Geothermal Heating and Cooling

Eric Sinson '23, Kevin Clark '23, Maalik McPherson '23
Professor John D. Mertens, Ph.D.
Trinity College Department of Engineering

Abstract

This project objectives were to design, test, and construct a reversible geothermal system that can provide supplemental heating and cooling. The vapor-compression thermodynamic cycle utilizes the constant temperature, approximately 55°F 4-6 feet below the earth's surface, to dump or take in heat. A phase change in the R-410a refrigerant maximizes the temperature difference with respect to the ground's constant temperature. The system includes two unique heat exchangers which have been designed, built, and tested to predict the heat transfer to and from the fluid. In heating mode, the system provides Q = 1.801 kW to the space at a coefficient of performance of β = 2.58. In cooling mode, the system provides Q = -683 W space at a coefficient of performance of β = 1.58. Based on New England climate requirements, the system is capable of heating and cooling a 116 sq. ft room or can provide supplemental heating and cooling of a larger space.

Background

Below the surface of the earth, about four to six feet, the underground temperature remains constant at approximately 55°F, regardless of the above ground conditions/season. In the winter, the system pumps a low temperature, low pressure refrigerant into the ground, taking in heat as the fluid becomes vapor. The fluid now passes through a compressor, where work is done, and the then superheated vapor enters the residential space, where heat is transferred from the fluid into the room. For warm conditions, the cycle is reversed, and the fluid will instead take heat from the room and dump it into the ground.

Testing Procedures

Test #1: Above Ground Heat Exchanger

A car condenser and a fan were used to create a cross flow, single path, unmixed heat exchanger. Hot water was run through the condenser while the fan blew room temperature air across the condenser to determine its Heat Transfer Coefficient (UA) using NTU-effectiveness analysis.

$$UA\left(\frac{W}{K}\right) = \left[\frac{1}{\eta_o h_{air} A_{external}} + \frac{1}{h_{fluid} A_{internal}}\right]^{-1} \qquad Q = UA\varepsilon\Delta T$$

$$\varepsilon = 1 - exp\left[\frac{1}{C_r} * NTU^{.22} * \{exp(-C_r * NTU^{.78}) - 1\}\right]$$

Water Test				
UA	Predicted Q	Actual Q	Effectiveness	Error Factor
191.3 W/K	2520 W	3026 W	.700	1.2

Test #2: Below Ground Heat Exchanger

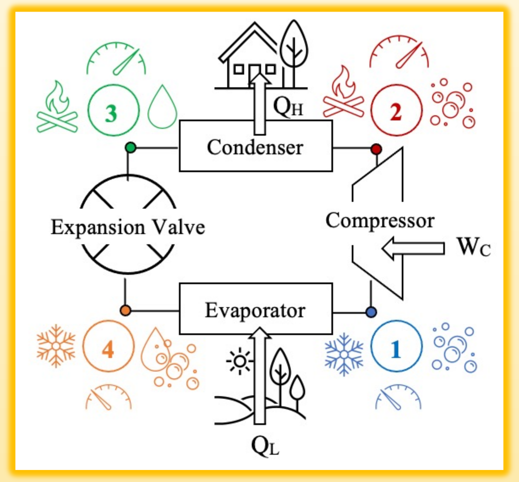
Hot water was flowed through 3 different pipe configurations (straight pipe, fins, dirt) in an ice bath to characterize Thermal Resistance (R) as a function of mass flow rate ($\dot{m}$). The performance of the designed heat exchanger in the ground with refrigerant could be predicted by nondimensionalizing the test results. The Maalik # (Ma) is the non dimensional heat transfer coefficient and Reynolds # (Re) is the non dimensional fluid flow.

