## COMPACT HEAT EXCHANGERS — PART 1:

# Designing Plate-and-Frame Heat Exchangers

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Use these design charts for preliminary sizing.

Considering available heat exchanger technologies at the outset of process design (at the process synthesis stage) is not general practice. In fact, procedures established in some companies preclude it. For instance, some purchasing departments' "nightmare" is having to deal with a single supplier — instead they want a general specification that can be sent to all equipment vendors, in the mistaken belief that they are then operating on a "level playing field."

This omission is both unfortunate and costly. It results in unnecessary capital expenditure and in reduced energy efficiency. It also hinders the development of energy saving technology.

Pinch analysis is the key tool used by engineers to develop flowsheets of energy-intensive processes, where heat exchanger selection is particularly important. Yet, this tool is hindering the adoption of a more-progressive approach because of the way it is restricted to traditional heat exchangers.

Numerous articles have been published regarding the advantages of compact heat exchangers. Briefly, their higher heat-transfer coefficients, compact size, cost effectiveness, and unique ability to handle fouling fluids make them a good choice for many services.

A plate-and-frame heat exchanger (Figure 1) consists of pressed, corrugated metal plates fitted between a thick, carbon-steel frame. Each plate flow channel is sealed with a gasket, a weld or an alternating combination of the two. It is not uncommon for plate-and-frame heat exchangers to have overall heat-transfer coefficients that are three to four times those found in shell-and-tube heat exchangers.

This three-part series outlines the lost opportunities and the importance of proper heat exchanger selection. This article discusses some general aspects of plate-and-frame heat

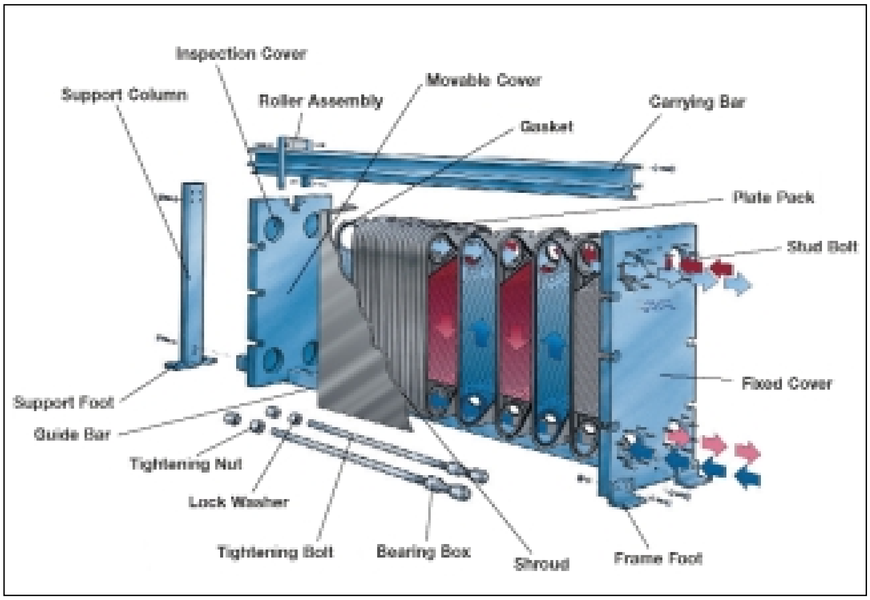


Figure 1. Cutaway drawing of a plate-and-frame heat exchanger.

| Table. Basis for heat exchanger quotation. | Tubeside | Shellside |
|---|---|---|
| **Flowrate, gal/min** | 500 | 1,800 |
| **Temperature In, °F** | 280 | 80 |
| **Temperature Out, °F** | 150 | 92 |
| **Allowable Pressure Drop, psi** | 15 | 15 |

exchangers, outlines a procedure for accurately estimating the required area, and shows how these units can be used to simplify processes. Part 2 (which will appear next month) covers integrating plate-and-frame exchangers (and other compact technologies) into pinch analysis for new plants, while Part 3 (which also will appear next month) deals with the application of plate-and-frame exchangers and pinch technology to retrofits.

