ESTIMATE HEAT TRANSFER and FRICTION in DIMPLE JACKETS

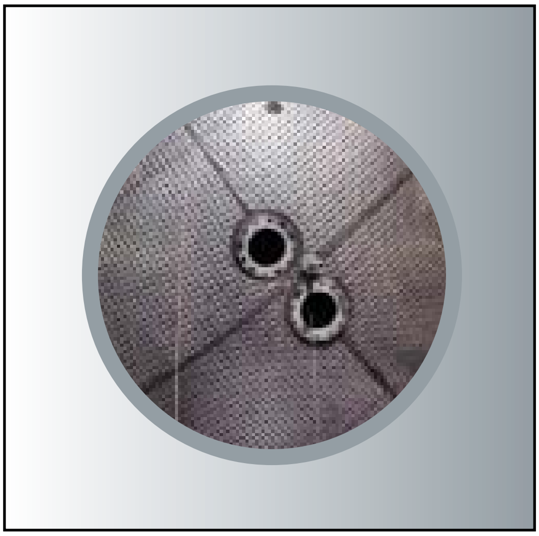

Photo courtesy of Paul Mueller Co.

Use these correlations to determine the overall heat-transfer coefficient and the frictional pressure drop in dimple jackets.

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Dimple jackets are frequently used to heat or cool process vessels in the chemical process industries (CPI). Among their advantages: They are relatively inexpensive to fabricate. Relatively light-gauge stainless steel can be used for cooling medium pressures of 6 to 8 bar, since the jacket is supported on a grid of plug welds. Dimple jackets have far better heat-transfer characteristics than conventional jackets due to the turbulence generated by the dimples. Heat transfer can be augmented by increasing jacket velocities through the use of jacket baffles. For differential heating or cooling, jackets can be segmented into zones. Dimple jackets can be fitted to both cylindrical and dished portions of a process vessel.

Unlike for other jacket types (*e.g.,* half-pipe, conventional, baffled conventional, agitated conventional), generalized prediction methods for heat transfer and frictional pressure drop in dimple jackets have not been available. This article presents correlations that are suitable for general use.

Database and correlations

Correlations for heat transfer and pressure drop in dimple jackets are derived by analyzing other hydraulically similar nonuniform ducted flows. Flows in dimple jackets, continuous fin-tube exchangers, and pin-fin exchangers are all examples of nonuniform ducted flows. They are characterized by flow through banks of rods, tubes, dimples, or similar obstructions confined between parallel plates.

Heat-transfer and pressure-drop correlations were obtained from Refs. 1–3 for pin-fin plate-fin surfaces, finned circular tubes, and finned flat tubes *(1),* pin-fin exchangers *(2),* and fin-tube exchangers

(3). The data were validated and experimentally derived based on known test conditions.

The Coburn heat-transfer factor is given by the following correlation:

$$j = 0.0845(w/x)^{0.368}(A_{min}/A_{max})^{-0.383}Re^{-0.305} \quad (1)$$

All fluid properties are evaluated at bulk flow conditions. The equation is valid for Reynolds numbers from 1,000 through 50,000, with an average absolute error of 9.8% and a maximum error of 30% over 116 data points.

From a knowledge of the Coburn *j*-factor, the jacket-side film coefficient can immediately be calculated by:

$$j = Nu/RePr^{0.33} \quad (2)$$

$$Nu = hd_0/k \quad (3)$$

This correlation was compared with proprietary heat-transfer information for dimple jackets obtained from a major manufacturer of dimple jackets. The *j*-values from the vendor were an average of 15–20% lower than the values predicted by the correlation over the normal range of operation. The deviation was largest at low jacket velocities (less than 0.5 m/s). At high flow rates (average jacket velocities greater than 1.5 m/s), there was good agreement between the vendor's data and the correlation.

Nomenclature

Any consistent set of units may be used. Refer to Figure 1 for clarification of dimple jacket geometry.

