

## Design Procedure

- Calculate the area for cross-flow  $A_s$  for the hypothetical row of tubes at the shell equator, given by

$$A_s = \frac{(p_t - d_o)D_s l_B}{p_t}$$

where  
 $l_B$  = baffle spacing, m.

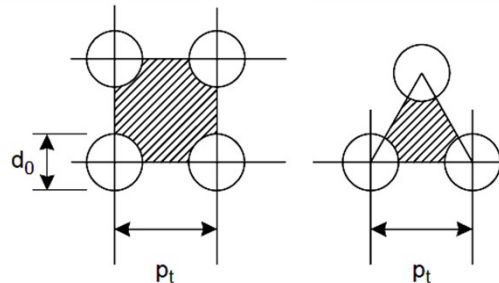
The term  $(p_t - d_o)/p_t$  is the ratio of the clearance between tubes and the total distance between tube centers.

- Calculate the shell-side mass velocity  $G_s$  and the linear velocity  $u_s$ :

$$G_s = \frac{W_s}{A_s} \quad u_s = \frac{G_s}{\rho}$$

where  
 $W_s$  = fluid flow rate on the shell-side, kg/s;

- Calculate the shell-side equivalent diameter (hydraulic diameter), as shown in Figure. For a square pitch arrangement:



$$d_e = \frac{4 \left( \frac{p_t^2 - \pi d_o^2}{4} \right)}{\pi d_o} = \frac{1.27}{d_o} (p_t^2 - 0.785 d_o^2)$$

For an equilateral triangular pitch arrangement:

$$d_e = \frac{4 \left( \frac{p_t}{2} \times 0.87 p_t - \frac{1}{2} \pi \frac{d_o^2}{4} \right)}{\frac{\pi d_o}{2}} = \frac{1.10}{d_o} (p_t^2 - 0.917 d_o^2)$$

- Calculate the shell-side Reynolds number, given by

$$Re = \frac{G_s d_e}{\mu} = \frac{u_s d_e \rho}{\mu}$$

- For the calculated Reynolds number, read the value of  $j_h$  from Figure for the selected baffle cut and tube arrangement, and calculate the shell-side heat transfer coefficient  $h_s$  from

$$Nu = \frac{h_s d_e}{k_f} = j_h Re Pr^{1/3} \left( \frac{\mu}{\mu_w} \right)^{0.14}$$

For the calculated shell-side Reynolds number, read the friction factor from Figure and calculate the shell-side pressure drop from

$$\Delta P_s = 8 j_f \left( \frac{D_s}{d_e} \right) \left( \frac{L}{l_B} \right) \frac{\rho u_s^2}{2} \left( \frac{\mu}{\mu_w} \right)^{-0.14}$$

The term  $(L/l_B)$  is the number of times the flow crosses the tube bundle =  $(N_b + 1)$ , where  $N_b$  is the number of baffles.

### Shell Nozzle-Pressure Drop

The pressure loss in the shell nozzles will normally be significant only with gases.

The nozzle pressure drop can be taken as equivalent to 1.5 velocity heads for the inlet and 0.5 for the outlet, based on the nozzle area or the free area between the tubes in the row immediately adjacent to the nozzle, whichever is the least.

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## Bell's method

- In Bell's method the heat-transfer coefficient and pressure drop are estimated from correlations for flow over ideal tube-banks, and the effects of leakage, bypassing and flow in the window zone are allowed for by applying correction factors.
- This approach will give more satisfactory predictions than Kern's method; and, as it takes into account the effects of leakage and bypassing, can be used to investigate the effects of constructional tolerances and the use of sealing strips.
- The procedure in a simplified and modified form to that given by Bell (1963), is outlined below.
- The method is not recommended when the by-pass flow area is greater than 30% of the cross-flow area, unless sealing strips are used.

## Heat-transfer coefficient

The shell-side heat transfer coefficient is given by:

$$h_s = h_{oc} F_n F_w F_b F_L$$

The total correction will vary from 0.6 for a poorly designed exchanger with large clearances to 0.9 for a well-designed exchanger.

where  $h_{oc}$  = heat transfer coefficient calculated for cross-flow over an ideal tube bank, no leakage or bypassing.

$F_n$  = correction factor to allow for the effect of the number of vertical tube rows,

$F_w$  = window effect correction factor,

$F_b$  = bypass stream correction factor,

$F_L$  = leakage correction factor.

### hoc, ideal cross-flow coefficient

The heat-transfer coefficient for an ideal cross-flow tube bank can be calculated using the heat transfer factors  $j_h$  given in Figure.

The Reynolds number for cross-flow through a tube bank is given by:

$$Re = \frac{G_s d_o}{\mu} = \frac{u_s \rho d_o}{\mu}$$

The heat-transfer coefficient is given by:

$$\frac{h_{oc} d_o}{k_f} = j_h Re Pr^{1/3} \left( \frac{\mu}{\mu_w} \right)^{0.14}$$

**$F_n$ , tube row correction factor**

The mean heat-transfer coefficient will depend on the number of tubes crossed. Figure is based on data for ten rows of tubes.

