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Energy Efficient Process Heating: Insulation and Thermal Mass

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ABSTRACT

Open tanks and exterior surfaces of process heating equipment lose heat to the surroundings via convection, radiation, and/or evaporation. A practical way of reducing heat loss is by insulating or covering the surfaces. This paper presents methods to quantify heat loss and energy savings from insulating hot surfaces and open tanks. The methods include radiation and evaporation losses, which are ignored by simplified methods. In addition, thermal mass, such as refractory, conveyor and racking equipment, acts as a heat sink and increases heating energy use in process heating applications. This paper presents lumped capacitance and finite-difference methods for estimating heat loss to thermal mass, and savings from reducing this loss.

The methods described above have been incorporated in free software, and are demonstrated using case study examples. The examples demonstrate the magnitude of the potential error from using simplified methods.

INTRODUCTION

This paper discusses and illustrates methods to quantify savings from insulating hot surfaces, covering open tanks and reducing thermal mass. A companion paper, Energy Efficiency Process Heating: Managing Air Flow, discusses improving process heating energy efficiency by managing air flow.

Process heating systems lose heat through the systems' shell. Heat is transferred through the shell by conduction, and then to the surroundings by convection and radiation. This heat loss can be reduced by insulating hot surfaces. Several methods exist to quantify the heat loss reduction from insulating hot surfaces. Most methods consider the thermal resistance of the insulation, but many do not consider the reduction in convection coefficient due to a cooler surface and do not explicitly consider radiation heat loss. Because of these deficiencies, simplified methods frequently underestimate the benefits of adding insulation. This paper describes a method to calculate energy savings from adding insulation, which increases the accuracy of the calculation by explicitly accounting for the reduction in convection due to decreased surface temperatures and radiation heat loss.

Even more heat is lost from a process heating system if hot liquids are exposed to ambient air, such as in from a heated rinse tank, because evaporation heat loss takes place in addition to convection and radiation. Evaporation heat loss from uncovered tanks is between 2 and 6 times greater than convection and radiation heat loss combined. When an open heated tank is covered

with a lid or insulating floats, heat loss by convection, radiation, and most prominently evaporation is reduced. This paper develops a method for calculating total heat loss from open tanks, and the benefits of covering or adding insulation to the tanks.

In most process heating systems, some of the heat that could be absorbed by the product is absorbed by the structure of the oven/furnace or by the machinery used to transport the material into and out of the oven/furnace. We define the effective thermal mass of a process heating system as the mass capacitance that absorbs heat that could otherwise be absorbed by the product. The quantity of heat loss is dependent on the quantity of thermal mass and the frequency that the thermal mass is heated. In continuous-flow heating systems, the thermal mass may be conveyors that enter the system at one temperature and exit the system at a higher temperature. In batch heating systems, the thermal mass includes structures to transport the product into and from the system, and the interior surfaces of the oven/furnace. Reducing the quantity of energy lost to thermal mass in a process heating system saves energy. This paper develops a method to calculate savings from reducing energy lost to thermal mass.

INSULATING HOT SURFACES

Savings Calculation Methodology

The following procedure describes how to calculate energy savings from insulating hot surfaces, while explicitly taking radiation and the dependence of the convection coefficient on geometry and surface temperature into account. The required input variables for this procedure are easily measured in the field. They are the area, temperature, emissivity and geometry of the hot surface, the temperature of the air and surroundings, and the temperature inside the oven, furnace or pipe.

To calculate heat loss savings, the heat loss from both the uninsulated surface and from the surface with the proposed insulation must be calculated. The savings from adding insulation are difference between the uninsulated and insulated heat loss.

$$Q_{\text{savings}} = Q_{\text{uninsulated}} - Q_{\text{insulated}} \quad (1)$$

Hot surfaces lose heat to the surroundings via convection and radiation. The equation for heat loss, Q , to the surroundings at T_a , from a hot surface at T_s , with area A is:

$$Q_{\text{uninsulated}} = h \cdot A \cdot (T_s - T_a) + \sigma \cdot A \cdot \epsilon \cdot (T_s^4 - T_a^4) \quad (2)$$

where h is the convection coefficient, σ is the Stefan-Boltzman constant (0.1714×10^{-8} Btu/ft²-hr-R⁴), ϵ is the emissivity of the surface. The emissivity of most materials is a function of their color. Very light and shiny surfaces have emissivities of about 0.1; dark surfaces have emissivities of about 0.9.

