Predict storage-tank heat transfer precisely

Use this procedure to determine the rate of heat transfer from a vertical storage tank when shortcut methods are inadequate.

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□ Heating or cooling storage tanks can be a major energy expense at plants and tankfarms. Though many procedures for calculating such heat-transfer requirements have been published [*1,3,5,7,8,10*], the simplifying assumptions that they use can lead to significant errors in computed heat-transfer rates. This is of concern because efficient sizing of tanks, insulation, heaters and coolers depends on accurate estimates of heat transfer to and from the various tank surfaces. And the ultimate value of being accurate increases as energy costs continue to rise.

The procedure presented here determines the heat transfer to or from a vertical-cylindrical storage tank seated on the ground—like the one in Fig. 1. It includes the effects of tank configuration, liquid level, ambient temperature and wind speed, as well as temperature variations within the tank and between air and ground. A partially worked example shows how to use the technique, and how to do the calculations on a computer.

The theory

Storage tanks come in many different shapes and sizes. Horizontal-cylindrical and spherical tanks are used for storage of liquids under pressure; atmospheric tanks tend to be vertical-cylindrical, with flat bottoms and conical roofs as shown in Fig. 1. The example presented here is for the latter configuration, but the procedure applies to any tank for which reliable heat-transfer correlations are available.

For the sake of simplicity, we assume that the tank contents are warmer than the ambient air, and that we are concerned with heat loss from the tank rather than heat gain. But the method may, of course, be applied to either case.

Consider, then, the categories of surfaces from which heat may be transferred across the tank boundaries: wet or dry sidewalls, tank bottom, and roof. In the context used here, "wet" refers to the portion of the wall submerged under the liquid surface, whereas "dry" refers to the portion of the wall in the vapor space, above the liquid surface.

In general, the heating coils would be located near the bottom of the tank, in the form of flat "pancakes." Therefore, the temperature of the air (or vapor) space above the liquid level may be expected to be lower than the liquid itself. Experience has shown that the average bulk temperatures of the liquid and vapor space may be significantly (i.e., more than 5°F) different, and they are treated accordingly in our procedure. Use of different liquid and vapor temperatures is an important departure from the traditional approach, which assumes the same value for both.

Our basic approach is to develop equations for calculating the heat loss from each of the four categories of surfaces, and then add the individual heat losses to get the total heat loss. Thus:

For dry sidewall $$q_d = U_d A_d (T_V - T_A) \quad (1)$$

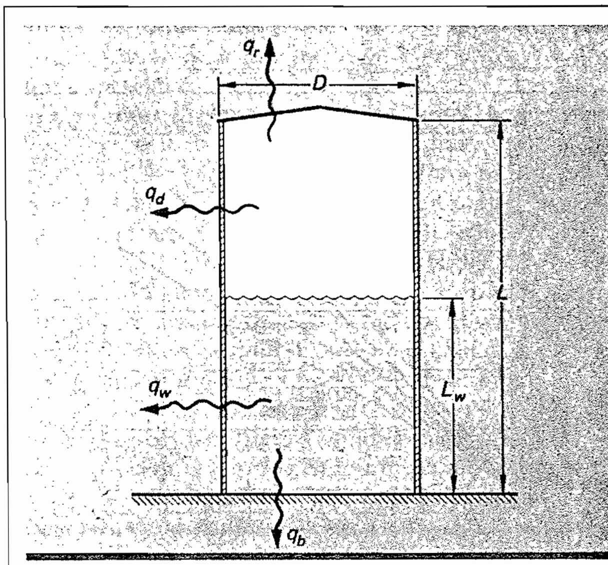
For wet sidewall $$q_w = U_w A_w (T_L - T_A) \quad (2)$$

For tank bottom $$q_b = U_b A_b (T_L - T_G) \quad (3)$$

For tank roof $$q_r = U_r A_r (T_V - T_A) \quad (4)$$

Total $$Q = q_d + q_w + q_b + q_r \quad (5)$$

When using these equations in design or rating problems, we either assume the various temperatures for typ-



