

Unlocking the Secrets of Plate-and-Frame Heat Exchangers

An understanding of the design, sizing, specification and installation of plate-and-frame heat exchangers is necessary to evaluate vendors' proposed designs

Grittaya Srinaphasawadi
Process Equipment Technology

Wiroom Tanthapanichakoon
SCG Chemicals

Plate-and-frame heat exchangers, or plate heat exchangers (PHEs), are a type of compact heat exchanger. PHEs have been widely used in the food industries, mainly because they easily meet health and sanitation requirements due to their simple disassembly for cleaning. They have recently become more common in the chemical process industries (CPI) for certain applications and operational conditions. Generally, a PHE is designed and supplied by vendors exclusively, making them a proverbial “black box” for engineers. However, it is a poor practice to rely only on vendors, who, at times, may not fully understand actual fluid properties or process requirements at a specific site. This can lead to design mistakes or equipment inadequacies.

This article covers the practical design aspects of PHEs, and explains how to design a PHE using a generally accepted method derived from heat-exchanger design fundamentals. This method can be applied to the preliminary design of a PHE and can also be used to review vendors' proposed equipment for its suitability for a required service.

What engineers should know

Using convective heat-transfer principles, a PHE achieves heat transfer via two streams of fluid that are separated by a stainless-steel corrugated plate, forming a small (usually 2–5 mm in size) flow passage. This small flow passage, along with the geometry of the PHE plates, induces turbulence, contributing to a very high convective heat-transfer coefficient — much higher than that seen in shell-and-tube (S&T) heat exchangers. Figure 1 shows the major components of a PHE.

Plate-corrugation geometry is an important feature for PHEs. Figure 2 shows the two typical types of plate-corrugation geometry, interlocking and chevron (herringbone). Chevron geometry produces greater heat-transfer enhancement for a given pressure drop and thus is more commonly used.

Gaskets are installed in a groove around the heat-transfer surface on a PHE plate to prevent fluid leakage to the outside and to avoid contamination of process fluids. Also, the gaskets help to control fluid movement by creating an alternating flow sequence between the two

fluids in the plate channels. A fluid may be prevented from entering some plate channels, but will be required to enter others.

As mentioned previously, PHEs have much higher convective heat-transfer coefficients compared to S&T heat exchangers. Some additional advantages in using PHEs rather than S&T heat exchangers are as follows:

- True countercurrent flow is easily achieved
- Surface area per unit volume is very high — usually greater than $700 \text{ m}^2/\text{m}^3$
- Equipment is lightweight and compact; only a small plot space is required
- Modifications to meet specific process requirements can be achieved by simply changing the number or form of plates
- A wide range of fluids, including ones that are very viscous, can be processed with relatively little expense
- The heat-transfer area can be easily increased or decreased
- When leaks occur, contamination of the process fluid is prevented
- Leaks are easily detectable

