Does Your Application Call for an F-Shell Heat Exchanger?

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If the process has a temperature cross or a low flowrate, an F shell might prove beneficial.

Several shell configurations, designated E, F, G, H, J, K and X by the Tubular Exchanger Manufacturers Assocation, Inc. (TEMA), are available for shell-and tube heat exchangers. E shells are by far the most common throughout the chemical process industries. In certain situations, though, F shells offer advantages. This article discusses two such situations — when there is a temperature cross and when the shellside flowrate is low — and demonstrates why an F shell would be beneficial in those applications.

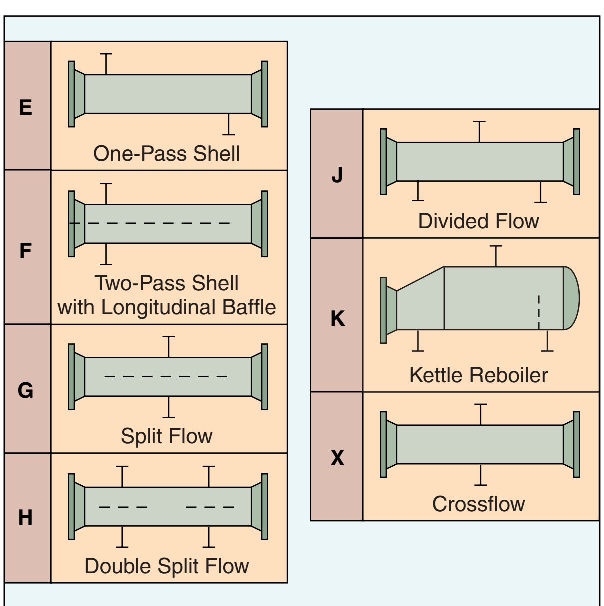

Figure 1. TEMA nomenclature for shell designs. Source: www.tema.org/TEMAfaqshtm.htm#nomenclature

Shell types

In shell-and-tube heat exchangers, there are various patterns of flow through the shellside, each with its special features and applications. TEMA has developed a nomenclature for shell types, as shown in Figure 1 (as well as for front and rear heads, which due to space limitations are not shown here).

A TEMA E shell is a single-pass shell. The shellside fluid enters at one end and leaves through the other end.

A TEMA F shell (Figure 2) is a two-pass shell that is divided into two passes, an upper pass and a lower pass, by a longitudinal baffle. The shellside stream enters at one end in either the upper or the lower half (first pass), traverses the entire length of the shell through that half of the shell, turns around and flows through the other half of the shell (second pass), and finally leaves at the same end of the shell through which it entered. The longitudinal

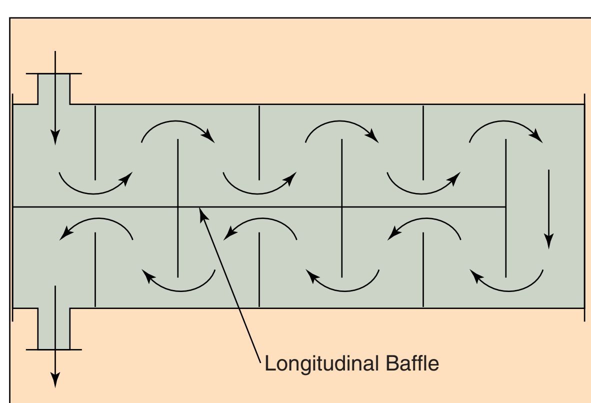

Figure 2. An F shell has a longitudinal baffle that divides it into two passes.

baffle does not extend to the tubesheet at the far end, but stops somewhat short of it so that the shellside fluid can flow from the first pass into the second pass. This construction is used for temperature-cross situations, that is, where the cold fluid leaves at a temperature higher than the outlet temperature of the hot stream. If a two-pass (F) shell has only two tube passes, it becomes a true countercurrent configuration and it can handle a large temperature cross.

