Gooch Thermal Systems

SPIRAL PLATE HEAT EXCHANGERS:

Sizing Units for Cooling Non-Newtonian Slurries

This article presents step-by-step guidance to demystify the sizing of these exchangers

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Spiral plate heat exchangers are ideal for cooling slurries and viscous fluids. The performance of these units is characterized by increased turbulent heat transfer, reduced fouling, greater ease of maintenance, and more compact size compared to many competing options.

In a spiral plate heat exchanger, the hot fluid enters at the center of the unit and flows from the inside outward (Figure 1). The cold fluid enters at the periphery and flows toward the center. Heat transfer is carried out by the countercurrent flow that is achieved. Both fluid streams flow in identical passage configurations, and therefore have the same heat transfer and pressure drop characteristics.

Heat transfer analysis

The amount of heat exchanged between the hot and cold fluids inside a spiral plate heat exchanger can be found by performing a simple energy balance around the appropriate section of the exchanger using this general relationship:

Energy lost by the hot fluid = Energy gained by the cold fluid + Energy lost to the surroundings

The energy lost to the surroundings is assumed to be negligible.

The actual heat balance in a spiral plate heat exchanger is calculated by applying the first law of thermodynamics (see p. 48 for Nomenclature):

$$Q = m_h c_h (T_{hi} - T_{ho}) = m_c c_c (T_{co} - T_{ci}) \quad (1)$$

The overall heat transfer coefficient, U, provides a common way to express the heat transfer rate for a given system. A detailed derivation of the overall heat transfer coefficient can be found in any heat transfer textbook. The result can be written as:

$$Q = U A f \Delta T_{LM} = U A f (LMTD) \quad (2)$$

The overall heat transfer coefficient, U, is:

$$U = \frac{1}{\frac{1}{h_h} + \frac{t}{k_P} + \frac{1}{h_c} + R_f} \quad (3)$$

[Ref. 2, p. 60]

The logarithmic mean temperature difference (LMTD) between the inlet and outlet streams is determined using Equation (4).

$$LMTD = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln \frac{(T_{hi} - T_{co})}{(T_{ho} - T_{ci})}} \quad (4)$$

The heat transfer surface area, A, represents both sides of the plate that is used to form the spirals inside the heat exchanger:

$$A = 2LH \quad (5)$$

The maximum plate width that fabricators of today's spiral-plate heat exchangers have available is 72 in. The length of this plate is adjusted to provide an optimum heat transfer surface and acceptable pressure drop. The values of these parameters are assumed at first in this iterative process, and then checked to ensure that the proposed values will provide a good heat transfer surface and an allowable pressure drop.

Combining Equations (2), (4) and (5) (and setting f = 1; f is a correction factor for countercurrent flow in a spiral plate heat exchanger) gives the following relationship:

$$Q = U A (LMTD)$$
$$= U(2LH) \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln \frac{(T_{hi} - T_{co})}{(T_{ho} - T_{ci})}} \quad (6)$$

Calculate the hot-side and cold-side film heat transfer coefficients. One of the most recent correlations to determine the film heat transfer coefficient in a spiral plate heat exchanger handling well slurries and water is the Morimoto and Hotta correlation [Ref. 3, Equation (38), p. 62]:

$$Nu = 0.0239 \left(1 + 5.54 \frac{D_H}{R_M}\right) Re^{0.806} Pr^{0.268} \quad (7)$$

Equations (7) and (17) will be used twice to determine the heat transfer coefficients defined in Equation (3) for the hot side and cold side. The equations that follow [Equations (8) through (18)] are written generically — that is, the fluid specific parameters will not be written as h_h or h_c but rather as just h.

