Double Pipe Heat Exchangers

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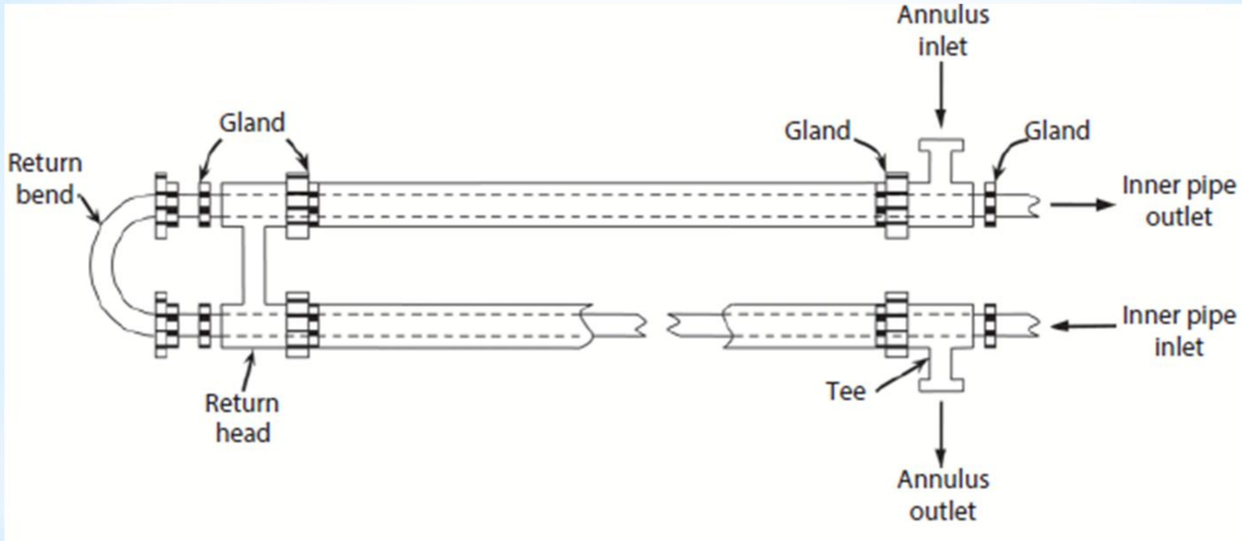
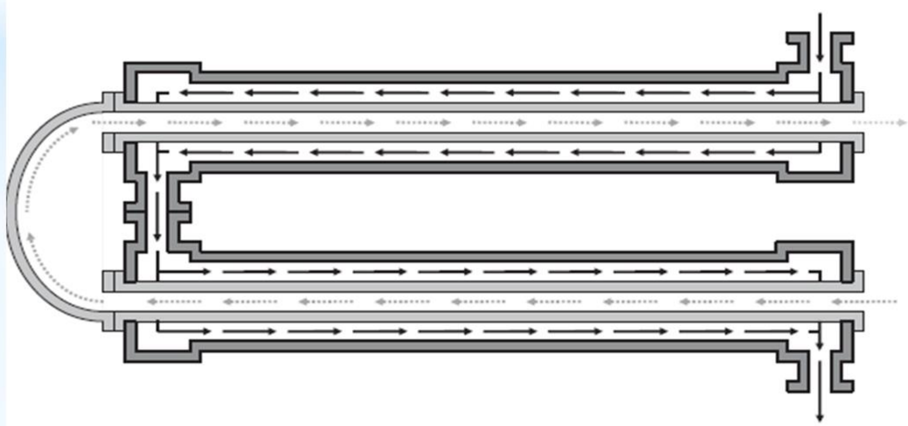


Double pipe exchanger.



Fluid flow inside a double pipe exchanger.

