THE EFFECTIVENESS OF A GROUND-COUPLED

HEATING AND COOLING SYSTEM

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ABSTRACT
A municipality water reticulation R22 ground-coupled reversible heat pump, was investigated as an alternative to conventional air source systems. The investigation was conducted by developing analytical models that were used for the design of a ground-coupled reversible heat pump and a conventional, also reversible air-to-air system. The models were verified with a commercially available computer program as well as with measurements on the two systems. The results indicate that a ground-coupled reversible heat pump system can provide a cost effective alternative to the more conventional air-to-air systems.

KEY WORDS

air-conditioning	condenser	evaporator	ground-coupled
heat exchanger	heat transfer	heat pump	pressure loss

INTRODUCTION

Ground source heat pumps have started to appear on the markets in the domestic and commercial sectors. These units consist of a reversible vapour-compression cycle linked to a closed loop heat exchanger buried in the soil (ASHRAE 1995a). There are many different types of ground source systems available as discussed in ASHRAE (1996). The most conventional one is the water-to-air heat pump which circulates water or a water-antifreeze solution through a liquid-to-refrigerant heat exchanger and a buried thermoplastic piping network which is the ground source.

A method that can be implemented so as to decrease the capital cost of a ground source heat pump is to connect the liquid-to-refrigerant heat exchanger directly to the municipality water supply making the ground water loop obsolete. The implication is that in winter in the heating mode heat will be extracted from the municipality water supply and in summer in the cooling mode heat will be added to the municipality water supply. This is not a new idea, utilising municipality water as a ground source is one of many methods discussed and rated by ASHRAE (1996). They have found that this is an excellent source, the only limitation is the regulation that enforces double wall heat exchangers to prevent possible refrigerant contamination to water and ground sources thus increasing cost.

The aim of this study is therefore to investigate the effectiveness at normal operating conditions of a municipality ground-coupled heat pump system in both the heating and cooling mode which can be accomplished by reversing the direction of the vapour-compression cycle through the two heat exchangers, namely the evaporator and condenser. The system effectiveness will be compared to that of the more conventional air-to-air system which is also reversible. A tool which would aid in the design of and evaluation of these two systems would be useful. Therefore, two models have been developed to predict the performances of an air source and a ground source system. These models predict the heat exchanger sizes, capacity and power requirements for specified operating conditions. With the two models the two systems are also designed, built, tested and compared analytically and experimentally. A comparison is therefore given between the predicted and measured results.

AIR AND GROUND SOURCE SYSTEMS

The air source and ground source systems that will be described are all based on the basic vapour-compression cycle through the evaporator and condenser, schematically shown with negligible pressure drops in Figure 1. In this cycle heat is absorbed at the evaporator (Q_L) which is a heat exchanger between stations six and one while heat is released at the condenser (Q_H) which is also a heat exchanger between stations two and five. The compressor input power (W_c) added to the refrigerant is between stations one and two while expansion occurs through a capillary tube between stations five and six. In practise the cycle is reversed to make either heating or cooling possible at any of the two heat exchangers. This is accomplished by a reversing valve. With the reversing valve heat exchangers A and B can both operate as an evaporator or condenser depending on the need for either cooling or heating on the inside of a zone as shown in Figure 2.

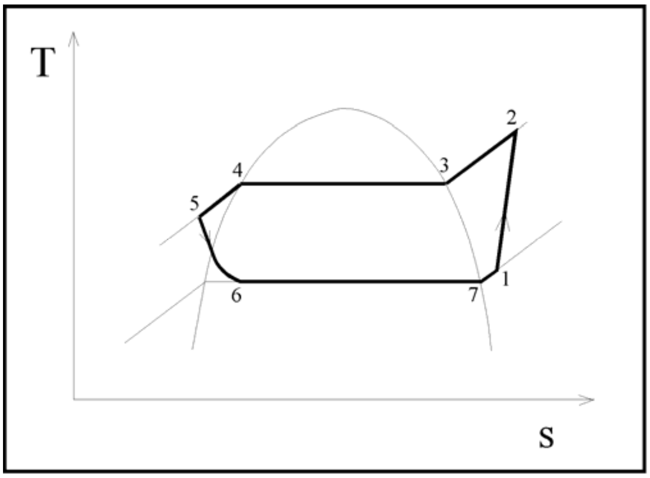

Figure 1. Schematic representation of the vapour-compression cycle on a temperature-entropy diagram.

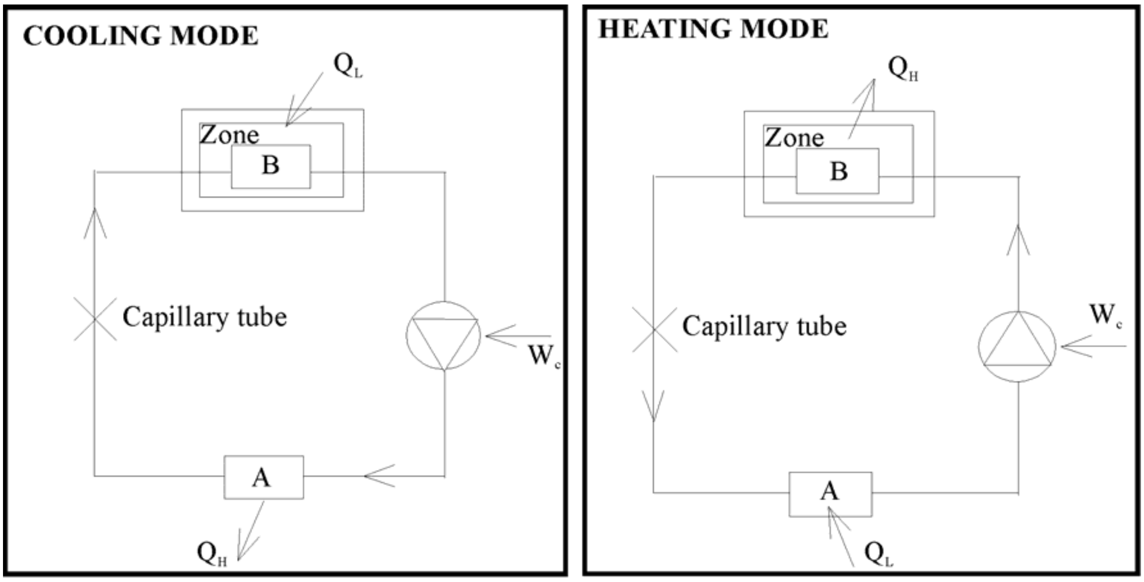

Figure 2. Schematic representation of the cooling and heating operating modes.