The value of the convection coefficient, h , depends on whether the air movement over the hot surface is forced, such as by a fan, or natural due to the buoyancy of warm air next to the surface compared to the cooler surrounding air. This paper considers only natural convection. For warm surfaces, the natural convection coefficient is about 1.5 Btu/ft²-hr-F (Mitchell, 1983). However,

as the surface temperature increases, so does the buoyancy effect and convection coefficient. To include this effect, various relations have been developed. This method uses the relations from the ASHRAE handbook (ASHRAE Fundamentals, 1989). In this method, the value of the convection coefficient can be approximated as a function of the orientation of the surface and the temperature difference between the surface and the surrounding air. The appropriate relation depends on whether the air flow is laminar or turbulent. Dimensional approximations for determining whether the flow is laminar or turbulent are shown in Equation 3. In these relations, L is the characteristic length (ft) and ΔT is temperature difference between the surface and the surrounding air (F).

$$\begin{aligned} \text{Laminar: } L^3 \Delta T < 63 \\ \text{Turbulent: } L^3 \Delta T > 63 \end{aligned} \quad (3)$$

After the nature of the flow is determined, the convection coefficient can be estimated using the relations in Equation 4 (ASHRAE Fundamentals, 1989). In these relations, D is characteristic diameter (ft), B is tilt angle of the surface from horizontal, and h is convection coefficient (Btu/hr-ft²-F).

$$\begin{aligned} \text{Horizontal Surfaces Losing Heat Upwards: } h_{\text{lam}} &= 0.27 \cdot (\Delta T / L)^{0.25}; \quad h_{\text{tur}} = 0.22 \cdot (\Delta T)^{0.33} \\ \text{Tilted/Vertical Surfaces: } h_{\text{lam}} &= 0.29 \cdot [\Delta T \cdot (\sin B) / L]^{0.25}; \quad h_{\text{tur}} = 0.19 \cdot [\Delta T \cdot (\sin B) / L]^{0.33} \\ \text{Horizontal Pipes and Cylinders: } h_{\text{lam}} &= 0.27 \cdot (\Delta T / D)^{0.25}; \quad h_{\text{tur}} = 0.18 \cdot (\Delta T)^{0.33} \end{aligned} \quad (4)$$

Using these relations, Equation 2 can be solved for $Q_{\text{uninsulated}}$ to estimate the current heat loss.

Similarly, heat loss from the insulated surface can be calculated from:

$$Q_{\text{insulated}} = h \cdot A \cdot (T_i - T_a) + \sigma \cdot A \cdot \varepsilon \cdot (T_i^4 - T_a^4) \quad (5)$$

where T_i is the temperature of the outside surface of the insulation. Unfortunately, in Equation 5, the values of T_i and h are not known. To determine T_i and h, the first step is to determine the thermal resistance of the current wall, R_c , based on the temperature of the fluid inside the heating system, T_f , and the current surface temperature T_s . Thermal resistance of the current wall includes both the conduction thermal resistance through the wall and the convection thermal resistance at the wall's inner surface.

$$Q_{\text{uninsulated}} = A \cdot (T_f - T_s) / R_c \quad (6)$$

Next, an equation can be written from a steady-state energy balance on the surface of the insulation:

$$h \cdot A \cdot (T_i - T_a) + \sigma \cdot A \cdot \varepsilon \cdot (T_i^4 - T_a^4) = A \cdot (T_f - T_i) / (R_c + R_i) \quad (7)$$

where R_i is the thermal resistance of the insulation. The relations for convection coefficient as a function of the temperature difference between T_i and T_a form a second equation. Thus, this system has two equations (Equation 4 and Equation 7) and two unknowns and can be solved to determine T_i and h.

One of the easiest ways to solve this system of nonlinear equations is successive substitution. In the successive-substitution method, an initial value for T_i is estimated and used to determine the convection coefficient h . The value for h is then substituted into Equation 4 to determine a new value of T_i . This procedure is repeated until the value of T_i varies by only a small amount after each iteration. The final values of T_i and h can then be substituted into Equation 5 to find $Q_{insulated}$. The heat loss savings, Q_{sav} is the difference between $Q_{uninsulated}$ and $Q_{insulated}$. This method has been incorporated into the free software HeatSim (www.engr.udayton.edu/udiac).

The energy savings, $E_{savings}$, are the heat loss savings, $Q_{savings}$, divided by system efficiency, η .

$$E_{savings} = Q_{savings} / \eta \quad (8)$$

Simplified Method

Many analyses use simplified methods to calculate savings from insulating hot surfaces. Most of these simplified methods assume a constant convection coefficient and do not explicitly consider radiation. For example, the simplified method below involves calculating a thermal resistance of a system's shell, R_{shell} , by approximating a convection coefficient between the outer shell surface and ambient air, h , and finding the new shell thermal resistance when a layer of insulation is added. The heat lost from the uninsulated surface, $Q_{uninsulated}$, is the heat transferred between the inner fluid and the outer shell surface, which also equals the heat flux between the outer shell surface and ambient air.

$$Q_{uninsulated} = (T_f - T_s) \cdot A / R_{shell} = h \cdot A \cdot (T_s - T_a) \quad (9)$$

The thermal resistance of a system's shell, R_{shell} , can be calculated in Equation 9. The total thermal resistance is the sum of the thermal resistance of the shell and the thermal resistance of the convection coefficient:

$$R_{current} = R_{shell} + 1 / h \quad (10)$$

Thus, the uninsulated heat loss is:

$$Q_{uninsulated} = (T_f - T_a) \cdot A / R_{current} \quad (11)$$

The thermal resistance of the insulated system would be the sum of the thermal resistance of the shell, R_{shell} , and the insulation, R_i . Thus, the heat lost from the insulated surface, $Q_{insulated}$, would be written as:

$$Q_{insulated} = (T_f - T_a) \cdot A / (R_{current} + R_i) = (T_f - T_a) \cdot A / (R_{new}) \quad (12)$$

