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SCOPE

This section covers heat transfer to or from the contents of agitated process vessels. The primary function of agitation in these vessels is mixing. Thus they are designed to satisfy a given mixing requirement. Heat transfer is considered a secondary requirement and is calculated for an already determined mixer configuration.

REFERENCES

DESIGN PRACTICES

[Section IX](#) Heat Exchange Equipment
[Section XIV](#) Fluid Flow

OTHER REFERENCES

Edwards, M. F. and Wilkinson, W. L., *Heat Transfer to Newtonian and Non-Newtonian Fluids in Agitated Vessels*, HTFS - DR 27, Heat Transfer and Fluid Flow Service, AERE, Harwell, 1972.
Uhl, V. W. and Gray, J. B., eds. *Mixing-Theory and Practice, Vol. 1*, Academic Press, 1966.
Nagata, S., *Mixing, Principles and Applications*, Halsted Press, 1975.

BACKGROUND

Frequently the contents of an agitated vessel must be heated or cooled to a given operating temperature by intentional heat transfer to or from the fluid in the vessel. Additionally, in some mixing situations, the mixing power dissipation to the fluid is high enough to cause a significant temperature increase. This may require a design, which includes intentional cooling.

Since the vessels are actually designed for their mixing duties, the agitators used are those appropriate for this mixing. In general, agitators which force large amounts of fluid to circulate near the heat transfer surfaces provide the most efficient heat transfer. In the case of close proximity impellers used for high viscosity ($> 5,000$ cP or $5 \text{ Pa}\cdot\text{s}$) mixing, heat transfer is promoted by the thinning of the amount of stagnant fluid near the wall. Typical clearance between impeller and tank wall is 1/2 in. (12.5 mm). Rubber scrapers can additionally be added to close proximity impellers to keep the wall surface free from deposits.

The surface needed for heat transfer can take a variety of forms. For close proximity impellers, the vessel wall itself is used in conjunction with jacketing. For turbines and propellers, the vessel walls or internal coils can be used. The various arrangements in common use are discussed below.

DEFINITIONS

The definitions given in [Section XIII-A](#) also pertain to this section.

HEAT TRANSFER SURFACE ARRANGEMENTS

The commonly used heat transfer surfaces are the vessel walls, through the use of external jackets, and internal coils. In general, coils are preferred over jackets because of lower costs, higher operating pressures, and higher inside heat transfer coefficients. Coils are particularly suitable for low viscosity fluids in combination with impellers, which give high radial flows normal to the coils. On the other hand for high viscosity fluids, a close clearance impeller in a jacketed vessel is probably more effective. Also coils are unsuitable with process liquids that foul. For "standard geometry," coils have about the same heat transfer area as jackets. However, by increasing the helix diameter, decreasing the spacing between turns, or using concentric coils, the heat transfer area provided by coils can be increased. Since these changes in coil design restrict fluid circulation in the vessel and thus heat transfer coefficient, proportional increases in total heat transfer are not achieved.

Tank baffling should be used for other than close proximity impellers as dictated by the mixing situation. For jacketed vessels, wall baffles should be used if the fluid viscosity is at any time less than 5000 cP ($5 \text{ Pa}\cdot\text{s}$). For vessels equipped with coils, wall baffles should be used if the clear space between turns is at least twice the OD of the coil tubing and the fluid viscosity is less than 1000 cP ($1 \text{ Pa}\cdot\text{s}$). Otherwise the baffles should be located inside of the coil helix.

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HEAT TRANSFER SURFACE ARRANGEMENTS (Cont)

JACKETS FOR UTILIZING THE VESSEL WALLS

A typical full vessel side and bottom jacketing configuration is shown in **Figure 1**. For this configuration the heat transfer area is given by:

$$A = \pi T^2 \left(\frac{L}{T} + 0.268 \right) \quad \text{Eq. (1)}$$

where: A = Heat transfer area, ft² (m²)
T = Vessel diameter, ft (m)
L = Vessel straight side, ft (m)

For a given vessel volume, a higher L/T ratio results in higher surface area. For simpler configurations, the area can be directly calculated for the cylindrical or circular geometry involved.

Conventional Jackets - A conventional jacket is shown in **Figure 1**. Spiral baffling within the jacket as shown in **Figure 2** should be used. Conventional jackets are a good choice for small vessels (< 500 gal or 1.9 m³) and for vessels with high internal pressures (internal pressure > 2 x jacket pressure). They are not recommended for jacket pressures greater than 100 psi (680 kPa) or high temperatures.

