2.6. Delaware Method for Shell-Side Rating of Shell and Tube Heat Exchangers

2.6.1. Introduction

The Delaware method for calculating the heat transfer coefficient and the pressure drop for a single-phase fluid flowing on the shell side of a shell and tube heat exchanger is based on the extensive experimental and analytical research program carried on in the Department of Chemical Engineering at the University of Delaware from 1946 to 1963. The procedure given here is modified from that originally presented in Ref. (5) and more recently in Ref. (6). As discussed previously, the method has been further modified to apply to Trufin tubes by following Ref. (4).

The Delaware method assumes that the flow rate and the inlet and outlet temperatures (also pressures for a gas or vapor) of the shell-side fluid are specified and that the density, viscosity, thermal conductivity, and specific heat of the shell-side fluid are known or can be reasonably estimated as a function of temperature. The method also assumes that the following minimum set of shell-side geometry data are known or specified:

Tube root and outside diameters, d_r and d_o
Fin spacing and thickness, s and Y
Tube geometrical arrangement (unit cell)
Shell inside diameter, D_i
Shell outer tube limit (diameter), $D_{ot\ell}$
Effective tube length (between tube sheets), L
Baffle cut (distance from baffle tip to shell inside surface) ℓ_c
Baffle spacing (face-to-face). ℓ_s
Number of sealing strips/side, N_{ss}

From this geometrical information all remaining geometrical parameters needed in the shell-side calculations can be calculated or estimated by methods given here. However, if additional specific information is available (e.g., tube-baffle clearance), the exact values of certain parameters may be used in the calculations probably with some improvement in accuracy.

In order to complete the rating of a shell and tube exchanger, it is necessary to calculate the tube-side heat transfer and pressure drop characteristics from the methods given previously.

Not all of the fluid flow rates and temperatures can be independently specified, but are connected through the heat balance on the exchanger. Similarly, the overall rate equation, Q = UA (MTD) must be satisfied, and it may well be that U calculated by this design method does not equal that required by the heat balance and rate equation. If this happens when one is designing an exchanger to perform a given service, it is necessary to change one or more of the geometrical parameters (the tube length is a particularly popular choice because changing it does not require complete recalculation of the coefficient) until the calculated and required performances are in substantial agreement. If an existing exchanger is being rated, disagreement between calculated and required performance can only be resolved by changing flow rates and/or terminal specifications until agreement exists.

Finally, it should be remembered that this method, though apparently generally the best in the open literature, is not extremely accurate, even for plain tubes. An exhaustive study by HTRI (Ref. 9) testing the various methods against 972 heat transfer data points and 1332 pressure drop data points covering a very wide range of fluids and geometrical parameters showed that this method predicted shell-side coefficients from about 50 per cent low to 100 percent high, while the pressure drop range was from about 50 per cent low to 200 per cent high. The mean error for heat transfer was about 15 per cent low (conservative) for all Reynolds numbers, while the mean error for pressure drop was from about 5 per cent low (unsafe) at Reynolds numbers above 1000 to about 100 percent high at Reynolds numbers below 10.

It is necessary to be very careful about the units used in the following equations and graphs. The graphs and equations used for generating estimates of the geometrical parameters are in the units most commonly used in the U.S. for the quantity in question. But dimensional consistency is required in the equations used for calculating heat transfer coefficient and pressure drop. Therefore, it is advisable to check the units used in each equation before assigning units to the final answer.

2.6.2. Calculation of Shell-Side Geometrical Parameters

1. Total number of tubes in the exchanger, N_t: If not known by direct count, find in the tube count table, Table 2.6, as a function of the shell inside diameter D_i, the tube pitch, p, and the layout. The tube count is for a fixed tube-sheet, fully-tubed bundle. In order to estimate the tube counts for a different bundle geometry, the factor F_3 from Table 2.4 may be used as a *divider* for the numbers in Table 2.6.

Example: The tube count for 3/4 in. 0 D tubes on a 15/16 in. triangular pitch in a 39 in. ID shell, with a split backing ring floating head and 4 tube side passes can be estimated to be:

from table 2.6 tube count = 1338
from table 2.4 F_3 = 1.06

$$\frac{1338}{F_3} = \frac{1338}{(1.06)} = 1262 \text{ tubes}$$

2. Tube-pitch parallel to flow, p_p, and normal to flow p_n:

These quantities are needed only for the purpose of estimating other parameters. If a detailed drawing of the exchanger is available, or if the exchanger itself can be conveniently examined, it is better to obtain these other parameters by direct count or calculation. The pitches are defined in Fig. 2.27 and some values tabulated in Table 2.7.

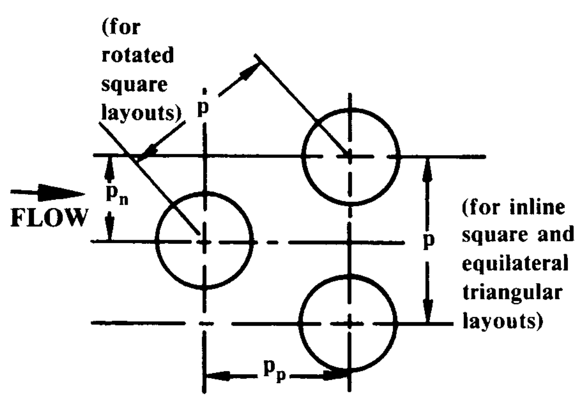

Figure 2.27 Definition of Various Tube Pitches

3. Number of tube rows crossed in one crossflow section, N_c: Count from exchanger drawing or estimate from

$$N_c = \frac{D_i\left[1-2\left(\frac{\ell_c}{D_i}\right)\right]}{p_p} \tag{2.38}$$

This is the number of rows of tubes between the tips of adjacent baffles, counting each row at the tip as one-half of a row.

4. Fraction of total tubes in crossflow, F_c. F_c can be calculated from

$$F_c = \frac{1}{\pi}\left\{\pi + 2\left(\frac{D_i - 2\ell_c}{D_{ot\ell}}\right)\sin\left[\cos^{-1}\left(\frac{D_i - 2\ell_c}{D_{ot\ell}}\right)\right] - 2\cos^{-1}\left(\frac{D_i - 2\ell c}{D_{ot\ell}}\right)\right\} \tag{2.39}$$

where all the angles are read in radians. For convenience, F_c has been plotted to an acceptable degree of precision in Fig. 2.28 as a function of percent baffle cut and shell diameter D_i. For pull-through floating head construction or other designs with large clearances between the shell inside diameter and the outer tube limit, $(D_i - D_{ot\ell})$, the value of F_c is a little higher than that shown.

5. Number of effective crossflow rows in each window, N_{cw}. Estimate from:

$$N_{cw} = \frac{0.8\ell_c}{p_p} \tag{2.40}$$

This assumes that the tube field covers about 80 percent of the distance from the baffle cut to the shell, and that the flow on the average penetrates about half-way through this part of the tube field before turning to flow parallel to the tubes through the baffle cut and then turning again to begin the next crossflow section.

6. Number of baffles, N_b. Calculate from

$$N_b = \frac{L}{\ell_s} - 1 \tag{2.41}$$

In design procedure, the length may not be (and need not be) specified precisely at this point. The heat transfer coefficient calculation does not (for most cases) require values of either L or N_b, so it may be convenient to calculate the shell-side and overall coefficients, followed by the required length to satisfy the thermal specification. Then this length (rationalized to an integral number of baffle spaces) can be used to calculate N_b and the pressure drops. Of course, it may then be necessary to choose a new shell diameter and start over again if the allowable pressure drops have been exceeded, or if they are so much over the calculated value as to suggest that a smaller shell diameter might suffice.