Equation 1 could then be used to calculate heat loss savings, $Q_{savings}$. Or, more directly, the heat loss savings would be:

$$Q_{savings} = (T_f - T_a) \cdot A / (1 / R_{current} - 1 / R_{new}) \quad (13)$$

Other Methods/Software

3E Plus (DOE, 2003) is a widely-available simulation program that calculates savings from insulating hot surfaces. The method used to calculate heat loss is not apparent from the software's Help features. However, the primary input parameters in 3E Plus are process temperature (which is analogous to the fluid temperature, T_f , in this procedure), ambient temperature, wind speed, insulation material used, and surface geometry. This indicates that a principle difference between 3E Plus and the method described here is that 3E Plus calculates surface temperature from the value of insulation input by the user, while the method described here directly asks for surface temperature as an input variable. If the surface temperature can be accurately measured, which it typically can using an IR thermometer, then the method described here will be more accurate than the 3E Plus method.

Comparison of Proposed and Simplified Methods

Consider a rectangular-shaped heat treat oven with dimensions of 10 ft x 10 ft x 10 ft, an inside air temperature of 1,600 F, an external surface temperature of 250 F, and an outer surface emissivity of 0.9. The combustion efficiency of the oven is 50% (Carpenter and Kissock, 2005). The oven is located in a room with air and surroundings at 70 F. The oven is to be covered with a layer of 2-inch thick insulation with a thermal conductivity of 0.44 Btu-in/hr-ft²-F.

The overall weighted convection coefficient, including both the vertical sides and the horizontal top of the oven was found to be 2.36 Btu/hr-ft²-F using Equations 3 and 4. The energy savings calculations using the proposed method are as follows. The uninsulated heat loss is (Equation 2):

$$Q_{\text{uninsulated}} = h \cdot A \cdot (T_s - T_a) + \sigma \cdot A \cdot \varepsilon \cdot (T_s^4 - T_a^4)$$
$$Q_{\text{uninsulated}} = 2.36 \cdot 500 \cdot (250 - 70) + 0.1714 \times 10^{-8} \cdot 500 \cdot 0.9 \cdot (250^4 - 70^4) = 233,031 \text{ Btu/hr}$$

The current thermal resistance of the oven including the internal convection coefficient and structure of the oven, but not external convection coefficient would be (Equation 6):

$$R_c = A \cdot (T_f - T_s) / Q_{\text{uninsulated}}$$
$$R_c = 500 \cdot (1,600 - 250) / 233,031 = 2.90 \text{ hr-ft}^2\text{-F/Btu}$$

An energy balance on the surface (Equation 7) gives:

$$h \cdot A \cdot (T_i - T_a) + \sigma \cdot A \cdot \varepsilon \cdot (T_i^4 - T_a^4) = A \cdot (T_f - T_i) / (R_c + R_i)$$
$$h \cdot 500 \cdot (T_i - 70) + 0.1714 \times 10^{-8} \cdot 500 \cdot 0.9 \cdot (T_i^4 - 70^4) = 500 \cdot (1,600 - T_i) / (2.90 + 4.55)$$

Combining the above equation with Equations 3 and 4, the values for convection coefficient and insulation surface temperature are $h = 2.06$ Btu/hr-ft²-F and $T_i = 163$ F. Thus the heat loss from the insulated surface would be (Equation 5):

$$Q_{\text{insulated}} = h \cdot A \cdot (T_i - T_a) + \sigma \cdot A \cdot \varepsilon \cdot (T_i^4 - T_a^4)$$
$$Q_{\text{insulated}} = 2.06 \cdot 500 \cdot (1,600 - 70) + 0.1714 \times 10^{-8} \cdot 500 \cdot 0.9 \cdot (163^4 - 70^4) = 96,546 \text{ Btu/hr}$$

The heat loss and energy savings from adding insulation would be (Equation 1, Equation 8):

$$Q_{\text{savings}} = Q_{\text{uninsulated}} - Q_{\text{insulated}} = 233,031 \text{ Btu/hr} - 96,546 \text{ Btu/hr} = 136,485 \text{ Btu/hr}$$

$$E_{\text{savings}} = 136,485 \text{ Btu/hr} / 50\% = 272,970 \text{ Btu/hr}$$

The HeatSim output screen in Figure 1 shows the current heat loss from the furnace's shell, the heat loss if it was insulated, and the heat loss savings using the proposed method.

