

Atascadero Community Center Event Space Heating system

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Introduction:

In this report we show how we determined the required components to maintain the temperature of a new community center in Atascadero California. Our primary goal is to design an insulation and radiant floor heating system to maintain this 800 square foot event space at a comfortable 68 degrees throughout the year. The room has 4 insulated walls, a concrete floor through which hot water will be pumped inside polyethylene tubes, and a large window along the 40ft length of the room's footprint.

Our process of designing this system starts with determining the heat load of the space at an estimated operating temperature. We made several key assumptions that helped us simplify our model: we assumed that the 3 walls and roof could be simplified as adiabatic surfaces, and that the radiation exchange within the surfaces inside the room could be modeled as diffuse gray with no transmitted radiation. This model of the event space lets us solve for the total heat transfer coefficient that accounts for both natural convection and radiation exchange. Since we know that each surface is adiabatic except the floor and windows, we know that the heat leaving the building only leaves through the windows and is equal to the heat coming through the floor into the event space.

Next we used the Gause Sidel method to create a steady state 2D nodal plot of temperature distribution in the floor. We looked specifically at a cross section with a single pipe through the center and insulation beneath. This allowed us to calculate heat load and the average surface heat flux. We also create a Gause Sidel transient model to determine the time to heat the system from a worst-case 55 degree temperature throughout the system. These models allowed us to size insulation and tubing spacing throughout the floor.

With specified tube properties we began to calculate pressure and temperature drop across the loops as well as the supply system and valves that circulate and heat the fluid. We then validated these values with alternative methods and specified the pump and heat exchanger and expansion tank required to create the system fluid properties. Finally we calculated the cost to install and power the system with a gas boiler. We also compared our system to a comparable electrical radiant heating system.

Heat Load:

We chose to design our system for a day with 30 degree Fahrenheit air and using our model we calculated a heat load of 13224 Btu/Hr. The average floor temperature of the concrete in our system at the calculated head load was 77.3 degrees F. This floor temperature was only based on

a single day temperature estimation of 30 degrees so we needed to check that the heat required keep the room at the target 68 degrees F can be supplied. Here is a plot of floor surface temperature and outside temperature based on data about the minimum and maximum temperatures annually in Atascadero. This plot shows that even from the all-time historic low of 11 degrees Fahrenheit, the required floor temperature is still within a safe range. If we had designed this improperly, a floor temperature above a level that is safe to stand on may be required to maintain the target temperature.

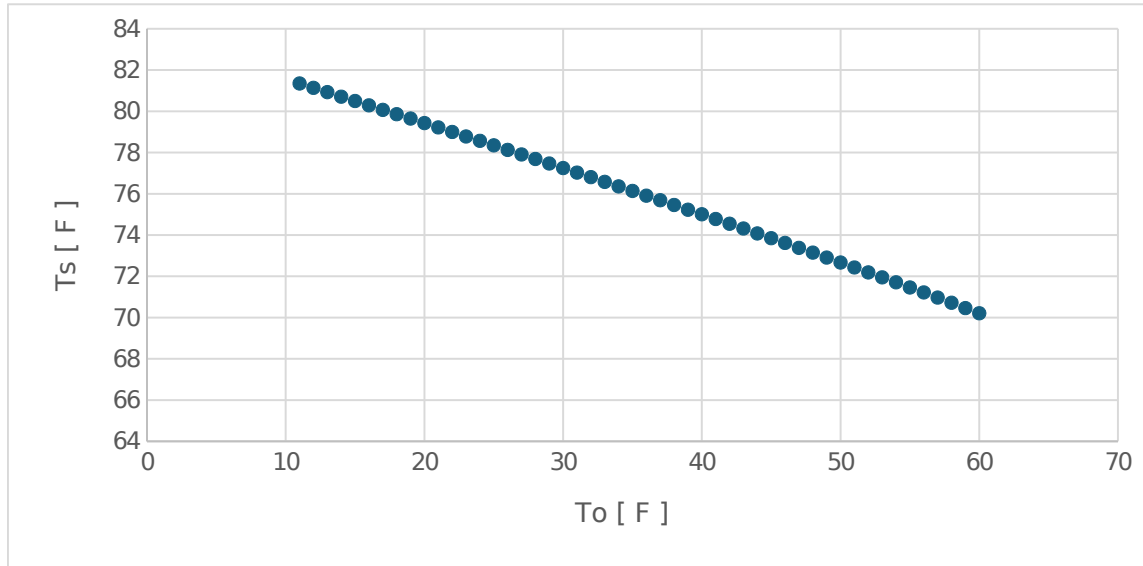


Figure 1. Plot of variation in required surface temperature to maintain internal temperature at 68 degrees Fahrenheit for a given outside air temperature

The following plot shows that the heat flux required to maintain the temperature of the system. The trends exhibited in these plots are both logical, as a higher outside air temperature requires a lower q_{load} to maintain the desired temperature and a cooler floor surface temperature to convect and radiate heat to the insides of the event space.

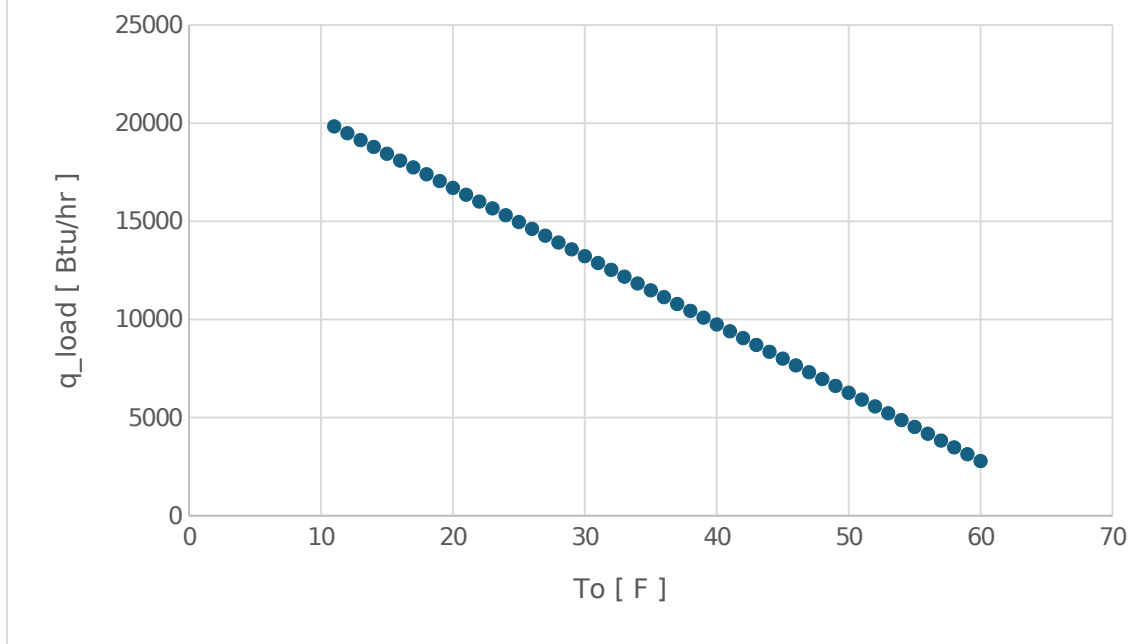


Figure 2. Plot of variation in required heat load to maintain internal temperature at 68 degrees Fahrenheit for a given outside air temperature