Typical vertical-cylindrical tank **Fig. 1**

Individual film heat-transfer coefficients — **Table I**

Type/surface	Dry wall	Wet wall	Roof	Bottom
Inside	h_{Vw}	h_{Lw}	h_{Vr}	h_{Lb}
Wall conduction	$\left(\frac{t_M}{k_M}+\frac{t_I}{k_I}\right)^{-1}$	$\left(\frac{t_M}{k_M}+\frac{t_I}{k_I}\right)^{-1}$	$\left(\frac{t_M}{k_M}\right)^{-1}$	$\left(\frac{t_M}{k_M}\right)^{-1}$
Outside	$W_f h'_{Aw} + h_{Rd}$	$W_f h'_{Aw} + h_{Rw}$	$W_f h'_{Ar} + h_{Rr}$	h_G
Fouling	h_{Fd}	h_{Fw}	h_{Fr}	h_{Fb}

Note: Tank roof and bottom are uninsulated.

ical conditions or determine them by measurement. The area values are also easy to obtain:

$$A_d = \pi D (L - L_w) \quad (6)$$

$$A_w = \pi D L_w \quad (7)$$

$$A_b = \pi D^2/4 \quad (8)$$

$$A_r = (\pi D/2)(D^2/4 + d^2)^{1/2} \quad (9)$$

The complications arise when we try to estimate the overall heat-transfer coefficients U_d, U_w, U_b and U_r, for the four surfaces of the tank. For the tank geometry chosen, these can fortunately be calculated from the individual film heat-transfer coefficients in the conventional manner, using published correlations.

The overall coefficients

Table I shows the component coefficients for each surface. The overall heat-transfer coefficient for the dry sidewall of the tank (U_d) is calculated as the sum of the resistances of vapor film, fouling, metal wall, insulation (if any), and outside air (convection plus radiation).

The outside-air heat-transfer coefficient (h_{Aw}) is a function of wind velocity as well as temperature gradient. Data on the effect of wind velocity and ΔT have been presented by Stuhlbarg [10] and Boyen [2]. With a little bit of manipulation, their data were replotted, yielding the "wind enhancement factor" (W_f) in Fig. 2. By definition:

$$W_f = h_{Aw}/h'_{Aw} = h_{Ar}/h'_{Ar} \quad (10)$$

Therefore, once the outside-air coefficient for *still* air (h'_{Aw}) is known, the overall dry-sidewall coefficient at various wind velocities can be computed as:

$$1/U_d = 1/h_{Vw} + t_M/k_M + t_I/k_I + 1/(W_f h'_{Aw} + h_{Rd}) + 1/h_{Fd} \quad (11)$$

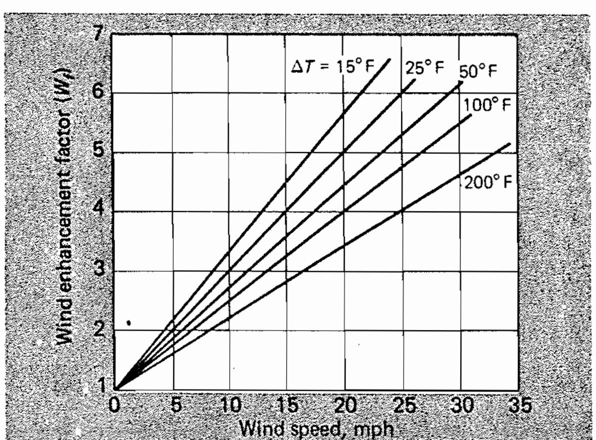
Similarly, the overall coefficients for the wet sidewall, bottom and roof surfaces are:

$$1/U_w = 1/h_{Lw} + t_M/k_M + t_I/k_I + 1/(W_f h'_{Aw} + h_{Rw}) + 1/h_{Fw} \quad (12)$$

$$1/U_b = 1/h_{Lb} + t_M/k_M + 1/h_G + 1/h_{Fb} \quad (13)$$

$$1/U_r = 1/h_{Vr} + t_M/k_M + 1/(W_f h'_{Ar} + h_{Rr}) + 1/h_{Fd} \quad (14)$$

Eq. 13 and 14 assume that the roof and bottom are not insulated, which is generally the case in temperate climates. We shall now review correlations for the individual heat-transfer coefficients needed to obtain the overall coefficients.