- A_{min} = minimum flow area = $z(w - d_0)$
- A_{max} = maximum (unrestricted) flow area = zw
- A = heat-transfer area
- c = specific heat of jacket fluid
- d_e = equivalent diameter of minimum flow area = $4A_{min}/P_{min}$
- d_0 = mean or equivalent external diameter of dimple = $(d_1 + d_2)/2$
- d_1 = smaller dimple diameter
- d_2 = larger dimple diameter
- f = Fanning friction factor = $d_eK/(4x)$
- h_j = film coefficient in dimple jacket
- h_{jf} = fouling coefficient in dimple jacket
- h_w = coefficient for vessel wall
- h_v = film coefficient in vessel
- j = Coburn heat-transfer factor = $Nu/RePr^{1/3}$
- k = thermal conductivity of jacket fluid
- K = flow coefficient = $\Delta P_t/(\rho V_{max}^2/2) = 4fx/d_e$
- Nu = Nusselt number = h_jd_0/k
- P_{min} = perimeter of minimum flow area
- Pr = Prandtl number = $c\mu/k$
- Q = heat-transfer rate
- Re = Reynolds Number = $d_0V_{max}\rho/\mu$
- U = overall heat-transfer coefficient
- V_{max} = maximum jacket velocity = Q/A_{min}
- w = transverse center-to-center distance between adjacent dimples
- x = center-to-center distance between dimples parallel to flow
- z = height of minimum flow area
- ΔP_t = pressure drop per dimple row
- ΔT = temperature difference between jacket and vessel fluids
- ρ = density of jacket fluid
- μ = viscosity of jacket fluid

Flow coefficient correlation

No useful correlation for the flow coefficient, *K,* for Reynolds numbers less than 5,000 could be developed. For $Re > 5{,}000$, Eq. 4 can be used, with an average absolute error of 10.6% and a maximum error of 25% over 62 datapoints.

$$K = 0.135 + 0.937(w/z)^{0.575}(A_{min}/A_{max})^{-2.10}Re^{-0.33} \quad (4)$$

With a knowledge of the flow coefficient *K,* the pressure drop per dimple (and total pressure drop across the jacket) can be calculated from:

$$K = \Delta P_t/(\rho V_{max}^2/2) \quad (5)$$

Additional allowances for entry and exit losses should be made.

Proprietary pressure-drop data from three manufacturers showed significant scatter about this correlation. There was good agreement with one firm's dimple plate data, however.

Sample calculation

Figure 1 defines the geometry of a dimple jacket.

A stainless steel vessel with a 1,000-mm internal diameter, 1,000-mm tan-to-tan measurement, and 7-mm wall has a dimple jacket fitted to 75% of its cylindrical portion. The dimples are laid out on a square 75-mm pitch (*i.e.,* $x = w = 0.075$ m). Dimple diameters d_1 and d_2 are 30 mm and 60 mm, respectively. The dimple depth is 25 mm (*i.e.,* $z = 0.025$ m). There is a single baffle fitted in the jacket. The jacket is supplied with 250 L/min of chilled water at 5°C. The vessel is filled with water at 80°C. Assuming a vessel-side film coefficient of 2,400 W/m²•K, calculate the overall heat-transfer coefficient and heat-transfer rate for clean and fouled conditions.

Flow rate per dimple. On entry to the jacket, the 250 L/min splits into two flows of 125 L/min, one passing clockwise around the jacket and the other passing counterclockwise. These flows meet at the far side of the vessel, reverse in direction around the jacket baffle, and flow toward the outlet. The overall height of the jacket is 750 mm split into two passes by the jacket baffle. There are five dimples (5 mm × 75 mm) across each pass of 375 mm, resulting in 125/5 = 25 L/min per dimple = 0.0004167 m³/s.