	AIR SOURCE		**GROUND SOURCE**	
HEAT EXCHANGER	COOLING MODE	HEATING MODE	COOLING MODE	HEATING MODE
A	Condenser Air-Cooled	Evaporator Air-Heated	Condenser Water-Cooled	Evaporator Water-Heated
B	Evaporator Air-Heated	Condenser Air-Cooled	Evaporator Air-Heated	Condenser Air-Cooled

Table 1: Heat exchanger identification for air and ground source reversible heat pumps.

In the case of an air source or ground source reversible heat pump system the only differences between the two systems is the type of heat exchangers used on the outside of the zone to be heated or cooled as can be observed from Table 1. In the case of an air source system this heat exchanger will be an air-to-refrigerant heat exchanger which should be air-cooled in the cooling mode and air-heated in the heating mode. In the case of a ground source system it will be a water-to-refrigerant system which should be water-cooled in the cooling mode and water-heated in the heating mode. For both systems the heat exchanger used on the inside of the zone to be air-conditioned is a refrigerant-to-air heat exchanger. In the heating mode it will be air-cooled and in the cooling mode it will be air-heated.

DESIGN METHODOLOGY

Since the basic elements of the air source and ground source systems differ only in the types of heat exchangers used the same design methodology is used for both. The two systems are designed by the procedure outlined in Figure 3. The procedure starts by selecting a compressor that will give an adequate cooling and heating capacity at selected evaporating and condensing temperatures. This enables the determination of the compressor characteristics and the discharge conditions of the compressor. Thereafter, the condenser size is determined. The condenser size refers either to the pipe diameters and length of a tube-in-tube heat exchanger or the face area of an air-coil. These values are first determined by assuming no pressure drop through the condenser. Once the condenser size is known the pressure drop is determined and the size is recalculated. This procedure is followed until the condenser size converges and the condenser outlet conditions are known.

The capillary size and diameter are calculated before the evaporator size is determined. The evaporator size is predicted in the same manner as the condenser. However, each time the calculated pressure drop through the evaporator is added to the inlet pressure of the evaporator to ensure that the evaporator temperature is equal to the originally selected value. The implication is that the capillary tube also has to be redesigned until the evaporator size converges. Once all the sizes of the heat exchangers and refrigerant mass flow are known the coefficients of performance can be calculated for the heating and cooling modes of the two systems.

Each one of the primary components shown in the flow chart of Figure 3 namely the compressor, condenser, capillary tube and evaporator are discussed in more detail in the next few sections with reference to Figure 1. After all the components have been discussed this section will be concluded on the methodology of determining the different coefficients of performance.

Compressor

The compressor is the "heart" of any vapour-compression system and in practise the capacity is dependent on the compressor size as well as on the evaporating (T_7) and condensing temperatures (T_3). Therefore, with the design the compressor is first selected. For this study a two cylinder hermetically sealed reciprocating compressor with a displacement of 0.0724 litres per revolution and a nominal heating capacity of 10 kW at ARI conditions of 7.2 ^{o}C evaporating temperature and 54.4 ^{o}C condensing temperature was selected. The compressor curves as supplied by the manufacturer and are valid for a superheat of 11.11 ^{o}C and a subcooling value of 8.33 ^{o}C.

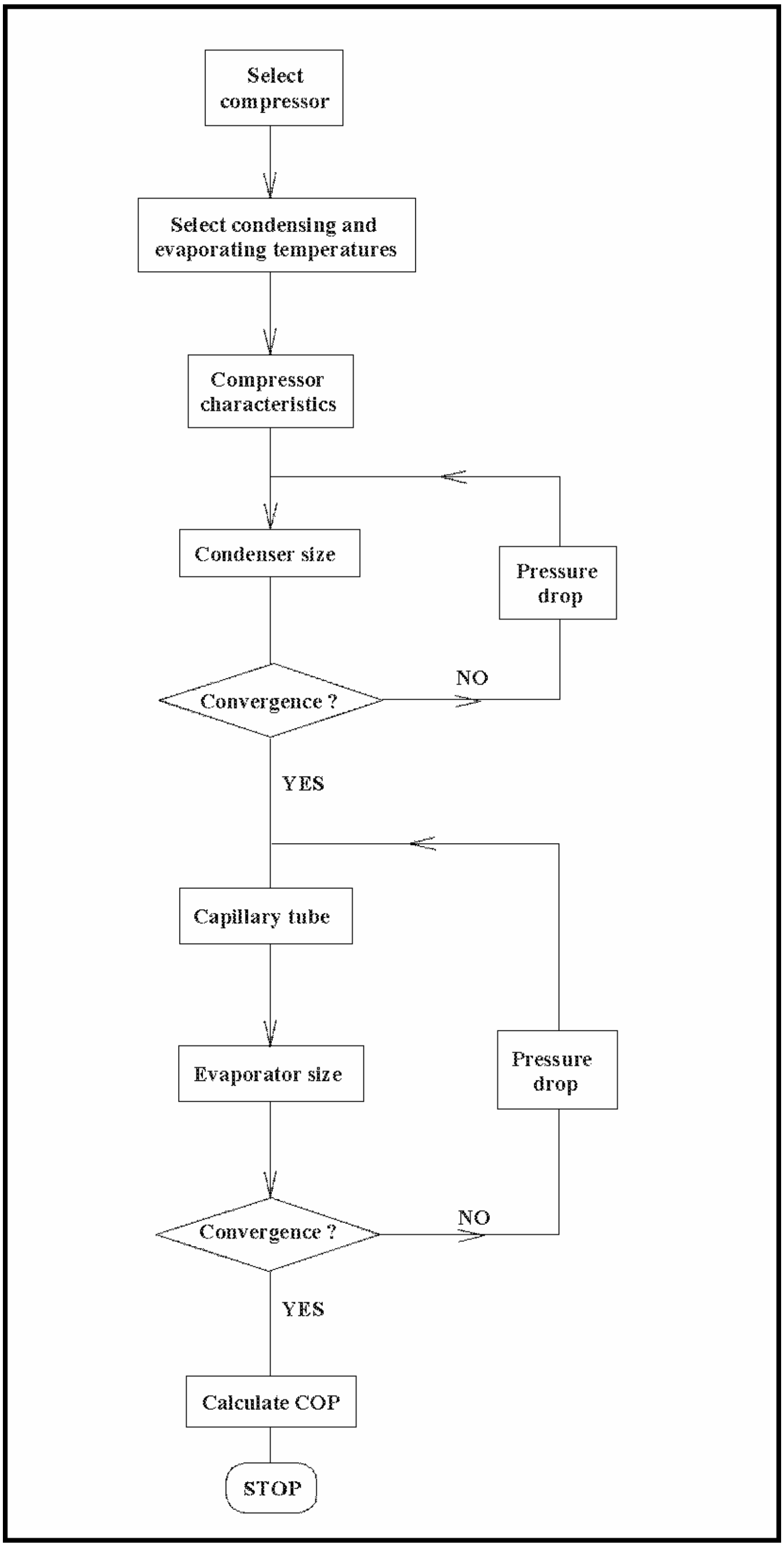

Figure 3. Flow chart outlining the design procedure.