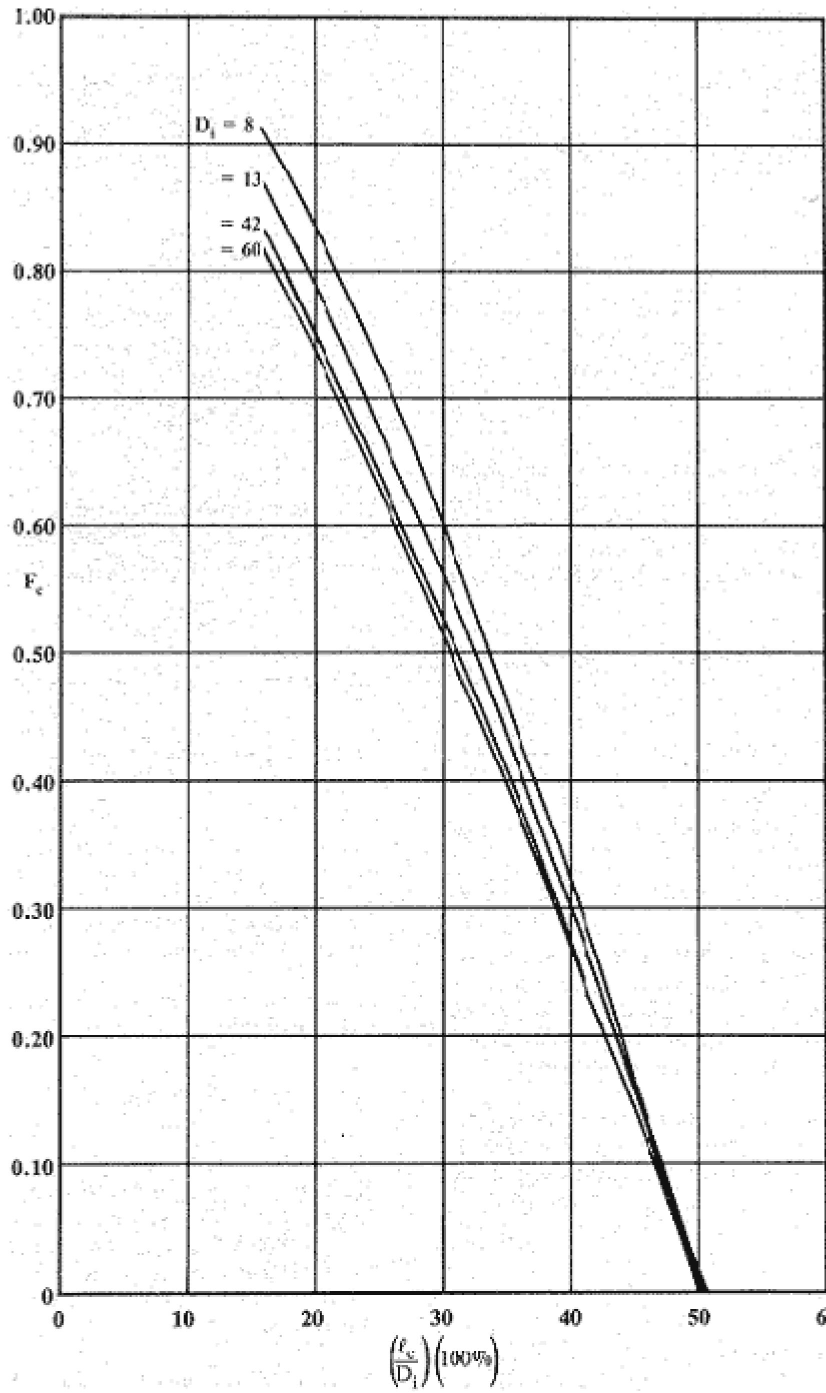

Fig. 2.28 Estimation of Fraction of Tubes in Cross-Flow.

7. Crossflow area at or near the centerline for one crossflow section, S_m. Estimate from eq. 2.42 or eq. 2.43.

$$S_m = \ell_s \left\{ D_i - D_{ot\ell} + \left(\frac{D_{ot\ell} - d_o}{p_n} \right) \left[(p - d_o) + 2H \left(\frac{s}{s+Y} \right) \right] \right\} \qquad (2.42)$$

for rotated and inline square layouts; or from

$$S_m = \ell_s \left\{ D_i - D_{ot\ell} + \left(\frac{D_{ot\ell} - d_o}{p} \right) \left[(p - d_o) + 2H \left(\frac{s}{s+Y} \right) \right] \right\} \qquad (2.43)$$

for triangular layouts.

8. Fraction of crossflow area available for bypass flow, F_{sbp}. Estimate from

$$F_{sbp} = \frac{(D_i - D_{ot\ell})\ell_s}{S_m} \qquad (2.44)$$

This is the area between the outermost tubes and the shell and can constitute a major path for flow to largely escape contact with heat transfer surface. In constructions with a large clearance between the shell and the outer tube limit, it is practically essential to block this flow with sealing devices.

9. Tube-to-baffle leakage area for one baffle, S_{tb}. Estimate from

$$d_o = \frac{5}{8} in.: S_{tb} = 0.0152 N_t (1 + F_c), in.^2 \qquad (2.45)$$

$$d_o = \frac{3}{4} in.: S_{tb} = 0.0184 N_t (1 + F_c) in.^2 \qquad (2.46)$$

$$d_o = 1 \ in.: S_{tb} = 0.0245 N_t (1 + F_c) in.^2 \qquad (2.47)$$

These values are based on TEMA Class R construction which specifies 1/32 in. diametrical clearance between tube and baffle as the usual standard (Ref. 10): Values should be modified if extra tight or loose construction is specified, or if clogging by dirt is anticipated.

10. Shell-to-baffle leakage area for one baffle, S_{sb}. If the diametrical clearance between the outside of the baffle and the inside of the shell, δ_{sb}, is known, S_{sb} can be calculated from

$$S_{sb} = \frac{D_i \delta_{sb}}{2} \left[\pi - \cos^{-1} \left(1 - \frac{2\ell_c}{D_i} \right) \right], \ in.^2 \qquad (2.48)$$

where the value of the term $\cos^{-1}(1-2\ell_c/D_i)$ is in radians and is between 0 and 2. This area has been calculated and plotted in Fig. 2.29 as a function of percent baffle cut and inside diameter. Fig. 2.29 is based on TEMA Class R Standards (Ref. 10):

D_i, in.	**Diametral shell-baffle clearance, in.**	
8 – 13	0.100	These values are for pipe shells; if rolled shells are used, add 0.125 in.
14 – 17	0.125	
18 – 23	0.150	
24 – 39	0.175	
40 – 54	0.225	
55 – 60	0.300	

Since pipe shells are generally limited to diameters below 24 in., the larger sizes are shown using the rolled shell specification. Again, allowance should be made for especially tight or loose construction. No specification is given in the TEMA Standards for shells above 60 in. in diameter.

11. Area for parallel flow through window, S_w. This area is obtained as the difference between the gross window area, S_{wg}. and the window area occupied by tubes, S_{wt} :

$$S_w = S_{wg} - S_{wt} \tag{2.49}$$

The value of S_{wg} can be calculated from

$$S_{wg} = \frac{D_i^2}{4}\left[\cos^{-1}\left[1-2(\frac{\ell_c}{D_i})\right]-\left[1-2(\frac{\ell_c}{D_i})\right]\sin\left\{\cos^{-1}\left[1-2(\frac{\ell_c}{D_i})\right]\right\}\right],\ in.^2 \tag{2.50}$$

For convenience, however, the values of S_{wg} are plotted in Fig. 2.30 as a function of (ℓ_c/D_i) and D_i.

The window area occupied by the tubes, S_{wt} , can be calculated from

$$S_{wt} = \frac{N_t}{8}(1-F_c)\pi\, d_o^{\,2}, in.^2 \tag{2.51}$$

For convenience, the value of S_{wt} can be found from Fig. 2.31. To use this figure, enter the lower abscissa at the appropriate value of N_t and proceed vertically to the solid line corresponding to d_o, Then proceed horizontally to the dashed line corresponding to the estimated value of F_c and thence vertically to read the value of S_{wt} on the upper abscissa.