## Specifying plate-and-frame heat exchangers

Engineers often fail to realize the differences between heat transfer technologies when preparing a specification to be sent to vendors of different types of heat exchangers. Consider the following example.

A process stream needs to be cooled with cooling water before being sent to storage. The stream requires C276, an expensive high-nickel alloy, to guard against corrosion; this metallurgy makes the stream a candidate for the tubeside of a shell-and-tube heat exchanger. The cooling water is available at 80°F and must be returned at a temperature no higher than 115°F. The process engineer realizes that with the water flow being placed on the shellside, larger flowrates will enhance the heat-transfer coefficient. The basis for the heat exchanger quotation was specified as shown in the table. According to the engineer's calculations, these basic parameters should result in a good shell-and-tube design that uses a minimum amount of C276 material.

A typical plate-and-frame exchanger designed to meet the specification would have about 650 ft$^2$ of area, compared to about 420 ft$^2$ for a shell-and-tube exchanger. A plate-and-frame unit designed to the above specification is limited by the allowable pressure drop on the cooling water. If the cooling water flow is reduced to 655 gal/min and the outlet water temperature allowed to rise to 115°F, the plate-and-frame heat exchanger would contain about 185 ft$^2$ of area. The unit is smaller and less expensive, and it uses less water. The load being transferred to the cooling tower is the same.

With shell-and-tube heat exchangers, increasing water flow will minimize heat-transfer area. However, with compact technologies, the effect is exactly the opposite. The larger water flow actually drives up the cost of the unit.

Rather than supplying a rigid specification to all heat exchanger manufacturers, the engineer should have explained the goal for the process stream. This could have been in the form of the following statement: "The process stream is to be cooled with cooling water. Up to 2,000 gal/min of water is available at 80°F. The maximum return temperature is 115°F." This simple statement could result in vastly different configurations compared with the designs that would result from the original specification.

## Design charts for plate-and-frame exchangers

When it comes to compact heat-transfer technology, engineers often find themselves at the mercy of the equipment manufacturers. For example, limited literature correlations are available to help in the preliminary design of plate-and-frame heat exchangers.

This article introduces a series of charts (Figures 2–7) that can be used for performing preliminary sizing of plate-and-frame exchangers. Examples will help clarify their use.

The following important points should be noted regarding the charts and their use:

1. The heat-transfer correlations apply to single-phase, liquid-liquid designs.
2. These charts are valid for single-pass units with 0.50-mm-thick plates. The accuracy of the charts will not be compromised for most materials of construction.
3. Wetted-material thermal conductivity is taken as 8.67 Btu/h-ft-°F (which is the value for stainless steel).
4. The following physical properties for hydrocarbon-based fluids were used for the basis: thermal conductivity ($k$) = 0.06 Btu/h-ft-°F, density ($\rho$) = 55.0 lb/ft$^3$, heat capacity ($C_p$) = 0.85 Btu/lb-°F. The following physical properties for water-based fluids were used for the basis: thermal conductivity = 0.33 Btu/h-ft-°F, density = 62.0 lb/ft$^3$, heat capacity = 0.85 Btu/lb-°F.
5. Accuracy should be within ±15% of the service value for the overall heat-transfer coefficient, assuming a nominal 10% excess heat-transfer area.
6. For fluids with viscosities between 100 and 500 cP, use the 100 cP line on the graphs. For fluids in excess of 500 cP, consult equipment manufacturers.