In the paper titled “On the heat transfer coefficients between heated/cooled radiant floor and room” from Tomasz Cholewa it is assumed that the reference temperature for the convective heat transfer coefficient is at a height of 1.1 meters as it has been found to best approximate the air temperature. The reference temperature is also assumed to be constant. The calculated h_{total} values in figure 3 are based on the assumptions that all boundary conditions are steady, the dew point is not reached, air speed is less than 0.2 m/s, with insulated walls that convect heat from the room.

In our model we assumed that the air temperature is at a constant 68 degrees Fahrenheit (design temp), still air is assumed and thus we used a natural convection correlation for horizontal flat plates. We also assumed adiabatic walls and roof with a constant heat flux. This means that the heat flux through the floor is equal to the flux through the window. We also did not account for moisture in the air, assuming dry air only.

When we compare our model to the more in-depth thermal model (equation 10) from Cholewa, et al (2013) we find that our is extremely similar to that of the experimental methods used in the paper to validate their thermal model.

$$h_{total} = 7.67(T_s - T_{air})^{0.1} \text{ and } q_{surface} = h_{total}A_s(T_s - T_{air})$$

The calculated heat transfer coefficient of a radiant floor heating system at our temperature delta is $9.038 [W * m^{-2} * K^{-1}]$ (with Cholewa model) whereas our total heat transfer coefficient value was $9.95 [W * m^{-2} * K^{-1}]$ using our own model. The $q_{surface}$ calculated using our model was $3777.9 [Watts]$ and with Cholewa it was $3898.7 [Watts]$ with our own.

This makes sense because our model assumes that the walls are adiabatic and so heat transfer is reduced overall. The diffuse gray, no-transmittance assumptions of our model cause our window to behave more similarly to a wall that is irradiating the interior space. The effective difference between our models is that the Cholewa model has more surfaces to convect heat to. The figure presented shows the relationship between the total heat transfer coefficient of the floor and a temperature delta between the surface and the air temperature.

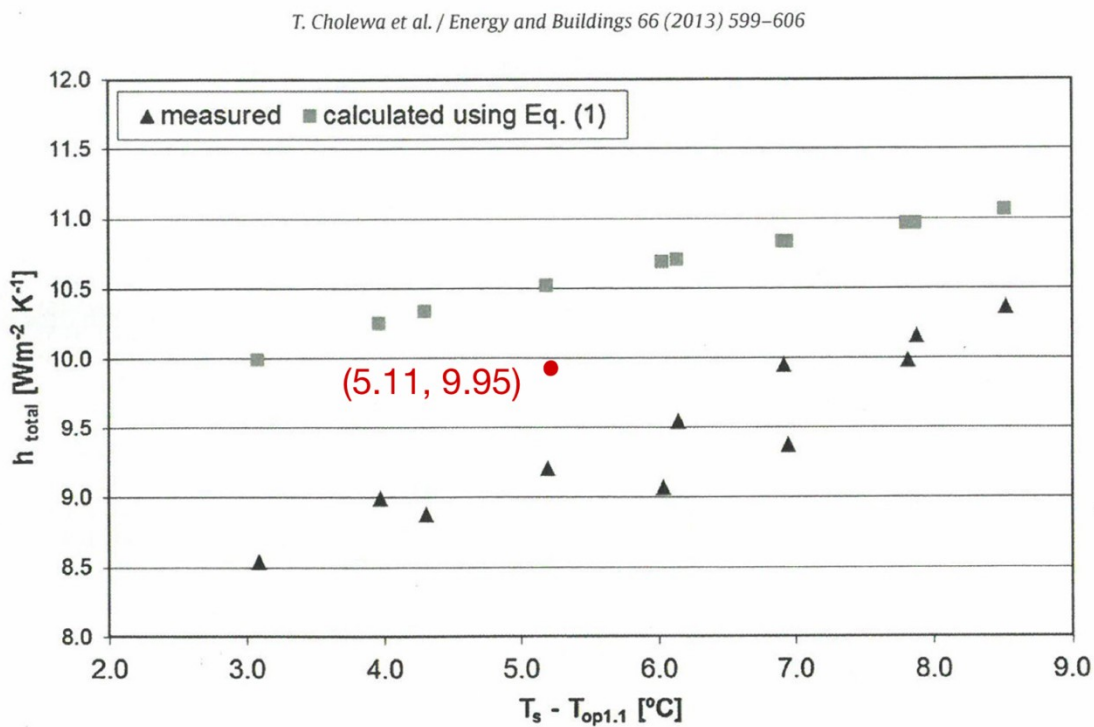


Figure 3. Calculated h_{total} value (using equation 10) and floor-air temperature difference plotted over Cholewa, et al (2013) plot of h_{total} (using equation 1) vs $(T_s - T_{air})$ for a heated radiant floor and room.

The data for h_{total} in figure 3 uses equation 1 which is based on commonly used engineering practices that are shown in the figure to be overestimates of the actual value. When we plot our own model's h_{total} we see that it closely aligns to the measured h_{total} values. When we calculate h_{total} using equation 10 we get a value that is far more conservative, hence why Cholewa proposes it be adopted as a standard when calculating heat transfer coefficients for radiant heating systems.

Unfortunately, this data and associated new model cannot be taken at face value for two reasons. The measured data has no reported error or uncertainty. This means that the data could actually have been measured to be far closer or even the same as the existing model. Furthermore, the spread of the data around a linear or second order fit is large showing that the repeatability and of the experimental setup is low. The difference between the furthest experimental datapoints around 7 Celsius is nearly that of the difference between the experimental and calculated values at 8.25 Celsius. The quality of the experimental results and the method of reporting them are not sufficient to draw any meaningful conclusions.

Steady State 2-D Conduction Model:

Once we determined the total heat transfer coefficient (h_{total}) for the floor of our system to be $1.753 \text{ [Btu/hr-ft}^2\text{-R]}$, we used that value to create a model of the pipes that run through the concrete. We selected $\frac{1}{2}$ in PEX tubing at a 12 inch spacing to heat the floors of the community center. Since most of the heat transfer happens around straight long sections of PEX tubing we can approximate this with a 2d cross section of a single pipe. We selected a 0.1 inch nodal spacing our calculations and the resulting model was 121 by 51 nodes.