12. Equivalent diameter of window, D_w. (Required only if laminar flow, defined as $\mathrm{Re}_s \le 100$, exists.)

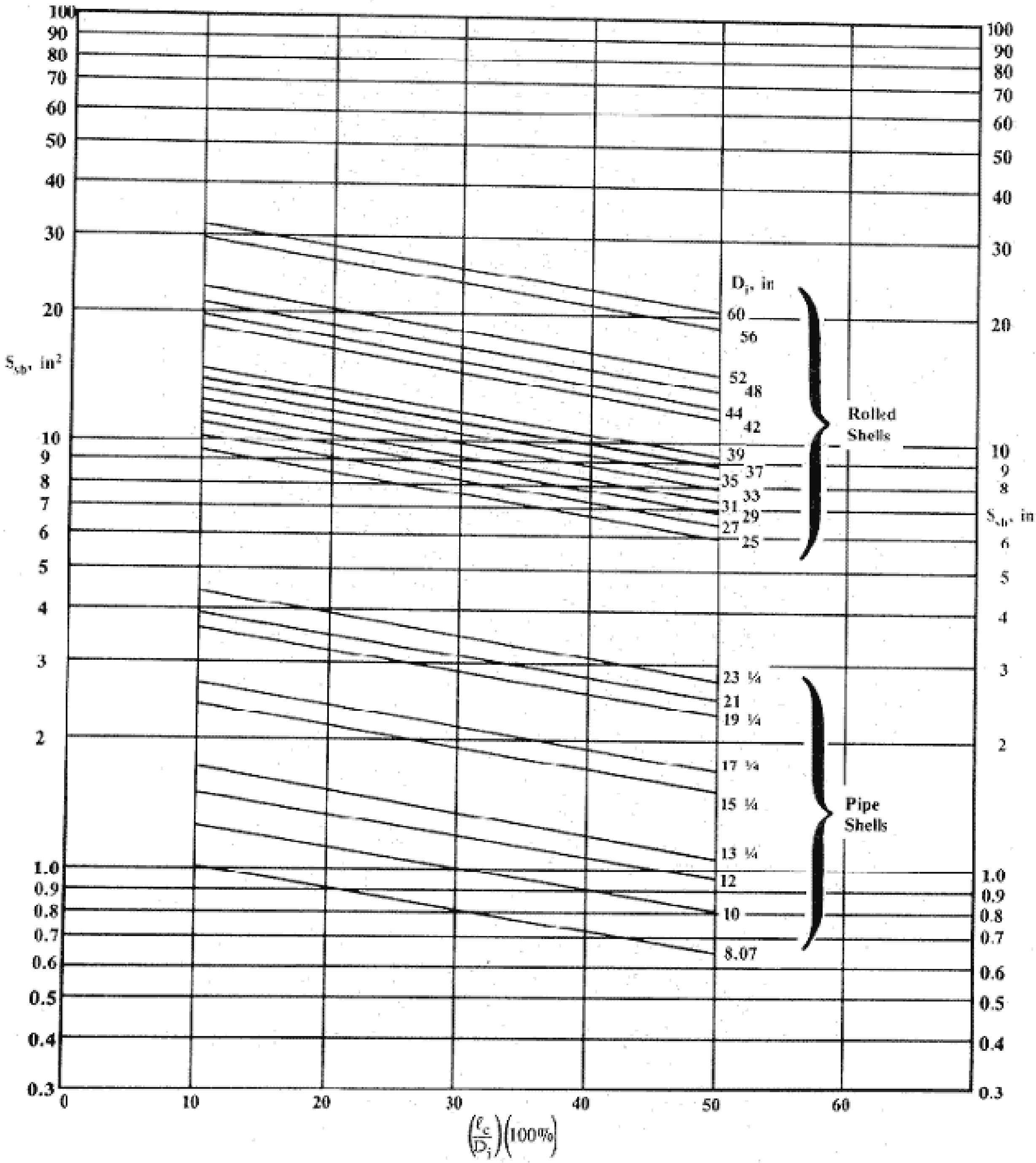

Fig 2.29 Estimation of Shell-to-Baffle Leakage Area.