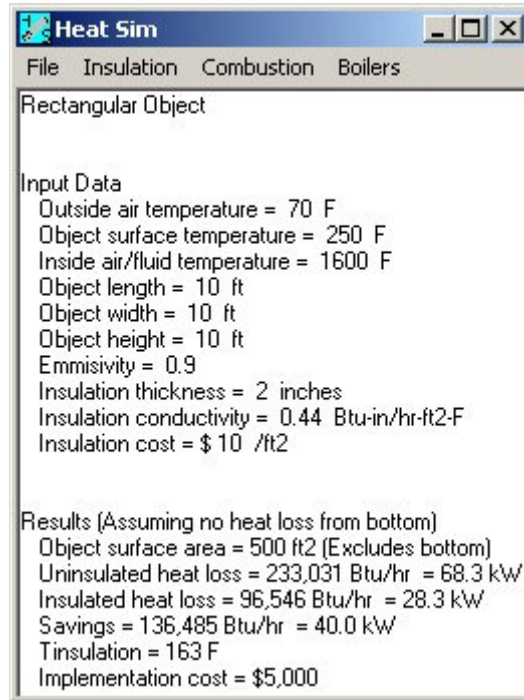


Figure 1. HeatSim output screen for insulation application

Using the simplified method and assuming the convection coefficient, including radiation, is 1.5 Btu/ft²-hr-F (Mitchell, 1983), the energy savings from insulating the oven would be:

$$R_{\text{shell}} = (T_f - T_s) / [h \cdot (T_s - T_a)] = (1600 - 250) / [1.5 (250 - 70)] = 5.0 \text{ Btu/ft-hr-F}$$

$$R_{\text{current}} = R_{\text{shell}} + 1 / h = 5.0 + 1 / 1.5 = 5.67 \text{ Btu/ft-hr-F}$$

$$R_i = t / k = 2 / 0.44 = 4.55 \text{ Btu/ft-hr-F}$$

$$R_{\text{new}} = R_{\text{current}} + R_i = 5.67 + 4.55 = 10.22 \text{ Btu/ft-hr-F}$$

$$Q_{\text{savings}} = (T_f - T_a) \cdot A / (1/R_{\text{current}} - 1/R_{\text{new}}) = (1,600 - 70) 500 (1/5.67 - 1/10.22) = 60,067 \text{ Btu/hr}$$

$$E_{\text{savings}} = 60,067 \text{ Btu/hr} / 50\% = 120,134 \text{ Btu/hr}$$

This is only 44% of the result obtained from the more thorough method developed here and incorporated into HeatSim. The reason that the simplified method results in fewer savings is because it does not account for radiation effects and the dependence of the convection coefficient on the temperature of the surface. The magnitude of the difference in savings between the proposed and simplified methods indicates the importance of using the proposed method.

COVERING HEATED TANKS

Savings Calculation Methodology

The total heat loss from the top of an open heated liquid tank is the sum of convection, radiation and evaporation heat loss from the tank.

$$Q_{\text{tank}} = Q_{\text{conv}} + Q_{\text{rad}} + Q_{\text{evap}} \quad (14)$$

The method for quantifying these losses follows from Incropera and DeWitt (1985).

The value of the convection coefficient, h , for free convection from the top of a hot surface can be estimated by calculating the thermal conductivity, k , kinematic viscosity, ν , Prandtl number, Pr , and Raleigh number, Ra , for the appropriate film temperature, T_f ; then using the appropriate relation for the Nusselt number, Nu , depending on whether the flow is laminar or turbulent. The film temperature is assumed to be the average temperature between the water temperature inside the tank, T_w , and ambient air temperature, T_a .

$$T_f = (T_w + T_a) / 2 \quad (15)$$

The thermal conductivity, k , Prandtl number, Pr , and kinematic viscosity, ν , of air near the surface is obtained from the following regressions derived from gas property tables.

$$k \text{ (W/m-K)} = [23.9886 + 0.0782 \cdot T_f \text{ (C)}] \cdot 10^{-3} \quad (16)$$

$$Pr = 0.716956 - 0.000228 \cdot T_f \text{ (C)} \quad (17)$$

$$\nu \text{ (m}^2\text{/s)} = [13.90552 + 0.09424 \cdot T_f \text{ (C)}] \cdot 10^{-6} \quad (18)$$

The Raleigh number, Ra , can be calculated as:

$$Ra = [g \cdot (T_w - T_a) \cdot L^3] / [T_f \text{ (K)} \cdot \nu^2] \quad (19)$$

where g is the acceleration due to gravity and L is the characteristic length. Characteristic length, L , is calculated from tank length, l , and tank width, w , as:

$$L = (l \cdot w) / [2 \cdot (l + w)] \quad (20)$$

The convection coefficient, h , can be calculated with the following equation.

$$\begin{aligned} h &= (k \cdot 0.54 \cdot Ra^{1/4}) / L && (Ra < 10^7) \\ h &= (k \cdot 0.15 \cdot Ra^{1/3}) / L && (Ra > 10^7) \end{aligned} \quad (21)$$

Convection heat loss, Q_{conv} , can be calculated as:

$$Q_{\text{conv}} = h \cdot l \cdot w \cdot (T_w - T_a) \quad (22)$$

Radiation heat loss can be calculated using the terms of Stefan-Boltzmann constant, $\sigma = 5.67 \cdot 10^{-8} \text{ W/m}^2\text{-K}^4$, emissivity, ϵ (about 0.96 for water), water temperature, T_w , and air temperature, T_a .

$$Q_{\text{rad}} = \varepsilon \cdot \sigma \cdot l \cdot w \cdot [T_w^4 \text{ (K)} - T_a^4 \text{ (K)}] \quad (23)$$