Effect of wind velocity and ΔT on heat-transfer rate — **Fig. 2**

Individual film heat-transfer coefficients

The film heat-transfer coefficients may be divided into four categories: convection from vertical walls, convection from horizontal surfaces, pure conduction, and radiative heat transfer. Within each category, correlations are presented for several flow regimes.

Vertical-wall film coefficients. These apply to the inside wall (wet or dry) and the outside wall (still air). For vertical plates and cylinders, Kato et al. [6] recommend the following for liquids and vapors:

$$N_{Nu} = 0.138\, N_{Gr}^{0.36}(N_{Pr}^{0.175} - 0.55) \quad (15)$$

where $0.1 < N_{Pr} < 40$ and $N_{Gr} > 10^9$.

For isothermal vertical plates, Ede [4] reported the following for liquids:

$$N_{Nu} = 0.495(N_{Gr}N_{Pr})^{0.25} \quad (16)$$

where $N_{Pr} > 100$ and $10^4 < (N_{Gr}N_{Pr}) < 10^9$, and for gases:

$$N_{Nu} = 0.0295\, N_{Gr}^{0.40}\, N_{Pr}^{0.47}\,(1 + 0.5\, N_{Pr}^{0.67})^{-0.40} \quad (17)$$

where $N_{Pr} \approx 5$ and $(N_{Gr}N_{Pr}) > 10^9$.

For vertical plates taller than 3 ft, Stuhlbarg [10] recommends:

$$h = 0.45\, k\, L^{-0.75}\,(N_{Gr}N_{Pr})^{0.25} \quad (18)$$

where $10^4 < (N_{Gr}N_{Pr}) < 10^9$.

Horizontal-surface heat-transfer coefficients. These coefficients apply to the roof and inside-bottom surfaces of the tank. The bottom is assumed to be flat. For surfaces facing up [8]:

$$N_{Nu} = 0.14\,(N_{Gr}N_{Pr})^{0.33} \quad (19)$$

For surfaces facing down:

$$N_{Nu} = 0.27\,(N_{Gr}N_{Pr})^{0.25} \quad (20)$$

Nomenclature

- A — Area of heat-transfer surface, ft^2; A_b for bottom, A_d for dry wall, A_w for wet wall, A_r for roof
- c_p — Specific heat at constant pressure, Btu/lb-°F
- D — Diameter of tank, ft
- d — Height of conical roof at center, ft
- g — Acceleration due to gravity, 4.17×10^8 ft/h^2
- h — Individual film coefficient of heat transfer, Btu/ft^{2}h-°F; h_{Aw} for air outside the walls, h_{Ar} for air above the roof, h'_{Aw} and h'_{Ar} for still air, h_{Lw} for liquid between the walls, h_{Lb} for liquid near the bottom, h_{Vw} for vapor near the walls, h_{Vr} for vapor near the roof
- h_F — Fouling coefficient, Btu/ft^{2}h-°F; h_{Fw} for liquid at the walls, h_{Fb} for liquid at the bottom, h_{Fv} for vapor at the walls or the roof
- h_G — Heat-transfer coefficient for ground, Btu/ft^{2}h-°F
- h_I — Heat-transfer coefficient for insulation, Btu/ft^{2}h-°F
- h_M — Heat-transfer coefficient for metal, Btu/ft^{2}h-°F
- h_R — Heat-transfer coefficient for radiation, Btu/ft^{2}h-°F; h_{Rb} for bottom, h_{Rd} for dry wall, h_{Rw} for wet wall, h_{Rr} for roof
- k — Thermal conductivity, Btu/ft-h-°F; k_G for ground, k_I for insulation, k_M for metal wall
- L — Total length for heat-transfer surface, ft
- L_w — Total length for wetted surface, ft
- N_{Gr} — Grashof number, $L^3\rho^2 g\beta\Delta T/\mu^2$
- N_{Nu} — Nusselt number, hD/k or hL/k
- N_{Pr} — Prandtl number, $c_p\mu/k$
- Q — Rate of heat transfer, Btu/h
- q — Individual rate of heat transfer, Btu/h; q_b for bottom, q_d for dry wall, q_w for wet wall, q_r for roof
- T — Temperature, °F; T_A for ambient air, T_L for bulk liquid, T_V for vapor, T_G for ground, T_w for inside wall, T_{ws} for outside wall
- ΔT — Temperature difference, °F
- t — Surface thickness, ft; t_I for insulation, t_M for metal
- U — Overall heat-transfer coefficient, Btu/ft^{2}h-°F; U_b for bottom, U_d for dry wall, U_w for wet wall, U_r for roof
- W_f — Wind enhancement factor
- β — Volumetric coefficient for thermal expansion, °F^{-1}
- μ — Viscosity of fluid, lb/ft-h
- ρ — Density of fluid, lb/ft^3
- ε — Emissivity