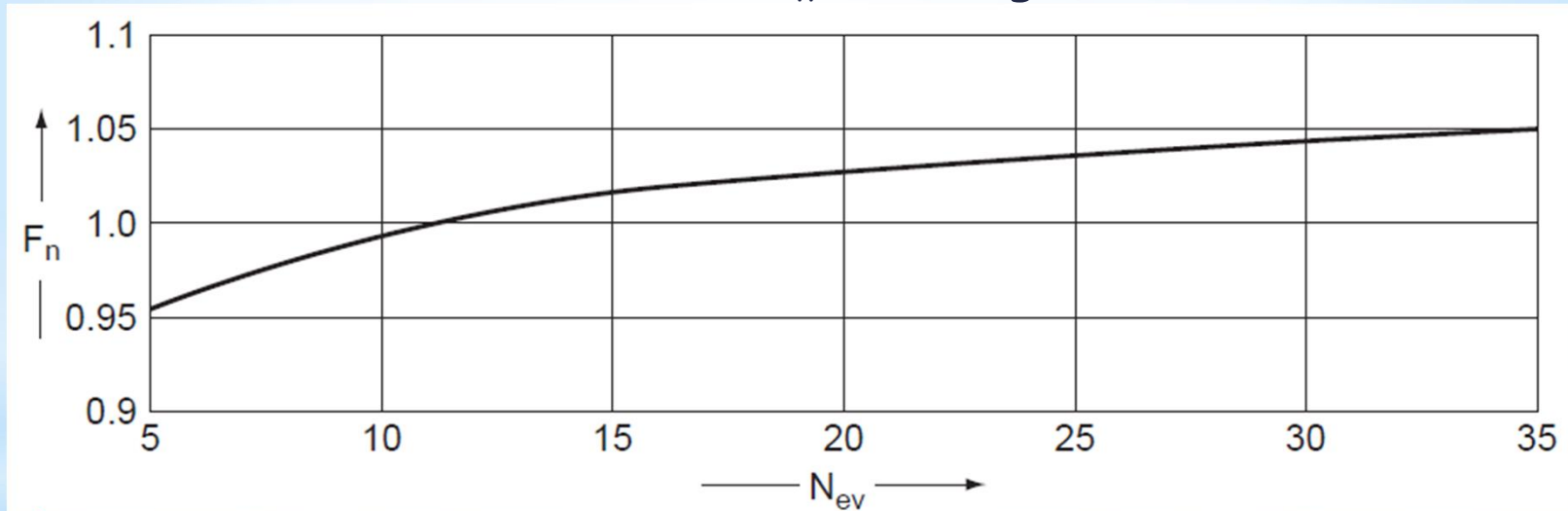
For turbulent flow the correction factor  $F_n$  is close to 1.0.

In laminar flow the heat-transfer coefficient may decrease with increasing rows of tubes crossed, due to the build up of the temperature boundary layer.

The factors can be used for the various flow regimes; the factors for turbulent flow are based on those given by Bell (1963).

$N_{cv}$  is number of constrictions crossed = number of tube rows between the baffle tips; see Figure.

1.  $Re > 2000$ , turbulent; take  $F_n$  from Figure below



2.  $Re > 100$  to 2000, transition region, take  $F_n = 1.0$ ;

3.  $Re < 100$ , laminar region,  $F_n \propto (N'_c)^{-0.18}$

where  $N'_c$  is the number of rows crossed in series from end to end of the shell, and depends on the number of baffles.

## $F_w$ , Window Correction Factor

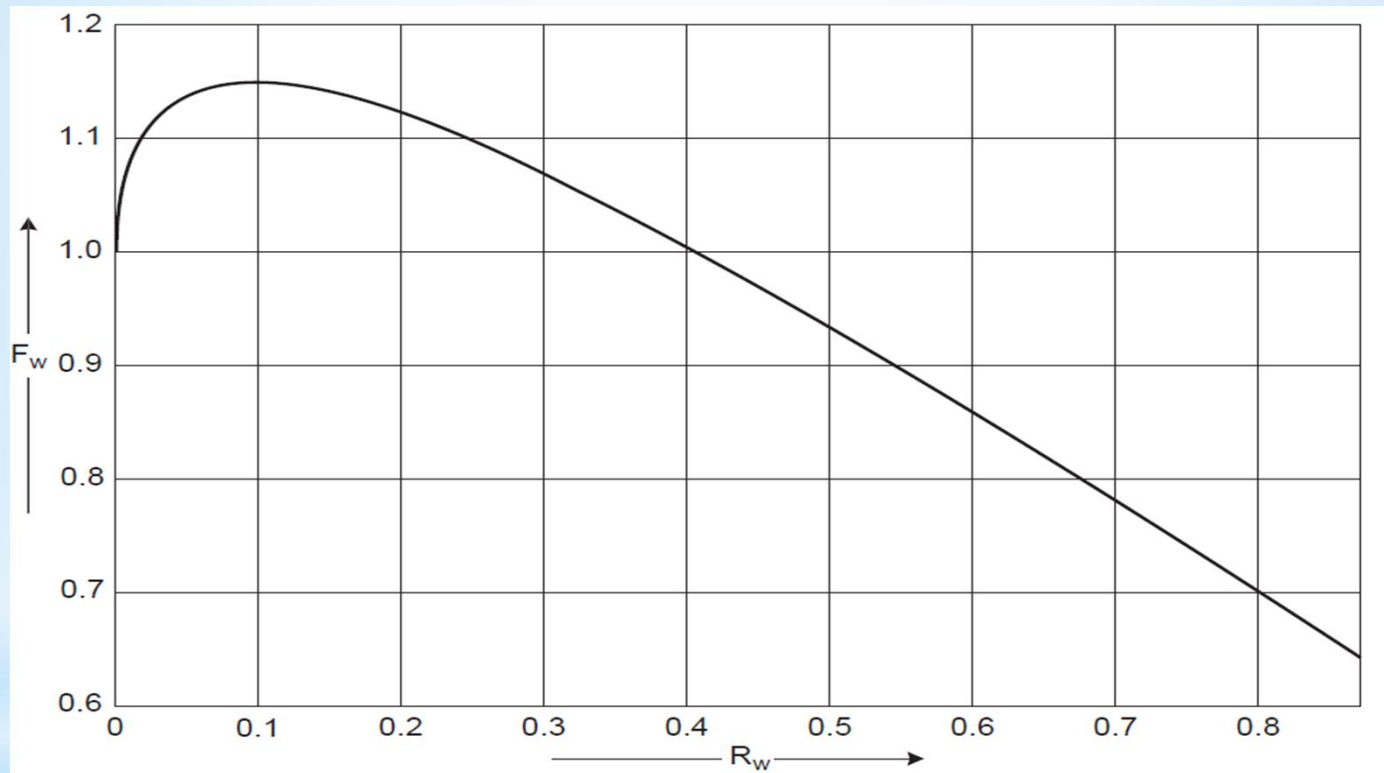
The window correction factor corrects for the effect of flow through the baffle window and is a function of the heat transfer area in the window zones and the total heat transfer area.

The correction factor is shown in Figure plotted versus  $R_w$ ,

The ratio of the number of tubes in the window zones to the total number in the bundle, determined from the tube layout diagram.

For preliminary calculations,  $R_w$  can be estimated from the bundle and window cross-sectional areas .