Equations 1–3 are used to calculate the log-mean temperature difference ($LMTD$) and number of transfer units ($NTU$) for the hot and cold streams. After the local heat-transfer coefficients ($h$) are read from the charts, the overall heat-transfer coefficient ($U$) is calculated by Eq. 4.

$$LMTD = \frac{\left(T_{hot,in} - T_{cold,out}\right) - \left(T_{hot,out} - T_{cold,in}\right)}{\ln\left(\dfrac{T_{hot,in} - T_{cold,out}}{T_{hot,out} - T_{cold,in}}\right)} \quad (1)$$

$$NTU_{hot} = \frac{T_{hot,in} - T_{hot,out}}{LMTD} \quad (2)$$

$$NTU_{cold} = \frac{T_{cold,out} - T_{cold,in}}{LMTD} \quad (3)$$

$$\frac{1}{U} = \frac{1}{h_{hot}} + \frac{\Delta x}{k} + \frac{1}{h_{cold}} \quad (4)$$

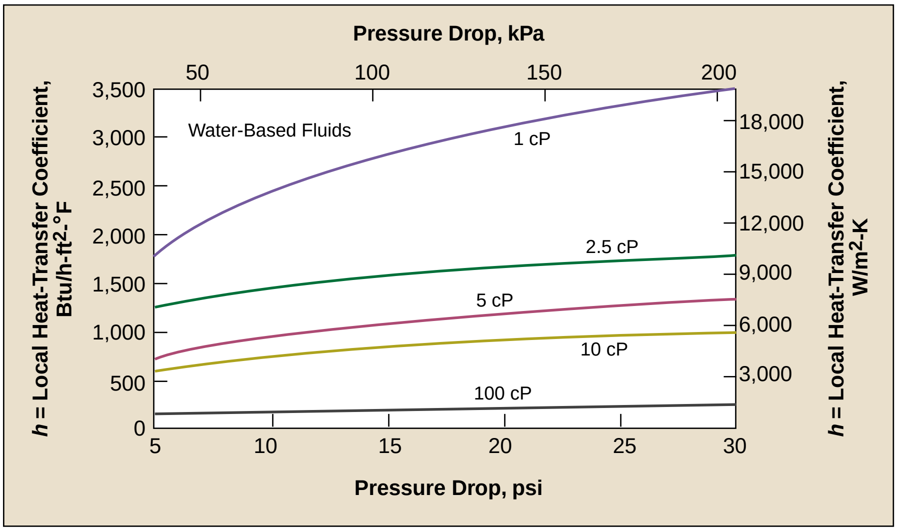


Figure 2. Heat-transfer correlations for water-based fluids, 0.25 < NTU < 2.0.

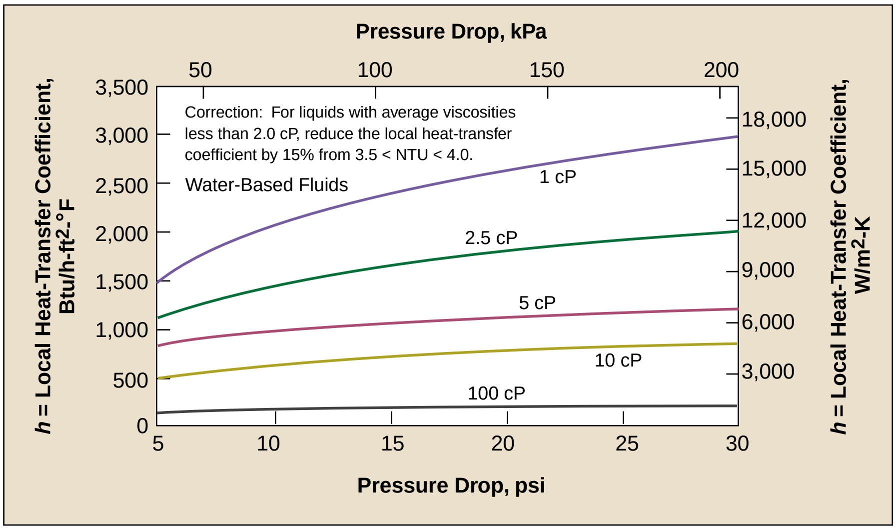


Figure 3. Heat-transfer correlations for water-based fluids, 2.0 < NTU < 4.0.

