Resource Assessment of Geothermal Reservoir in Western Alberta and Evaluation of Utilization Options Using Non-Renewable Energy Displacement

Casey Lavigne

Thesis of 60 ECTS credits
Master of Science (M.Sc.) in Sustainable Energy Engineering

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by

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Thesis of 60 ECTS credits submitted to the School of Science and Engineering at Reykjavík University in partial fulfillment of the requirements for the degree of Master of Science (M.Sc.) in Sustainable Energy Engineering

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Abstract

Geothermal resources can be employed in service of either direct-use heating applications or indirect-use power generation. This study applies a non-renewable energy savings approach to the evaluation of a prospective geothermal reservoir near the town of Hinton in Western Alberta, Canada. The energy content of the resource is estimated and two exclusive development options – power generation and space heating – were modelled and analysed.

Monte Carlo simulations were used to determine an estimated wellhead thermal output of 226 MWth at 95% cumulative probability for a project lifetime of 50 years. The thermal power was converted to a brine flow rate of 540 kg/s at the reservoir average temperature of 118°C. A binary power plant was modelled and optimized in EES using the estimated geothermal flow. The resulting n-butane power plant model produced 12.1 MWe net power with a seasonal range of 9.5-16.1 MWe. The model operated at a thermal efficiency of 9.2% and functional and overall second law efficiencies of 36.4% and 20.0% respectively. A residential district heating system was modelled in EES using a 80/40/-20 design criteria. The design resulted in a system capable of heating over 18,000 houses year-round, with excess energy available to potential industry partners. The heating system operates at 92.4% thermal efficiency under design conditions.

The power plant scenario provides 108 GWh annually, translating to a fossil fuel energy content savings of 649 TJ per year. The district heating option provides a potential of 3840 TJ of thermal energy annually, resulting in a fossil fuel savings of approximately 4267 TJ – an increase of 557% from the power generation scenario.

In total, the prospective heating network would then save 213 PJ (5.76B m³ natural gas) of non-renewable energy, more than six times the 32.4 PJ (0.87B m³ natural gas) saved by the power generation scenario. Alternatively, if designed to serve only the residential energy needs of the nearby town of Hinton, the power plant could provide 100 TJ/year for almost 200 years and the heating system 301 TJ/year for over 630 years.

The results of this case study analysis are applicable to any similar community located within an energy economy dominated by fossil fuels. The seminal finding from the study is the conclusion that the non-renewable energy payback of direct-use application was 6.6 times that of the indirect application. Using available geothermal resources to replace space heating most readily fulfills the objective of displacing the maximum amount of non-renewable energy.
Jarðvarmamatur svæðis í Vestur Alberta og samanburður á nýtingarmöguleikum miðað við skipti úr öndurnýjanlegum orkugjöfum

Casey Lavigne

mai 2018

Útdráttur

Jarðvarmi er ýmist nýttur í beina nýtingu s.s. húshitun eða til rafrorkuframleiðslu. Í þessu verkefni var jarðhitasvæði nærri bænum Hinton í Vestur Alberta, Kanada metið með því að meta hversu mikið af öndurnýjanlegri orku myndi sparast við að nýta það. Varmamagn jarðhitasvæðisins er metið og tveir kostir, rafrorkuframleiðsla og húshitun – voru metnir með útreikningum og líkanagerð.

Monte Carlo líkön voru notuð til að meta varmaorku sem reyndist vera 226 MWth m.v. 95% líkur og liftíma verkefnisins 50 ár. Það orkumagn samsvarar 540 kg/s flæði jarðhitavatns við meðalhitastig 118°C. Sett var upp líkan í EES fyrir orkuver sem nýtir tvívökva orkuferli sem nýtur það flæði. Með því að nota n-butane vinnslubvökva mætti framleiða 12.1 MWe af rafrorku eða á bilinu 9.5-16.1 MWe eftir árstíðum. Líkanid gefur orkunýtni upp á 9.2% en rekstrarnýtni og annars lógmáls nýtni 36.4% og 20.0%. Líkan fyrir hitaveitu var búið til í EES sem notar hönnunarforsenduna 80/40/-20. Niðurstöður úr því líkan voru að hægt var að hita upp meira en 18,000 hús allt árið um kring, þar sem umframorka væri einnig til staðar sem mætti nýta í íðnaði. Hitaveitukerfið gefur orkunýtni upp á 92.4% miðað við þessi skilyrði.

Sviðsmyndin fyrir orkuverið gefur árlega 108 GWh af rafrorku sem samsvarar sparnaði á jarðefnaeldsneyti upp á 649 TJ á ári. Hitaveitan gæti gefið 3840 TJ af varmaorku á ári, sem samsvarar sparnaði á jarðefnaeldsneyti upp á 4267 TJ á ári – sem er um 557% meira en rafrorkusviðsmyndin gæfi. Í heildina, þá myndi hitaveitan spara 213 PJ (5.76B m³ af jarðgasi) af öndurnýjanlegri orku eða meira en sex sinnum meira en þau 32.4 PJ (0.87B m³ jarðgas) sem orkuverið myndi spara. Að auki, ef þetta væri hannað fyrir bæinn Hinton, þá myndi orkuverið geta framleitt 100 TJ/ári í allt að 200 ár, en hitaveitan 301 TJ/ár í yfir 630 ár.