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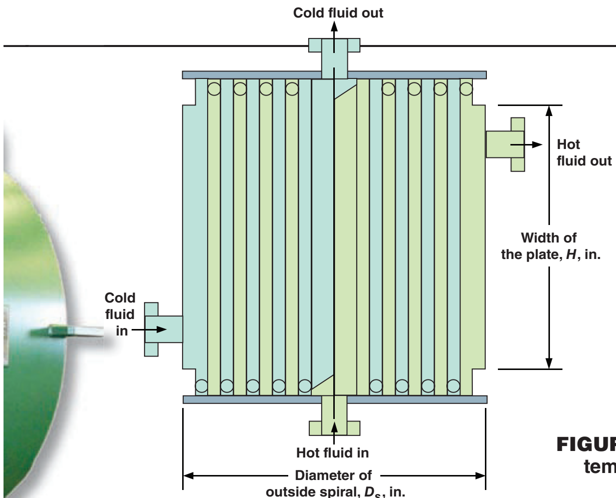

FIGURE 1. As shown in this cross-section of spiral plate heat exchanger, countercurrent flow achieved inside the unit enables efficient heat transfer

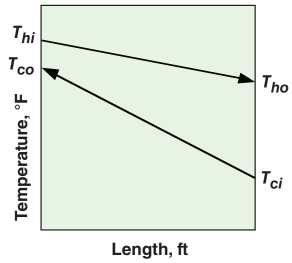

FIGURE 2. Shown here is the typical temperature-distribution profile for a counterflow heat exchanger

The Reynolds number, Re, is defined as follows [Ref. 4, p. 16]:

$$Re = \frac{GD_H}{\mu} \quad (8)$$

The mass flux, G, is defined by Equation (9) [Ref. 4, p. 16]:

$$G = \frac{m}{A_C} = \frac{m}{HS} \quad (9)$$

The average hydraulic diameter, D_H, is defined as:

$$D_H = \frac{2HS}{H+S} \cong 2S \quad (10)$$

Thus, the hydraulic diameter is approximately twice the spacing. This is justified because in most spiral-plate heat exchangers, the width of the passage, H, is considerably larger than its spacing [Ref. 4, p. 16]. If needed, the spacing can be increased to provide a pressure drop lower than the maximum allowable.

The apparent viscosity, μ, is determined as follows:

$$\mu = \frac{\tau}{\gamma}\ 1{,}000 \quad (11)$$

$$\tau = \tau_o + \frac{\eta\gamma}{1{,}000} \quad (12)$$

The strain rate, γ, is determined from the relationship provided in [Ref. 7, p. 6-13]. In the application of sizing spiral plate heat exchangers for cooling non-Newtonian slurries, the diameter, D, of the pipe must be replaced by the average hydraulic diameter of the channel D_H. Thus:

$$\gamma = \frac{8V}{D_H} \quad (13)$$

The velocity is calculated by dividing the mass flux by the density:

$$V = \frac{G}{\rho} \quad (14)$$

The spiral mean radius, R_M, is defined by Equation (15) [Ref. 3, p. 63]:

$$R_M = \frac{R_{max} + R_{min}}{2} \quad (15)$$

The Prandtl number, Pr, is defined as:

$$Pr = \frac{\mu c_P}{k} \quad (16)$$

As shown above, the Nusselt number, Nu, can be calculated from Equation (7). It can also be calculated from Equation (17) [Ref. 7, pp. 3–90, Table 3-8]:

$$Nu = \frac{hD_H}{k} \quad (17)$$

Solving for h gives:

$$h = \frac{k\,Nu}{D_H} \quad (18)$$

By substituting the value for Nu from Equation (7), the heat transfer coefficient can be calculated. After the heat transfer coefficients are obtained, U can be calculated from Equation (3). Using the value of U, the total heat transfer, Q, can be calculated from Equation (6). This value is defined as the "actual duty."

The percentage over-surface design, OS, which is typically added to provide a margin of safety, is determined from Equation (19):

$$OS = 100\left(\frac{Actual\ duty}{Operating\ duty} - 1\right) \quad (19)$$

The value of OS is increased by assuming a larger value for L in Equation (6).

Analyze pressure drop. The pressure drop in a spiral plate heat exchanger with studs is presented in [Ref. 7, Equation (11–81), p. 11-55]:

$$\Delta P = \frac{1.45(LV^2\rho)}{1.705x10^3} \quad (20)$$

To convert the pressure drop from kPa to psi, multiply the result in kPa by 1.45. The constant 1.45 is used for 60-x-60-mm studs (such dimensions are typical, as noted in [Ref. 7]).

Determine the outside spiral diameter. The diameter of the outside spiral is determined using the empirical Equation (21), as presented in [Ref. 6, p. 112]:

$$D_S = \left[\ 15.36L(S_h + S_c + 2t) + C^2\ \right]^{\frac{1}{2}} \quad (21)$$

Sample calculation

The assumptions described below are referenced in the next sections.