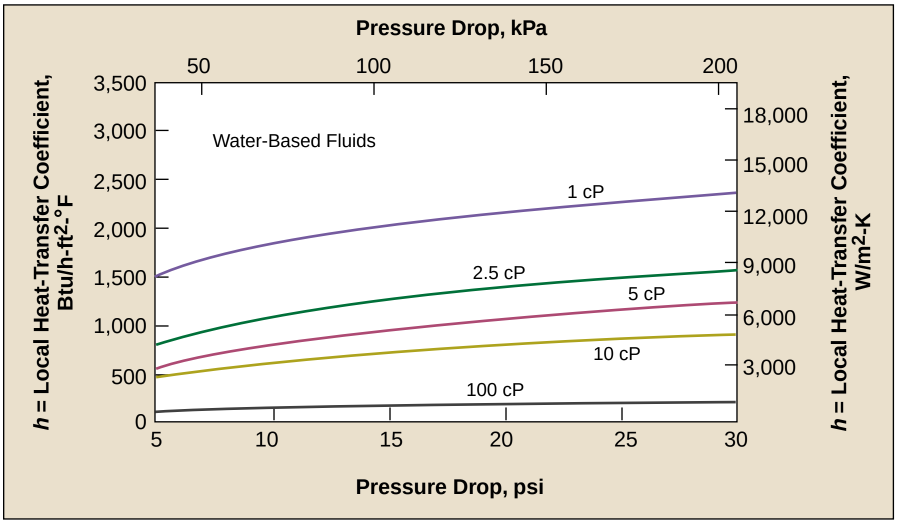


Figure 4. Heat-transfer correlations for water-based fluids, 4.0 < NTU < 5.0.

## Using the charts

Consider the following example. 150,000 lb/h of water is being cooled from 200°F to 175°F by 75,000 lb/h of SAE 30 oil. The oil enters the exchanger at 60°F and leaves at 168°F. The average viscosity of the water passing through the unit is 0.33 cP and the average viscosity of the oil in the unit is 215 cP. The maximum-allowable pressure drop through the plate heat exchanger is 15 psi on the hot and cold sides.

*Step 1: Calculate the LMTD.* From Eq. 1, *LMTD* = [(200 – 168) – (175 – 60)]/ln[(200 – 168)/(175 – 60)] = 64.9°F.

*Step 2: Calculate* $NTU_{hot}$ *and* $NTU_{cold}$. From Eqs. 2 and 3, $NTU_{hot}$ = (200 – 175)/64.9 = 0.38 and $NTU_{cold}$ = (168 – 60)/64.9 = 1.66.

*Step 3: Read* $h_{hot}$ *from the appropriate chart.* Use Figure 5, the chart for hydrocarbons when 0.25 < *NTU* < 2.0. Although there is not a viscosity line for 215 cP, the line representing 100 cP can be used for viscosities up to about 400–500 cP. The heat exchanger will be pressure-drop-limited and the heat-transfer coefficient will not change appreciably over this viscosity range for plate-and-frame exchangers. Reading from the chart, a pressure drop of 15 psi corresponds to $h_{hot} \approx 50$ Btu/h-ft$^2$-°F.

*Step 4: Read* $h_{cold}$ *from the chart.* Use Figure 2, which applies to water-based liquids when 0.25 < *NTU* < 2.0. Again, the exact viscosity line needed for pure water (0.33 cP) in this case is not available. However, the 1.0 cP line provides a very good estimate of the heat-transfer coefficient for pure water. Reading from the chart, a pressure drop of 15 psi corresponds to $h_{cold} \approx 3{,}000$ Btu/h-ft$^2$-°F.

*Step 5: Calculate U.* Assume a stainless steel plate with a thickness of 0.50 mm is being used. Type 316 stainless steel has a thermal conductivity of 8.67 Btu/h-ft-°F. Then from Eq. 4, $1/U$ = (1/50 + 0.000189 + 1/3,000) and $U$ = 49 Btu/h-ft$^2$-°F.

