More accurate exchanger shell-side pressure drop calculations

Use this method to reduce equipment size and pumping power cost

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The growing importance of energy conservation requires accurate pressure drop calculations. One type of pressure drop calculation that can be improved is for the shell side of shell-and-tube heat exchangers. This revised and improved method for calculating shell-side pressure drop in a shell-and-tube heat exchanger has an accuracy of -6% to +9% when compared to data available in the literature. The improved accuracy of the method means cost savings in heat exchanger surface area and pumping power. The improved accuracy also means avoiding under capacity.

The friction factor for a tube bundle depends on the tube pitch/tube OD ratio when the ideal Reynolds number is below approximately 30,000. The ratios found in the literature are from 1.208 to 1.25. The 1.208 ratio was used by Tinker in his pioneering study. The 1.25 ratio is the most commonly used in industry and is the minimum recommended by TEMA. The calculation method presented here is for heat exchangers that meet TEMA specifications. The calculated pressure drop may be low if there are leakage paths that exceed these specifications. The ideal (nonleakage) Reynolds number range studied is from 750 to 185,500, although most of the experimental data is for Reynolds numbers from 8,700 to 66,000. The method proposed here starts to have a negative deviation of calculated pressure drop compared to experimental when the Reynolds number drops below 850. There is a -5% deviation at a Reynolds number of 750, and a more negative deviation below 750.

Three well-known calculation procedures for calculating shell-side pressure drop in a shell-and-tube heat exchanger are the Bell-Delaware, HEDH and stream analysis methods. HEDH is the section in the Heat Exchanger Design Handbook that is written by Taborek. The Bell-Delaware method for shell-side pressure drop has been a standard for many years in the literature. The Bell-Delaware method was modified and improved in HEDH. HEDH has improved the turbulent friction factors, leakage factors and added a correction factor for the baffle spacing in the end zones. In addition, HEDH has the equations for computer applications.

The Bell-Delaware and its modification (HEDH) are more widely published than the stream analysis method and, therefore, will be utilized here. There have been improvements made in HEDH for turbulent flow pressure drop, but there is still a scatter of ± 40% of data tested and larger scatter for unusual tube layouts.

Use this new shell pressure drop method by first calculating the cross-flow pressure drop as shown in such recent reference books. Then, a new baffle window pressure drop, developed here, is added to the cross-flow pressure drop and the shell nozzle pressure drops. Included here are guidelines for calculating the shell nozzle pressure drops.

The baffle window pressure drop is the Achilles’ heel of shell-side pressure drop calculations. The following improves the accuracy with new turbulent pressure drop equations for the baffle window. You can use the new baffle window pressure drop with either the HEDH or the stream analysis methods. One new variable is needed for the baffle window pressure drop: a stream distortion factor of the shell-side flow stream profile. The stream distortion factor will be explained later.

**Bundle bypass areas.** Shell-side pressure drop is difficult to predict with a high degree of accuracy. Calculating shell-side pressure drop is complicated because of the constant changing fluid velocity across the bundle and leakage flows that bypass the bundle nest. There are leakage paths throughout the shell-side because of manufacturing restrictions. Generally, the largest leakage area is that around the bundle. Instead of the fluid flowing into the bundle, it bypasses around it (Fig. 1).

The bundle bypass flow area depends on the bundle-diameter-to-shell clearance, baffle spacing and the number of tube pass lanes. If there is more than one tube pass, there will be an open lane or lanes in the tube layout (Fig. 2). Those tube pass lanes in line with the cross-flow need to be taken into account. An effective portion of the pass lane area is added to the bundle bypass area. HEDH suggests using 0.5 of the tube pass lane area. Usually a pass lane is 0.625 in. wide.

The next largest leakage path is between the tubes and baffle holes (Fig. 3). TEMA requires the baffle holes to be ½ in. larger than the tube diameter when the baffle spacing is 18 in. or less and the tube size is 1¼ in. or smaller. The smallest leakage path is between the baffles and the shell (Fig. 4). The baffle-to-shell clearance is usually ⅛ in. to ⅜ in. and this clearance leakage path varies with the shell size.