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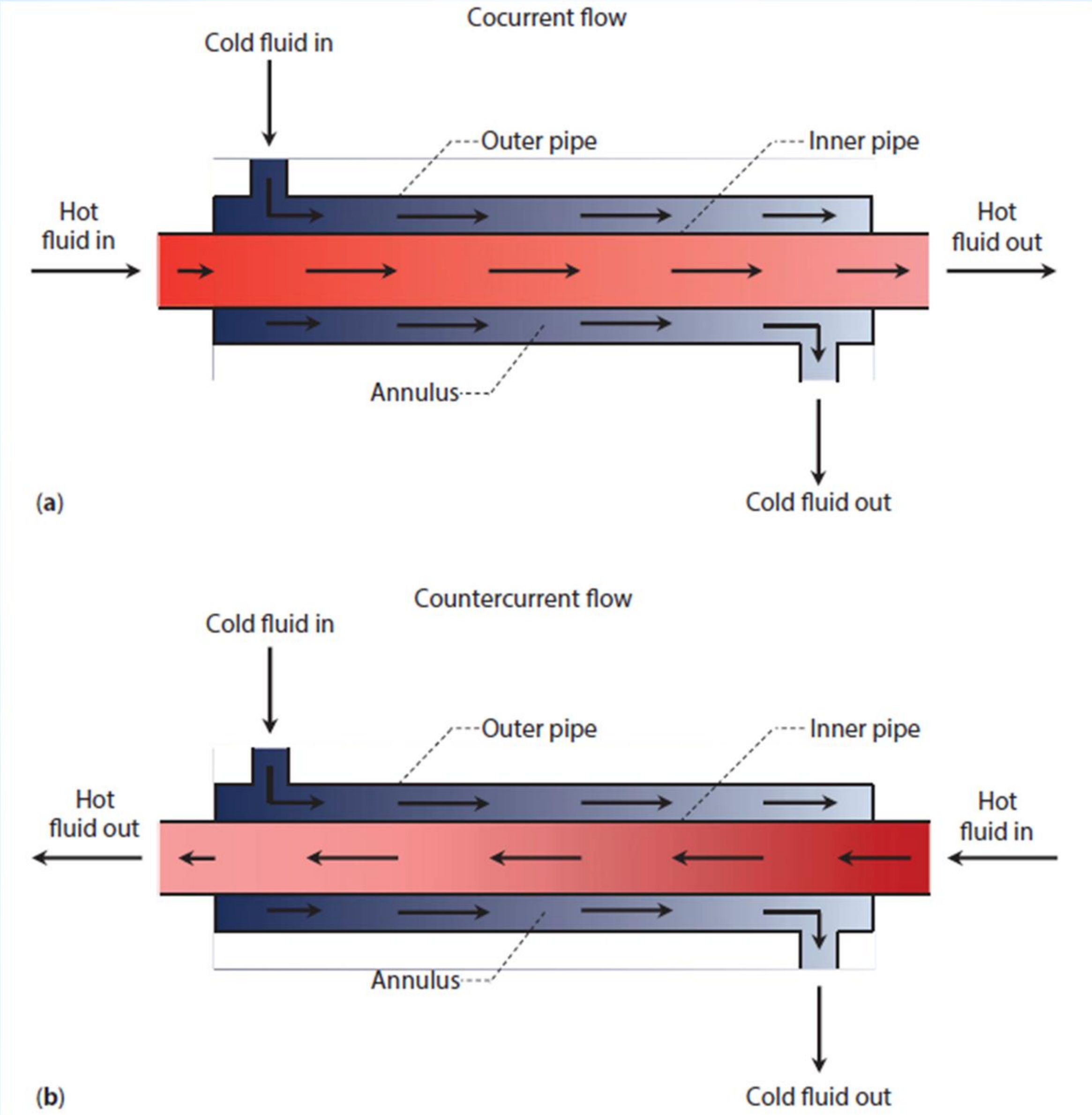

Double pipe schematic.

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Types of tubes or pipes available include:

1. Plain tubes
2. Duplex tubes
3. Finned tubes

Double pipe exchanger fittings.

Outer pipe, IPS	Inner pipe, IPS
2	1 ¼
2 ½	1 ¼
3	2
4	3

This class of exchanger is usually assembled in 12-, 15-, or 20-ft effective lengths.

Where effective length is defined as the distance in each leg over which heat transfer occurs and excludes inner pipe protruding beyond the exchanger section.

In other words, a *20 ft hairpin* is *40 ft long*.

As noted earlier, hairpins should not be designed for pipes in excess of 20 ft in effective length.

Whenever the linear pipe distance of each hairpin exceeds 40 ft, the inner pipe tends to sag and touch the outer pipe in the middle of the leg, thereby causing a poor flow distribution in the annulus.

The principal disadvantage to the use of double pipe exchangers lies in the small amount of heat-transfer surface contained with a single hairpin.

In an industrial process, a very large number of hairpins are required. These require considerable space, and each double pipe exchanger introduces no fewer than 14 points at which leakage might occur.

calculations for flow in the annular region require the use of the aforementioned characteristic or *equivalent diameter.*

By definition, this diameter, *De* , is given by 4 times the area available for flow divided by the "wetted" perimeter.

