Tube banks, crossflow over
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TUBE BANKS, CROSSFLOW OVER.

Following from: Cross flow; External flows, overview; Tubes, crossflow over.

Tube banks are commonly-employed design elements in heat exchangers. Both plain and finned tube banks are widely found. Tube bundles are a sub-component in shell-and-tube heat exchangers, where the flow resembles crossflow at some places, and longitudinal flow elsewhere. (The term tube *bank* is often used, in the literature, to denote a crossflow situation and *bundle* to indicate longitudinal flow, however this convention is far from universal.) The flow may be *single-phase* or *multi-phase*; boilers and condensers containing tube banks find a wide range of applications in industry, in addition there may be *combustion* eg. in a furnace heat exchanger.

Figure 1 shows the two basic tube-bank patterns involving either a rectangular or a rhombic primitive unit. These are referred to as *in-line tube banks* and *staggered tube banks* respectively. These are characterized by cross-wise and stream-wise pitch-to-diameter ratio's, *a* and *b,*

$$a \equiv \frac{s_T}{D} \tag{1}$$

$$b \equiv \frac{s_L}{D} \tag{2}$$

where D is the cylinder diameter, s_T the cross-wise (transverse) pitch, and s_L the stream-wise pitch. Most commonly encountered tube banks are the in-line square (a = *b*), rotated square (a = 2b) and equilateral triangle (a = 2b/$\sqrt{3}$). Tube banks with the product axb < 1.25^2 are referred to as compact, while those with axb > 4 are considered widely-spaced.

Analytical expressions for the flow of an ideal fluid in in-line and staggered tube banks have been derived in the form of power series (Beale, 1993). For widely spaced banks, the pressure coefficients are similar to the sinusoidal distribution observed in single cylinders. For compact banks there are significant differences in both in-line and staggered tube banks, in the latter case two additional pressure extrema can occur at $\phi = \pm 45°$, where ϕ is the angle measured from the front of the cylinder.

The flow of a real fluid in the body of a tube bank also resembles flow past a single cylinder, though with significant differences. The flow Reynolds number, *Re*, is defined by,
where ρ is the fluid density, $\_{max}$ is the maximum bulk velocity ie. the bulk velocity in the minimum cross-section (see Fig. 1), *D* is the cylinder diameter, and η is the fluid viscosity. At low *Re*, the flow is laminar with separation occurring at around $\phi = 90°$, resulting in stable vortices forming behind each cylinder. For staggered banks, the upstream flow is typically a maximum between the preceding tubes, so the impinging flow bifurcates at the front-leading edge of each cylinder. For in-line banks, cylinders are in a comparatively dead zone downstream of the preceding cylinder's wake, and there are two re-attachment points at around $\phi = \pm 45°$.

$$\mathrm{Re} \equiv \frac{\rho \bar{u}_{\mathrm{max}} D}{\eta} \qquad \textbf{(3)}$$

As Re increases, vortices are shed from each cylinder in an alternate fashion. Due to the overall pressure gradient, transient motion tends to occur at higher Re than for single tubes, particularly for compact banks. The number of vortices present and the motion of the streams depends on a and b; neighbouring streams may be in-phase, out-of-phase, or uncorrelated. Wake-switching is observed in staggered tube banks, while in in-line banks there is an instability in the shear-layer between the wake and main flow with some of the detached vortices being entrained by the free-stream flow.

As *Re* further increases, the wake becomes turbulent, and there is substantial free-stream turbulence, due to the influence of the preceding upstream rows. The boundary layer region remains laminar up to the critical *Re* of around 2×10^5, at which point transition to turbulence occurs and the separation point moves downstream. The engineer should be aware that both vortex-shedding and turbulent buffeting can induce vibrations in tube banks. (*See Vibration in Heat Exchangers*.)