Both equations apply in the range $2 \times 10^7 < (N_{Gr} N_{Pr}) < 3 \times 10^{10}$.

Equivalent coefficients for conductive heat transfer. The wall and insulation coefficients are derived from the thermal conductivities:

$$h_M = k_M/t_M \tag{21}$$

$$h_I = k_I/t_I \tag{22}$$

The coefficient for heat transfer to and from the ground is the coefficient for heat conduction from a semi-infinite solid [9]:

$$h_G = 8\, k_G/\pi D \tag{23}$$

Fouling coefficients. The coefficients h_{Fd}, h_{Fw} and h_{Fb} apply to the vapor and liquid at the wall, and the liquid at the bottom of the tank, respectively. These are empirical, and depend on the type of fluid and other factors such as tank cleaning. Generally, h_{Fd} is the greatest of the three, and h_{Fb} the least, indicating that the greatest fouling resistance is at the bottom of the tank.

Equivalent coefficient for radiative heat transfer. The coefficient for sidewalls and roof depends on the emissivity of these surfaces, and is given by [8]:

$$h_R = \frac{0.1713\varepsilon}{(T_{ws} - T_A)}\left[\left(\frac{T_{ws} + 460}{100}\right)^4 - \left(\frac{T_A + 460}{100}\right)^4\right] \tag{24}$$

With these relationships, we now have the tools to calculate heat transfer to or from the tank.

Example

ABC Chemical Corp. has a single manufacturing plant in the U.S., and exports a high-viscosity specialty oil product to Europe. The oil is offloaded in Port City, and stored in a flat-bottom, conical-roof tank rented from XYZ Terminal Co. Ltd. The tank is located outdoors and rests on the ground. It is equipped with pancake-type steam-heating coils because the oil must be maintained above 50°F in order to preserve its fluidity. Other pertinent data are: tank diameter is 20 ft; tank height is 48 ft (to the edge of the roof); roof incline is 3/4 in. per foot; tank sidewalls are 3/16-in. carbon steel; insulation is 1½-in. fiberglass, on the sidewall only.

XYZ Terminal Co. does not have metering stations on the steam supply to individual tanks, and proposes to charge ABC Chemical for tank heating on the basis of calculated heat losses, using the conventional tables [1], and assuming a tank wall temperature of 50°F. The project engineer from ABC Chemical decided to investigate how XYZ's estimate would compare with the more elaborate one described in this article.

First, the engineer collected basic data on storage and climate. Oil shipments from the U.S. arrive at Port City approximately once a month, in 100,000-gal batches. Deliveries to local customers are made in 8,000-gal tanktrucks, three times a week on average. The typical variation in tank level over a 30-day period is known from experience.