The binary diffusion coefficient, D_{AB} , for water at atmospheric pressure is $0.26 \cdot 10^{-4} \text{ m}^2/\text{s}$. From the steam tables, the regression equations for specific volume of water vapor, v , at the temperature of water in the tank, T_w , and ambient air temperature, T_a , are as follows.

$$v_{T_w} \text{ (m}^3/\text{kg)} = e^{[5.285152 - 0.06458115 \cdot T_w \text{ (C)} + 0.000172321 \cdot T_w^2 \text{ (C)}]} \quad (24)$$

$$v_{T_a} \text{ (m}^3/\text{kg)} = e^{[5.285152 - 0.06458115 \cdot T_a \text{ (C)} + 0.000172321 \cdot T_a^2 \text{ (C)}]} \quad (25)$$

The mass transfer coefficient, h_m , is calculated as follows.

$$h_m = h \cdot D_{AB} \cdot [v / (Pr \cdot D_{AB})]^{1/3} \quad (26)$$

The mass of water evaporating from the tank, m_w , is calculated as:

$$m_w = h_m \cdot (l \cdot w) \cdot [(1 / v_{T_w}) - (rh / v_{T_a})] \quad (27)$$

where rh is the relative humidity of ambient air. From the steam tables, the regression equations for latent heat of vaporization, h_{fg} , at the temperature of water in the tank, T_w , is:

$$h_{fg} \text{ (kJ/kg)} = 2,503.863 - 2.443774 \cdot T_w \text{ (C)} \quad (28)$$

The heat lost due to evaporation, Q_{evap} , is the product of water mass evaporating from the tank, m_w , and latent heat of vaporization, h_{fg} .

$$Q_{\text{evap}} = m_w \cdot h_{fg} \quad (29)$$

A common practice in industry is to cover heated open tanks with spherical insulation floats. The surface area covered by a spherical insulation float with a diameter, dia , is:

$$A_{\text{float}} = \pi \cdot \text{dia}^2 / 4 \quad (30)$$

The surface area covered by a square with a side of length dia is:

$$A_{\text{square}} = \text{dia}^2 \quad (31)$$

Thus the fraction of surface area, f_{float} , covered by spherical floats in an open tank is about:

$$f_{\text{float}} = A_{\text{float}} / A_{\text{square}} = 0.79 \quad (32)$$

When insulation floats are placed on the top of the tank, we assume no evaporative heat is lost from the covered area. We assume that heat loss from the uncovered area is the same as without floats.

To calculate the convection and radiation heat loss from the surface area of the floats, float surface temperature, T_{fs} , and convection coefficient, h_{fs} , on the surface must be found. To do so, an energy balance can be applied to the floats in which the sum of the radiation and convection energy flux from the float surface equals the conduction heat transfer flux through the float. The conduction flux through the floats is the difference between the water temperature, T_w , and float surface temperature, T_{fs} , divided by the thermal resistance of the floats, R_{float} . Equation 33 demonstrates this energy balance.

$$Q'_{conv} + Q'_{rad} = Q'_{cond}$$

$$h_{fs} \cdot (T_{fs} - T_a) + \varepsilon \cdot \sigma \cdot [T_{fs}^4 \text{ (K)} - T_a^4 \text{ (K)}] = (T_w - T_{fs}) / R_{float} \quad (33)$$

HeatSim solves Equation 33 by choosing values for T_{fs} until the equation is balanced. HeatSim uses Equations 15 through 21 to solve h_{fs} for each chosen value of T_{fs} . Equation 34 calculates the heat lost by the tank when insulated with floats, $Q_{tank,ins}$. This equation accounts for the heat lost by both the area covered with floats and the area not covered with floats.

$$Q_{tank,ins} = (Q_{conv} + Q_{rad} + Q_{evap}) \cdot (1 - f_{float}) + \{h_{fs} \cdot (T_{fs} - T_a) + \varepsilon \cdot \sigma \cdot [T_{fs}^4 \text{ (K)} - T_a^4 \text{ (K)}]\} \cdot (1 \cdot w \cdot f_{float}) \quad (34)$$

The heat loss savings, $Q_{savings}$, is the difference between the heat lost by the tank when uninsulated, Q_{tank} , and when insulated with floats, $Q_{tank,ins}$.

$$Q_{savings} = Q_{tank} - Q_{tank,ins} \quad (35)$$

Savings Example

For example, consider an open tank of dimensions 6 ft x 2 ft holding 158 F water. The ambient air surrounding the tank has a temperature of 50 F and a relative humidity of 50%. The tank is to be covered with balls with an R-value of 3 hr-ft²-F/Btu-in and a thickness (diameter) of 1.5 inches. The HeatSim output screen in Figure 2 shows that the current heat lost from the tank is 13,807 Btu/hr, and would reduce to 3,321 Btu/hr if covered with insulation floats; which is a savings of about 10,485 Btu/hr, or 76%. In addition, the current water lost from the tank is 10.8 lb/hr, and would reduce to 2.2 lb/hr; which is a savings of about 8.6 lb/hr, or 80%.