The most widely adopted *reference bulk velocity* for use in Eq. 2 is the bulk interstitial velocity which occurs in the *minimum cross-section* ie. the maximum bulk velocity, $\bar{u}_{max}$, Bergelin et al. (1950, 1958), Žukauskas et al. (1988), ESDU (1974) . Some authors eg. Kays and London (1984) base *Re* on a hydraulic radius. The Engineering Sciences Data Unit, ESDU (1979), base *Re* on the so-called *mean superficial velocity*, $\bar{U}_{mean}$, ie. the average velocity that would occur if the tubes were removed, (see Figure 1). The mean superficial velocities is useful when simulating large-scale flow in heat exchangers. It can readily be shown that,

$$\bar{u}_{max} = \left(\frac{a-1}{a}\right)\bar{U}_{mean} \tag{4}$$

for all in-line banks, and staggered banks with $a < 2b^2 - ½$. For some compact staggered banks, where $a > 2b^2 - ½$, the minimum cross-section occurs across the diagonal and so,

$$\bar{u}_{max} = \frac{\sqrt{4b^2 + a^2} - 2}{a}\bar{U}_{mean} \tag{5}$$

The hydrodynamic parameter of most interest to the heat exchanger designer is the *overall pressure loss coefficient*, often expressed in terms of an *Euler number, Eu*,

$$\mathrm{Eu} \equiv \frac{\Delta \bar{p}_{row}}{\frac{1}{2}\rho\,\bar{u}_{max}^2} \tag{6}$$

where $\Delta \bar{p}_{row}$ is the mean pressure drop across a single row.

Numerous alternative definitions abound in the literature, for instance Bergelin et al. (1950, 1958) define a *friction factor* f = 4Eu. Others simply use the symbol *f* to denote Eu. The friction factor of Kays and London (1984) is such that
f = (a-1)Eu/π for large banks. ESDU (1979) use a different definition for pressure loss coefficient, again, based on $\overline{U}_{mean}$, ie there has been little effort to-date to standardize parameters.

Many *experimental studies* have been conducted on flow in tube banks, starting in the early part of this century. Comprehensive data were gathered by workers at the University of Delaware in the 40's and 50's. Their data were primarily in the low and intermediate range of Re. These were summarized in two reports, Bergelin et al. (1950, 1958), containing original data, and several papers (see ESDU, 1974) for a concise bibliography of experimental data).

A group at the Institute of Physical and Technical Problems of Energetics of the Lithuanian Academy of Sciences, have also published a large number of papers on both flow and heat transfer in tube banks over a wide *Re* range. The book by Žukauskas et al. (1988) contains detailed discussions on numerous aspects of the subject, while the article by Žukauskas (1987) is a substantially shorter, but comprehensive review of fluid flow and heat transfer in tube banks. Charts of *Eu* vs. *Re* are provided for in-line and staggered banks at various pitch ratios. Achenbach has conducted research on tube banks in the high *Re* turbulent flow regime (for references, see Žukauskas et al.,1987)

Various *empirical correlations* of pressure drop data have been devised over the years. The Eu vs. Re correlations of the Lithuanian group are commendable; they have been reconciled with numerous sources of externally-gathered experimental data, in addition to data gathered by the authors themselves. They are included here. Others such as those based on the Delaware groups work, could equally-well have been reproduced, having formed an integral part of the thermal design of shell-and-tube exchangers, in the West for many years; see the articles by Taborek in the Heat Exchanger Design Handbook (1983) and Mueller in the Handbook of Heat Transfer (Rohsenhow and Hartnett, 1972).

Figures 2 and 3 show Eu *vs.* Re (based on $\_{max}$) for in-line square and equilateral triangle tube banks. The Heat Exchanger Design Handbook (1983) contains analytical expressions approximating these curves. These take the form of a power series,

$$Eu = \sum_{i=1}^{4} \frac{c_i}{Re^i} \quad \textbf{(7)}$$

for all in-line and rotated square banks, except in-line banks with b = 2.5, for which,

$$Eu = \sum_{i=1}^{4} c_i \; Re^i \quad \textbf{(8)}$$

Values of the coefficients, c_i for in-line and staggered tube banks are given in Tables 1 and 2, respectively. The reader is cautioned that these equations render poor continuity across certain ranges of application. Other mathematical correlations also exist, for example, ESDU (1979). Agreement between ESDU/Delaware, and the Lithuanian group's curves is not particularly good, especially in the low-intermediate *Re* range (Beale, 1993).

In most practical tube banks it is necessary to modify *Eu* for several effects. This usually done as follows,

$$Eu = k_1 k_2 k_3 \ldots k_n E'u \quad \textbf{(9)}$$

where Eu' is the value calculated from the correlation for an ideal bank, and the k_i are correction factors. Corrections are typically required to account for the geometry, size, and location of the tube bank, deviations from normal incidence of the working fluid, and variations in the fluid properties due to temperature and pressure changes. These are detailed below.

