INTEGRATION OF A GEOTHERMAL HEAT PUMP WITH A LOW-GRADE HEAT STORAGE SYSTEM

HYBRID ENERGY SYSTEMS STUDY (HESS)

PHASE II

FINAL REPORT

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January 15, 2009

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ACKNOWLEDGEMENTS AND CERTIFICATION
The research described herein has been performed in partnership with Rochester Public Utilities with funding provided by them and other sponsors. This financial support is gratefully acknowledged, however, the Principal Investigators assume complete responsibility for the contents herein.
EXECUTIVE SUMMARY

The Hybrid Energy Systems Study (HESS) Phase II was a research study focusing on the systems level integration of alternative energy technologies. In this initial work, the primary emphasis was on the combination of a heat pump and low grade thermal storage system. Low grade heat is available from sources such as fuel cells, solar thermal or hybrid collectors, biomass conversion plants, etc. The study encompassed comparisons between typical geothermal wells and stored thermal energy as heat sources, heat pump operation in heating and cooling environments, and the control systems necessary for the functioning of a hybrid plant. The goal of the study was to understand what energy efficiency gains are possible using stored low grade heat; what the requirements would be to implement a hybrid design as a fully functioning residential space conditioning system; and the cost savings possible by reducing the number of geothermal wells that are required without thermal storage. In performing the research, the following systems were designed and built:

- a stratified thermal storage system,
- a data acquisition system with real-time graphic data display capabilities, and,
- a manual control system for interconnecting the system components in various ways.

HESS was a partnership between the University of Minnesota Rochester and Rochester Public Utilities (City of Rochester, MN). Project funding and support was also provided by the American Public Power Association (APPA), the Initiative for Renewable Energy and the Environment (IRRE) from the U of MN, Southeastern Minnesota Public Power Association (SMMPA), the University of MN Digital Technology Center (DTC), and the University of MN Rochester (UMR). In addition, construction and funds for the HESS II Laboratory with a highly energy efficient envelope at the Quarry Hill Nature Center in Rochester, MN were provided by various Rochester contractors, the City of Rochester and the Friends of Quarry Hill. The TRANE Corporation provided the heat pump. The total project cost, including the value of the laboratory, was $375,000.

The results of the HESS project confirm the viability of using stored low grade heat as a heat pump thermal source in heating mode as well as the improved heat pump performance possible in such conditions. These results have encouraged the consideration of several future projects. First, would be the development of a heat pump technology/system optimized to make full use of efficiency gains demonstrated by using stored thermal energy as the primary source for the heat pump. Second, would be the demonstration of a HESS based technology heat pump/solar heat energy system to provide heating and cooling to a small group of homes (3 or 4) all connected to a small central energy plant—effectively a small microgrid combined heating and cooling (CHC) energy supply system. This would include sensor and related developments along with control algorithms to optimize and assure reliable system operation. Finally, the concept of a fully integrated microgrid energy system (combined heating, cooling and electrical power (CHCP)) would provide an optimized, comprehensive alternative energy supply to new housing developments (several hundred homes) with the objective of having more than 50% of the energy supplied from internal alternative energy sources.
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<table>
<thead>
<tr>
<th>Acronym</th>
<th>Description</th>
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<tbody>
<tr>
<td>COP</td>
<td>Coefficient of Performance for Heat Pump heating</td>
</tr>
<tr>
<td>CPVC</td>
<td>Chlorinated polyvinyl chloride</td>
</tr>
<tr>
<td>DAS</td>
<td>Data acquisition system</td>
</tr>
<tr>
<td>EER</td>
<td>Energy Efficiency Ratio for Heat Pump cooling</td>
</tr>
<tr>
<td>FC</td>
<td>Fuel cell</td>
</tr>
<tr>
<td>HDP</td>
<td>High density polyethylene</td>
</tr>
<tr>
<td>HESS</td>
<td>Hybrid Energy System Study</td>
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<tr>
<td>HP</td>
<td>Heat pump</td>
</tr>
<tr>
<td>HRV</td>
<td>Heat recovery ventilation unit</td>
</tr>
<tr>
<td>Lab VIEW</td>
<td>Laboratory Virtual Instrumentation Engineering Workbench</td>
</tr>
<tr>
<td>NI</td>
<td>National Instruments, Inc.</td>
</tr>
<tr>
<td>PEM</td>
<td>Proton exchange membrane</td>
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<tr>
<td>PVC</td>
<td>Polyvinyl chloride</td>
</tr>
<tr>
<td>STSS</td>
<td>Stratified thermal storage system</td>
</tr>
<tr>
<td>WH</td>
<td>Water heater</td>
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<tr>
<td>WHGD</td>
<td>Low-grade heat generating device Measurements</td>
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<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>A</td>
<td>Current in Amperes</td>
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<tr>
<td>°C</td>
<td>Temperature in degrees Celsius</td>
</tr>
<tr>
<td>°F</td>
<td>Temperature in degrees Fahrenheit</td>
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<tr>
<td>fpm</td>
<td>Air flow in cubic feet per minute</td>
</tr>
<tr>
<td>gpm</td>
<td>Liquid flow in gallons per minute</td>
</tr>
<tr>
<td>psi</td>
<td>Pressure in pounds per square inch</td>
</tr>
<tr>
<td>V</td>
<td>Voltage in volts</td>
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<tr>
<td>W, kW</td>
<td>Watts, kilo-Watts</td>
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INTRODUCTION

1.1 **Historical Context**

Clean, efficient, reliable and readily available energy is not only critical to the future of the United States but to the World given our global economy. The national concerns of not having energy independence and the environmental concerns of global warming and clean air are propelling the US and other countries towards alternative energy sources.

One of several alternative energy sources that is currently receiving significant attention are fuel cells for producing electricity. The basic concept of the fuel cell has been known for decades and it has been used in very special applications such as manned space travel. Although there have been many US patents issued relative to Proton Exchange Membrane (PEM) and Silicon Oxide fuel cell technology, much work remains on the fundamental technology of the fuel cell, how to efficiently produce the hydrogen to fuel the cell, and how to efficiently and safely introduce the technology as a primary source of electricity or power for the average consumer or business. Alternative energy devices or sources such as fuel cells, solar power, etc have significant available thermal energy that can be combined with other alternative energy options to form highly efficient, non fossil fuel energy solutions.

The presence of commercial and residential fuel cell systems for the generation of electricity and heat offer the electric power industry a great opportunity to increase their competitive position, while contributing to reduced environmental emissions. The vast variety of alternative energy generation methods, like geothermal, fuel cells, solar cells and wind turbines, yield opportunities for development of new hybrid energy systems that will be unique for the geographical environment of a specific region. For example, the most cost-efficient residential energy system for Minnesota could be different from a system in Colorado. Various techniques available in digital control and digital signal processing, using inexpensive microprocessors, offer the ability to dynamically control a hybrid energy system for optimal cost-performance and minimization of the need for fossil based energy. These types of systems allow the power industry to move in the direction to become full service energy solution providers.

The use or proposed use of fuel cell technology is starting to have an impact in a number of products and applications. These include of course the automobile, use as camping generators, to power rural homes and farms, as backup power for critical system applications, in mobile devices that require extended battery life, and to become a core piece of future microgrid energy systems. To prepare for the broad applications and use of fuel cells, a partnership between Rochester Public Utilities and the University of Minnesota Rochester was formed. This partnership has as its cornerstone the Hybrid Energy Systems Study or HESS project.

Partnership discussions explored various opportunities for joint work between the two organizations. These ultimately focused on a project researching the feasibility of a systems study focusing on combining fuel cells and geothermal heat pump (GX) technologies. It has become more and more evident in recent years, that any alternative energy technology, by itself, cannot meet the needs of replacing fossil fuel based technologies. As a result, the HESS project was defined as a study of how the integration of various alternative energy options can improve the availability and efficiencies over a single alternative energy option. In addition, to make the system more viable, options for the storage and retrieval of thermal energy were studied as part of the project. Fuel cells and geothermal technologies were chosen at the time
because of the maturity of the geothermal technology and the strong interest in fuel cell technology.

To fully develop the basic fuel cell technology there are still several segments or ‘sub-technologies’ that require significant research focus. While one can divide the work in various ways, three main components of the sub-technologies include: the production of hydrogen gas from clean sources for use in the fuel cell; the development of the fuel cell system itself which includes materials development, system controls and monitoring and integration into an operating unit; and finally how best to integrate basic alternative energy technologies for ultimate consumer applications.

HESS was planned to have three distinct Phases. Phase I, the basic study of fuel cells to become familiar with the technology, started in January 1, 2003 and reported in Ottesen 2004[1]. Phase II, the subject of this report, has focused on a systems level prototype with a high degree of instrumentation operated by digital control software resulting in the integration of the fuel cell with the geothermal heat pump based system. Phase III could involve two parts: a) product prototype demonstration of concepts and capabilities developed in HESS Phase II, that involves modified Heat Pump designs and development of a small central energy plant to serve 3-4 homes, and b) a large system application of integrated alternative energy solutions with some current fossil fuel sources, to maximize the use of green energy options that meet or exceed recent government directives via a microgrid application.

1.2 Project Support

Two types of funding and support have been provided to the HESS project. The first is direct support for the project research and development effort. The second is support to construct the HESS Phase II Laboratory at Quarry Hill Nature Center in Rochester, Minnesota.

Relative to the first, the U of MN Rochester and Rochester Public Utilities (RPU) have formed the primary partnership for HESS. In addition to RPU, significant funding has been provided by the Southern Minnesota Municipal Power Association (SMMPA), the American Public Power Association (APPA) with a grant to RPU, the Institute for Research on Energy and the Environment (IREE)-U of MN, the Digital Technology Center (DTC)-U of MN, and the University of Minnesota Rochester (UMR).

The funding and in-kind time and materials to construct and outfit the HESS Laboratory as an addition to the Quarry Hill Nature Center in Rochester, MN was provided by the following companies and institutions:

AJ Benike  
City of Rochester  
Friends of Quarry Hill Nature Center  
Haley Comfort Systems  
Rochester Public Utilities  
Sebesta Bloomberg and Associates-Engineers  
Superior Mechanical Systems, Inc.  
Superior Plumbing & Heating  
Trane Corporation  
University of Minnesota Rochester  
Yaggy Colby—Architects
We would also like to acknowledge the Quarry Hill Nature Center Staff for all of their support, and Jim Walters, Director of Customer Relations at Rochester Public Utilities, serving as the RPU partnership contact. Finally we would like to thank Ann Bottorff, University of Minnesota Rochester for her administrative and operations support throughout this project.

The Quarry Hill Laboratory facilities and project sign are shown in Figures 1, 2, and 3 below.
CHAPTER 2: GOALS AND OBJECTIVES

The HESS Phase II objectives resulted from the three phase HESS project goals defined in the partnership agreement with Rochester Public Utilities, and the research completed as part of HESS Phase I.

The specific goals of HESS Phase II included:

- Verification of the Hybrid Energy Systems Study patent\(^2\) submitted on January 31, 2005 that resulted from the HESS Phase I project completed in 2004 and validation of the physics outlined in the patent.

- Determination and demonstration of the impact of low grade heat in increasing the thermal performance of a vapor compression cycle heat pump in heating mode via experimental measurement of the Coefficient of Performance (COP).

- Investigation and implementation of an optimal means for storing low grade heat for prolonged periods including the hypothesis that an adequately sized, stratified thermal storage system can substitute for geothermal wells when the heat pump is in heating mode.

- Investigation of the operating dynamics of a hybrid energy system with a view to developing optimal control strategies applicable to renewable energy based systems.

- Investigation of the performance limitations of a conventional vapor compression cycle heat pump when connected to a thermal source of higher temperature than a geothermal well.

- Demonstration of the importance of a high thermal integrity building envelope to make the use of alternative energy practical in building space conditioning systems.

- Demonstration of the importance of Heat Recovery Ventilation (HRV) systems to minimize building envelope heat losses.
CHAPTER 3: HESS II PLANT

3.1 DESCRIPTION

The HESS II plant is situated in a recently built 24’x24’ annex to the Quarry Hill Nature Center located in the South-Eastern part of Rochester, Minnesota. The Nature Center is built on Quarry Hill Park, a 290-acre city park with hiking trails, fossil bed, fishing pond, caves, and a picnic area. Quarry Hill offers kindergarten through 8th grade curriculum based field trips during the school year for more than 35,000 annual student visits.

The original HESS II idea was to improve the efficiency of a ground-source heat pump using low-grade heat from a fuel cell that also was powering the heat pump. A US Patent Application was filed on January 31, 2005 and issued as a US Patent #7,334,406 on February 26, 2008[2]. The patent abstract describes: A hybrid energy system heats or cools a plant with a geothermal unit powered at least partly by a fuel cell, which may also power other devices. The thermal fluid for the geothermal unit also cools the fuel cell via a heat exchanger. A digital controller bypasses a variable portion of the thermal fluid around the heat exchanger to regulate the fuel-cell temperature.
The HESS II idea was funded and the original concepts were implemented in the HESS II Plant which over time had new and important additions added to the original idea. One of the most important additions was the 400-gallon Stratified Thermal Storage System (STSS), see Section 3.1.2-f. The next subsections describe the current HESS II plant, including the building, electrical-mechanical subsystems, instrumentation, and hardware and software components of the data acquisition system.

3.1.1 HESS II Building

a. Design
The Quarry Hill laboratory was designed and built to exceed the code requirements of the Minnesota Commercial Energy Code. In particular, there was a continuous sealed air barrier and vapor retarder over the entire interior surface area and the thermal insulation system was designed to minimize thermal breaks. The overall foundation design detail is shown in Figure 3.1.1-1.

![Foundation design](image)

Note the continuous insulation extending from the above the foundation stem wall sill plate and down the interior masonry block wall surface to the frost footing top surface. The slab was cast against the interior insulation surface. The sub-slab insulation butts against the vertical insulation and completely surrounds the perimeter supply duct. This arrangement eliminates the thermal breaks often found at the sill plate and the slab/wall interface. The exterior stem wall insulation increases the net R-value of the foundation wall system particularly in the vicinity of the sub-slab air supply duct. Similar attention to detail was a characteristic of the design of the rest of the envelope.
b. Duct System
The design of the duct system is shown in Figure 3.1.1-2. The return air system comprised a length of 18” diameter duct suspended beneath the ceiling on an east-west axis with the intake about 3 ft from the west wall. This duct was connected directly to the heat pump intake plenum.

![Figure 3.1.1-2. Duct system](image)

The supply air system was provided by a sub-slab perimeter continuous loop with floor registers along the north, east and south walls.

c. Ventilation
Laboratory ventilation was provided by a Fantech Model HRV-450 XI heat recovery ventilator (HRV) suspended beneath the ceiling with the exterior intake and exhaust ports positioned on either side of the window on the east wall. The interior intake was at the end of a duct extension while the interior exhaust was via a louvered, adjustable damper attached directly to the unit. The separation between the interior intake and exhaust ports was sufficient to eliminate short circuiting of the ventilation air. The HRV-450 is a dual speed unit with rated flow rates of 270 and 450 cfm at 0.2 in. water column static pressure. The heat recovery efficiency is rated by the manufacturer to be 60% at 270 cfm and 52% at 450 cfm. The purpose of an HRV is to provide adequate external fresh air while minimizing the conditioned space sensible enthalpy loss to the exterior (and resulting envelope thermal inefficiency) as shown by the manufacturer’s schematic in Figure 3.1.1-4. The exhaust and supply air streams exchange heat as they pass through the inclined counter-flow heat exchanger. Condensate collected on the exchanger plates drips down to a pan beneath the heat exchanger and is removed by tubing connected to an appropriate drain (on the floor in the HESS laboratory).
Figure 3.1.1-3. Fantech Model HRV-XI 450 light commercial heat recovery ventilator

Figure 3.1.1-4. Manufacturer’s HRV air flow schematic (Fantech article no. 600642 dated 5/17/06)
3.1.2 Major HESS II Subsystems

The major electrical-mechanical hardware components that make up the HESS II plant are discussed in this section:

a. Geothermal Heat Pump

The heart of any standard heat pump is the vapor-compression refrigeration cycle.