Now let's consider another example. 150,000 lb/h of water is being cooled from 200°F to 100°F by 150,000 lb/h of NaCl brine. The brine enters the exchanger at 50°F and leaves at 171°F. The average viscosity of the water passing through the unit is 0.46 cP and the average viscosity of the brine in the unit is 1.10 cP. The maximum-allowable pressure drop through the plate heat exchanger is 10 psi on the hot (water) side and 20 psi on the cold (brine) side.

The *LMTD* is calculated to be 38.5°F. $NTU_{hot}$ and $NTU_{cold}$ are 2.59 and 3.14, respectively. From the charts for $2.0 < NTU < 4.0$ (water based), $h_{hot} \approx 2{,}000$ Btu/h-ft$^2$-°F and $h_{cold} \approx 2{,}500$ Btu/h-ft$^2$-°F. Although the material of choice may be titanium or palladium-stabilized titanium, the properties for stainless steel are used for preliminary sizing. *U* is calculated to be 918 Btu/h-ft$^2$-°F.

## Implications of size reduction

Alternative technologies offer significant size advantages over shell-and-tube heat exchangers. Let's now consider the implications of this.

The individual exchangers are smaller, and the spacing between process equipment can be reduced. Thus, a smaller plot is needed for the process plant. If the plant is to be housed in a building, the building can be smaller. The amount of structural steel used to support the plant can be reduced, and because of the weight saving, the load on that structure is also reduced. The weight advantage extends to the design of the foundations used to support the plant. Since the spacing between equipment is reduced, piping costs are lower.

However, we stress again that the savings associated with size and weight reduction can only be achieved if these advantages are recognized and exploited at the earliest stages of the plant design.

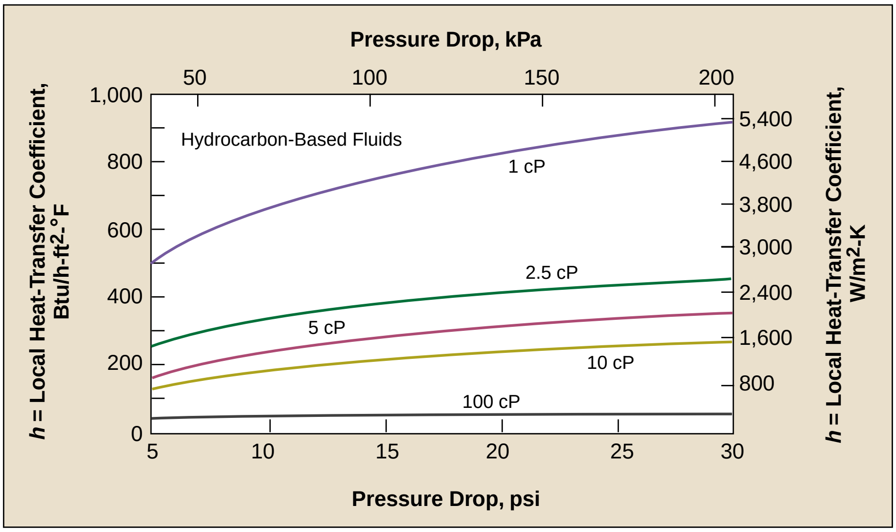


Figure 5. Heat-transfer correlations for hydrocarbons, 0.25 < NTU < 2.0.