Influence of pitch ratio, a/b ≠ 1

This may be accounted for by multiplying Eu′ by k_1 as is shown in the insets to Figures 2 and 3. Mathematical expressions for k_1 may be found in the Heat Exchanger Design Handbook (1983).

Influence of temperature on fluid properties

For non-isothermal conditions it is necessary to account for variations in viscosity across the boundary layer. Many authors achieve this by setting,

$$k_2 = \left(\frac{\eta_w}{\eta}\right)^p \tag{10}$$

where η_w is the viscosity at T_w and η is the viscosity at the mean bulk temperature, T_M, of the tube bank. Žukauskas and Ulinskas (Heat Exchanger Design Handbook, 1983) recommend,

$$\begin{aligned} p &= 0.776\exp\left(-0.545\,Re^{0.256}\right) \quad \eta_w > \eta \\ p &= 0.968\exp\left(-1.076\,Re^{0.196}\right) \quad \eta_w < \eta,\ Re < 10^3 \end{aligned} \tag{11}$$

For many applications, the temperature effects on ρ, μ etc. are negligible, and it is sufficient to calculate these at the arithmetic mean of the inlet and exit bulk temperatures (if known). However if the temperature change in the bank is such as to affect the fluid properties significantly, it is necessary to compute Re and Eu on a row-by-row basis.

Entry-length effects

If the number of rows, N_{row}, in the stream-wise direction are sufficiently small, it is necessary to modify Eu for entrance-length effects. Figure 4 shows the entrance-length factor k_3, as a function of N_{row}.

Deviations from normal incidence

The flow may deviate from pure crossflow in either of two ways: (i) The angle of attack, α, may not be 0° ie. the bank may be rotated at some arbitrary angle to the flow. (ii) There may be a component in the azimuthal (longitudinal) *z*-direction, $\beta \neq 90°$, sometimes referred to as inclined crossflow.

Rotated crossflow, $\alpha \neq 0°$

Butterworth (1978) used the analogy of fluid flow in porous media and noted the mean pressure drop to be nominally the same for in-line and rotated banks with *a/b* ≈ 1, concluding the overall pressure gradient to be independent of α. Thus the mean pressure drop may be calculated using the normal incidence correlations. The physical significance of the reference velocity $\overline{u}_{max}$ is vague, however it may still be may still be calculated from the mean superficial velocity $\overline{U}_{mean}$ using Eq. 4 or 5. $\overline{U}_{mean}$ should be regarded as a *local volume-averaged* quantity (see 5)

$$\overline{U}_{mean} = \left| \frac{1}{V} \int_V \vec{u}\, dV \right| \qquad \textbf{(12)}$$

where the volume V includes the space occupied by the cylinders. Since Re and Eu are based on velocity and pressure gradients in the α-direction, the normal component of the pressure drop is reduced by a factor,

$$k_4 = \cos\alpha \qquad \textbf{(13)}$$

It being understood that there is also a pressure gradient in the crosswise-direction. For a/b ≠ 1 anisotropy may be a problem, but the α-direction flow-resistance can still be estimated from the normal-incidence, α = 0°, 90°, values.

Inclined crossflow, $\beta \neq 0°$

For this case the flow resistance is anisotropic, ie. p is not in the β-direction, due to the axial (longitudinal) drag being less than the cross-wise component. 6 shows k_5 vs. β, where k_5 is the ratio of the normalised streamwise component of the pressure drop to the value if the same mass flow had been in pure crossflow.

Rough surfaces

The use of roughened and other enhanced surfaces to increase heat transfer is widespread. In addition to being deliberately employed, surface roughness tends to naturally increase when smooth tubes become fouled. When the mean height of the roughness elements, k*,* is sufficient, turbulence is enhanced. It is generally maintained that the influence of surface roughness is more significant in staggered banks than in-line banks. The Heat Exchanger Design Handbook (1983) contains some guidelines for calculating k_5 as a function of k/D*,* for staggered tube banks.

Finned tubes

Banks employing finned tubes are often found. 8 is a schematic showing some of the more common arrangements, with examples of circular, spiral, axial and plate-fins. It is common to distinguish between so-called low-finned and high-finned tubes, according to the fin-height to diameter ratio. Fins may be straight or tapered. Correlations for Eu vs. Re*,* or equivalent, may be found in The Heat Exchanger Design Handbook (1983), Z̆ukauskas et. al, (1988), Stasiulevi_ius and Skrinska (1988), ESDU (1984, 1986) and elsewhere.