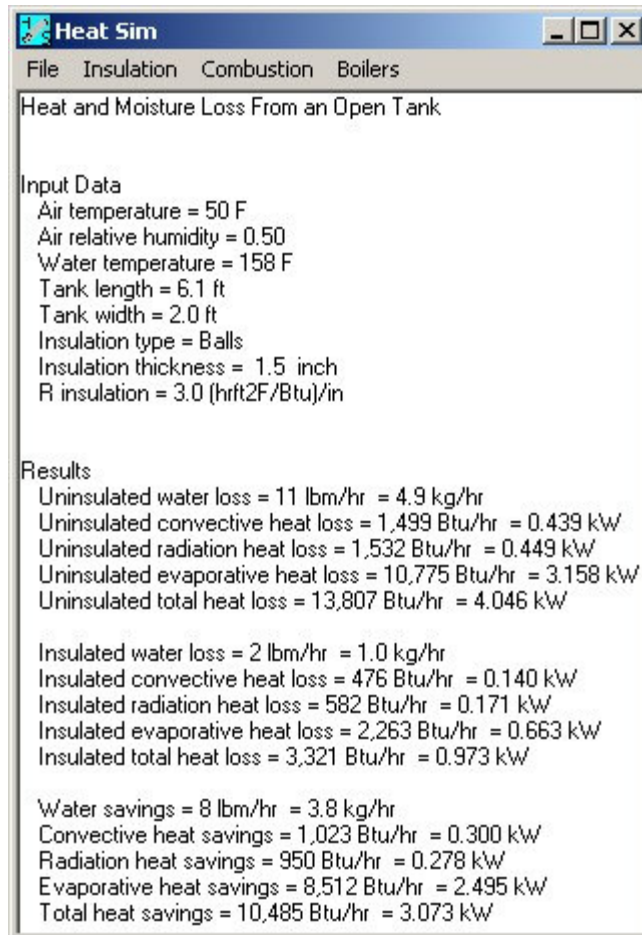


Figure 2. HeatSim output screen for covering open tank with insulation balls

Comparison of HeatSim and Experimental Results

We compared the above HeatSim results to the following experimental results (Euro-Matic Plastics Inc.). These results indicate that HeatSim's estimates of energy and liquid loss are slightly conservative, but are certainly close enough for estimating savings. Some of the deviation between HeatSim and experimental results may be attributed to the difference between the ambient conditions in the experiment, which were not published, and the ambient conditions assumed in the HeatSim simulation.

Table 1. Comparison of HeatSim and experimental results

	Experimental Results	HeatSim Results
<i>Open Tank</i>		
Energy loss (kW)	4.61	4.05
Liquid loss (kg/h)	5.52	4.90
<i>With Balls</i>		
Energy loss (kW)	1.31	0.97
Liquid loss (kg/h)	0.46	1.00
<i>Percent Reduction</i>		
Energy loss	72%	76%
Liquid loss	92%	80%

REDUCING THERMAL MASS

Continuous Process Systems

In continuous process heating systems, conveyors or other machinery that transport the material through the oven/furnace carry heat out of the oven/furnace and contribute to the heating load. Although the interior walls and ceiling also absorb heat, they remain warm as long as the oven/furnace is in operation; thus, the quantity of heat absorbed by these components over time is typically negligible compared to other types of heat loss.

Savings Calculation Methodology

Conveyors in continuous heating systems often run continuously, even when not loaded with parts. If so, a simple measure to reduce energy heat loss via thermal mass is to slow or stop the conveyor when not loaded. Another simple savings measure is to decrease the thermal mass of the conveyor, if possible, by using a lighter material or narrowing the width of a conveyor belt.

The energy lost to a conveyor in a continuous heating system, Q_{conv} , is the product of conveyor velocity, V , mass per linear length, m , specific heat of conveyor material, $C_{p\text{conv}}$, and the temperature difference of the conveyor leaving, $T_{\text{conv}2}$, and entering, $T_{\text{conv}1}$ the system.

$$Q_{\text{conv}} = V \cdot m \cdot C_{p\text{conv}} \cdot (T_{\text{conv}2} - T_{\text{conv}1}) \quad (36)$$

When the conveyor moves slowly, the temperature of the conveyor leaving the system, $T_{\text{conv}2}$, frequently approaches the interior temperature of the oven/furnace, and the temperature of the conveyor entering the system, $T_{\text{conv}1}$, approaches room temperature. If conveyor velocity, V , or mass, m , were reduced, the energy savings would be the difference between energy absorbed by the conveyor before and after the change.

Savings Example

A brazing oven with a set-point temperature of 1,900 F has a stainless steel conveyor belt weighing 5 lbs/ft traveling 42 ft per hour. The specific heat of stainless steel is 0.12 Btu/lb-F.