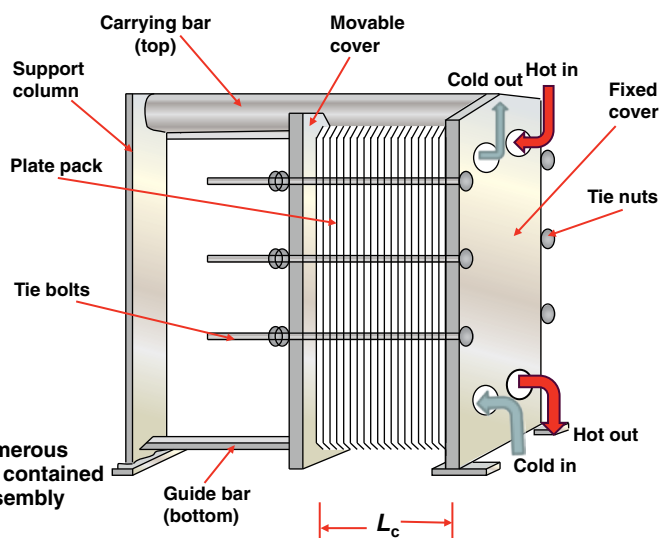


FIGURE 1. Numerous components are contained within a PHE assembly

NOMENCLATURE

<p>A_{ch} Cross-sectional flow area of one channel, m^2</p> <p>A_e Total effective heat-transfer area, m^2</p> <p>A_1 Plate true surface area, m^2</p> <p>A_{1p} Plate planar projected area, m^2</p> <p>b Mean channel flow gap, m</p> <p>β Chevron angle, deg</p> <p>C_h Heat-transfer correlation coefficient</p> <p>C_p Specific heat capacity, $J/kg\cdot K$</p> <p>D_e Hydraulic diameter, m</p> <p>D_p Port diameter, m</p> <p>Δp Pressure drop, Pa</p> <p>f Fanning friction factor</p> <p>F LMTD correction F-factor</p> <p>F_m Fouling margin, %</p> <p>h Convective heat-transfer coefficient, $W/m^2\cdot K$</p> <p>k Fluid thermal conductivity, $W/m\cdot K$</p> <p>k_w Plate thermal conductivity, $W/m\cdot K$</p> <p>K_p Empirical coefficient of the Fanning friction factor for each type of chevron plate and Reynolds number</p> <p>L_c Compressed-plate pack length measured between the two head plates, m</p>	<p>L_{horiz} Horizontal port-to-port center distance, m</p> <p>L_p Vertical distance between top port bottom edge to bottom port top edge, m</p> <p>L_{vert} Effective flow length between the vertical ports, m</p> <p>L_w Effective channel width (between gasket grooves), m</p> <p>$LMTD$ Log-mean temperature difference, $^{\circ}C$</p> <p>m_{ch} Mass flowrate through each flow channel, kg/s</p> <p>μ Fluid viscosity, $N\cdot s/m^2$</p> <p>μ_b Bulk fluid viscosity, $N\cdot s/m^2$</p> <p>μ_w Fluid viscosity at the wall, $N\cdot s/m^2$</p> <p>N_{cp} Number of channels per pass</p> <p>N_e Effective number of plates</p> <p>N_{pass} Number of passes</p> <p>N_t Total number of plates</p> <p>p Compressed pitch per plate, m</p> <p>ϕ Surface enlargement factor</p> <p>Pr Prandtl number</p> <p>Q Heat duty, W</p> <p>R_f Fouling resistance, m^2K/W</p> <p>Re Reynolds number</p>	<p>ρ Density, kg/m^3</p> <p>t Plate thickness, m</p> <p>T_1 Hot side inlet temperature, $^{\circ}C$</p> <p>T_2 Hot side outlet temperature, $^{\circ}C$</p> <p>t_1 Cold side inlet temperature, $^{\circ}C$</p> <p>t_2 Cold side outlet temperature, $^{\circ}C$</p> <p>U Overall heat-transfer coefficient, $W/m^2\cdot K$</p> <p>V_{ch} Fluid velocity in a flow channel, m/s</p> <p>V_p Fluid velocity through the port, m/s</p> <p>W Mass flowrate through either hot side or cold side, kg/s</p>
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Superscripts

- m Power exponent for Reynolds number effect on pressure drop (for calculation of Fanning friction factor f)
- n Power exponent for Reynolds number effect on convective heat-transfer coefficient

Subscripts

- | | |
|---------------|--------------|
| c Cold side | h Hot side |
| ch Channel | p Port |
| cl Clean | t Total |
| e Effective | |
| f Fouled | |

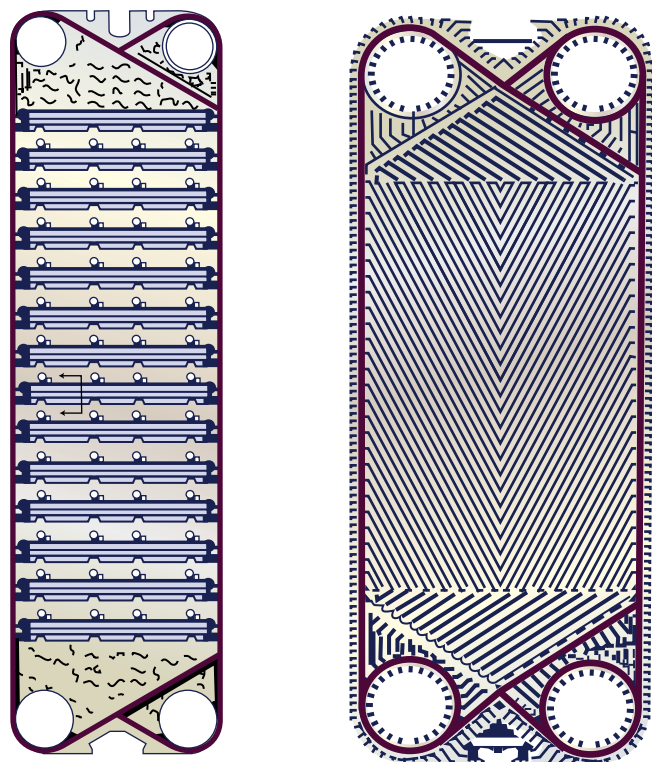


FIGURE 2. There are two typical plate types used in a PHE: interlocking type (left) and chevron type (right)