We calculated energy balance equations for each node summing the total heat going into the node. Notably we placed node centers on the boundaries and corners themselves as opposed to interior material. This model has 9 unique nodal equations to calculate the temperature of each node based on its neighboring nodes. There are 3 pairs of symmetric nodes for which the equations were mirrored in accordance. The notation used is as follows: The temperature of any node is referred to as T . T_T refers to the temperature of the node above the node current node. T_B refers to the temperature of the node below the current node. T_L and T_R refer to the temperature of the nodes to the left and right. K_c , K_i , B_i , and T_∞ are the thermal conductivities of concrete and insulation materials, Biot Number and room air temperature.

Finite Difference Equations:

1. Interior Node Equation:

$$T = \frac{1}{4}(T_L + T_T + T_R + T_B)$$

2. Insulated Boundary Node Left Equation

$$T = \frac{1}{4}(2*T_R + T_T + T_B)$$

3. Insulated Boundary Node Right Equation

$$T = \frac{1}{4}(2*T_L + T_T + T_R)$$

4. Top Boundary Node with Convection Equation

$$T = \frac{1}{4 + 2*B_i}(2*B_i*T_\infty + T_R + T_L + 2*T_B)$$

5. Insulated Top Corner Left Equation

$$T = \frac{1}{B_i + 2}(B_i*T_\infty + T_L + T_B)$$

6. Insulated Top Corner Right Equation

$$T = \frac{1}{B_i + 2}(B_i*T_\infty + T_R + T_B)$$

7. Interior Boundary Node between Concrete and Insulation

$$T = \frac{1}{4}(T_R + T_L) + \frac{1}{2*(K_c + K_i)}*(K_c*T_T + K_i*T_B)$$

8. Left Insulated Boundary Node between Concrete and Insulation

$$T = \frac{1}{2}*T_R + \frac{1}{2*(K_c + K_i)}*(K_c*T_T + K_i*T_B)$$

9. Right Insulated Boundary Node between Concrete and Insulation

$$T = \frac{1}{2}*T_L + \frac{1}{2*(K_c + K_i)}*(K_c*T_T + K_i*T_B)$$

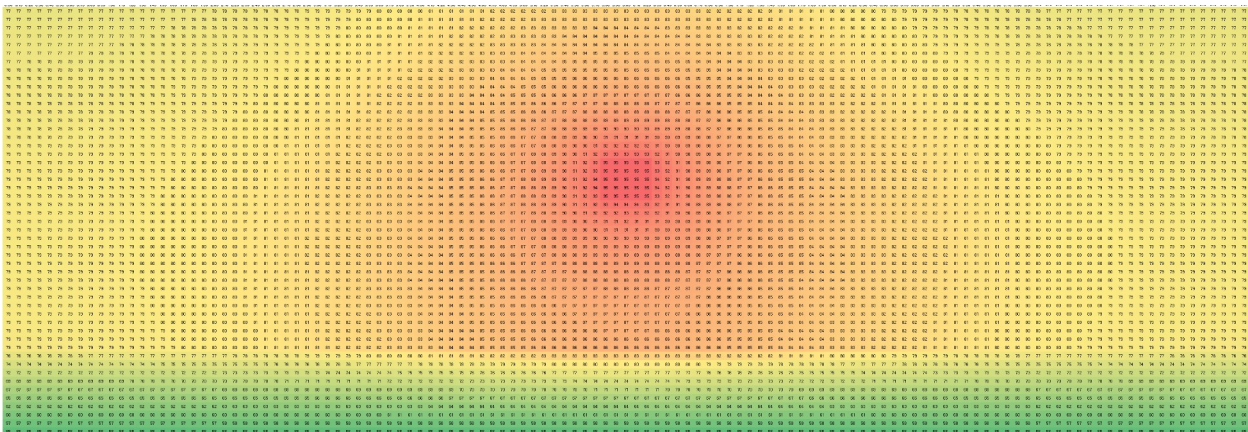


Figure 4. Steady State 2D conduction nodal model using Gause-Seidel method of 12-inch-wide cross section with 4 inches deep concrete on top of 1 inch of insulation, and constant surface temperature 0.625" OD PEX tubing.

In this model, the green represents colder temperatures and red represents hotter ones. The maximum temperature in this model is at the pipe surface at 95 degrees F. The minimum temperature is where the insulation contacts the ground at 55 degrees F.

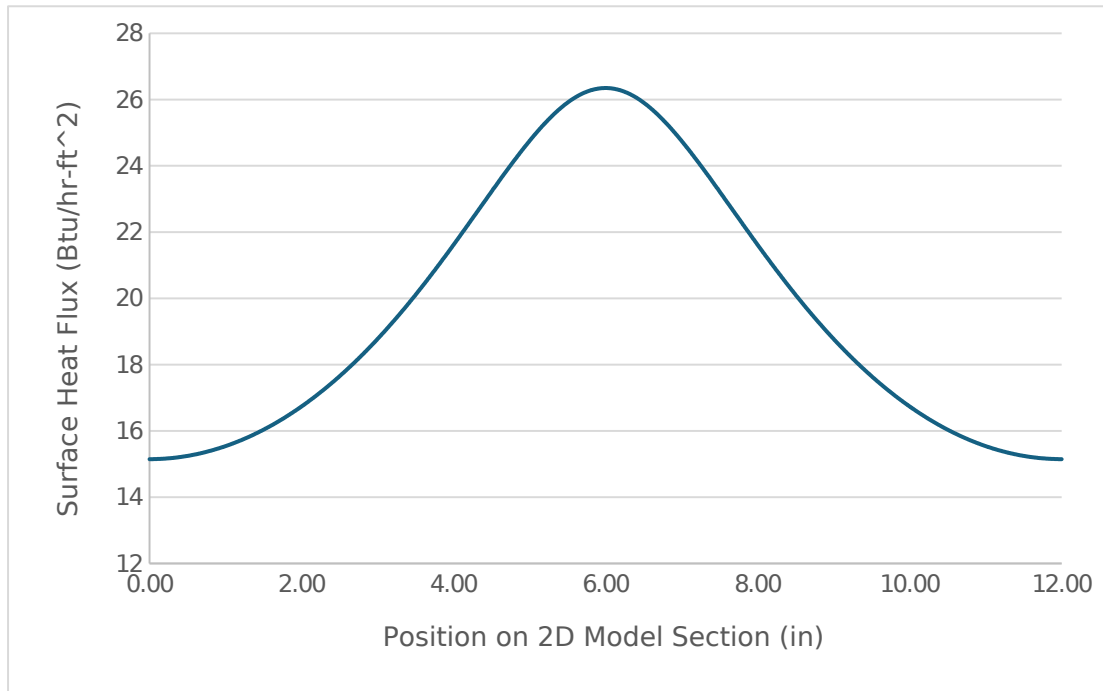


Figure 5. Plot of surface heat flux versus position along 2D floor cross section surface at steady state conditions.