A TEMA J shell is a divided-flow shell typically used for minmizing shellside pressure drop. The shellside fluid enters at the center (along the length) and divides into two halves, one flowing to the left and the other to the right; the streams leave separately and are combined into a single stream by external piping. This is referred to as a J1-2 shell. Alternatively, the stream may be split into two halves and enter the shell at the two ends, flow toward the center and leave as a single stream; this is referred as a J2-1 shell.

A TEMA G shell is a split-flow shell usually employed for horizontal thermosyphon reboilers. It has only one central support plate and no baffles. Because TEMA specifies a maximum unsupported tube length of about 60 in. or 1,500 mm for 1-in.-OD tubes, a G shell cannot be used for heat exchangers having a tube length greater than 120 in. or 3,000 mm, as the unsupported length would then exceed the TEMA limit. (The TEMA unsupported span limit varies with tube OD, thickness and material.)

When a larger tube length has to be employed, a TEMA H shell is adopted. An H shell is really two G shells placed side-by-side, so that it has two full support plates. The description for this configuration is double-split, as the flow is split twice and recombined twice. This construction, too, is invariably employed for horizontal thermosyphon reboilers. The advantage of TEMA G and H shells is that the shellside pressure drop is drastically lower than that in an E shell.

A TEMA K shell is a special crossflow shell employed for kettle reboilers (K for kettle). It has an integral vapor disengagement space in the shape of an enlarged shell. Here, too, full support plates can be employed as required.

A TEMA X shell is a pure crossflow shell. The shellside fluid enters at the top (or bottom) of the shell and flows across the tubes, then exits from the opposite side of the shell. The flow may be introduced through multiple nozzles located along the length of the shell in order to achieve a better distribution. Because of the low pressure drop (in fact, there is hardly any pressure drop in the shell, the only pressure drop being in the nozzles), this configuration is employed for cooling or condensing vapors at very low pressure, particularly at vacuum. Full support plates can be located as required for structural integrity; they do not interfere with the shellside flow since they are parallel to the flow direction.

The merits of the F shell

If a temperature cross exists, a single E shell is thermodynamically incapable of accomplishing the specifed heat duty. Depending on the degree of temperature cross, two or more E shells in series are required. In most CPI heat exchanger services, the degree of temperature cross is moderate, so two E shells can generally accomplish the task.

In such situations, a single F shell can be used instead of two E shells in series, thereby leading to savings in exchanger and piping cost. Should an F shell require only two tube passes to yield the desired tubeside velocity, it represents true countercurrent flow, which can handle any degree of temperature cross.

Often, when the shellside flowrate is relatively low, the shellside velocity is also low, even with the smallest baffle spacing. In such situations, the allowable shellside pressure drop cannot be utilized properly. The shellside heat-transfer coefficient, and thus the overall heat-transfer coefficient, are low, resulting in an unduly large and expensive heat exchanger. Additionally, if the shellside stream is dirty, the low shellside velocity will result in heavy shellside fouling, which will translate into high operating costs.

In such cases, many designers use two E shells in series. This yields a higher shellside velocity and thereby a higher heat-transfer coefficient. This leads to not only a smaller and cheaper heat exchanger, but also lower operating costs due to the reduction in fouling.

As an alternative, an F shell can often yield a comparable shellside velocity and heat transfer area. By virtue of being a single shell, such a design can have a lower capital cost than two E shells in series. There will also be a reduction in the piping cost, and the lower overall vertical height can be an advantage in many situations.