Figure 3.1.2-1. A closed-loop diagram of a heat pump's vapor-compression refrigeration cycle: (1) condenser heat exchanger, (2) expansion valve, (3) evaporator heat exchanger and (4) compressor. The arrows show the direction of circulation of the fluid/vapor state refrigerant.

Referring to Figure 3.1.2-1, the refrigerant in its gaseous state is pressurized and circulated through the system by a compressor (4). On the discharge side of the compressor, the now hot and highly pressurized gas is cooled in a heat exchanger (1) called a condenser until it condenses into a high pressure, moderate temperature liquid. The condensed refrigerant then passes through a pressure-lowering device like an expansion valve (2), a capillary tube, or possibly a work-extracting device such as a turbine. This device then passes the low pressure, barely liquid (saturated vapor) refrigerant to another heat exchanger (3), the evaporator where the refrigerant evaporates into a gas via heat absorption. The refrigerant then returns to the compressor and the cycle is repeated.

The reversible geothermal heat pump was a residential, 1.5-ton, Trane GEV-018 “Vertical Water-Source Comfort System” that was donated to the HESS II project by the Trane Corporation. The GEV-018 operates on single-phase, 60 Hz, 230-volt power and includes a fixed speed Grundfos, 1/6 hp circulation pump and a digital programmable thermostat as standard equipment. In the HESS II plant, the Grundfos pump was shown to be capable of providing a geothermal loop flow of 2.4 gpm only. However, the GEV-018 has a rated Coefficient of Performance (COP) of 3.34 and an Energy Efficiency Ratio (EER) of 14.9 at a

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1 A useful discussion of the geothermal heat pumps can be found in: [http://en.wikipedia.org/wiki/Heat_pump#Geothermal_heat_pumps](http://en.wikipedia.org/wiki/Heat_pump#Geothermal_heat_pumps)

loop flow rate of 4.2 gpm. This discrepancy was accommodated by the plant design of adding a variable speed pump, named P2, in parallel with the Grundfos pump (P1) as shown in Figure 3.2-1 and discussed in Section 3.2. Alone the ½ HP variable speed pump was able increase the loop flow rate to 5.1 gpm.

b. Geothermal Wells
Two 200-foot, 4.5-inch diameter, vertical wells were drilled about 25 feet north of the Quarry Hill Nature Center. The wells were spaced about 15 feet apart. The first 25 feet below the surface consisted of clay, sand and gravel, in that order. Below the 25-foot level there were various layers of gray and brown limestone and some sandstone. During the drilling it was also discovered that there was a 3-foot tall subterranean cavern connecting the two wells. One 200-foot, U-tube of 1-inch outer diameter (OD), black High Density Polyethylene (HDP) tube was inserted into each well then grouted with Portland cement. Thermocouples were installed at each of the wellheads to measure the temperatures of the inlet and outlet liquid flows. To make a good thermal conduction for the four wellhead thermocouples, a 8”-piece of ¾” copper tubing was spliced into each of the HDP tubes, the thermocouples were installed to make contact with the copper tubes and the splices were fitted with a moisture proof seal. The U-tubes were fused to similar HDP tubes that, together with a shielded thermocouple cable, were routed via a 85-foot long, 6-foot deep trench to a manifold located inside the laboratory. The tube pair for each well was placed in a plastic sleeve to keep them dry. The thermal coupling between the tube pairs was minimized by separating them on the base of the trench by a horizontal distance of 2-feet.

The pipes were connected to the manifold as shown in Figure 3.2-2 and discussed in Section 3.2.1. The fluid used in the well loops was a 20% solution of food-grade propylene glycol in water.
c. Fuel Cell and Hydrogen Storage

The 5-kW fuel selected for the HESS II project was the GenCore 5T48 manufactured by Plug Power Corp\textsuperscript{3}. This fuel cell runs on 99.95\% pure, dry hydrogen and has a Proton Exchange Membrane (PEM) fuel cell stack with 60 individual PEM cells sandwiched together to produce 48 volts DC and a maximum current of 109 Amperes. Under full power the GenCore 5T48 emits 2 quarts of water per hour. The fuel cell stack is liquid cooled and has a liquid-to-air heat exchanger to eliminate the excess low-grade heat developed in the fuel cell stack. The normal operating temperature of the GenCore 5T48 is 55 degrees Celsius (C) or 131 degrees Fahrenheit (F). All of the fuel cell operations and safety systems are under computer control.

The principal application of this type of fuel cell has been as an outdoor power backup for telecommunication providers (both wireless and wired). Figure 3.1.2-2 shows a typical outdoor installation of the backup power system with the 5-kW GenCore 5T48 PEM fuel cell to the left and a hydrogen storage cabinet to the right. The hydrogen storage cabinet can store up to six cylinders of hydrogen divided into two banks.

Figure 3.1.2-2. 5-kW PEM Fuel Cell (left) and Hydrogen Storage Cabinet

For more information on the GenCore 5T48 PEM fuel cell, see the following links:
http://www.fuelcellmarkets.com/content/images/articles/GenCore_Telecom_Datasheet.pdf
c1. Indoor Fuel Cell
One of the main objectives of using a fuel cell in the HESS II project was to capture and use the heat cogenerated when the fuel cell produces power. Approximately 35% of the input hydrogen energy is converted into electricity, while 65% is rejected as low-grade heat. This low-grade heat was to be captured and used to provide spatial heating or be stored in an insulated tank farm (Section 3.1.2-f). To minimize the cogenerated heat losses it was decided to bring the fuel cell inside the HESS II laboratory building, but to keep the hydrogen storage cabinet outside the building. This decision resulted in several technical and other safety issues to be discussed in a later section. Several modifications were made to the GenCore 5T48 fuel cell. To capture cogenerated low-grade heat from the circulating 55-degree (C) cooling liquid in the fuel cell, a flat-plate, liquid-to-liquid heat exchanger (HX) was placed in series with the temperature-controlled closed-loop fuel cell cooling system. Provision also was made for a direct outside air supply to the fuel cell, while the water emitted was piped to a drain and the fuel cell cathode was vented to the outside.

![Modified GenCore 5T48 fuel cell with a flat-plate heat exchanger (in the blue cooler, left) and partially built duct for outside air supply.](image)

Figure 3.1.2-3.

c2. Outdoor Hydrogen Storage Cabinet
Safety and Rochester building code requirements require that the hydrogen storage cabinet be placed at least 25 feet away from the HESS II building. Hydrogen gas was piped into the laboratory through a buried ¼-inch stainless steel pipe encased in a sealed 1” diameter CPVC, Schedule 80, pipe. The elevation of the base of the hydrogen storage cabinet is about six feet above the HESS II laboratory floor. Thus, any hydrogen leakage in the ¼” stainless steel pipe slanting upward, would naturally vent away from the building. The above-ground portion, a gray connection box and a vertical CPVC pipe, of the venting system is shown in Figure 3.1.2-4. The cabinet also had electronic sensors that measured the hydrogen pressure in the tanks. The cabinet holds up to six hydrogen gas cylinders (29.4 liters @ 2400 psi) in two parallel banks.
d. DC/AC Inverter
As mentioned in the previous section, the 5-kW GenCore 5T45 produces 48 volts (DC) at a maximum of 109 Amperes. The Trane GEV-018 heat pump requires 230 volts, single phase AC. It was therefore necessary to convert the power from DC to AC using a 5.5 kW/48-volt (DC), Xantrex SW5548 Series II DC/AC Inverter with a grid-tie interface (GTI).  Unfortunately, this unit produced only 120 volts/60 Hz, but due to the budget cost constraints we could not afford to buy two of these Xantrex SW5548 units and series couple them to produce 240 volts/60 Hz. So the output of the SW5548 was directly connected to the power grid, and 240 volts was obtained from the power grid to supply the Trane GEV-018 heat pump.

e. Water Heater
An electric, 80-gallon, 4.5 kW Rheem Solaraid HE, 81V80HE-1, water heater with a built-in heat exchanger is part of the HESS II plant. The heat exchanger is made of 120 feet of 5/8” copper tubing wound into a coil and attached inside the tank wall with convenient inlet and outlet connections. Measurements showed that this type of heat exchanger was not very effective at the “high” flow rates used on the project. The thermostat of the tank was set to 55 °C (to mimic the fuel cell operating temperature) when the water heater coil was in service as the heating source. The water heater electric coil was used as fuel cell substitute to reduce the high cost of

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4 For more technical information, see links:
http://www.affordable-solar.com/xantrex.sw5548.series.II.inverter.htm

5 More technical information can be found in:
http://waterheating.rheem.com/content/resources/documents/specsheets/RHSolaraidHE.pdf
the significant volume of hydrogen consumed by the GenCore 5T48 PEM fuel cell (one cylinder of hydrogen at 50 Amperes (half-power) in 3.8 hours). One 29.4-liter cylinder of 99.95% pure, dry hydrogen at 2,400 psi with delivery and tank rental, costs $64.00 in Rochester, Minnesota. This computes to a hydrogen fuel cost for continuous running of about $19.00 per hour or $3,183 per week.

f. Stratified Thermal Storage (Tank farm)
A 400-gallon insulated stratified thermal storage system (STSS) was designed, built and tested. The STSS consisted of four identical 100-gallon, 28.5" x 51.25", closed-head polypropylene tanks, Part # 5319, US Plastic Corp. The tanks were mounted on a wooden platform as shown below in Figure 3.1.2-5.

The initial objective was to bury the STSS in the ground outside the HESS II laboratory about four feet below the surface. As a result, the propylene tanks needed a cage-like frame structure to prevent any damage to the non-structural tanks. Figure 3.1.2-6 shows a 3-dimensional solid model of the overall treated wooden frame.

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6 Specific information can be found in: http://www.usplastic.com/catalog/product.asp?catalog%5Fname=usplastic&category%5Fname=30&product%5Fid=11984
The four 100-gallon, closed-dome polypropylene tanks were all identically modified to include a flange mounted drain with double rubber gaskets at the bottom of the tank and a flange mounted inlet connection with a rubber gasket at the top of the tank. The tanks were carefully positioned on the wooden platform and plumbed with 1” CPVC Schedule 80 pipes in a parallel configuration. Great care was taken to have the same lengths of pipe from the common inlet and outlet supply pipes to each tank inlet and outlet respectively. This enabled very evenly divided flows through each of the tanks. The stratified thermal storage system inlet and outlet were connected to the heat supply system manifold with 1” CPVC Schedule 80 pipes.

The initial decision to bury the STSS below grade outside the HESS laboratory was later reevaluated owing to the cost of the excavation and soil removal and the disruption of to the normal operation of the Quarry Hill Nature Center. It was decided instead to leave the STSS inside the HESS II laboratory. A modified non-load bearing wooden frame shown in Figure 3.1.2-6 was then constructed around the tanks. It was mounted on the wooden platform of Figure 3.1.2-5 that was placed above two 2” thick sheets of 25 psi extruded polystyrene insulation with a total R-value of 20. The wooden frame was insulated on the four sides and top with nominally R-15 extruded polystyrene. The insulated enclosure was thoroughly air sealed with modified bituminous rubber membrane (Grace Vycor) and all piping and wiring penetrations sealed with acoustic sealant. The overall insulated tank enclosure had a base of 82.5” x 71.5” and a height of 72.5”, yielding an enclosed volume of about 9.2 cubic yards.
Each tank was instrumented with a T-type thermocouple probe to measure the bulk temperature of the tank. The probe was 28” long and vertically mounted from the top. One of the tanks was equipped with a 40” long, vertically mounted, water level transmitter\(^7\). The temperature of the fluid input and output to the heavily insulated enclosure was measured by T-type thermocouple probes. The tanks and the tank farm loop were purposely not pressurized.

\(^7\) Information about the water level transducer can be found in link: [http://www.dwyer-inst.com/htdocs/level/SeriesCLTPrice.cfm](http://www.dwyer-inst.com/htdocs/level/SeriesCLTPrice.cfm)
g. Heat Exchanger
A brazed flat-plate heat exchanger was used to effectively capture the low-grade heat from the fuel cell. The brazed flat-plate heat exchanger consists of stainless steel plates with a special high efficiency heat transfer surface. The plates are then copper-brazed together at high temperature, allowing the heat exchanger to be leak tight and rugged. Analysis resulted in the selection of a liquid-to-liquid, 36-plate heat exchanger\textsuperscript{8}. Model # FP10x20-36 with 1-1/2” MPT ports manufactured by FlatPlate, Inc. Y-strainers were placed in front of both input ports to the heat exchanger to prevent particles and debris in the circulating fluids from dislodging the space between the 36 plates.

\textsuperscript{8} For more information about the flat-plate heat exchanger, see product catalog on page 14 on link: http://www.flatplate.com/pdf/refrig/Refrigeration\%20Catalog.pdf
The A-side of the heat exchanger was connected in series with the circulating closed-loop cooling liquids in the GenCore 5T48 PEM fuel cell. Note also that a manual bypass was installed on the B-side of the heat exchanger, see Figure 3.2-1 with the heat exchanger HX1 and pump P3 OFF. This would allow the fuel cell to attain proper operating temperature of 55 degrees C during startup before the heat exchanger was coupled to the heat supply manifold.
h. Fuel Cell Low-grade Heat Emulator
h.1 Fuel Cell Failure
On the afternoon of Tuesday, September 25, 2007, the 5T48 Gencore fuel cell exhibited strange behavior after it had been working well in the morning during normal data collection. From the fuel cell data terminal display, it appeared that not enough hydrogen was reaching the fuel cell stack, while the computer readout still showed 730 psi of hydrogen supply pressure. The fuel cell shut down with an error code indicating that excess hydrogen was present in the fuel cell cabinet. Further, the individual cell voltage bar chart readout indicated that only half the cells were producing power which led to the conclusion that the likely source of failure was a stack hydrogen leak at the first cell not producing power (about half way down the stack).

While checking the outside hydrogen storage cabinet, the observed output pressure of the pressure / flow control regulator slowly increased from 80 psi to 150 psi, went back to 80 psi and slowly increased again to 150 psi, and then repeated the cycle. When the pressure reached 150 psi, hydrogen was vented to the atmosphere. This regulator apparently has a dual function (although this has not been verified independently):

(a) As a pressure regulator to reduce the hydrogen tank pressure that can be as high as 2400 psi, down to a fixed 80 psi.
(b) As an excess flow cutoff, that is, when the gas flow exceeds the factory specified minimum (signifying a leak), the flow is shut off.

The ability of the flow control regulator to keep the outside hydrogen pressure constant at 80 psi before it enters into the HESS II lab is crucial for the operation of the fuel cell stack. Within the fuel cell enclosure there is another pressure regulator that reduces the hydrogen pressure from 80 psi to 10 psi, which is the correct hydrogen pressure for the fuel cell stack. The maximum input pressure tolerance of this onboard regulator is unknown and is crucial to the durable operation of the stack – it is unlikely that this regulator is able to tolerate the full 2400 psi supply pressure.

