-effectiveness correction factors for externally and internally-finned tube surfaces (equal to unity for plain tubes), and $R_{f\,bank}$ and $R_{f\,tube}$ are fouling factors which may be estimated using the recommendations in the Standards of the **Tubular Exchanger Manufacturers Association** (TEMA, 1988). R_w is the tube-wall resistance which for plain tubes, of length L, is given by,

$$R_w = \frac{1}{2\pi\lambda_w L}\ln\frac{D}{D_i} \qquad \textbf{(23)}$$

where D_i is the inner diameter, and λ_w is the wall thermal conductivity. For finned tubes, R_w is configuration-dependent.

Commonly-employed heat exchanger design methods are (1) the LMTD-based F-correction-factor method, (2) the P-NTU method and (3) the θ method. For the F-correction method, ΔT_{LM}, is defined by Eq. 12, but with ΔT_{in} and ΔT_{out} the temperature differences between the two working fluids at the inlet and outlet. Heat transfer is reduced by a factor F, which is a function of the capacitance ratio, R, and the thermal effectiveness, P, as defined in Table II, which also gives with the modified rate equation used to calculate . The P-NTU method, presumes the thermal effectiveness, P, (similar to E) to be a function of R, as well as the number of transfer units, NTU = $AU/_{cold}$, of the cold fluid. For the third method, $\theta = \theta(R, NTU_{cold})$ but a modified energy balance, instead of a modified rate equation (see Table II) is used to compute . Charts of F , P, and θ may be found in standard references on heat exchangers, often in combined form (see Taborek, *Heat Exchanger Design Handbook, 1983*). Regardless of which method is used, two Nu correlations are needed to calculate *U*: One, as per Eq. 5, to calculate the bank-side crossflow heat transfer coeffiecient, $\overline{\alpha}_{bank}$, the other to calculate the internal tube-side heat transfer coefficient, $\overline{\alpha}_{tube}$, (See **Tubes Single Phase Heat Transfer in**). The temperatures at the solid-fluid interfaces, $T_{w\,bank}$

and $T_{w\ tube}$, are obtained as a linear combination of $T_{M\ bank}$ and $T_{M\ tube}$, using the ratio of the resistances in Eq. 22 as weighting factors. T_w may be taken as being the temperature between the fluid and the fouling resistance. A typical calculation would then proceed as follows.

1. Given $T_{in\ bank}$ and $T_{in\ tube}$, guess values for $T_{out\ bank}$,$T_{out\ tube}$, $T_{w\ bank}$, $T_{w\ tube}$ (unless known) .
2. Calculate Re and Pr for tube-side and bank-side fluids at the arithmetic-mean bulk temperatures, and values of Pr_w for the two fluids based on $T_{w\ bank}$ and $T_{w\ tube}$.
3. Compute Nu_{bank} and Nu_{tube} using the appropriate correlations applying corrections k_1, k_2 as necessary, to obtain $_{bank}$ and $_{tube}$. Calculate the overall heat transfer coefficient, U , Eq. 22
4. Compute the overall rate of heat transfer, : Either calculate P and R based on the guessed values and obtain F(P,R) or compute R and NTU_c to obtain P(R,NTU_c) or θ(R,NTU_c) from a chart. Calculate from the formulae in Table II.
5. Compute $T_{out\ bank}$ and $T_{out\ tube}$ from , by means of heat balances applied to both tube-side and bank-side fluids Eq. 8. Hence calculate $T_{w\ bank}$ and $T_{w\ tube}$ (a single value, T_w , will often suffice).
6. Re-iterate steps 2-5 based on the new values of $T_{out\ bank}$, $T_{out\ tube}$ (if necessary) until satisfactory convergence is obtained.

The choice of whether to use the F-correction, P-NTU, or θ methods will depend on the application and may result in substantial simplifications to the general procedure detailed above. If all temperatures are known, the F-correction method can be used with properties enumerated at T_M. Conversely, if fluid property variations are negligible and upstream values may be used, the P-NTU and θ methods can be used advantageously. If property variations are significant, but small enough to be considered perturbations about a mean, all the above lumped-parameter-type schemes will necessarily be iterative. For large variations in T_M, a numerical scheme is preferred.