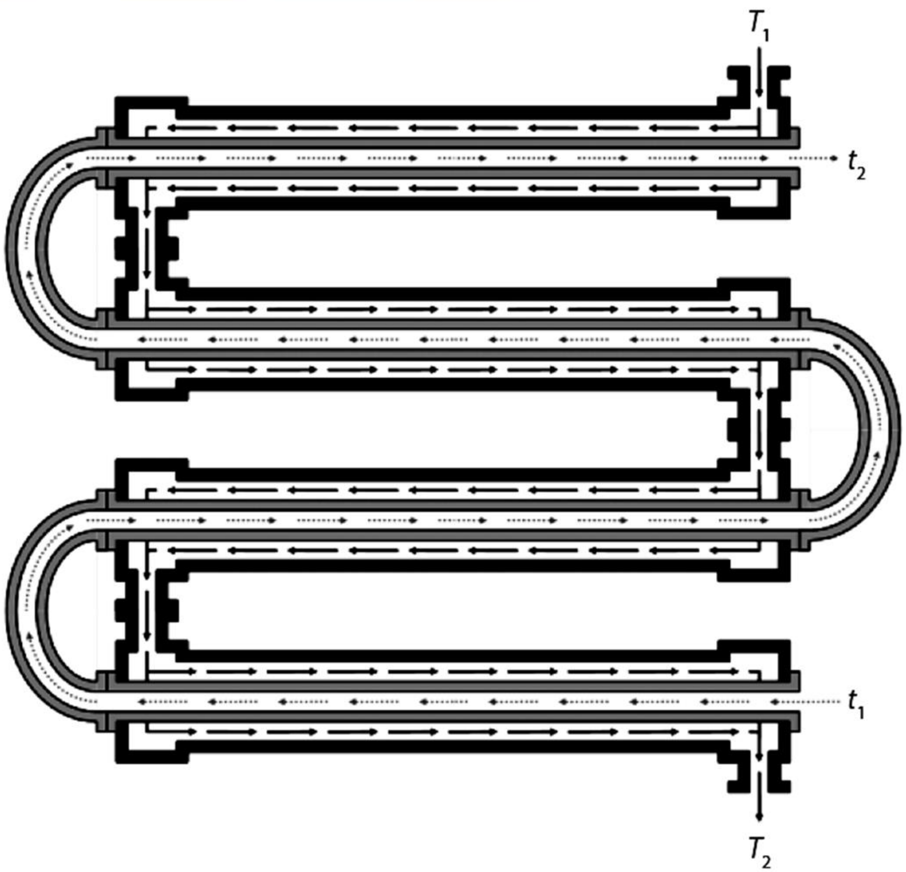
$$D_e = 4\left(\frac{\pi}{4}\right)\left[\frac{D_{o,i}^2 - D_{i,o}^2}{\pi\left(D_{o,i} - D_{i,o}\right)}\right]$$

where $D_{o,i}$ is the inside diameter of the outer pipe and $D_{i,o}$ is the outside diameter of the inner pipe. This equation reduces to,