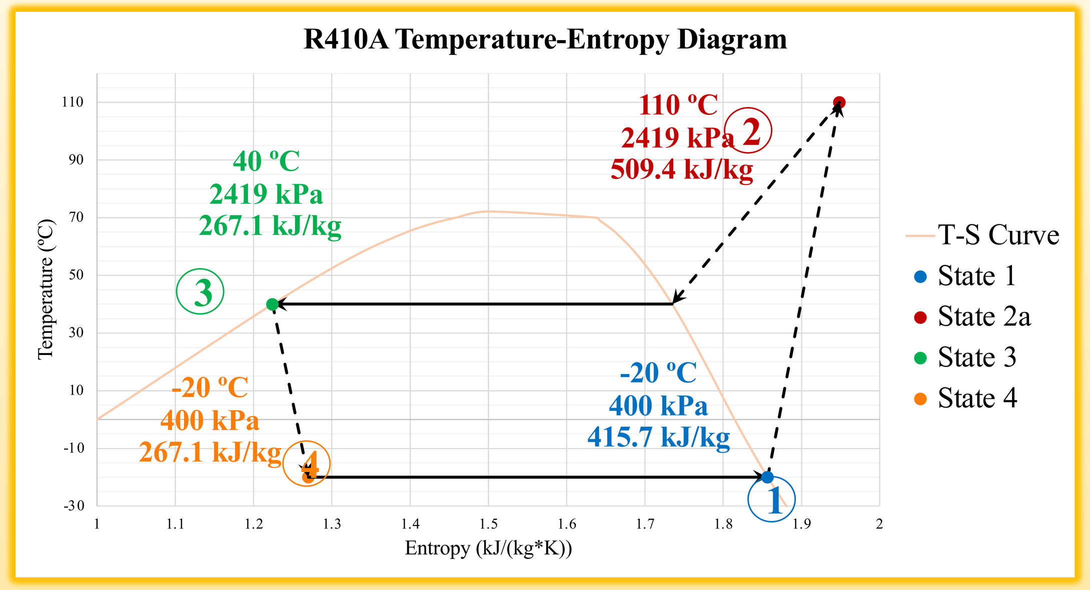
$$Ma = \frac{lc_p\Delta T\rho^2}{Rk\mu^2}$$