## $F_b$ , Bypass Correction Factor

The bypass correction factor corrects for the main bypass stream.

the flow between the tube bundle, and the shell wall and is a function of the shell to bundle clearance and whether sealing strips are used:

$$F_b = \exp \left[ -\alpha \frac{A_b}{A_s} \left( 1 - \left( \frac{2N_s}{N_{cv}} \right)^{1/3} \right) \right]$$

where

$\alpha = 1.5$  for laminar flow,  $Re < 100$ ;

$\alpha = 1.35$  for transitional and turbulent flow  $Re > 100$ ;

$A_b$  = clearance area between the bundle and the shell, see Figure 12.41 and Section 12.9.5;

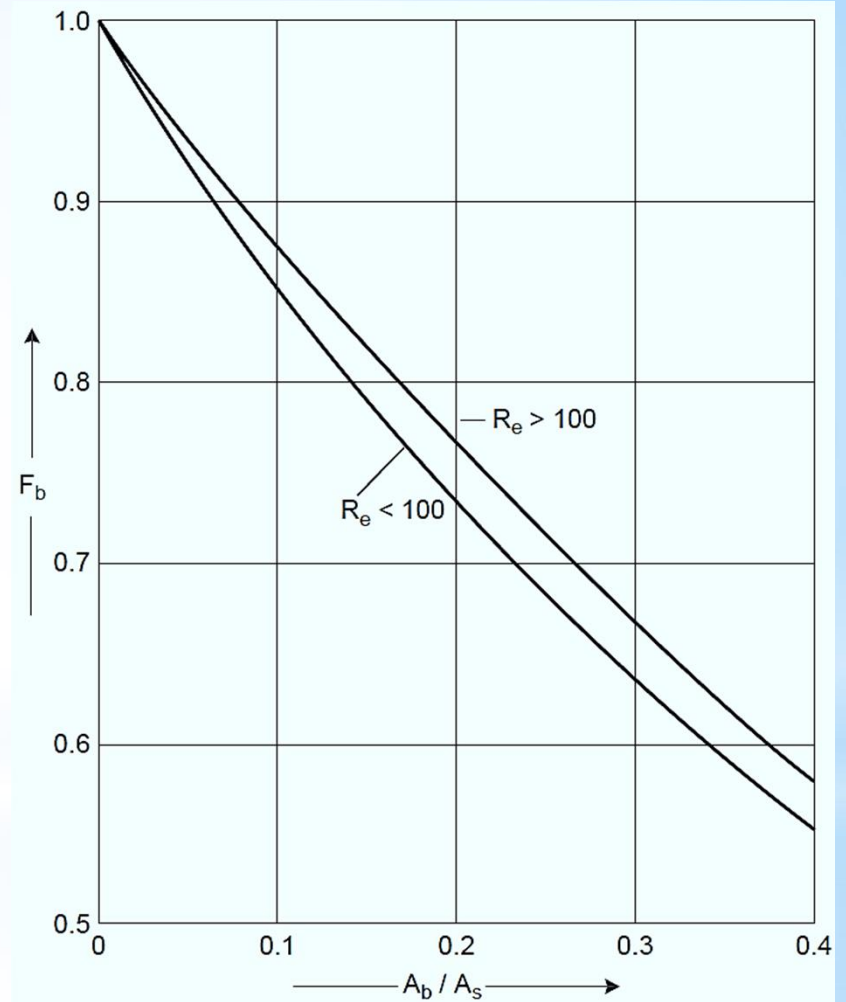
$A_s$  = maximum area for cross-flow, equation 12.21;

$N_s$  = number of sealing strips encountered by the bypass stream in the cross-flow zone;

$N_{cv}$  = the number of constrictions, tube rows, encountered in the cross-flow section.

Above Equation applies for  $N_s \leq N_{cv}/2$ .

Where no sealing strips are used,  $F_b$  can be obtained from Figure.



## $F_L$ , Leakage Correction Factor

The leakage correction factor corrects for the leakage through the tube-to-baffle clearance and the baffle-to-shell clearance:

$$F_L = 1 - \beta_L \left[ \frac{(A_{tb} + 2A_{sb})}{A_L} \right]$$

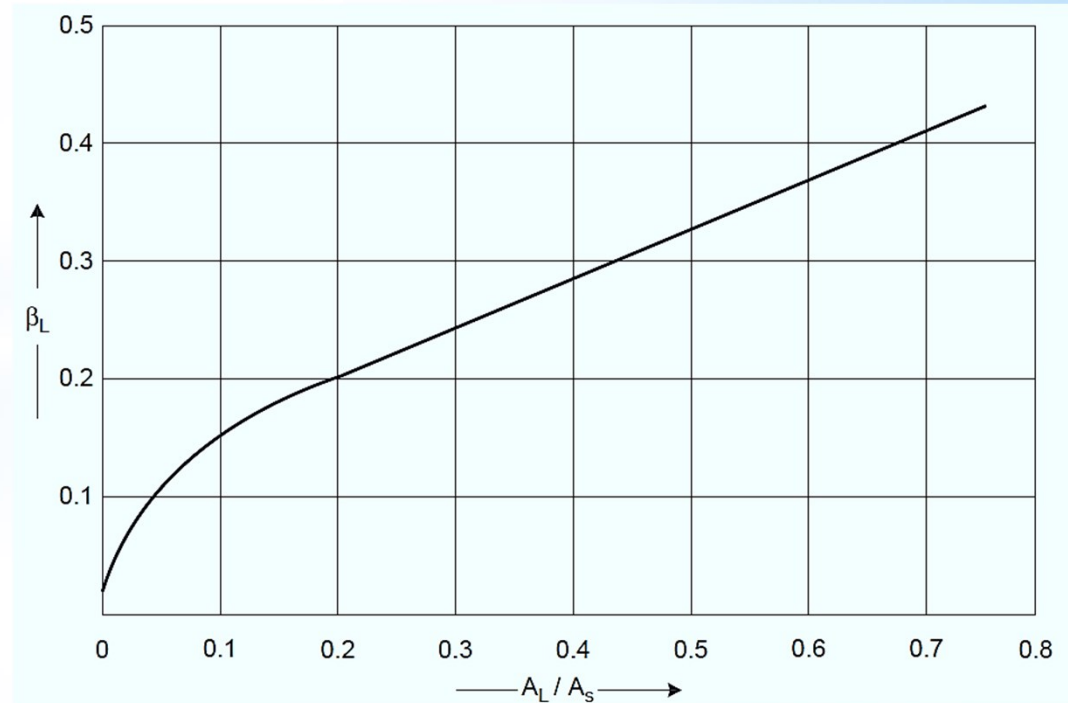
where

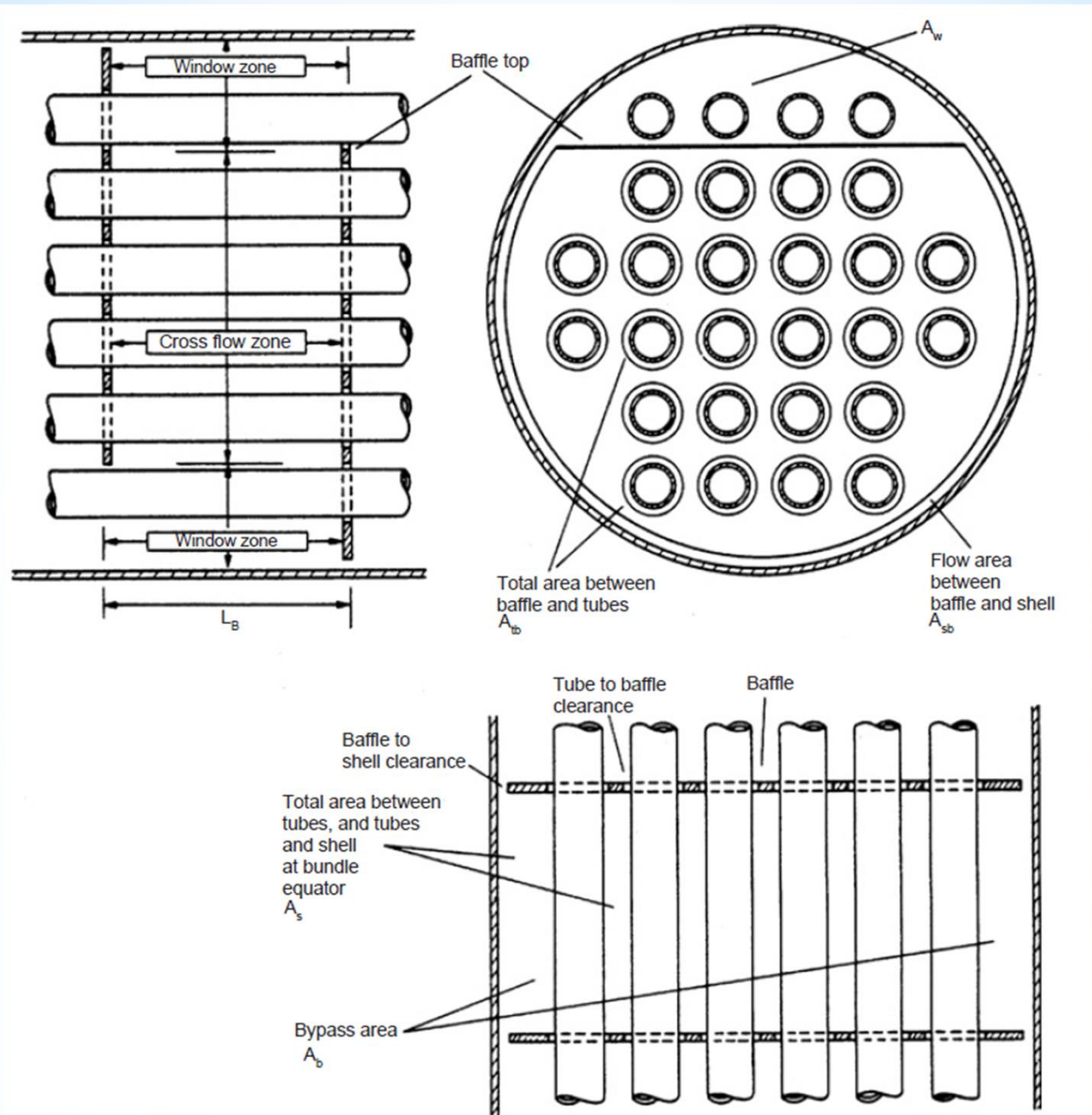
$\beta_L$  = a factor obtained from Figure 12.37;

$A_{tb}$  = the tube to baffle clearance area, per baffle, see Figure;

$A_{sb}$  = shell-to-baffle clearance area, per baffle, see Figure;

$A_L$  = total leakage area =  $(A_{tb} + A_{sb})$ .





## Pressure Drop

The pressure drops in the cross-flow and window zones are determined separately and summed to give the total shell-side pressure drop.

### Cross-Flow Zones

The pressure drop in the cross-flow zones between the baffle tips is calculated from correlations for ideal tube banks and corrected for leakage and bypassing:

$$\Delta P_c = \Delta P_i F'_b F'_L$$

where

$\Delta P_c$  = the pressure drop in a cross-flow zone between the baffle tips, corrected for bypassing and leakage;

$\Delta P_i$  = the pressure drop calculated for an equivalent ideal tube bank;

$F'_L$  = leakage correction factor;

$F'_b$  = bypass correction factor.

## $F'_b$ , Bypass Correction Factor for Pressure Drop

Bypassing will affect the pressure drop only in the cross-flow zones.