TABLE 1. TYPICAL MAXIMUM OPERATING TEMPERATURES FOR COMMON GASKET MATERIALS

Gasket material	$^{\circ}C$
Natural rubber/Neoprene	70-90
Nitrile/Viton	125-135
Butyl materials	100-155
Silicone	180-250

Some of the disadvantages associated with the use of PHEs are as follows:

- If gaskets deteriorate, especially in the presence of hazardous fluids or hydrocarbon mixtures, the equipment is more susceptible to atmospheric leaks
- A higher pressure drop is required to induce turbulent flow at smaller flow passages (2-5 mm); at the same pressure drop, a PHE may not provide the desired heat-transfer enhancement effect.
- PHEs have limited use for gas-to-gas heat-exchanger or boiling services where volume expansion is large, because the required outlet nozzle will be too big
- Gasket materials selection can impose constraints on operating pressure and temperature (See Table 1 for specific temperature limitations); conversely, in

- The ability to individually remove and clean plates decreases maintenance and cleaning efforts for both fluid sides; S&T heat exchangers are more difficult to open, especially on the shellside
- Turbulence can be achieved at a relatively low Reynolds number,

- typically less than 500
- Lower sensitivity to vibration than S&T heat exchangers
- Very heat-sensitive process fluids can be capably handled

However, there are some applications where S&T heat exchangers are a better option than PHEs.

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S&T heat exchangers, the design temperature is based upon the metallurgy of the materials of construction

- Process fluids must be clean, non-flammable, non-toxic and nearly free of particulate matter; for dirty fluids, inlet strainers must be installed
- Non-corroding metallurgy, such as austenitic stainless steel, titanium, Inconel or Hastelloy, is a requirement; S&T heat exchangers may be constructed of less expensive carbon steel, since corrosion allowances are more relaxed
- Large exposed bolts and typically elastomeric (rubber-based) gasket materials decrease fire resistance; a fireproofing shroud should be specified if fire resistance is needed
- Suitable applications may be limited by the compatibility of gasket materials with operating fluids. For example, rubber or neoprene gaskets cannot be used with aromatic oils

Based on these advantages and disadvantages, one can begin to determine in which specific applications and under which operating conditions PHEs will be appropriate alternatives to S&T heat exchangers. While S&T heat exchangers remain ubiquitous across the CPI, PHEs are gaining ground in some specific applications, including food processing and lube-oil coolers for compressors and turbines. However, it must be noted that PHEs are not suitable for gas-to-gas heat exchange, and must be used with caution in boiling and condensation services, especially where there is a large volume expansion. S&T heat exchangers remain the preferred option for these phase-change applications. Table 2 compares, in further detail, the specific operating requirements of PHEs and S&T heat exchangers.

Design methodology

PHE design remains proprietary in nature, because exact design correlations used by vendors are often undisclosed. The PHE convective heat-transfer coefficient, h , can be calculated using an in-tube convec-

TABLE 2. COMPARISON OF OPERATING AND DESIGN REQUIREMENTS FOR PHE AND S&T HEAT EXCHANGERS

	PHE	Shell & tube (S&T)
Maximum design temperature, °C	250 (based on elastomeric gasket materials)	much greater than 250 (based on metallic material)
Maximum design pressure, barg	20-25	200-300
Heat-transfer area, m ²	0.1-2,200	> 2,500-3,500 per shell is possible
Heat-transfer coefficient, W/m ² -K	3,500-7,500 (mainly liquid and water service)	100-2,500 (gas-gas, liquid-liquid, gas-liquid and phase-change services)
Pressure drop, bars	2-3	usually < 1
Heat-transfer ratio	3-5	1
Plot-space ratio	0.2-0.5	1
Fouling-factor ratio	0.1-0.25	1
Temperature cross/ Minimum temp. approach, °C	Acceptable/ 1°C	Unacceptable or needs multiple shells/ 5°C
Operating weight ratio	0.1-0.3	1
Hold-up volume	Low	High
Design standards	API 662	TEMA, API 660

tive heat-transfer coefficient correlation, but the tube inner diameter must be changed to hydraulic diameter, D_e . This design method has been validated with commercial PHE design and rating software.