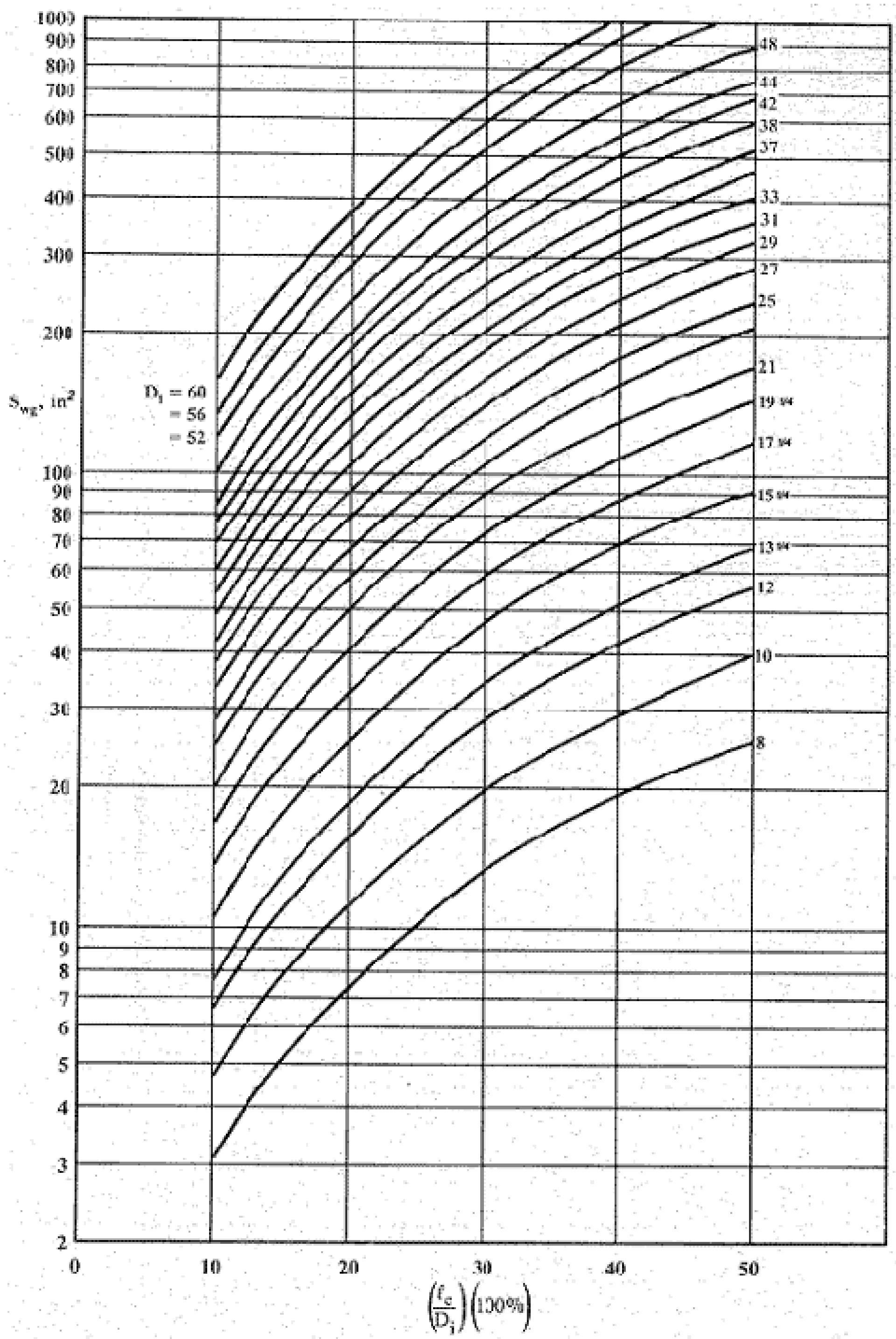

Fig. 2.30 Estimation of Window Gross Flow Area

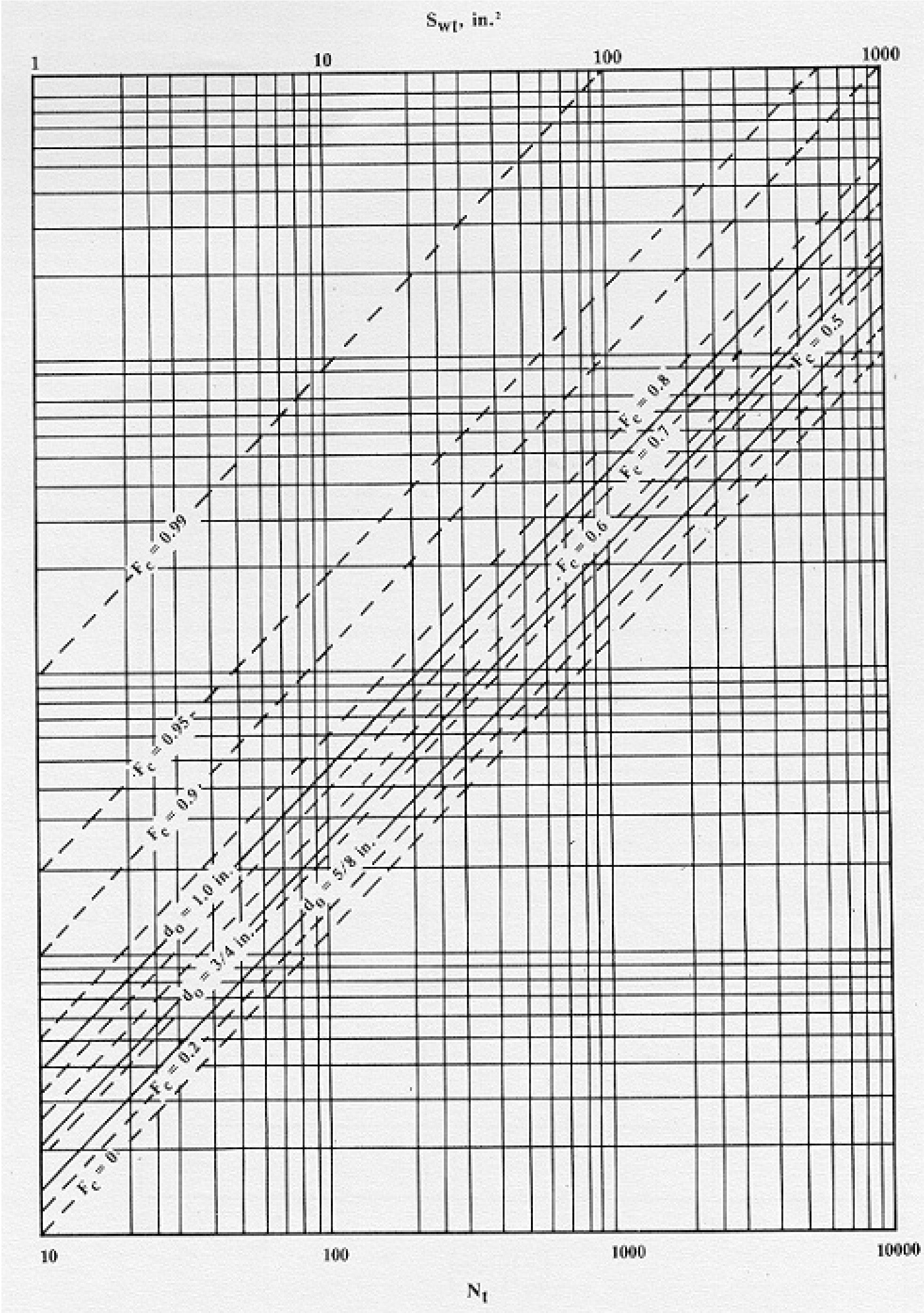

Fig. 2.31 Estimation of Cross-Sectional Area of Tubes in Window.

Calculate from

$$D_w = \frac{4S_w}{\frac{\pi}{2} N_t (1 - F_c) d_o + D_i \theta}, in. \tag{2.52}$$

where θ is the baffle cut angle and is given by

$$\theta = 2\cos^{-1}\left(1 - \frac{2\ell_c}{D_i}\right) \tag{2.53}$$

For convenience, θ is plotted in Fig. 2.32 as a function of $\left(\frac{\ell_c}{D_i}\right)(100\%)$.

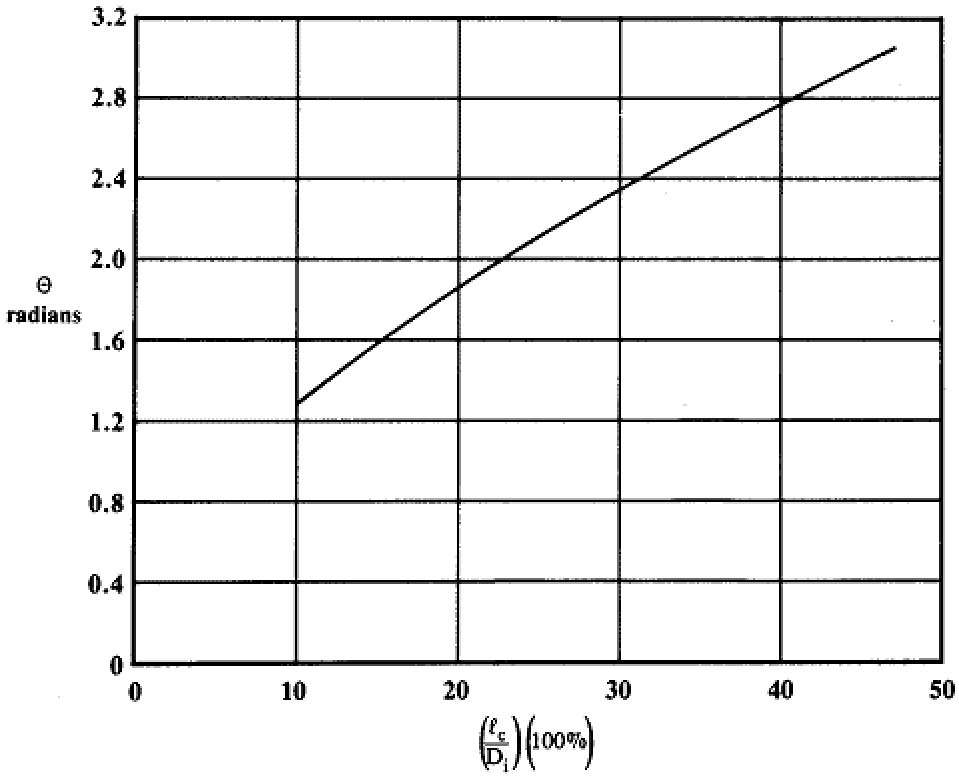

Fig. 2.32 Baffle Cut Angle.

2.6.3. Shell-Side Heat Transfer Coefficient Calculation

1. The Delaware method first calculates the heat transfer coefficient for crossflow of the fluid (at the design flow rate) in an ideal tube bank bounded by the shell on the sides and two adjacent baffles on the ends. This is done using the ideal tube bank correlation previously presented in this Chapter.

   This coefficient must then be corrected for the effects of the baffle geometry, tube-to-baffle and shell-to-baffle leakage, bundle by-pass effects (including sealing strip effects) and, at low Reynolds numbers, the build-up of an adverse temperature gradient. The resulting coefficient is then the effective shell-side heat transfer coefficient and is used with the other heat transfer terms to calculate the overall heat transfer coefficient. The overall coefficient is then used with the heat duty and the Mean Temperature Difference to calculate the area required. If the effective tube length has not been previously specified, the required length may now be calculated to determine the feasibility of the design. If the length has been given (as in the case of rating an existing exchanger), the required area may be compared to the available area to determine the suitability of the given exchanger.

2. Calculate shell-side Reynolds number, Re_s. The shell-side Reynolds number is defined as

$$\mathrm{Re}_s = \frac{d_r W_s}{\mu_s S_m} \tag{2.54}$$

   where

   W_s = weight flow rate of shell-side fluid, Ib$_m$/hr
   μ_s = bulk viscosity of shell-side fluid, Ib$_m$/ft hr
   d_r = root dia. of Trufin tube, ft.

   It is important to verify that the quantities used in Eq. (2.54) are in such units that the resulting value of Re_s is dimensionless. It is usually adequate to use the arithmetic mean bulk shell-side fluid temperature (i.e., halfway between the inlet and exit temperatures) to evaluate all bulk properties of the shell-side fluid. In the case of long temperature ranges or for a fluid whose viscosity is very sensitive to temperature change, special care must be taken (such as breaking the calculation into segments, each covering a more limited temperature range). Even then, the accuracy of the procedure is less than for more conventional cases.