The correction factor is calculated from the equation used to calculate the bypass correction factor for heat transfer are used but with the following values for the constant  $\alpha$ :

Laminar region,  $Re < 100$ ,  $\alpha = 5.0$ ;

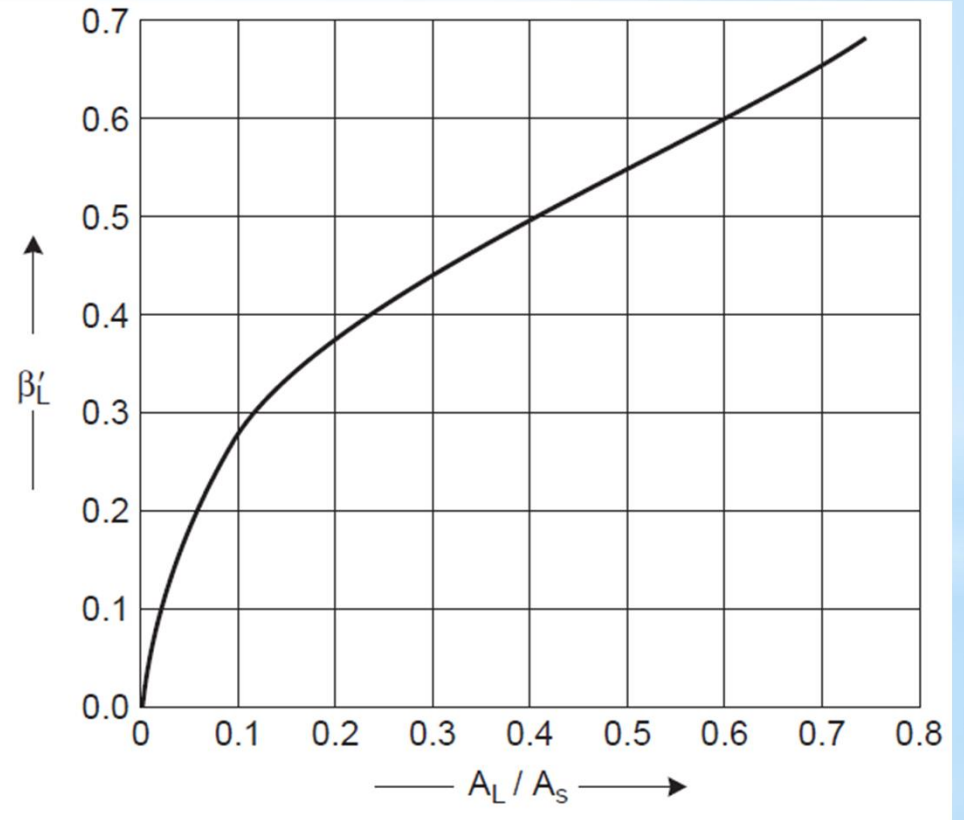
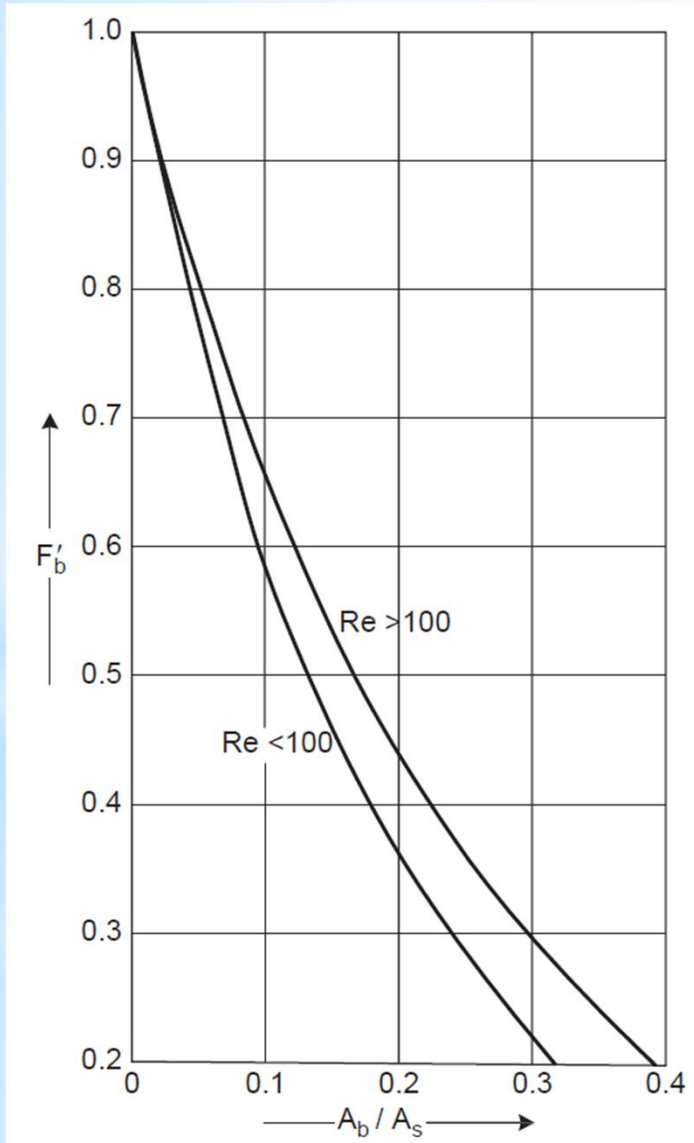
Transition and turbulent region,  $Re > 100$ ,  $\alpha = 4.0$ .

The correction factor for exchangers without sealing strips is shown in Figure.

## $F'_L$ , Leakage Factor for Pressure Drop

Leakages will affect the pressure drop in both the cross-flow and window zones.

The factor is calculated using the equation for the heat transfer leakage-correction factor with the values for the coefficient  $\beta'_L$  taken from Figure.



## Window-Zone Pressure Drop

Any suitable method can be used to determine the pressure drop in the window area.

Bell used a method proposed by Colburn. Corrected for leakage, the window drop for turbulent flow is given by

$$\Delta P_w = F'_L (2 + 0.6N_{ww}) \frac{\rho u_z^2}{2} \quad \text{where}$$

$u_z = \text{the geometric mean velocity;}$   
 $u_z = \sqrt{u_w u_s},$

$u_w = \text{the velocity in the window zone, based on the window area less the area occupied by the tubes } A_w;$

$$u_w = \frac{W_s}{A_w \rho};$$

$W_s = \text{shell-side fluid mass flow, kg/s;}$

$N_{ww} = \text{number of restrictions for cross-flow in window zone, approximately equal to the number of tube rows.}$



## End Zone Pressure Drop

There will be no leakage paths in an end zone (the zone between tube sheet and baffle).

Also, there will be only one baffle window in these zones, so the total number of restrictions in the cross-flow zone will be  $N_{cv} + N_{wv}$ .