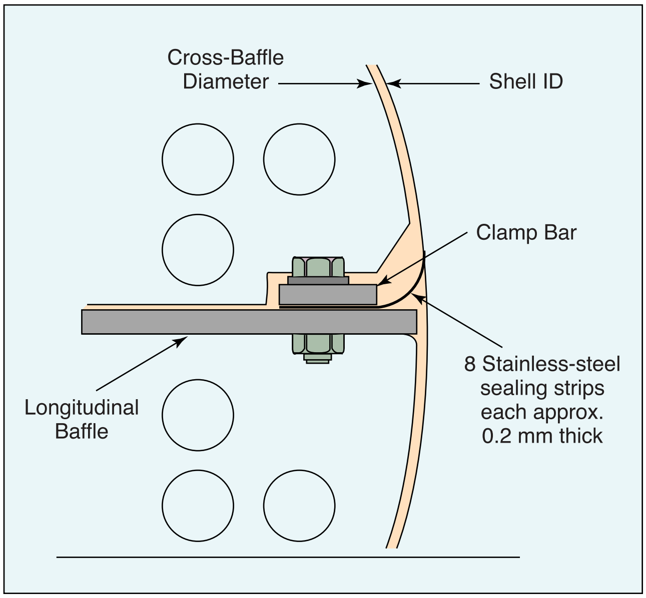

Figure 3. Stainless steel strips along the baffle can eliminate leakage.

Physical leakage

One of the limitations of the F shell is the potential for leakage. Engineers are generally apprehensive of the possibility of physical leakage of the shellside stream from the inlet pass to the outlet pass across the longitudinal baffle and the consequent deterioration in performance due to the loss in shellside performance and the loss in mean temperature difference (MTD).

However, with eight to ten pairs of thin stainless steel strips pressing against the longitudinal baffle in the first (inlet) pass (Figure 3), it is difficult to appreciate this concern. Of course, the strips are very likely to get damaged when such a tube bundle is removed from the shell and should therefore be replaced every time the bundle is taken out. In a cost-effective F shell application, this cost will be insignificant compared to the cost savings the F shell generates.

In order to minimize the possibility of physical leakage across the longitudinal baffle of an F shell, some licensors specify a maximum pressure drop of 0.35 kg/cm^2 on the shellside. This is because the higher the shellside pressure drop, the greater will be the tendency of the shellside stream to leak across the longitudinal baffle. Thus, limiting the permissible shellside pressure drop is good practice, although 0.35 kg/cm^2 might be somewhat conservative and 0.5 kg/cm^2 may be more realistic.

It follows that for services with a low allowable shellside pressure drop that conform to either of the conditions described above (temperature cross and low shellside flowrate), the use of an F shell will be more advantageous. Reactor feed/bottom exchangers with condensation and/or vaporization are a good example.

Thermal leakage

Another potential problem with F shells is thermal leakage across the longitudinal baffle from the hotter shell pass to the colder shell pass. This will adversely affect both the shellside heat-transfer coefficent and the MTD.

Thermal leakage is usually not appreciable unless the difference between the shellside inlet and outlet temperatures is high. Even if the temperature difference is high, thermal leakage can be avoided by providing a little extra heat-transfer area. Another option is to employ an insulated longitudinal baffle if thermal leakage is expected to be significant.

Commercially available software that is usually employed for heat-exchanger thermal design can evaluate both physical and thermal leakage across a longitudinal baffle.

Case study 1: Temperature cross

Consider the heat exchanger service specified in Table 1a. This was a process condensate interchanger, that is, a heat exchanger wherein heat is exchanged between the feed and effluent streams of a stripper. The streams were virtually water and as such, the physical properties of water were considered for the thermal design.

Table 1a. Process parameters for the process condensate interchanger in case study 1.

	Shellside	Tubeside
Flowrate, kg/h	7,100	6,700
Temperature In/Out, °C	255/80	40/236
Heat Duty, MM kcal/h	1.333	1.333
Allowable Pressure Drop, kg/cm^2	0.4	0.4
Viscosity In/Out, cP	Same as Water	Same as Water
Density, kg/m^3	Same as Water	Same as Water
Thermal Conductivity In/Out, kcal/h-m-°C	Same as Water	Same as Water
Specific Heat In/Out, kcal/kg-°C	Same as Water	Same as Water
Fouling Resistance, h-m-°C/kcal	0.0002	0.0002
Nominal Line Size, mm	50	50
Material of Construction	Carbon Steel	Type 304L Stainless Steel

Note the extremely high temperature cross between the two streams. This called for a pure countercurrent heat exchanger. The heat exchanger had to be designed with a TEMA BEU construction and stainless steel tubes having 25-mm O.D. and 2.5-mm thickness. Due to the U-tube construction, the tube length was not specified and any suitable length up to 6,000 mm could be selected.