The fuel cell was shut down, and after contacting the supplier, H.M. Cragg, Minneapolis, MN, about the problem a service technician came down to inspect the system on October 26, 2007 and observed the same problem. The fuel cell enclosure was opened and the fuel cell stack inspected. Leakage of fuel cell coolant (propylene glycol) was observed on the stack base plate indicating leakage from the stack itself which is consistent with the hydrogen leak hypothesis noted above. A significant amount of coolant had pooled on the bottom of the enclosure. The radiator liquid level was very low and the overflow bottle was empty. All the coolant seals going into and out of the stack were dry as was all the coolant piping and pump equipment. Further, on attempting to start the system, an additional error was produced indicating that the stack voltage sensing array had failed, no doubt caused by the coolant leak short-circuiting the electronics. Thus the conclusion that indeed there is a hole in the stack causing leakage of hydrogen and coolant was confirmed.

It was agreed that the failure of the flow control regulator in the outside hydrogen storage cabinet was the source of the stack seal breach. Once this regulator failed, a pressure shock wave traveled through the hydrogen pipe and into the fuel cell. Apparently there are 3 additional safety systems in the fuel cell (although specifics of their design are unknown):

(a) The first system is a solenoid valve operated off a pressure transducer. The shock wave propagation speed is much faster then the response of the valve / pressure transducer / onboard data acquisition system and hence this system failed.
(b) The second system is the onboard pressure regulator. If this regulator is unable to withstand pressures much higher than 80 psi, then it too would have failed which is what the evidence indicates. Further because it is a spring loaded mechanical device, it has a slow response time to pressure pulses and would not have been able to react quickly enough.

(c) There is also a spring-loaded check valve that is supposed to close if the pressure exceeds 10 psi (presumably) which also was ineffective in restraining the pressure pulse for the same reason as in (b) above, a mechanical system with a response time that is too slow.

Thus the high pressure pulse reached the stack and breached the stack seal at its weakest point. The design flaw apparent from this sequential failure is the absence of any safety system with a response time fast enough (of order tens of nanoseconds) to respond to incident high pressure pulses. This can only be regarded as a severe system design oversight given the criticality to maintain the stack incident pressure at 10 psi to ensure durable operation.

As a result, the progress of the HESS II research project was delayed by several weeks. It was decided that a repair of the fuel cell would take too much time (because of a dispute with the supplier about the warranty) and thus severely impact the HESS II schedule, so the solution devised was to fabricate an emulator to simulate the low-grade heat produced by the GenCore 5T48 fuel cell.

h.2 Fuel Cell Low-Grade heat Emulator Design
The very successful emulator design provided more test flexibility than the fuel cell by supplying heat at a variable temperature and flow rate. Note that the GenCore 5T48 has a constant flow rate of 7.5 gpm that starts only when the temperature of the coolant reaches 55°C. A block diagram of the HESS II emulator design is shown in Figure 3.1.2-11.
Figure 3.1.2-11. Block diagram of the emulator simulating HESS II fuel cell low-grade heat system.

Referring to the A-side of the heat exchanger, a solution of water and food grade propylene glycol is pumped in a closed-loop consisting of the heat exchanger, a variable speed pump, an in-line flow meter, and a custom designed flow-through fluid heater with a variable power electric heating element.

The heat exchanger was discussed in Section 3.1.2-g. The variable speed pump was a Dayton Centrifugal Pump Head (pedestal pump) – 1/3 HP with a GE 1/3 HP DC motor: 1725 RPM, 90-Volt armature, permanent magnet field, NEMA 56C frame, rigid base mounting. A 1-10 GPM in-line flow meter allowed manual monitoring of the liquid flow in the closed-loop system. The flow-through fluid heater was fabricated from a of a 12-inch long, 4-inch diameter, copper tube with one end capped and the other end fitted with a flanged closing plate containing a screw-in water heater element (Incoloy sheathed electric resistance with a rating of 5.5-kW at 208 Volts). Copper tubes of ¾” diameter were attached to the end cap and flange plate to provide the fluid inlet and outlet ports. Controls allowed adjustment of the power applied to the heating element using a clamp-on current meter. A Dayton DWT-pre-charge variable pressure 2.1-gallon water tank was used to maintain constant loop pressure and absorb any steam while a Bourdon pressure gauge allowed manual pressure monitoring. The flow-through heater and loop piping were thermally insulated. A photograph of the assembled fuel cell low-grade heat emulator is shown in Figure 3.1.2-12.
i. Heat Recovery Ventilation (HRV) system
The main purpose of the heat recovery ventilation (HRV) system besides normal ventilation was to purge the air quickly in the laboratory in the case of a hydrogen gas leak when operating the fuel cell. This was a City of Rochester code requirement when using hydrogen inside. The multiple speed HRV blowers were wired up in two modes, high speed in an emergency and low speed for normal ventilation. The HRV complements the energy efficiency of the modern building by filtering incoming fresh outdoor air before it enters the heat-recovery core where it is preheated by the outgoing, stale contaminated air. The HRV then distributes the preheated fresh filtered air throughout the laboratory by direct ductwork installed specifically for the HRV. The return used air intake to the HRV was passed through a 10-foot, 1-foot diameter duct before entering into the HRV unit.
3.2. METHODS OF OPERATION

The objective with the HESS II Project is to research various ways one could improve the efficiency of different configurations of a standard ground-sourced (geothermal) heat pump system using cogenerated low-grade heat. Several modifications were made to the basic ground-sourced heat pump system to facilitate the integration of variable amounts of low-grade heat into the system. The efficiency was measured in terms of the Coefficient of Performance (COP) for heating and the Energy Efficiency Ratio (EER) for cooling.

The heating efficiency of ground-source heat pump systems is indicated by their coefficient of performance (COP), which is the ratio of heat provided in BTU per BTU of energy input. Their cooling efficiency is indicated by the Energy Efficiency Ratio (EER), which is the ratio of the heat removed (in BTU per hour) to the electricity required (in watts) to run the unit.

Discussed in this section are three methods of operation:

- Standard Heat Pump Operation with Wells
- Heat Pump Operation with a Combination of Wells and Low-grade Heat
- Heat Pump with Low-grade Heat Only

Before we look at the three methods of operation, it is important to give a description of the HESS II plant relative to the modes of operation. These modes of operation are described by the configuration of various loops of liquid flow. The HESS II plant is capable of implementing several modes of operations by automatic remote control of 3-way valves, variable and fixed speed pumps, etc. A simple overall diagram of the HESS II plant, where the directions of flows are shown by arrows\(^9\), is illustrated in Figure 3.2-1.

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\(^9\) Note that some flow directions have been insured by the means of check valves.
Refer now to the overall HESS II plant diagram in Figure 3.2-1, and let us start on the left side of the diagram at the fluid output of the heat pump. Here 3-way valve S1 has two settings, A and B. In the A-setting, the fluid from the heat pump flows through the heat exchanger (HX2) housed inside the hot water heater\textsuperscript{10}. Here it can either pick up heat (heating mode) or deliver heat.

\textsuperscript{10} Rheem residential solar electric water heater, 4.5 kW, with internal heat exchanger
(cooling mode). The flow rate in the loop is either controlled by either a fixed speed pump (P1)\(^\text{11}\) or a variable speed pump (P2).\(^\text{12}\) The output from either pump passes the point E into a proportional valve\(^\text{13}\). Here the range of control variable x is 0 ≤ x ≤ 1. When x = 0, the total flow at point E passes down at point G and into the manifold and into one or both wells. The return flow from the wells is then circulated back to the heat pump. On the other hand, when x = 1, the wells are bypassed and the total flow at point E passes by point F and back into the heat pump. If the 3-way valve S1 is in the B-setting, the hot water heater is bypassed and typically, the proportional control setting x = 0. This last mode can be considered to be the standard operating mode for the ground-source heat pump.

Next, let us continue with the upper right side of the diagram, where the low-grade heat generator could be from any source, for example from a fuel cell\(^\text{14}\) or a fuel cell emulator\(^\text{15}\). The heated fluid, which is a byproduct of cooling some type of equipment or other types of low-grade heat, are circulated into the A-side of a heat exchanger (HX1).\(^\text{16}\) The liquid flow through the B-side of HX1, provided by a variable speed pump (P3)\(^\text{17}\) or fixed speed pump (P4)\(^\text{18}\), picks up the heat from the heat exchange and passes it into the top of the hot water heater tank. (The electric heating element in the tank would normally be off in this situation.) The output flow at the bottom the hot water tank is either circulated into the top of the 400-gallon tank farm,\(^\text{19}\) if 3-way valve is in the C-setting or the tank farm is bypassed if the 3-way valve is in the D-setting. This completes a brief description of the variable modes of the HESS II plant.

A more detailed description of the three main modes used for testing are described in the next three subsections.

3.2.1 Standard Heat Pump Operation with Wells

The method used for the HESS II project for the standard heating and cooling with the Trane GEV-018 reversible ground-source heat pump system is shown diagrammatically in Figure 3.2-2. Starting with the output of the heat pump, the liquid water/glycol mix is pumped at a controlled flow rate through the manifold and down into either one of the wells or both (Wells 1 and 2). The selection of wells is done manually with ball valves in the manifold. The return flow from the well(s) is combined in the manifold and circulated back to the heat pump. Temperature measurements of the circulating liquids are made at the two well heads. Well input flow temperature (T1 and T3) and well return flow temperature (T2 and T4). The temperature is also measured at the input (T5) and the output (T6) of the heat exchanger. The total loop flow is measured by flow-meter at the pump output, F1, and the flow in Well #2 is measured by flow-meter F2. The flow in Well #1 is computed as the difference between F1 and F2 flow measurements. The laboratory air is circulated through the ductwork and into an air-to-liquid heat exchanger located in the heat pump. The temperature is measured both for the input (T7) and output air (T8). The airflow is also measured at the input duct section to the air-to-liquid heat exchanger.

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11 Grundfos centrifugal pump, 1/6 HP
12 Procon brass vane (positive displacement) pump, ½ HP
13 Dwyer modulating electric ¾” NPT 3-way brass, automated ball valve
14 Plug Power PEM GenCore 5T48, 5 kW, 48 VDC
16 See Section 3.1.2h.
16 Flat Plate heat exchanger, FP 10x20-36
17 Procon brass vane (positive displacement) pump, ¼ HP
18 Dayton self-priming centrifugal stainless steel pump
19 See Section 3.1.2f.
(a) During the heating season heat is being extracted from the wells and circulated into the heat pump’s compressor system that “lifts” the temperature from the liquid input (T5) to the output heated air temperature (T8). In the heating season the following temperature conditions exist. For the wells: T2 < T1 and T4 < T3, for the heat pump: T5 > T6 and for the air duct system: T7 < T8.

(b) In the cooling season, the operation of the vapor-compressor refrigeration cycle is reversed, see Figure 3.1.2-1. Now the wells act as heat sinks and heat from the room air is deposited into the cool wells. During the cooling season the opposite temperature conditions exist. For the wells: T2 > T1 and T4 > T3, for the heat pump: T5 < T6 and for the air duct system: T7 > T8.

Figure 3.2-2. Standard ground-source heat pump configuration with wells
3.2.2 Heat Pump Operation with a Combination of Wells and Low-grade Heat

This mode of operation for the heat pump is typically meant for the heating season, where the heat energy drawn from the geothermal wells is being supplemented by low-grade heat from any source. For example, the low-grade heat could come from the excess heat being developed by a fuel cell, cf. the HESS II patent, or the heat could come from thermal solar panels, gas turbines, biomass operations, etc. The source of low-grade heat is irrelevant. In Figure 3.2-3 below the low-grade heat is being simulated by a Rheem 80-gallon, 4.5 kW, electric Solaraide HE water heater with an internal heat exchanger consisting of 120 feet of 5/8" copper tubing wrapped around inside the tank and secured to the tank.

![Diagram showing heat pump operation utilizing a variable mixture of the wells and stored low-grade heat]

Figure 3.2-3. Heat pump operation utilizing a variable mixture of the wells and stored low-grade heat
Referring to Figure 3.2-3 at the output of the Reversible Heat Pump the cooled outlet liquid water/glycol mix (at temperature $T_5 = T_9$) is passed through the heat exchanger and picks up heat from the low-grade heat storage ($T_6 > T_5$). Then through the pump and to the proportional valve that will under remote control split the flow ($F_1$) such that one portion ($F_2$) is passed through the wells, while the remainder of the flow ($F_1 - F_2$) is bypassing the wells. The pump used in this configuration could be either the fixed speed pump $P_1$ or the variable speed pump $P_2$, as defined in the previous section. The flow from the wells and the bypassed flow are combined at point M and returned to the fluid input of the heat pump at temperature $T_8$.

In other words, at one extreme ($x=0$), the proportional valve allows all the loop fluid to pass through the wells, in effect discarding the stored low-grade heat to the ground. At the other extreme ($x=1$), the circulating fluid bypasses the wells and allows all of the heat collected by the heat exchanger in the low-grade heat storage to be delivered to the heat pump (discussed in the next section).

The bypass diversion $x$ of the proportional valve is very important to control the amount of heat to be delivered to the heat pump. At one extreme end, $x = 0$, the proportional valve can let all circulating geothermal fluid pass through the well system which will reduce the heat supplied to the inlet of the heat pump. At the other extreme, $x = 1$, the circulating fluid bypasses the wells and allows all of the heat picked up by the heat exchanger located inside the low-grade heat storage to yield the highest input temperature $T_8$ at the fluid input to the heat pump input. The temperature $T_8$ at the input of the reversible heat pump as a function of the output fluid temperatures $T_1$ and $T_3$ of well 1 and 2, respectively, and the output fluid temperature $T_6$ of the heat exchanger of can be expressed as:

$$T_8 = x \times T_6 + (1-x) \times (T_1 + T_3) / 2.$$  

Note that the fluid input temperature $T_8$ to the heat pump can be continuously varied between the thermal storage and average well outlet temperatures. That completes the closed-loop circulation system.

Of course during the cooling season, the fluid output temperature $T_9$ of the heat pump is much higher than the fluid input temperature $T_8$. This low grade heat could, for example, be used to preheat water to a hot water tank, a swimming pool or some other heat storage.

### 3.2.3 Heat Pump Operations with Low-grade Heat Only

This mode of operation for the heat pump typically is applicable in the heating season, where the thermal energy drawn from the geothermal wells is supplemented by a variable quality of low-grade heat supplied from any source. For example, the low-grade heat could come from the excess heat being developed by a fuel cell, cf. the HESS II patent, or the heat could come from solar thermal panels, gas turbines, etc, that is, the source of low-grade heat is arbitrary. In Figure 3.2-4 below, the low-grade heat is being simulated by a 4.5 kW electrical resistance heating element within an 80-gallon hot water tank (Section 3.1.2-e) or a fuel cell low-grade heat emulator (see Section 3.1.2-h).
Figure 3.2-4. Heat pump operation with a source of low-grade heat

As with Section 3.2.2, this operation also is applicable to the heating mode of operation. The low-grade heat can, for example, be generated from a fuel cell, a gas turbine, etc. and is stored in the low-grade heat thermal storage system which was described in Section 3.1.2f and referred to henceforward as the "tank farm". As can be seen from Figure 3.2-4, the main elements in this experimental system are: a low-grade heat generating device (WHGD) with an internal pump, a 400-gallon tank farm, a 80-gallon hot water tank, two heat exchangers, two constant speed pumps, two variable speed pumps and, not least, a reversible heat pump. There are three closed-loop fluid systems shown in Figure 3.2-4 that will be designated as the Low-
grade Heat Generating Loop, the Low-grade Heat Storage Loop and the Heat Pump Loop. By individually controlling the flow in each of the three loops, we can optimize the efficiency of the overall system. Referring to Figure 3.2-4, the three loops are:

(a) Low-grade Heat Generating Loop
Starting with the hot output flow from the WHGD at temperature (T1), which is pumped by a variable or fixed speed pump located inside the WHGD. The flow from the WHGD enters the A-side of the heat exchanger 1 (HX1), transfers heat to the B-side and flows back to the WHGD at a cooler temperature (T2), where T1 > T2. The HX1 was discussed in Section 3.1.2g.