Bypassing

Tube banks seldom extend to the walls of the containing vessel: Also in staggered tube banks, alternate cylinders are missing near a wall. (Experimentalists may employ dummy half-tubes or corbels to eliminate this effect.) Near the wall the flow is faster, due to decreased resistance. An iterative procedure, similar to Wills and Johnson's method, in shell-and-tube heat exchanger analysis (Hewitt et al., 1994) is recommended. Let the subscripts `bank' and `bypass' refer to the main and bypass flow lanes, as illustrated in 7, and suppose that,

$$\Delta p = \Delta p_{bypass} = \Delta p_{bank} \tag{14}$$

The fractional mass flow through the bank, F_{bank} is,

$$F_{bank} = \frac{\dot{M}_{bank}}{\dot{M}_{total}} = \left(\frac{1}{1 + 2\frac{n_{bank}}{n_{bypass}}} \right)^{\frac{1}{2}} \tag{15}$$

where,

$$n = \frac{1}{2}\frac{N_{row}}{\rho A^2} Eu \tag{16}$$

The procedure is iterative because Eu_{bank} and Eu_{bypass} are functions of $\dot{M}_{bank}$ and $\dot{M}_{bypass}$. The pressure drop may be calculated from either n_{bank}, n_{bypass}, or the average value, n,

$$\Delta p = \bar{n}\,\dot{M}_{total}^2 = \frac{n_{bank}\, n_{bypass}}{\left(2\sqrt{n_{bank}} + \sqrt{n_{bypass}}\right)^2}\dot{M}_{total}^2 \tag{17}$$

Eu_{bank} is obtained using Eq. 9, Figures 2 or 3, in the usual fashion. Bypass-pressure drop correlations are uncommon, and often proprietary (but see ESDU, 1974). As a rough approximation, Eu_{bypass} may be estimated from 2 assuming effective values of *a* and b, namely, a = $2s_{bypass}/D$, b = s_L (inline bank) or a = $2s_{bypass}/D$, b = $2s_L/D$ (staggered bank). (NB: In the latter case the effective number of rows is $N_{row}/2$.)

Equation 14 is based on the premise that pressure variations across the inlet and outlet are insignificant, something which may or may not be the case. For this reason, the engineer should attempt to use data incorporating bypass effects, if available. The lack of such data suggest that more research is needed on this important subject.

The procedure for calculating the overall pressure drop in tube banks is as follows;

1. Calculate Re based on values of u_{max}, ρ, μ evaluated at arithmetic mean of inlet and outlet bulk temperatures
2. Obtain the value of Eu for an ideal bank using Figs. 2 and 3
3. Correct Eu for factors k_1, k_2, k_3, k_4, k_5, as discussed above

If either variations in fluid properties due to changes in temperature obtained from a heat transfer calculation (*see Tube Banks, Single-phase heat transfer in*), or bypass effects are significant, the calculation procedure are iterative.

Numerical studies

Numerical studies have gained popularity in recent years. It is convenient to differentiate between (a) detailed calculations of flow within the passages of tube banks and (b) overall performance calculations.

Detailed calculations of flow within the passages of tube banks

Many results have now been obtained for tube banks using both finite-volume and finite-element methods. Numerical methods have been used to simulate laminar transient flow, large-eddy turbulent flow, inclined flow $\beta \neq 0°$, and also 3D secondary flow effects in finned tubes. Beale (1993) contains a review of recent work.

Most detailed numerical simulations have been conducted for either laminar or high-speed turbulent regimes. Far less numerical-data are available in the intermediate Re range, in which most heat exchangers operate, where the influence of free-stream turbulence is of paramount importance: Because of the complex nature of flow within the passages of tube banks, and the inadequacies of existing turbulence models, numerical experiments have not yet, and probably never will, render laboratory work obselete. The two activities are not however, mutually exclusive, and many of the problems described above are readily amenable to the methods of computational fluid dynamics.

