SIGMA EXPERT SOLUTIONS LLC

RESIDENTIAL HOT WATER SYSTEM
WHITE PAPER

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Disclaimer:

This is a white paper used to propose a concept and does not represent a design calculation. The results presented are based on careful consideration of inputs, review of calculation results, and comparison with other data, but independent check and review has not been performed. Use of this information as a basis for decision making should only be made after independent verification of calculations and independent review.

Table of Contents

1. Introduction

1.1. Background on Problem

This white paper investigates the use of an alternative means of providing heat to a residence for various services. Traditionally, heat is provided by various different systems, tailored for the application. These systems are essentially independent. The proposed alternative consists of a centralized hot water circulating loop for transport of heat throughout the residence.

1.1.1. Current Heating Systems

In residences, heat is used mostly for provision of climate control, heating of water, and cooking. Other uses unique to specific residences may include heating of swimming pools, exterior heating, etc. This white paper will focus on provision of climate control and hot water.

Climate Control

Heating for residences is generally performed either by transfer of heat through air ducts from an air handler, or radiant floor heating. Duct heating is the most common and is addressed here, but the proposed system would be well leveraged to provide radiant floor heating.

Hot Water

Hot water is provided either through a central hot water heater or “on-demand” water heaters. Both can use either gas or electric as their source of heating. Incoming cold water is elevated in temperature by the introduction of energy, either in a centralized and insulated tank that serves all of a house, or at discrete locations when hot water is in demand.

1.2. Proposed Solution

A hot water loop is proposed to replace the heat source for both potable hot water and climate control. It consists of a closed loop system of pipes containing water and potentially some additives to minimize corrosion, several heat exchangers, a circulation pump, and a head tank. The exact installation of the loop will be dependent on the requirements of the residence, but the proposed version introduced for analysis has the following characteristics:

1. Circulates hot water at approximately 140F around the residence.
    a. This water temperature is high enough to ensure efficient heat transfer for use in providing potable hot water, and the return temperature is likely low enough to allow for heat recovery from the chill water system.
2. Hot water is transferred through a highly insulated piping system.
    a. The piping system is primarily a pipe-in-pipe system that will be described in this white paper.

3. Water pressure is slightly below potable water pressure.

   a. Any leak in a hot potable water heat exchanger will result in leakage into the loop as opposed to leakage from the loop which may contain anti-corrosive chemicals.

4. Has a regenerative heat exchanger that extracts heat from a chill water heat rejection heat exchanger and a heat exchanger that provides heating through solar energy.

   a. The concept is premised on a complimentary chill water system (to be described in another white paper) that will have heat rejection from the refrigeration cycle high pressure side. This heat will be primarily rejected to the hot water loop with the remainder being rejected to atmosphere.
   b. Solar heating is included as it is easily integrated into the system and highlights some ways that the system can improve overall efficiency of the residence.

5. Has electrical resistance as the means of heating.

   a. The majority of the heating would be through solar heating. High efficiency electrical heaters are the simplest manner to provide any supplemental heating, but heat pumps could also be used as a more energy efficient solution in mild climates.

6. Uses a variable speed circulating pump that maintains the system differential pressure.

   a. As services are put under demand, flow through the system will increase as required to meet the demand. The circulating pump will maintain a constant pressure between the high pressure and low pressure side of all downstream services so they work consistently, independent of the operation of other parallel services.

7. Services hot potable water heat exchangers and point air conditioning units.

   a. The hot potable water heat exchangers and point air conditioning units will be described later as part of the white paper calculations.

1.3. Results and Conclusions

Based on analysis, the proposed solution would use 9.48 kWh of electricity to provide heating and hot water during a harsh winter day in Zone 4. By contrast, a perfect (reversible) heat pump operating across the outside temperature of 20°F and inside temperature of 72°F would have a coefficient of performance of 10.2. For the house heating requirements conservatively estimated as 437400 BTU (128kWh), that would require electrical input of at least 12.5kWh, though typically coefficients of performance are more typically 3-4 for heat pumps, so the expected electrical usage would be much higher. Additionally, the hot water loop provides water heating services for a total of over 48000 BTU of heating energy, which, if there was perfect conversion from electrical heating would require 4kwh. Therefore the 9.48kWh of this solution takes the place of 16.5kWh of electricity that would be commonly used in the analyzed house. These savings would add up over time, especially if a similar chill water system was provided for summer months.

In this white paper, the details of design are mostly ignored except where they are required to define process conditions for performance of calculations. The concept of a system that transports heat by use of water is at the core of the concept, and details of system design will be specific to application in terms of services, level of use of sustainable energy sources, integration with other concepts such as geothermal heating and heat pumps, etc. It is beyond the scope of this document to delve into these types of design choices and instead determine the feasibility and economics of operating such a system.

The proposed solution rests mostly on the ability to store and distribute heat energy in the form of a circulating loop, and the ability to provide basic solar heating to the loop. No significant research into new technology is needed as the concept is built on commonly available process equipment, re-imagined for use in residential home construction. The application shown here is basic, and other uses of a hot water loop may be appropriate such as heating of auxiliary spaces, de-icing structures, pool heating, etc. The more services that can be accommodated by a mostly sustainable hot energy source, the higher the value will be to the homeowner.

This means of heating would require significant upfront investment, and is therefore economically impractical for small, low cost homes given the current market. It could still be a model for larger houses, where decrease in energy usage is desired, or in housing where services are shared. If this type of heating concept were to be more universally implemented, cost decreases could be seen in many of the components to where installation on typical residences is economically practical.

2. ANALYSIS

2.1. Method of Analysis

The proposed solution will be analyzed to meet the needs of a typical American household in terms of heating in the winter and hot water usage. The analysis will consider preliminary concepts for components of the system based on commercially available items or preliminary performance parameters of the components based on basic calculations. The result will be a determination of process parameters and energy consumption for a typical residence.

2.1.1. Description of Residence

The residence used in the analysis will be a single story 3000ft^2 house with a square footprint and 10 foot vertical walls. The roof is minimally sloped and has an attic ventilated to the atmosphere. It will consist of a crawlspace that is perfectly ventilated to the outside air. Insulations values for the house will be based on DOE recommendations for Zone 4 and typical window and door insulation values. The result is:

- Vertical Wall Insulation value – R15 for cavity and R5 for sheathing
- Ceiling Insulation value – R30
- Floor Insulation value – R30
- Window Insulation value (triple paned) – R5
- Door Insulation value – R5

A total of two exterior doors with a 36inch width would be included in the house, and 10% of the total external surface area is covered with windows.

2.1.2. Description of Residence Usage

For purposes of the analysis, the residence is assumed to maintain an internal temperature of 72F during the winter. Hot water usage includes four showers per day, running a dishwasher once per day, and doing one load of laundry with hot water. A 10% additional hot water usage is used to address ancillary usage, such as cooking, cleaning, and hand washing.