Table 1. Steady State Conduction 2D Model values

Variable	Value
H_{total}	1.753 [Btu/hr-ft ² -F]
Nodal Spacing	0.1 [in]
Average Surface Heat Flux	17 [Btu/hr-ft ²]
Heat Transfer to Floor $q_{surface}$	13303 [Btu/hr]
Average Surface Temp T_s	77.4 [F]
Heat Loss to Ground $q_{ground-loss}$	492 [Btu/hr]
Thermal resistance of concrete	0.81 [ft ² -F-hr/Btu]

$R_{concrete}$	
Thermal resistance of Concrete and Insulation $R_{insulation}$	59.4 [ft ² -F-hr/Btu]
Tube surface temperature [F] $T_{tube-surface}$	91 [F]

The average surface temperature from the 2-D model was 77.5 F for a pipe outside temperature of 95 F. This closely matches the surface temperature of 77.4 F from the EES heat load model. We did not refine the pipe surface temperature any further because the surface temperatures matched with < 1%. In the 2-D model the heat load and surface temperature are determined by the PEX pipe outside temperature, so we adjusted the pipe temperature until the heat loads were close in magnitude. By doing this we found that for the desired heat load we needed a pipe temperature of 95 F to maintain steady state conditions.

To determine the insulation thickness we started with 1 inch of insulation at \$2400 for the whole floor area. The heat loss through 1 inch of insulation is 492 Btu/hr so after 15 years of operating every day for 8 hours/day, the heat loss costs about \$323 assuming \$1.5 per therm. Adding more insulation is not financially viable since it would cost another \$2400 to save a couple hundred dollars in electricity over the 15 yr operating life of the system.

From our 2D conduction model we calculate the $q_{ground-loss}$, $T_{tube-surface}$, $T_{surface}$, T_{ground} , and q_{load} . Using these values, we can calculate the thermal resistance of the floor and concrete-insulation combination that resist heat transfer under the tubes.

$$q_{ground-loss} = \frac{T_{tube-surface} - T_{ground}}{\frac{R_{concrete-insulation}}{A_{floor}}}$$

$$q_{surface} = q_{load} = \frac{T_{tube-surface} - T_s}{\frac{R_{concrete}}{A_{floor}}}$$

$$492 \left[\frac{Btu}{hr} \right] = \frac{95 [F] - 55 [F]}{\frac{R_{concrete-insulation}}{800 [F t^2]}}$$

$$13303 \left[\frac{Btu}{hr} \right] = \frac{95 [F] - 77.5 [F]}{\frac{R_{concrete}}{800 [F t^2]}}$$

$$R_{concete - insulation} = 58.6 \left[\frac{F t^2 * F * hr}{BTU} \right]$$

$$R_{concrete} = 0.81 \left[\frac{F t^2 * F * hr}{BTU} \right]$$

We initially chose a tube spacing of 16in as discussed in our preliminary report but downsized to 12in because the pipe temperature needed start up the system in under 4 hours was very high. We only added an extra ten tubes which has a negligible effect on the pex cost and allowed us to specify a smaller piping and heating system. ½ in PEX pipe was chosen because it is the cheapest option and most widely available size.

Transient 2-D Conduction Model:

To evaluate the system start up and cooldown times, we performed a transient 2D conduction analysis using Gauss-Seidel iteration in Excel. The finite difference equations are outlined below, and the temperature names are the same as in the steady state model. T_p is the node temperature at the previous time step and F_o is the Fourier number, ΔT is the time step, Δx is the node spacing, and α is the thermal conductivity of the material (concrete or insulation).

$$F_o = \frac{\alpha * \Delta T}{\Delta x^2}$$

Finite Difference Equations:

1. Interior Node Equation:

$$T = \frac{1}{4 + F_o} (T_L + T_T + T_R + T_B + \frac{1}{F_o} * T_p)$$

2. Top Left Corner Boundary Node Equation:

$$T = \frac{1}{2 + Bi + \frac{1}{F_o}} (T_R + T_B + Bi * T_\infty + \frac{1}{2 * F_o} * T_p)$$

3. Top Right Corner Boundary Node Equation:

$$T = \frac{1}{2 + Bi + \frac{1}{F_o}} (T_L + T_B + Bi * T_\infty + \frac{1}{2 * F_o} * T_p)$$

4. Top Convection Node Equation:

$$T = \frac{1}{2 + Bi + \frac{1}{Fo}} (T_L + T_B + 2 * T_B + 2 * Bi * T_\infty + \frac{1}{Fo} * T_p)$$

5. Insulated Left Boundary Node Equation:

$$T = \frac{1}{4 + Fo} (T_T + 2 * T_R + T_B + \frac{1}{Fo} * T_p)$$

6. Insulated Left Boundary Node Equation:

$$T = \frac{1}{4 + Fo} (T_T + 2 * T_L + T_B + \frac{1}{Fo} * T_p)$$

7. Internal Node Between Materials Equation:

$$T = \frac{1}{4 + 4 \frac{Ki}{Kc} + \frac{1}{Fo_c} + \frac{Ki}{Kc} * \frac{1}{Fo_i}} * \left(2 T_T + \left(1 + \frac{Ki}{Kc} \right) * (T_R + T_L) + \left(2 \frac{Ki}{Kc} \right) * T_B + \left(\frac{1}{Fo_c} + \frac{Ki}{Kc} * \frac{1}{Fo_i} \right) * T_p \right)$$

8. Left Boundary Node Between Materials Equation:

$$T = \frac{1}{2 + 2 \frac{Ki}{Kc} + \frac{1}{2 Fo_c} + \frac{Ki}{Kc} * \frac{1}{2 Fo_i}} * \left(2 T_T + \left(1 + \frac{Ki}{Kc} \right) * T_R + \frac{Ki}{Kc} * T_B + \left(\frac{1}{2 Fo_c} + \frac{Ki}{Kc} * \frac{1}{2 Fo_i} \right) * T_p \right)$$

9. Right Boundary Node Between Materials Equation:

$$T = \frac{1}{2 + 2 \frac{Ki}{Kc} + \frac{1}{2 Fo_c} + \frac{Ki}{Kc} * \frac{1}{2 Fo_i}} * \left(2 T_T + \left(1 + \frac{Ki}{Kc} \right) * T_L + \frac{Ki}{Kc} * T_B + \left(\frac{1}{2 Fo_c} + \frac{Ki}{Kc} * \frac{1}{2 Fo_i} \right) * T_p \right)$$

Startup From 55 F Uniform Temperature:

The first transient analysis we performed was a startup of the system from a uniform concrete and ground temperature of 55F. This represents the first time starting the system assuming the rest of the building is also cold.