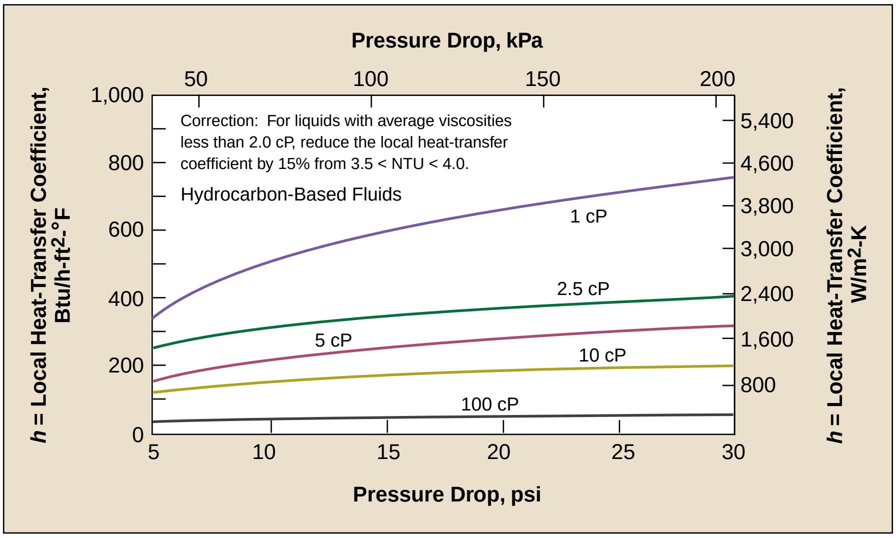


Figure 6. Heat-transfer correlations for hydrocarbons, 2.0 < NTU < 4.0.

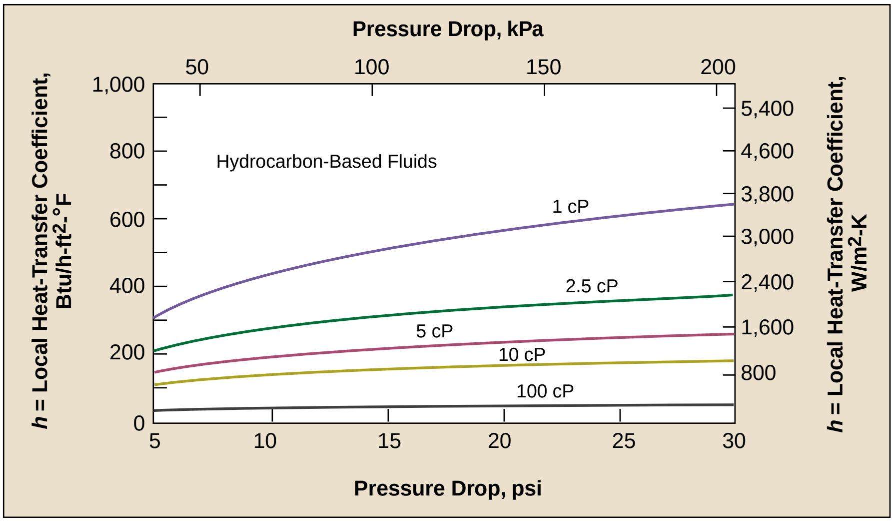


Figure 7. Heat-transfer correlations for hydrocarbons, 4.0 < NTU < 5.0.

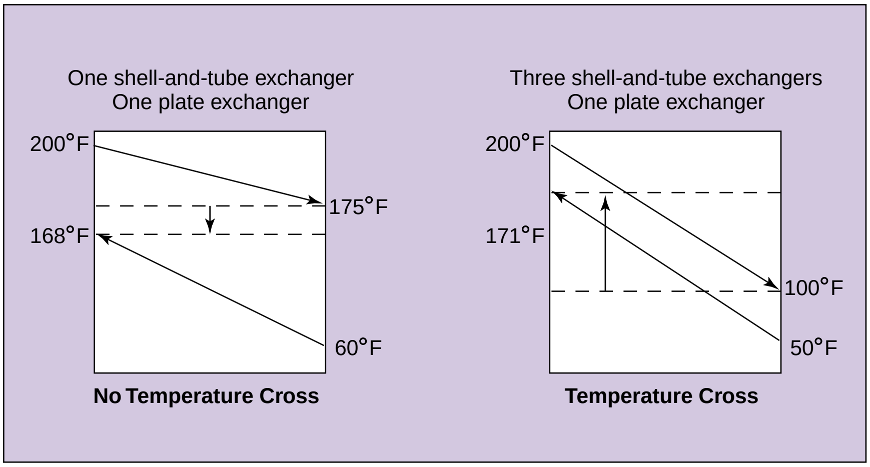


Figure 8. The situation on the left does not involve a temperature cross, while the one on the right does.