3. Find j_s from the ideal tube bank curve for a given tube layout at the calculated value of Re_s, using Fig. 2.15.

4. Calculate the shell-side heat transfer coefficient for an ideal tube bank, $h_{o,i}$

$$h_{o,i} = j_s C_{p,s}\left(\frac{W_s}{S_m}\right)\left(\frac{k}{C_p \mu}\right)_s^{2/3}\left(\frac{\mu}{\mu_w}\right)_s^{0.14}\left(\frac{Btu}{hr\ ft^{2}\,{}^{\circ}F}\right) \tag{2.55}$$

5. Find the correction factor for baffle configuration effects, J_c. J_c is read from Fig. 2.33,as a function of F_c.

6. Find the correction factor for baffle leakage effects, J_ℓ. J_ℓ is found from Fig. 2.34 as a function of the ratio of the total baffle leakage area, $(S_{sb} + S_{tb})$, to the crossflow area, S_m, and of the ratio of the shell-to-baffle leakage area S_{sb} to the tube-to-baffle leakage area, S_{tb}.

7. Find the correction factor for bundle bypassing effects, J_b. J_b is found from Fig. 2.35 as a function of F_{sbp} and of N_{ss}/N_c (the ratio of the number of sealing strips per side to the number of rows crossed in one baffle crossflow section). The solid lines on Fig. 2.35 are for $\mathrm{Re}_s \geq 100$; the dashed lines for $\mathrm{Re}_s < 100$. Sealing strips are always mounted in pairs, arranged symmetrically with respect to both the bundle diameter parallel to the flow direction and the bundle diameter perpendicular to the flow direction. The best compromise between the improved heat transfer coefficient and the greater complexity of construction seems to be achieved when one pair of sealing strips is used for approximately every six tube rows.

8. Find the correction factor for adverse temperature gradient buildup at low Reynolds numbers, J_r. This factor is equal to 1.00 if Re_s is equal to or greater than 100. For Re_s equal to or less than 20, the correction factor is fully effective and a function only of the total number of tube rows crossed. For Re_s between 20 and 100, a linear proportion rule is used.

   Therefore:

   a. If $\mathrm{Re}_s < 100$, find J^*_r from Fig. 2.36, knowing N_b and (N_c+N_{cw}).
   b. If $\mathrm{Re}_s \leq 20$, $J_r = J^*_r$.
   c. If $20 < \mathrm{Re}_s < 100$, find J_r from Fig. 2.37, knowing J^*_r and Re_s.

9. Calculate the shell-side heat transfer coefficient for the exchanger, h_o Btu/hr ft^2°F, from the equation:

$$h_o = h_{o,i} J_c J_\ell J_b J_r \tag{2.56}$$

2.6.4. Shell-Side Pressure Drop Calculation

1. From the Delaware experimental work, we know the correlations for calculating the pressure drop for a single ideal cross-flow section as well as for a single ideal window section. Further studies there indicated: a) that the pressure drop across the inlet and exit sections was affected (reduced) by bundle bypass but not by baffle leakage, b) that the pressure drop across internal crossflow sections was affected by both bundle bypass and baffle leakage, and c) that the pressure drop through a window was affected by baffle leakage but not by bypass.

   The calculational structure is then to calculate the ideal crossflow and window pressure drops, correct each of those terms by the effective correction factors, then multiply the effective pressure drops by the number of sections of that kind found in the exchanger, and finally to sum the effects to give the total shell side pressure drop (exclusive of nozzles.)

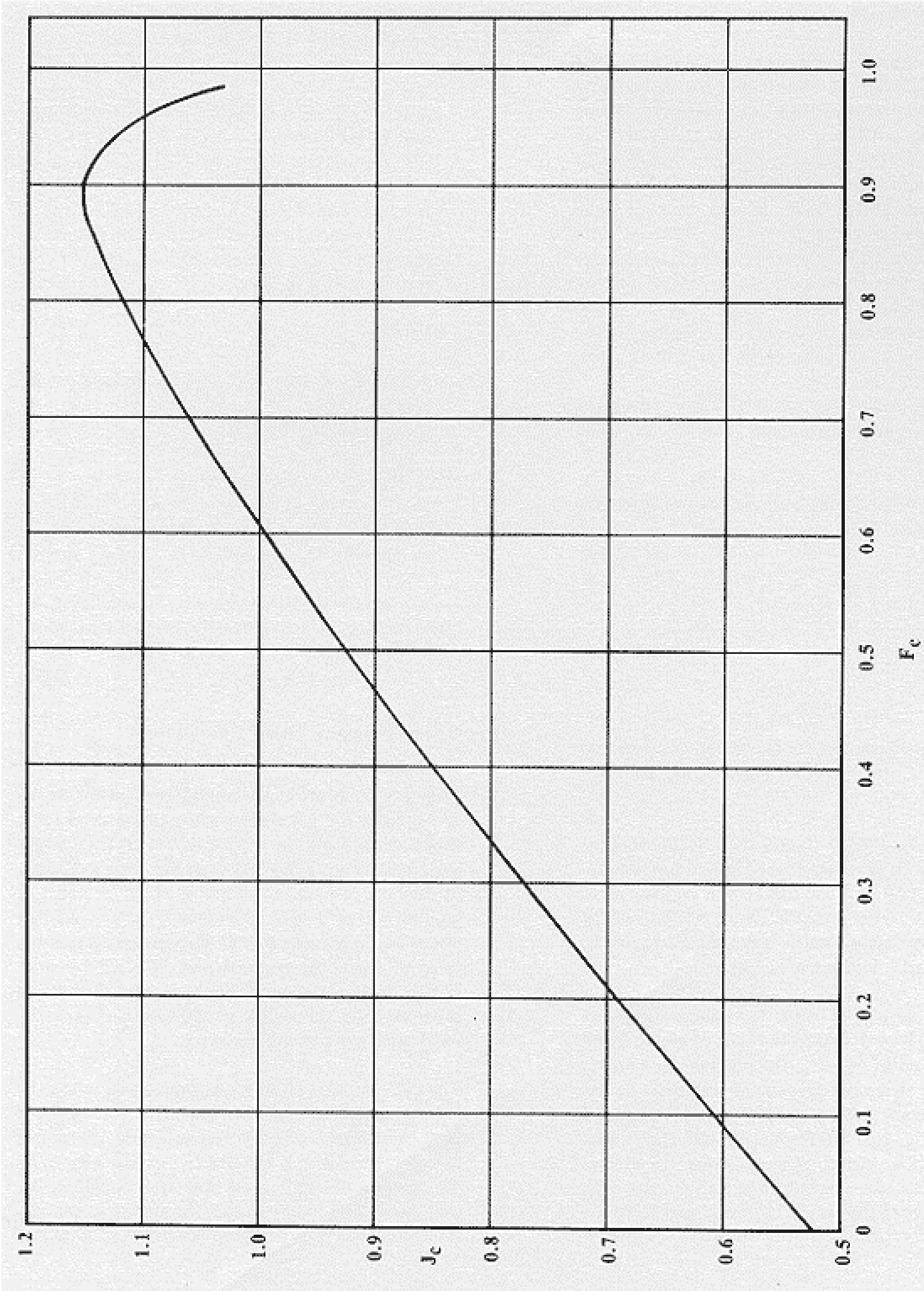

Fig. 2.33 Correction Factor for Baffle Configuration Effects.