(b) Low-grade Heat Storage Loop
Heated fluid at temperature (T4) flows from the B-side of HX1 to a hot water tank with a bulk temperature (T6) that has an internal heat exchanger (HX2). See Section 3.1.2e for more information about the hot water tank and HX2. Cooler liquid from the bottom of the hot water tank flows into the 400-gallon thermal storage (tank farm) that has a bulk temperature of T5. Cooler water from the bottom of the tank farm is pumped back to the B-side of the HX1 at temperature T3 through either a fixed speed pump (P4) or a variable speed pump (P3) at a flow as measured by flow-meter F2. This completes the closed Low-grade Heat Storage Loop in which the liquid temperatures are T4 > T6 > T5 > T3.

(c) Heat Pump Loop
The third closed-loop system is defined as the cool water, at temperature T7, is returned from the reversible heat pump and flows into the heat exchanger (HX2) and picks up heat from the 80-gallon water tank attaining a liquid temperature of T8, where T8 > T7. The liquid flow in this loop is provided by either the fixed speed pump (P1) or the variable speed pump (P2) and is measured by flow-meter F1. Finally, the fluid flows back to the input of the reversible heat pump at a liquid temperature T9. This completes the closed Heat Pump Loop in where the liquid temperatures are T8 ~ T9 > T7.

3.3 Instrumentation

This section covers most of the important sensors used in the HESS II plant. Most of these sensors were connected to the computer controlled data acquisition system (DAS). A few sensors on the fuel cell emulator (Section 3.1.2-h) were read and adjusted manually. All cabling were shielded and carefully grounded to minimize the effects of noise. Most sensor inputs showed very high signal-to-noise ratios (SNR), while some sensor signals, like high powered AC currents and wattage from the heat pump compressor, etc. needed additional filtering by means of median filters.

3.3.1 Thermocouples

There are a total of 46 T-type thermocouples sensors wired up in the HESS II plant. Most of the thermocouple sensors are of the T-type thermocouple probes, but there are a few T-type thermocouple wire junction sensors.
(a) T-type thermocouple probes\textsuperscript{20}

The T-type thermocouple probes have stainless steel probes that are in contact with the circulating fluids in the HESS II system. They also have a quick connect handle with an integral locking mechanism that secures any miniature probe handle in use. Most of the thermocouple sensors are securely grounded and are connected to the input of the DAS using shielded extension wires and cables.

(b) T-type thermocouple wire-junction sensors\textsuperscript{21}

The T-type thermocouple wire-junction sensors are used in areas where they do not come in contact with water. In this project, the wire-junction sensors were used below grade, like at input and the output of the two well heads. A special non-invasive design was used at the well heads where an 8-inch piece of $\frac{3}{4}$" copper pipe was spliced into the 1-inch outer diameter (OD), black High Density Polyethylene tubes. The copper pipe make good thermal connection with circulating fluids and the wire-junction in thermal contact with the copper tube picks up the correct temperature. The outside of the copper tube is insulated with a water tight plastic sleeve to keep the wire-junction dry.

Due to the decision not to bury the insulated tank farm into the ground, it was necessary to estimate the thermal energy loss effects of the below ground temperature on the insulated tank farm. As a result, an eleven-foot rod with twelve T-type thermocouple wire-junctions used to measure below ground temperatures at 1-foot intervals. The top portion of the white 1" diameter, PVC rod is buried 1-foot below the ground surface. Hence, the contraption will measure below ground temperatures from 1 foot to 12 feet. See Figure 3.3-1 for more detail on the 11-foot rod.

\textsuperscript{21} http://www.omega.com/pptst/GG_T_TC_WIRE.html
3.3.2 Flow Sensors

The use of flow sensors is very important to the HESS II research. The Hess II plant hydronic manifold was built and modified in stages as the research progressed. When new brass parts, unions and copper pipes were fitted together it was impossible to prevent residue of pipe dope to contaminate the geothermal fluids. This residue and other residue particles has a tendency to clog up the flow sensors. It was therefore necessary to place Y-strainers in front of some flow sensors and other sensitive equipment.

(a) Paddlewheel flow sensor\textsuperscript{22}

The paddlewheel liquid flow sensor is a reliable and low cost device for measuring flow. The Kobold, DRG1165N5L343, uses a Hall Effect sensor to detect the passing of permanent magnets imbedded in the rotating paddle. The DRG flow sensor is supplied with a 4-20 mA output range and will measure liquid flow in the range 0.6 – 12.0 gpm. The reason the sensor will not measure flows below 0.6 gpm is due to friction/stiction effects in the mechanical paddlewheel structure.

\textsuperscript{22} For more details, see: http://www.maselmon.com/assets/63/k_drg.pdf
(b) Turbine flow sensor
This is another flow sensor made by Kobold Instrument, Inc. Instead of using a paddlewheel, it uses a vane-axial turbine with a Hall Effect sensor to detect the passing of permanent magnets imbedded in the rotating turbine blades. This sensor has a 4-20 mA output range and will measure liquid flow in the range 0.6 – 10.5 gpm. The reason the sensor will not measure flows below 0.6 gpm is due to friction/stiction effects in the mechanical turbine structure.

(c) In-line flow meter
This in-line flow meter feature rugged construction, easy installation, but requires manual reading of the liquid flow rates. Range of liquid flow is 1 – 10 gpm. The in-line flow meter is used in the fuel cell emulator.

3.3.3 Pressure transducer
The fluid pressure in any of the loops involving the heat pump is important. Usually the pressure should be between 30 psi – 60 psi. To automatically measure the loop pressure the HESS II project is using a solid state pressure transducer with amplification and range 0 – 100 psi.

3.3.4 Water level transducer
The water level in each of the tanks of the tank farm is important. Due to the great care taken in making the four-tank design symmetrical, it was only necessary to measure the water level in one of the tanks. The vertically mounted continuous water level transmitter with a 40” indicating length was partially submerged into the liquid. The water level was electrically measured by a floater and had a current output of 4 – 20mA. The liquid input entered each tank from the top and would freefall down to the water surface. This caused a lot of wave action on the liquid surface in the tank which resulted in erratic up-down motion of the floater and output current. To minimize this effect, a surface-wave-damper was constructed. It consisted of a 4-foot long, 6” diameter PVC tube vertically mounted inside the tank with the 40” long water level transducer inserted inside. This solved the problem and greatly smoothed out the noise effects of the surface waves on the water level reading.

More can be found in the link:
http://www.maselmon.com/assets/63/K_DRS.pdf

More information about in-line flow meters is found in link:

See the link for more detailed information:

Link to the water level transducer:
http://www.dwyer-inst.com/htdocs/level/SeriesCLTPrice.cfm
http://www.dwyer-inst.com/htdocs/level/SeriesCLTSpec.CFM
3.3.5 Watt transducer 27

In order to determine the Coefficient of Performance (COP) for heating and the Energy Efficiency Ratio (EER) for cooling, the power used by the heat pump compressor and fan must be determined. This unit is a watt transducer that utilizes a Hall-effect multiplier to provide continuous multiplication of voltage and current to accurately measure real AC power delivered to the load. Output voltage is 0 – 5 VDC.

3.3.6 Air flow sensor 28

The Ebtron airflow measuring station is a hot wire probe that measures the airflow in the HESS II heat pump air duct system. To achieve close to laminar flow at the measuring station, a 15-foot, 16-inch diameter round duct was mounted in front of the return air path to the heat pump. The hot wire probe was located 14 feet away from the inlet to the round air duct. Operationally, this 15-foot duct addition has been working satisfactory.

3.4 Data Acquisition system Hardware

To analyze and understand the dynamics of the underlying processes in a physical plant, like the HESS II plant, it has to be instrumented with a variety of sensors placed in strategic locations around the plant. The data acquisition system is electronics based, and it is made of hardware and software. The HESS II real-time Data Acquisition System (DAS) is the link between the plant sensors and a computer system that measures and logs a multitude of electronic sensor values in real time. Under computer program control, all electronic sensor signals are properly filtered, periodically sampled and the measured sensor values are stored in computer memory for later analysis. The system has also output features that allow the changing of set-points for the various actuators, like the position of valves, the speed of pumps and turning ON/OFF various subsystems. The hardware part is made of sensors, cables and electronics components. The software part is made of the data acquisition logic and the analysis software is discussed in the next subsection (Section 3.5).

For the hardware implementation, a decision was to use a high quality PC-based DAS made by National Instruments Corp. The National Instruments SCXI system provides high-performance signal conditioning and switching. It can measure sensor outputs and raw signals, generate currents or voltages, monitor digital lines, or route signals with switching in a single integrated system. The system has modular capabilities like analog inputs and outputs, and digital I/O. The SCXI offer several means for filtering and conditioning options, like amplification, isolation, simultaneous sampling, and programmable filtering. Below are some of the major hardware components in the HESS II DAS:

27 More information about the watt transducer can be found on link: https://www.ohiosemitelectronics.com/productDetails.asp?sku=PC5-059CX5
28 For more detailed information about the Ebtron, see Ebtron Product Catalog, pages 114 – 130, on link: http://www.ebtron.com/Web_Pdfs/Ebtron_Catalog_v4.05.pdf
3.4.1 **SCXI -1000, 4-slot chassis**\(^{29}\), 120VAC

The low-noise, rugged SCXI-1000 chassis houses, power, and control the SCXI modules and conditioned signals. The SCXI chassis architecture includes the SCXIbus, which routes analog and digital signals and acts as the communication conduit between modules. Chassis control circuitry manages this bus, synchronizing the timing between each module and the DAS device. It enables to scan input channels from several modules in up to 333,000 samples/second.

3.4.2 **Two SCXI -1102, 32-Channel Thermocouple Amplifier**\(^{30}\)

The SCXI-1102 module is designed for high-accuracy, low voltage, thermocouple measurements. Each input channel includes an instrumentation amplifier and a 2 Hz lowpass filter. The HESS II project used two of the SCXI-1102 modules with the accessory of two SCXI-1303\(^ {31}\), 32-channel isothermal terminal blocks to attach the shielded thermocouple sensor wires. The terminal block is a shielded device with 32 pairs of screw terminals that connect to the SCXI-1102/B/C and SCXI-1100 modules. The SCXI-1303 has a high-accuracy thermistor cold-junction temperature sensor, and an isothermal copper plane to minimize the temperature gradients across the screw terminals when thermocouple measurements are taken.

3.4.3 **SCXI -1163R, 32 Channel Solid-State Relay Multiplexer**\(^{32}\)

The SCXI-1163R module includes 32 normally open, or Form A, optically isolated solid-state relays, arranged in eight banks of four relays with one common pole for each bank. The device can switch high-voltage loads, up to 240 VAC/VDC, and up to 200mA. The solid-state relay channels can be used to switch a wide range of AC and DC voltage and power signals to control field devices. The signal cables are attached to the SCXI-1326\(^ {33}\) terminal block, which is a shielded board with supports to connect it to the SCXI-1163R module front connector. The terminal block has 48 screw terminals for easy signal connections.

3.4.4 **SCXI -1124, 6-channel DAC module**\(^{34}\)

The SCXI-1124 has six isolated digital-to-analog converters with voltage or current outputs. It provides 0 to 20 mA current outputs using the SCXI Process current resistor kit, Part number: 776582-01. The SCXI-1325 shielded terminal block has screw terminals for easy signal attachment to the SCXI-1124. The SCXI-1325\(^ {35}\) high-voltage terminal block is a shielded board with 26 screw terminals for easy connection to the SCXI-1124 input connector. One pair of screw terminals connects to the SCXI-1124

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\(^{29}\) For more information, see:  
http://nees.unr.edu/lab_files/SCXI_1001Chassis.pdf

\(^{30}\) Information about the thermocouple amplifier module is found on link:  

\(^{31}\) More detailed information can be found on link:  

\(^{32}\) More information can be found on link:  

\(^{33}\) More detailed information on link:  

\(^{34}\) Information can be found on link:  

\(^{35}\) This link will give more information:  
chassis ground. The remaining 24 screw terminals are for signal connection to the six module-output channels.

Figure 3.4-1. Installation of data acquisition system (DAS) chassis SCXI-1000, with four input modules, four terminal blocks, sensor and control wiring.

The modules described above were inserted into a SCXI-1000 backplane shown in Figure 3.4-1. The backplane was connected to a PCI-6220 M-series multifunction DAQ card inserted into a 32-bit PCI slot in the laboratory computer. The PCI-6220 controls the SCXI-1000 multiplexing functions and converts the conditioned analog signals output by the SCXI-1000 to digital signals via a 16-bit analog-to-digital (A/D) converter. The modules in the SCXI mainframe as well as the channels in each module were configured manually via the National Instruments “Measurement Explorer” application.

3.5 Data Acquisition System Software

3.5.1 LabVIEW Software

LabVIEW (short for Laboratory Virtual Instrumentation Engineering Workbench) is a platform and development environment for a visual programming language from National Instruments. The programming language used in LabVIEW, called G, is a dataflow programming language. Execution is determined by the structure of a graphical block diagram (the LV-source code) on which the programmer connects different function-nodes by drawing wires. These wires propagate variables and any node can execute as soon as all its input data become available. Since this might be the case for multiple nodes simultaneously, G is inherently capable of parallel execution. Multi-processing and multi-threading hardware is automatically exploited by the built-in scheduler, which multiplexes multiple OS threads over the nodes ready for execution.
3.5.2 Real-time monitoring and control facility for the HESS II system

A valuable tool was designed and made available during performance testing. This was two real-time interactive monitoring screen images showing the updated values of temperature, liquid and air flow, and pressure around the HESS II plant. Other computed values, like the COP, were also displayed. The sensor values were updated every 20 seconds. Pump speeds and valve settings could also be controlled directly from the computer the two computer screens. This real-time monitoring facility was a great feedback tool when we were adjusting valves and pump speed settings.

Figures 3.5-1 and 3.5-2 depict the interactive image screens for the heat pump circuit and the low-grade heat emulator, respectively. These two interactive image screens allowed for adjusting pump speed and valve settings and also to keep track of sensor readings of temperature (green), flow (yellow) and pressure (pink) values for feedback of the HESS II process during system test.

Figure 3.5-1 Real-time interactive screen image of the heat pump circuit allowed for adjusting pump speed and valve settings and also keep track of sensor readings of temperature (green), flow (yellow) and pressure (pink) values for feedback of the HESS II process during system test. (All sensor values have been set to zero.)
Figure 3.5-2 Real-time interactive screen image of the low-grade heat emulator circuit allowed for adjusting pump speed and valve settings and also keep track of sensor readings of temperature (green), flow (yellow) and pressure (pink) values for the HESS II process during system test. (All sensor values have been set to zero.)
CHAPTER 4: RESULTS AND DISCUSSION

The experimental test results are presented in two major sections. The first section describes the heating season performance, that is, when the heat pump is adding heat to the conditioned space. The second section is structured in the same way as the first and describes the cooling season performance in which the heat pump removes heat from the conditioned space.