Half-Pipe Jackets - A typical half-pipe jacket is shown in **Figure 3**. They are useful for high pressures (up to 600 psi or 4082 kPa) especially when jacket pressure determines the vessel wall thickness. Half-pipe jackets are better for liquid than for vapor service fluid. They can be easily zoned so that only a portion of the heat transfer area is used. A 3 in. (76 mm) nominal pipe size is typically used for forming the jacket. For maximum heat transfer, the space between the half-pipe coils is 3/4 in. (19 mm). However, if the inside heat transfer film coefficient is low, the space between the half-pipe coils can be increased without significant loss in heat transfer surface.

Dimple Jackets - A typical dimple jacket is shown in **Figure 4**. Such jackets are suitable for larger vessels, particularly when jacket pressure determines the vessel wall thickness ($P_{\text{jacket}}/P_{\text{vessel}} > 0.6$). They are satisfactory up to 300 psi (2041 kPa) and 700°F (371°C). High jacket pressure drops are encountered with liquid flows. Typically the dimples are on a square pitch of 2.5 in. (63.5 mm).

External Electrical Heaters - The vessel wall can also be used as the heat transfer surface by placing external electrical heaters around the vessel. For small vessels, strap-on electrical heaters are available.

INTERNAL COILS

Helical Coils - Helical coils are the most common type of coil. Typical layout and relative dimensions are given in **Figure 5**. The heat transfer area provided by a helical coil is given by:

$$A = \pi d_c n_t \left[(\pi D_c)^2 + (S_c + d_c)^2 \right]^{1/2} \quad \text{Eq. (2)}$$

where: A = Heat transfer area (process side), ft² (m²)
d_c = Coil tubing outside diameter, ft (m)
n_t = Number of helical turns
D_c = Coil helix diameter, ft (m)
S_c = Open space between turns, ft (m)

Baffle Coils - A typical baffle coil is shown in **Figure 6**. They are less common than helical coils but they provide heat transfer area while also serving as baffles.

HEAT TRANSFER CORRELATIONS AND DESIGN PROCEDURE

Heat transfer rates in agitated vessels have been correlated in a manner similar to that in conventional heat exchangers; namely, by the use of process side and service fluid side heat transfer coefficients. Correlations for the coefficients are normally given in terms of products of dimensionless groups and are basically empirical fits of experimental data. The correlations given below are from literature sources and are believed to be reliable. However, they have not been validated by ExxonMobil operating or laboratory data.

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HEAT TRANSFER CORRELATIONS AND DESIGN PROCEDURE (Cont)

PROCESS SIDE HEAT TRANSFER COEFFICIENTS

The nomenclature used in all of the following Eqs. (3) through (14) is:

C	=	Impeller elevation (to center of impeller) above bottom of vessel, ft (m)
C _p	=	Fluid heat capacity, Btu/lb°F (J/kg°C)
D	=	Impeller diameter, ft (m)
D _c	=	Diameter of coil helix, ft (m)
d _c	=	Outside diameter of tube coil, ft (m)
h _c	=	Heat transfer coefficient, vessel fluid to coil wall, Btu/(hr•ft ² •°F) (W/m ² °C)
h _j	=	Heat transfer coefficient, vessel fluid to jacket wall, Btu/(hr•ft ² •°F) (W/m ² °C)
i	=	Number of impellers
k	=	Thermal conductivity, Btu/(hr•ft ² •°F/ft) (W/m°C)
N	=	Impeller rotational speed, rpm (rps)
n _b	=	Number of baffles
n _p	=	Number of blades on turbine impeller
T	=	Vessel diameter, ft (m)
w	=	Blade width of impeller in direction parallel to axis of rotation, ft (m)
Z	=	Fluid depth in vessel, ft (m)
θ	=	Turbine blade angle from horizontal, °
μ	=	Fluid viscosity at bulk temperatures, lb/(ft•hr) (Pa•s*)
μ _w	=	Fluid viscosity at wall temperature, lb/(ft•hr) (Pa•s*)
ρ	=	Fluid density, lb/ft ³ (kg/m ³)

The physical properties used are those of the process fluid.

* Viscosity in Pa•s is obtained by multiplying viscosity in centipoises by 10⁻³.