2.1.3. Description of Analysis Conditions

The environmental condition is that of a house in a relatively harsh winter day in Zone 4. The exterior temperature is assumed to be 20F with a prevailing wind of 10mph.

2.2. Determination of Ventilation Heating Requirements

To determine the requirements for climate control heating, the total energy loss through the exterior surfaces is calculated. In a steady state condition, the energy input is equal to the energy loss to maintain a constant temperature.

2.2.1. Calculation of Exterior Vertical Wall Overall R Value

The vertical walls consist of insulated void spaces, studs, insulation sheathing, windows, and doors. The ratio of windows to overall surface area has been provided as 10%. Overall surface area is calculated by using the perimeter of the square house and the vertical wall height of 10ft. Both the overall surface area and the window surface area can be calculated.

$$Vertical\ Surface\ Area = \sqrt{3000ft^2} \times 4 \times 10ft = 2190.9ft^2$$
$$Window\ Surface\ Area = 2190.9ft^2 \times 10\% = 219.1ft^2$$

There are two exterior doors measuring 3ft wide. Exterior doors are traditionally 80 inches high resulting in:

$$Door\ Area = 2 \times 3ft \times 6.67ft = 40ft^2$$

The total sheathed portion of the exterior walls is calculated by subtracting the windows and doors from the overall vertical surface area:

$$Sheathed\ Vertical\ Surface\ Area = 2190.9ft^2 - (219.1ft^2 + 40ft^2) = 1931.8ft^2$$

The average R value for the vertical surfaces is determined by using the parallel resistance method. Firstly we just consider the vertical walls. They consist of the cavity R value and the stud R value. From the DOE, wood R value varies from 1.41 to .71 per inch. For convenience, a value of 1 per inch is used. The exterior wall is assumed to be a 6inch stud wall (5.5 inch length). Additionally it is assumed that the studs are spaced 24 inches on center.

From this information we can calculate the R value of a section of "clear" vertical exterior wall not including sheathing:

$$\frac{1}{R} = \frac{93.7\%}{15} + \frac{6.3\%}{5.5} = 7.4E - 2$$
$$R = 13.5$$

Sheathing is applied on top of this value for the total clear wall rating:

$$R = 13.5 + 5 = 18.5$$

Using the same parallel path approach, the entire vertical wall system (inclusive of doors and windows) can have its R value calculated:

$$\frac{1}{R} = \frac{1931.8/2190.9}{18.5} + \frac{219.1/2190.9}{5} + \frac{40/2190.9}{5} = .0713$$
$$R = 14$$

2.2.2. Calculation of Heat Transfer through Vertical Walls

The vertical walls have a heated interior surface with relatively stagnant air and an exterior surface subjected to low temperatures and prevailing winds. The heat transfer is calculated with the simplifying assumption that all exterior surfaces will have convective losses based on the prevailing wind as though they were flat plates parallel to the wind direction. This is conservative as stagnation would occur on two of the surfaces, or angled surfaces would be provided. It is also assumed that the exterior sheathing is relatively smooth, and leak tight, allowing the use of the flat plate model. Even so, a turbulent model will be used to capture the fact that some irregularities will occur to create turbulence along the surface. Radiation heat transfer, either from external heating from the sun or radiation internally is neglected, including solar heating through windows.

Convective Heat Transfer Coefficient

There are many correlations for convective heat transfer, but they all give similar answers. A fully turbulent model will be used (even if the Re number does not support) to increase the heat transfer rate. From Incropera and DeWitt, page 368, the Nusselt number in this situation is:

$$Nu = 0.0296 \times Re^{4/5} Pr^{1/3}$$

Using inputs from above (5mph, 20F), we can calculate the Re and substituting (note that υ taken from engineering toolbox and Pr estimated):

$$Re = \frac{\frac{14.7ft}{s} \times 54.8ft}{1.368E - \frac{4ft2}{s}} = 58.9E5$$

$$Nu = 0.0296 \times (58.9E5)^{4/5} \times 0.71^{1/3} = 6882$$

The average convection coefficient can be calculated as:

$$h = \frac{Nu \times k}{L} = \frac{6882 \times .01378BTU/(hrftF)}{54.8ft} = 1.73BTU/(hrft2F)$$

Combined Heat Transfer Calculation

The total resistance to heat flow can be combined for convection and conduction to make an overall heat transfer coefficient:

$$U = \frac{1}{\frac{1}{1.73BTU/(hrft^2F)} + 14\frac{ft^2hrF}{BTU}} = 6.86 \times 10^{-2}\ BTU/(hrft^2F)$$

Combined with the inside and outside temperatures and the total wall area, the total rate of heat loss can be calculated.

$$Vertical\ Wall\ Heat\ Transfer = (6.86 \times 10^{-2}\ BTU/(hrft^2F))(72F - 20F) \times 54.8ft \times 4 \times 10ft = 7819\ BTU/hr$$

2.2.3. Calculation of heat transfer through floor and ceiling

For the floor and ceiling, the air is stagnant and the convective heat transfer impact is considered negligible. This is conservative as there will be some temperature insulation effect formed by the boundary layer itself, but it is neglected. The area is the same for both the ceiling and the floor (3000ft2). This results in the following equation:

$$Floor\ and\ Ceiling\ Heat\ Transfer = \frac{72F - 20F}{30\ (Fft^2hr)/BTU} \times 3000ft^2 \times 2 = 10400BTU/hr$$

2.2.4. Total Heat Loss

The heat loss from all surfaces can be combined for the overall heat loss. This is then multiplied by the number of hours in the day for the total heat loss in a single day. This results in:

$$Total\ Residence\ Heat\ Loss = \left(\frac{7819BTU}{hr} + \frac{10400BTU}{hr}\right) \times 24hr = 437256BTU$$

2.3. Determination of Hot Water Requirements

Hot water is used primarily for showers, dishwashers, and washing machines. Each will be evaluated for total energy input based on heating water from mean ground temperature to the required temperature for the system. For purposes of this discussion, mean ground temperature is assumed to be 55F, which is lower than what is typically associated with zone 4 housing on which this residence is based.

2.3.1. Water Usage

Values for the average shower seems to be all over the map, but based on some quick research, a value of 20 gallons per shower seems to be bounding. Similarly, water usage for a dishwasher can vary, but assuming a relatively modern appliance, 6 gallons is bounding (more than an energy star appliance). There is even higher variability in washing machines, but 30 gallons seems to bound most modern machines (and is significantly higher than high efficiency washing machines). All of this water is assumed to be at a maximum “safe” water temperature for residential hot water at 105F.

2.3.2. Energy Usage

The total water consumption can be calculated based on the largest loads outlined above as:

$$Daily\ Hot\ Water\ Usage = 20gallon \times 4 + 6gallons + 30gallons = 116gallons$$

This water usage can be combined with the heat capacitance of water to determine the water heating energy requirements

$$Water\ Heating\ Requirement = \frac{1BTU}{lbmF} \times 116gal \times \frac{8.33lbm}{galH2O} \times (105F - 55F) = 48314BTU$$

A 10% increase in hot water is considered to capture any other uses for hot water resulting in a total energy requirement of 53145BTU.