**Pressure drop components.** Total shell-side pressure drop is the sum of four components:

\[ \Delta P_{total} = \Delta P_c + \Delta P_{ends} + \Delta P_w + \Delta P_{noz} \]  

The terms \( \Delta P_c \) for the interior cross-flow sections and \( \Delta P_{ends} \) for the end zones are still the same as calculated by Taborek. \( \Delta P_{noz} \) is for the nozzles and entrance effects and will be discussed later. \( \Delta P_w \) is the baffle window pressure drop.
Existing baffle window pressure drop equation. The baffle window pressure drop equation has remained basically the same since published by Bergelin et al. in 1954:

$$\Delta P_w = \frac{W_s^2 (2 + 0.6N_{cw}) N_p R_l}{2gcS_mS_w\rho}$$

Friction losses in the baffle window are not taken into account. The $R_l$ term for baffle leakage was not in the original equation. Two velocity heads are used as a base and an additive factor for the number of tube rows crossed in the window is used. The 0.6 factor is based on results for pure cross-flow. Actually, there will be angle components to the flow, and the calculated pressure drop results will usually be high. Both the two velocity heads and the 0.6 factor will contribute to a high-pressure drop calculation result at high Reynolds numbers. The new method makes Eq. 2 obsolete by incorporating more variables for better accuracy.

**Required factors.** To improve accuracy, friction and stream distortion need to be considered. The factors that affect the window pressure drop are:

- Window friction factor
- Number of effective tube rows crossed in the baffle window
- Fluid stream distortion
- Leakage streams in the cross-flow section.

**Friction factors.** Friction factors for flow in the baffle windows are not available in the literature. As an alternative, friction factors for an ideal cross-flow tube bank are used in this revised method and supplemented by other variables. The two common literature sources for friction factors in the laminar zone but not in the turbulent zone. In this revised method, HEDH friction factors are used. They are used because the research is for the larger and more commonly used shell sizes and because the friction factors are curve-fitted for computer applications.

HEDH is lower in the turbulent zone than in reference 3. HEDH friction factor equations are available in that publication for developing software. When the friction factor equations in that publication are plotted, the curve fits are good in areas where the friction factor is a straight line on log-log graphs. For the curve in the transition zone, however, the new method incorporates adjustments to improve accuracy in that region. Contact the author for the curve fit adjustments.

The graphs and curve fits of Tabor in HEDH do not extend above a $Re$ number of 100,000. Data from various sources show that the friction factor for 30° triangular tube pitch has a slight slope above 100,000. The friction factor for 90° square tube pitch has no slope. In this study the friction factor for 100,000 was used for all $Re$ numbers higher than 100,000.

**Shell fluid stream distortion.** The velocity profile is distorted. It is not like the tube side, where there is a symmetrical paraboloid profile. There are chaotic velocity fluctuations, recirculating regions and shedding of vortices. Rather than trying to calculate the expansion and contraction losses in a baffle window, a variable was developed named a distortion factor. An ideal baffle cut and spacing is shown in Fig. 5. There is very little distortion of the fluid stream. At small and large baffle cuts, inefficient eddy currents create stream distortion.

When the baffle cut, based on diameter, is in the range of 24% and 29%, there is little distortion of the shell-side stream due to eddies and recirculation.

Outside this baffle cut range the stream profile becomes more chaotic and inefficient. This distorted stream profile adds extra pressure drop. There are limited data available for large and small baffle cuts. The rudimentary equations for the distortion factor are:

- Below a baffle cut of 24%: $D = 1.0 + 2.35(0.24 - B_c)$
- Above 29% baffle cut: $D = 1.0 + 1.9(B_c - 0.29)$

where $B_c$ is the baffle cut as a fraction of the shell ID. Continued
Baffle window pressure drop. The baffle window pressure drop equation was developed from data in the literature. A good data source is the Argonne National Laboratory, Halle, Chenoweth and Wambsganss,9 using Argonne data, published a large set of experimental data on an industrial size 24-in. OD heat exchanger. They measured the shell-side pressure drop for 24 different bundle configurations. Water was used as a fluid, and the resulting Reynolds numbers varied from 8,700 to 66,000 for a full bundle and bare tubes. Other data sources are Tinker1 and Silvester and Doyle.10 Most of the literature data is for 30° triangular layout, and 90° square is next.