The ambient temperature goes through a more complex cycle, of course. Within the primary cycle of 365 days, there are daily temperature variations. But in the seasonal cycle, heat supply is required only during the winter months, when temperatures fall well below 50°F.

Wind conditions at the storage site are not as well defined, and therefore much harder to predict. However, we can assume that the wind speed will hold constant for a short period of time, and calculate the heat loss for this unit period under a fixed set of conditions. The wind speed to be used must be based on the known probability distribution of wind speeds at the site.

The procedure for determining the annual heat loss consists of adding up the heat losses calculated for each unit period (which could be an hour, 12 hours, 24 hours, or 30 days, as appropriate). This example demonstrates the calculation of heat loss for only one unit period, of 12 hours, using an ambient temperature of

Data for ABC Chemical Co. example — Table II

Physical properties	Liquid	Air	Vapor*
Density, lb/ft^3	4.68	0.08	0.08
Specific heat, Btu/lb-°F	0.6	0.25	0.25
Viscosity, cP	40	0.007	0.007
Thermal conductivity, Btu/ft-h-°F	0.12	0.0151	0.015
Coefficient of volumetric expansion per °F	1 X 10^{-6}	0.002	0.002
Assumed fouling coefficients			
Dry wall	1,000 Btu/ft^{2}h-°F		
Wet wall	800		
Roof	1,000		
Bottom	500		
Thermal conductivities			
Metal walls	10 Btu/ft-h-°F		
Insulation	0.028		
Ground	0.80		
Surface emissivity			
Wall and roof	0.9		
Temperatures			
Vapor in tank	50°F		
Liquid in tank	55		
Outside air	35		
Ground	40		

*Since the liquid has low volatility, the vapor space is assumed to be mostly air.

Heat-transfer coefficients after first iteration — Table III

Coefficient	Dry wall	Wet wall	Roof	Bottom
h_{Vw}	0.5815	–	–	–
h_{Lw}	–	1.415	–	–
h_{Vr}	–	–	0.1537	–
h_{Lb}	–	–	–	1.105
h_G	–	–	–	0.102
h'_{Ar}	–	–	0.6635	–
h_{Ar}*	–	–	2.057	–
h'_{Aw}	0.51	0.51	–	–
h_{Aw}*	1.683	1.683	–	–
h_M	640	640	640	640
h_I	0.224	0.224	–	–
h_F	1,000	800	1,000	500
h_R	0.7565	0.7594	0.7651	–
U*	0.1516	0.1828	0.1457	0.0933

*For 10-mph wind

Second iteration yields closer temperature estimates — Table IV

Temperature	Iteration	Dry wall	Wet wall	Roof	Bottom
T_W (inside), °F	2	46.0	52.7	35.75	53.7
	1	42.5	45	42.5	47.5
T_{WS} (outside), °F	2	35.9	36.5	35.75	–
	1	38.75	40	42.5	–

35°F, a wind velocity of 10 mph, and a liquid level of 50%. The other data required are given in Table II. Note that the liquid temperature is controlled at 55°F to provide a 5°F margin of safety.

Since the Prandtl and Grashof numbers occur repeatedly in the film heat-transfer coefficient equations, and remain relatively unchanged for all the conditions of interest, let us first calculate their values. Thus, for the liquid phase:

$$N_{Gr} = L^3\rho^2 g\beta\, \Delta T/\mu^2 = 97.5\, L^3\, \Delta T$$
$$N_{Pr} = C_p\mu/k = 484$$

Similarly, for the vapor phase, $N_{Gr} = 1.90 \times 10^7 L^3\, \Delta T$, and $N_{Pr} = 0.28$. We can now calculate the individual film heat-transfer coefficients, using the appropriate L and ΔT values in the Grashof-number equations. This is an iterative process that requires initial estimates for wall and ground temperatures, plus wall temperatures.