Overall performance calculations

These methods range from simple automatations of earlier methods to detailed three-dimensional flow calculations using the techniques of **Computational Fluid Dynamics**. Overall performance predictions are still based on empirically-based correlations of pressure drop [Eqs. 7 and 8] and heat transfer. However these are embedded in a computer code which is used to predict the overall performance of the heat exchanger as a whole. Important effects such as entrance phenomena, bypassing, variable properties etc. can thus readily be accommodated. In modern heat-exchanger design, computer-based methods have already supplanted hand-calculation techniques to some extent, a trend which will doubtless continue in the future. The challenge is to incorporate the physics and engineering experience into future heat exchanger design software.

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Leading to:

<u>Cross reference terms</u>

In-line tube banks
Staggered tube banks
Inclined tubes
Rotated tubes
Entry-length effects
Finned tubes
Rough tubes
Bypassing

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Figure 1 Schematic of (a) an inline and (b) a staggered tube bank illustrating nomenclature, and showing location of minimum cross-section.

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Figure 2 Pressure drop coefficient vs. Reynolds number for in-line tube banks. From Heat Exchanger Design Handbook (1983).

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Figure 3 Pressure drop coefficient vs. Reynolds number for staggered tube banks. From Heat Exchanger Design Handbook (1983).

Figure 4. Influence of finite-number of rows on overall pressure drop in tube banks. Adapted from Heat Exchanger Design Handbook (1983).

Figure 5. Rotated tube bank showing angle of attack, α, with respect to mean superficial velocity.

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Figure 6 Effect of angle of attack, _, on overall pressure drop for inclined crossflow in tube banks. From Žukauskas et al. (1988).

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Figure 7 Schematic showing main and bypass streams in a tube bank, together with a network diagram.

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Figure 8 Some typical types of finned tubes.

a	*Re* Range	c_0	c_1	c_2	c_3	c_4
1.25	$3\text{-}2x10^3$	0.272	$0.207x10^3$	$0.102x10^3$	$-0.286x10^3$	-
1.25	$2x10^3\text{-}2x10^6$	0.267	$0.249x10^4$	$-0.927x10^7$	$0.10x10^{11}$	-
1.5	$3\text{-}2x10^3$	0.263	$0.867x10^2$	$-0.202x10^0$	-	-
1.5	$2x10^3\text{-}2x10^6$	0.235	$0.197x10^4$	$-0.124x10^8$	$0.312x10^{11}$	$-0.274x10^{14}$
2	7-800	0.188	$0.566x10^2$	$-0.646x10^3$	$0.601x10^4$	$-0.183x10^5$
2	$800\text{-}2x10^6$	0.247	-0.595	0.15	-0.137	0.396

Table I Coefficients, c_i, for use in Eqs. 7 and 8 to generate pressure drop coefficients for in-line square banks. From Heat Exchanger Design Handbook (1983).

b	*Re* Range	c_0	c_1	c_2	c_3	c_4
1.25	$3\text{-}10^3$	0.795	$0.247x10^3$	$0.335x10^3$	$-0.155x10^4$	$0.241x10^4$
1.25	$10^3\text{-}2x10^6$	0.245	$0.339x10^4$	$-0.984x10^7$	$0.132x10^{11}$	$-0.599x10^{13}$
1.5	$3\text{-}10^3$	0.683	$0.111x10^3$	$-0.973x10^2$	$-0.426x10^3$	$-0.574x10^3$
1.5	$10^3\text{-}2x10^6$	0.203	$0.248x10^4$	$-0.758x10^7$	$0.104x10^{11}$	$-0.482x10^{13}$
2	$7\text{-}10^2$	0.713	$0.448x10^2$	$-0.126x10^3$	$-0.582x10^3$	-
2	$10^2\text{-}10^4$	0.343	$0.303x10^3$	$-0.717x10^5$	$0.88x10^7$	$-0.38x10^9$
2	$10^4\text{-}10^6$	0.162	$0.181x10^4$	$0.792x10^8$	$-0.165x10^{13}$	$0.872x10^{16}$
2.5	$10^2\text{-}5x10^3$	0.33	$0.989x10^2$	$-0.148x10^5$	$0.192x10^7$	$-0.862x10^8$
2.5	$5x10^3\text{-}2x10^6$	0.119	$0.498x10^4$	$-0.507x10^8$	$0.251x10^{12}$	$-0.463x10^{15}$

Table II Coefficients, c_i, for use in Eq. (7) to generate pressure drop coefficients for equilateral triangle banks. From Heat Exchanger Design Handbook (1983).