Since the evaporating temperature is a selected value the pressure at point 1 on Figure 1 is known. The temperature at this point is equal to the evaporating temperature plus the recommended superheat value. With the pressure and temperature known all the other values can be determined from the thermodynamic superheat chart for R22.

From the selected evaporating and condensing temperatures the refrigerant mass flow (m_R), cooling capacity (Q_L) and compressor input work (W_c) can be read from the compressor curves. In the case of a hermetically sealed compressor it is fair to assume that all the input power is transferred to the refrigerant - either in the form of heat or work. Since the compressor discharge pressure is known from the condensing temperature the enthalpy at point 2 can be determined from

$$W_c = m_R \cdot \left(h_2 - h_1\right) \quad (1).$$

With the enthalpy and pressure known at point 2 all the other thermodynamic properties can be determined from the REFPROP database (Gallagher et al, 1993).

Condenser

The thermodynamic properties of points 3 and 4 in Figure 1 follow from the R22 properties chart at the selected condensing temperature. The temperature at point 5 is determined from the temperature at point 4 minus the subcooled value recommended by the compressor manufacturer. With the temperature and pressure at point 5 known all the other thermodynamic properties can be determined (Gallagher et al, 1993). Two different types of condensers (water-cooled and air-cooled) have been identified previously (Table 1) and since different procedures are used to determine their sizes they are discussed separately.

Water-cooled condenser

The water-cooled condenser is a tube-in-tube heat exchanger of the counterflow configuration with the warm refrigerant R22 flowing in the inner tube and the colder water flowing in the outer annulus. The condenser is split into three regions; superheat, two phase and subcool. To obtain the overall condenser length the length of each region must be calculated. The first step in the design process is to determine the heat transfer in each individual region, this is done with the aid of Equation 2.

$$Q_r = m_R \cdot \left(h_{ri} - h_{ro}\right) \quad (2).$$

The first region pertains to the superheat region, in this region the inlet enthalpy is the enthalpy at point 2 in Figure 1 and the outlet enthalpy refers to point 3. The second region is the two phase region with an inlet enthalpy at point 3 and the outlet enthalpy at point 4. The final region is the subcool region with the inlet enthalpy at point 4 and the outlet enthalpy at point 5. For all three regions the refrigerant mass flow rate is a constant value and is equal to the mass flow through the compressor. In each region it is possible to rewrite the previously calculated heat transfer in terms of the overall heat transfer coefficient, the total heat transfer area and the log mean temperature difference of the specific region. This equation is given as

$$Q_r = U_r \cdot A_r \cdot \Delta T_{mr} \quad (3).$$

The first term to be defined is the log mean temperature difference for a specific region.

$$\Delta T_{mr} = \frac{\left[GTTD_r - LTTD_r\right]}{\left[\ln\cdot\left(GTTD_r / LTTD_r\right)\right]} \quad (4).$$

The two values in Equation 4 are described as the greater terminal temperature difference of a region and the lesser terminal temperature difference of a region. At the inlet and outlet to each region there are two distinct temperatures, the first is that of the refrigerant and the second is the water temperature. The difference between the refrigerant and water temperatures at the inlet and outlet are either a maximum or a minimum value and this value is then defined as either the greater terminal temperature difference or lesser terminal temperature difference of the individual region. To determine these values the refrigerant and water temperatures must be known at the inlets and outlets of each region. The refrigerant temperatures at points 2, 3, 4 and 5 are known. The water inlet temperature at point 5 is 22.16^{o}C. This value is the summer average ground temperature at a depth of 1.8 m (Climate of South Africa, 1954). The summer value was used since the system will most probably be operated in the cooling mode in summer in which case the condenser will have to be cooled by water for the ground source system. Using this water inlet temperature the outlet temperature at point 4 is calculated from:

$$m_R \cdot \left(h_4 - h_5\right) = m_{water} \cdot C_{pwater} \cdot \left(T_4 - T_5\right) \quad (5).$$

The outlet temperature of this region is then the inlet temperature of the successive region. The same procedure is then used to calculate the water temperatures at points 3 and 2. The final parameter in Equation 3 is the overall heat transfer coefficient for each region. This term is composed of further parameters that incorporate the heat transfer coefficients of each fluid for each region as well as the heat exchanger dimensions. The parameter U_r is expanded as follows:

$$U_r = \frac{1}{\left[\left(\frac{1}{h_{R22}}\right) + \left\{\frac{d_{ii} \cdot \ln\cdot\left(d_i / d_{ii}\right)}{2 \cdot k_{Cu}}\right\} + \left(\frac{d_{ii}}{d_{oi}}\right) \cdot \left(\frac{1}{h_{water}}\right)\right]_r} \quad (6).$$

The middle term in Equation 6 for each of the three regions is the same for an assumed pipe diameter, as it is a function of the inner and outer pipe diameters and the thermal conductivity of the tubes, namely copper, used in the pipe-in-pipe heat exchanger construction. The first and the third term which relate to the heat transfer coefficients of the refrigerant and water for each region will now be explained in more detail.

The first region under investigation is the superheat region (state 2-3 on Figure 1). The first heat transfer coefficient (h_{R22}) is that of a single phase fluid. The equation describing this coefficient is the Petukhov equation (Holman, 1992) with all properties calculated at the film temperature. The accuracy of the Petukhov equation is at best 10 %, which is considerable better than the more conventional Dittus-Boelter equation (Holman, 1992).

The second heat transfer coefficient (h_{water}) is also that of a single phase fluid and is also determined using the Petukhov equation (Holman, 1992). Once the two heat transfer coefficients for the two fluids of the superheat region have been calculated the length of the water-cooled tube-in-tube condenser for this region can be calculated using Equation 3. The equation is rewritten:

$$L_r = Q_r \Big/ \left(U_r \cdot \Delta T_{mr} \cdot \pi \cdot d_{ii}\right) \qquad (7),$$

where L_r represents the length of a particular region, in this first case the superheat region. The same procedure was used to calculate the heat exchanger lengths in the other regions except for the refrigerant in the two phase region between points 3 and 4. In this region condensation occurs and therefore the heat transfer coefficient is described by Fujii (Shizuya et al, 1995) with all properties based on an average condensing temperature.