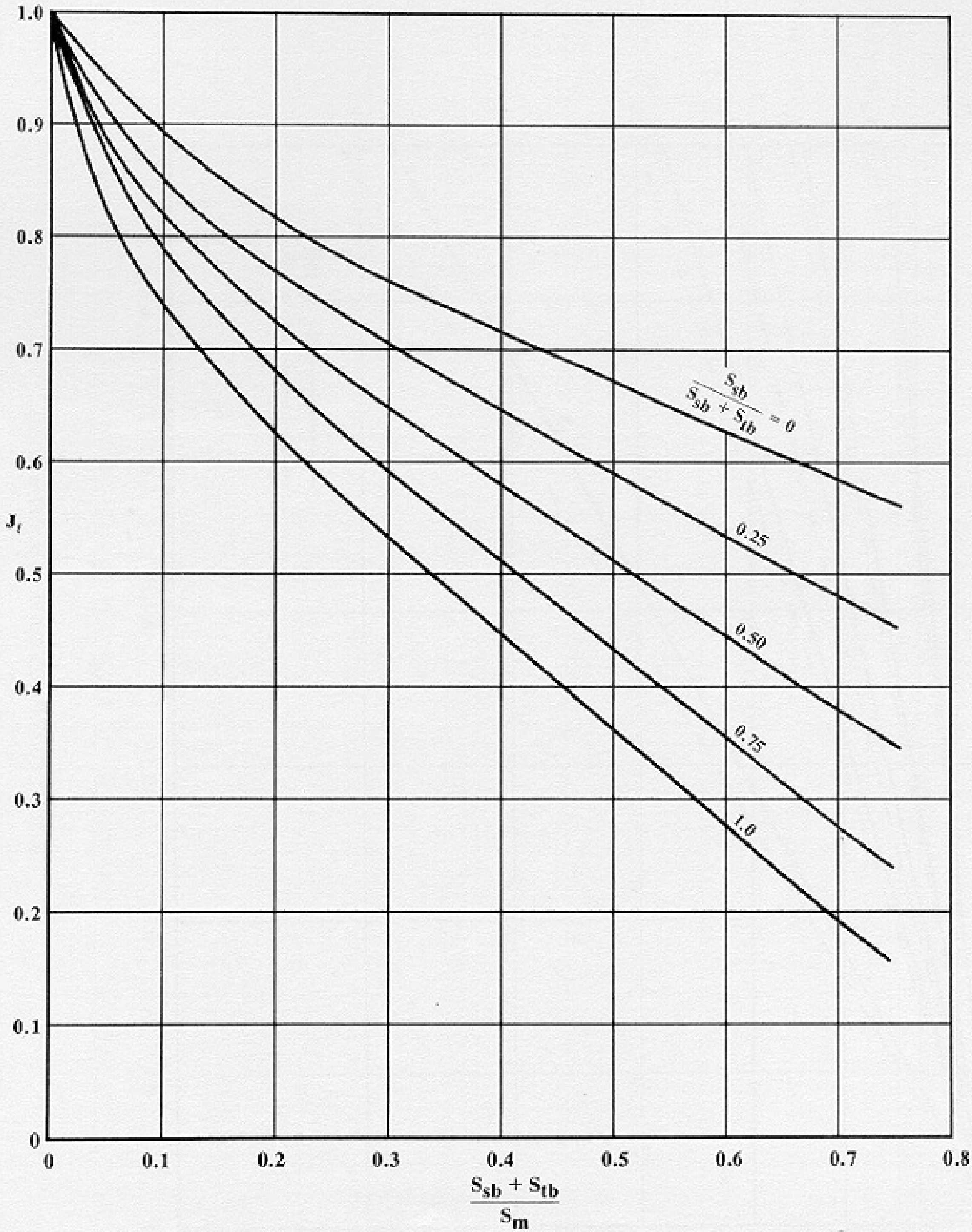

Fig. 2.34 Correction Factor for Baffle Leakage Effects.

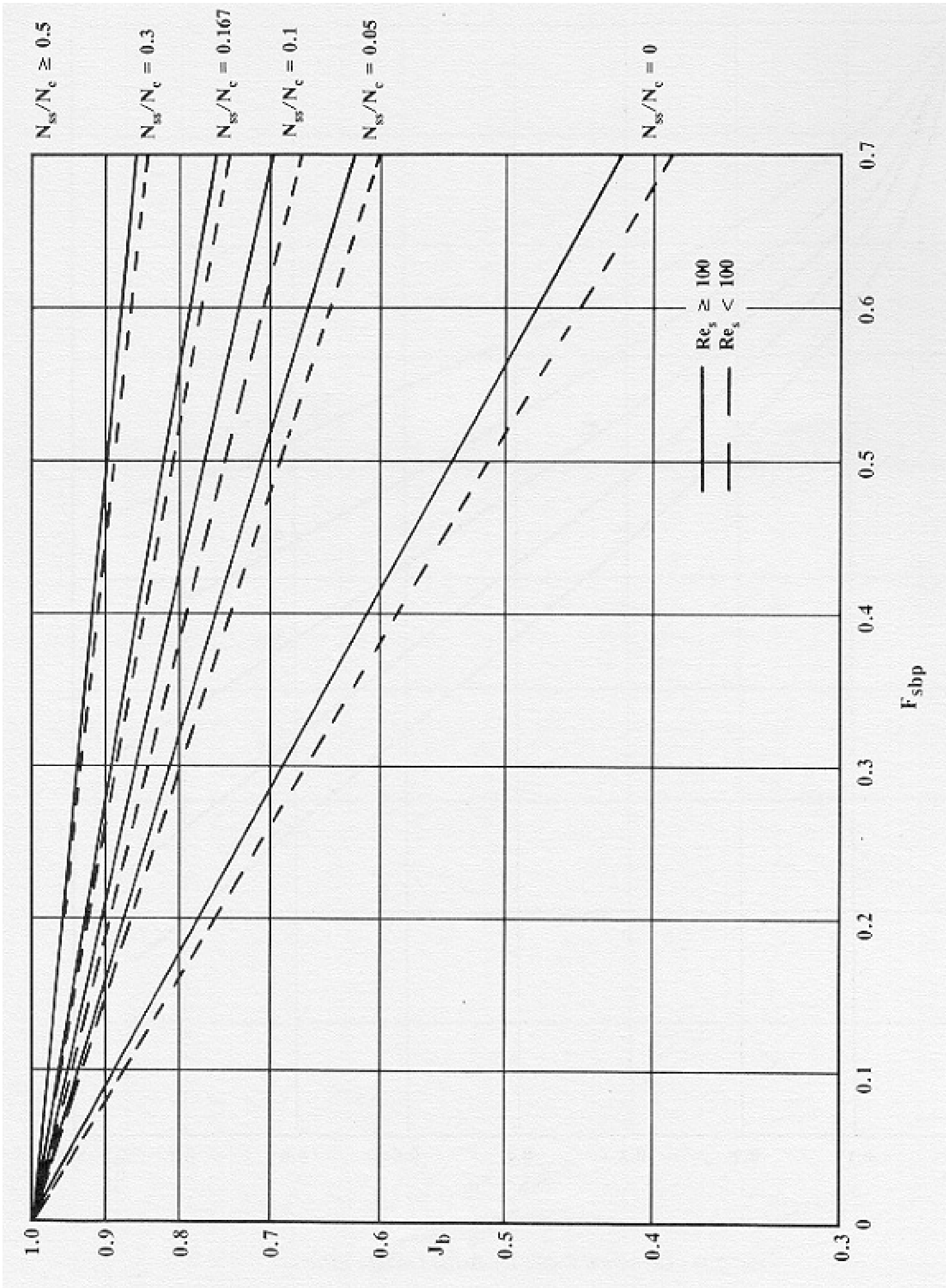

Fig. 2.35 Correction Factor for Bypass Flow Effects.