1.1 The physical properties of the hot slurry through the heat exchanger are assumed as:
- Specific gravity S_g = 1.35
- Yield stress τ_o = 30 Pa
- Consistency viscosity η = 30 *cP*
- Specific heat c_p = 0.9 Btu/lb$_m$°F
- Thermal conductivity k = 0.36 Btu/h ft°F

The assumed dimensions of the proposed spiral plate heat exchanger are:
- Plate width = 36 in.
- Plate thickness = 0.125 in.
- Core diameter = 12 in.
- Minimum radius of spiral = 6 in.
- Maximum radius of spiral = 10.75 in.
- Channel spacing, hot side = 1.25 in.
- Channel spacing, cold side = 0.25 in.

1.2 The operating heat duty is assumed to be 750,000 Btu/h.

1.3 The operating volume flowrate for the hot side is 1,500 gal/min.

1.4 The operating volume flowrate for the cold side is 300 gal/min.

1.5 The outlet temperature of the slurry in the vessel is 77°F.

1.6 The maximum allowable pressure drop is 25 psi.

1.7 The heat losses to the surroundings are negligible. This assumption is based on engineering judgment and consultation of [Ref. 1, pp. 17–27].

1.8 The inlet temperature of the cooling water is 50°F.

1.9 Cooling water specific gravity, S_g = 0.9992 ≅ 1.0.

1.10 The physical properties of water at 50°F and 60°F are those specified in [Ref. 5, Appendix 35.A, p. A-93]. The physical properties of water at 55°F were obtained by interpolation:

- Viscosity μ = 0.00082 lb_m/ft s = 1.219 cP
- Specific heat c_c = 1.0 Btu/lb_m °F
- Thermal conductivity k = 0.336 Btu/h ft °F = 9.33(10)$^{-5}$ Btu/s ft°F
- Density ρ = 62.35 lb_m/ft^3

1.11 For countercurrent flow through a spiral heat exchanger, the correction factor is equal to 1 [Ref. 2, p. 60].

1.12 The fouling factor, R_f, for spiral heat exchangers varies between 0.0003 and 0.001 h°Fft²/Btu [Ref. 2, Table 1, p. 62]. In this calculation, a mid-value between the two extremes is used. Thus, R_f = 0.0006 h°Fft²/Btu

1.13 To decrease the investment cost and still satisfy the process specifications, an over-surface design between 20 and 30% is specified.

1.14 Thermal conductivity of 316 stainless steel is 14.538 W/mK at 293.15K [Ref. 1, pp. 2–59]. The corresponding value in the English system is 8.4 Btu/h ft°F at 68°F

Determine the slurry inlet temperature.

Q = 750,000 Btu/h (an operational requirement, Assumption 1.2)

c_h = 0.9 Btu/lb_m°F (Assumption 1.1)

S_g = 1.35 (Assumption 1.1)

T_{ho} = 77°F (Assumption 1.5)

1 gal (water) = 8.34 lb_m (Standard conversion factor)

1 h = 60 min (Standard conversion factor)

Then for the operating volume flowrate of 1,500 gal/min (Assumption 1.3), the mass flowrate, m_h, is:

$$m_h = (1{,}500\,\frac{\text{gal}}{\text{min}})(8.34\,\frac{\text{lb}_\text{m}}{\text{gal}})\bullet(\frac{60\,\text{min}}{\text{h}})(1.35) = 1{,}013{,}310\ \text{lb}_\text{m}/\text{hr}$$

Using Equation 1:

$$Q = m_h c_h (T_{hi} - T_{ho}) \Rightarrow$$

$$T_{hi} = \frac{Q}{m_h c_h} + T_{ho}$$

$$= \frac{750{,}000\ \text{Btu/h}}{(1{,}013{,}310\,\text{lb}_\text{m}/\text{h})(0.9\ \text{Btu/lb}_\text{m}{}^\circ\text{F})} + 77\,^\circ\text{F} \cong 77.82\,^\circ\text{F}$$

T_{hi}=77.82°F

Determine the chilled water outlet temperature. From Equation (1):