2.4. Hot Water Loop Capacity Calculation

The size of the hot water system is dependent on the rate of heating energy to be delivered to the house. This considers both the total flow requirement, as well as the pressure differential required to transfer heat from the hot water loop to the serviced systems.

Appendices A and B perform preliminary design calculations on the heat exchangers required to provide the services noted. The resulting heat exchangers have the following characteristics:

- HVAC Heat Exchanger
  - Heating – 1886 BTU/hr
  - Hot Water Flow Rate – 1gpm
  - Differential Pressure – 1.28 psi
- Hot Water Heat Exchanger
  - Heating – 54231 BTU/hr
  - Hot Water Flow Rate – 4.5gpm
  - Differential Pressure – 2.8 psi

2.4.1. System Flow Requirement

To address the heat loss from the residence, heating must be provided at a rate of 18219 BTU/hr (on average). Adding a 15% increase to address air leaks, opening doors, etc, this would indicate provision of 20950 BTU/hr as appropriate for a comfortable environment. Based on the heating capabilities of the climate control systems described in Appendix A, a minimum of 12 heaters would be required, which would result in a system flow rate of 12 gpm if all were providing maximum heating.

At peak hot water usage, it is assumed that two showers are being taken as well as running the dishwasher and washing machine. A single heat exchanger can supply the total energy necessary for hot water in a day previously calculated in a single hour. Therefore, just to meet the peak flow requirements, a total of three heat exchangers working concurrently is considered. This allows for two showers, and the flow requirements for both the dishwasher and washing machine which are significantly lower than a shower. This results in a flow of 13.5 gpm.

2.4.2. System Pressure Drop Requirement

Pressure drop is determined by adding all the resistance within one most restrictive path. The design of the system is envisioned to have all services in parallel, so only the most restrictive element must be addressed. This must be added to the pressure drop caused by piping losses and all process equipment that operate the system.

The most restrictive service is the hot water heater at 2.8 psi. This includes 40 feet of tubing to connect to the main hot water header. This pressure drop would be required to drive the highest anticipated heat transfer demand and is therefore considered bounding.

Piping Pressure Drop

The hot water system would consist of a large main pipe-in-pipe design to minimize heating losses. The cross-section would consist of a central pipe surrounded by a larger pipe. The inner pipe would carry the supply hot water and return hot water would flow through the annular region. The entire assembly would be heavily insulated.

To minimize convective heat transfer from the piping and flow losses, large piping diameters would be chosen. The maximum system flow has been estimated to be approximately 25.5 gpm. Use of a 2 inch inner pipe would result in a pressure drop of approximately .786 psi per 100 feet at a velocity of 2.87 ft/sec (source of Crane for SCH40 steel pipe at 30 gpm). The outer pipe would be selected to have a similar hydraulic diameter, resulting in an outer pipe of 6 inches (SCH40). To be bounding, it is assumed that the orientation of the residence would require 100 feet of this pipe in pipe system.

The provision of four elbow fittings is considered appropriate to meet any routing requirements for the main piping. The pressure drop of the fittings and piping are added together as described in Crane:

$$(4)\ \left[20(.019)\frac{\left(2.87\,{}^{ft}/_{sec}\right)^2}{2\left(32.2\,{}^{ft}/_{sec^2}\right)}\frac{62\,{}^{lbm}/_{ft^3}}{144\,{}^{in^2}/_{ft^2}}\right]+0.786\,psi=.870psi$$

This value is for one leg. To capture the return leg, it must be doubled. For conservatism, assume that the total distribution pressure drop is 2 psi.

Pressure Drop of Heating Equipment

There are three other major pieces of equipment that will impart pressure drop. These are the heat exchangers to reclaim heat from the chill water compressor, solar heating, and resistance heating. It is beyond the scope of this white paper to define all of these components, but the pressure drop required to perform the heat exchange in the hot water heater is considered a good approximation, as it is not optimized and would be similar in structure to the first two, and likely more restrictive than a resistance heater.

The hot water heater in Appendix B has a pressure drop of .63 psi for a heating capacity of approximately 54231 BTU/hr. Combining the maximum momentary heating requirement of the 12 climate control units and three water heaters results in a total energy input of 185325 BTU/hr, that could be handled by approximately 3.4 water heaters. Therefore, the total pressure drop for heating equipment is estimated as 4 times a single water heater pressure drop at 2.5 psi.

2.4.3. Total Hot Water Pumping Requirements

The total pressure drop and flow requirement of the system define the operating conditions for the pump. Note that we are combining the maximum flow with the maximum pressure drop, which will define the worst case condition. At most times, only flow required to operate climate control units will be required, though the differential pressure may be retained the same to respond to demands on the system. This implies that a multiple pump system or variable speed system is appropriate, and any pumping power calculations are significantly higher than reality if only the highest flow rate is considered. For this reason, two pumping conditions are considered viable.

- High Pumping Power (approximately 20 minutes per day)
  - Flow rate of 25.5 gpm
  - Differential pressure of 7.3 psi
- Low Pumping Power (balance of day)
  - Flow rate of 12 gpm
  - Differential pressure of 7.3 psi

Pump Selection

A simple search for a hot water circulating pump resulted in finding the Grundfos MAGNA2 pump line. Model 32-100 would be able to meet the pumping conditions shown here. With variable speed, it could operate efficiently in both regimes. Based on the data sheet, in constant pressure operation, the high flow condition would result in approximately 145W of power, and the low flow condition in approximately 80W. Using these numbers at the duty cycle outlined above, a total of 1252800J of energy being used in a day, or 3.48 kWh.

2.5. Heat Balance of the Hot Water Loop

Heat input into the hot water loop is an energy balance of heat recovery from the chill water system, solar heating, electrical heating, pumping energy, and heat loss to atmosphere.

2.5.1. Heat Losses

Losses of heat to the ambient would occur from the piping and connected components. All piping and components would be heavily insulated. A comprehensive calculation of the losses would require more design information on all components, which is beyond the scope of this white paper. Therefore, the losses will be estimated as double the losses from the main distribution pipe. Note that losses between the distribution piping and services (climate control and hot water heaters) would be minimal as there is only flow on demand and the flow is a relatively small.