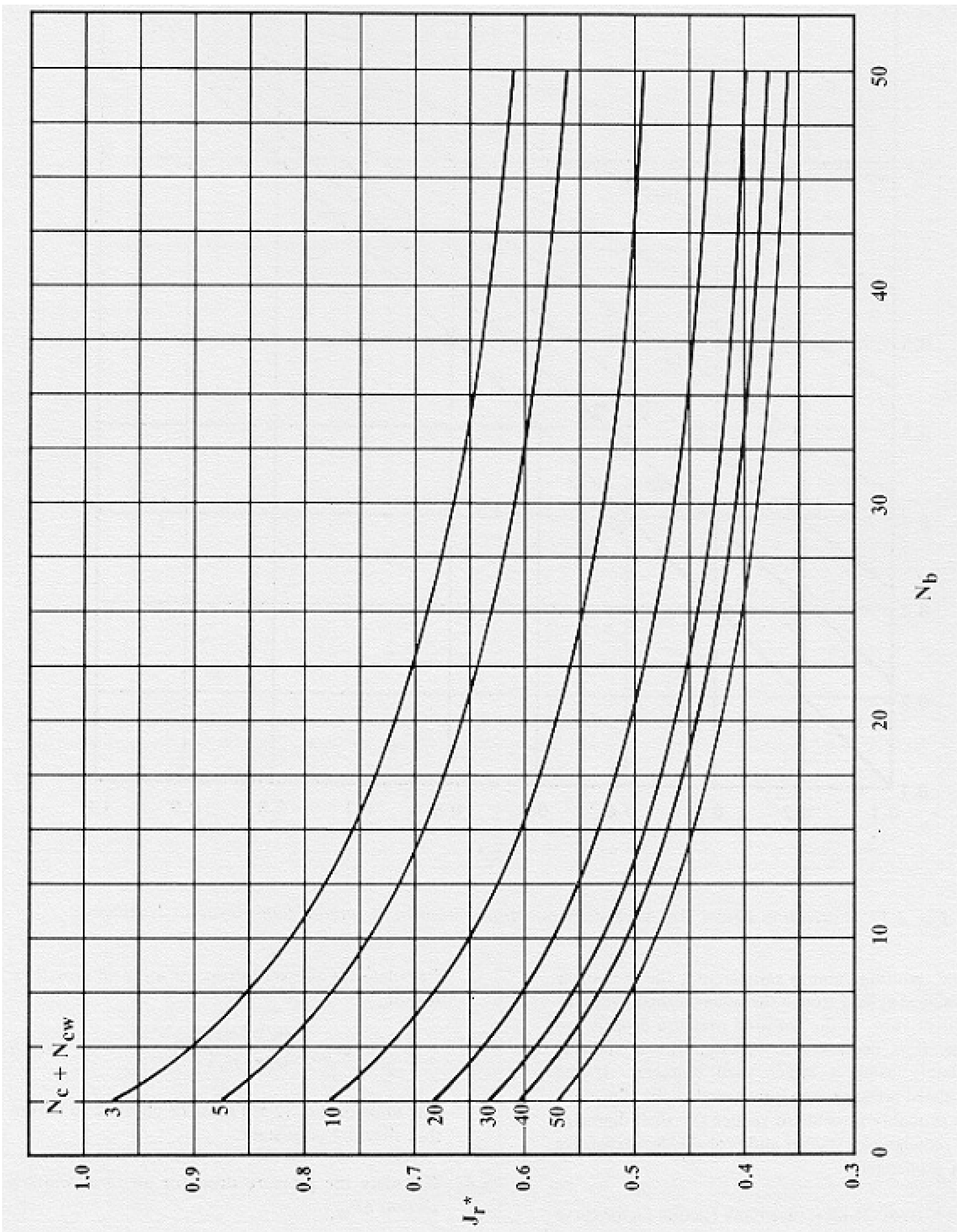

Fig. 2.36 Basic Correction Factor for Adverse Temperature Gradient at Low Reynolds Numbers

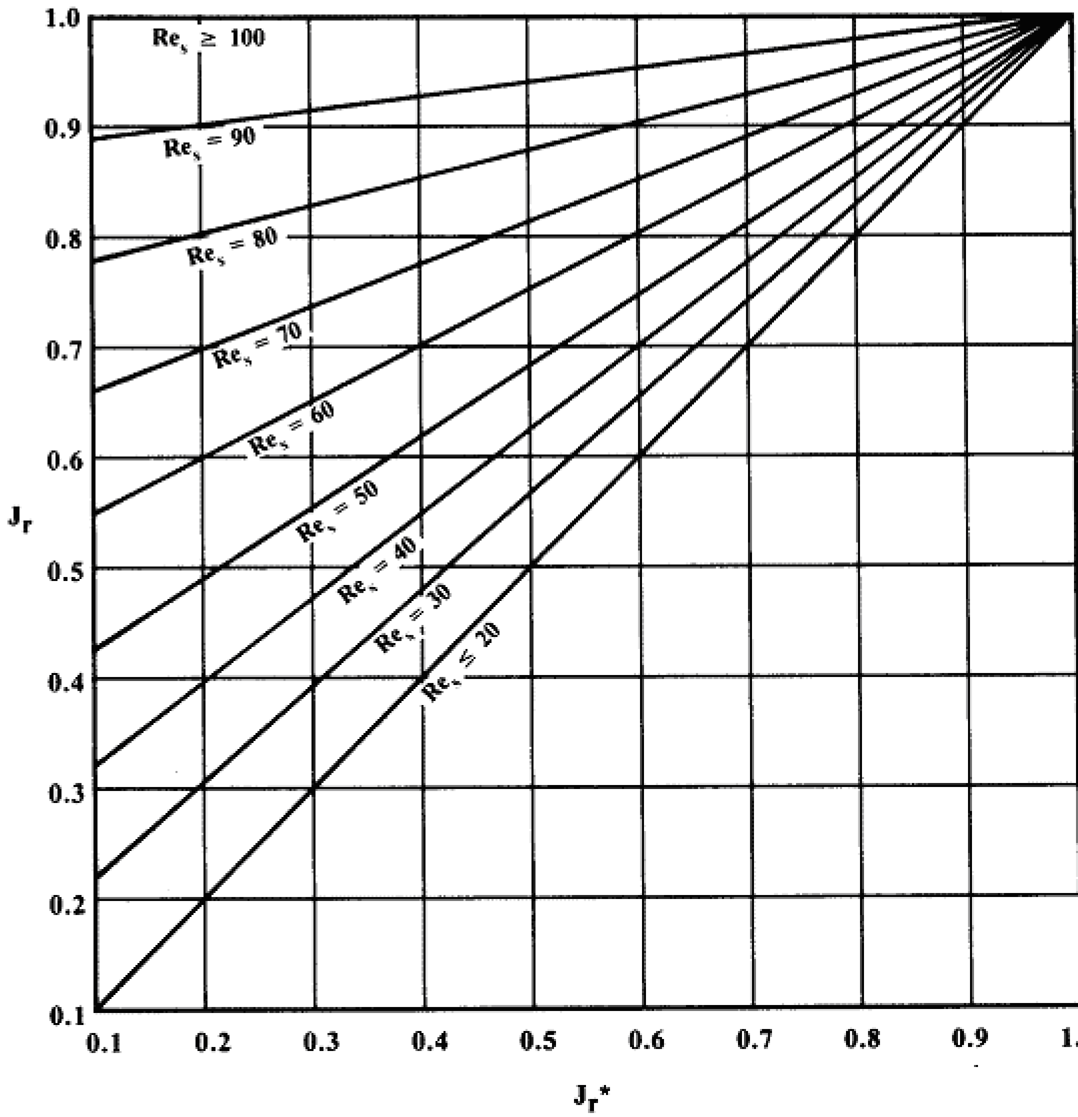

Fig. 2.37 Correction Factor for Adverse Temperature Gradient at Intermediate Reynolds Numbers.

If the resulting value is satisfactory, the exchanger is designed at least from a shell-side thermal-hydraulic point of view. If the required pressure drop is too large, it is necessary to redesign the exchanger, probably using a larger shell diameter. If the calculated pressure drop is much below the allowable, it is probably possible to reduce the shell diameter and redesign to a smaller and probably less expensive heat exchanger.

2. Find f_s from the ideal tube bank friction factor curve for the given tube layout at the calculated value of Re_s, using Fig. 2.17.

3. Calculate the pressure drop for an ideal crossflow section,

$$\Delta P_{b,i} = \frac{4 f_s W_s{}^2 N_c}{2\rho_s g_c S_m{}^2}\left(\frac{\mu_w}{\mu}\right)_s^{0.14} \tag{2.57}$$

The units on Eq. (2.57) must be checked to ensure that they are consistent.

4. Calculate the pressure drop for an ideal window section $\Delta P_{w,i}$.

   a. If $Re_s \geq 100$:

$$\Delta P_{w,i} = \frac{W_s{}^2(2+0.6N_{cw})}{2g_c S_m S_w \rho_s} \tag{2.58}$$

   b. If $Re_s < 100$:

$$\Delta P_{w,i} = 26\frac{\mu_s w_s}{g_c\sqrt{S_m S_w}\rho_s}\left[\frac{N_{cw}}{p-d_o}+\frac{\ell_s}{D_w{}^2}\right]+\frac{w_s{}^2}{g_c S_m S_w \rho_s} \tag{2.59}$$

5. Calculate correction factor for effect of baffle leakage on pressure drop, R_ℓ. Read from Fig. 2.38 as a function of $(S_{sb} + S_{tb})/S_m$ with parameter of $S_{sb}/(S_{sb} + S_{tb})$. The curves are not to be extra polated beyond the points shown.

6. Find the correction factor bundle bypass, R_b. Read from Fig. 2.39 as a function of F_{sbp} and N_{ss}/N_c. The solid lines are for $Re_s \geq 100$; the dashed lines are for $Re_s < 100$.