$Q = m_h\, c_h(T_{hi} - T_{ho}) = m_c\, c_c\,(T_{co} - T_{ci})$

where:

m_h = 1,013,310 lb_m/h (from the section above: Determine the slurry inlet temperature)

$$m_c = (300\,\frac{\text{gal}}{\text{min}})(8.34\,\frac{\text{lb}_\text{m}}{\text{gal}})\bullet(\frac{60\,\text{min}}{\text{h}})(1.0) = 150{,}120\ \text{lb}_\text{m}/\text{h}$$

(Assumption 1.4)

c_c = 1.0 Btu/lb_m°F (Assumption 1.10)

T_{hi} = 77.82°F (from the section above: Determine the slurry inlet temperature)

T_{ho} = 77°F (Assumption 1.5)

T_{ci} = 50°F (Assumption 1.8)

Substituting in Equation (1), we have:

$m_h\, c_h\,(T_{hi} - T_{ho}) = m_c\, c_c\,(T_{co} - T_{ci})$

(1,013,310 lb_m/h)(0.9 Btu/lb_m°F) (77.82 °F – 77 °F) = (150,120 lb_m/h) (1.0 Btu/lb_m°F)(T_{co} – 50°F)(747,822.78 Btu/h) = 150,120 Btu/h°F (T_{co} – 50°F)

4.982°F = (T_{co} – 50°F)

T_{co} = 50°F + 4.982°F = 54.982°F ≅ 55°F

Calculate the LMTD. From Equation (4):

$$LMTD = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln \dfrac{(T_{hi} - T_{co})}{(T_{ho} - T_{ci})}}$$

$$= \frac{(77.82-55)-(77-50)}{\ln\dfrac{(77.82-55)}{(77-50)}} = \frac{22.82-27}{\ln\dfrac{22.82}{27}}$$

$$= \frac{-4.18}{\ln(0.8452)} = 24.85\,^\circ\text{F}$$

Calculate the film heat transfer coefficient for the hot side. The film heat transfer coefficient for the hot side is determined as follows:

From Equation (10):

$(D_H)_h$ ≅ $2S_h$ = 2(1.25 in.) = 2.5 in. ≅ 0.210 ft ([Ref. 4, p. 16] and Assumption 1.1)

From Equation (15):

$$R_M = \frac{R_{\max} + R_{\min}}{2} = \frac{10.75\,\text{in.} + 6\,\text{in.}}{2}$$

$$= 8.375\,\text{in.} = 0.698\,\text{ft}$$

([Ref. 3, p. 63] and Assumption 1.1)

To determine the Reynolds number, the mass flux is determined first from Equation (9):

$$G_h = \frac{m_h}{A_C} = \frac{m_h}{H S_h}$$

$A_c = HS_h$ = (36 in.)(1.25 in.) = 45 in.² = 0.3125 ft² (Assumption 1.1)

m_h = 1,013,310 lb_m/h = 281.48 lb_m/s (see the section above, Determine the slurry inlet temperature)

$$G_h = \frac{281.48\ \text{lb}_\text{m}/\text{s}}{0.3125\ \text{ft}^2} = 900.74\ \text{lb}_\text{m}/\text{s ft}^2$$

The velocity is obtained from Equation (14):

$$V = \frac{G}{\rho} = \frac{900.74\ \text{lb}_\text{m}/\text{s ft}^2}{(1.35)(62.4)\ \text{lb}_\text{m}/\text{ft}^3}$$

$$= 10.70\ \text{ft/s}$$

The viscosity is obtained from Equations (11), (12) and (13):

$$\mu = \frac{\tau}{\gamma}$$

$\tau = \tau_o + \eta\gamma$

τ_o = 30 Pa (Assumption 1.1)

η = 30 cP (Assumption 1.1)

$$\gamma = \frac{8V}{D_H} = \frac{8(10.70\ \text{ft/s})}{0.210\,\text{ft}}$$

$$= 407.619\ \text{s}^{-1} \cong 408\ \text{s}^{-1}$$

$$\tau = \tau_o + \eta\gamma$$

$$= 30\ \text{Pa} + (30\ \text{cP})(408\ \text{s}^{-1})(\frac{0.001\ \text{Pa s}}{\text{cP}})$$

$$= 30\ \text{Pa} + 12.24\ \text{Pa} = 42.24\ \text{Pa}$$

$$\mu = \frac{\tau}{\gamma} * 1{,}000 = \frac{(42.24\ \text{Pa})(1{,}000)}{408\ \text{s}^{-1}}$$

$$= 103.53\ \text{cP}$$

The apparent viscosity, μ, in lb_m/ft s is:

μ = 103.53 cP = 103.53(0.0006725) = 0.0696 lb_m/ft s

Then from Equation (8):

$$Re = \frac{GD_H}{\mu}$$

$$= \frac{(900.74\ \text{lb}_m/\text{s ft}^2)(0.210\ \text{ft})}{0.0696\ \text{lb}_m/\text{ft s}} \cong 2{,}718$$

The Prandtl number is determined by Equation (16):

$$Pr = \frac{\mu c_P}{k}$$

μ = 0.0696 lb_m/ft s
k = 0.36 Btu/hft°F
= 0.0001 Btu/s ft°F (Assumption 1.1)
c_p = 0.9 Btu/lb_m°F (Assumption 1.1)

$$Pr = \frac{\mu c_P}{k}$$

$$= \frac{(0.0696\ \text{lb}_m/\text{ft s})(0.9\ \text{Btu/lb}_m\ °\text{F})}{0.0001\ \text{Btu/s ft °F}}$$

$$\cong 626$$

The Nusselt number is determined using Equation (7):

$$Nu = 0.0239\left(1+5.54\frac{D_H}{R_M}\right)Re^{0.806}\,Pr^{0.268}$$

$$= 0.0239\left(1+5.54\frac{0.210}{0.698}\right)\bullet$$

$$(2{,}718)^{0.806}\,(626)^{0.268} \cong 210$$

Then from Equation (18), the film heat transfer coefficient for the hot fluid is:

$$h_h = \frac{k\,Nu}{D_H} = \frac{(0.36\ \text{Btu/h ft}^2\,°\text{F})(210)}{0.210\ \text{ft}}$$

$$= 360\ \text{Btu/h ft}^2\,°\text{F}$$

Calculate the film heat transfer coefficient for the cold side. The film heat transfer coefficient for the cold side is determined as follows:
From Equation (10):
$(D_H)_c \cong 2S_c$ = 2(0.25 in.) = 0.50 in. = 0.0417 ft ([Ref. 4, p. 16] and Assumption 1.1)

To determine the Reynolds number, the mass velocity is determined first from Equation (9):

$$G = \frac{m_c}{H\,S_c}$$

m_c = 150,120 lb_m/h = 41.7 lb_m/s (Assumption 1.4)
μ = 0.00082 lb_m/ft s (Assumption 1.10)
$A_c = HS_c$ = (36 in.) (0.25 in.) = 9 in.^{2} = 0.0625 ft^2 (Assumption 1.1)

$$G = \frac{41.7\ \text{lb}_m/\text{s}}{0.0625\ \text{ft}^2} = 667.2\ \text{lb}_m/\text{s ft}^2$$

The velocity is obtained from Equation (14):

$$V = \frac{G}{\rho} = \frac{667.2\ \text{lb}_m/\text{s ft}^2}{62.35\ \text{lb}_m/\text{ft}^3} = 10.70\ \text{ft/s}$$

From Equation (8):

$$Re = \frac{GD_H}{\mu}$$

$$= \frac{(667.2\ \text{lb}_m/\text{s ft}^2)(0.0417\ \text{ft})}{0.00082\ \text{lb}_m/\text{ft s}}$$

$$= 33{,}930$$

The Prandtl number is determined by Equation (16):

$$Pr = \frac{\mu c_P}{k}$$

μ = 0.00082 lb_m/ft s (Assumption 1.10)
k = 0.336 Btu/h ft °F = 9.33(10)$^{-5}$ Btu/s ft°F (Assumption 1.10)
c_p = 1.0 Btu/lb_m°F (Assumption 1.10)

$$Pr = \frac{\mu c_P}{k}$$

$$= \frac{(0.00082\ \text{lb}_m/\text{ft s})(1.0\ \text{Btu/lb}_m\ °\text{F})}{9.33(10)^{-5}\ \text{Btu/s ft °F}}$$