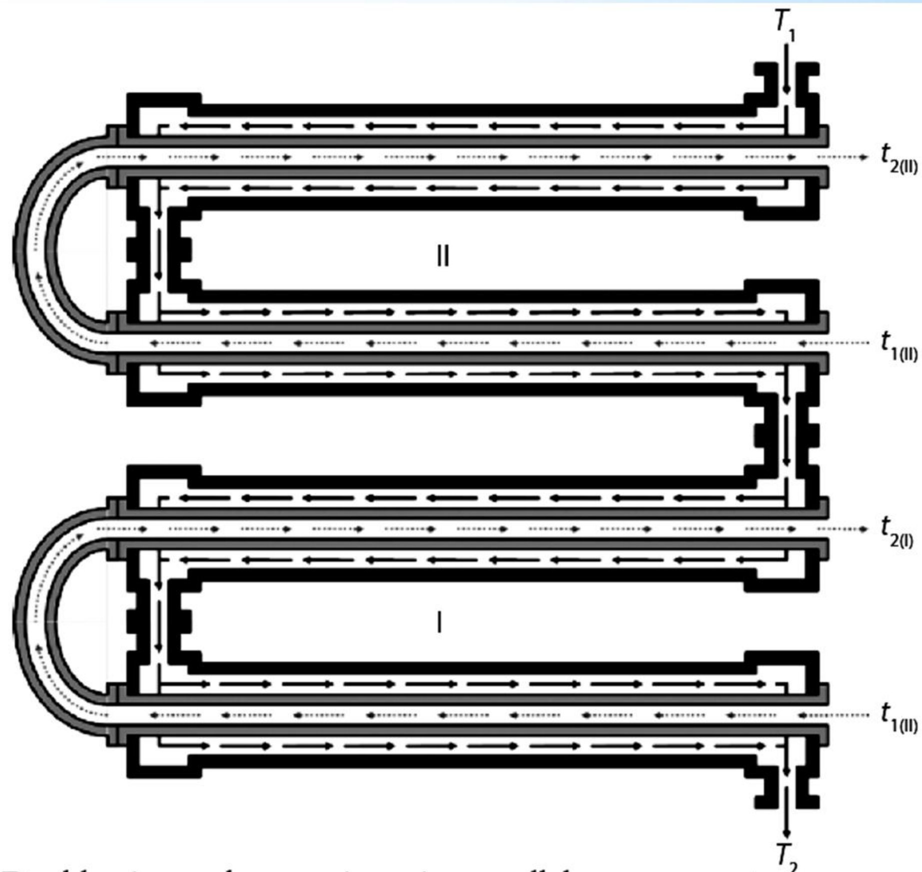
$D_e = D_{o,i} - D_{i,o}$

which is four times the hydraulic radius, r_H, i.e.,

$D_e = 4r_H$



Double pipe exchangers in series.



Double pipe exchangers in series-parallel arrangement.

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Key Equations for Double pipe Heat Exchanger design

For heating or cooling of a process fluid, the heat transfer equation is employed to find a suitable heat exchanger, using one or more rearrangements of the following steps:

(i) Solve for unknown variables in the heat balance.

$$Q = \dot{m}c\Delta t = \dot{M}C\Delta T$$

(ii) Calculate ΔT_{LM} from known and calculated temperatures.

$$\Delta T_{\mathrm{LM}} = \frac{\Delta T_a - \Delta T_b}{\ln\left(\Delta T_a / \Delta T_b\right)}$$

(iii) Calculate the film coefficients from fluid properties and U_C (as U_o or U_i).

$$\frac{1}{U_o A_o} = \frac{1}{h_i A_i} + \frac{\Delta x_w}{k A_{\mathrm{LM}}} + \frac{1}{h_o A_o}$$

(iv) When applicable, neglect pipe wall resistance; assume $A_i = A_0$.

$$\frac{1}{U_C} = R_i + R_o = \frac{1}{h_i} + \frac{1}{h_o}$$

(v) Assume an allowable dirt factor in order to calculate U_D.

$$\frac{1}{U_D} = \frac{1}{U_C} + R_d$$

(vi) Use the heat transfer equation to solve for unknown variables such as heat transfer surface area, *A*.

$$Q = U_D A \Delta T_{LM}$$

Film Coefficient Calculations.

Individual coefficients, h_i and h_0, can be calculated by using empirical equations.

All of the physical properties in the equations that follow are evaluated at bulk temperatures. Except for the viscosity term at the wall temperature.

For the hot stream in the inner tube, the bulk temperature is

$$T_{H,\,bulk} = (T_1 + T_2)/2.$$

For the cold stream in the annulus, the bulk temperature is

$$t_{C,\,bulk} = (t_1 + t_2)/2.$$

Employing viscosity values at the wrong temperature can occasionally lead to substantial errors.

The Reynolds number for both the cold and hot process streams must be calculated in order to determine whether the flow rate for each stream is in the laminar, turbulent, or transition region.

Flow regimes for various Reynolds numbers

Reynolds number, Re	Flow region
$Re < 2100$	Laminar
$2100 < Re < 10{,}000$	Transitional
$Re > 10{,}000$	Turbulent

Inner pipe:

$$Re_i = \frac{4\dot{m}}{\pi D_{i,i} \mu_i}$$

Annulus between pipes:

$$Re_o = \frac{4\dot{m}_o}{\pi \left(D_{o,i} - D_{i,o} \right) \mu_o} = \frac{D_e \dot{m}_o}{\mu_o S}$$

S = cross-sectional annular area, ft^2

μ = viscosity of hot or cold fluid at bulk temperature, lb/ft · hr

Similarly, the Nusselt numbers, Nu_i and Nu_o, are defined by the following equations:

Inner pipe:

$$Nu_i = \frac{h_i D_{i,i}}{k_i}$$

Annulus between pipes:

$$Nu_o = \frac{h_o \left(D_{o,i} - D_{i,o}\right)}{k_o}$$

For laminar flow:

Nusselt number = 4.36 (for uniform surface heat flux Q/A)
= 3.66 (for constant surface temperature)

This value should only be used for Graetz numbers, Gz, < 10.