The following definitions are used throughout this Chapter:

Heat pump Coefficient of Performance (COP) = \frac{\text{Air enthalpy change}}{\text{Compressor + blower electrical energy}} \quad (4.1)

During the heating season, the air enthalpy change is the heat supplied to the air at the condenser, while during the cooling season, it is the heat removed from the air at the evaporator. The air enthalpy change rate (\dot{\dot{H}}) (gain in heating mode, loss in cooling mode) is determined from:

\dot{\dot{H}} = \dot{V}_{\text{air}} \rho_{\text{air}} C_P (T_{bd} - T_{cu}) \quad (4.2)

where \dot{V}_{\text{air}} is the volumetric air flow rate, \rho_{\text{air}} is the air density, \( C_P \) is the heat capacity at constant pressure, \( T_{bd} \) is the air temperature downstream of the blower, and \( T_{cu} \) is the temperature upstream of the condenser (in heating mode) / evaporator (in cooling mode) coil.

The average COP is simply an average of the COP with respect to an independent variable such as time or flow rate.

The Carnot heating Coefficient of Performance (\text{COP}_{\text{Car}}) is the maximum performance that can be attained by a reversible heat pump operating between given hot and cold temperatures limits and is defined as:

\text{Heat pump Carnot heating Coefficient of Performance (COP}_{\text{Car}}) = \frac{T_h}{T_h - T_c} \quad (4.3)

where \( T_h \) is the temperature of the hot sink (conditioned space air) and \( T_c \) is the cold source (circulating fluid) temperature. The heat ratio in heating mode is a measure of the effective heat exchange across the heat pump. In heating mode, this is given by:

\text{Heat ratio}_{\text{heating}} = \frac{\text{Heat supplied to air}}{\text{Heat removed from liquid}} \quad (4.4)

Conversely, in cooling mode, the heat ratio is given by:
Heat ratio_{cooling} = \frac{\text{Heat removed from air}}{\text{Heat rejected to liquid}} \quad (4.5)

The effectiveness ($\xi_E$) of the heat exchanger coupling the emulator and thermal storage fluid circuits is defined as:

$$\xi_E = \frac{\dot{V}_{\text{thermal storage circuit}}}{\dot{V}_{\text{min}}} \left( \frac{T_{\text{circuit, out}} - T_{\text{circuit, in}}}{T_{\text{emulator, in}} - T_{\text{emulator, out}}} \right)$$

where $\dot{V}$ is the volumetric flow rate and $T$ is the temperature. $\dot{V}_{\text{min}}$ is the minimum of $\dot{V}_{\text{thermal storage circuit}}$ and $\dot{V}_{\text{emulator}}$.

Similarly, the effectiveness ($\xi_T$) of the coil-in-tube heat exchanger within the hot water tank is given by:

$$\xi_T = \frac{\dot{V}_{\text{heat pump circuit}}}{\dot{V}_{\text{min}}} \left( \frac{T_{\text{coil, in}} - T_{\text{coil, out}}}{T_{\text{tank, in}} - T_{\text{bulk}}} \right)$$

where $\dot{V}$ is the volumetric flow rate and $T$ is the temperature. $\dot{V}_{\text{min}}$ is the minimum of $\dot{V}_{\text{heat pump circuit}}$ and $\dot{V}_{\text{thermal storage circuit}}$.

“Inlet” and “outlet” (or “in” and “out”) are defined strictly in terms of liquid flows relative to the component under discussion and not in terms of temperature or other components in a given loop. Hence, a component “inlet” is a port in which the flow originates upstream of the port external to the component, and a component “outlet” is a port in which the flow originates upstream of the port internal to the component. Thus for example, consider a loop in which a geothermal well is connected to a heat pump. The well outlet is connected to the heat pump inlet and the heat pump outlet is connected to the well inlet under all conditions regardless of whether the heat pump inlet temperature is greater than the heat pump outlet temperature (during heating mode) or vice versa (during cooling mode).

4.1 HEATING SEASON

The heating season performance is described in two subsections. The first shows the behavior of the plant under long term operating conditions. In this mode, the heat pump was controlled by a standard fixed state (that is, “on” or “off”) thermostat in response to the conditioned space (HESS laboratory) interior temperature. As the space lost heat to the exterior, the interior temperature declined until it reached the set point temperature minus half the “dead-band” temperature at which point the heat pump was turned on. The interior temperature increased as...
heat was added until it reached the set point temperature plus half the dead-band temperature when the heat pump was turned off.

The second subsection describes the short term performance of the plant under various operating conditions. In this mode, the heat pump was not under the control of the thermostat but operated continuously while various plant parameters (principally flow rates) were changed.

4.1.1 Continuous operation

Figure 4.1 parts (a) and (b) shows the long term performance of the heat pump when the geothermal wells were the source of the heat. Figure 4.1(a) essentially defines the system operating conditions during the six day test period. The upper two panels describe the inlet and outlet temperatures of the two geothermal wells. Note that each temperature profile consists of a lighter shaded band about an average. The lighter shaded band depicts the individual heating cycles that the heat pump experienced over time as the thermostat cycles through its on/off sequence. The upper panels show that during this test in which the wells functioned as thermal sources, Well 1 delivered heat at a temperature about 1°F higher than Well 2 for the same inlet temperature conditions. Also of note is that the outlet temperatures of both wells remained essentially constant over the test period showing that under transient (that is, thermostat cycling) conditions for a prolonged period, the wells could accommodate the thermal loading of the space without becoming unstable. In such a situation, the outlet temperatures would gradually decline over time.

The bottom panel of Figure 4.1(a) shows that the exterior temperature during the test period ranged between fairly cold (0°F) to mild (~32°F). Note the correlation between the average inlet and outlet temperatures of the wells with the dips in exterior temperature at days 0.75, 3.75 and 4.75. Of particular note in panel 3 is the rather poor thermostat calibration. With an indicated set point of 68°F (solid black line) the actual set point was around 69.5°F with a dead band temperature of about 1.5°F. This is important to bear in mind in the subsequent discussion of a transient cycle.

Figure 4.1(b) reveals the response of the heat pump itself to the prevailing liquid and air side temperatures. The top panel shows the relationship between the heat source liquid inlet and the room air supply temperatures. The liquid supply temperature experienced a range of about 46 to 47.5 °F during a transient cycle, while the air output temperature varied over a much larger 12°F range.
Thus with an average inlet temperature of about 48 °F and an average air outlet temperature of approximately 88 °F, the reverse Carnot coefficient of performance \( \text{COP}_{\text{Car}} \) (equation 4.3) in heat pumping mode was 13.7.
Figure 4.1(b) Long term heating performance with the geothermal wells as the heat source

This may be compared with the bottom panel in which the measured COP varies on average between 2 and 4. Thus at best, the heat pump had a reverse Carnot efficiency ratio or “figure of merit” (COP / COP_{Car}, Equations 4.1 and 4.2) of 29%. The figure of merit shows that, relatively speaking, the Trane GEV 018 heat pump used was rather inefficient on its own (or homogeneous) terms under transient conditions. Note that this evaluation is not relevant to
comparing heat pumps with other heating equipment such as natural gas furnaces (heterogeneous terms) but simply gives an indication for the scope of possible improvement in heat pump technology (an ideal heat pump would have a figure of merit of 100%). However, this observation has to be qualified by the liquid flow rate that is shown in the bottom panel of Figure 4.2(b) to be fairly steady at around 2.9 gpm (ignoring the pump start-up spike). The manufacturer rates the heat pump at 4.2 gpm which is greater than the long-term average and this could increase the measured COP and hence the order of merit, slightly. However, counteracting this argument are the data of Figures 4.6 to 4.8 that show under continuous operating conditions, the measured COP is not significantly sensitive to flow over a 2.9 to 4.2 gpm range.

The center panel of Figure 4.2(b) indicates a supply air enthalpy gain (Equation 4.2) with a maximum value of about 5200W at the end of each transient cycle. This is a reflection of the 93 °F air outlet temperature under essentially constant volumetric air flow rate conditions maintained by a fixed speed air circulation fan.

Figure 4.2 describes two consecutive cycles during the long term heating performance test shown in Figure 4.1. The panels in Figures 4.2(a) and (b) correspond to those in Figures 4.1(a) and (b). Looking at the bottom panel of Figure 4.2(b), the cycle dynamics are best shown by the flow rate temporal profile. Thus the two cycles depicted reflect thermostat “on” periods of about 6 minutes and 7 minutes respectively while the intervening quiescent (or thermostat “off”) period lasts about 23 minutes for an average cycle period of 29.5 minutes. Thus the heat pump was active about 22% of the time. Clearly this ratio is a function of the thermal integrity (or level of energy conservation) of the building envelope and, in fact, is one of the techniques that can be used to assess the energy performance of a building envelope. The lower the active time ratio, the higher the thermal integrity and vice versa. However, this ratio also has a significant impact on the actual heat pump performance under dynamic conditions when compared with that measured under typical steady-state rating test conditions. This theme will be amplified as the discussion of the results continues.

Returning to Figure 4.2(a), the top and middle panels show the transient temperature response of the wells. The well1 outlet temperature shows a stable, well-damped reaction to the inlet temperature transient in that the outlet temperature linearly declines to its value prior to the commencement of a cycle. This confirms the inference of well stability made previously, since in an unstable condition, the outlet temperature would gradually increase at the start of each cycle. Well 2 also shows stable behavior, except the damping is greater with the inlet temperature reaching is pre-cycle start value more quickly. Also the, transient temperature rise experienced by Well 2 was about 1.4 °F compared with 1 °F for Well 1, a subtle indication of the mismatch of the heat transfer characteristics of the two wells that nominally were identical. As shown in the bottom panel, the exterior temperature remained fairly constant while the thermostat demonstrated an actual setpoint 69 °F with a deadband of 2.2°F, fairly large in control terms.
The impact of transient operation in heating mode can be seen in Figure 4.2(b). The top panel shows essentially that the heat pump only started approaching the steady-state outlet temperature at the end of a cycle and spent most of the cycle essentially "warming up". This effect is amplified in the center panel in which the air enthalpy gain consequently also reached the steady-state condition at the end of the cycle. The COP in the bottom panel therefore shows the same highly transient behavior with the consequence that in fact, the heat pump essentially never operated at a steady-state.

Figure 4.2(a) two transient cycles during the long term heating performance test (well heat source)
In this case, the COP ranged from less than 3 to 3.9 compared with the manufacturer's rating of 3.6 at the same liquid inlet temperature and flow conditions. Note that the decline in the COP during the last half minute of the second cycle was a systemic effect in which data continued to be recorded after the thermostat turned the compressor off.

Figure 4.2(b) Two transient cycles during the long term heating performance test (well heat source)
These data raise interesting questions about the optimality of conventional thermostat control of heat pumps with high thermal integrity building envelopes. This suggests that, in these circumstances, either the control scheme or the design of conventional vapor compression cycle...
heat pumps (or both) is not ideal in terms of optimizing the COP in long-term operating mode with transient cycling that is the normal heating season operating mode. When operating off the thermal storage system, the long term performance of the heat pump is shown in Figures 4.3. Figure 4.3(a) depicts the operating boundary conditions while the heat pump performance under these conditions is given in Figure 4.3(b).

The top panel of Figure 4.3(a) shows the temperatures of the thermal storage system during the 7 day test. For the first 1.5 days, heat was added to the thermal storage system from an emulator, thereafter, heat was withdrawn from storage for the remaining 5.5 days. During the entire test, liquid was circulated through all five tanks in the thermal storage system (four 100 gal. and one 80 gal., see sections 3.1.2(e) and (f)) at a constant rate of about 8.7 gpm so explaining the closely correlated tank temperatures. The heat removal commenced at 135°F and continued until the stored temperature reached about 54 °F producing a dynamic thermal time constant of about 0.6°F/hour. The bottom panel of Figure 4.3(a) indicates that the exterior temperature averaged about 0°F for most of the test except for a warm spell around freezing from day 0.75 through day 2. There were also episodes of below zero temperatures peaking at days 3.5, 5.25 and 6.5. Hence, overall, this was a fairly severe test of operating the heat pump using stored heat as the thermal source.

As the thermal storage system was located within the laboratory and thus lost heat to the laboratory air, during the warm period from day 0.75 through day 2.5, there was sufficient heat loss to maintain the interior temperature above the set point, so requiring no space conditioning heat from the heat pump. This is shown by the absence of cycling of the interior temperature during this period as well as by the gaps in the performance data of Figure 4.3(b). This phenomenology is not a detriment to the overall system performance but demonstrates the value of including the thermal storage within the heated envelope allowing any leaked heat to be passively captured thus reducing the heating demand from the heat pump. At day 2.5, the interior temperature decreased to the point where the heat pump began cycling once more. As in Figure 4.1(a), the thermostat did not cycle about the nominal setpoint temperature of 68 °F, but in this case, yielded an average temperature of about 70 °F with a deadband of about 1.5 °F.

Note that Figure 4.3(b) reveals the same transient operating phenomenology discussed above for Figure 4.1. Thus each vertical line represents a discrete transient operating cycle. The top panel of Figure 4.3(b) shows a rather interesting phenomenon that is a hallmark of vapor compression cycle heat pumps. During the heat storage period, after about half a day, the liquid inlet temperature began to exceed the air output temperature. This also occurred during the heat removal period, from day 2.5 through day 3.25. These conditions arise when the phase change (liquid to gas) heat absorption capacity of the working fluid at maximum pressure in the evaporator is reaches its upper limit. Thus under these conditions (inlet liquid temperature greater than air outlet temperature) the heat pump should not be operated at all, rather, there should be provision for direct liquid to air heat exchange. However, in mitigation, from day 0.5 though day 3.25, the GEV-018 heat pump was being operated at liquid inlet
temperatures as high as 120°F, well outside the nominal maximum design inlet temperature of 86 °F.

The center panel of Figure 4.3(b) shows the significantly higher air enthalpy gains of up to 7300 W than the 5300W maximum realized with the thermal wells as the heat source (Figure 4.1(b)). These larger enthalpy gains are reflected in the improved COP in the bottom panel of Figure 4.3(b) so that at the end of each transient cycle during the period from day 0.75 to day 3.5 (inlet liquid temperature higher than outlet air temperature), the COP reached values as high as 6.5. As before, the liquid flow rate was about 2.8 - 2.9 gpm after the start-up transient of each cycle.

Figure 4.3(b) Long term heating performance with the thermal storage system as the heat source
A single cycle during day 3.5 is extracted from Figure 4.2 and shown in isolation in Figure 4.4. The top panel shows the air outlet temperature rising logarithmically from 83 °F to 104 °F with 14 °F of the 21 °F total rise occurring during the first minute as the liquid inlet temperature increased to its maximum value of 95 °F before declining. The entire “on” cycle lasted less than 6.25 minutes, again showing the rapid increase of interior temperature as a result of the high thermal envelope integrity.

Figure 4.4 A transient cycle during the long term heating performance test (thermal storage heat source)
The center panel reveals that the air also gained enthalpy logarithmically as dictated by the air outlet temperature while the interior temperature increased more or less linearly. The bottom panel indicates a constant liquid flow rate of about 2.8 gpm while the COP increased from 1 to about 4.25 during the first 2 minutes, remained at about 4.5 through minute 5.5 before increasing to greater than 5 during the final half minute of the cycle. This is more or less the same pattern as shown by Figure 4.2(b) for the well heat source except for the very last segment in which the COP declined in that figure owing to systemic effects (as explained previously). The only major difference is the higher maximum COP reached in Figure 4.4 (>5 versus <4), the cycle times in both cases were about the same (6 to 7 mins.) The discontinuous rise in the COP during the last half-minute indicates that the heat pump did not reach steady-state during each 6-minute cycle so that the maximum achievable COP is greater than that realized at elevated liquid inlet temperatures as steady-state is approached (as shown is the subsequent discussion). This argues for altering the heat pump control system to achieve longer operating cycles (by derating the compressor speed) allowing the unit to operate at higher COP’s for longer periods so yielding higher electrical energy consumption efficiency.