Turbines - Paddles (All Styles)

1. Unbaffled

For jackets:

$$\frac{h_j T}{k} = 0.525 \left(\frac{D^2 N \rho}{\mu} \right)^{2/3} \left(\frac{C_p \mu}{k} \right)^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14} \left(\frac{D}{T} \right)^{-0.25} \left(\frac{w}{T} \right)^{0.15} n_p^{0.15} \left(\frac{C}{Z} \right)^{0.15} (\sin \theta)^{0.5} \quad \text{Eq. (3)}$$

For helical coils with the impeller inside the coil array:

$$\frac{h_c T}{k} = 0.825 \left(\frac{D^2 N \rho}{\mu} \right)^{0.56} \left(\frac{C_p \mu}{k} \right)^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14} \left(\frac{D}{T} \right)^{-0.25} \left(\frac{w}{T} \right)^{0.15} n_p^{0.15} \left(\frac{d_c}{T} \right)^{-0.3} \quad \text{Eq. (4)}$$

For helical coils with the impeller below the coil array:

$$\frac{h_c T}{k} = 1.05 \left(\frac{D^2 N \rho}{\mu} \right)^{0.62} \left(\frac{C_p \mu}{k} \right)^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14} \left(\frac{D}{T} \right)^{-0.25} \left(\frac{w}{T} \right)^{0.15} n_p^{0.15} \left(\frac{D_c}{T} \right) \quad \text{Eq. (5)}$$

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HEAT TRANSFER CORRELATIONS AND DESIGN PROCEDURE (Cont)

2. Baffled

(4 baffles except as included in correlation for baffle coils)

For jackets:

$$\frac{h_j T}{k} = 1.40 \left(\frac{D^2 N \rho}{\mu} \right)^{2/3} \left(\frac{C_p \mu}{k} \right)^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14} \left(\frac{D}{T} \right)^{-0.3} \left(\frac{w}{T} \right)^{0.45} n_p^{0.2} \left(\frac{C}{Z} \right)^{0.2} \left(\frac{Z}{T} \right)^{-0.6} (\sin \theta)^{0.5} \quad \text{Eq. (6)}$$

For helical coils:

$$\frac{h_c T}{k} = 2.68 \left(\frac{D^2 N \rho}{\mu} \right)^{0.56} \left(\frac{C_p \mu}{k} \right)^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14} \left(\frac{D}{T} \right)^{-0.3} \times \left(\frac{w}{T} \right)^{0.3} n_p^{0.2} \left(\frac{C}{Z} \right)^{0.15} \left(\frac{Z}{T} \right)^{-0.5} (\sin \theta)^{0.5} \quad \text{Eq. (7)}$$

For baffle coils:

$$\frac{h_c d_c}{k} = 0.10 \left(\frac{D^2 N \rho}{\mu} \right)^{0.65} \left(\frac{C_p \mu}{k} \right)^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14} \left(\frac{D}{T} \right)^{1/3} n_b^{-0.2} \quad \text{Eq. (8)}$$

For multiple turbines (i in number) the sum of impeller blade widths ($\sum w_i$) should be used for w and the average impeller height $\frac{\sum C_i}{i}$ should be used for C in the equations which include these terms. n_p remains unchanged at the number of blades per turbine. With turbines having different diameters on the same shaft, a weighted average diameter based upon the exponents of D in the appropriate equation should be used.

Propellers (baffled or unbaffled)

For jackets:

$$\frac{h_j T}{k} = 0.33 \left(\frac{D^2 N \rho}{\mu} \right)^{2/3} \left(\frac{C_p \mu}{k} \right)^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14} \left(\frac{D}{T} \right)^{-0.25} \left(\frac{C}{Z} \right)^{0.15} \quad \text{Eq. (9)}$$

For helical coils:

$$\frac{h_c T}{k} = 1.31 \left(\frac{D^2 N \rho}{\mu} \right)^{0.56} \left(\frac{C_p \mu}{k} \right)^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14} \left(\frac{D}{T} \right)^{-0.25} \left(\frac{C}{Z} \right)^{0.15} \quad \text{Eq. (10)}$$

Close Proximity Impellers

The following correlations apply to installations without wall scrapers. If scrapers are used, the values given by the equations should be multiplied by a factor of 1.3.

Anchors:

For $N_{Re} < 200$

$$\frac{h_j T}{k} = 1.50 \left(\frac{D^2 N \rho}{\mu} \right)^{0.5} \left(\frac{C_p \mu}{k} \right)^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14} \quad \text{Eq. (11)}$$

For $N_{Re} > 200$

$$\frac{h_j T}{k} = 0.36 \left(\frac{D^2 N \rho}{\mu} \right)^{2/3} \left(\frac{C_p \mu}{k} \right)^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14} \quad \text{Eq. (12)}$$