For laminar flow with Gz from 10 to 1000, the following equation applies

$$Nu = 2.0 Gz^{1/3} \left(\frac{\mu}{\mu_{wall}}\right);$$

$$Gz = \frac{\dot{m} c_p}{kL}$$

where,
μ_{wall} = viscosity at wall temperature, lb/ft · hr
L = total length of tubular exchanger, ft

For turbulent flow (Re > 10,000):

The Nusselt number may be calculated from

Dittus-Boelter equation (if $0.7 \leq Pr \leq 160$)

$$Nu = 0.023\, Re^{0.8} Pr^{n}$$

$$St = 0.023\, Re^{0.8} Pr^{-2/3}$$

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

where
$St = Nu/(Re \bullet Pr) = h/\rho \tilde{v} c_p$.

n = 0.4 for heating or 0.3 for cooling

Sieder -Tate equation (if $0.7 \leq Pr \leq 16,700$)

$$Nu = 0.023\, Re^{0.8} Pr^{1/3} \left(\frac{\mu}{\mu_{wall}} \right)^{0.14}$$

This correction factor is included to account distortion in the velocity profile that arises due to variation in viscosity with temperature.

Both equations are valid for L/D greater than 10.

The Dittus-Boelter equation should only be used for small to moderate temperature differences. The Sieder-Tate equation applies for larger temperature differences.

Errors as large as 25% are associated with both equations.

The wall temperature can be estimated by the average of the four known temperatures:

$$\Delta T_i = \frac{t_{C,i} + t_{C,o} + T_{H,i} + T_{H,o}}{4} = \frac{t_1 + t_2 + T_1 + T_2}{4}$$

After the individual heat transfer coefficients are calculated, a new wall temperature can be calculated with the following equation.

$$\Delta T_i = \frac{1/h_i}{(1/h_i) + (D_{i,i} / D_{i,o})h_o} \left(T_{H,bulk} - t_{C,bulk}\right)$$

With $$T_{wall} = T_{H,bulk} - \Delta T_i$$

Based on experimental data, equation obtained for heating of several oils in a pipe:

$$\frac{h_i D}{k} = 0.0115 \left(\frac{DG}{\mu}\right)^{0.90} \left(\frac{c_p \mu}{k}\right)^{1/2}$$

Sieder and Tate made a correlation for both heating and cooling of number of fluids, principally petroleum fractions in horizontal and vertical tubes (streamline flow $DG/\mu < 2100$).

$$\frac{h_i D}{k} = 1.86\left[\left(\frac{DG}{\mu}\right)\left(\frac{c\mu}{k}\right)\left(\frac{D}{L}\right)\right]^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

$$= 1.86\left[\frac{4}{\pi}\left(\frac{\dot{m}c_p}{kL}\right)\right]^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

L is the total length of the heat-transfer path before mixing occurs.

Equation gave maximum mean deviations of approximately ±12% from Re 100 to Re 2100, except for water.

Beyond the transition range, the data may be extended to turbulent flow, resulting Equation

$$\frac{h_i D}{k} = 0.027\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c_p\mu}{k}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

Equation gave maximum mean deviations of approximately ±15 and ±10 percent for the Reynolds numbers above 10,000.

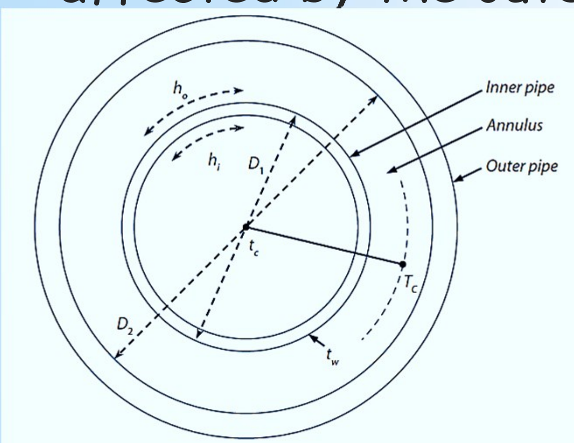
These equ. for tubes but they have been used indiscriminately for pipes.