Table 2. Transient system properties for cold start up.

Variable	Value
Pipe Surface Temperature	91 F
Start Up Time (95% Tss)	7 hr
Simulation Time Step	1 hr

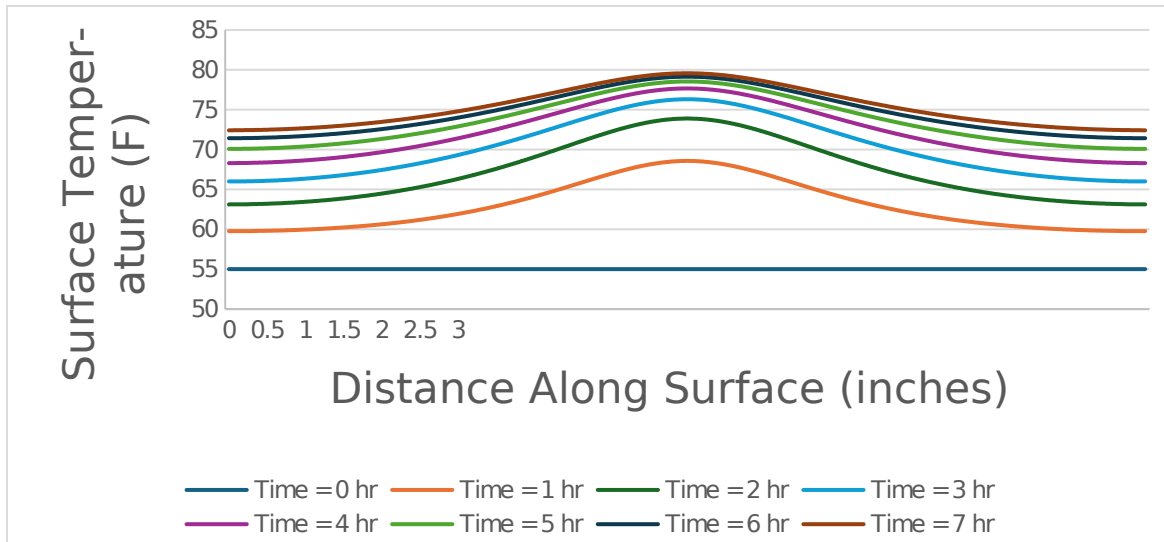


Figure 6. Transient surface temperature profile along 2-dimensional nodal cross section when starting from a system at 55 [F] state.

24 Hour Cooldown:

We analyzed the system cooldown over a 24 hr period assuming the rest of the building is still heated. This represents a full day of cooling off before being started up again. For this estimate we ignored the tubes and replaced them with more interior nodes that do not generate heat.

Table 3. Transient system properties for cooldown after steady state operation.

Variable	Value
Ts After 24 hr Cooldown	67 F
Total Cooldown Time	24 hr
Simulation Time Step	3 hr

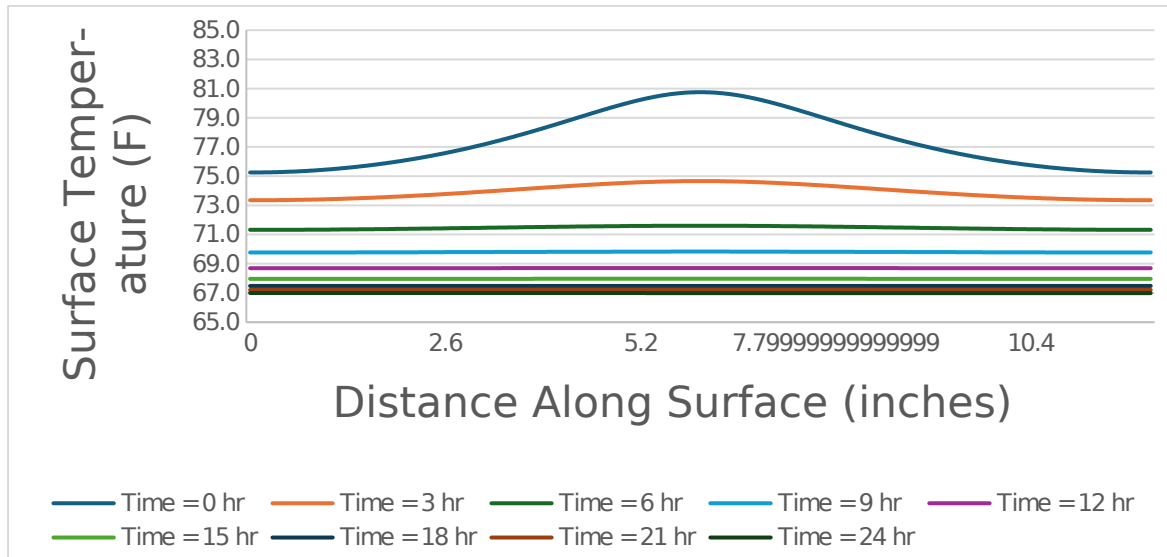


Figure 7. Transient surface temperature profile along 2-dimensional nodal cross section of cooldown from system at steady state

Start Up After 24 hr Cooldown:

Often the system will be used consecutively day after day in which the system will not be required to heat from the worst-case cold startup scenario. We assumed the room will be used 5 days per week so we wanted to calculate the time to reheat the system after a night of cooldown. Our goal was to be able to start up the system in 4 hours. Initially our pipe temperature was too low to reach steady state conditions in under 4 hours so the start up temperature was increased to 95 F. At 95 F the average concrete surface temperature reaches 95% of the average steady state temperature in 3.5 hours. This is worst case scenario assuming an outside temperature of 30F on most days the outside temperature will be higher and the start up time will be smaller. We chose a maximum of 4 hours to start the system in case it needs to be started manually. Assuming staff arrive at 6am, the system can be heated by 10 am which is when we expect most events to start.

Table 4. Transient system properties for startup after 24 hour cooldown.