7. Calculate the pressure drop across the shell-side (excluding nozzles), ΔP_s from:

$$\Delta P_s = \left[(N_b-1)(\Delta P_{b,i})R_b + N_b \Delta P_{w,i}\right]R_\ell + 2\Delta P_{b,i}R_b(1+\frac{N_{cw}}{N_c}) \tag{2.60}$$

The application of the Delaware method to heat exchanger design is illustrated by the following example.

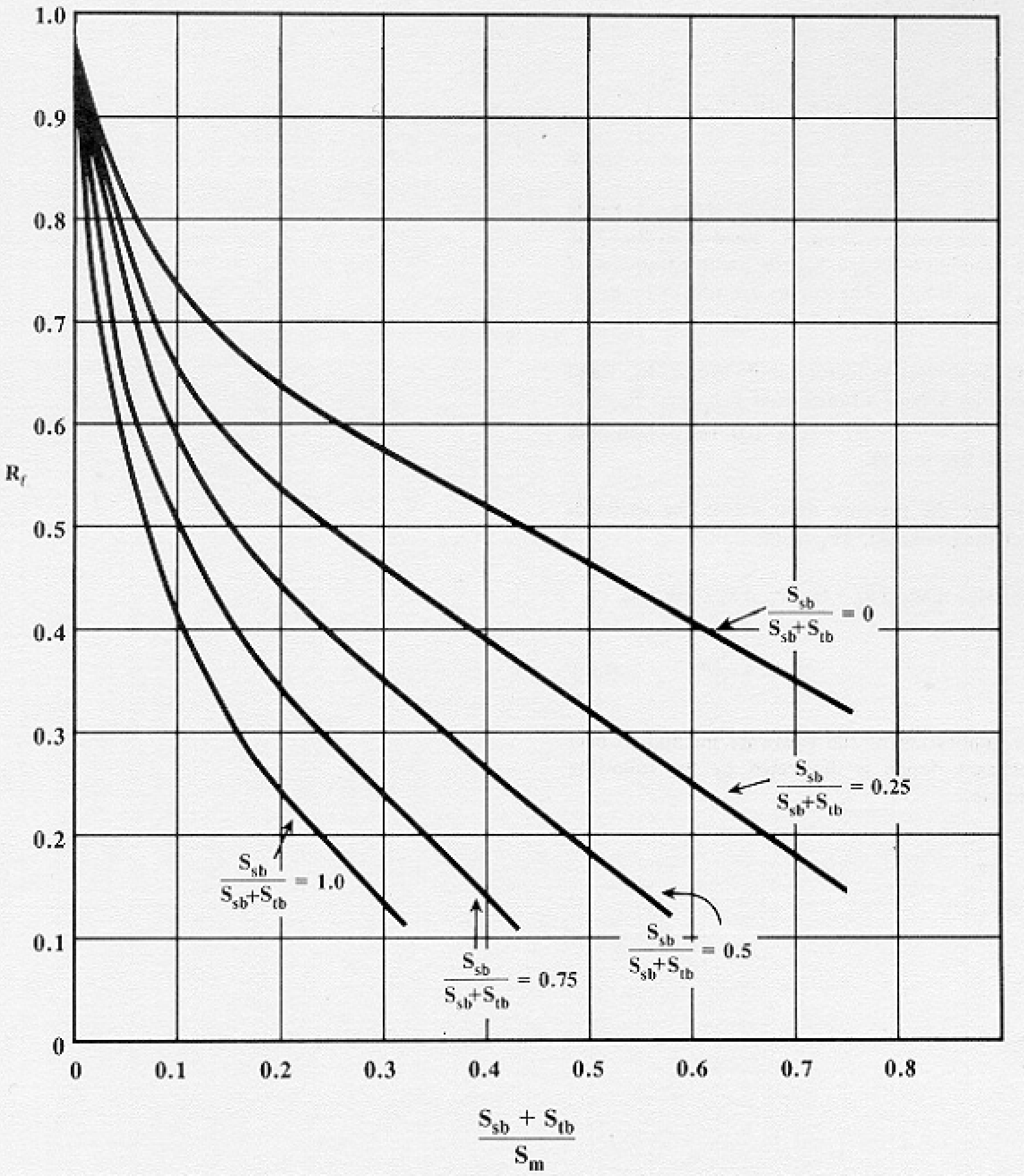

Fig. 2.38 Correction Factor for Baffle Leakage Effect on Pressure Drop

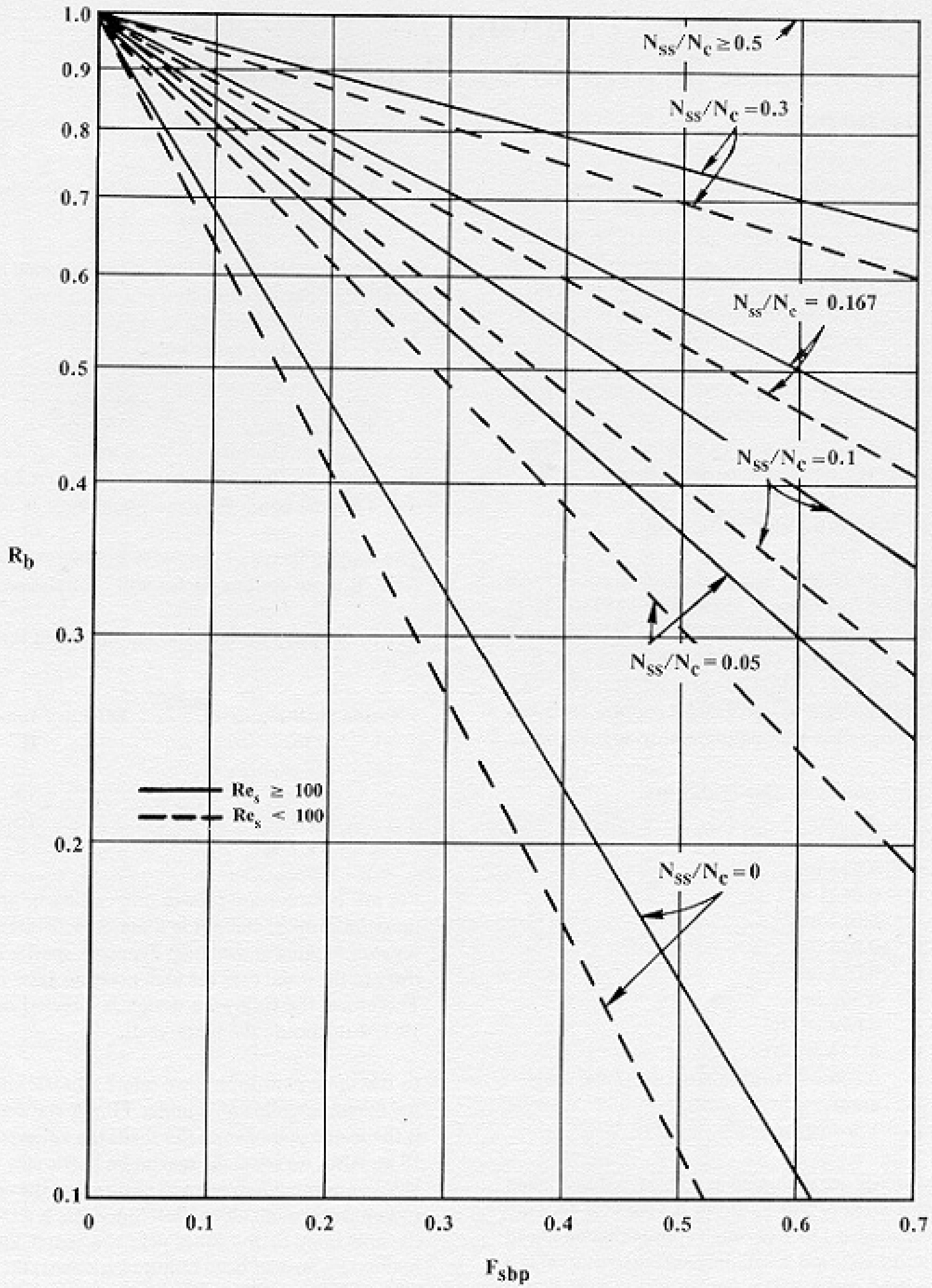

Fig. 2.39 Correction Factor on Pressure Drop for Bypass Flow