The following equations use heat-exchanger fundamentals to develop a methodology for designing a typical PHE. The nomenclature section, as well as Figures 3 and 4, explain the meaning of the parameters in the following equations. First, the process requirements must be defined for the heat duty (Q) on the hot and cold sides of the exchanger, as well as the log-mean temperature difference ($LMTD$).

$$Q_h = W_h C_{p,h} (T_1 - T_2) \quad (1)$$

$$Q_c = W_c C_{p,c} (t_2 - t_1) \quad (2)$$

$$LMTD = \frac{[(T_1 - t_2) - (T_2 - t_1)]}{\ln[(T_1 - t_2)/(T_2 - t_1)]} \quad (3)$$

Equations (4) through (13) define geometry for the PHE, beginning with the effective number of plates (N_e), which is two less than the total number of plates, due to the absence of flow between the end plate and the cover, shown in Equation (4). Equation (5) gives the compressed pitch (p), which is a function of the compressed-plate pack length measured between the two

head plates (L_c).

$$N_e = N_t - 2 \quad (4)$$

$$p = L_c / N_t \quad (5)$$

Equations (6) through (9) regard channel parameters. The mean channel flow gap (b) is given by Equation (6), the vertical port distance by Equation (7) and the effective channel width by Equation (8).

$$b = p - t \quad (6)$$

$$L_p = L_{vert} - D_p \quad (7)$$

$$L_w = L_{horiz} + D_p \quad (8)$$

The flow area of one channel, shown in Equation (9), is the product of the channel's flow gap and its width.

$$A_{ch} = b \times L_w \quad (9)$$

The single-plate true heat-transfer area is defined as the ratio of the equivalent area and the number of effective plates. The total effective heat-transfer area of a PHE (or total effective area), A_e , depends on plate types and number of plates, as shown in Equation (10). The planar projected area is given in Equation (11).

$$A_1 = A_e / N_e \quad (10)$$

$$A_{1p} = L_p \times L_w \quad (11)$$

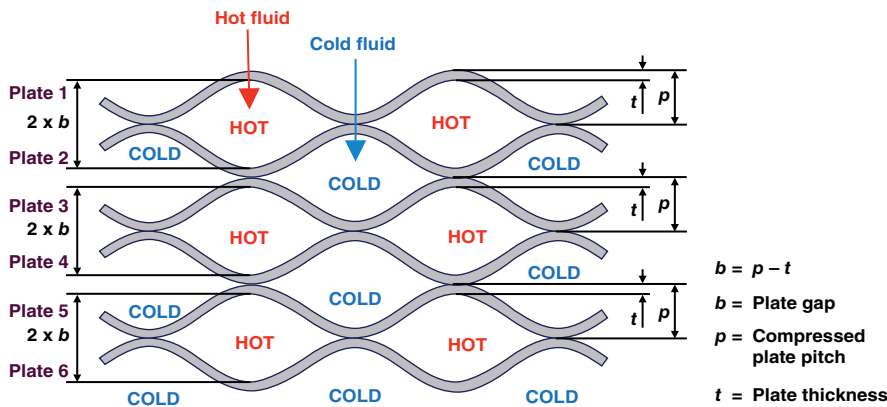


FIGURE 3. The gasket is compressed until there is metal-to-metal contact between adjacent plates, forming flow channels on the hot and cold sides

The surface enlargement factor, ϕ , is the ratio of plate true surface area to plate planar projected area. Defined in Equation (12), this value is normally between 1.20 and 1.25.

$$\phi = A_1 / A_{1p} \quad (12)$$

The hydraulic diameter is defined as four times the channel flow area divided by the wetted perimeter. In Equation (13), it is assumed that b is much smaller than L_w .

$$D_c = \frac{4bL_w}{2 \times (b + L_w\phi)} = 2b / \phi \quad (13)$$

The expressions required for heat-transfer calculations for the design of a PHE are shown in Equations (14) through (24). Equations (14) through (16) relate to the channels. Equation (14) gives the number of channels per pass, Equation (15) gives the mass flowrate to each channel and Equation (16) defines the fluid velocity in a given flow channel.