$$= 8.79$$

The Nusselt number is determined using Equation (7):

$$Nu = 0.0239\left(1+5.54\frac{D_H}{R_M}\right)Re^{0.806}\,Pr^{0.268}$$

$$= 0.0239\left(1+5.54\frac{0.0417}{0.698}\right)\bullet$$

$$(33{,}930)^{0.806}\,(8.79)^{0.268} \cong 255$$

Then from Equation (18), the film heat transfer coefficient for the cold fluid is:

$$h_c = \frac{k\,Nu}{D_H}$$

$$= \frac{(0.336\ \text{Btu/h ft °F})(255)}{0.0417\ \text{ft}}$$

$$\cong 2{,}055\ \text{Btu/h ft}^2\,°\text{F}$$

Calculate the overall heat transfer coefficient. The overall heat transfer coefficient is determined from Equation (3):

$$U = \frac{1}{\frac{1}{h_h}+\frac{t}{k}+\frac{1}{h_c}+R_f}$$

$$h_h = 360\ \text{Btu/h ft}^2\,°\text{F} \Rightarrow \frac{1}{h_h}$$

$$= 0.00278\ \frac{\text{h ft}^2\,°\text{F}}{\text{Btu}}$$

$$h_c = 2{,}055\ \text{Btu/h ft}^2\,°\text{F} \Rightarrow \frac{1}{h_c}$$

$$= 0.00049\ \frac{\text{h ft}^2\,°\text{F}}{\text{Btu}}$$

t = 0.125 in. = 0.01042 ft (Assumption 1.1)
k = 8.4 Btu/h ft°F [Ref. 1, pp. 2–59]

$$\frac{t}{k} = \frac{0.01042\ \text{ft}}{8.4\ \text{Btu/h ft °F}} = 0.0012\ \frac{\text{h ft}^2\,°\text{F}}{\text{Btu}}$$

$$R_f = 0.0006\ \frac{\text{h ft}^2\,°\text{F}}{\text{Btu}}$$

(Assumption 1.12)

$$U = \frac{1}{\frac{1}{h_h}+\frac{t}{k_P}+\frac{1}{h_c}+R_f}$$

$$= \frac{1}{\left[(0.00278+0.0012+0.00049+0.0006)\frac{\text{h ft}^2\,°\text{F}}{\text{Btu}}\right]}$$

$$= \frac{1}{0.00507\ \frac{\text{h ft}^2\,°\text{F}}{\text{Btu}}} \cong 197\ \frac{\text{Btu}}{\text{h ft}^2\,°\text{F}}$$

Calculate the length of the heat transfer plate. The total heat transfer is determined from the energy absorbed by the chilled water. Substituting in Equation (1), we have:
$Q = m_c\,c_c\,(T_{co} - T_{ci})$ = 150,120 lb_m/h (1.0 Btu/lb_m°F)(55–50)°F
=750,600 Btu/h

Using Equation (6):

$$L = \frac{Q}{2\,H\,U\,(LMTD)}$$

$$Q = 750{,}600\ \frac{\text{Btu}}{\text{h}}$$

H = 36 in. = 3 ft (Assumption 1.1)

$$U = 197\ \frac{\text{Btu}}{\text{h ft}^2\,°\text{F}}$$

(see the section above, Calculate the overall heat transfer coefficient)
$LMTD$ = 24.85°F (see the section