To calculate the losses to ambient from the piping system, we consider the pipe configuration discussed earlier, namely the pipe-in-pipe design. This design means that only the outermost pipe, carrying the return hot water is a heat loss boundary. Any heat lost from the inner pipe is recovered in the return leg. As a conservative approximation, we will assume that the return leg is at 200F, which is the maximum temperature of the hot water loop in an energy storage situation. This temperature will be applied directly to the pipe wall, which is also conservative as the resistance of the boundary layer is neglected. It will be assumed that 2 inches of insulation is provided on the exterior of the pipe, made up of fiberglass batting. Conductivity values for carbon steel and fiberglass were taken from engineering toolbox. The pipe is assumed to run in a location subject to ambient temperature of 20°F. With this information we can calculate the heat loss from the pipe. The first step is to calculate the overall heat transfer coefficient for a cylinder:

$$U = \frac{1}{\frac{\ln\left(\frac{6.625}{6.065}\right)\left(\frac{6.065in}{12\,{}^{in}/_{ft}}\right)}{20\,\left(\frac{BTU}{hr}\right)/(ft*F)} + \frac{\ln\left(\frac{8.625}{6.625}\right)\left(\frac{6.065in}{12\,{}^{in}/_{ft}}\right)}{.0231\,\left(\frac{BTU}{hr}\right)/(ft*F)}} = .173\left(\frac{BTU}{hr}\right)/(ft^2*F)$$

This is then used to calculate the total heating loss from 100 ft of piping:

$$Q_{loss} = \frac{.173\left(\frac{BTU}{hr}\right)}{ft^2*F} \times \frac{\pi}{4}\left(\frac{6.065in}{12\,{}^{in}/_{ft}}\right)^2 \times 100ft \times (200F - 20F) = 625\left(\frac{BTU}{hr}\right)$$

This loss will be doubled to account for losses at components and connections.

2.5.2. Heating Input

As discussed, heating input is from a solar hot water heater, regeneration from the chill water system, electrical heating, and pumping energy. For conservatism, the pumping energy input is neglected.

Regenerative Heating

The regenerative heating is capturing the waste heat from the compressed condenser coil of the chill water heat exchanger. The chill water system will mostly only service refrigeration loads during the winter. The refrigeration loads are hard to calculate as they are highly variable, and little information is provided. As the chill water system will be the ultimate heat sink for the refrigerator, all heat removed will result in heating of the chill water system.

As an initial estimate, assume a refrigerator of 15ft^3 with a freezer of 5ft^3. This is a rather standard size for purposes of determining total heat rejection. A quick scoping calculation shows that this comprises only around 10% of the energy usage of a single climate control unit, and is relatively inconsequential. As it is conservative, the addition to heating will be neglected.

Solar Heating

It is assumed that the solar heating array can be sized as necessary to provide adequate heating for the entire loop during daylight hours. For purposes of this study, it will be assumed that eight hours of usable light are available. Note that the services are rated for operation with a hot loop temperature of 140°F. This is easily exceeded by a solar heating unit, so excess heating capacity during the day can be stored as increased temperature within the hot water loop with control valve at services properly throttling heat exchanger flows. A maximum temperature of 200°F could be established during the daylight hours to be used during the night. The amount of heating that could be stored is based on the sizing of the hot water loop. At a minimum, 200ft of distribution piping is provided, with some water in the service heat exchangers, an expansion tank, etc. The piping represents a volume of 4.66 ft^3. An expansion tank similar in size to a large hot water heater would be 60 gallons, or approximately 8 ft^3. Assuming a total system volume of 13 cubic feet, the excess energy storage is calculated as:

$$Excess\ Energy = (8ft^3)\left(61.467\frac{lbm}{ft^3}\right)\left(1\frac{BTU}{lbm*F}\right)(200F - 140F) = 47944BTU$$

During the day, solar heating will account for all heat loads if the system is properly sized. Note that an increased volume of stored water would allow for this system to be operated purely by solar heating. It is also acknowledged that significant investment in solar water heating panels would be required to collect the required heating.

Electrical Heating

Having neglected all other sources of heating other than solar, electrical heating must provide the balance of the heating requirement. This would only be for the period when the solar heating is not providing all heat loads (16 hours). The remaining heating requirements can be calculated as:

$$437256BTU(110\%)\left(\frac{16hrs}{24hrs}\right) + 48314BTU(115\%)\left(\frac{16hrs}{24hrs}\right) - 47944BTU = 309751BTU$$

Over the 16 hours, that is 19359 BTU/hr of heating required from electrical resistance heaters. This can be converted to approximately 5.7kW. Assuming a heating efficiency of 95%, a total heating power of 6kW should be provided.

A. Climate Control Heat Exchanger

A.1. Problem Statement

A preliminary design is needed for a water to air heat exchanger that can meet the requirements as outlined in the Residential Hot Water System white paper. The climate control concept outlined consists of distributed air handlers, each containing their own heat transfer capabilities, fed by a hot water loop. To support development of the white paper, the anticipated load on the hot water system must be estimated, including the total flow requirement as well as the pressure drop caused by the heat exchanger.

A.1.1. Preliminary Climate Control Design Concept

The means of designing, installing, and operating distributed air handlers was considered to develop a preliminary design concept so that the boundaries of the design are established. The resulting physical design consists of a vertical duct that is housed entirely inside of a wall void containing fans, heat exchangers, registers, and controls. The resulting design constrains the space available, the types of airflows that could be anticipated, and the number of units that could be anticipated.

A.1.2. Constrained Design Parameters

Based on the design, it is shown that little horizontal space is available for the heat exchanger, therefore a heat exchanger consisting of several rows of u-tubes oriented horizontally is established as the basis for the design. This puts the tubes in crossflow and allows for increasing the rows of u-tubes to increase heat transfer area as necessary.

Due to the compact nature of the air handler, it is anticipated that there will be at least one air handler per space, and on average it would serve a space of 100-300 ft^2. Multiple air handlers would be anticipated in larger spaces.

A.1.3. Results

The analysis performed determined that a heat exchanger that contains 12 tube rows of ¼ inch tubing with a straight length of 16 inches would be able to provide approximately 1886 BTU/hr of heating with an average hot water temperature of 135F and an air side temperature of 72F. The hot water system would see a pressure drop of 1.28 psi from the heat exchanger and tube and fittings leading to the main hot water line. Hot water flow would be 1gpm to a single climate control unit in this condition. Printouts from the spreadsheet used to perform the calculations is at the end of this appendix.

Several simplifications were made in the analysis to provide a reasonable estimate without performing complex analyses. Of highest significance is that the heat exchanger consists of unfinned tubing, which lessens the potential heat transfer area. As the air side of the heat exchanger is limiting, there is potential for lowering the air side air flow and still allowing sufficient heat transfer. That is not fully explored in this preliminary study.

A.2. Heat Exchanger Analysis

The heat exchanger analysis consists of determining the amount of heat transfer that occurs from the hot water system to the air handler given set process conditions. The method used is to determine the following heat transfer coefficients as functions of controlled and derivative process inputs:

- The convective heat transfer on the inside of the tubes from the hot water system to the tubing wall.
- The conductive heat transfer through the tubing wall.
- The convective heat transfer from the outside of the tubing wall into the airstream of the air handler.