Because the temperature cross was very high, five shells in series were required. The design that emerged is described in Table 1b. Although the baffle spacing is 40% of the shell I.D. (which is usually an optimum value), the stream analysis is not very good, with a main crossflow fraction of 43.6% and a shell/baffle fraction of 15.4%. The temperature profile distortion-correction factors in the five shells, from the hottest shell to the coldest, are 0.937, 0.905, 0.905, 0.907 and 0.918. Because of this and because the flow pattern between the two streams is not pure countercurrent, the overall MTD is only 16.3°C. The MTDs for the individual shells, from the hot end to the cold end, are 13°C, 14°C, 15.7°C, 18°C and 21.9°C.

The heat-transfer coefficients are quite high — 3,903 kcal/h-m^2-°C on the shellside and 1,868 kcal/h-m^2-°C on the tubeside — thereby leading to an overall heat-transfer coefficient of 702 kcal/h-m^2-°C. The shellside heat-transfer coefficient is much higher than the tubeside heat-transfer coefficient, since the allowable pressure drop is much better utilized on the shellside. There are two passes on the tubeside, and this cannot be increased to four because the tubeside pressure drop will then exceed the allowable maximum of 0.4 kg/cm^2. The net result is an exchanger with five shells, each having a heat-transfer

Table 1b. Principal construction and performance parameters for the process condensate interchanger in case study 1.

	E Shell Design, TEMA Type BEU	F Shell Design, TEMA Type BFU
Number of Shells in Series	5	2
Shell ID, mm	305	438
Total Heat-Transfer Area, m²	5 × 23.8 = 119	2 × 51.7 = 103.4
Number of Tubes × Tube Passes	50 × 2	118 × 2
Tube O.D. × Thickness × Length, mm	25 × 2.5 × 6,000	25 × 2.5 × 5,500
Tube Pitch, mm	26 Rotated Square (45-deg)	26 Rotated Square (45-deg)
Baffle Spacing, mm	120	150
Baffle Cut, % diameter	25 (Horizontal)	20 (Vertical)
Nominal Shellside Velocity, Crossflow/Window, m/s	0.24/0.21	0.3/0.3
Stream Fraction, Tube-to-Baffle Hole	0.122	0.205
Stream Fraction, Main Crossflow	0.436	0.464
Stream Fraction, Bundle-Shell	0.122	0.055
Stream Fraction, Baffle-to-Shell	0.154	0.227
Stream Fraction, Pass Partition	0.166	0.049
Shellside Heat-Transfer Coefficient, kcal/h-m²-°C	3,903	4,166
Tubeside Heat-Transfer Coefficient, kcal/h-m²-°C	1,868	937
Overall Heat-Transfer Coefficient, kcal/h-m²-°C	702	516.5
Shellside Pressure Drop, kg/cm²	0.32	0.22
Tubeside Pressure Drop, kg/cm²	0.12	0.036
Mean Temperature Difference, °C	16.3	26.1
Overdesign	3.5	6.2

area of 23.8 m². The total heat-transfer area is 119 m².

Let us now consider a design with an F shell with two tube passes. This will provide true countercurrent flow, so a single shell may suffice. However, upon analysis it was found that although a single shell would suffice, an inordinately long tube length would be required; a larger number of shorter tubes led to an unacceptably low tubeside velocity and a much larger heat-transfer area. Thus, the design that emerged has two TEMA BFU shells in series, each having tubes 5.5 m long (straight length). This is equivalent to having a single shell with 11.0-m-long tubes. Some of the more important construction and performance parameters of this configuration are indicated in the second column of Table 1b.

The heat-transfer coefficient on the tubeside is very low at 937 kcal/h-m²-°C, because the tubeside velocity is only 0.11 m/s. The allowable tubeside pressure drop is hardly utilized, but it was important to have only two tube passes so as to have pure countercurrent flow. Otherwise, the number of shells and thereby the capital cost would be considerably higher. Because the stream is clean process condensate, there are no fouling implications.