The next important step in the design process is to determine the pressure losses through each length of the water-cooled tube-in-tube condenser. In the superheat region (state 2-3) and the subcool region (state 4-5) both fluids are single phase fluids and their pressure losses are given by the Fanning equation (Saunders, 1988). All the fluid properties are based on the fluid film temperature. Using the length of the region and the area of the specific pipe for each fluid the pressure loss through the heat exchanger is determined for each individual fluid.

In the two phase region (state 3-4) the refrigerant is a two phase fluid whilst the water is a single phase fluid. It is therefore possible to calculate the pressure loss in the water in the outer annulus using the Fanning equation (Saunders, 1988). The pressure drop in the inner pipe, R22, however is given by the Martinelli equation (Jung and Radermacher, 1993) with all fluid properties based once again on an average condensing temperature. The calculation of the pressure losses in the water-cooled tube-in-tube condenser are done so as to assist in the selection of suitable pipe diameters in the iterative design process.

Air-cooled condenser

The air-cooled condenser is a plate fin-and-tube heat exchanger with plane fins and a cross-flow configuration. The refrigerant R22 is flowing inside the tube and the air is moving outside the tube. The air-cooled condenser is designed according to the same design methodology as the water-cooled condenser laid out in Figure 3. The condenser has various geometrical parameters, they include the face area, coil depth, the number of rows transverse and vertical, the pipe diameter, the fin spacing, the number of circuits and the tube type configuration. Each standard air-coil unit is split into a set number of circuits and the unit is then designed separately for each circuit.

To obtain the overall area of an air-coil unit the area of the coil for each circuit must be calculated. The design process is now explained for one such circuit. The condenser is split into three regions; superheat, two phase and subcool. The first step in the design process is to determine the heat transfer in each individual region, this is done using a variation of Equation 2. The difference is the mass flow rate, in this case the mass flow is divided by the number of circuits as the design is done for each separate circuit. The first region is the superheat region, in this region the inlet enthalpy is the enthalpy at point 2 in Figure 1 and the outlet enthalpy refers to point 3. The second region is the two phase region with an inlet enthalpy at point 3 and the outlet enthalpy at point 4. The final region is the subcool region with the inlet enthalpy at point 4 and the outlet enthalpy at point 5. For all three regions the refrigerant mass flow rate is a constant value equal to the mass flow in a circuit.

The calculated heat transfer of each region is rewritten according to Equation 3 where the log mean temperature difference is defined as in Equation 4. This value is calculated differently as the temperature differences must be calculated between the refrigerant (inside the tube) and the air (outside the tube). The refrigerant temperature at point 2, 3, 4 and 5 are known. The air inlet temperature is known, it is equal to the required indoor temperature. This value is constant for each region due to crosswise flow of the air. The air outlet temperature differs for each region as the heat transferred in each region is not the same. The air outlet temperature for each region is calculated from the following equation:*

$$\frac{m_R}{n_c} \cdot \left(h_{ri} - h_{ro}\right) = \frac{m_{air}}{n_c} \cdot C_{pair} \cdot \left(T_{ia} - T_{roa}\right) \qquad (8).$$

The overall heat transfer coefficient for each region is composed of various heat transfer coefficients and the heat exchanger dimensions. It is defined as follows:

$$U_r = 1 \Big/ \left[\left(1 \big/ \eta_o \cdot h_o \cdot A_o \right) + \left(\delta_w \big/ k_w \cdot A_w \right) + \left(1 \big/ h_i \cdot A_i \right) \right] \qquad (9).$$

The middle term in Equation 9 is the same for each region as it is a function of the tube area and its thermal conductivity. The first and third terms pertain to the air side and tube side heat transfer coefficients. The first heat transfer coefficient to be calculated is the air side heat transfer coefficient which is calculated from the Colburn j-factor defined by Wang et al. (1996). The fluid properties are evaluated at the average values of the inlet and outlet air temperatures. The finned surface effectiveness (η_o) in Equation 9 is taken as a constant value of 0.8 which is acceptable for the air-coil units used in this design (Jones, 1985).

The second heat transfer coefficient that must be calculated is the tube side heat transfer coefficient which refers to the refrigerant. The calculations are split again as with the water-cooled condenser into three regions, namely that of the single phase refrigerant (superheat and subcool regions) and the two phase refrigerant (two phase region). They are calculated according to the same method as the water-cooled condenser except for two differences. Firstly the mass flow considered is not the total refrigerant mass flow but the mass flow in a circuit which is equal to the total refrigerant mass flow divided by the number of circuits. Secondly the heat transfer coefficient in the two phase region is described in the equation of Ackers and Rosson (ASHRAE, 1993) since this equation was specifically developed for condensation in an air-cooled tube with fins. All the fluid properties are evaluated at an average condensing temperature. The single phase heat transfer coefficients are calculated from the Petukhov equation (Holman, 1992) with all R22 properties evaluated at an average superheat/subcool temperature.

It is now possible from Equation 3 to determine the area of a region and subsequently the area of each circuit which is equivalent to the sum of the region areas. To obtain the total area of the air-coil unit it is the sum of the areas of each circuit. Since each circuit area is the same the total area can be written as:

$$A_o = n_c \cdot A_{cir} \qquad (10.$$

The next important step in the design process is to calculate the pressure losses of the air and the refrigerant. The air side pressure drop is solved for from the equation of Kays and London (Wang et al., 1996) once the f-factor has been calculated from Wang et al. (1996).

* The parameter n_c in Equation 8 cancels out. It is shown in the equation to highlight the importance of this parameter in the formulation of the relevant equations.

All the fluid properties are evaluated at the average values of the inlet and outlet temperatures.

The second pressure losses to be calculated are for the refrigerant through the tubes for each tube circuit and each region. The same equations are used as in the water-cooled condenser with again one difference, namely that the refrigerant mass flow considered is that of a circuit which is equal to the total refrigerant mass flow divided by the number of circuits. In the superheat and subcool regions the pressure losses are calculated from the Fanning equation (Saunders, 1988), with the fluid properties based on the fluid film temperature. In the two phase region the pressure drop is calculated from the Martinelli equation (Jung and Radermacher, 1993) with the fluid properties evaluated at the average condensing temperature. Based on the new total area, as determined from Equation 10 it is possible to redo the calculations using different geometrical parameters until the area of the air-coil unit converges to give the best possible design.