The baffle window pressure drop is calculated by using a velocity head equation:

$$\Delta P_w = \frac{K_p 0.000108 G_w^2 N_b}{\rho}$$

(5)

where $K_p$ is a pressure loss coefficient, $G_w$ is the nonleakage mass velocity in the baffle window, $N_b$ is the number of baffles and $\rho$ is the fluid density.

A basic part of the new pressure drop equation is a nonleakage expression:

$$f_l C_l N_{cw} D$$

(6)

where $f_l$ is the friction factor for an ideal tube bank, $C_l$ is a constant, $N_{cw}$ is a function of the number of tube rows in the baffle window and $D$ is a dimensionless distortion factor that is defined above. $N_{cw}$ is defined as originally expressed by Bell (0.8 $L_i/P_p$).3 This Bell expression uses the entire baffle window cut dimension rather than the more geometrically accurate expression, where the gap between the bundle and shell is subtracted from the cut. The advantage of the Bell expression is to allow data from smaller test heat exchangers to be used.

The leakage term, $K_p$, depends on the ratio of $S_l/S_w$, where $S_l = S_{lb} + S_{lb}$. The baffle leakage term, $R_l$, in Eq. 2 only depends on $S_l$. Ishigai et al.11 used 3.5($S_l/S_w$). A slightly more complicated term is used here. It includes the effect of friction. It is:

$$2.0 f_l \left(\frac{S_l}{S_w}\right)^2$$

(7)

This leakage term is subtracted from the nonleakage expression above. The friction factor, $f_l$, is factored out and:

$$K_p = f_l \left[ (C_l N_{cw} D) - 2 \left(\frac{S_l}{S_w}\right)^2 \right]$$

(8)

The $C_l$ constant depends on the tube layout pattern. The values are:

- 30° triangular: 2.2
- 90° square: 3.64
- 45° square rotated: 2.29
- 60° triangular: 1.79 estimate

There are no published charts for 60° triangular tube pattern friction factors. The above constant was derived from using 0.78 of the square rotated friction factor. The 45° square rotated tube layout gives higher pressure drops than the 30° triangular tube pattern when the Reynolds number is below approximately 100,000. Above 100,000, the pressure drops are nearly equal.

This $K_p$ equation is good above a Reynolds number of 800. Below 800, it calculates low pressure drops. The minimum number of tube rows effective in the baffle window investigated was four.

Nozzle pressure drops. $\Delta P_{max}$ is the total pressure drop at the shell entrance and exit. The nozzle pressure drop is difficult to predict accurately. It involves more than calculated nozzle pressure drops using expansion and contraction calculations. There is also a complex flow pattern involving eddies, recirculation and a matrix of tubes.

The HEDH12 has a simple equation for the total shell-side nozzle pressure drops. The handbook uses 1.0 velocity heads for expansion and 0.58 for contraction for a total of 1.58. As will be explained later, this is only approximately true and that is when the nozzle flow area equals the shell entrance or exit area.

Nearly all the available shell-side pressure drop data in literature are from test heat exchangers that have the tubes encroaching on the shell nozzle entrance and exits. Then the entrance and exit areas are not typical. They are smaller than the shell nozzle flow areas. Usually the shell entrance and exit areas are designed to be approximately equal to the nozzle flow areas.

It is apparent from measured data from Argonne National Laboratory9 that always basing the pressure drops on the cross-sectional area of the shell nozzle gives erratic results. The type of tube pattern involved also has an effect. Therefore, it was decided to use the shell entrance and exit areas rather than the shell nozzle.
flow areas as a basis to calculate the $\Delta P_{noz}$ term. The entrance and exit areas are also required to calculate their TEMA shell velocity limits. TEMA\textsuperscript{2} has equations for shell entrance and exit areas.