On the shellside, the heat-transfer coefficient is far better, 4,166 kcal/h-m²-°C, as the permitted pressure drop is far better utilized. Thus, the overall heat-transfer coeffcient is 516.5 kcal/h-m²-°C. This is 26.4% less than the overall heat-transfer coefficient in the BEU design.

The stream analysis reveals that both the main crossflow fraction and the baffle/shell flow fraction are higher than in the BEU design. The temperature profile distortion-correction factor is 0.963, which is much better than that for the BEU design. More importantly, there is no LMTD correction factor (F_t), as the flow is true countercurrent flow. The net result is that the overall MTD for the two shells is 26.1°C, which is 60% higher than that of the BEU design. Consequently, the total heat-transfer area is 103.4 m², compared to the 119 m² of the BEU design. Because the heat-transfer area is somewhat lower and there are only two shells, vs. the five shells in the BEU design, the BFU design will be far less expensive.

An additional advantage is that while the two shells of the BFU design can easily be stacked, it might not be possible to stack the five shells of the BEU design because it might be too difficult to remove the upper tube bundles for maintenance work. Thus, the five shells of the BEU design might require two stacks, one having three shells and the other having two, which requires a larger plot area.

Table 2a. Process parameters for the liquid-liquid heat exchanger in case study 2.

	Shellside, Stream 1	Tubeside, Stream 2
Flowrate, kg/h	22.5	344
Temperature In/Out, °C	369/300	291.6/296.2
Heat Duty, MM kcal/h	1.11	1.11
Allowable Pressure Drop, kg/cm²	0.5	0.5
Viscosity In/Out, cP	1.85/3.3	3.02/2.49
Density, kg/m³	752/780	768/761
Thermal Conductivity In/Out, kcal/h-m-°C	0.0612/0.0632	0.1/0.1
Specific Heat In/Out, kcal/kg-°C	0.739/0.719	0.7/0.7
Fouling Resistance, h-m-°C/kcal	0.001	0.001
Nominal Line Size, mm	100	300
Material of Construction	5 Cr 1/2 Mo	5 Cr 1/2 Mo

Case study 2: Low shellside flowrate

Consider the heat exchanger service specified in Table 2a. This exchanger transfers heat between two rather dirty liquid streams (fouling resistance = 0.001 h-m^2-°C/kcal) of moderate viscosity. The tubeside flowrate is rather large, and the shellside flowrate is very low.

A first design was made with a single shell, as indicated in Table 2b. What stands out about this design is that the shellside velocity is extremely low and the allowable shellside pressure drop has been very poorly utilized: while 0.5 kg/cm^2 is permitted, only 0.023 kg/cm^2 is consumed. As a result of the low velocity, the shellside heat-transfer coefficient is only 159 kcal/h-m^2-°C.

On the tubeside, too, the allowable pressure drop has not been properly utilized, being only 0.144 kg/cm^2 vs. the allowable 0.5 kg/cm^2. This is because if the number of tube passes is increased from two to four in order to increase the heat-transfer coefficient, the tubeside pressure will exceed the permissible value. Therefore, the tubeside velocity is constrained to be only 0.86 m/s, and as a result, the tubeside heat-transfer coefficient is only 317.3 kcal/h-m^2-°C.

The low tubeside and shellside heat-transfer coefficients produce an overall heat-transfer coefficient of only 85 kcal/h-m^2-°C, and the heat-transfer area is unduly high at 533 m^2.

Not only is the heat-transfer area high, the shellside velocity is unacceptably low. As the shellside stream is dirty, this would lead to severe fouling problems.

In order to improve the design by having a higher shellside velocity, the use of two shells in series was considered. However, this is a hopeless alternative, since the tubeside pressure drop, which is already 0.144 kg/cm^2 in the single-shell design, becomes excessive.