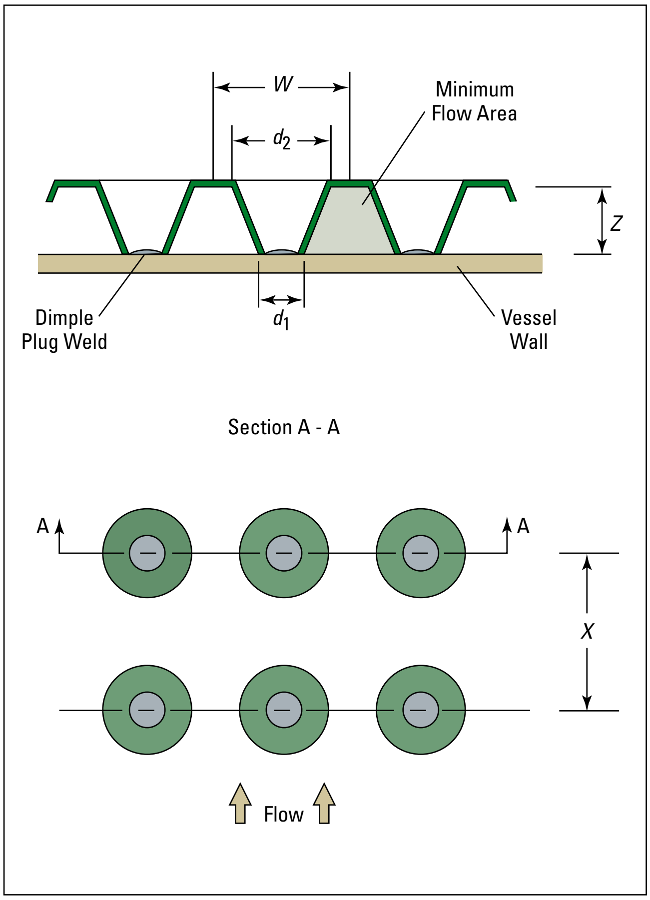

Figure 1. Dimple jacket geometry.

Physical properties. For water at 5°C, ρ = 1,000 kg/m³, μ = 1.507 cP = 0.001507 kg/ms, k = 0.5758 W/m•K, and Pr = 11.04. For stainless steel, k = 16 W/m•K.

Geometry and Reynolds number. d_0 = (0.030 + 0.060)/2 = 0.045 m. A_{min} = 0.025 × (0.075 – 0.045) = 0.00075 m². A_{max} = 0.025 × 0.075 = 0.001875 m². A_{min}/A_{max} = 0.40. w/x = 1. V_{max} = 0.0004167/0.00075 = 0.556 m/s. Re = (0.045)(1,000)(0.556)/ 0.001507 = 16,604.

Jacket film coefficient. The Coburn j-factor is given by the correlation $j = 0.0845(1)^{0.368}(0.40)^{-0.383}(16{,}604)^{-0.305}$ = 0.0662. The jacket film coefficient is then $h_j = Nu \times k/d_0$ = $(jRePr^{0.33})k/d_0$ = $(0.0662 \times 16{,}604 \times [11.04]^{0.33})$ (0.5758/0.045) = 3132 W/m²•K = 551 Btu/ft²•h•°F.

Vessel wall coefficient. h_w = 16/0.007 = 2,286 W/m²•K = 403 Btu/ft²•h•°F.

Jacket fouling coefficient. A typical jacket-side fouling coefficient for chilled-water duty is h_{jf} = 1,000 Btu/ft²•h•°F = 5,680 W/m²•K.

Overall coefficient. The overall heat-transfer coefficient under clean conditions is given by $1/U = 1/h_j + 1/h_w + 1/h_v$ = 1/3,132 + 1/2,286 +1/2,400. U = 852 W/m²•K = 150 Btu/ft²•h•°F.

With design fouling in the jacket (the vessel is assumed to be clean), $1/U = 1/h_j + 1/h_{jf} + 1/h_w + 1/h_v$ = 1/3,132 + 1/5,680 + 1/2,286 + 1/2,400. U = 741 W/m²•K = 131 Btu/ft²•h•°F.

Rate of heat transfer. The mean chilled water temperature = 7.5°C (the chilled water temperature rises 5°C across the jacket). The vessel temperature is 80°C. The jacket heat-transfer area is π(1)(0.75) = 2.36 m². Note that the total dimple jacket area is included in the heat-transfer area; the area occupied by the plug welds is also effective for heat-transfer purposes because of the fin effect.

With no fouling in the jacket, $Q_{clean} = UA\Delta T$ = 852(2.36)(80 – 7.5) = 145 kW.

With design fouling in the jacket, Q_{fouled} = 741(2.36)(80 – 7.5) = 127 kW. CEP

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