The end zone pressure drop  $\Delta P_e$  will therefore be given by

$$\Delta P_e = \Delta P_i \left[ \frac{(N_{wv} + N_{cv})}{N_{cv}} \right] F'_b$$

## Total Shell-Side Pressure Drop

Summing the pressure drops over all the zones in series from inlet to outlet gives

$\Delta P_s = 2$  end zones +  $(N_b - 1)$  cross-flow zones +  $N_b$  window zones

$\Delta P_s = 2\Delta P_e + \Delta P_c(N_b - 1) + N_b\Delta P_w$  where  $N_b$  is the number of baffles =  $[(L/l_B) - 1]$ .

An estimate of the pressure loss incurred in the shell inlet and outlet nozzles must be added to total pressure drop equation.

### End Zone Lengths

The spacing in the end zones will often be increased to provide more flow area at the inlet and outlet nozzles.

The velocity in these zones will then be lower, and the heat transfer and pressure drop will be reduced slightly.

### Shell and Bundle Geometry

- The bypass and leakage areas, window area, and the number of tubes and tube rows in the window and cross-flow zones can be determined precisely from the tube layout diagram.
- For preliminary calculations, they can be estimated with sufficient accuracy by considering the tube bundle and shell geometry. With reference to Figures in slide 247 and 248:

$H_c$  = baffle cut height =  $D_s \times B_c$ , where  $B_c$  is the baffle cut as a fraction;

$H_b$  = height from the baffle chord to the top of the tube bundle;

$B_b$  = "bundle cut" =  $H_b/D_b$ ;

$\theta_b$  = angle subtended by the baffle chord, rads;

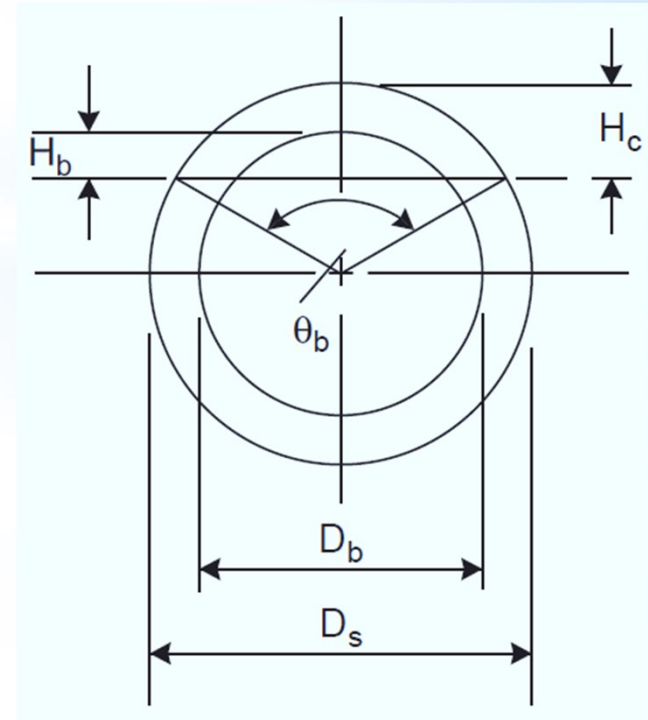
$D_b$  = bundle diameter.

Then:

$$H_b = \frac{D_b}{2} - D_s(0.5 - B_c)$$

$$N_{cv} = \frac{(D_b - 2H_b)}{p'_t}$$

$$N_{wv} = \frac{H_b}{p'_t}$$



where

$P'_+$  is the vertical tube pitch;

$P'_+ = p_+$  for square pitch;

$P'_+ = 0.87 p_+$  for equilateral triangular pitch.

The number of tubes in a window zone  $N_w$  is given by

$$N_w = N_t \times R'_a$$

where  $R'_a$  is the ratio of the bundle cross-sectional area in the window zone to the total bundle cross-sectional area;

$R'_a$  can be obtained from Figure, for the appropriate "bundle cut",  $B_b$ .

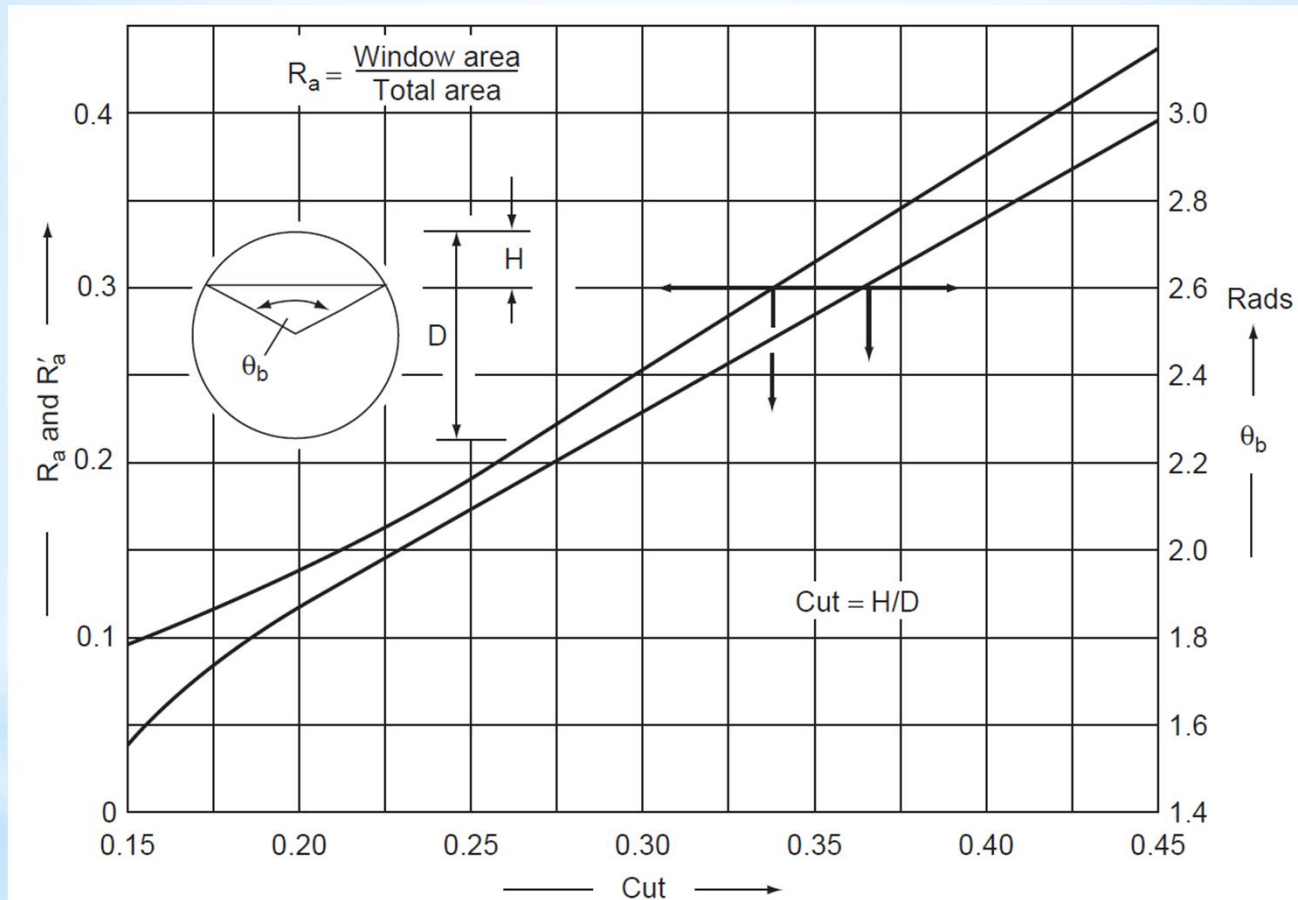
The number of tubes in a cross-flow zone  $N_c$  is given by