Capillary Tubes

The selection of a capillary tube depends on the application and anticipated range of operating conditions. For all four situations, as laid out in Table 1, a set capillary tube inside diameter and set number of capillary tubes is chosen. This is to reduce the cost of the overall system due to a reduction in the amount of parts that need to be replaced when changing the operating condition of the system. It is apparent form the design procedure that the total refrigerant mass flow in the system is known. The number of selected capillary tubes is known and thus the mass flow through each capillary tube can be determined by dividing the total refrigerant mass flow by the number of capillary tubes.

It is now possible with the chosen inside diameter and the mass flow per tube to obtain the pressure drop per metre through the use of the pressure drop chart for capillary liquid. The next step is to determine the pressure drop through the capillary tube, this is the difference between the condenser inlet pressure and the evaporator outlet pressure of the capillary tube. Now by dividing the obtained pressure drop per metre by the pressure drop the length of the capillary tube is calculated. This is the length of a single capillary tube.

Evaporator

The thermodynamic properties of points 6 and 7 are determined from the known evaporating temperature and pressure (Gallagher et al, 1993). The vapour quality at point 6 is obtained from these thermodynamic properties and the condenser design. An assumption is made that the process of expansion through the capillary tubes occurs isenthalpic, meaning there is no addition or absorption of heat or work. Thus the enthalpy at points 5 and 6 on Figure 1 are equal. As with the condenser only two types of evaporators have to be considered in this study, namely a water-heated evaporator and an air-heated evaporator as already identified in Table 1.

Water-heated evaporator

The water-heated evaporator is a tube-in-tube heat exchanger of the counterflow configuration with the cold refrigerant R22 flowing in the inner tube and the warmer water flowing in the outer annulus. The evaporator is split into two regions; two phase and superheat. The overall length of the evaporator is obtained by determining the length of each individual region. The amount of heat transferred in each region is calculated from Equation 2 based on the parameters for the evaporators regons.

The calculated heat transfer in each region (Q_r) can be rewritten as in Equation 3 with the log mean temperature difference defined as in Equation 4. The calculation of the greater terminal temperature difference and lesser terminal temperature difference values for each region depends on the refrigerant and water temperatures at the inlet and outlet of each region. The refrigerant temperatures at points 6, 7 and 1 are known, the water temperatures are calculatd the same as the water-cooled condenser except that the inlet temperature is now 18.65^{o}C. This value is the winter average ground temperature at a depth of 1.8 m (Climate of South Africa, 1954). The remaining water temperatures are calculated as before.

It is now necessary to explain the individual components of the overall heat transfer coefficient as it is defined in Equation 6. As for the condenser design the middle term is constant for both regions for an assumed pipe diameter, as it is a function of the inner and outer pipe diameters and the thermal conductivity of the tubes, namely copper. The first and third term which relate to the heat transfer coefficients of the refrigerant and water for both regions will now be explained in more detail.

The first region under investigation is the two phase region (state 6-7 on Figure 1). The first heat transfer coefficient in this region (h_{R22}) is that of a two phase fluid. The process of evaporation occurs in this region and the heat transfer coefficient is described by the equation of Jung et al. (Jung and Radermacher, 1993) with all the fluid properties evaluated at an average evaporating temperature. The second heat transfer coefficient in this region (h_{water}) is that of a single phase fluid and it is determined using the Petukhov equation. The same conditions apply as were defined in the condenser design. Once the two heat transfer coefficients for the two fluids of the two phase region have been calculated the length of the water-heated tube-in-tube condenser for this region is calculated using Equation 7. In the next region, superheat region, both fluids are single phase and so the heat transfer coefficients are calculated using the Petukhov equation (Holman, 1992). Upon calculation of the two heat transfer coefficients the heat exchanger length is calculated using Equation 8.

The next important step in the design process is the calculation of the pressure losses through the water-heated tube-in-tube evaporator for each individual region. The single phase fluids pressure losses are calculated using the Fanning equation (Saunders, 1988) and the two phase fluids pressure drop is calculated using the Martinelli equation (Jung and Radermacher, 1993). All fluid properties are evaluated at the average superheat/evaporating temperatures. The calculation of the pressure losses in the water-heated tube-in-tube evaporator are done so as to assist in the selection of suitable pipe diameters in the iterative design process.

Air-heated evaporator
The air-heated evaporator is a plate fin-and-tube heat exchanger with plane fins and a cross-flow configuration. The refrigerant R22 is flowing inside the tube and the air is moving outside the tube. The air-cooled condenser is designed according to the same design methodology as the water-heated condenser laid out in Figure 3. The evaporator has various geometrical parameters, they include the face area, coil depth, the number of rows transverse and vertical, the pipe diameter, the fin spacing, the number of circuits and the type tube

configuration. Each standard air-coil unit is split into a set number of circuits and the unit is then designed separately for each circuit.

To obtain the overall area of an air-coil unit the area of the coil for each circuit must be calculated. The design process is now explained for one such circuit. The evaporator is split into two regions; two phase and superheat. The first region pertains to the two phase region from point 6 to 7 and the second region pertains to the superheat region from point 7 to 1 on Figure 1. The calculated heat transfer of each region is rewritten according to Equation 3 where the log mean temperature difference is defined as in Equation 4. This value is calculated differently as the temperature differences must be calculated between the refrigerant (inside the tube) and the air (outside the tube). The refrigerant temperatures at points 6, 7, and 1 are known. The air inlet temperature is known, it is equal to the required indoor temperature. This value is constant for each region due to crosswise flow of the air. The air outlet temperature differs for each region as the heat transferred in each region is not the same. The air outlet temperature for each region is calculated from the following equation:

$$\frac{m_R}{n_c} \cdot \left(h_{ri} - h_{ro}\right) = \frac{m_{dryair}}{n_c} \cdot \left(h_{dryao} - h_{dryai}\right) \quad (11).$$

The mass flow of the dry air over the fins can be determined from the fan curve of the selected fan once the pressure drop over the fins are known. From the fan curve the volume flow of the air is determined but this mass flow includes the water vapour. The mass flow of the dry air is the volume flow multiplied by the density of the air in kilograms dry air per cubic metre obtained from the psychometric chart at its specific dry and wet bulb temperatures. The enthalpy of the air at the inlet is read directly from a psychometric chart for air due to the fact that both the wet bulb and dry bulb temperatures of the air are known for a certain height above sea-level. The enthalpy at the outlet can then be calculated from Equation 11. The apparatus dew point temperature is taken as the average between T_6 and T_7 in the two phase region and in the superheat region it is taken as the average between T_7 and T_1. Through the construction of a straight line through the apparatus dew point temperature on the saturation curve of a psychometric chart and the inlet condition of the air, the outlet enthalpy and thus the value of the dry bulb outlet temperature is read directly from the psychometric chart. The outlet temperature of the two phase region is therefore not the same as the outlet temperature of the single phase region.