$$Re = \frac{4\dot{m}}{\pi D\mu}$$

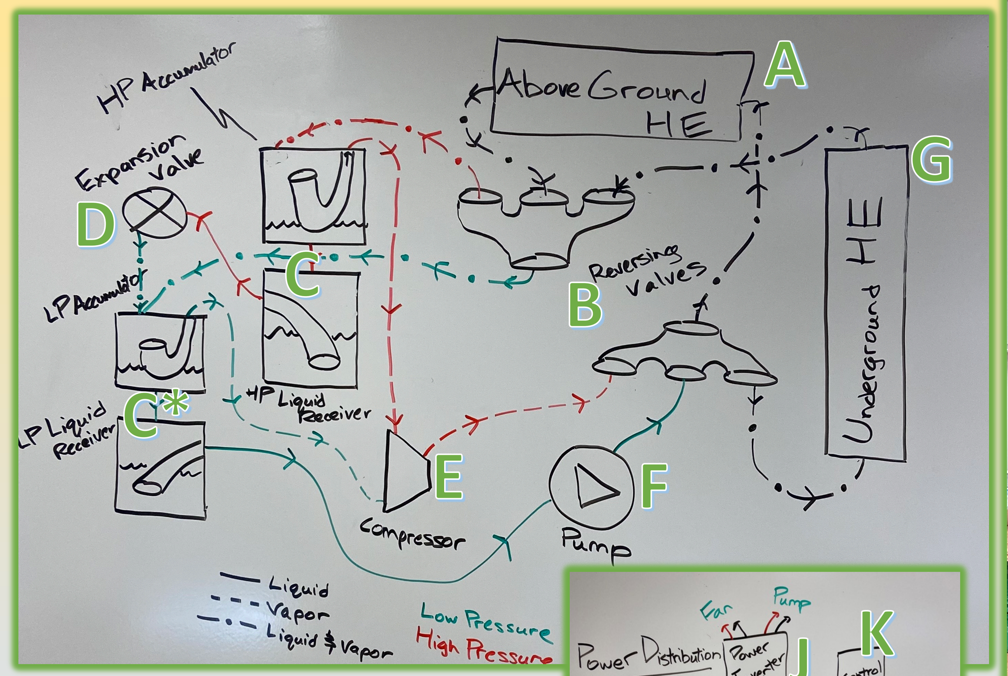
Vapor Compression Cycle

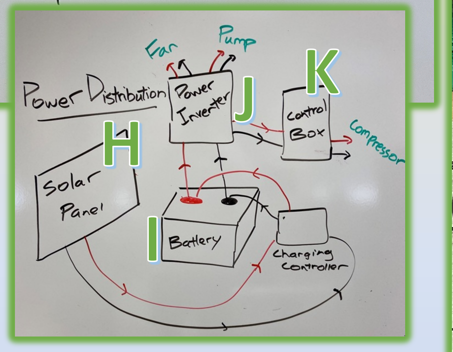
In this type of Thermodynamic cycle, the evaporator takes in heat as the refrigerant boils, the work done at the compressor raises the pressure and temperature. The condenser dumps heat as the refrigerant condenses, and the expansion valve decreases the pressure and temperature of the refrigerant, and the cycle continues.



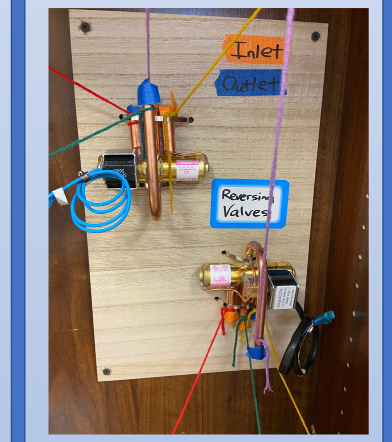


System Components





Unique Designs



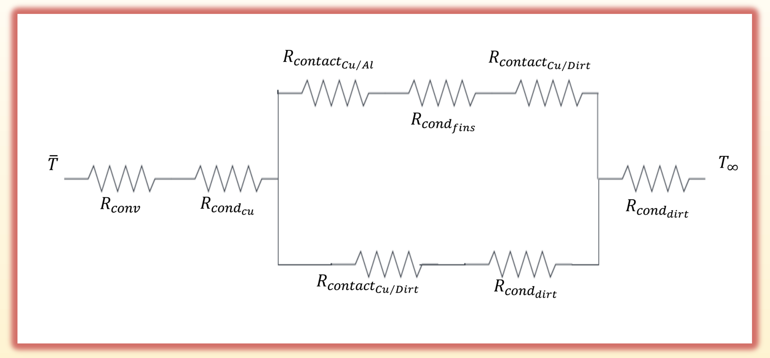
The two reversing valves allow for the system to be switched from heating to cooling modes without changing any other element of the system.

The accumulator and liquid receiver paired together allow for physical separation vapor and liquid so that refrigerant can be recirculated through the heat exchangers.

A solar power subsystem allows the compressor and other devices to be powered renewably.

The below ground heat exchanger is a copper coated aluminum sheet bent around a copper pipe and attached by stainless steel zip ties. This design reduces the length of the heat exchanger by five times.