The next alternative considered was interchanging the two fluid sides, that is, to route stream 1 through the tubeside and stream 2 through the shellside. This design is also depicted in Table 2b. The shell diameter was kept the same, at 1,200 mm. In order to hamdle the much smaller flowrate on the tubeside, the number of tube passes was increased to 16. Accordingly, the number of tubes decreased from 1,446 to 1,210. The baffle spacing and cut were increased to handle the higher shellside flowrate. While the pressure drop on both the tubeside and shellside are within the specified limits, the exchanger is still highly undersurfaced. It is clear that it will not be possible to catch up with this underdesign by increasing the size of the exchanger. The tubeside film resistance is highly controlling, and the number of tube passes is already 16. This design was therefore abandoned.

Finally, an F shell was considered, which is also shown in Table 2b. The stream analysis for the F shell design is far better than the original E shell de-

Table 2b. Principal construction and performance parameters for the liquid-liquid heat exchanger in case study 2.

	E Shell Design, TEMA Type AES	E Shell with Sides Interchanged, TEMA Type AES	F Shell Design, TEMA Type AFS
Shell ID, mm	1,200	1,200	810
Heat-Transfer Area, m^2	533	447	233
Number of Tubes	1,446	1,210	630
Number of Tube Passes	2	16	2
Tube O.D. × Thickness × Length, mm	20 × 2 × 6,000	20 × 2 × 6,000	20 × 2 × 6,000
Tube Pitch, mm	26 Rotated Square (45 deg)	26 Rotated Square (45 deg)	26 Rotated Square (45 deg)
Baffle Spacing, mm	250	350	170
Baffle Cut, % diameter	21	25	21
Tubeside Velocity, m/s	0.86	0.82	1.96
Nominal Shellside Velocity, Crossflow/Window, m/s	0.078/0.071	0.87/0.82	0.34/0.3
Stream Fraction, Tube-to-Baffle Hole	0.128	0.185	0.293
Stream Fraction, Main Crossflow	0.448	0.434	0.617
Stream Fraction, Bundle-Shell	0.067	0.053	0.09
Stream Fraction, Baffle-to-Shell	0.313	0.168	0
Stream Fraction, Pass Partition	0.044	0.16	0
Shellside Heat-Transfer Coefficient, kcal/h-m^2-°C	159.3	975	449.9
Tubeside Heat-Transfer Coefficient, kcal/h-m^2-°C	317.4	80.8	918.2
Overall Heat-Transfer Coefficient, kcal/h-m^2-°C	85	63.6	177.3
Shellside Pressure Drop, kg/cm^2	0.023	0.44	0.31
Tubeside Pressure Drop, kg/cm^2	0.144	0.45	0.495
Mean Temperature Difference, °C	24.3	28.1	28.8
Overdesign	Nil	–28.3	7.3
Nominal Nozzle Size, Shellside, mm	100	100	100
Nominal Nozzle Size, Tubeside, mm	300	300	300

sign. The allowable pressure drops are far better utilized with much higher shellside and tubeside velocities, thereby leading to a much higher shellside heat-transfer coefficient (449.9 kcal/h-m^2-°C vs. 159.3 kcal/h-m^2-°C) and a much higher tubeside heat-transfer coefficient (918.2 kcal/h-m^2-°C vs. 317.4 kcal/h-m^2-°C). Consequently, the overall heat-transfer coefficient is far greater, 177.3 kcal/h-m^2-°C vs. 85 kcal/h-m^2-°C, and the heat-transfer area was reduced from 533 m^2 to 233 m^2 — a much less expensive design.

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Final recommendations

In many services involving temperature cross and in numerous situations where the permitted shellside pressure drop cannot be properly utilized with a single E shell, F shells offer a more cost-effective design with lower first cost and/or lower operating cost due to higher shellside velocity and lower fouling. Whether an F shell design is going to be superior in these situations cannot be predicted but must be established during the design stage. It would, therefore, be best if the process data sheets for heat exchangers in such services specify a choice in the shell style (E or F) and leave it to the heat exchanger designer to make the final choice. Process licensors, engineering contractors and plant owners may all work toward this eventuality. CEP