Equivalent Diameter and Film Coefficients for Fluids in Annuli

For heat transfer, the wetted perimeter is the outer circumference of the inner pipe with diameter D_1. Therefore, for heat transfer in annuli,

$$D_e = 4r_h = \frac{4\times \text{flow area}}{\text{wetted perimeter}} = \frac{4\pi\left(D_2^2 - D_1^2\right)}{4\pi D_1} = \frac{D_2^2 - D_1^2}{D_1}$$

However, in pressure-drop calculations the friction not only results from the resistance of the outer pipe but is also affected by the outer surface of the inner pipe.



$$D_e' = 4r_h = \frac{4\times \text{flow area}}{\text{frictional wetted perimeter}} = \frac{4\pi\left(D_2^2 - D_1^2\right)}{4\pi\left(D_2 + D_1\right)} = D_2 - D_1$$

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Once the equivalent diameter has been obtained, the outside or annulus coefficient, h_o, is found in the same manner as h_i described earlier.

In designing a double pipe heat exchanger, mass balances, heat balances, and the applicable heat transfer equation(s) are used.

After heat duties are known (or have been calculated), values for the overall heat transfer coefficient, U, can be calculated as follows:

$$U_o = \frac{Q_C}{A_o \Delta T_{LM}}$$

$$U_i = \frac{Q_H}{A_i \Delta T_{LM}}$$

U_o = overall heat transfer coefficient based on the outside area of the inner pipe, $Btu/ft^2 \cdot hr \cdot F$

U_i = overall heat transfer coefficient based on the inside area of the inner pipe, $Btu/ft^2 \cdot hr \cdot F$

In addition, one may write,

$$\frac{1}{UA}=\frac{1}{U_iA_i}=\frac{1}{U_oA_o}$$

The overall heat transfer coefficient, U, for flow in a tube is related to the individual coefficients by the following equation:

$$\frac{1}{UA}=\frac{1}{h_iA_i}+\frac{R_{d,i}}{A_i}+\frac{\ln\left(D_{i,i}/D_{o,i}\right)}{2\pi kL}+\frac{R_{d,o}}{A_o}+\frac{1}{h_oA_o}$$

If the resistance of the metal wall is small compared to the other resistances then

If the outside area of the inner pipe is used, A_o, then the relationship is simplified as:

$$\frac{1}{U_o}=\frac{D_o}{h_iD_i}+\frac{1}{h_o}$$

When the inside area, A_i, is used, then the relationship is simplified as:

$$\frac{1}{U_i}=\frac{1}{h_i}+\frac{D_i}{h_oD_o}$$

Pressure Drops in Pipes and Annuli

The pressure drop in pipes can be computed from the Fanning Equation using an appropriate value of f from Table in next slide, depending upon the type of flow.

For the pressure drop in fluids flowing in annuli, replace D_e in the Reynolds number by D_e' to obtain f. The Fanning equation may then be modified to give,

$$\Delta P_{\text{Fanning}} = \Delta F = \frac{4fL\overline{v}^2}{2gD_e'} = \frac{4fG^2L}{2g\rho^2D_e'}$$

Standard allowable pressure drops are as follows:

1. 5 to 10 psi is a customary allowable pressure drop for an exchanger or battery of exchangers fulfilling a single process service (except where the flow is by gravity).

2. 10 psi is fairly standard for each pumped system.

3. For gravity flow, the allowable pressure drop is determined by the elevation of the storage vessel above the final outlet, z_h,

in feet of fluid. The feet of fluid may be converted to pounds per square inch by multiplying z_h by 144 , where is the density of the fluid.

	Correlation	Conditions
	Darcy friction factor: [2]	
(1)	$f = \frac{64}{\mathrm{Re}}$	Laminar, fully developed
(2)	$\mathrm{Nu} = 4.364$	Laminar, fully developed, constant Q', UHF, $\mathrm{Pr} \geq 0.6$
(3)	$\mathrm{Nu}_D = 3.658$	Laminar, fully developed, constant T_S, UWT, $\mathrm{Pr} \geq 0.6$
	Seider and Tate equation:	
(4)	$\overline{\mathrm{Nu}} = 1.86\left(\mathrm{Re}_D \mathrm{Pr}\frac{D}{L}\right)^{1/3}\left(\frac{\mu}{\mu_s}\right)^{0.14}$	Laminar, combined entry length, properties at mean bulk temperature of the fluid, $\left[\left(\mathrm{Re}_D \mathrm{Pr}\frac{D}{L}\right)^{1/3}\left(\frac{\mu}{\mu_s}\right)^{0.14}\right] \geq 2$, Constant wall temperature, T_S, $0.48 < \mathrm{Pr} < 16{,}700$, $0.0044 < (\mu/\mu_s) < 9.75$
	$= 1.86\mathrm{Gz}^{1/3}\left(\frac{\mu}{\mu_s}\right)^{0.14}$	
(5)	$f = 0.316\mathrm{Re}_D^{-0.25}$	Turbulent, fully developed, smooth tubes, $\mathrm{Re} \leq 2\times10^4$
(6)	$f = 0.184\mathrm{Re}_D^{-0.2}$	Turbulent, fully developed, smooth tubes, $\mathrm{Re} \geq 2\times10^4$
	Dittus-Boelter equation:	
(7)	$\mathrm{Nu} = 0.023\mathrm{Re}_D^{0.8}\mathrm{Pr}^n$	Turbulent, fully developed, $0.6 \leq \mathrm{Pr} \leq 160$, $\mathrm{Re} \geq 10{,}000$, $L/D \geq 10$, $n = 0.4$ for $T_S > T_m$ and $n = 0.3$ for $T_S < T_m$