Numerical schemes

While traditional methods are meritorious, the majority of heat exchangers are now designed using computer-software. Many approaches are possible; for example the *E-NTU* method may be performed over a discrete number of cells corresponding to one or more cylinders: An alternative procedure is to start from first principles, and solve pairs of equations of the form,

$$\begin{aligned} \dot{C}_{bank}\,\Delta T_{bank} &= U\Delta A(T_{tube} - T_{bank}) \\ \dot{C}_{tube}\,\Delta T_{tube} &= -U\Delta A(T_{tube} - T_{bank}) \end{aligned} \qquad \textbf{(24)}$$

on a cell-by-cell basis. If an upwind-difference scheme is used,

$$\begin{aligned} \Delta T_{bank}(i,j) &= \{T_{bank}(i,j) - T_{bank}(i-1,j)\} \\ \Delta T_{tube}(i,j) &= \{T_{tube}(i,j) - T_{tube}(i,j-1)\} \end{aligned} \qquad \textbf{(25)}$$

and upstream values are used to ennumerate properties, no iteration is required. Better accuracy can be achieved with a central-difference scheme,

$$\begin{aligned} \Delta T_{bank}(i,j) &= \frac{1}{2}\{T_{bank}(i+1,j) - T_{bank}(i-1,j)\} \\ \Delta T_{tube}(i,j) &= \frac{1}{2}\{T_{tube}(i,j+1) - T_{tube}(i,j-1)\} \end{aligned} \qquad \textbf{(26)}$$

Some iteration is now necessary, though by proceeding as indicated in Figure 5, a simple Jacobi point-by-point method will converge rapidly. It is now possible to account for variations of U due to changes in T_b on a cell-by-cell basis. T_w may also be obtained, at an intermediate calculation in each cell to enumerate the dependance of U on Pr_w. Variations in other properties may be dealt with on a cell-by-cell basis as necessary, depending on the nature of the problem.

The rate equations defined Eq. 23 may be considered like a set of simple enthalpy-conservation equation (convection-source-term equations). It is possible to devise much more sophisticated two-phase enthalpy equations, which account for bank-side mixing, 3D phenomena etc., in addition to bulk convection It is also possible to generate momentum equations to account for flow-related effects, bypassing, finite-number of rows, etc. For complex flows, the definition of U may be of limited use. Numerical methods differ from traditional methods in that there is no need to rely on the premiss that heat transfer is governed by a rate equation. The reader is referred to the articles by D.B. Spalding in the Heat Exchanger Design Handbook (1983), and elsewhere. The use of spreadsheet programs, specialised heat-exchanger-design programs and general-purpose computational fluid dynamics software should be considered prior to writing source code.

Simple lumped-parameter type schemes once formed the basis for all practical heat exchanger design. The situation is changing, and the application of computer software, is not only increasing in use and range of application, but is also enhancing our understanding of the physics of flow within the passages of heat exchangers.

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Leading to:

Cross reference terms

Rough tubes
Inclined tubes
FInned tubes
Thermal performance in tube banks
Numerical methods

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	In-line a ≥ 1.15, 1.2 ≤ b ≤ 4		Staggered a ≥ 1.15, 1.2 ≤ b ≤ 4	
***Re* Range**	***c***	***m***	***c***	***m***
10- 3x10^2	**0.742**	**0.431**	**1.309**	**0.360**
3x10^2- 2x10^5	**0.211**	**0.651**	**0.273**	**0.635**
2x10^5- 2x10^6	**0.116**	**0.700**	**0.124**	**0.700**

Table I Coefficients used to calculate overall heat transfer in tube banks using Eq. 5 with $n = 0.34$, p = 0.26. From ESDU International plc, 1973.

Dependent Variable	Independent Variables	Rate of Heat Transfer
F	$R = \frac{T_{in\ hot} - T_{out\ hot}}{T_{out\ cold} - T_{in\ cold}}$ $P = \frac{T_{out\ cold} - T_{in\ cold}}{}$	$\dot{Q} = UAF\Delta T_{LM}$
P	$R,\ NTU_{cold} = \frac{AU}{\dot{C}_{cold}}$	$\dot{Q} = P\dot{C}_{cold}(T_{in\ hot} - T_{in\ cold})$
θ	$R,\ NTU_{cold}$	$\dot{Q} = UA\theta(T_{in\ hot} - T_{in\ cold})$

Table II Design methods commonly used to calculate thermal performance in heat exchangers

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Figure 1 Variation of local heat transfer around 1) a single tube 2) a tube in a staggered bank 3) a tube in an in-line banks. From Žukauskas and Ulinskas (1988)

Figure 2 Average heat transfer for in-line and staggered tube banks. From ESDU (1973).

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Figure 3 Influence of number of rows on overall heat transfer in tube banks. From Žukauskas and Ulinskas (1988).

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Figure 4 Influence of angle of inclination, β, on overall heat transfer for inclined crossflow in tube banks. From ESDU (1973).

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Figure 5 Fin Efficiency

Figure 6 Discretized version of tube-bank heat exchanger