Variable	Value
Pipe Surface Temperature	95 F
Start Up Time (95% Tss)	3.5 hr
Simulation Time Step	1 hr

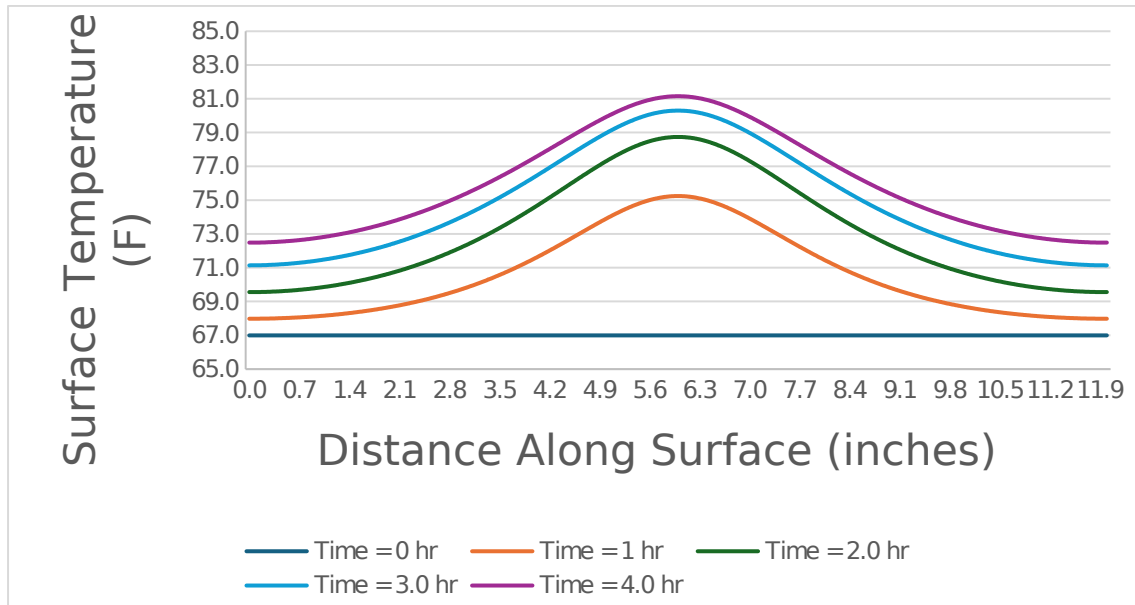


Figure 8. Transient surface temperature plot for 2-dimensional nodal cross section of

Updated Steady State Values for Increased Startup Temperature

Since the pipe temperature was increased, the system must be sized for the higher heat load. Updated steady state values were calculated with the new temperature of 95 F and are listed below. The updated total heat transfer rate and thermal resistances were used to size the piping system and heat exchanger.

Table 5. Nodal model properties and resulting Heat transfer, Flux, Temperature, and Thermal resistance values of the Gause Sidel model.

Variable	Value
H_{total}	1.753 [Btu/hr-ft ² -F]
Nodal Spacing	0.1 [in]
Average Surface Heat Flux	20 [Btu/hr-ft ²]
Heat Transfer to Floor $q_{surface}$	15758 [Btu/hr]
Average Surface Temp T_s	79.2 [F]
Heat Loss to Ground $q_{ground-loss}$	537 [Btu/hr]
Thermal resistance of concrete $R_{concrete}$	0.80 [ft ² -F-hr/Btu]
Thermal resistance of Concrete and Insulation $R_{insulation}$	60.4 [ft ² -F-hr/Btu]

Tube surface temperature [F] $T_{tube - surface}$	95 [F]
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Heat Transfer in Pipes:

To determine the heat transfer through the pipes of our system we need to select the length and number of pipes as well as how many times each loop will traverse the 20 ft length of the room. Each loop as a number of tubes that are connected at each end with a bend. To select the number of loops in our system we created a parametric table of all tube amounts that divide into our 40 foot room length and allow us to keep the inlet and outlet T joints parallel. This makes it easy for each fluid path through the system to have the same resistance and thus same flow without an expensive valve system.

Table 6. Parametric design table of system properties for various numbers of tubes per loop in our thermal system model.

	1 tubes _{per,loop} [-]	2 x _{loop,length} [ft]	3 m _{loop} [lbm/s]	4 dP _{loop} [lbf/ft ²]	5 H _{loop} [ft]	6 T _{m,water} [F]	7 T _{mi} [F]	8 T _{mo} [F]
Run 1	2	40	0.04866	15.3	0.2468	99.48	101.8	97.16
Run 2	4	80	0.08452	78.39	1.264	99.29	102	96.62
Run 3	8	160	0.1059	230.2	3.713	100.1	104.3	95.82
Run 4	10	200	0.1049	282.2	4.553	100.9	106.3	95.49
Run 5	20	400	0.08628	396.4	6.405	107.9	121	94.86

We selected 8 tubes per loop with a total of 5 loops. This value was determined by looking at the water inlet and outlet temperatures. Our loop thermal system is modeled with a constant temperature across its length. For this type of model, we need to keep the difference between the inlet and outlet water temperature less than 10% of the mean water temperature for our model to be valid. A large temperature drop across the length of the loop could lead to uneven heating of the floor which may be uncomfortable for patrons of the event space. The thermal properties of our system at the specified tube are shown in table 2 below.

Table 7. Total system and Loop Flow properties for heat transfer analysis.

Variable	Value
Loop Length	160 [ft]
Flow Rate per Loop	381[lbm/hr]
Number of Loops	5
Total Flow Rate	3.84 [gpm]
Total Flow Rate	1907 [lbm/hr]
Q _{total}	16224 [btu/hr]

Required Loop Inlet Temperature	104.3 [F]
Loop Temperature Drop	8.5 [F]
Single Loop Pressure Drop in psi	1.6 [psi]
Single Loop Pressure Drop in Feet Head	3.71 [ft]

To validate that this water temperature is valid for our system design we compared our system analysis to the standards outlined in the 2016 ASHRAE handbook.