Kern's Original Design Methodology

- Hot- and cold-fluid temperatures are represented by upper and lower case letters, respectively.
- All fluid properties are indicated by lower case letters
- The diameter of the pipes must be given or assumed.

With references to Kern's design methodology from the first edition, the following calculation procedure involves an application where a hot fluid is to be cooled by contact with a cooler fluid.

Process conditions (variables) that needs to be given, obtained, or calculated:

Hot fluid: T_1, T_2, M, C, S, ρ_H, μ_H, k_H, ΔP, $R_{d,o}$, or $R_{d,i}$

Cold fluid: t_1, t_2, m, c, s, ρ_C, μ_C, k_C, ΔP, $R_{d,o}$, or $R_{d,I}$

Process conditions that are usually specified or estimated:

Hot fluid: Pmax, Rd,o or Rd,i (from past design and/or experience)

Cold fluid: Pmax, Rd,o or Rd,i (from past design and/or experience)

A convenient order of calculation provided by Kern [2] follows:

(1) From T_1, T_2, t_1 and t_2, check the heat balance, Q, using *C* at T_{avg} and c at t_{avg}

$$T_{avg} = \frac{T_1 + T_2}{2} \quad and \quad t_{avg} = \frac{t_1 + t_2}{2}$$

$$Q = \dot{M}C\left(T_2 - T_1\right) = \dot{m}c\left(t_2 - t_1\right)$$

(2) LMTD, assuming countercurrent flow (countercurrent flow arrangements tend to produce a larger temperature driving force, *TLM)*.

(3) Viscosity considerations:

If the fluid is nonviscous and Newtonian, then can be assumed to equal 1, where $\Phi = (\mu/\mu_w)^{0\ 14}$.

However, if the fluid is viscous like that of petroleum or heavy hydrocarbons, the viscosity may behave differently depending on where the fluid is because it is often sensitive to temperature, i.e.,

The viscosity of the fluid at the inner pipe wall can differ from that of the bulk flow area of the fluid.

Inner pipe (p):

(4) Flow area, $a_p = \dfrac{D^2\pi}{4}; \quad \mathrm{ft}^2$

(5) Mass velocity, $G_p = \dfrac{\dot{m}_p}{a_p}; \quad \mathrm{lb}/\left(\mathrm{hr}\cdot\mathrm{ft}^2\right)$

(6) Obtain at T_{avg} or t_{avg} depending upon which flows through the inner pipe.

$$\mu[=]\text{lb}/(\text{ft}\cdot\text{hr})$$

(7) From *D ft, Gp lb/(hr · ft²), and μ lb/(ft · hr), obtain the Reynolds* number, *Re_p* .

$$Re_p = \frac{DG_p}{\mu}$$

(8) From c_p Btu/lb °F, μ lb/ft hr , and k Btu/(hr ft² °F/ft) , all obtained at T_{avg} or t_{avg} , compute h_i using the appropriate equation based on the range of Re_p.

For Laminar Flow: $$\frac{h_i D}{k} = 1.86\left[\left(\frac{DG_p}{\mu}\right)\left(\frac{c_p\mu}{k}\right)\left(\frac{D}{L}\right)\right]^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

For Turbulent Flow: $$\frac{h_i D}{k} = 0.027\left(\frac{DG_p}{\mu}\right)^{0.8}\left(\frac{c_p\mu}{k}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

Annulus (a):

(4') Flow area, $$a_a = \frac{\pi}{4}\left(D_2^2 - D_1^2\right)[=]\text{ft}^2$$

Equivalent diameter, $D_e = 4 \times \frac{Flow\ Area}{Wetted\ Perimeter} = \frac{D_2^2 - D_1^2}{D_1} [=] \text{ft}$