4.1.2 Transient testing

Figures 4.5 through 4.7 describe three repeated experiments (referred to as Experiments 1, 2 and 3 respectively) during which the heat pump performance was measured under continuous operating conditions. In this mode, the heat pump was not controlled by the thermostat but operated continuously, or more specifically, the compressor, air circulation blower and liquid circulation pump remained on during the entire test. Experiment 1 (Figure 4.5) reports the performance at a constant liquid flow rate while Experiments 2 and 3 show the results for a varying liquid flow rate. In all three experiments, thermal energy was withdrawn from the thermal storage exclusively. The experiment was repeated three times as the heating performance using stored low-grade heat is arguably the most salient result of the project and thus it is important to define the repeatability error.

The conditions for Experiment 1 were different from those of Experiments 2 and 3 as follows:

- Heat was withdrawn from the 80 gal. hot water tank only (high temperature section of thermal storage) – the 400 gal. low temperature section of the thermal storage system was not in the circuit.
- The liquid flow rate was constant at 4 – 4.2 gpm.

Experiment 1 was 260 minutes (4.3 hours) in duration and was one of the first tests conducted after the instrumentation system was commissioned. During the initial 20 minute transient start-up period, the liquid inlet temperature declined from about 105 °F to 96 °F. There was an interruption at minute 41 during which the compressor stopped owing to the thermostat setpoint temperature being set too low for continuous operation. After restarting, there was a second transient lasting through minute 60 after which the inlet temperature declined gradually from 96 °F to 91 °F. Thereafter, the inlet temperature declined more steeply and uniformly through the end of the experiment reaching a minimum temperature of less than 50 °F at the end of the test. This inlet temperature response was a direct consequence of the limited (80 gals) amount of
thermal storage available in the hot water tank. The liquid outlet temperature response was similar to that of the liquid inlet temperature with about a 10 °F offset during the first 100 minutes after which the offset gradually decreased. On the air side, the air inlet temperature remained fairly constant at about 80 °F over the duration of the test after the start-up transient, while the air outlet temperature remained steady at 117 °F through minute 120 before declining gradually in response to the decreasing liquid inlet temperature.

Figure 4.5 Short term performance of heat pump under continuous operating conditions with the thermal storage system as the heat source—Experiment 1
In examining the performance profiles in the lower panel of Figure 4.5, it is important to note that the COP (Equation 4.1) and heat ratio (Equation 4.4) profiles each are shown as two overlapping curves.

Figure 4.6 Short term performance of heat pump under continuous operating conditions with the thermal storage system as the heat source – Experiment 2
The dotted fuzzy profile represents the actual unsmoothed data, while the solid discrete profile shows the smoothed mean using a negative exponential least squares fit.

In response to the inlet temperature profile, the mean COP varied from a high of about 6.1 during the startup transient (when the inlet liquid temperature was in excess of 100 °F) to a maximum steady-state value of 5.95 at 102 minutes before declining to a low of about 4.8 at 240 minutes as the inlet temperature decreased. This demonstrates how strongly the COP is dependent on the inlet temperature under constant liquid flow rate conditions. These values of COP represent the highest values measured during the experimental test program. In contrast, the heat ratio shows very little dependence on inlet temperature remaining within the range of 1.8 to 2 after the startup and compressor stoppage transients were damped out at 72 minutes.

Experiments 2 and 3 showed much steadier continuous performance when the full thermal storage capacity (480 gals) was used. In essence, there was sufficient stored heat available so as not to deplete the thermal storage to any significant degree during the experiments.

In Experiment 2 (Figure 4.6) that was 93 minutes in duration, the liquid flow rate was decremented steadily from 5 to 0.8 gpm. The liquid input temperature from the thermal storage system remained within the range 100-110 °F, declining from 110 °F to 100 °F during the initial 10 minute start-up transient, then remaining essentially constant through minute 60 before gradually increasing back to 108 °F during the last 33 minutes of the experiment. In response, the inlet and outlet air temperatures also remained fairly constant after the 10 minute start-up transient at about 80 and 118 °F respectively. In contrast, the outlet liquid temperature, declined fairly steadily after 40 minutes from 90 °F to 66 °F. Thus the liquid outlet temperature was most responsive to the flow rate as a consequence of the dynamics of a vapor compression cycle. The start of the decline in the outlet temperature corresponded with a flow rate of about 3.3 gpm. However, examination of the COP profile shows that after the initial 10 minute transient, the mean value remained almost constant at about 5.5 through minute 70 (1.8 gpm) before declining to a value of 5 at the end of the test. This indicates that from a COP perspective, the operating working liquid flow rate range is between 1.8 and 5 gpm. The heat ratio remained fairly constant after 10 minutes at a value of 1.5 to 1.6 gently rising to a peak of 2 after 70 minutes, mirroring the COP profile.

A repetition of Experiment 2 is shown in Figure 4.7 (Experiment 3). In this case, the experiment lasted about 125 minutes and repeated the pattern of Experiment 2 with some minor differences. Notably, the thermal transients on the liquid side show increasing values during the 10 minute start-up transient compared with decreasing values in Experiment 2. Further, the liquid inlet temperature was a little colder on average in Experiment 3 (98 °F compared with 100 °F). This mostly explains the lower average COP of about 5.3 in Experiment 3 compared with 5.5 for Experiment 2. The heat ratio response in Experiment 3 is similar to that of Experiment 2.
Taken together, the three continuous heating mode experiments consistently showed that with a steady liquid supply temperature of 100 °F and a liquid flow rate in excess of 1.8 gpm, COP's of 5.3 or greater were maintained for significant periods.

Figure 4.7 Short term performance of heat pump under continuous operating conditions with the thermal storage system as the heat source – Experiment 3
For comparison with Figures 4.6 and 4.7 (Figure 4.5 is excluded because of the non-constancy of the liquid input temperature), continuous operation in heating mode with the thermal energy being withdrawn from the wells only is described in Figure 4.8. The topology of Figure 4.8 is very similar to that of Figures 4.6 and 4.7 with two principle differences:
  
- The liquid input temperature had a range of 48 to 50 °F compared with 100 to 110 °F in Figure 4.6 and 94 to 104 °F in Figure 4.7.
- After the warm-up transient, the COP had a range of 4.2 to 5.2 compared with 5 to 5.5 in Figures 4.6 and 4.7.

Figure 4.8 Short term performance of heat pump under continuous operating conditions with the geothermal wells as the heat source
The performance of the heat pump as reflected by the heating COP with the wells and thermal storage system as heat sources is compared as a function of flow rate in Figure 4.9. The heavy lines represent a negative exponential weighted least squares fit of the data. Thus the maximum thermal storage/wells COP ratio occurs at a flow of 1.8 gpm and has a value of 1.24, while the minimum ratio of about 1.05 occurs at 4.8 gpm. The ratio generally increases as the flow rate decreases.

By way of comparison, the manufacturer's COP data for the average temperature of the wells and the maximum source temperature reported (86 °F) is shown in Figure 4.10, also as a function of flow rate. These curves yield a high/low temperature COP ratio that is essentially constant in the range 1.21 to 1.25. Hence the experimental ratiometric data bound the manufacturer's data, and both sets of profiles show COP increasing with flow rate. However, while the manufacturer's COP ratio tends to remain essentially constant, the experimental ratio decreases markedly with increasing flow rate.

It needs to be noted that in absolute terms, the manufacturer's COP data are less than those measured experimentally. For example, consider that at 3 gpm and 48.5 °F (blue profiles in Figures 4.9 and 4.10), the manufacturer to experimental COP ratio is 0.74 (3.6/4.85). There is
no obvious explanation for this discrepancy in the steady-state data given that the long-term experimental COP’s under intermittent operating conditions (for example, Figure 4.3(b)) appear to bound the manufacturer’s data quite well, and, the experimental and manufacturer’s steady state cooling data are in substantial agreement (Figure 4.17). Thus additional research is necessary to investigate this phenomenon in terms of some of the more likely explanations such as differences in heating mode testing protocol (under both steady-state and transient conditions) and other factors. Hence the performance comparisons can only be done using the ratiometric approach of Figures 4.9 and 4.10.

In this context, there are a number of reasons that likely explain the differences between the experimental and manufacturer’s ratiometric performance data as follows:

- Different testing boundary conditions. Most notably, the manufacturer’s data is generated for a constant air inlet temperature of 68 °F, whereas the experimental inlet air temperatures varied between 72 and 82 °F for the thermal storage heat source and between 68 and 78 °F for the wells heat source.
- Different air flow rates. The manufacturer’s performance data is rated at 570 cfm while the experimental air flow rate averaged about 690 – 700 cfm.
• Different refrigerant operating pressure.
• Different compression ratios.
• Different heat exchanger characteristics.

Thus it may be noted that the experimental and manufacturer’s data concur that the maximum COP increase attainable with the GEV-018 heat pump tested as the source temperature is increased appears to be about 25%, although the manufacturer’s maximum ratio occurs for double the flow rate of the experimental maximum (3.3 to 3.7 gpm). Hence, the question arises as to whether this takes advantage of the potential of using higher temperature source heat. Further insight to this question can be gained by examining the COPCar’s (Equation 4.3) at the test conditions. Thus at a 3 gpm flow rate with the thermal storage source, Figures 4.6 and 4.7 yield COPCar’s of 32.1 and 28.9 respectively, while with the well source, the COPCar is reduced to 10.8 – roughly a three-fold theoretical increase. Thus clearly, the potential performance benefits possible from an increased source temperature were not realized with the vapor compression cycle heat pump used. This suggests an area for additional research to determine whether it is possible to modify a vapor compression heat pump to improve its performance at elevated source temperatures and whether alternate forms of heat pump (such as a Stirling cycle heat pump) would be more appropriate in a hybrid system application.

In addition to the heat pump performance, several other aspects of the plant were tested including the performance dynamics of the geothermal wells and the heat transfer effectiveness of the flat plate heat exchanger coupling the emulator to the thermal storage circuit.

The characteristics of the wells when acting as thermal sources providing heat to the heat pump under steady state conditions during the heating season are shown in Figure 4.11. There was a 10 minute startup transient (evident in the top and center panel profiles) during which the well outlet temperatures spiked owing to the inlet fluid displacing the volume in the U-shape well tubing that has reached equilibrium with the surrounding ground temperature. Clearly, the thermal characteristics of the wells were different in terms of flow resistance (hydraulic pressure drop) since the flow rate through Well 2 was consistently less than that through Well 1 by about 0.4 gpm (had the wells been matched, their flow rates would have been equal).

These different flows contributed to the slightly different temperature draw down characteristics shown in the center panel in which after the 10 minute startup transient, the outlet temperature of Well 2 (lower flow rate) increased slightly during the test while that of Well 1 (higher flow rate) remained almost constant. However, the maximum outlet temperature difference of about 1 °F at the end of the test is not significant. The center panel (as well as the dashed line in the bottom panel) also shows that the heat pump outlet temperature declined fairly steadily during the 80 minute test. However, over the range of flow rates measured, the wells had sufficient heat transfer capacity to absorb this declining inlet temperature without a corresponding decline in the outlet temperature.
The top panel of Figure 4.11 reveals how the flow rate and temperatures combined to yield slightly higher ground heat transfers for Well 1 than for Well 2. Evidently, the increased flow rate through Well 1 more than offset the increased outlet/inlet temperature difference of Well 2.

Figure 4.11 Characteristics of wells when acting thermal sources (heating mode)
The performance of the flat plate heat exchanger coupling the low-grade heat source to the thermal storage loop is shown in Figure 4.12. During this test as shown in the bottom panel, the emulator was tested at flow rates of 3, 4 and 5 gpm while at each of these flow rates, the thermal storage circuit flow rate spanned a range of 1.5 to 6 gpm for all 3 tests. During the second test (emulator flow rate of 4 gpm), there was a flow reversal at about 70 minutes which produced a transient in the thermal effectiveness yielding a fictitious heat exchanger effectiveness less than unity. The circuit flow rate range was increased during the third test (emulator flow of 3 gpm) by decreasing the minimum flow to about 0.7 gpm.

![HEAT EXCHANGER PERFORMANCE](image)

**HEAT EXCHANGER PERFORMANCE**

Inlet and Outlet Temperatures

Flows and Effectiveness

Figure 4.12 Emulator circuit heat exchanger performances
The top panel of Figure 4.12 describes the temperature profiles measured during the test. The thermal storage circuit inlet temperature remained more or less constant during the 140 minute test within a range of about $112 \pm 0.5 \, ^\circ F$. At each emulator flow rate, the emulator side inlet and outlet temperatures increased with decreasing circuit flow rate as a result of the emulator input power being held constant at about 2640 W (emulator outlet temperature increased as the heat removed in the heat exchanger decreased in order to keep the heat transfer within emulator approximately constant). The thermal storage circuit side outlet temperature thus also increased with decreasing circuit flow rate. These effects combined to yield the exchanger effectiveness (Equation 4.6) profile shown in the bottom panel.

At the maximum circuit flow rate of 6 gpm, the heat exchanger minimum effectiveness was 0.64 at an emulator flow rate of 3 gpm. At the maximum loop flow rate of 6 gpm, the minimum effectiveness increased to 0.67 and 0.77 at emulator flow rates of 4 and 5 gpm respectively. At all emulator flow rates, the effectiveness increased with decreasing circuit flow reaching unity at circuit flow rates of 2 gpm or less. Thus, the sizing and performance of the heat exchanger were adequate to enable control of the heat exchange process to achieve minimized entropy heat transfer as shown by the convergence of the emulator side input and circuit side outlet temperatures in the upper panel of Figure 4.12.

4.2 COOLING SEASON

As with the heating season, the performance is shown in two sections. The first section describes the long term cooling performance with the heat pump being controlled by the thermostat while the second section highlights the short term performance under various operating conditions.

4.2.1 Continuous operation

Figure 4.13 describes the performance of the heat pump and wells over a 27 day period during July in two parts. The first part (Figure 4.13(a)) gives the temperature boundary conditions and the well temperatures while the second part (Figure 4.13b)) shows the heat pump performance. The format of the graphs is the same as that of Figure 4.1 in which each vertical line represents a separate cooling cycle.

The inlet and outlet temperatures of Wells 1 and 2 during the test are shown in the top two panels of Figure 4.13(a). In these panels, the heavy smoothed lines represent the average inlet and outlet temperatures while the dashed fine lines show the cycle data. In cooling mode, the wells appear to be fairly well matched with the outlet temperature of both wells (excluding the transients) being approximately 52 °F. Of interest is the 180° phase difference between the inlet and outlet temperature transients for both wells in which a trough in the inlet temperature corresponds to a peak in the outlet temperature compared with the inverse case in which a peak in the inlet temperature does not produce a corresponding trough in the outlet temperature. The
peaks in the well inlet temperatures are in phase with peaks in the ambient temperature (bottom panel) that in turn produced peaks in the laboratory interior temperature. The troughs in the inlet well temperatures occurred when the heat pump was quiescent (gaps in between the vertical blocks in the interior temperature profile corresponding with lower ambient temperatures) allowing the ground immediately around well sleeves to recover to an equilibrium temperature of 56 °F or more. This indicates that the well fiduciary heat exchange temperature was 52 °F, that is, the temperature at which sustained heat rejection could be maintained.