Coefficient for vapor at wall (h_{Vw}). As an initial approximation, assume that the wall temperature is the average of the vapor and outside-air temperatures: $T_w = (50 + 35)/2 = 42.5$°F. Then find the Grashof number:

$$N_{Gr} = 1.90 \times 10^7(L - L_w)^3(T_V - T_w)$$
$$= 1.90 \times 10^7(24)^3(7.5)$$
$$= 1.97 \times 10^{12}$$

Employing Eq. 15, find the Nusselt number and then the coefficient ($k = 0.0151$, $L = 48$ ft, $L_w = 24$ ft):

$$N_{Nu} = 0.138(N_{Gr})^{0.36}(N_{Pr}^{0.175} - 0.55) = 921.1$$
$$h_{Vw} = (921.1)(k)/(L - L_w) = 0.581 \text{ Btu/ft}^2\text{h-°F}$$

Coefficient for liquid at the wall (h_{Lw}). Here, neither N_{Pr} nor $(N_{Gr}\, N_{Pr})$ falls within the range of the applicable correlations (Eq. 16,18). Let us try both, again using an average for T_w:

$$T_w = (T_L + T_A)/2 = 45\text{°F}$$
$$N_{Gr} = 97.47L^3(T_L - T_w) = 1.35 \times 10^7$$

Using Eq. 16 and 18, we get two estimates for the heat-transfer coefficient ($k = 0.12$, $N_{Pr} = 484$):

$$h_{Lw} = (0.495k/L_w)(N_{Gr}\, N_{Pr})^{0.25} = 0.704 \text{ Btu/ft}^2\text{h-°F}$$
$$h_{Lw} = (0.45k/L_w^{0.75})(N_{Gr}\, N_{Pr})^{0.25}$$
$$= 1.415 \text{ Btu/ft}^2\text{h-°F}$$

To be conservative, we use the higher value: $h_{Lw} = 1.415$ Btu/ft^{2}h-°F.

Coefficient for vapor at roof (h_{Vr}). We consider this a flat plate, with a diameter of 20 ft, and use Eq. 20, again with an average T_w of 42.5°F ($k = 0.0151$):

$$N_{Gr} = 1.9 \times 10^7 D^3(T_V - T_w) = 1.14 \times 10^{12}$$
$$h_{Vr} = (0.27k/D)(N_{Gr}\, N_{Pr})^{0.25} = 0.154 \text{ Btu/ft}^2\text{h-°F}$$

Coefficient for liquid at tank bottom (h_{Lb}). Assume that the ground temperature (T_G) is 5°F above ambient, and use an average of liquid and ground temperatures as a first approximation for the tank-bottom temperature:

$$T_w = (T_L + T_G)/2 = (T_L + T_A + 5)/2 = 47.5\text{°F}$$

Then, figure the Grashof number, and use Eq. 19 to get the coefficient:

$$N_{Gr} = 97.47D^3(T_L - T_w) = 5.85 \times 10^6$$
$$N_{Gr}N_{Pr} = 2.83 \times 10^9$$
$$h_{Lb} = 1.105 \text{ Btu/ft}^2\text{h-}^\circ\text{F}$$

Coefficient for outside air at roof (h'_{Ar}). Assume $T_{ws} = T_w$ since the roof is uninsulated, and get the coefficient for still air from Eq. 19:

$$N_{Gr} = 1.9 \times 10^7 D^3(T_{ws} - T_A) = 1.14 \times 10^{12}$$
$$h'_{Ar} = 0.663 \text{ Btu/ft}^2\text{h-}^\circ\text{F}$$

Coefficient for outside air at wall (h'_{Aw}). Assume that the temperature drop across the film is one-fourth of the drop from the inside fluid to the outside air (averaged for the wet and dry walls), and use Eq. 15 to find the coefficient:

$$\Delta T = 17.5/4 = 4.375^\circ\text{F}$$
$$N_{Gr} = 1.9 \times 10^7 L^3 \Delta T = 9.19 \times 10^{12}$$
$$h'_{Aw} = 0.51 \text{ Btu/ft}^2\text{h-}^\circ\text{F}$$