The conveyor was loaded with parts during about 30% of oven operating hours. If the conveyor was slowed to 18 ft per minute when not loaded, total oven energy savings would be about:

$$Q_{\text{conv},1} = V_1 \cdot m \cdot C_{p_{\text{conv}}} \cdot (T_{\text{conv}2} - T_{\text{conv}1}) = 42 \cdot 5 \cdot 0.12 \cdot (1900 - 70) = 46,116 \text{ Btu/hr}$$

$$Q_{\text{conv},2} (\text{avg}) = [V_1 \cdot m \cdot C_{p_{\text{conv}}} \cdot (T_{\text{conv}2} - T_{\text{conv}1})] \cdot 30\% + [V_2 \cdot m \cdot C_{p_{\text{conv}}} \cdot (T_{\text{conv}2} - T_{\text{conv}1})] \cdot 70\%$$

$$Q_{\text{conv},2} (\text{avg}) = [42 \cdot 5 \cdot 0.12 \cdot (1900 - 70)] \cdot 30\% + [18 \cdot 5 \cdot 0.12 \cdot (1900 - 70)] \cdot 70\% = 27,670 \text{ Btu/hr}$$

$$Q_{\text{conv},\text{sav}} = 46,116 \text{ Btu/hr} - 27,670 \text{ Btu/hr} = 18,446 \text{ Btu/hr}$$

This is a savings of about 40% of the energy lost to the conveyor.

Batch Systems

Unlike continuous systems, the interior structure of batch ovens and furnaces absorb energy during the heating cycle. Carts and racking that hold or transport the product also absorb energy. If the heat absorbing material is highly conductive, such as metal, the temperature is approximately uniform throughout the component. However, if the heat absorbing material is less conductive, such as refractory, the temperature decreases from the outside to the inside of the material. Finite-difference models can determine the temperature profile at the end of the cycle. The energy absorbed by the mass can then be determined from this profile. Energy savings can be realized if the quantity of thermal mass can be reduced.

Savings Calculation Methodology

To demonstrate the finite-difference method, consider a layer of fire brick that is exposed to interior of an oven on one side and insulated on the other (Figure 3). The thermal mass is divided into eight layers. Each layer is called a node and is assumed to have a uniform temperature.

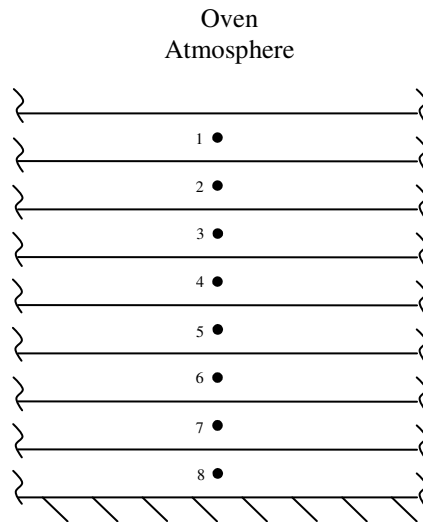


Figure 3. Layer of thermal mass with temperature nodes

The thickness of each node, dx , is equal to the thickness of the entire layer, x , divided by 8.

$$dx = x / 8 \quad (37)$$

The thermal resistance between the oven atmosphere and node 1, R_1 , is the sum of the convection and conduction thermal resistances. The convection thermal resistance is the inverse of the product of convection coefficient, h , and the surface area, A . The conduction thermal resistance is the distance between the center of node 1 and the surface, divided by thermal conductivity, k , and cross-sectional area, A .

$$R_1 = 1 / (h \cdot A) + dx / (2 \cdot k \cdot A) \quad (38)$$

The thermal resistance between each consecutive node is the distance between each node, dx , divided by thermal conductivity, k , and the cross-sectional area, A . If material properties of the thermal mass is constant throughout, the thermal resistances between each consecutive node are equal.

$$R_2 = R_3 = R_4 = R_5 = R_6 = R_7 = R_8 = dx / (k \cdot A) \quad (39)$$

From the First Law of Thermodynamics, the energy into each node, E_{in} , minus the energy out of each node, E_{out} , equals the change of energy stored in each node, ΔE_{store} .

$$E_{in} - E_{out} = \Delta E_{store} \quad (40)$$

Figure 4 demonstrates the interaction between a node and the nodes above and below it in terms of temperatures, T , thermal resistances, R , and energy, E .

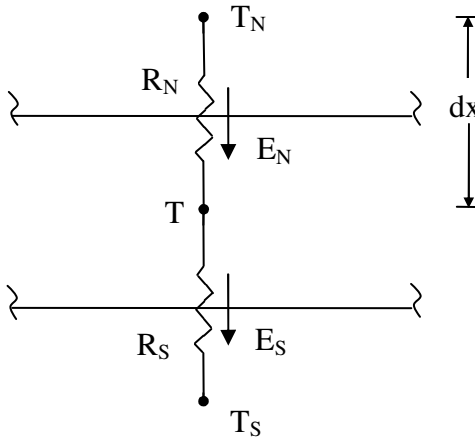


Figure 4. Finite difference analysis on temperature node

The E_{in} and E_{out} terms in Equation 40 can be rewritten using the terms in Figure 4. The change in energy stored at a node, ΔE_{store} , is the product of distance between the nodes, dx , cross-sectional area, A , material density, ρ , material specific heat, C_p , and the difference between node

temperature after the iterative step, T' , and current node temperature, T , all divided by the time period of iteration, dt . Thus, Equation 40 can be rewritten as:

$$(T_N - T) / R_N - (T - T_S) / R_S = [dx \cdot A \cdot \rho \cdot Cp \cdot (T' - T)] / dt \quad (41)$$

Rearranging Equation 41, the temperature after each iterative step, T' , can be calculated.

$$T' = dt \cdot (T_N - T) / (R_N \cdot dx \cdot A \cdot \rho \cdot Cp) - dt \cdot (T - T_S) / (R_S \cdot dx \cdot A \cdot \rho \cdot Cp) + T \quad (42)$$

When T' is calculated, it can be inserted into the T term in Equation 42 to calculate the next T' value in the following time iteration. This step must be repeated until enough time iterations have been performed to equal the time duration of a cycle. The temperatures of each node at the end of the cycle are calculated by performing all time iterations.