All calculations were performed in a spreadsheet in both English and SI units. These calculations were done in parallel to minimize the potential for errors and unit conversion considerations.

A.2.1. Input Parameters

Heat Exchanger Design Parameters

To establish the heat exchanger to be analyzed, inputs that define the heat exchanger design were selected and varied after the spreadsheet was completed to achieve the desired thermal performance. The following design aspects were used as inputs:

- Tube Outer Diameter (0.25 inches)
- Tube Wall Thickness (0.016 inches)
- Number of Tube Rows (12)
- Tube Straight Portion Length (16 inches)
- Air Duct Width (2 inches)
- Tube Internal Roughness (1.5E-4 feet)

Only the number of tube rows was varied from the initial values selected for these parameters so that an adequate heat exchange rate could be evaluated.

Process Parameters

The attributes of the air and water side process conditions were also established to allow for determination of the overall heat transfer of the heat exchanger. The following process conditions were established:

- Air Volume Served by Air Handler (300 ft^2)
- Number of Air Exchanges per hour (30)

- Hot Water Flow Rate (1 gpm)
- Ambient Air Temperature (72F)
- Average Temperature of Hot Water (135°F)

Only the number of air changes was varied to establish high enough flow for adequate heat exchange and control total duct flow rate.

A.2.2. Sources of Information

Three major sources were used to provide the necessary information and formulas to perform the calculation. This is not a formal calculation, so each usage of sources in not cited, but in general the information was derived as follows:

- Thermophysical properties for air and water were from www.engineeringtoolbox.com. This allowed for a single location where temperature dependent properties could be easily found. The spreadsheet notes at what temperature properties were obtained.
- Heat transfer formulas came from “Introduction to Heat Transfer”, by Incropera and DeWitt. The formula number will be cited in sections below.
- Flow conditions within the pipe (head loss calculation) were calculated with information from “Flow of Fluids”, published by Crane as Technical Paper No. 410.

It is possible that some conversion factors and properties were derived from other available textbooks that were handy at the time, but all properties used were compared with the online tool cited and are similar. Similarly conversion factors are expected to be similar and the source is inconsequential for this level of analysis.

A.2.3. Calculation Process

The calculation was performed on a spreadsheet. Initially all of the inputs previously discussed were entered. They were then manipulated as discussed in the sections below.

Process parameter calculations

These calculations use the process parameters combined with the heat exchanger design to establish local conditions.

- Duct Volumetric Flow. Calculated using the air exchange rate and size of conditioned space.
- Average Air Velocity. Uses the volumetric flow combined with the duct flow area, which is assumed to be the straight tube length multiplied by the duct width, to establish velocity outside of the heat exchanger bundle.
- Maximum Air Velocity. The cross-sectional area of the heat exchanger tubes is subtracted from the duct cross-sectional area to establish the maximum flow that will be seen in the heat exchanger section. This assumes the u-bend is outside of flow and that the heat exchanger tubes are aligned.

- Air Maximum Reynolds Number. Uses the maximum velocity, tube outer dimensions, and fluid properties to calculate the Reynolds Number.
- Mean Velocity of Water. This is calculated by taking the internal area of the tubing and number of tubes combined with the total flow to determine average velocity through each tube.
- Water Reynolds Number. The water mean velocity and water properties and tube inner dimensions are combined to calculate the Reynolds Number.
- Friction Factor. The friction factor is calculated with a formula from Crane in Appendix A based on fully developed turbulent flow using roughness and internal diameter. The Reynolds number is relatively low for this approach, but the result is conservative relative to a laminar friction factor.
- Water dP. The pressure drop through the tubes can be calculated directly with the information developed, using the Darcy formula, tube dimensions and friction factors, and the form factor in Crane Appendix A for a close pattern return bend.

Water Side Convection Calculation

To calculate water side convection, the Nusselt number had to be determined based on empirical relationships. The Reynolds number indicated turbulent flow, near the transition region. For this reason, the formulation in Incropera and DeWitt in equation 8.63 was used to allow for maximum flexibility. Initially the Reynolds number was within range of the correlation, but increasing the number of tube rows put is just below the range of applicability. It is still considered an appropriate usage since any sort of fouling will ensure flow is turbulent and the final Reynolds number is very close to the stated range of applicability.

The Nusselt number is used to determine the convective heat transfer coefficient, which is multiplied by the internal tube area and inverted to determine the resistance to heat transfer.

Air Side Convection Calculation

The Nusselt number for the air side was calculated using equation 7.46 in the Incropera and DeWitt text. Initially, equation 7.57 was considered, but the constants used for the correlation indicate that for the most likely flow regime, the tubes should be treated as single cylinders. Because of this, the correction factors for less than 20 tubes was also not used. As a further simplification, the ratio of surface to freestream air temperature Prandtl term was neglected as it has little consequence. Instead, a Prandtl number of .707 was used.

The resulting Nusselt number was used to calculated the convective heat transfer coefficient, and along with the area of the outside of the tube the resistance to heat flow.

Determination of Heat Transfer

The heat transfer from the hot water to the air handler airstream was calculated by using the differential temperature combined with the thermal resistance of the two convective layers and the tube wall. The formulation was of the form:

$$Q = \frac{T_{Water} - T_{Air}}{\frac{1}{hA_{Water\ Side}} + \frac{1}{hA_{Air\ Side}} + \frac{Tube\ Wall}{kA_{Stainless\ Tube}}}$$

The number of tube rows and the number of air exchanges were iterated until it was deemed that sufficient heat transfer resulted from the heat exchanger.

Water Pressure Drop

The water pressure drop from the tubes themselves was calculated previously, but there are additional losses. Losses to be added to this pressure drop are for entrance and exit of the tubes from plenums, control valve pressure drop, and flow losses between the main hot water piping to the heat exchanger.

Entrance and exit of the u-tubes from the supply and return header are considered abrupt area changes. Specifically, the headers are envisioned as larger diameter pipes with the u-tubes exiting perpendicular to the direction of flow. This can be best modeled as an entrance and exit with a sharp corner from a plenum. The entrance and exit losses in Appendix A of Crane of 0.5 and 1 respectively are used to describe this form loss.

Flow losses in the tubing are calculated by establishing the length of the supply tube, its inner diameter, internal roughness, ID of the supply tube, and fittings. The loss from the tubing itself was calculated as fL/D. The loss was multiplied by two to account for the supply and return tubing. The fittings were estimated to contain three elbows on both the supply and return. The value for form loss for an elbow was taken from Crane, Appendix A.

The control valve pressure drop was calculated using information from Crane, Appendix A for a diaphram style valve. It is assumed that it would be installed in the return line tubing.

Tubing velocity was calculated the same way it was calculated for heat exchanger tubing and then used to calculate the dynamic pressure term. The dynamic pressure was multiplied by the summation of all of the flow losses, then added to the pressure drop caused by a coil. This resulted in the final pressure drop caused by the entire heat exchanger system.