It is now necessary to explain the individual components of the overall heat transfer coefficient as it is defined in Equation 9. As for the condenser design the middle term is constant for both regions as it is a function of the tube area and its thermal conductivity. The first and third terms pertain to the air side and tube side heat transfer coefficients. The first heat transfer coefficient to be calculated is the air side heat transfer coefficient.

For the calculation of the outside heat transfer coefficient it is first vital to determine whether the air-heated evaporator (air-coil) operates under wet (dehumidification) or dry conditions. This fact can be determined by evaluating the following equation and then comparing it to various limits which then determine the conditions under which the coil operates (ASHRAE, 1996). During the design process it was determined that the two conditions (see the discussion later on reversibility) for which the air-coil units must be designed are the fully wetted and completely dry conditions. For each of these conditions a j- and f-factor can be defined from the literature (McQuiston, 1978). There are a separate set of equations for the dry and wet conditions respectively.

Depending on which condition is prominent the specific j-factor is now used in the Colburn j-factor equation (Wang et al., 1996) and thus the air side heat transfer coefficient is calculated for the air-coil unit. All the fluid properties are evaluated at the average values of the inlet and outlet temperatures. The finned surface effectiveness (η_o) in Equation 10 is defined as before for an air-coil unit as a constant value of 0.8 (Jones, 1985).

The second heat transfer coefficient that must be calculated is the tube side heat transfer coefficient which refers to the refrigerant. The calculations are split again as with the water-heated evaporator into two regions, namely that of the single phase refrigerant (superheat region) and the two phase refrigerant (two phase region). They are calculated according to the same method as the air-cooled condenser. The single phase and two phase heat transfer coefficients are defined as for the air-cooled condenser using Petukhov (Holman, 1992) and Ackers and Rosson (ASHRAE, 1993). The area of a region is calculated from Equation 3 and the total area of the air-coil unit is calculated from Equation 10.

The next important step in the design process is the calculation of the pressure losses on the air side and the tube side. The air side pressure drop is calculated from the equation of Kays and London (Wang et al., 1996) with the fluid properties evaluated at the average values of the inlet and outlet temperatures after the evaluation of the friction factor. The friction factor refers to either the fully wetted or completely dry f-factor depending on the conditions.

The tube side pressure losses are calculated for each tube circuit and each region; two phase and superheat. The same equations are used as in the water-cooled condenser with one difference, namely that the refrigerant mass flow considered is that of a circuit which is equal to the total refrigerant mass flow divided by the number of circuits. The single phase regions pressure loss is calculated from the Fanning equation (Saunders, 1988) with the fluid properties based on the fluid film temperature. The two phase regions pressure drop is calculated from the Martinelli equation (Jung and Radermacher, 1993) with the fluid properties evaluated at the average evaporating temperature. Based on the new total area, as defined by Equation 21 it is possible to redo the calculations using different geometrical parameters until the area of the air-coil unit converges to give the best possible design.

Coefficient of performance

The coefficients of performance for each system, air and ground source will be discussed separately. The first set of equations apply to the air source system and they are given for both operating conditions, namely heating and cooling. The coefficient of performance for the air-coil unit for heating is given as (Stoecker and Jones, 1982):

$$COP_{aH} = \frac{Q_H}{W_c + P_{FE} + P_{FC}} \tag{12},$$

where P_{FE} and P_{FC} refer to the evaporator and condenser fan power respectively. These can be calculated from the theory (Stoecker and Jones, 1982). The coefficient of performance for the air-coil unit for cooling is defined as follows (Stoecker and Jones, 1982):

$$COP_{aC} = \frac{Q_L}{W_c + P_{FE} + P_{FC}} \quad (13),$$

with the properties defined as for Equation 12.

The second set of equations apply to the ground source system and they are given for both operating conditions, namely heating and cooling. The coefficient of performance for the ground-coupled system for heating is given as (Stoecker and Jones, 1982):

$$COP_{gH} = \frac{Q_H}{W_c + P_{FC} + P_p} \quad (14),$$

where P_{FC} refers to the condenser fan power and P_p refers to the pump power. These can be calculated from the theory (Stoecker and Jones, 1982). The coefficient of performance for the ground-coupled system for cooling is defined as follows (Stoecker and Jones, 1982):

$$COP_{gC} = \frac{Q_L}{W_c + P_{FE} + P_p} \quad (15),$$

where P_{FE} refers to the evaporator fan power and P_p refers to the pump power and is defined as before. These four equations are now used to determine the coefficients of performance of each system and subsequently help in determining the overall effectiveness of the two systems.

DESIGN GEOMETRY

Based on the design methodology described previously two systems were designed, an air source system and a ground source system. The boundary values used were ambient conditions and water supply conditions which are given later when the results are discussed. It is not the aim of this study to optimise these systems. Therefore, current practices as found mostly in industry were used in the selection of most variables except the area of the air-coil and the area or more specifically the length of the tube-in-tube heat exchanger.

For the air source and ground source systems three heat exchangers were therefore designed with corresponding capillary tubes. The heat exchangers are an inside air-coil, an outside air-coil and an outside water tube-in-tube heat exchanger (water coil). The only difference between the air source and ground source system is on the outside where an air-coil is used for the air source system and a water coil is used for the ground source system. The final results is that three heat exchangers are needed for the two systems. Firstly there is an inside air-coil that has the same geometry for the air source and ground source system. This unit is used on the inside of the zone that is to be heated or cooled. Secondly there is an outside air-coil that is used as either a heat source or heat sink for the air source system. Lastly there is a tube-in-tube heat exchanger connected to a water loop which acts as either a heat source or heat sink for the ground source system. Since the two systems also operate between the same pressure levels the diameters and lengths of the capillary tubes are also the same. Only the most important geometrical variables of these heat exchangers and capillary tubes are now given: *Inside air-coil:* Width 900 mm, height 280 mm, depth 88 mm, 4 rows deep, 11 rows down, pipe outside diameter 9.5 mm, fin spacing - 472 fins/m, 4 circuits and a staggered configuration; *Outside air-coil:* Width 960 mm, height 610 mm, depth 65 mm, 3 rows deep, 23 rows down, pipe outside diameter 9.5 mm, fin spacing - 472 fins/m, 4 circuits and a staggered configuration; *Outside water coil:* Outside nominal pipe diameter of 25.4 mm, inside nominal pipe diameter of 15.88 mm and an overall calculated length of 43.875 m; *Capillary tubes:* Number of tubes - 3, tube outside diameter 3.3 mm, tube inside diameter 1.4 mm and a calculated length of 1.5 m. The calculated length is the length of each of the three capillary tubes as explained in the design process.