Conduction coefficients for ground, metal wall, and insulation (h_G, h_M and h_I). These are straightforward, from Eq. 21-23:

$$h_M = k_M/t_M = 640 \text{ Btu/ft}^2\text{h-}^\circ\text{F}$$
$$h_I = k_I/t_I = 0.224 \text{ Btu/ft}^2\text{h-}^\circ\text{F}$$
$$h_G = 8\,k_G/\pi D = 0.102 \text{ Btu/ft}^2\text{h-}^\circ\text{F}$$

Radiation coefficients for dry and wet sidewall, and roof (h_{Rd}, h_{Rw}, h_{Rr}). As for the outside-air film coefficients, assume that $T_{ws} = T_A + 0.25\ (T_{bulk} - T_A)$, where T_{bulk} is the temperature of the liquid or vapor inside the tank, if the surface is insulated. For the uninsulated roof, assume that $T_{ws} = T_A + 0.5(T_V - T_A)$. Then $T_{ws} = 38.75^\circ$F for the (insulated) dry sidewall, $T_{ws} = 40^\circ$F for the wet sidewall, and $T_{ws} = 42.5^\circ$F for the roof. Using Eq. 24, find the coefficient for each of the three cases:

$$h_{Rd} = 0.757 \text{ Btu/ft}^2\text{h-}^\circ\text{F}$$
$$h_{Rw} = 0.759 \text{ Btu/ft}^2\text{h-}^\circ\text{F}$$
$$h_{Rr} = 0.765 \text{ Btu/ft}^2\text{h-}^\circ\text{F}$$

Closing in on results

Table III summarizes the heat-transfer coefficients just calculated, including the corrections for wind—h'_{Aw} and h'_{Ar} are multiplied by 3.3 and 3.1, respectively, based on data for 10-mph wind in Fig. 2. Substituting these individual coefficients in Eq. 11-14, we obtain the U values listed in Table III.

What remains to be done? When we began the calculations, we assumed that the outside-wall temperatures were related to the bulk-fluid temperatures by:

$$T_w = T_A + 0.5(T_{bulk} - T_A) \text{ for uninsulated surfaces}$$
$$T_{ws} = T_A + 0.25(T_{bulk} - T_A) \text{ for insulated surfaces}$$

In order to calculate accurate coefficients for heat transfer, we must now obtain better estimates of these wall temperatures. This requires an iterative procedure that can be programmed and run on a computer.

Revised coefficients after second iteration **Table V**

Coefficient	Dry wall	Wet wall	Roof	Bottom
h_{Vw}	0.463	–	–	–
h_{Lw}	–	0.98	–	–
h_{Vr}	–	–	0.181	–
h_{Lb}	–	–	–	0.619
h_G	–	–	–	0.102
h'_{Ar}	–	–	0.31	–
h_{Ar}*	–	–	0.96	–
h'_{Aw}	0.317	0.317	–	–
h_{Aw}*	1.047	1.047	–	–
h_M	640	640	640	640
h_I	0.224	0.224	–	–
h_F	1,000	800	1,000	500
h_R	0.7500	0.7514	0.7500	–
U*	0.1392	0.1655	0.1636	0.0875

*For 10-mph wind

For dry wall, the rate of heat loss is given by all three of the following:

$$q_d = U_d A_d(T_V - T_A) \quad (25)$$
$$= h_{Vw} A_d(T_V - T_w) \quad (26)$$
$$= (h_{Rd} + h_{Aw})A_d(T_{ws} - T_A) \quad (27)$$

Solving Eq. 25 and 27 for T_{ws} yields:

$$T_{ws} = (U_d/(h_{Rd} + h_{Aw}))(T_V - T_A) + T_A \quad (28)$$

Similarly, solving Eq. 25 and 26 for T_w yields:

$$T_w = T_V - (U_d/h_{Vw})(T_V - T_A) \quad (29)$$

Using the same approach, now calculate T_w and T_{ws} for the wet wall, and T_w for the roof and bottom of the tank.