$$N_{cp} = (N_t - 1) / 2N_{pass} \quad (14)$$

$$m_{ch} = W / N_{cp} \quad (15)$$

$$V_{ch} = (m_{ch} / \rho) / A_{ch} \quad (16)$$

The Reynolds Number (Re) and the Prandtl Number (Pr), defined in Equations (17) and (18), respectively, are important parameters in determining the flow behavior of the process fluids in a PHE. Note that flow is defined as laminar at $Re < 10$, transitional at $10 < Re < 500$ and turbulent at $Re > 500$.

$$Re = (\rho V_{ch} D_e / \mu) \quad (17)$$

$$Pr = (C_p \mu / k) \quad (18)$$

Equation (19) relates a heat-transfer coefficient, h , in terms of Re and Pr . C_h and n can be found from Table 10.6 in Ref. [1] at various Chevron angles (β) and different Reynolds number values.

$$(h D_e / k)_h = C_h (Re)^n (Pr)^{1/3} (\mu_f / \mu_w)^{0.17} \quad (19)$$

Equations (20) and (21) define the overall heat-transfer coefficient (U) for clean and fouled conditions, respectively.

$$1/U_{cl} = 1/h_h + t/k_w + 1/h_c \quad (20)$$

$$1/U_f = 1/h_h + R_{f,h} + t/k_w + R_{f,c} + 1/h_c \quad (21)$$

Fouling factors for S&T heat exchangers should not be used for PHE design, because the resulting area margin would be too large. Fouling factors for PHEs are typically only 10–25% of those for S&T heat exchangers of the same service. Alternatively, a fouling margin F_m can be specified, and is defined by Equation (22). Typically, a minimum of 10% fouling margin should be specified.

$$F_m = (U_{cl} / U_f - 1) \times 100 \quad (22)$$

Understanding the effects of fouling on PHE design is crucial. This is why the heat duty must be calculated for both clean and fouled conditions, as shown in Equations (23) and (24). Note that a PHE design is acceptable when $Q_{cl} > Q_c (= Q_h)$ and $Q_f > Q_c (= Q_h)$, where F is a correction factor applied to the $LMTD$. F is equal to one when flow is countercurrent.

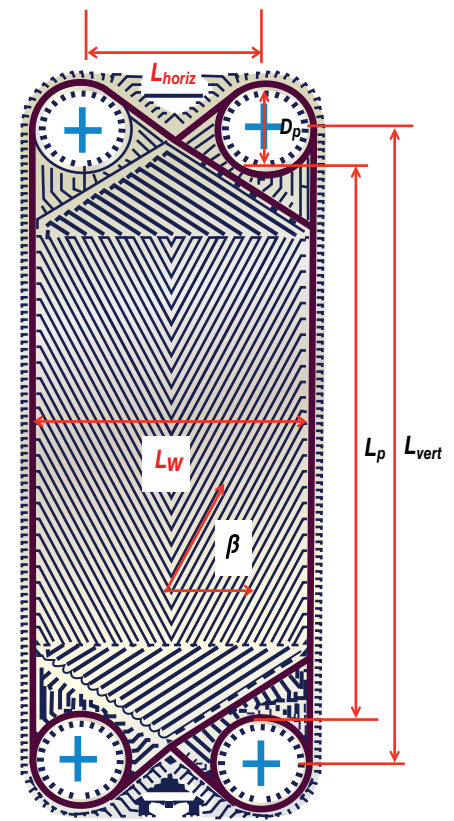


FIGURE 4. The gasket is installed in the groove near the edge of each plate

$$Q_{cl} = U_{cl} A_e F (LMTD) \quad (23)$$

$$Q_f = U_f A_e F (LMTD) \quad (24)$$

Finally, equations (25) through (29) deal with pressure-drop calculations in a PHE. First, the Fanning friction factor is defined in Equation (25), where K_p and m are found in Table 10.6 in Ref. [1]. The friction factor is crucial in determining the pressure drop in the channels, which is defined by Equation (26).

$$f = K_p / Re^m \quad (25)$$

$$(\Delta p_{ch}) = 4f(L_{vert} N_{pass} / D_e) (\rho V_{ch}^2 / 2) \quad (26)$$

Equations (27) and (28) relate to flow at the port. Equation (27) defines the fluid velocity through the port and Equation (28) defines the pressure drop at the port.