VERIFICATION

Two methods have been used to verify the calculated results obtained using the design procedure outlined in this paper. The first method is through the use of a commercially available simulation computer programme HPSIM (Greyvenstein, 1988). The second method is from the actual experimental testing. Each one of these verification procedures will be discussed separately before the results are given.

The programme HPSIM is an analysis programme that simulates the operation of a specific heat pump system for any operating mode. The programme covers a large range of available heat pumps and is very useful as a verification method. The important thing is that it is not a design program, rather it uses predetermined geometrical heat exchanger parameters as its input. In this case the dimensions used were those given in the previous section. For more detailed information on this programme the reader is referred to Greyvenstein (1988).

For the experimental set-up the two designed systems were built and installed for testing. The systems were installed in a house, thus the indoor unit and outdoor unit were exposed to different ambient conditions (which are given later). The basic test procedure includes the measuring of the temperatures (dry bulb and wet bulb), the mass flows (air and water) and the heating and cooling capacities.

RESULTS

The results will be split into two sections, the first are those of the air source system (Table 2) and the second are those of the ground source system (Table 3). For each system the results are also given in the heating mode and the cooling mode. For each set of results three different values are given they are the experimental results, the calculated results and the simulated (HPSIM) results. The experimental results which are taken directly from the experimental tests are accepted to be the correct value. The error of the calculated and simulated results compared to the experimental results are then also tabulated. This error is either positive or negative. If the error is positive then the calculated or simulated value overpredicts the experimental result but if the error is negative then the values underpredict the experimental results.

Air Source

From the results in Table 2 it is evident that the accuracy of the simulated results are better than those of the calculated results when compared with the experimental results. The accuracy of both values are still within reason. It is further noted that in the heating mode the heating coefficient of performance (COP) is underpredicted by only 3 % for the simulated results and by 12 % for the calculated results whilst in the cooling mode the cooling COP is underpredicted by 6 % for the simulated results and by 17 % for the calculated results. This affirms the better accuracy achieved with the simulation program when compared to the actual operating conditions.

HEATING MODE					
Property	**Experimental Value**	**Calculated Value**	**Calculated Error**	**Simulation Value**	**Simulation Error**
Evaporating temperature	6°C	6°C	0°C	5.84°C	- 2.7°C
Condensing temperature	58°C	58°C	0°C	58.23°C	+ 0.4°C
Cooling capacity	8190 W	6671 W	- 18.5 %	8286 W	+ 1.12 %
Compressor power	3400 W	3950 W	+ 13.9 %	3821 W	+ 11 %
Evaporator: Inlet air temperature	19°C	19°C	0°C	18.58°C	- 2.2°C
Inlet air temperature (WB	12°C	12°C	0°C	12°C	0°C
Outlet air temperature	12°C	8.18°C	- 31.8°C	11.28°C	- 6°C
Air mass flow	1.05 kg/s	1.0 kg/s	- 4.8 %	1.11 kg/s	+ 5.4 %
Condenser: Inlet air temperature	22°C	22°C	0°C	22°C	0°C
Outlet air temperature	50°C	50.76°C	+ 1.5°C	50.45°C	+ 0.9°C
Air mass flow	0.4 kg/s	0.38 kg/s	- 5 %	0.4 kg/s	0 %
Heating COP	3.41	3.0	- 12 %	3.3	- 3.3 %
Cooling COP	2.41	2.0	- 17 %	2.35	- 2.3 %
COOLING MODE					
Property	**Experimental Value**	**Calculated Value**	**Calculated Error**	**Simulation Value**	**Simulation Error**
Evaporating temperature	0°C	0°C	0°C	- 0.35°C	- 0.01°C
Condensing temperature	36°C	36°C	0°C	35.51°C	- 1.4°C
Cooling capacity	8253 W	7452 W	- 9.7 %	8780 W	+ 1.5 %
Compressor power	2400 W	2648 W	+ 9.4 %	2381 W	- 0.8 %
Evaporator: Inlet air temperature	22°C	22°C	0°C	22.15°C	+ 0.7°C
Inlet air temperature (WB	14°C	14°C	0°C	14°C	0°C
Outlet air temperature	6°C	5.42°C	- 9.7°C	6.01°C	+ 0.2°C
Air mass flow	0.38 kg/s	0.38 kg/s	0 %	0.4 kg/s	+ 5 %
Condenser: Inlet air temperature	19°C	19°C	0°C	19°C	0°C
Outlet air temperature	29°C	33.48°C	+ 13.4°C	28.85°C	- 1.7°C
Air mass flow	1.12 kg/s	1.12 kg/s	0 %	1.12 kg/s	0 %
Heating COP	4.61	4.0	- 13.2 %	4.32	- 6.1 %
Cooling COP	3.61	3.0	- 16.9 %	3.4	- 5.6 %

Table 2: Experimental, calculated and simulated results for the air source system.

Ground Source

From the results shown in Table 3 it is evident that the accuracy of the calculated results are better than those of the simulated results when compared with the experimental results. The accuracy of the calculated values is very good but the accuracy of the simulated results is very poor. values are still within reason. It is further noted that in the heating mode the heating coefficient of performance (COP) is underpredicted by only 6 % for the calculated results and by 10 % for the simulated results whilst in the cooling mode the cooling COP is underpredicted by 4 % for the calculated results and by 18 % for the simulated results. This affirms the better accuracy achieved with the design procedure described in this document when the calculated results are compared to the results of the experimental tests. The larger error in the simulated results is attributed to the different equations used in the proposed design procedure and the simulation program's design procedure followed in the program.