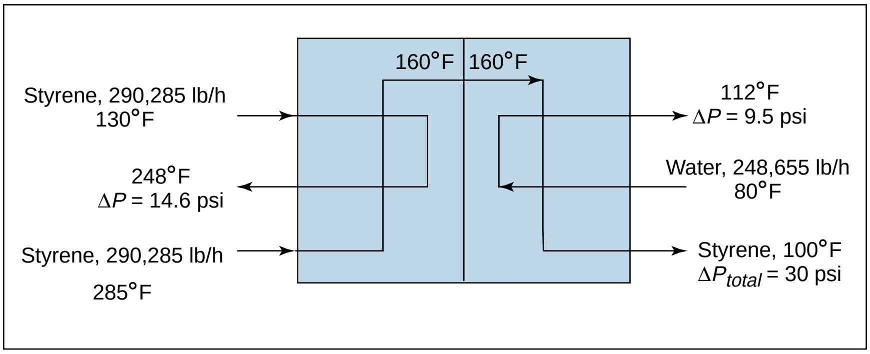


Figure 9. A single plate-and-frame exchanger handles three process streams …

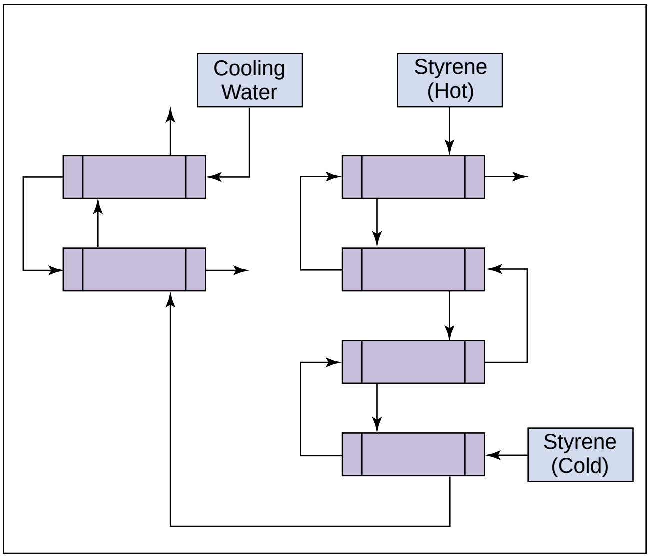


Figure 10. … whereas the equivalent shell-and-tube design requires six units.

## Reduced plant complexity

The use of alternative heat exchanger technologies can significantly reduce plant complexity by reducing the number of heat exchangers through improved thermal contacting and multi-streaming. This adds to the savings associated with reduced size and weight and also has safety implications. The simpler the plant structure, the easier it is for the process operator to understand the plant. In addition, plant maintenance will be safer, easier and more straight-forward.

Mechanical constraints play a significant role in the design of shell-and-tube heat exchangers. For instance, it is common to find that some users place restrictions on tube length. Such a restriction can have important implications for the design. In the case of exchangers requiring large surface areas, the restriction drives the design toward large tube counts. If such large tube counts lead to low tubeside velocity, the designer is tempted to increase the number of tubeside passes in order to maintain a reasonable tubeside heat-transfer coefficient. Thermal expansion considerations can also lead the designer to opt for multiple tube passes, because the cost of a floating head is generally lower than the cost of installing an expansion bellows in the exchanger shell.