Hx Design Parameters				
Variable Inputs	**Value**	**Units**	**Value**	**Units**
tube OD	0.25	inches	0.00635	m
Wall Thickness	0.016	inches	0.0004064	m
# tube rows	12	N/A	12	
Tube Length	16	inches	0.4064	m
Duct Width	2	inches	0.0508	m
Tube Spacing	1	inches	0.0254	m
Tube Roughness	0.00015	ft	0.00004572	m

Process Parameters				
Variable Inputs	**Value**	**Units**	**Value**	**Units**
Air Volume Ex	300	ft3	8.495053978	m3
# Exchanges/hr	30	N/A	30	N/A
Water Flow	1	gpm	6.30902E-05	m3/s
Air Temp In	72	F	22.22222222	C
Water Temp Average	135	F	57.22222222	C

Properties				
Properties	**Value**	**Units**	**Value**	**Units**
Air Kinematic (72F)	0.0001643	ft2/s	0.00001527	m2/s
Water Kinematic (135F)	0.000005316	ft2/s	4.939E-07	m2/s
Water Density	61.467	lbm/ft3	984.5	kg/m3
Air Conductivity	0.01504	(BTU/hr)/(ft*F)	0.02604	W/(m*K)
Water Prandtl (137.5F)	3.095		3.095	
Water Conductivity (140	0.3761	(BTU/hr)/(ft*F)	0.65091	W/(m*K)
Stainless Conductivity	8.609518	(BTU/hr)/(ft*F)	14.9	W/(m*K)
Water Cp(135F)	1	BTU/(lb*R)	4190	J/(kg*K)
Air Density (72F)	0.07459	lbm/ft3	1.195	kg/m3
Air Cp(72F)	0.2402	BTU/(lb*R)	1006	J/(kg*K)

Calculations				
Quantity	Value	Units	Value	Units
Duct Vol Flow	2.5	CFS	0.070792116	m/s
Air Average Vel	11.25	ft/s	3.429	
Max Air Vel	15	ft/s	4.572	
Air Re,max	1902.008521		1901.257367	
Mean Velocity Water	0.716300689	ft/s	0.21832845	m/s
Water Re	2807.172878		2807.016921	
Friction Factor	0.035562656		0.035562656	
Water dP	0.023800133	psi	164.2106695	Pa
Water Side Convection				
Nu,D	12.29927833		12.29821692	
h,D	254.6289127	(BTU/hr)/(ft2*F)	1445.682362	W/(m2*K)
Tube Area	1.826312529	ft2	0.169669986	m2
Resistance	0.00215039	F/(BTU/hr)	0.004076825	K/W
Air Side Convection				
Nu,D	21.22198514		21.21695606	
h,D	15.32057551	(BTU/hr)/(ft2*F)	87.00622613	W/(m2*K)
Tube Area	2.094395102	ft2	0.194575672	m2
Resistance	0.031164941	F/(BTU/hr)	0.059069205	K/W
Determine Heat Transfer				
Heat Transfer	1886.220291	BTU/hr	552.8633311	W
Water Pressure Drop				
K Entrance	0.5		0.5	
K Exit	1		1	
Length Tubing run	40	ft	12.192	m
ID Tube	0.485	in	0.012319	m
Roughness	0.00015	ft	0.00004572	m
F tube	0.027802406		0.027802406	
K Tube	55.03156659		55.03156659	
K Bends	5.004433087		5.004433087	
K Valve	1.084293836		1.084293836	
K summation	62.62029352		62.62029352	
Tube Velocity	1.736625305	ft/s	0.529323393	m/s
Re	13203.27428		13202.54075	
dynamic pressure	0.019989683	psi	137.9202069	Pa
Total Losses	1.275559951	psi	8800.814506	Pa
Temperature Changes				
dT Water	3.825876116	F	2.124350451	C
dT Air	11.69758724	F	6.496313925	C

B. Hot Water Heat Exchanger

B.1. Problem Statement

A preliminary design is needed for a water to water heat exchanger that can meet the requirements as outlined in the Residential Hot Water System white paper. The hot water supply concept outlined consists of distributed point water heaters fed by the hot water loop and providing on-demand heating of cold potable water. To support development of the white paper, the anticipated load on the hot water system must be estimated, including the total flow requirement as well as the pressure drop caused by the heat exchanger.

B.1.1. Preliminary Water Heater Design Concept

A basic concentric tube hot water heater is envisioned that has the cold potable water entering from one end and hot loop water entering an outer jacket from the opposite end (opposed flow). The pipe for the cold potable water would be smooth and consistent with regular plumbing to minimize pressure drop caused by heating. This results in a long, skinny assembly that is installed underneath the floor served by hot water supply.

B.1.2. Constrained Design Parameters

There are few constraints on the size of the heat exchanger as it would be located in a void space, but the design would require that the cold water section minimize pressure drop, necessitating standard size piping that would be found in the residential installation. Similarly, the outer pipe would be of a standard size to simplify the design. The length of the heat exchanger would be determined based on the heat transfer requirements.

Process conditions have been defined in the white paper. The required outlet temperature of the potable hot water, as well as the anticipated inlet temperature are known. Additionally the supply temperature of the hot water loop is defined. Flow rate for the hot potable water is defined based on the 20 gallon shower for 10 minutes (2gpm). Other services are anticipated to have a lower required flow rate.

For purposes of performing the calculation, hot loop heat exchanger outlet temperature and total flow are estimated, but are iterated during the design process to meet heat exchange requirements.

B.1.3. Results

The analysis performed determined that a heat exchanger with a length of approximately 12 feet would be sufficient to supply 110F hot potable water at 2gpm. The resulting hot water loop outlet temperature would be 116F at a flow of 4.5gpm. Accounting for fittings, supply tubing, and the heat exchanger, the total pressure drop caused by installation of the heat exchanger to the main hot loop header is approximately 2.8 psi.

Several simplifications were made in the analysis to provide a reasonable estimate without performing complex analyses. Of highest significance is that the heat exchanger consists of unfinned tubing, which lessens the potential heat transfer area. Fins on the potable water side are likely infeasible, but addition of fins on the hot water loop size would decrease overall heat exchanger length with minimal increase in flow resistance. That is not fully explored in this preliminary study.

B.2. Heat Exchanger Analysis

The heat exchanger analysis consists of determining the amount of heat transfer that occurs from the hot water system to the hot potable water given set process conditions. The method used is to determine the combined heat transfer coefficient for the heat exchanger and use that to iteratively solve for hot loop outlet temperature and heat exchanger length. The following heat transfer coefficients are developed independently and combined to establish the overall heat transfer coefficient:

- The convective heat transfer on the inside of the tube from the potable water system to the tubing wall.
- The conductive heat transfer through the tubing wall.
- The convective heat transfer on the outside of the tubing wall from the hot water system into the potable water system.

All calculations were performed in a spreadsheet in both English and SI units. These calculations were done in parallel to minimize the potential for errors and unit conversion considerations.