HEATING MODE					
Property	**Experimental Value**	**Calculated**		**Simulation**	
		Value	**Error**	**Value**	**Error**
Evaporating temperature	11°C	9.04°C	- 17.8°C	11.03°C	+ 0.3°C
Condensing temperature	47°C	46.73°C	- 0.6°C	46.86°C	- 0.3°C
Cooling capacity	12600 W	12488 W	- 0.9 %	11388 W	- 9.6 %
Compressor power	3100 W	3245 W	+ 4.5 %	3129 W	+ 0.9 %
Evaporator: Water inlet temperature	19°C	18.65°C	- 1.8°C	18.7°C	- 1.6°C
Water outlet temperature	14°C	13.67°C	- 2.4°C	14.16°C	+ 1.1°C
Condenser: Inlet air temperature	22°C	22°C	0°C	22°C	0°C
Outlet air temperature	40°C	42.03°C	+ 4.8°C	34.57°C	- 13.6°C
	5.07	4.77	- 5.9 %	4.57	- 9.9 %
Cooling COP	4.07	3.79	- 6.9 %	3.58	- 12 %
COOLING MODE					
Property	**Experimental Value**	**Calculated**		**Simulation**	
		Value	**Error**	**Value**	**Error**
Evaporating temperature	7°C	6.48°C	- 7.4°C	3.73°C	- 46.7°C
Condensing temperature	29°C	28.17°C	- 9.3°C	28.3°C	- 2.4°C
Cooling capacity	15278 W	14132 W	- 7.5 %	11379 W	- 25.5 %
Compressor power	2600 W	2444 W	- 6 %	2302 W	- 11.5 %
Evaporator: Inlet air temperature	22°C	22°C	0°C	22°C	0°C
Inlet air temperature (WB)	14°C	14°C	0°C	14°C	0°C
Outlet air temperature	10°C	9.9°C	- 1°C	13°C	+ 23.1°C
Outlet air temperature (WB	9°C	9.18°C	+ 1.9°C	9.2°C	+ 2.2°C
Condenser: Water inlet temperature	22°C	22.16°C	+ 0.7°C	22.16°C	+ 0.7°C
Water outlet temperature	29°C	28.77°C	- 0.8°C	27.63°C	- 4.7°C
Heating COP	6.88	6.64	- 3.5 %	5.82	- 15.4 %
Cooling COP	5.88	5.66	- 3.7 %	4.84	- 17.7 %

Table 3: Experimental, calculated and simulated results of the ground source system.

Air Source and Ground Source Systems

The final set of results are the comparison between the two systems. It is vital to compare the systems for both operating conditions to determine the effectiveness of the municipality water reticulation ground-coupled reversible heat pump over the air-to-air system. This is done through the comparison of the most vital variables of the experimental results. The systems experimental results are compared as these are the values are considered the most accurate. Table 4 is a summary of these values as well as the absolute percentage differences between these two values. If this value is positive then the air source system has a higher value than the ground source system and if the value is negative then the ground source system is higher.

For the heating mode it is evident that with less power input the heating capacity and heating coefficient of performance (COP) of the ground source system is better than the air source system. For the cooling mode it is evident a slight increase in input power the cooling capacity and cooling COP of the ground source system is better than the air source system. It is thus evident that for both operating conditions the reversible ground source system has a higher rate of performance (within 40 %) than the more conventional air-to-air system.

HEATING MODE			
Property	**Air Source**	**Gound Source**	**Absolute Difference**
Cooling capacity	8190 W	12600 W	- 35 %
Compressor power	3400 W	3300 W	+ 2.9 %
Heating capacity	11590 W	15900 W	- 27.1 %
Outlet air temperature	50°C	42°C	+ 16 %
Heating COP	3.41	4.82	- 29.2 %
COOLING MODE			
Property	**Air Source**	**Gound Source**	**Absolute Difference**
Cooling capacity	8653 W	15278 W	-43.4 %
Compressor power	2400 W	2600 W	- 7.7 %
Heating capacity	11053 W	17878 W	- 38.2 %
Outlet air temperature	6°C	10°C	- 40 %
Cooling COP	3.61	5.88	- 38.7 %

Table 4: Comparison of air source and ground source variables.

CONCLUSIONS AND RECOMMENDATIONS

In conclusion it has been determined through experimental tests that the proposed system, ground-coupled, greatly improves in its effectiveness over the more conventional air-to-air system. The proposed municipality water reticulation ground-coupled reversible heat pump system has been designed, built and tested. The design procedure (laid out in Figure 3) has been verified by two methods and they have proven that the design procedure is a feasible method that can be applied in further work.

The improvement in performance between the proposed ground-coupled system and the more conventional air-to-air system is significant. In the heating mode the system performance has improved by more than 29 % and in the cooling mode it improves by more than 38 %. It can be deduced that the ground-coupled system performance rates one third better than the air-to-air system and the system hasn't even been optimised yet. This improvement also proves that the choice of three standard heat exchangers for both systems, that is operating with the largest areas/length is a good design choice. Thus, the reversibility of the ground-coupled system is possible at an improvement in the system's performance.

There is still room for improvement by adjusting the structure of the system and optimising it so that the best possible performance can be achieved at the lowest possible cost. This can be accomplished by further research and testing of the existing system that will highlight the problem areas which can then be improved on and maybe modified if need be.

NOMENCLATURE

A heat exchanger as denoted in Figure 2
A_{cir} area of a circuit [m^2]
A_i tube side area [m^2]
A_o total surface area [m^2]
A_r total heat transfer area of a region [m^2]
A_w wall of tube area [m^2]
B heat exchanger as denoted in Figure 2
COP coefficient of performance
C_{pair} specific heat of air based on the air inlet temperature [J/kg K]
C_{pwater} specific heat of water based on the water inlet temperature [J/kg K]
d_i outer diameter of inner pipe [m]
d_{ii} inside diameter of inner pipe [m]
d_{oi} inner diameter of outer pipe [m]
$GTTD$ greater terminal temperature difference [K]
h enthalpy [J/kg]
h_i tube side heat transfer convection coefficient [W/m^2 K]
h_o air side heat transfer convection coefficient [W/m^2 K]
h_{R22} refrigerant heat transfer convection coefficient [W/m^2 K]
h_{water} water heat transfer convection coefficient [W/m^2 K]
k thermal conductivity [W/m K]
L_r length of heat exchanger in a region [m]
$LTTD$ lesser terminal temperature difference [K]
m mass flow rate [kg/s]
n_c number of circuits
P power [W]
Q_H heat released at condenser [W]
Q_L heat absorbed at evaporator [W]
Q_r heat transferred in region [W]
s specific entropy [J/kg K]
T temperature [K]
ΔT_{mr} log mean temperature difference of a region [K]
U_r overall transfer coefficient of a region [m^2 K/W]
W_c compressor input power [W]
δ_w tube wall thickness [m]

Subscripts

$1-7$ different states on Figure 1
aC cooling, air source system
aH heating, air source system
air air
Cu copper
$dryair$ dry air
$dryai$ dry air at inlet
$dyao$ dry air at outlet
FC condenser fan
FE evaporator fan
gC cooling, ground-coupled system
gH heating, ground-coupled system
ia inlet air
p pump
r region
ri inlet of region
roa air outlet of region
ro outlet of region
$R, R22$ refrigerant
w wall of the tube
$water$ water

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