The use of multiple tube passes has four detrimental effects. First, it leads to a reduction in the number of tubes that can be accommodated in a given size shell, thereby leading to increased shell diameter and cost. Second, for bundles having more than four tube passes, the pass-partition lanes introduced into the bundle give rise to an increase in the quantity of shellside fluid bypassing the tube bundle and a reduction in shellside heat-transfer coefficient. Third, it results in wasted tubeside pressure drop in the return headers. Finally, and most significantly, the use of multiple tube passes results in the thermal contacting of the streams not being pure counter-flow, which reduces the effective-mean-temperature driving force and possibly produces a temperature cross (*i.e.*, where the outlet temperature of the cold stream is higher than the inlet temperature of the hot stream, as shown in Figure 8). If a temperature cross occurs, the designer must split the duty between multiple heat exchangers arranged in series.

Many of the alternative heat-exchanger technologies allow the application of pure countercurrent flow across all size and flow ranges. This results in better use of the available temperature driving force and the use of single heat exchangers.

## Multi-streaming

The traditional shell-and-tube heat exchanger handles only one hot stream and one cold stream. Some heat exchanger technologies (most notably plate-fin and printed-circuit exchangers) can handle many streams. It is not uncommon to find plate-fin exchangers transferring heat between ten individual processes. (The principles behind the design of multi-stream exchangers and the operability of such units are discussed in Ref. 1.) Such units can be considered to contain a whole heat exchanger network within the body of a single exchanger. Distribution and recombination of process flows is undertaken inside the exchanger. The result is a major reduction in piping cost.

Engineers often overlook the opportunities of using a plate-and-frame exchanger as a multi-stream unit. As mentioned earlier, this is a common oversight when exchanger selection is not made until after the flowsheet has been developed.

A good example of multi-streaming is a plate heat exchanger that serves as a process interchanger on one side and a trim cooler on the other. This arrangement is particularly useful for product streams that are exiting a process and must be cooled for storage.

Another popular function of multi-streaming is to lower material costs. Some streams, once they are cooled to a certain temperature, pose much less of a corrosion risk. One side of the exchanger can be made of a more-expensive corrosion-resistant alloy while the other side can utilize stainless steel or a lower alloy.

Figure 9 shows a plate-and-frame unit applied to three process streams. A single exchanger with 1,335 ft$^2$ of effective surface area is used. Figure 10 is the equivalent shell-and-tube solution — to avoid temperature crosses, six individual exchangers are needed: the cooler having two shells in series (each with 1,440 ft$^2$ of effective surface area) and the heat recovery unit having four shells in series (each with 2,116 ft$^2$ of surface area). So, the plate-and-frame design involves the use of 1,335 ft$^2$ of surface area in a single unit, whereas the equivalent shell-and-tube design has 11,344 ft$^2$ of surface area distributed across six separate exchangers.

## Budget pricing correlations

For plate-and-frame heat exchangers with a design pressure up to 150 psi and a design temperature up to 320°F, the following cost equations can be used to estimate the purchased cost.

For areas ($A$) less than 200 ft$^2$:

$C = 401\,A^{0.4887}$ for Type 316 stainless steel (5)

$C = 612\,A^{0.4631}$ for Grade 1 titanium (6)

For areas larger than 200 ft$^2$:

$C = 136\,A^{0.6907}$ for Type 316 stainless steel (7)

$C = 131\,A^{0.7514}$ for Grade 1 titanium (8)

Typical installation factors for plate-and-frame heat exchangers can range from 1.5 to 2.0, depending on the size of the unit.

## Up next

Being able to estimate the area and prices of plate-and-frame heat exchangers is an important first step in including alternative heat-transfer technology in process synthesis. The remaining articles in this series (which will appear next month) focus on the integration of plate-and-frame (and other compact technology) into pinch analyses. CEP

### Literature Cited


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Correlations used to formulate charts were developed by Alfa Laval (formerly Alfa Laval Thermal) through comprehensive research and development and decades of industrial experience.