$$V_p = (W / \rho) / (\pi D_p^2 / 4) \quad (27)$$

$$(\Delta p_p) = (1.4 N_{pass} \rho V_p^2 / 2) \quad (28)$$

Finally, we arrive at the total pressure drop, shown in Equation

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TABLE 3. PROCESS SPECIFICATIONS FOR A GIVEN PHE

Parameter	Hot side	Cold side
Process fluid	Hot oil	Cold water
Mass flowrate, W , kg/s	140	130
Inlet temperature, T_1, t_1 , °C	85	20
Outlet temperature, T_2, t_2 , °C	45.0	41.5
Max allowable Δp , bars	3	3
Specific heat, C_p , J/kg-K	2,089	4,178
Viscosity, μ , N-s/m ²	5.11×10^{-4}	7.68×10^{-4}
Thermal conductivity, k , W/m-K	0.190	0.616
Density, ρ , kg/m ³	950	995
Prandtl number, Pr	5.618	5.210
Fouling margin, F_m (%)	10	10

TABLE 4. VENDOR'S PROPOSED PHE DESIGN (BASED ON 304 STAINLESS STEEL)

Parameter	Value
Plate thickness, t , mm	0.6
Chevron angle, β , deg	45
Total number of plates, N_t	105
Enlargement factor, ϕ	1.25
Number of passes, N_{pass}	1
Overall heat-transfer coefficients, U_{ci}/U_{cf} , W/m ² -K	3,520/3,200
Total effective heat-transfer area, A_e , m ²	110
Port diameter, D_p , mm	200
Compressed plate pack length, L_c , m	0.38
Vertical port distance, L_{vert} , m	1.55
Horizontal port distance, L_{horiz} , m	0.43
Effective channel width, L_w , m	0.63
Plate thermal conductivity, k_w , W/m-K	17.5

(29), which is the sum of the pressure drop at the port and the channel pressure drop.

$$(\Delta p_t) = (\Delta p_c) + (\Delta p_p) \quad (29)$$

Design example

The design method derived in the previous section will now be applied in an example scenario to confirm a vendor's design. In this example, cold water is to be heated by a hot oil stream in a PHE. Using the process specifications in Table 3 and the vendor's proposed design in Table 4, we can use Equations (1) through (29) to determine if the proposed exchanger satisfies the design requirements. Table 5 gives a summary of the calculation results using the previously derived design methodology. It can be seen from Table 5 that the proposed design is expected to achieve its desired heat transfer and not exceed the allowable pressure drop. However, it is determined that the vendor's proposed area of 110 m² is oversized for this service and can be further optimized to reduce costs.

Best practices

Design and installation. In addition to sizing a PHE, it is also beneficial for engineers to understand some practical considerations in the purchasing and installation of PHEs. A summary of best practices for installing PHEs is as follows:

- The frame and tie bolts of a PHE should be designed to permit future installation of at least 20% additional plates
- The nominal thickness of gasketed plates should be a minimum of 0.5 mm to meet design conditions. Gasketed plates shall be individually replaceable without removing any other plate
- The PHE must be designed to withstand the design pressure of each side when the pressure on the other side is atmospheric pressure, or vacuum, if specified. Thus, either hydrotest or pneumatic pressure tests can be undertaken on only one side at a time, with the other side at atmospheric pressure or vacuum
- A corrosion allowance shall be applied to unlined connections only, and there is zero corrosion allowance for plates
- Gaskets should be positioned in the groove around the heat-transfer surface
- Gaskets should be compressed to achieve metal contacts between the plates
- Each sealing gasket should be one integral piece to allow for better pressure resistance

Specifying and purchasing. Despite the dependence on vendors to provide PHEs, an understanding of PHE design data is crucial to purchasing engineers. For proper PHE design, the purchaser should spec-

ify the following design data to be provided by vendors:

- Completed datasheets
- Details of the construction and assembly
- Anticipated life of the gaskets in the specified service and in storage, and special storage guidelines, if needed to maintain gasket life
- The method used to support the removable cover
- Recommended spare-parts list
- Fireproof testing certificate for a fireproof shroud, if specified
- Verification of compatibility of the gasket material and glue with the specified fluids, including any specified chemical cleaning

Problems with very large PHEs

There are some hidden structural constraints that often cause problems with the use of PHEs. These structural issues are especially concerning when manufacturers offer very large equipment to purchasers. The constraints in question are the total number of plates used and the total weight of the plate pack assembly.