$$N_c = N_t - 2N_w$$

and

$$R_w = \frac{2N_w}{N_t}$$

$$A_w = \left( \frac{\pi D_s^2}{4} \times R_a \right) - \left( \frac{N_w \pi d_o^2}{4} \right)$$



$R_a$  is obtained from Figure, for the appropriate baffle cut,  $B_c$

$$A_{tb} = \frac{c_t \pi d_o}{2} (N_t - N_w)$$

where  $c_t$  is the diametrical tube-to-baffle clearance, the difference between the hole and tube diameter, typically 0.8 mm.

$$A_{sb} = \frac{c_s D_s}{2} (2\pi - \theta_b)$$

where  $c_s$  is the baffle-to-shell clearance; see in Table.

$\theta_b$  can be obtained from Figure, for the appropriate baffle cut,

$B_c$

$$A_b = l_B (D_s - D_b)$$

## Effect of Fouling on Pressure Drop

**Table** Ratio of Fouled to Clean Pressure Drop

Fouling Coefficient (W/m <sup>2</sup> °C)	Shell Diameter/Baffle Spacing		
	1.0	2.0	5.0
<i>Laminar flow</i>			
6000	1.06	1.20	1.28
2000	1.19	1.44	1.55
<1000	1.32	1.99	2.38
<i>Turbulent flow</i>			
6000	1.12	1.38	1.55
2000	1.37	2.31	2.96
<1000	1.64	3.44	4.77

## Shell and Tube Exchangers: Worked as Condensers

The construction of a condenser will be similar to other shell and tube exchangers, except a **wider baffle spacing**,

typically  $l_B = D_s$ .

**Four condenser configurations** are possible:

1. **Horizontal**, with **condensation in the shell**, and the cooling medium in the tubes;
  2. **Horizontal**, with **condensation in the tubes**;
  3. **Vertical**, with **condensation in the shell**;
  4. **Vertical**, with **condensation in the tubes**;
- Horizontal shell-side and **vertical tube-side** are the **most commonly used** types of condensers.
  - A horizontal exchanger with condensation in the tubes is rarely used as a process condenser, but is the usual arrangement for heaters and vaporizers using condensing steam as the heating medium.

## Heat Transfer Fundamentals

The normal mechanism for heat transfer in commercial condensers is filmwise condensation.

Dropwise condensation will give higher heat transfer coefficients but is unpredictable; therefore, not considered for practical design of condensers for general purposes.

## Physical Properties

The physical properties are evaluated at the average condensate film temperature:

the mean of the condensing temperature and the tube-wall temperature.



## Condensation Outside Horizontal Tubes

$$(h_c)_1 = 0.95 k_L \left[ \frac{\rho_L (\rho_L - \rho_v) g}{\mu_L \Gamma} \right]^{1/3}$$

where

$(h_c)_1$  = mean condensation film coefficient, for a single tube,  $W/m^2 \text{ } ^\circ C$ ;

$k_L$  = condensate thermal conductivity,  $W/m^\circ C$ ;

$\rho_L$  = condensate density,  $kg/m^3$ ;

$\rho_v$  = vapor density,  $kg/m^3$ ;

$\mu_L$  = condensate viscosity,  $Ns/m^2$ ;

$g$  = gravitational acceleration,  $9.81 \text{ m/s}^2$ ;

$\Gamma$  = the tube loading, the condensate flow per unit length of tube,  $kg/m \text{ s}$ .

In a bank of tubes, the condensate from the upper rows of tubes will add to that condensing on the lower tubes.

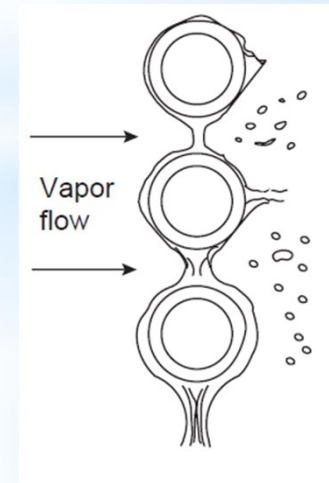
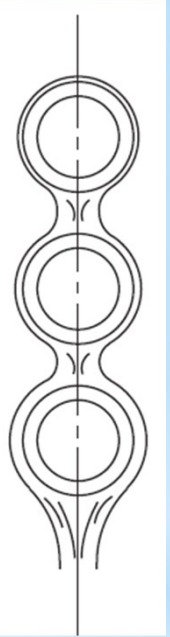
If there are  $N_r$  tubes in a vertical row and the condensate is assumed to flow smoothly from row to row, and if the flow remains laminar, the mean coefficient predicted by the Nusselt model is related to that for the top tube by

$$(h_c)_{N_r} = (h_c)_1 N_r^{-1/4}$$

Using Kern's method, the mean coefficient for a tube bundle is given by

$$(h_c)_b = 0.95 k_L \left[ \frac{\rho_L (\rho_L - \rho_v) g}{\mu_L \Gamma_b} \right]^{1/3} N_r^{-1/6}$$

Where  $\Gamma_b = \frac{W_c}{LN_t}$



$L$  = tube length,

$W_c$  = total condensate flow,

$N_t$  = total number of tubes in the bundle,

$N_r$  = average number of tubes in a vertical tube row.