The control response in the bottom panel is the inverse of that shown in heating mode in Figure 4.1(a) in which the interior temperatures were maintained at levels below that of the set point with a 5 °F dead band skew centered at about 68.2 °F. So, again while this shows a poorly calibrated thermostat, it also points to one of the key issues in optimal heat pump control which is that larger dead bands are necessary to produce longer operating runs to achieve higher efficiencies at the expense of an even interior temperature and lower overall energy consumption (short runs at low COP use less energy than long runs at higher COP).

This can be seen clearly in the bottom panel of Figure 4.13(b) in which the cooling COP\textsuperscript{36} (Equation 4.5) varied from 1 to a maximum of about 4.5 during day 17. Thus under these conditions, the full benefits of ground sourced cooling cannot be realized\textsuperscript{37} and suggest that research into correcting this deficiency would be worthwhile.

\textsuperscript{36} It has become commonplace to use the dimensional EER (Energy Efficiency Ratio = COP x 3.413 Btu/Watthour) as a substitute for the non-dimensional cooling COP. While this may have marketing value by reporting the cooling performance with inflated numbers, in the authors’ opinion, it has no technical merit and is not used here.

\textsuperscript{37} Exactly the same issues arise with conventional air-sourced heat pumps (usually called air-conditioners), except that the COP of such units is lower and so the energy advantages of long runs at higher COP likely are not as economically compelling.
Figure 4.13(a) Long term cooling performance with the geothermal wells as the heat sink
The top panel of Figure 4.13(b) shows consistently overlapping heat pump inlet and outlet temperature ranges compared with Figure 4.1(b) for the heating season in which the air outlet temperature was always greater than the liquid inlet temperature for the entire duration of a cycle. The maximum enthalpy loss over a single cycle was about 5000 W, 200W or so less than the enthalpy gain during the heating season (Figure 4.1(b)).

Figure 4.13(b) Long term cooling performance with the geothermal wells as the heat sink

This represents a 4% discrepancy in heat exchanger performance that can be accounted for by enthalpy lost to condensation drainage during the cooling season.
Two transient heat pump operating cycles extracted from day 17 of Figure 4.13 are shown in Figures 4.14.

**LONG TERM COOLING FROM WELLS: TWO CYCLES - PART A**

- **Geothermal Well 1 Temperatures**
- **Geothermal Well 2 Temperatures**
- **Temperature Boundary Conditions**

Figure 4.14(a) two transient cycles during the long term cooling performance test (well heat sink)
Figure 4.14(b) reveals that each cycle had duration of about 5 minutes with an interval of 44 minutes between starts or an 11% duty cycle. The top two panels in Figure 4.14(a) confirm the heat transfer characteristics of the wells in cooling mode were matched. During the actual cycle starting at the discontinuity, the well inlet temperature rose sharply to 66 °F by the end of the cycle. During the same interval the well outlet temperature decreased by less than a degree F.

Figure 4.14(b) two transient cycles during the long term cooling performance test (well heat sink)
Compare this to the heat pump liquid inlet temperature range (top panel of Figure 4.14(b)) that starts at 57.5 °F and ends at 52.3 °F. These two observations show that during the operating cycle, liquid from the wells never reaches the heat pump and liquid from the heat pump never reaches the wells so the heat sink is provided entirely by the liquid in the pipes linking the heat pump to the wells. In other words, during long term cooling operation, the wells in fact do not contribute directly to the heat exchange and are more or less redundant. This points to another often overlooked aspect of geothermal design, namely that “dead” or non-active heat exchange volume in the liquid system can impose a large performance penalty on heat pump performance especially at low duty cycles.

This is evident in the bottom panel of Figure 4.14(b) in which the COP rises steeply from 3.5 to a quasi-steady state 4.3 as the liquid inlet temperature decreases from a shallow ground warmed 57.5 °F to the approximate well outlet of 52 °F. This again provides fodder for improving the design and control of heat pumps in cooling mode to realize a greater level of long-term operating efficiency. Note that in the second panel of Figure 4.14(b) the large increases in the air enthalpy loss occur after compressor shutdown while the blower is still running. The heat exchange enthalpy loss decreases from -4125 W to -4800 W during the duration of the cycle.

4.2.2 Transient testing

Figure 4.15 describes the performance of the heat pump when operating continuously during the cooling season. The bottom panel shows a clear correlation between liquid flow rate and COP with the COP decreasing with flow rate. This is not an inlet temperature (or well heat exchange) effect since the top panels shows that after the start-up transient (5 minutes) as discussed above, the liquid inlet temperature that reflects the well outlet temperature remained fairly constant. The steadily increasing liquid outlet temperature was required to keep the net heat lost to the liquid constant as required by the fairly constant, liquid flow rate independent, air side temperature difference. Thus the increasing liquid outlet temperature necessary to compensate for the decreasing liquid flow rate requires additional compressor work that in turn decreases the COP. However, the pumping power increases with liquid flow rate so that there is an inverse relationship between pumping and compressor power – the compressor power increases as the pumping power decreases. The maximum cooling COP measured at about 4.8 gpm was about 6 compared with 5 in heating mode (Figure 4.8).

A comparison of the manufacturer’s and experimental cooling COP at the average experimental heat pump inlet temperature of 55 °F is given in Figure 4.17. Over the manufacturer’s test flow rate range of 2.7 to 5.0 gpm, the manufacturer’s and smoothed experimental COP performance profiles agree within the experimental variation. This is quite unlike the COP discrepancy encountered in heating mode (discussion of Figure 4.8) and highlights the need for additional research to resolve the heating COP disagreement. The agreement in the cooling mode data make an argument that the discrepancy in the heating mode data is caused by systemic experimental errors highly improbable.
As with the heating season tests (Figures 4.5 through 4.7) the cooling mode heat ratio (equation 4.5) remains fairly constant after the start up transient within a range of 0.75 to 1.2. This is lower than the range of 1.4 to 2 measured during the heating season. It is interesting to speculate whether the cooling heat ratio could be increased to the heating heat ratio if the inlet liquid temperature were decreased below the well outlet temperatures (the inverse of heating season operation from higher temperature thermal storage).

Figure 4.15 Short term performance of heat pump under continuous operating conditions
Figure 4.16 Comparison of experimental and manufacturer's heat pump cooling mode performance

The characteristics of the thermal wells when acting as sinks are shown in Figure 4.17. With reference to the bottom panel of the figure, there was a significant transient at 188 minutes after the combined well flow reached 4 gpm. This transient represented the clearance of a sand blockage in Well 1 since before the transient, the Well 2 flow was always greater than the Well 1 flow but after the blockage cleared the flows were more uniform with the Well 1 flow exceeding the Well 2 flow. This was a second instance of a sand blockage that in this case, did no damage. In the previous instance when a blockage was cleared by a 5 gpm flow through a single well, the sand from the blockage reached the positive displacement vane pump and destroyed the rotor.
Thereafter, the experimental apparatus was modified to include a basket filter to trap and remove any additional liquid loop particulates. This experience highlights the importance of flushing the liquid loop piping at high flow rates before placing the system in service, an exercise that was not performed during the heat pump installation.

The effect of clearing the blockage on the heat transfer performance of the wells can be seen in the middle panel of Figure 4.17 in which before the clearance of the blockage, the inlet/outlet
temperature difference for Well 2 was less than that for Well 1, while after the clearance, the
temperature differences for both wells were approximately equal. This effect is carried over to
the top panel in which the ground heat transfer rates reversed in magnitude after the blockage
clearance with the difference after the clearance a result of the 0.6 gpm difference in the well
flow rates. Note that the thermal source experiment reported in Figure 4.11 was carried out
after the experiment reported in Figure 4.17, hence only the well heat transfer phenomenology
of Figure 4.17 after the blockage clearance may be compared with that shown in Figure 4.11 in
which the wells operated as thermal sources. On this basis, the heat transfer performance of
the wells was fairly symmetric between thermal sink and source operation.

The isolated thermal sink characteristics of Wells 1 and 2 are shown in Figures 4.18 and 4.19
respectively. In these experiments, only one well was in the circuit, the other being dormant.
Figure 4.18 shows that the damped sinusoidal transient in the ground heat transfer during the
first 14 minutes of the test was caused by the transient in the well manifold supply temperature
measured at the outlet of the variable speed pump (oscillations through minute 9) with the
peaks in the heat transfer profile thereafter being caused by the step changes in the flow rate
profile. The final transient in the heat transfer profile at minute 27.8 resulted from the step
decrease in the flow rate from 2.9 to 2.3 gpm, but after the transient was damped, the heat
transfer at 2.3 gpm was almost the same as that at 2.9 gpm. Note that during the first minute of
operation, the well outlet temperature exceeded the inlet temperature producing a net positive
heat transfer in the well, another manifestation of a “dead” volume effect, in this case no doubt a
result of warmed fluid in the upper length of the well outlet tubing.

After about 10 minutes of operation, Well 1 yielded a very slowly rising outlet temperature
through the end of the test (2 °F over 23 minutes or 0.09 °F/min) in response to a more quickly
rising inlet temperature (11 °F over 23 minutes, 0.5 °F/min). For Well 2, Figure 4.19 shows that
after the initial transient at 10 minutes, the outlet temperature rise was 0.14 °F/min for an inlet
temperature rise again of 0.5 °F/min. Hence both wells exhibited essentially the same heat
transfer characteristics even when subjected to different flow rate time profiles. Well 2 was
subjected to a different flow rate time profile than used for Well 1 but with exactly the same
transient performance pattern in terms of inlet temperature and flow rate change perturbations.
This well also exhibited a positive net heat transfer during the first 1.5 minutes of operation.
Figure 4.18 Isolated characteristics of Well 1 when acting as a thermal sink (cooling mode – Well 1 only in the circuit)
Figure 4.19 Isolated characteristics of Well 2 when acting as a thermal sink (cooling mode – Well 2 only in the circuit)

Figure 4.20 describes the performance of thermal storage system in terms of recovering the heat rejected to the liquid by the heat pump in cooling mode. The bottom panel of Figure 4.19 shows that during the test the storage or bulk fluid flow rate was constant at 11.5 gpm while the heat pump circuit flow rate profile was symmetric in that the heat pump loop flow was increased
from 1 to 5 gpm during the first half of the test and then reduced back to 1 gpm during the second half. This produced a symmetric heat pump liquid outlet temperature profile (top panel) with high outlet temperatures at low flow rates and vice versa. However, the heat pump liquid inlet temperature remained relatively constant over the test sequence within a range of about 3 °F.

Figure 4.20 Cooling heat pump mode heat recovery
The center panel of Figure 4.20 shows the heat exchange temperatures in the hot water tank. The heat pump outlet fluid flowed through a 200 ft long coil immersed in the bulk fluid within the hot water tank so the coil temperature difference is an indication of how much heat was transferred to the bulk fluid during the test. This temperature difference has a more or less parabolic trend on which is superimposed an oscillation produced by the heat pump compressor cycling. Note that the coil temperature difference is a very strong function of heat pump circuit flow rapidly declining to zero with increasing flow rate. As a consequence, the bulk hot water tank temperature increased significantly for flows of 3 gpm or less with a much lower increase at higher flows. These effects combined to yield the heat exchange effectiveness (Equation 4.7) characteristic delineated by the black dashed profile in the bottom panel. For the most part during the higher heat transfer regions at flows less than 3 gpm, the effectiveness was less than 0.7 with essentially no effectiveness at flows above 4 gpm. This is relatively much worse than the flat plate heat exchanger performance discussed with reference to Figure 4.12 in which the effectiveness at 3 gpm exceeded 0.8. Hence the data shows that using a coil-in-tank heat exchanger is not the best choice for the use in the thermal storage system tested in terms of performance (as opposed to other factors such as cost).

4.3 Closure

The experimental test program provided a comprehensive evaluation of the hybrid geothermal / low-grade heat space-conditioning system in both heating and cooling modes. Long-term operation in which the system operates cyclically under control of a conventional thermostat was compared against short-term continuous operation.

The results show that, in heating mode, using stored low-grade heat as a source is a viable long-term alternative to geothermal wells. However, the results also show that the potential performance benefits of a higher temperature heat supply afforded by a thermal storage source are not realized with a conventional vapor compression cycle heat pump. The measured maximum average COP[3] increased by 24% with an increase in source temperature from about 49 to 102 °F at a liquid flow rate of 1.8 gpm. This was in substantial agreement with the manufacturer’s reported maximum COP increase of 25% (although at a flow rate of 3.3 to 3.7 gpm) with an increase in source temperature from 48.5 to 86 °F. This magnitude of increase does not reflect the approximately three-fold increase in reversed Carnot COP possible with the experimental source temperature increase.

Under steady-state test conditions, the experimental and manufacturer’s heating COP data do not agree in absolute terms, although there appears to be substantial agreement under cyclic, non-continuous operating conditions. However, again under steady-state conditions, the experimental and manufacturer’s cooling COP data do agree in absolute terms. This discrepancy was not resolved and thus requires additional research.
CHAPTER 5 SUMMARY AND CONCLUSIONS

The main goal for the hybrid energy system study (HESS) II project was to explore the potential efficiency improvements that could be made to a hybrid residential/commercial heating/cooling system by combining a commercially available reversible geothermal heat pump system with a low-grade heat generation and stratified thermal storage system. This required access to a source of low-grade heat as well as the design and construction of an insulated stratified storage system. Also required was a carefully designed data acquisition system with real-time graphic data display capabilities, a manual control system for transferring the thermal energy from the stratified thermal storage system, and various ways of connecting multiple heat sources or sinks to the heat pump. This operational hybrid energy system was tested in the newly built HESS II (24' x 24’) laboratory attached to the Quarry Hill Nature Center in Rochester, Minnesota. The laboratory building was designed to have a very energy efficient envelope.

A patent was filed for the original HESS II system concept on January 31, 2005 and was granted and issued by the US Patent Office on February 26, 2008 as US Patent 7,334,406, Hybrid geothermal and fuel-cell system [2]. Several conceptual system improvements were made to the patented HESS II system in the period between the filing date of the patent and the start of the HESS II project research. Some of these improvements were implemented in the experimental HESS II system.

The commercially available reversible geothermal heat pump used was a Trane GEV-018. This is a 1.5-ton water-source reversible heat pump connected to a closed-loop geothermal field of two 200-foot deep wells connected in parallel via tubing in a 90-foot long trench. The piping manifold interconnecting the wells to the balance of the system was located inside the laboratory. Except for the upper 20 feet, the well geotechnical environment consists mainly of limestone. Initially, the well tubing was not properly flushed for debris, resulting first in organic particles and pipe sealing compound clogging up flow meters, and ultimately in circulating sand dislodged at flow rates of 5 gpm causing a pump failure. Besides the re-flushing of the pipes, it was found that wire mesh filters installed upstream from flow meters and pumps were essential to ensure reliable operation of the closed-loop liquid system.