(5') Mass velocity, $G_a = \frac{\dot{m}_a}{a_a} [=] \text{lb}/\left(\text{hr} \cdot \text{ft}^2\right)$

(6') Obtain μ of the annulus's fluid at T_{avg} or t_{avg} depending upon which flows through the inner pipe.

$$\mu [=] \text{lb}/\left(\text{ft} \cdot \text{hr}\right)$$

(7') From D_e ft, G_a lb/(hr · ft2), and μ lb/(ft · hr), obtain the Reynolds number, Re_a.

$$Re_a = \frac{D_e G_a}{\mu}$$

(8) From c_p Btu/lb °F, μ lb/ft hr, and k Btu/hr ft^2 °F/ft , all obtained at T_{avg} or t_{avg} , compute h_o using the appropriate equation based on the range of Re_p.

For Laminar Flow: $\frac{h_o D_e}{k} = 1.86 \left[\left(\frac{D_e G_a}{\mu} \right) \left(\frac{c_p \mu}{k} \right) \left(\frac{D_e}{L} \right) \right]^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14}$

For Turbulent Flow: $$\frac{h_o D_e}{k} = 0.027\left(\frac{D_e G_a}{\mu}\right)^{0.8}\left(\frac{c_p \mu}{k}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

Overall coefficients:

(9) Compute U_C,

$$\frac{1}{U_o A_o} = \frac{1}{h_i A_i} + \frac{\Delta x_w}{k A_{LM}} + \frac{1}{h_o A_o}$$

$$= \frac{1}{h_i A_i} + \frac{1}{h_o A_o},$$ neglecting pipe wall resistance

$$= \frac{h_i h_o}{h_i + h_o},$$ assuming $A_i \approx A_o$

(10) Compute U_D, $$\frac{1}{U_D} = \frac{1}{U_C} + R_d$$

Compute *A from* $Q = U_D A \Delta T_{LM}$ *which may be translated into length.*

An additional task in this step is to recalculate the dirt factor. If the length should not correspond to an integral number of hairpins, a change in dirt factor will result.

The recalculated dirt factor should equal or exceed the required dirt factor by using the next larger integral number of hairpins.

Calculation of *P*

This requires a knowledge of the total length of the path satisfying the heat-transfer requirements.

For example, if the calculated pressure drop, *P, is too high, the rate of* heat transfer may be too low for it to operate as specified.

This situation arises because the fluid would require more energy for it to be flowing through the system. If the pump forcing the fluid through the system does not provide sufficient flow rate, this may result in a low *h, low U, and* therefore a reduced rate of heat transfer, *Q.*

ΔP for Inner pipe (p):

(1) Compute Re_p, $Re_p = \frac{DG}{\mu}$, identify type of flow

(2) Compute *f and* ΔF_p, $f = \frac{16}{(DG/\mu)}$, for laminar flow

$$f = 0.0035 + \frac{0.264}{\left(DG/\mu^{0.42}\right)}, \text{ for turbulent flow}$$

$$\Delta F_p = \frac{4fG^2L}{2g\rho^2 D}[=]\text{ft}$$

(3) Compute P_p,* $\quad \Delta P_p = \Delta F_p\left(\frac{\rho}{144}\right)[=]\text{psi}$

**Note that 144 converts units from ft² to in²*

ΔP for Annulus (a):

(1') *Compute D'_e and Re'_a, $\quad D'_e = \frac{4\pi\left(D_2^2 - D_1^2\right)}{4\pi\left(D_2 + D_1\right)} = D_2 - D_1$

$$Re'_a = \frac{D'_e G_a}{\mu}, \text{ identify type of flow}$$

*Note that D'_e for pressure drop differs from D_e used for the heat transfer rate calculation. The Reynolds number must be calculated based on D'_e .

(2) Compute f, ΔF_a, and ΔF_i, $f = \dfrac{16}{\left(D_e' G_a / \mu\right)}$, for laminar flow

$$f = 0.0035 + \frac{0.264}{\left(D_e' G_a / \mu^{0.42}\right)},$$

for turbulent flow

$$\Delta F_a = \frac{4 f G_a^2 L}{2 g \rho^2 D_e'} [=] \text{ft}$$

$$\Delta F_i = \frac{\overline{v}^2}{2g'} [=] \text{ft/hairpin}$$

(3) Compute ΔP_a, $\Delta P_a = \dfrac{\left(\Delta F_a + \Delta F_i\right)\rho}{144} [=] \text{psi}$

There is an advantage if both fluids are computed side by side, as per Kern's recommendation.