The total energy absorbed by the thermal mass during the cycle, Q_{TM} , can be calculated by summing the energy at all nodes relative to their energy before the cycle. This calculation involves each node temperature at the end of the cycle, T_f , and at the beginning of the cycle, T_i . Equation 43 calculates the total energy absorbed by the thermal mass during the cycle, Q_{TM} .

$$Q_{TM} = \Sigma dx \cdot A \cdot \rho \cdot Cp \cdot (T_f - T_i) \quad (43)$$

Savings Example

Consider a heat treat oven whose floor is mounted on rails so that it can be sled into and out of the oven. The floor is 100 ft² in area, and consists of fire bricks on top of a thick layer of insulation. The bricks are laid vertically such that the firebrick floor is about eight inches thick. The density of firebrick is about 36 lb/ft³, the specific heat is about 0.25 Btu/lb-F, and the thermal conductivity is about 1.2 Btu-in/ft²-hr-F. The heat treat cycle consists of raising the temperature to about 1,700 F for four hours, then decreasing the oven temperature by about 50 F per hour until the oven reaches 400 F.

As an energy-savings measure, the 8-inch thick firebrick is replaced with 4-inch firebrick. Using the methodology shown above, the temperature profile of 8-inch firebrick and 4-inch firebrick at the conclusion of the heating cycle is shown in Table 2.

Table 2. Temperature profile of 8-inch and 4-inch thick firebrick after heat treat cycle

	8-in thick	4-in thick
T1	484 F	477 F
T2	591 F	569 F
T3	674 F	631 F
T4	736 F	662 F
T5	782 F	
T6	813 F	
T7	832 F	
T8	841 F	

Using Equation 43, the energy absorbed by the 8-inch firebrick after the cycle is 389,600 Btu, and the energy absorbed by the 4-inch firebrick is 154,400 Btu. This is a savings of 235,200 Btu per cycle, or 60% of the energy absorbed by the original floor.

A simplified method may assume that the entire thermal mass obtains the oven temperature at the end of the cycle. In this case, the simplified method would underestimate savings to be more than half of the actual savings. Thus, we recommend using the full finite difference method to estimate savings from reducing thermal mass if the mass is not highly conductive.

$$Q_{abs1} = \text{thick} \times \text{area} \times \text{density} \times c_p \times (T_{\text{final}} - T_{\text{initial}}) = (8/12) \times 100 \times 36 \times 0.25 \times (400-70) = 198,000 \text{ Btu/cycle}$$

$$Q_{abs2} = \text{thick} \times \text{area} \times \text{density} \times c_p \times (T_{\text{final}} - T_{\text{initial}}) = (4/12) \times 100 \times 36 \times 0.25 \times (400-70) = 99,000 \text{ Btu/cycle}$$

$$Q_{\text{sav}} = Q_{abs1} - Q_{abs2} = 198,000 \text{ Btu/cycle} - 99,000 \text{ Btu/cycle} = 99,000 \text{ Btu/cycle}$$

SUMMARY AND CONCLUSIONS

This paper identified three energy losses in process heating systems: heat loss through hot solid surfaces, heat loss from hot liquids exposed to ambient air, and heat loss to thermal mass. Energy-savings measures to reduce these losses were discussed, and methods to quantify savings were presented.

The method for quantifying savings from insulating hot surfaces explicitly calculates convection coefficients and radiation heat loss. A simplified method, which did not explicitly account for these effects, was shown to significantly underestimate heat loss and savings. Thus, we recommend using the full method developed here to estimate savings from insulating hot surfaces.

The method presented for quantifying savings from insulating open liquid tanks accounts for convection, radiation, and evaporation heat losses. The method was shown to be comparable to experimental results. For open tanks, evaporation is the dominant type of heat loss; in the example shown above, evaporation accounted for about 78% of the total heat loss. Thus, we recommend using the method shown here, or a comparable method, when estimating the savings from insulating the surface of hot tanks.

The method presented for quantifying savings from reducing thermal mass in process heat systems uses a lumped capacitance model for conductive material and a finite-difference model for less-conductive material. The finite-difference method generates significantly different and more accurate results than simply assuming that all thermal mass is heated to the temperature of the oven. Thus, we recommend using this method to estimate savings from reducing thermal mass whenever the mass is not highly conductive.

Many of these methods have been incorporated into a free public-domain software application, HeatSim, which is available from the University of Dayton Industrial Assessment Center at www.engr.udayton.edu/udiac.

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