B.2.1. Input Parameters

Heat Exchanger Design Parameters

To establish the heat exchanger to be analyzed, inputs that define the heat exchanger design were selected and varied after the spreadsheet was completed to achieve the desired thermal performance. The following design aspects were used as inputs:

- Tube Outer Diameter (0.675 inches, 3/8SCH10S piping)
- Tube Wall Thickness (0.065 inches)
- Inner Diameter of the Outer Pipe (1.049 inches, 1SCH40 piping)
- Tube Internal Roughness (1.5E-4 feet)

Process Parameters

The attributes of the potable and hot water side process conditions were also established to allow for determination of the overall heat transfer of the heat exchanger. The following process conditions were established:

- Potable Water Inlet Temperature (55F)

- Potable Water Outlet Temperature (110F)
- Hot Water Inlet Temperature (140F)
- Hot Water Outlet Temperature (Variable, discussed in calculation)
- Potable Water Flow Rate (2gpm)
- Hot Water Flow Rate (4.5gpm)

The hot water outlet temperature was varied during the calculation to establish the appropriate log mean temperature difference to enable adequate heat exchange. Additional details are discussed in the calculation.

B.2.2. Sources of Information

Three major sources were used to provide the necessary information and formulas to perform the calculation. This is not a formal calculation, so each usage of sources in not cited, but in general the information was derived as follows:

- Thermophysical properties for air and water were from www.engineeringtoolbox.com. This allowed for a single location where temperature dependent properties could be easily found. The spreadsheet notes at what temperature properties were obtained.
- Heat transfer formulas came from "Introduction to Heat Transfer", by Incropera and DeWitt. The formula number will be cited in sections below.
- Flow conditions within the pipe (head loss calculation) were calculated with information from "Flow of Fluids", published by Crane as Technical Paper No. 410.

It is possible that some conversion factors and properties were derived from other available textbooks that were handy at the time, but all properties used were compared with the online tool cited and are similar.

B.2.3. Calculation Process

The calculation was performed on a spreadsheet. Initially all of the inputs previously discussed were entered. They were then manipulated as discussed in the sections below.

Convective Coefficients

The convective coefficients for the potable water side and the hot water side were performed in a similar manner. The process was to use the Dittus-Boelter equation (8.6 in Incropera and DeWitt) for both tubes. Due to geometry, its applicable was correction factors were different.

For the inner tube, the application of D-B was straightforward to determine the Nusselt number based on the internal diameter. As the overall heat transfer coefficient would be based on the area of the outer diameter, a correction was made based on the inner and outer perimeter to prevent overestimating heat transfer when using the outer surface area.

For the outer tube, the D-B equation provides a concept of how convective heat transfer relates to the Reynolds number of smooth tubes, but the applicable of the Reynolds number is not as obvious in this case. It was determined that the hydraulic diameter would be the appropriate characteristic length as its combination with flow velocity represents the dynamic forces at work in the fluid. The hydraulic diameter was calculated as:

$$D_h = \frac{4A_{flow}}{P_{wetted}}$$

where the total flow area and wetted perimeter are used. Using the resulting hydraulic diameter to calculate Reynolds number, D-B was applied to calculate the convective heat transfer coefficient.

Overall Heat Transfer Coefficient

The overall heat transfer coefficient was defined by combining the convective heat transfer coefficients and the conductivity of the inner tube wall.

$$U = \frac{1}{\frac{1}{h_{Potable\ Water}} + \frac{1}{h_{Hot\ Water}} + \frac{Tube\ Wall}{k_{Stainless\ Tube}}}$$

Heat Exchanger Calculation

The temperatures defined for the process conditions were used to calculated the log mean temperature difference across the counterflow heat exchanger. Note that the hot water outlet temperature was an initial guess in the process parameters.

The required heat transfer to raise the potable water the required amount was determined by simply using the defined flow rate, heat capacity of water, and temperature differential. This was then used to determine the resulting hot water outlet temperature. At this point, the initial guess of hot water outlet temperature may not match the result of this calculation. Therefore a goal seek function is used to iterate the calculation until the desired temperature is established for temperature outlet. Note that this will impact the log mean temperature difference.

After convergence of the outlet temperature, the area required for heat transfer is defined which results in the total length of the heat exchanger.

Pressure Drop Calculation

The pressure drop on the hot water side is required to provide information on the white paper. It is calculated as a combination of pressure drop within the heat exchanger and the piping and fittings leading to the heat exchanger.

Pressure drop for the heat exchanger is complicated on the hot water side due to the annular nature of the flow space. The hydraulic diameter was used in conjunction with the Darcy formula and calculated friction factor based on the pipe roughness previously defined.

The pressure drop for supply tubing is calculated by establishing the length of the supply tube, its inner diameter, and internal roughness. The velocity of the tubing was calculated by using the flow area and the total hot water flow, previously defined. The loss from the tubing itself was calculated as fL/D. The loss was multiplied by two to account for the supply and return tubing. Entrance and exit losses to the heat exchanger were sourced from Crane, Appendix A.

Fittings were estimated to contain three elbows on both the supply and return. The control valve pressure was assumed to be a diaphragm style valve. It is assumed that it would be installed in the return line tubing. The value for form loss for an elbow and valve was taken from Crane, Appendix A.

Tubing velocity was used to calculate the dynamic pressure term. The dynamic pressure was multiplied by the summation of all of the flow losses, then added to the pressure drop caused by the heat exchanger. This resulted in the final pressure drop caused by the entire heat exchanger system.

Process Parameters				
Variable Inputs	Value	Units	Value	Units
Tc,i	55	F	12.77777778	C
Th,i	140	F	60	C
Th,o	115.5555556	F	46.41975309	C
Tc,o	110	F	43.33333333	C
Flow Cold	2	gpm	0.00012618	m3/s
Flow Hot	4.5	gpm	0.000283906	m3/s

Heat Exchanger Characteristics				
Variable Inputs	Value	Units	Value	Units
OD,inner	0.675	in	0.017145	m
wall,inner	0.065	in	0.001651	m
ID,outer	1.049	in	0.0266446	m
Pipe Roughness	0.00015	ft	0.00004572	m

Properties				
Quantity	Value	Units	Value	Units
Water kinematic (100F)	7.39E-06	ft2/s	6.87E-07	m2/s
Water kinematic (130F)	5.55E-06	ft2/s	5.16E-07	m2/s
Water,Cp	1	(BTU)/(lb*R)	4.18E+03	J/(kg*k)
Water,K (120F)	0.3694	(BTU/hr)/(ft*F)	6.39E-01	W/(m*K)
Water,dynamic (100F)	1.65E+00	lbm/(ft*hr)	6.82E-04	Pa*s
Water,dynamic (130F)	1.23E+00	lbm/(ft*hr)	5.09E-04	Pa*s
Water,Pr (100F)	4.47E+00		4.46E+00	
Water,Pr (130F)	3.33E+00		3.32E+00	
Stainless Conductivity	8.609518	(BTU/hr)/(ft*F)	14.9	W/(m*K)
Water Density	61.467	lbm/ft3	984.5	kg/m3