To find the correct wall temperatures, use the initial estimates of U and h values in Eq. 28 and 29 (and in the parallel equations for the other surfaces) to get new T_w and T_{ws} values. Table IV shows these temperatures after a second iteration. Using these new temperatures, recompute Grashof numbers, individual heat-transfer coefficients and overall coefficients, and then iterate again to get a new set of T_w and T_{ws} values. When the current and previous iteration's temperature estimates are the same (within a specified tolerance), the iteration is completed.

Table V lists the individual and overall coefficients after the second iteration. Although it is clear that additional iterations are needed, let us accept these values as sufficiently accurate for the present purpose. Then we can obtain the total heat-transfer rate (Q) by using the U values in Eq. 1-5 and summing. Table VI shows the calculated heat-transfer rates through each boundary, and the total rate. Note that the roof and bottom of the tank account for only slight heat loss, despite being uninsulated.

This, of course, is for the unit period of time, when wind speed is 10 mph, the tank is half full, and the air is 35°F. Table VII shows how the results of unit-period

Rate of heat transfer during unit period — Table VI

Surface	U, Btu/ft²h-°F	Area, ft²	ΔT, °F	q, Btu/h
Dry wall	0.1392	1,508	15	3,148.7
Wet wall	0.1655	1,508	20	4,991.5
Roof	0.1636	315	15	773.0
Bottom	0.0875	314	15	412.1
Total		3,645		9,325.3

Note: Total for 12-h period is 111,904 Btu

Summing losses for unit periods yields heat loss for 30 days — Table VII

Period	Liquid level, %	T_A, °F	Wind speed, mph	Heat loss, Btu
1	50	35	10	111,904
2	50	27	5	392,407
3	43	42	0	42,591
.	.	.	.	.
.	.	.	.	.
.	.	.	.	.
42	93	55	30	0
.	.	.	.	.
.	.	.	.	.
.	.	.	.	.
59	56	48	20	12,368
60	49	60	15	0
Total for 30-day period				8,389,050

heat losses can be tabulated and added to get the cumulative heat loss for a month or year. Of course, this requires climatic data and tank-level estimates for the overall time-period.

Comparison with other methods

Aerstin and Street [*1*] offer a very simple method for calculating heat loss from tanks. For a tank with 1.5 in. of sidewall insulation, and a wind speed of 10 mph, the recommended overall U (based on $k = 0.019$ for the insulation) is 0.14 for $\Delta T = 60°F$ and 0.14 for $\Delta T = 100°F$. Adjusting these values for $k = 0.028$ and $\Delta T = 17°F$, as in our example, yields an overall U of 0.206 Btu/ft²h-°F. The total exposed surface is 3,331 ft² (tank bottom not included), and thus the overall rate of heat transfer by their method is:

$$Q = 0.206 \times 3{,}331 \times 17 = 11{,}666 \text{ Btu/h}$$

This compares with a heat loss of 8,913 Btu/h (for the exposed surface) calculated by the procedure of this article—see Table VI. Thus their method yields a result 31% too high in this case.

Stuhlbarg [*10*] takes an approach similar to that proposed here, but his method differs in how the outside tankwall film coefficient is computed. Stuhlbarg recommends the use of a manufacturer's data table, and does not explicitly distinguish between the bulk liquid temperature and the outside-wall surface temperature in calculating the proper heat-transfer coefficient.

The algebraic method of Hughes and Deumaga [5] resembles the one presented in this article in many ways. But it does not recognize differences between liquid and vapor temperatures inside the tank, nor does it account for the interaction between ΔT and wind speed in calculating a wind-enhancement factor. Finally, even though their procedure requires iteration, the focus of the iterative efforts is to get better estimates of fluid properties, not tankwall temperatures.

Conclusions

Our engineer at ABC Chemical was able to negotiate a significant reduction in the heating charges proposed by the XYZ Terminal Co., which had used a shortcut method for its estimate, because the procedure presented here is rational and defensible. A rigorous solution of the iterations can easily be reached on a digital computer or even a programmable calculator, and the effort pays off in better design or operation criteria.

Mark Lipowicz, Editor

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