NOMENCLATURE

A	=	Heat-transfer surface area, ft^2
A_c	=	HS = Flow area of channel, ft^2
C	=	Core diameter, in.
c_c	=	Specific heat of cold fluid, Btu/lbm°F
c_h	=	Specific heat of hot fluid, Btu/lbm°F
c_p	=	Specific heat of fluid, Btu/lbm°F
D_H	=	Average hydraulic diameter of channel, ft
D_s	=	Outside spiral diameter, in.
G	=	Mass flux, lbm/s ft^2
H	=	Width of the plate, ft
h	=	Film heat-transfer coefficient, Btu/h ft^2 °F
h_h	=	Hot-side convection heat-transfer coefficient, Btu/h ft^2 °F
h_c	=	Cold-side convection heat-transfer coefficient, Btu/h ft^2°F
k	=	Thermal conductivity of fluid, Btu/h ft °F
k_P	=	Thermal conductivity of the plate providing the heat-transfer surface, Btu/h°F ft
L	=	Length of the plate, ft (m)
$LMTD$	=	Log mean temp. difference, °F
m	=	Mass flowrate, lbm/s
m_h	=	Mass flowrate of hot fluid, lbm/h
m_c	=	Mass flowrate of cold fluid, lbm/h
Nu	=	Nusselt number, unitless
Pr	=	Prandtl number, unitless
ΔP	=	Pressure drop, kPa (psi)
Q	=	Total heat transfer, Btu/h
R_{max}	=	Maximum radius of spiral, ft
R_{min}	=	Minimum radius of spiral, ft
Re	=	Reynolds number, unitless
R_f	=	Fouling factor, h°F ft^2/Btu
R_M	=	Mean spiral radius, ft
S	=	Channel spacing, ft
S_h	=	Spacing of the hot side, in.
S_c	=	Spacing of the cold side, in.
t	=	Thickness of the plate providing the heat transfer-surface, ft (in.)
T_{ho}	=	Outlet temperature of hot fluid, °F
T_{hi}	=	Inlet temperature of hot fluid, °F
T_{co}	=	Outlet temperature of cold fluid, °F
T_{ci}	=	Inlet temperature of cold fluid, °F
ρ	=	Density, kg/m^3 (lbm/ft^3)
τ	=	Bingham plastic shear stress, Pa
τ_o	=	Bingham plastic yield stress, Pa
η	=	Bingham plastic consistency viscosity, cP
γ	=	Strain rate, s^{-1}
μ	=	Apparent viscosity of fluid, lbm/ft s

TABLE 1. RESULTS OF THE SAMPLE CALCULATION

Fluid	Hot-side slurry	Cold-side water
Volume flowrate, gal/min	1,500	300
Specific heat, Btu/lbm °F	0.9	1.0
Density, lbm/ft³	84.24	62.4
Specific gravity	1.35	1.0
Viscosity, cP	103.53	1.22
Thermal conductivity, Btu/ft h°F	0.36	0.336
Inlet temperature, °F	77.82	50
Outlet temperature, °F	77	55
Pressure drop, psi	16.95	12.83
Heat transfer Area, ft²	192	—
Heat exchanged, Btu/h	750,000	—
LMTD, °F	24.85	—
Heat transfer coefficient, Btu/h ft² °F	197	—
Percentage of over surface, %	25.32	—
Plate length, ft	32	—
Plate width, in.	36	—
Outside spiral diameter, in.	32	—

above, Calculate the LMTD)

$$L = \frac{Q}{2\,HU(LMTD)}$$

$$= \frac{750{,}600\ \text{Btu/h}}{2(3\text{ft})(197\ \text{Btu/h ft}^2\,°\text{F})(24.85\,°\text{F})}$$

$$= 25.6\,\text{ft}$$

$A = 2HL = 2(3\text{ ft})(25.6\text{ ft}) = 153.6\text{ ft}^2$

From Equation (2) the actual heat duty is:

$Q = U\,A\,(LMTD)$

$$= (197\frac{\text{Btu}}{\text{h ft}^2\,°\text{F}})(153.6\,\text{ft}^2)(24.85\,°\text{F})$$

$$\cong 751{,}941\ \text{Btu/h}$$

It is shown that the actual duty is greater than the required operating duty by a very small amount. The percentage of over-surface design is:

$$OS = 100\left(\frac{\text{Actual duty}}{\text{Operating duty}} - 1\right)$$

$$= 100\left(\frac{751{,}941}{750{,}000} - 1\right) = 0.26\ \%$$

An over-surface design between 20 and 30% is specified (Assumption 1.13). Therefore, the length of the plate must be increased. Assuming a new value of 32 ft and repeating the calculation, the new heat transfer area is:

$A = 2H\,L = 2(3\text{ ft})(32\text{ ft}) = 192\text{ ft}^2$

And the new actual duty is:

$Q = U\,A\,(LMTD)$

$$= (197\frac{\text{Btu}}{\text{h ft}^2\,°\text{F}})(192\,\text{ft}^2)(24.85\,°\text{F})$$

$$\cong 939{,}926\ \text{Btu/h}$$

$$OS = 100\left(\frac{\text{Actual duty}}{\text{Operating duty}} - 1\right)$$

$$= 100\left(\frac{939{,}926}{750{,}000} - 1\right) = 25.32\ \%$$

Thus, the over-surface design is between 20 and 30% as indicated in Assumption 1.13.