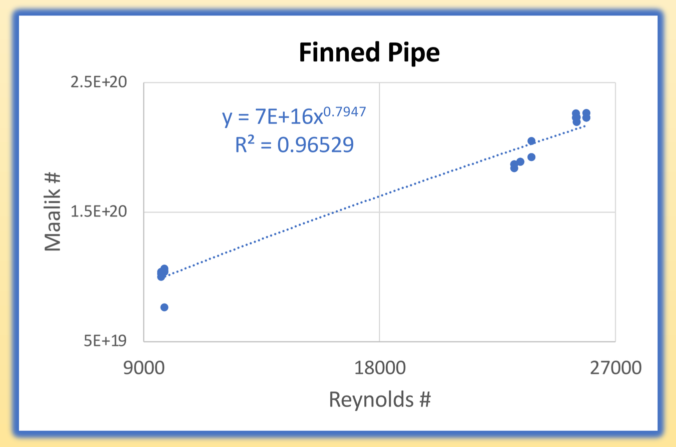
Analysis

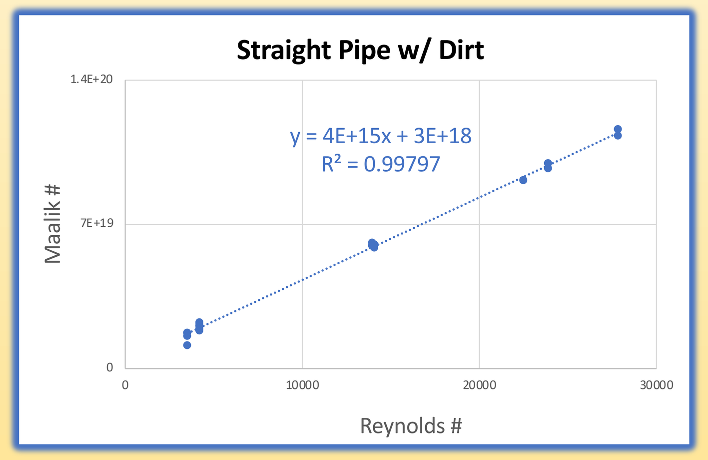


$$R_{contact_{Cu/Dirt}} = .0907\frac{Wm^2}{K}$$

$$R_{contact_{Cu/Al}} = .004518\frac{Wm^2}{K}$$

$$\Sigma R = \frac{\mu^{.8}k_{fluid}{}^{.3} * .002623}{l * \dot{m}^{.8} * (\mu c_p)^{.3}} + \frac{8.8 * 10^{-5}}{l} + \frac{1}{\frac{1}{7.5 * 10^{-5} * l + \frac{1.136 * 10^{-5}}{l} + 4.37 * 10^{-5} * l} + \frac{1}{.003013 * l + \frac{.7028}{l}}} + \frac{.208}{l}$$





$$Q = \dot{m}\Delta h = \frac{\Delta T}{\Sigma R} = UA\varepsilon\Delta T$$

$$W = \dot{m}(h_2 - h_1);\ \beta = {}^{Q}/_{W}$$

Results

Heating Cycle				
Q (W)	β	$\dot{m}\ (\frac{kg}{s})$	L_{fin}(m)	L_{Nofin}(m)
1801	2.58	.00744	6.94	32.96

Cooling Cycle				
Q (W)	β	$\dot{m}\ (\frac{kg}{s})$	L_{fin}(m)	L_{Nofin}(m)
-683	1.58	.00462	3.75	17.81

The heat transfer (Q) exiting the above ground heat exchanger was predicted using the analysis from the above and below ground heat exchanger testing. The required enthalpy difference determined by the thermodynamic cycle informed the ideal mass flow rate. Using the ideal mass flow rate and the below ground testing, the necessary length of finned pipe was determined to satisfy the needed enthalpy change in the fluid. This length when compared to a non-finned pipe demonstrates a five times improvement.

References

- Incropera, F. P. (2007). *Introduction to heat transfer*. J. Wiley.
- Sonntag, R.E. and Borgnakke, C. (2009) *Fundamentals of thermodynamics*. New Jersey: John Wiley.
- Carl, M., Guy, D., Leyendecker, B., Miller, A., & Fan, X. (2012, April 4). *THE THEORETICAL AND EXPERIMENTAL INVESTIGATION OF THE HEAT TRANSFER PROCESS OF AN AUTOMOBILE RADIATOR*. Retrieved May 1, 2023, from https://www.lamar.edu/engineering/_files/documents/mechanical/dr.-fan-publications/2012/No%2031%202012%20ASEE%20Radiator%20Final.pdf

Acknowledgements

- Professor Mertens
- Professor Byers
- Andrew Muslin
- Travelers Insurance
- NASA CT Space Grant Consortium