In a normal flanged joint, the stud bolts used to clamp the joint must be properly torqued to provide sufficient clamping force, giving the gasket the adequate seating stress to seal the internal pressure. With the required clamping force, the bolt stress will also be an acceptable value. With this principle,

TABLE 5. SUMMARY OF CALCULATION RESULTS

Variable	Equation	Calculations	Value
Q_h	(1)	$140 \times 2,089 \times (85-45)$	11,698.4 kW
Q_c	(2)	$130 \times 4,178 \times (41.5-20)$	11,677.5 kW
$LMTD$	(3)	$((85-41.5) - (45-20))/\ln((85-41.5)/(45-20))$	33.4°C
Heat balance: Q_h is within 1% of $Q_c \rightarrow$ Good			
N_e	(4)	$105 - 2$	103
p	(5)	$0.38/105$	0.00362 m
b	(6)	$0.00362 - 0.0006$	0.00302 m
L_p	(7)	$1.55-0.20$	1.35 m
L_w	(8)	$0.43 + 0.20$	0.63 m
A_{ch}	(9)	0.0030×0.63	0.00190 m ²
A_1	(10)	$110/103$	1.068 m ²
A_{1p}	(11)	1.35×0.63	0.85 m ²
ϕ	(12)	$1.068/0.85$	1.256
D_e	(13)	$2 \times 0.00302/1.256$	0.00481 m
N_{cp}	(14)	$(105-1)/2(1)$	52
$m_{ch,h}$	(15)	$140/52$	2.69 kg/s
$m_{ch,c}$	(15)	$130/52$	2.50 kg/s
$V_{ch,h}$	(16)	$(2.69/950)/0.00190$	1.49 m/s
$V_{ch,c}$	(16)	$(2.5/995)/0.00190$	1.32 m/s
Re_h	(17)	$950 \times 1.49 \times 0.00478/0.000511$	13,320 (turbulent)
Re_c	(17)	$995 \times 1.44 \times 0.00478/0.000768$	8,230 (turbulent)
$(hD_e/k)_h$	(19)	From Table 10.6 in [7], at $\beta = 45$ deg and $Re > 100$, $C_h = 0.3$, $n = 0.663$ Assume $\mu_b = \mu_w \rightarrow 0.3 \times (13,320)^{0.663}(5.618)^{1/3}$	289.4
h_h	(19)	$289.4 \times 0.190/0.00481$	11,436 W/m ² -K
$(hD_e/k)_c$	(19)	$0.3 \times (8,230)^{0.663}(5.21)^{1/3}$	205.1
h_c	(19)	$205.1 \times 0.616/0.00481$	26,272 W/m ² -K
U_{cl}	(20)	$1/(1/11436+0.0006/17.5 + 1/26272)$	6,258 W/m ² -K
U_f	(22)	$100\%/(100\%+10\%) \times 6258$	5,689 W/m ² -K
F		$F = 1$ for true countercurrent flow	1
Q_{cl}	(23)	$6,258 \times 110 \times 1 \times 33.4$	22,993 kW
Q_f	(24)	$5,689 \times 110 \times 1 \times 33.4$	20,903 kW
Heat transfer results: U_{cl} and U_f are greater than the vendor's proposed values \rightarrow Good			
Q_{cl} and Q_f are greater than the required heat duty of 11,698.4 kW \rightarrow Good.			
f_h	(25)	From Table 10.6 in [7], $K_p = 1.441$, $m = 0.206 \rightarrow 1.441/(13320)^{0.206}$	0.204
$(\Delta p_{ch})_h$	(26)	$4 \times 0.204 \times (1.55 \times 1/0.00481) \times (950 \times 1.492/2)$	276,989 Pa
$V_{p,h}$	(27)	$(140/950)/(\pi(0.2)^2/4)$	4.69 m/s
$(\Delta p_p)_h$	(28)	$1.4 \times 1 \times (950 \times 4.69^2/2)$	14,633 Pa
$(\Delta p_i)_h$	(29)	$276,989 + 14,633$	291,622 Pa (2.92 bars)
f_c	(25)	$1.441/(8230)^{0.206}$	0.225
$(\Delta p_{ch})_c$	(26)	$4 \times 0.225 \times (1.55 \times 1/0.00481) \times (995 \times 1.32^2/2)$	251,810 Pa
$V_{p,c}$	(27)	$(130/995)/(\pi(0.2)^2/4)$	4.16 m/s
$(\Delta p_p)_c$	(28)	$1.4 \times 1 \times (995 \times 4.16^2/2)$	12,046 Pa
$(\Delta p_i)_c$	(29)	$251,810 + 12,046$	263,856 Pa (2.64 bars)
Pressure drop results: $(\Delta p_i)_h$ and $(\Delta p_i)_c$ are less than 3 bars \rightarrow Good			