Total nozzle pressure drop is:

$$\Delta P_{noz} = K_n \times 0.000108 \times V_s^2 \times \rho$$ \hspace{1cm} (9)

where $K_n$ is the total pressure loss coefficient of the inlet and outlet nozzles. $V_s$ is the velocity in the entrance and exit areas based on the TEMA calculation.

In analyzing the nozzle pressure drop data,\textsuperscript{9} it became apparent that the value for $K_n$ to use for calculations varied significantly from the 1.58 mentioned previously. The value of $K_n$ to use varied from less than 0.7 to approximately 1.8. The lowest value is when the tubes are closest to the nozzles and when the slot area between the tubes is a large percentage of the shell entrance or exit area. An example of large slot area percentage is when the shell nozzle size is a large percentage of the shell size. The tubes act as straightening vanes to the shell stream, thus reducing turbulence and pressure drop in the inlet/outlet baffle zones. Martin et al.\textsuperscript{13} found in this case the tubes made the recirculating regions smaller.

Available data from the literature are from units with tubing that is closer than normal to the shell nozzles. As an example, the entrance and exit area of the 14 in. nozzle of the Argonne data is only 45.8 in.\textsuperscript{2} while the nozzle flow area is 137.9 in.\textsuperscript{2}. Therefore, a conservative value of 1.0 was used for $K_n$ with the Argonne data. For the square tube pattern, a factor of 0.77 was used.

These pressure loss coefficients check out when comparing different nozzle sizes and the same bundle configuration using the Argonne data.\textsuperscript{9} These low values of $K_n$ should not be used unless the shell entrance or exit area is much smaller than the nozzle flow area. None of the experimental data in the literature had an impingement plate. An impingement plate adds extra pressure drop due to turning of the inlet fluid stream.

**Example.** Using the same example as that in the *Hemisphere Heat Exchanger Design Handbook*:

- $\Delta P_c = 0.565$ psi (from HEDH)
- $\Delta P_{ends} = 0.493$ psi (from HEDH)
- $B_s = 9.84$ in.
- $D_s = 23.27$ in.
- $f_i = 0.11$ based on $Re$, of 20,695 for 30° triangular layout
- $L_s = 5.58$ in.
- $N_b = 8$
- $N_{cw} = 6.29$
- $S_s = 16.02$ in.\textsuperscript{2}
- $W_s = 177,780$ lb/hr
- $\rho = 46.2$ lb/ft\textsuperscript{3}

HEDH has a value of 43.6 in.\textsuperscript{2} for $S_w$, the net free area for flow in the baffle window. This study is based on the Bell equation, which gives a value of 41.9 in.\textsuperscript{2}. Then the value for the nonleakage mass velocity in the baffle window is:

$$G_w = 0.04 \times \frac{W_s}{S_w}$$
$$= (0.04)(177,780)/41.9$$
$$= 169.6 \text{ lb/ft}^2/\text{sec}$$
Since the baffle cut of 24% is in the ideal range, the distortion factor, \( D = 1.0 \).

\[
K_p = f_i \left[ 2.2 N_{cw} D - \left( \frac{S_l}{S_{sw}} \right)^2 \right]
\]

\[
= 0.11 \left( (2.2)(6.29)(1.0) - 2(0.382)^2 \right) = 1.49
\]

\[
\Delta P_w = \frac{1.49(0.000108)(169.6)^2(8)}{46.2} = 0.801 \text{ psi}
\]

Since the handbook excluded the nozzle pressure drop, then the total shell-side pressure drop, excluding \( \Delta P_{n子女} \), is:

\[
\Delta P_{total} = \Delta P_c + \Delta P_{ends} + \Delta P_w
\]

0.565 + 0.493 + 0.801 = 1.86 psi

The experimental pressure drop is 1.81 psi

**Discussion.** Using the HEDH method gives low pressure drops for the measured data of the Argonne data for small baffle cuts.\(^\text{14}\)

The HEDH method calculates about one-half the measured pressure drop for 16% baffle cuts. The procedure in this publication gives an average deviation of 0% for the Argonne bundle configuration of 45° square rotated tube pitch. The largest deviation is +8.6% deviation at the lowest Reynolds number of 8,700. The 60° triangular bundle gives a -0.6% average deviation.