$N_r$  can be taken as two-thirds of the number in the central tube row.

### Condensation Inside and Outside Vertical Tubes

For condensation inside and outside vertical tubes, the Nusselt model gives

$$(h_c)_v = 0.926k_L \left[ \frac{\rho_L(\rho_L - \rho_v)g}{\mu_L \Gamma_v} \right]^{1/3} \quad \text{Re} = 30$$

where

$(h_c)_v$  = mean condensation coefficient, W/m<sup>2</sup> °C;

$\Gamma_v$  = vertical tube loading, condensate rate per unit tube perimeter, kg/m s for a tube bundle:

$$\Gamma_v = \frac{W_c}{N_t \pi d_o} \text{ or } \frac{W_c}{N_t \pi d_i}$$

$Re > 30$ ; waves on the condensate film become important.  
The Reynolds number for the condensate film is given by

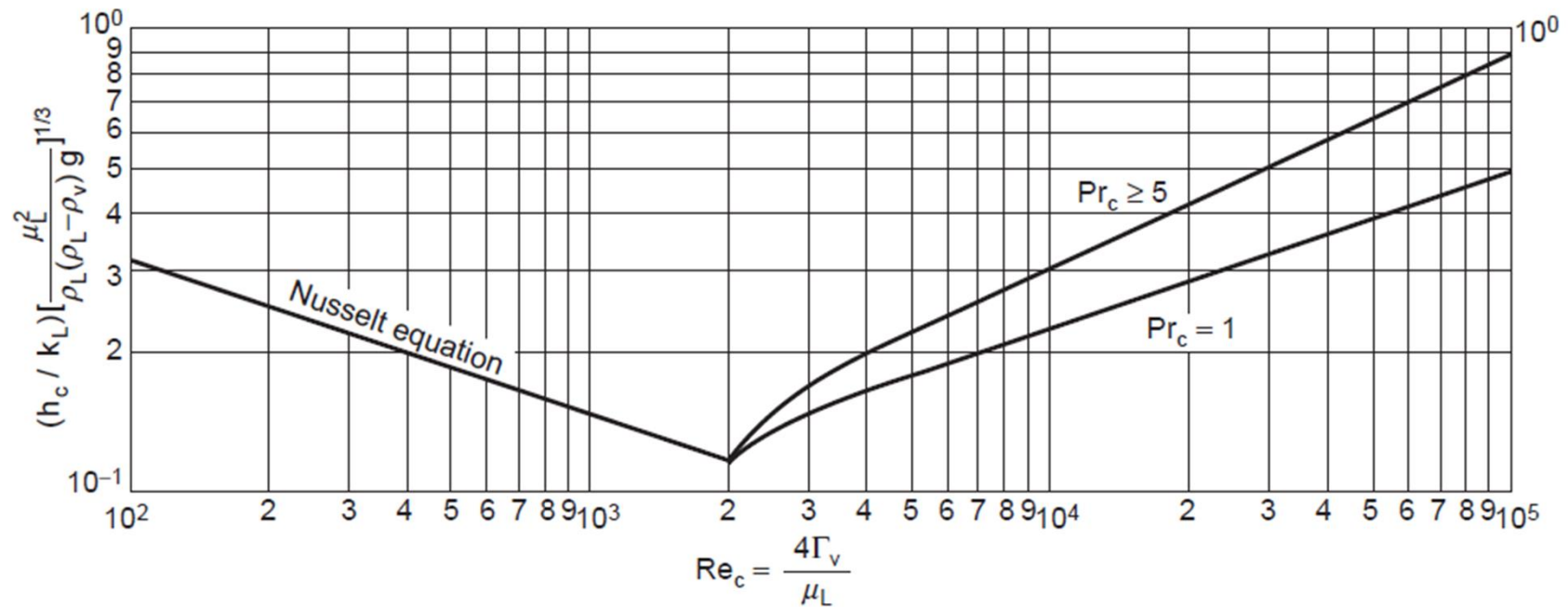
$$Re_c = \frac{4\Gamma_v}{\mu_L}$$

The presence of waves will increase the heat transfer coefficient, so the use of above equation will give conservative (safe) estimates for  $Re = 30$ .

Above a Reynolds number of around 2000, the condensate film becomes turbulent. The effect of turbulence in the condensate film was investigated by Colburn (1934), as shown in Figure.

The Prandtl number for the condensate film is given by

$$Pr_c = \frac{C_p \mu_L}{k_L}$$



Boyko and Kruzhilin (1967) developed a **correlation for shear-controlled condensation** in tubes that is simple to use.

Correlation gives the **mean coefficient between two points at which the vapor quality is known.**

The vapor quality  $x$  is the mass fraction of the vapor present. It is convenient to represent the Boyko-Kruzhilin correlation as

$$(h_c)_{BK} = h'_t \left[ \frac{J_1^{1/2} + J_2^{1/2}}{2} \right]$$

$$\text{where } J = 1 + \left[ \frac{\rho_L - \rho_v}{\rho_v} \right] x$$

suffixes 1 and 2 refer to the inlet and outlet conditions respectively.

$h'_t$  is the tube-side coefficient evaluated for single-phase flow of the total condensate

$$h'_t = 0.021 \left( \frac{k_L}{d_i} \right) Re^{0.8} Pr^{0.43}$$

In a condenser, where inlet stream fully saturated vapor and the vapor will be totally condensed.

For these conditions above equation becomes

$$(h_c)_{BK} = h'_t \left[ \frac{1 + \sqrt{\rho_L / \rho_v}}{2} \right]$$



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