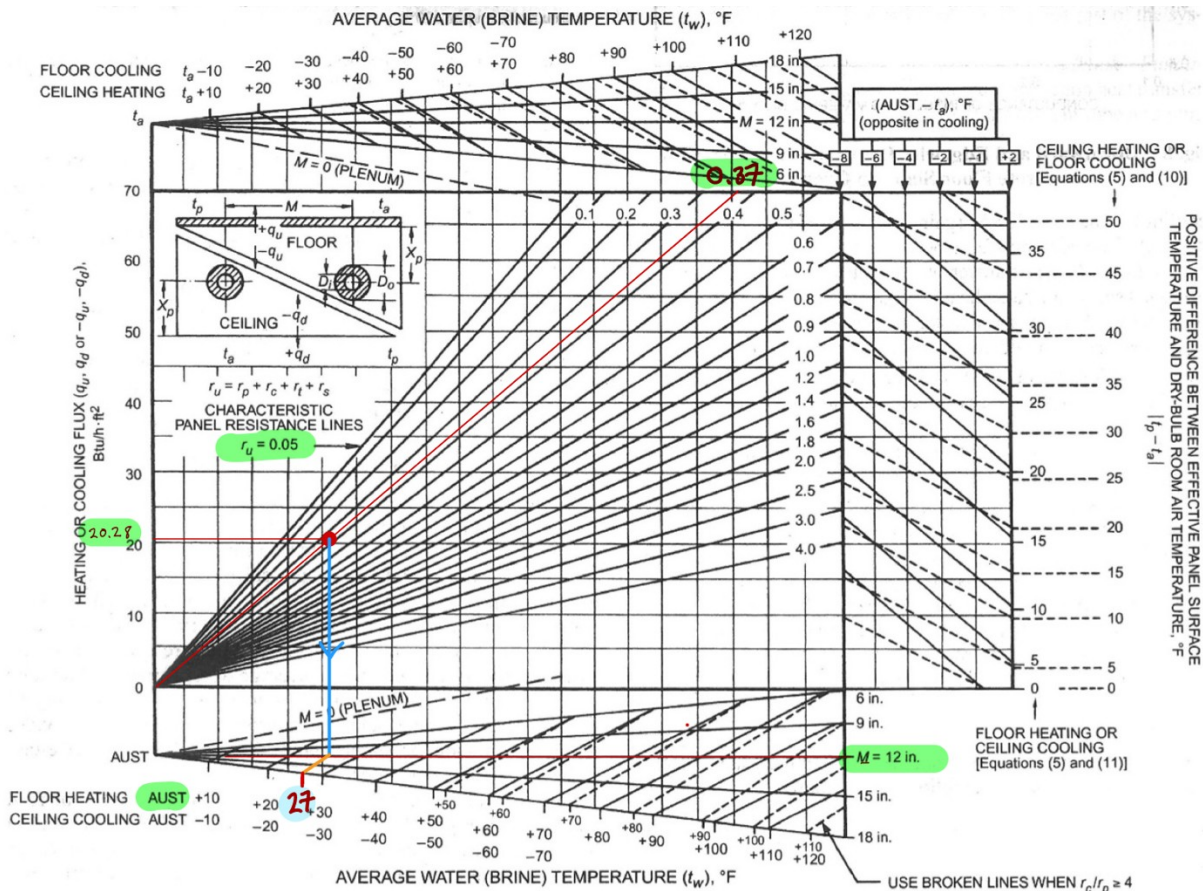


Figure 9. Design Graph for Sensible Heating and Cooling with Floor and Ceiling Panels from ASHRAE with system annotations.

Using Figure 9 from the panel design method (ASHRAE Handbook, Chapter 6) we can calculate the temperature of the water required to circulate through the system to meet our calculated q_{total} . We used a 12-inch pipe spacing, 68-degree target temperature (AUST), heat flux per unit area of 20.28 Btu/hr·ft². We determined the characteristic panel length of our system based on r_p , the thermal resistance of the body panel and r_t the thermal resistance of the tube. Because there is no carpet, and the tubes are mounted inside the concrete the r_c and Mr_s values can be neglected.

This yields characteristic panel resistance of $r_u = 0.37$. Using the ASHRAE design graph we plotted a 95-degree [F] suggested water temperature to meet our design heat flux.

Using our own model, we calculated a water temperature of 100.3 [F] which is very close. While both methods use a diffuse grey assumption, ASHRAE uses the MRT method which assumes a fictitious finite surface with the same heat flux as determined by a 1980 paper by Walton in his 1980 paper on *A new algorithm for radiant interchange in room loads calculations* (ASHRAE Transactions 86(2):190-208).

This means that designers don't need to consider the angle of surfaces in the room. The area of the fictitious surface is an area weighted sum of the emittance of all surfaces. For situations where surface emittances of enclosure are nearly equal this means that unheated surfaces are not accounted for in the AUST value. This means that the ASHRAE method may trap slightly more heat in the system because the walls are combined into a single representative surface and do not account for a slightly colder window. This discrepancy causes the required water temperature flowing through the tubes of our system to be slightly higher than a room with symmetric walls.

Another difference between the ASHRAE model and our model is that the ASHRAE model does not account for solar radiation entering the system. This effect may be less dominant due to the thermal reflective coating placed on the window which reflects radiation that would otherwise heat the system.

While there are some system differences, the ASHRAE model closely follows our own validating our water temperature.

Total System Pressure Drop and Pump Selection:

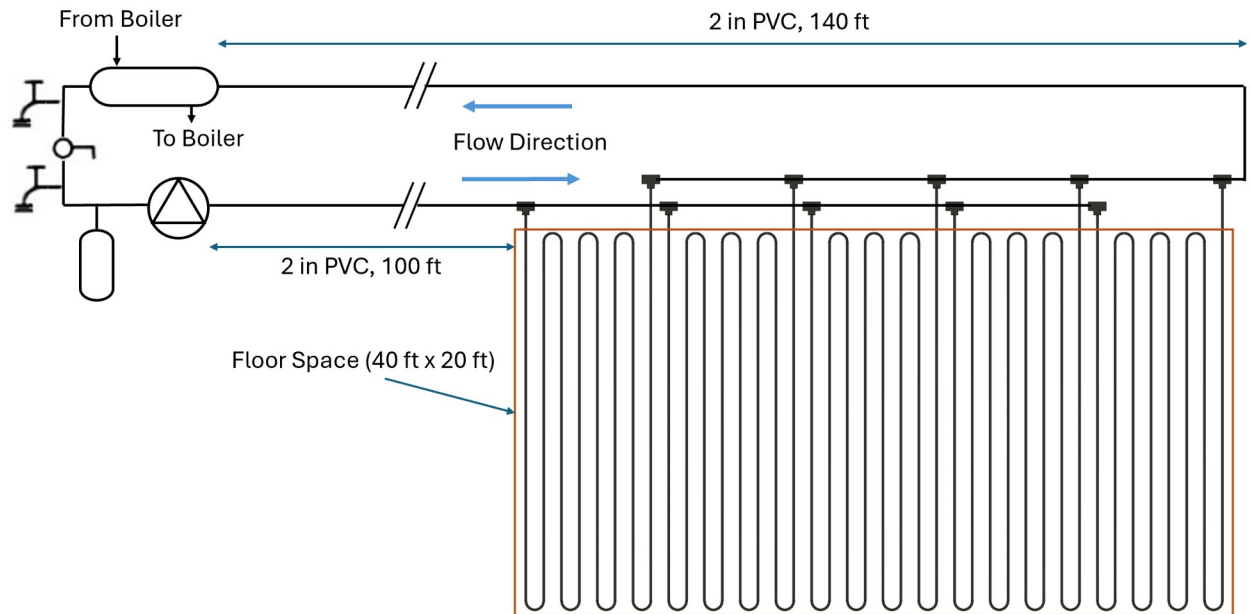


Figure 10. System drawing of 40 ft x 20 ft radiant floor heating system and components critical for operation.