for the same internal pressure and the same type and size of gasket, the flanged joint with two gaskets would need more clamping force than the joint with a single gasket — this means that the total number of bolts will have to be increased to result in that same bolt stress. The softer gasket (the one with a lower gasket factor) would normally need less clamping force than the harder one. With the rubber gaskets typically used in PHEs, it is easier to provide a small clamping force and properly torqued tie bolts.

However, if a large number of plates is present, an even higher number of tie bolts (and a larger total bolt cross-sectional area) would be required. A large number of small tie bolts will give a more uniform gasket pressure and result in much better sealing performance than a small number of large bolts having the same bolt total cross-sectional area.

In one case, a manufacturer designed a PHE with 500 plates assembled into one pack. This makes it extremely difficult to have all 500 gaskets properly compressed to seal the internal pressure of the process — no matter the torque value or number of tie bolts, and even though the manufacturer applied an ample amount of adhesive to the gasket to provide the seal. Bear in mind that the total weight of the entire plate assembly is hanging on the carrying bar. The higher the number of plates used, the heavier and longer the plate assembly is, imposing an even greater weight on the bar, which acts as a uniformly distributed load beam. According to the beam theory, the deflection of the beam varies linearly with the load and with the cube of the beam length. This makes the plate assembly more susceptible to leakage, especially at its bottom. Therefore, the purchaser should ensure that the PHE is not too large in size and the total number of plates in the assembly should be carefully considered.

Understanding the advantages and disadvantages of PHEs compared with S&T heat exchangers allows one to achieve a more

efficient and more cost-effective design of a heat-exchange system under specific and appropriate conditions. Even though PHE design remains proprietary to vendors, plant engineers should be able to evaluate a vendor's PHE design for correctness using the methods explained above. ■

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Authors



Grittaya Srinaphasawadi

is the managing director of Process Equipment Technology Co. (555 Moo 4 Tambol Makamkoo, Ampur Nikompattana, Rayong, 21180, Thailand; Phone: +66-38-893628-9, +66-81 8623910; Email: grittaya@petco.co.th). He obtained B.S. and M.S. degrees in mechanical engineering from King Mongkut University of Technology Thonburi (KMUTT), Thailand in 1977 and 1979, respectively. After spending one year working as a lecturer with the mechanical engineering faculty at KMUTT, he joined Thai Oil Co., Thailand's largest oil refinery, and continued working there until 1997, assuming the positions of maintenance engineer, project engineer, and head of Engineering Technical Services. He also worked as the plant manager of Thai Carbon Products Co., a subsidiary of the company. During his 17 years with the company, he gained experience in various process-plant equipment design, selection, construction and troubleshooting. His expertise includes centrifugal pumps, steam turbines, gas turbines, reciprocating and centrifugal compressors, heat exchangers, pressure vessels, fired heaters, boilers, piping systems, vibration analysis and more. He has maintained a level 3 (maximum-level) license as a professional mechanical engineer since 1988.



Wiroon Tanthapanichakoon

is a lead engineer working for the process technology center of SCG Chemicals Co. (ROC Site 3 Technic Building, 271 Sukhumvit Rd., Map Ta Phut, Rayong, 21150, Thailand; Phone: +66-38-911-240, +66-83-177-8108; Email: wiroont@scg.co.th, twiroon@gmail.com). He has authored several international papers and technical articles. He is currently involved in process and equipment design, as well as basic engineering for new process technologies from SCG Chemicals. He has several years of experience as a process engineer with an ExxonMobil subsidiary refinery in Thailand and as a process design and energy improvement engineer with SCG Chemicals. He obtained his B.S.Ch.E. and M.S.Ch.E. degrees from Kyoto University, Japan. He is a licensed engineer in Thailand and a senior member of AIChE. A Microsoft Excel spreadsheet developed by the author for the example problems is available by request to the author's email address.