The minimum number of effective tube rows crossed in the baffle window that was used in this procedure is four. The negative deviation from experimental data increases for fewer rows. The smallest value found for the baffle window pressure loss coefficient \( (K_p) \) was 1.0. At this time I would not recommend using any value lower than 1.0.

**Calculated vs. experimental pressure drop.** A shell pressure drop comparison for 30° triangular pitch expressed in psi follows:

<table>
<thead>
<tr>
<th>Case</th>
<th>Calculation</th>
<th>Experimental</th>
</tr>
</thead>
<tbody>
<tr>
<td>Argonne(^\text{8}) case 1—mid range flow</td>
<td>20.5</td>
<td>20.3</td>
</tr>
<tr>
<td>14 in. nozzle on 23.25 in. ID shell</td>
<td></td>
<td></td>
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<tr>
<td>17.5 in. baffle spacing and 26% cut</td>
<td></td>
<td></td>
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<tr>
<td>Argonne case 7—lowest flow</td>
<td>5.27</td>
<td>5.24</td>
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<tr>
<td>10 in. nozzle on 23.25 in. ID shell</td>
<td></td>
<td></td>
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<tr>
<td>23.5 in. baffle spacing and 29% cut</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tinker(^\text{1}) unit 9—maximum flow</td>
<td>19.7</td>
<td>19.5</td>
</tr>
<tr>
<td>3 in. nozzle on 7.95 in. ID shell</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.6 in. baffle spacing and 30% cut</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Doyle(^\text{10})—run no. 1</td>
<td>6.73</td>
<td>6.3</td>
</tr>
<tr>
<td>6 in. nozzle on 8.625 in. ID shell</td>
<td></td>
<td></td>
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<tr>
<td>12 in. baffle spacing and 40.6% cut</td>
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</tbody>
</table>

**NOMENCLATURE**

- \( B_i \) = baffle spacing, in.
- \( C_i \) = constant in \( K_p \) equation
- \( D \) = distortion factor of the shell fluid profile
- \( D_i \) = inside diameter of shell, in.
- \( f_i \) = friction factor for an ideal tube bank
- \( g \) = gravitational conversion constant, \( 32.2 \text{ ft/sec}^2 \)
- \( G_w \) = mass velocity in baffle window, \( \text{lb/ft}^2\text{sec} \)
- \( K_u \) = pressure loss coefficient for total shell nozzle pressure drop, dimensionless
- \( K_p \) = pressure loss coefficient for baffle window pressure drop, dimensionless
- \( L_i \) = baffle cut, in.
- \( N_i \) = number of baffles
- \( N_{sw} \) = number of effective tube rows crossed in baffle window
- \( P_i \) = tube pitch parallel to flow, in.
- \( Re \) = Reynolds number for shell fluid (no leakage)
- \( R_b \) = bundle leakage factor for pressure drop
- \( S_i \) = total of leakage areas, in.\(^2\)
- \( S_{sw} \) = minimum cross-flow area near bundle centerline, in.\(^2\)
- \( S_a \) = shell-to-baffle leakage area, in.\(^2\)
- \( S_b \) = tube-to-baffle leakage area, in.\(^2\)
- \( S_n \) = net flow area in baffle window, in.\(^2\)
- \( V_i \) = velocity in shell nozzle entrance and exit areas, \( \text{ft/sec} \)
- \( W_i \) = shell-side weight flow, \( \text{lb/hr} \)
- \( \Delta P_i \) = total cross-flow pressure drop excluding baffle end zones, psi
- \( \Delta P_{ends} \) = total cross-flow pressure drop in baffle end zones, psi
- \( \Delta P_{n子女} \) = total nozzle zone pressure drops, psi
- \( \Phi \) = ratio of average viscosity/wall viscosity
- \( \rho \) = density of shell fluid, \( \text{lb/ft}^3 \)

**LITERATURE CITED**


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