Calculate pressure drop for hot stream. Calculate the pressure drop for the hot stream using Equation (20):

$$\Delta P = \frac{1.45(LV^2\rho)}{1.705 \bullet 10^3}$$

Here:

L = 32 ft ≅ 9.80 m (see the above section, Calculate the length of the heat transfer plate)

$S_g = 1.35 \Rightarrow \rho = 84.24\ \text{lb}_m/\text{ft}^3 = 1{,}320.05\ \text{kg/m}^3$ (Assumption 1.1)

V = 10.70 ft/s ≅ 3.26 m/s (see the above section, Calculate the hot side film heat transfer coefficient)

$$\Delta P = \frac{1.45\left[\begin{array}{l}(9.80\,m)(3.26\ \text{m/s})^2 \bullet \\ (1{,}320.05\ \text{Kg/m}^3)\end{array}\right]}{1.705 \bullet 10^3}$$

$$= 116.92\ \text{kpa} = 16.95\ \frac{\text{lb}_f}{\text{in.}^2}$$

Calculate pressure drop for cold

stream. The pressure drop for the cold stream is calculated using Equation (20):

$$\Delta P = \frac{1.45(LV^2\rho)}{1.705 \bullet 10^3}$$

Here:
L = 32 ft ≅ 9.80 m (see the section above, Calculate the length of the heat transfer plate)
ρ = 62.35 lb_m/ft^3 = 998.85 kg/m^3 (Assumption 1.10)
V = 10.70 ft/s = 3.26 m/s (see the section above, Calculate the film heat transfer coefficient of the cold side)

$$\Delta P = \frac{1.45\begin{bmatrix}(9.80\ \text{m})(3.26\ \text{m/s})^2 \bullet \\ (998.85\ \text{kg/m}^3)\end{bmatrix}}{1.705 \bullet 10^3}$$

$$= 88.47\ \text{kpa} = 12.83\ \frac{\text{lb}_f}{\text{in.}^2}$$

Determine the outside spiral diameter. The diameter of the outside spiral is determined via Equation (21):
$D_s = [15.36L\,(S_h + S_c + 2t) + C^2]^{1/2}$
Here:
L = 32 ft (see the section above, Calculate the length of the heat transfer plate)
S_h = 1.25 in. (Assumption 1.1)
S_c = 0.25 in. (Assumption 1.1)
t = 0.125 in. (Assumption 1.1)
C = 12 in. (Assumption 1.1)
Then:
$D_s = [15.36L\,(S_h + S_c + 2t) + C^2]^{1/2}$
$= [15.36(32)(1.25 + 0.25 + 2(0.125)) + (12)^2]^{1/2}$
$D_s = [491.52(1.75) + 144]^{1/2} = [1,065.6]^{1/2} = 31.69 \Rightarrow$ Use 32 in.

The results are shown in Table 1. ■

Edited by Suzanne Shelley

References

1. W.M. Rohsenow, J.P. Harnett, Y.I. Cho, "Handbook of Heat Transfer," 3rd ed., McGraw-Hill, New York, 1988.
2. Kevin M. Bailey, Understand Spiral Heat Exchangers, *Chem. Eng. Prog*, May 1994.
3. E. Morimoto, K. Hotta, Study of the Geometric Structure and Heat Transfer Characteristics of a Spiral Plate Heat Exchanger, *Heat Transfer, Japanese Research*, 17,53-71, 1988.
4. Wen-Jei Yang, D. Rundle, Optimized Thermal Design of Plate and Spiral-Type Heat Exchangers, in HTD-Vol. 282, "Challenges of High Temperature Heat Transfer Equipment," ASME 1994.
5. M.R. Lindeburg, "Mechanical Engineering Reference Manual for PE Exam," 10th ed.
6. P.E. Minton, Designing Spiral Plate Heat Exchangers, *Chem. Eng.*, May 4, 1970.
7. "Perry's Chemical Engineers' Handbook," 7th Ed., McGraw-Hill, New York, 1997.