Convective Coefficients				
Quantity	Value	Units	Value	Units
Inner wall H				
Inner Velocity	2.750594645	ft/s	0.838381248	m/s
Re,inner	1.69E+04		1.69E+04	
Nu,D	2.48E+02		2.47E+02	
h,inner	2.01E+03	(BTU/h)/(ft2F)	1.14E+04	W/(m2K)
ID/OD Ratio	0.807407407		0.807407407	
h,in,corr	1.63E+03	(BTU/h)/(ft2F)	9.22E+03	W/(m2K)
Outer Wall H				
Outer Flow Area	0.003516708	ft2	0.000326713	m2
Outer Velocity	2.85097397	ft/s	0.868976866	m/s
Wetted P	0.451342145	ft	0.137569086	m
Re,outer	1.60E+04		1.60E+04	
Nu,D	1.77E+02		1.77E+02	
h,outer	2.10E+03	(BTU/h)/(ft2F)	1.19E+04	W/(m2K)

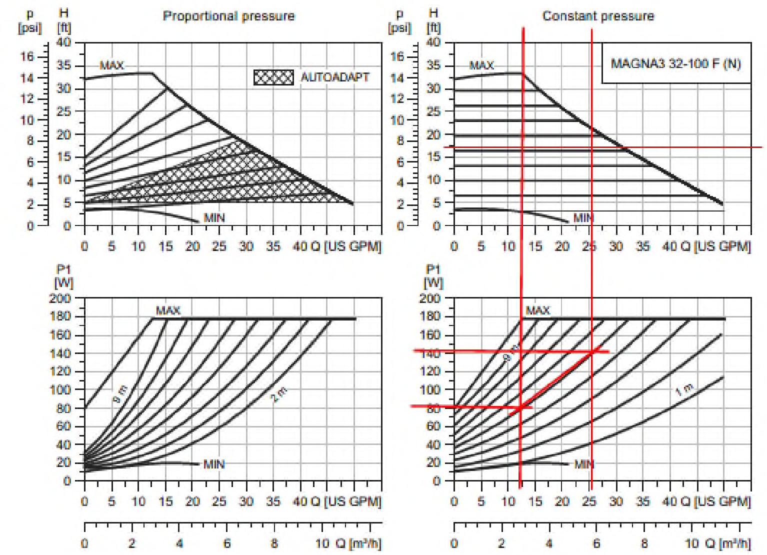
Overall U (Relative Inner OD Area)				
Quantity	**Value**	**Units**	**Value**	**Units**
Rinner	6.15E-04	(ft2F/(BTU/hr)	1.08E-04	m2K/W
Router	4.77E-04	(ft2F/(BTU/hr)	8.42E-05	m2K/W
Rwall	0.000629149	(ft2F/(BTU/hr)	0.000110805	m2K/W
Rtotal	1.72E-03	(ft2F/(BTU/hr)	3.03E-04	m2K/W
U	5.81E+02	(BTU/h)/(ft2F)	3.30E+03	W/(m2K)

Heat Exchanger Calculation				
Quantity	**Value**	**Units**	**Value**	**Units**
T1	30	F	16.66666667	C
T2	60.55555556	F	33.64197531	C
delT,lm	43.50388503	F	24.16882502	R
Establish Th,out				
delT,cold	55		30.55555556	
Q Req	54231.82188	BTU/hr	1.59E+04	W
Thot,out	115.5555556		4.64E+01	
goal	0		0	
Area OD	2.145367039	ft2	0.199169763	m2
Length	12.14029401	ft	3.69773722	

Pressure Drop Calculation				
Quantity	**Value**	**Units**	**Value**	**Units**
Heat Exchanger Pressure Drop				
Hydraulic Diameter	0.031166667	ft	0.0094996	ft
f Heat Exchanger	0.030019766		0.030019766	
dynamic Pressure	0.053874117	psi	371.7082108	Pa
dP Heat Exchanger	0.62997935	psi	4343.50358	Pa
Tubing and Fittings				
Tubing Diameter	0.681	in	0.0172974	m
Tubing Roughness	0.000005	ft	0.000001524	
f tubing	0.011696478		0.011696478	
Tubing Velocity	3.963764941	ft/s	1.208155554	m/s
Tubing Dynamic Pressure	0.10413805	psi	718.5077125	Pa
Tubing Run	40	ft	12.192	m
Tubing Loss	16.4884268		16.4884268	
Fitting Loss	2.105365997		2.105365997	
Entrance Loss	0.5		0.5	
Exit Loss	1		1	
Control Valve	0.456162633		0.456162633	
Total Losses	20.54995543		20.54995543	
dP Tubing	2.140032289	psi	14765.30147	psi
Total dP	2.770011638	psi	19108.80505	psi

C. Pump Data Sheet

MAGNA3 32-100 F (N) 1 x 115-230 V, 50/60 Hz

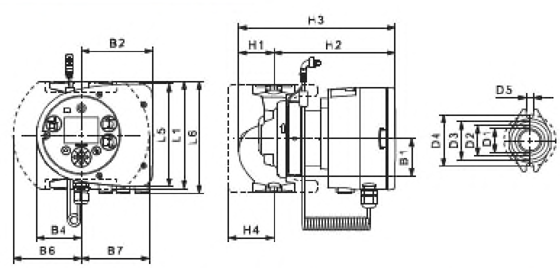


Speed	$I_{1/1}$ [A] 115 V	P1 [W] 115 V	$I_{1/1}$ [A] 230 V	P1 [W] 230 V
Min.	0.29	9.7	0.09	9
Max.	1.61	178.3	1.47	180

The pump incorporates overload protection.

System pressure:	Max. 175 psi (12 bar)
Liquid temperature:	14 to 230 °F (-10 °C to +110 °C)
Also available with:	Stainless-steel pump housing, type N
Specific EEI:	0.18

Net weights [lbs (kg)]	Gross weights [lbs (kg)]	Ship. vol. [ft³ (m³)]
12.3 (5.5)	14.4 (6.5)	0.46 (0.014)



Pump type	Dimensions [in (mm)]								
	L1	L5	L6	B1	B2	B4	B6	B7	D1
MAGNA3 32-100 F (N)	6.50 (165)	6.23 (158)	6.62 (168)	2.29 (58)	4.38 (111)	2.72 (69)	4.18 (106)	4.18 (106)	1.26 (32)
	D2	D3	D4	D5	H1	H2	H3	H4	
	1.82 (46)	2.29 (58)	3.15 (80)	0.46 (11.5)	2.13 (54)	7.37 (187)	9.49 (241)	2.76 (70)	

For product numbers, see page 7.