The total pressure drop across the system in figure 10 accounting for all fittings, valves, tubes, and other components was 2.7 psi or 6.17 feet of head. The design flow rate is 3.84 gpm. To circulate the water, we selected the UP-15-42F pump from Grundfos, operating at the lowest speed (left most H vs Q curve)

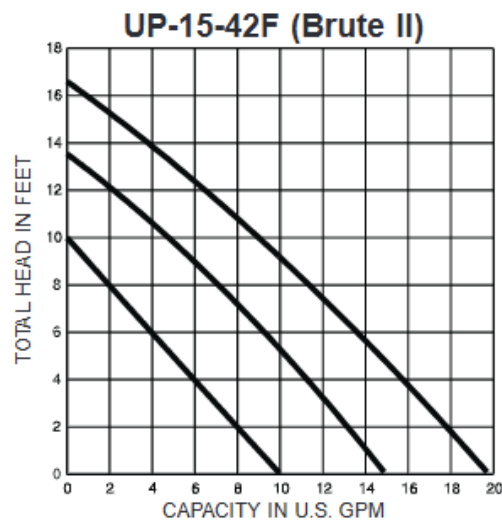


Figure 11. Pump curve for UP-15-42F given by Grundfos Technical Documentation.

Table 8. Grundfos UP-15-42F Pump operating point at system conditions

Variable	Value
Flow Rate	3.84 [gpm]
Pressure	6.17 [ft]
Mechanical Power	0.01 [BHP]
Pump Efficiency	0.6 [-]

Expansion Tank Selection:

To determine the size of the expansion tank we calculated the change in volume of our systems water across a conservative temperature range from the 55 degrees F ground temperature to the maximum PVC pipe temperature of 140 degrees F. We know that the specified expansion tank must have a volume greater than that of the maximum volume change the water will see.

$$V_{\text{tank}} = V_{\text{fluid}} * \frac{\frac{V_{\text{max}}}{V_{\text{min}}} - 1}{1 - P_{\text{min}} / P_{\text{max}}}$$

We calculated V_{fluid} to be 7.56 cubic feet for which a minimum of a 0.987 gallon expansion tank would be required at a maximum. Because our temperature range was so conservative it is acceptable to select an expansion tank with the volume of 1 gallon with minimal margin of safety

To select the expansion tank we had to ensure that the tank can withstand the pressures and temperatures our system will experience. We calculated that the maximum and minimum gauge pressures of our system to be 61.6 Psi and 1.29 Psi respectively.

We selected the Amtrol ST-1 Expansion tank which costs \$31.5 from suppliers like Home Depot. This tank has a 1-gallon capacity, maximum temperature of 140 [F], and a max pressure of 150 PSI which meet our design requirements. An added benefit of this tank is that it is compact with a pre-charged bladder technology that will reduce installation time and cost as opposed to a similarly priced 2-gallon system.

Heat Exchanger Selection:

We selected a CX40 heat exchanger from Allied Boilers. This heat exchanger at the manufacturers specification has an effectiveness of 0.244 and NTU of 0.328 with a coil inlet temperature of 96 [F] and shell inlet temperature of 180 [F] assuming 4 [gpm] flow through both shell and tube. The conductance specified by the manufacturer at our specified coil inlet temperature in the CX40 was 639.2 [btu/hr*F].

In our system, the tube flow and shell flow was 0.44 [gpm] and 3.8 [gpm] respectively. The heat exchanger in our system has an NTU of 2.98 and total heat transfer rate of 16224 [Btu/hr] to the water.

The heat exchanger is designed to operate at a maximum inlet flowrate of 4 gpm, but our system requires about 0.4 gpm. We operate the system at the percentages of the max flowrate as shown below.

Table 9. Heat Exchanger flow rate properties for steady state and startup conditions.

Variable	Value
Max hot side flowrate	4 [gpm]
Steady state % max flowrate	8.7 %
Steady state hot side flowrate	0.35 [gpm]
Startup % max flowrate	11 %
Startup hot side flowrate	0.44 [gpm]

At startup the correction factor for our heat exchanger is 0.875 and at steady state conditions the correction factor is 0.827. This correction factor indicates that the heat exchanger is operating on the lower end of an acceptable range. A correction factor lower than 0.8 typically as it is operating further away from an idealized counter flow heat exchanger. Because we have calculated both steady state and startup correction factors we can see that the system correction factor will be above the 0.8 threshold which indicates our heat exchanger is adequately sized.

Cost Estimation:

The capital cost to install the designed system came to a total of \$16,067. We estimated the operating costs of the system assuming the event space will be heated for 5 days of the week for a total of 260 days per year. We calculated the startup time of the system to be 3.5 hours after a 24 hour cooldown and assumed that the system will be used for 12 hours total per day. We conservatively calculated cost per day with 4 hours at startup conditions and 8 hours at steady state heat loads. We assumed the boiler efficiency of 0.85% when determining gas usage. We also included electrical cost to operate the pump in our system.

Table 10. Cost of power by utility for summer and winter months

Energy/Season	Winter	Summer
Gas	0.060420 \$/kwh	0.054270 \$/kwh
Electricity	0.34033 \$/kwh	0.39222 \$/kwh

Because of the large demand for electrical power from air conditioning units the cost of power rise during the summer as seen in table XX. Whereas gas prices decrease in summer due to less heating systems being used. Overall gas prices remain a fraction of electrical prices per Kwh.

Operation of the system using natural gas costs \$916.1 per year. The present value of the operating cost using a gas boiler for a 15 year lifetime is \$8,897.58. The cost of installing our system and operating it for the lifetime is a total of \$24,964.34

Installing a comparable electrical radiant heating system would cost 20\$ per square foot for a total of \$16,000 which is just slightly less than the gas boiler system we designed. Where we see a radiant floor heating system such as our own excel is when we look at the operating costs. The present value of the 15 year lifetime for the electrical system is \$55,103.26, which is more than 6 times more expensive than a gas powered system. The cost of installing and operating a comparable electrical system for 15 years is \$71,103.26

This vast operational cost difference negates the small \$67 cost savings made during the installation process making the radiant floor heating system with pumped water the superior long term economic decision.

The system can be controlled entirely by varying the hot water flowrate from the boiler to the heat exchanger. A pump with a variable frequency drive will be used to set the flowrate and PID control with air temperature feedback will be used to control the pump.

Conclusion:

Overall we were able to design a radiant floor heating system to keep the room at 68 degrees Fahrenheit during use. We used transient and steady state models to calculate system requirements throughout a day of operations. We validated the water temperature of our system using the panel design method from ASHRAE. Validated our tube sizes and insulation thickness through parametric tables and compared the effect of a range of values on our overall system cost and operational costs. We also confirmed our heat transfer coefficient for a given air floor surface temperature drop was acceptable by comparing to Chelewa et al. and the European standard for calculating the thermal performance of surface heating systems outlined in EN 1264-2. We used these models to iterate and evaluate how the insulation, tubing sizes, and other components affect the lifetime and operational cost of the system. The resulting system was adequate meeting our design requirements and cost \$16,067 to install and \$24,964 to operate for a 15 year span.