The original objective was to use low-grade heat generated from a hydrogen fueled Proton Exchange Membrane (PEM) fuel cell. A 5kW, 24-volt DC, 108-Ampere, 5T24 PEM fuel cell system manufactured by Plug Power, Inc. was purchased for this purpose. Rochester and State building codes required the tanks of hydrogen gas to be stored in an approved storage cabinet that was located at least 25-feet away from the laboratory building. Heat rejected from the fuel cell via circulating fluid at 130°F during fuel cell operation was captured by a flat-plate heat exchanger and transferred to a 480-gallon stratified thermal storage system. However, after about 10 hours of operation, the fuel cell failed due to a failure of a pressure regulator in the hydrogen storage cabinet. To quickly recover from this complication, a low-grade heat generation emulator was designed and built to produce heat for the heat pump and the thermal storage system. This emulator could simulate the heat rejected from a variety of low-grade
thermal energy sources including a PEM fuel cell. The heat emulator was a much better choice for this purpose than the fuel cell since both the rate of liquid flow and the liquid temperature could be varied over a large range. The operating fuel cell only provided a fixed flow rate at a fixed temperature.

The hybrid energy plant was arranged into three interlinked loops that enabled low-grade exhaust heat from the emulator to be captured and stored for later use by the heat pump to heat the laboratory space. In cooling mode, the geothermal wells were used as the heat sink. The heat generated by the heat pump in this mode was recovered and sequestered in the thermal storage system.

Data collected over a 10-month period fully characterized the steady state and transient dynamic behavior of the HESS II system. It was found that this hybrid energy system operating at a thermal source temperature of about 100 °F yielded a 24% improvement in heating COP at a flow rate of 1.8 gpm compared with operation at an average well source temperature of 48.5 °F. However, it was also determined that the conventional vapor compression cycle heat pump used is not ideal for this application since it does not maximize the heating COP that is theoretically possible with the increase in source temperature provided by using stored low-grade heat.

In a fairly severe test (as shown in Figure 4.3(a)), the 400-gallon stratified thermal storage system, initially at 135°F, was sufficient to maintain a setpoint temperature of 68°F in the laboratory for more than five days during which the ambient temperature averaged 5°F. It was observed that because the insulated thermal storage system was located inside the heated laboratory space, the heat loss from the thermal storage system aided in heating the laboratory so reducing heat pump energy consumption. Thus, in a HESS II product application it is advisable to locate the thermal storage system within the conditioned space of the building (typically in the basement or sub-basement if a residential or commercial building) instead of outside the building in a below-grade location.

5.1 Future directions

Most of the electrical power used by a conventional reversible heat pump is consumed in the compressor. For a typical residential 5-ton system with a two-speed compressor, the compressor uses 80% of the power required to operate the entire geothermal installation (compressor, circulating air blower and liquid pumps). During a very cold winter month, the energy used by the compressor can exceed 90% of the total geothermal system electrical power. In order to reduce the electrical operating power required by reversible heat pumps, the focus should be on minimizing the compressor power that most directly increases the heat pump COP or efficiency. One way of doing this is to use a heat pump with an optimally controlled variable speed compressor instead of the single speed compressor unit used in the HESS II project.
Another method of increasing the efficiency and reducing the operating cost of a HESS II type heat pump system during the heating season is to use the warm liquids from the thermal storage system during very cold winter nights and the geothermal field during the days. This alternating process would allow the ground surrounding the geothermal field to recover some of the local heat loss during the night.

A third possibility is to design a fuzzy logic control system that would maximize the heat transfer from the thermal storage to the heat pump by appropriately controlling the liquid flows within the interlinked HESS II system loops as shown in Figures 3.2-3 and 3.2-4. While conducting the experiments, several saddle points in various loop flows within the system were discovered that could be used for optimal control.
APPENDIX A

BIBLIOGRAPHY

The principal goal of the HESS II Project was to implement a workable prototype of the HESS II patent rather than repeat the research that had already been completed in developing the patent. Therefore, it was not considered necessary or financially prudent to repeat a literature search in the field of geothermal heat pump based hybrid energy systems in the HESS II project. Instead, the available limited resources were focused on fabricating the prototype plant and collecting experimental data. Relevant references are listed below.


APPENDIX B
EQUIPMENT SPECIFICATIONS

B1. **Heat Pump**
High Efficiency Vertical Water-Source Heat Pump System, GEV-018
Manufacturer: Trane Corporation
Part Number: GEV-B-018-2-1-N/A-0-1-2-0-T-L-D-0-0-0-0-1

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<td>High Efficiency Vertical</td>
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<tr>
<td>Digit 4:</td>
<td>B</td>
<td>Sequence B</td>
</tr>
<tr>
<td>Digit 5-7:</td>
<td>018</td>
<td>1.5 Ton</td>
</tr>
<tr>
<td>Digit 8:</td>
<td>2</td>
<td>230/60/1</td>
</tr>
<tr>
<td>Digit 9:</td>
<td>1</td>
<td>Copper-Water Coil</td>
</tr>
<tr>
<td>Digit 10:</td>
<td>N/A</td>
<td></td>
</tr>
<tr>
<td>Digit 11:</td>
<td>0</td>
<td>Heating and cooling</td>
</tr>
<tr>
<td>Digit 12:</td>
<td>1</td>
<td>Standard blower configuration</td>
</tr>
<tr>
<td>Digit 13:</td>
<td>2</td>
<td>Geothermal design for Trane Commercial Group</td>
</tr>
<tr>
<td>Digit 14:</td>
<td>0</td>
<td>Open digit</td>
</tr>
<tr>
<td>Digit 15:</td>
<td>T</td>
<td>Supply-air arrangement</td>
</tr>
<tr>
<td>Digit 16:</td>
<td>L</td>
<td>Return-air arrangement</td>
</tr>
<tr>
<td>Digit 17:</td>
<td>D</td>
<td>Deluxe 24 Volts control</td>
</tr>
<tr>
<td>Digit 18:</td>
<td>0</td>
<td>Wall mounted thermostat</td>
</tr>
<tr>
<td>Digit 19:</td>
<td>0</td>
<td>No fault sensor</td>
</tr>
<tr>
<td>Digit 20:</td>
<td>0</td>
<td>No additional temperature sensor</td>
</tr>
<tr>
<td>Digit 21:</td>
<td>0</td>
<td>No night setback relay</td>
</tr>
<tr>
<td>Digit 22:</td>
<td>1</td>
<td>Internal boilerless electric heat</td>
</tr>
</tbody>
</table>

www.r.umn.edu/HESS/TRANE_Heat_Pump.pdf

B2. **Fuel Cell System**
GenCore 5T48 5kW PEM Fuel Cell System with Hydrogen Storage Cabinet and Communication Package
Manufacturer: Plug Power, Inc.
Part number: GenCore 5T48
Technology: Proton Exchange Membrane (PEM)
Power: 5 kWatt (DC)
Voltage: 48 Volts
Current: 108 Amperes
Cooling: Liquid/Air Heat Exchanger
Gas: Hydrogen 99.95% pure, dry
Hydrogen Storage Cabinet: 408542261
Communication Package 1: 408542204 (modem)

http://www.fuelcellmarkets.com/content/images/articles/GenCore_Telecom_Datasheet.pdf
B3. DC/AC Inverter
Xantrex SW5548 Series II Inverter/Charger 5500 Watt 48 Volts with Grid-tie
AC input voltage 120 VAC
AC input voltage range 80-149 VAC
AC input current 60 amps AC pass through 35 amps AC charging
Continuous Power @ 25C 5500 VA
Efficiency (Peak) 96%
Output voltage (RMS) 120 VAC
Output voltage Regulation +/- 3%
Frequency (Nominal +/- 0.04% crystal controlled) 60 Hz
Continuous output @ 25C 46 amps AC
http://www.affordable-solar.com/xantrex.sw5548.series.II.inverter.htm

B4. Water Heater
Residential Solar Electric Water Heater with Inside Heat Exchanger
Manufacturer: Rheem
Part number: 81V-80HE-1 Solaraide SE
Capacity: 80 gallons
Wattage: 6.0 kW
Heat exchanger: 120 feet of coiled 5/8" copper tubing inside tank
http://www.aaasolar.com/ProdLit/Rheem/80galWHE.pdf

B5. Tank Farm
B5.1 Water Tanks
100-gallon, 28.5"x 51.25", Closed Head, Propylene Tank
Manufacturer: US Plastic Corp.
Part number: 5319
http://www.usplastic.com/catalog/product.asp?catalog%5Fname=usplastic&category%5Fname=30&product%5Fid=11984

B5.2 Water Level Transmitter
Continuous Level Transmitter, brass housing, 40” indicating length
Manufacturer: Dwyer Instruments, Inc.
Part number: CLT-V-B-2-F140-1/4-42.25
http://www.dwyer-inst.com/htdocs/level/SeriesCLTPrice.cfm
http://www.dwyer-inst.com/htdocs/level/SeriesCLTSpec.CFM

B6. Heat Exchanger
B6.1 Flat Plate Copper Brazed Liquid-to-Liquid Heat Exchanger
Manufacturer: FlatPlate, Inc.
Part number: FP10x20-36
B7. **Light Commercial Heat Recovery Ventilator**
Unit ventilator for code approves HESS II laboratory hydrogen safety system
Manufacturer: FanTech
Part number: HRV-XI 450

B8. **Pumps and Controllers**

B8.1 **Centrifugal Pumps**

Dayton Self-Priming Centrifugal Stainless Steel Pump
Manufacturer: Dayton Electrical Mfg. Co.
Part number: 4UA68
Pumping power: ½ HP
Electrical: 230V/60Hz/1Ph
Full load current: 4.0 Amps
Speed: 3500 RPM
Motor frame type: 56J

Dayton Portable Self-Priming Centrifugal Utility Pump
Manufacturer: Dayton Electrical Mfg. Co.
Part number: 4CB57
Pumping power: ½ HP
Electric power: 115V/8.0A/60Hz

B8.2 **Positive Displacement Pumps**

Two Procon Brass Vane Pumps, 330gph, bolt on
Manufacturer: Procon Products
Part number: 114E330F11XX
Flow: 5.5 gpm

Motor:
Part number: ACM01742
Power: ½ HP and ¼ HP
Speed: 1800 RPM
Electrical: 208-230V/3 phase inverter duty motor
Frame: NEMA 56C
Mounting: Rigid base
[http://www.appliedmembranes.com/propump1.htm](http://www.appliedmembranes.com/propump1.htm)
[http://www.appliedmembranes.com/promot1.htm](http://www.appliedmembranes.com/promot1.htm)
[http://www.proconpumps.com/Products.htm](http://www.proconpumps.com/Products.htm)
B9. Valves and Controllers

B9.1 Modulating electric ¾” NPT 3-way brass, automated ball valve (L1 port configuration)
Manufacturer: Dwyer Instruments, Inc.
Part number: 3ABV1U1103-L1
http://www.dwyer-inst.com/HTDOCS/Valves/Series3ABVPrice.CFM

B9.2 Two position electric ¾” NPT 2-way brass, automated ball valve
Manufacturer: Dwyer Instruments, Inc.
Part number: ABV101
http://www.dwyer-inst.com/HTDOCS/VALVES/SeriesABVSpec.CFM

B9.3 Automatic Proportional Valve
Modulating electric ¾” NPT 3-way brass automated ball valve
Manufacturer: Dwyer Instruments
Part number: 3ABV1V1103-L1
http://www.dwyer-inst.com/htdocs/valves/Series3ABVPrice.CFM

B9.4 Automated Globe Valve
1" globe control valve, 2-way, Cv of 9.3, max. differential pressure 145 psi (10 bar) fit with EVA2 or 203 psi (4 bar) fit with EVA3, 19/32” (15 mm) stroke.
Manufacturer: Dwyer Instruments, Inc.
Part number: GV221
http://www.dwyer-inst.com/htdocs/valves/SeriesGV2-GV3Price.cfm
GV221 Controller: Series EVA2M Electric Actuators
Electric actuator, modulating control inputs, 225 lb (1000 N) output force, 1" to 2-1/2" valve size, compatible with GV2 or GV3 globe control valves.
http://www.dwyer-inst.com/htdocs/valves/SeriesEVAPrice.cfm

B10. Ventilation Equipment
Fantech light commercial heat recovery ventilator
Manufacturer: Fantech Corp.
Part number: HRV-XI 450
http://www.fantech.net/light_hrv_erv_inst.pdf

B11. Thermocouples
B11.1 Wire type thermocouple-junction
Manufacturer: Omega Engineering, Inc.
Part number: PR-T-24 (ripcord)
http://www.omega.com/pptst/GG_T_TC_WIRE.html
B11.2 T-type, SS, Quick Disconnect type
Manufacturer: Omega Engineering, Inc.

B12. Flow sensors
B12.1 Paddle type
0.6 – 12 gpm, NPT, 4-20mA 3-wire output, paddle flow sensor
Manufacturer: Kobold Instruments, Inc.
Part number: DRG1165 N5 L343
http://www.maselmon.com/assets/63/k_drg.pdf

B12.2 Turbine type
0.6 – 10.5 gpm OEM turbine flow sensor, brass body, ¾” NPT ports with 4-20mA 3-wire output
Manufacturer: Kobold Instruments, Inc.
Part number: DRS-9180N5 L343
http://www.maselmon.com/assets/63/K_DRS.pdf

B12.3 In-Line Flowmeter 1 – 10 gpm
Manufacturer: Omega Engineering, Inc.
Part number: FL-510

B13. Pressure Transducer
Rugged Solid State Pressure Transducer
Manufacturer: Omega Engineering, In.
Part number: PX209-100G5V
Range: 0 – 100 psi
Output: 0 – 5 volts

B14. Data Acquisition System (Hardware)
B14.1 SCXI -1000, 4-slot chassis, 120VAC
Manufacturer: National Instruments Corp.
Part number: 776570-01
B14.2  Two SCXI -1102, 32-Channel Thermocouple Amplifier
    Manufacturer: National Instruments Corp.
    Part number:  776572-02

B14.3  Two SCXI -1303, 32-channel isothermal terminal blocks
    Manufacturer: National Instruments Corp.
    Part number:  777687-03

B14.4  SCXI -1124, 6-channel DAC module
    Manufacturer: National Instruments Corp.
    Part number:  776572-24

B14.5  SCXI -1325, High voltage screw terminal block
    Manufacturer: National Instruments Corp.
    Part number:  777687-25

B14.6  SCXI -1163R, 32 Channel Solid-State Relay Multiplexer
    Manufacturer: National Instruments Corp.
    Part number:  776572-63R

B14.7  SCXI -1326, High voltage screw terminal block
    Manufacturer: National Instruments Corp.
    Part number:  777687-26

B15.  Others
B15.1  AC/DC Hall Effect Current Transducer
    Manufacturer: CR Magnetics, Inc.
    Part number:  CR5410-20
    Input:  0 – 20 Amps
    http://www.crmagnetics.com/newprod/ProductViewPart.asp?ProdName=CR5410

B15.2  RMS to DC Converter
    Manufacturer: Analog Devices, Inc.
    Part number:  AD637JQ
    http://www.datasheetcatalog.org/datasheet/analogdevices/AD637BQ.pdf
B15.3 Airflow sensor
Duct airflow sensor for heat pump air input with transmitter.
Manufacturer: Ebtron
Part number: STA-102-PB/16x16-0-2-1-25
For more details see Ebtron Product Catalog, pages 114 – 130,
http://www.ebtron.com/Web_Pdfs/Ebtron_Catalog_v4.05.pdf

B15.4 Watt transducer
For measurements of power used by heat pump compressor and fan.
Manufacturer: Ohio Semitronics, Inc.
Part number: PC5-059CX5

B15.5 Barometric pressure transducer
Manufacturer: Yellow Springs Instrument Co., Inc
Model number: 2014-22/31.5-HA-3
Serial number: 16970
Range: 27" - 31.5" Hg
https://www.ysi.com/