Þessar niðurstöður mætti nýta í svipaða sviðsmynd þar sem orkuframleiðsla úr jarðefnaeldsneyti er ráðandi. Meginnið niðurstöðuðan er að sparnaður á jarðefnaeldsneyti er 6,6 sinnum meiri ef hitaveitað er notuð heldur en ef varminn er nýttur í rafrorkuframleiðslu. Með því að nýta jarðvarma til hitaveitu er skilyrði um hámarks orkuskipti úr jarðefnaeldsneyti uppfyllt.
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Casey Lavigne  
Master of Science
To my mom, for the never-ending support.
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ANSI      American National Standards Institute
ASHRAE    American Society of Heating, Refrigerating, and Air-conditioning Engineers
AB        Alberta
B         Billion
BC        British Columbia
CFC       Chlorofluorocarbon
EES       Engineering Equation Solver
FEED      Front-End Engineering
GHP       Geothermal Heat Pump
GWP       Global Warming Potential
HC        Hydrocarbon
HCFC      Hydrochlorofluorocarbon
HFC       Hydrofluorocarbon
HX        Heat Exchanger
MB        Manitoba
O&M       Operating and Maintenance
ODP       Ozone Depletion Potential
ORC       Organic Rankine Cycle
TDS       Total Dissolved Fluids
U of A     University of Alberta
YK        Yukon
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<td>Saturation</td>
</tr>
<tr>
<td>th</td>
<td>Thermal</td>
</tr>
<tr>
<td>turb</td>
<td>Turbine</td>
</tr>
<tr>
<td>wh</td>
<td>Wellhead</td>
</tr>
<tr>
<td>wf</td>
<td>Working Fluid</td>
</tr>
</tbody>
</table>
Chapter 1

Introduction

Globally, energy produced by geothermal power generation is roughly equivalent to that of geothermal direct-use applications. Both direct and indirect sectors have grown significantly in the past couple of decades, but, as a whole, geothermal development has not kept up with the exponential growth seen with other renewables, namely wind and solar energy. However, the fact that geothermal energy can be harnessed by two different methods - direct and indirect - makes it unique among renewable energy sources. As a result, individual applications of geothermal energy can be customized to suit different needs for various industries and can also be applied in a cascading manner to extract as much energy from the resource as is feasible.

Despite the fact that the two sectors contribute to geothermal energy production equally, power generation research and implementation projects often receive more publicity and funding than direct-use applications. The value of liquid-dominated low- to moderate-temperature reservoirs is often overlooked in the search for reservoirs suitable for large power projects. This is due to the fact that, in general, electricity is more marketable than thermal energy and power projects typically have a higher potential for return on investments. Nonetheless, this approach results in the under-utilization of smaller and/or cooler reservoirs which may contain a large amount of thermal energy.

This study aims to determine the value of a moderate-temperature reservoir under two exclusive development scenarios - power generation and direct-use space heating - by determining their equivalent displacement of non-renewable energy. A case study approach is taken to assess a potential resource in Western Alberta, Canada and evaluate the outcomes of development scenarios with respect to the regional energy market. The ultimate goal of the study is to quantify the difference between the non-renewable energy savings of the power and heating options. The disparity of the fossil fuel savings, or lack thereof, between the development options can be factored into development decisions of greenfield geothermal projects, especially those in energy markets saturated with fossil fuels.

1.1 Geothermal Energy

Geothermal energy utilization denotes the extraction and application of thermal energy stored in the Earth’s crust. The Earth’s molten iron-nickel core is a product of the materials and forces that formed the planet. It is this residual formation heat at the core, along with radioactive decay of elements in the crust, that comprise geothermal energy. It is the energy stored in the outermost 10 km of the crust that is available for exploitation by the world’s population. Temperature of the crust increases from the surface at a rate of 30°C/km on
average but can reach gradients in excess of 90°C/km in regions of thin crustal zones, such as tectonic boundaries [1] [2]. Geothermal energy is therefore present, in varying capacity, everywhere on Earth.

Due to various geological factors, geothermal reservoirs are found at a wide range of temperatures. It is common in industry to place reservoirs in general categories of high-temperature and low-temperature, however, the division between the two is not always consistent. Many academic and industry groups have included a medium- or moderate-temperature category to attempt to delineate the potential applications of the reservoirs more discretely. This study will follow the USGS reservoir categorization which describes high-temperature as above 150°C, moderate-temperature as 90°C to 150°C, and reservoirs at less than 90°C as low-temperature [3]. These categories loosely correspond to applications of flash steam power generation, binary power generation, and direct-use applications respectively, as shown in the Lindal diagram in Figure 1.1. The two main categories of geothermal energy applications are expanded upon in the following sections.

Figure 1.1: Potential applications of geothermal energy depending on reservoir temperature. [4]

1.1.1 Direct-Use

Direct-use describes applications in which the thermal energy present in a geothermal fluid is the end-user entity. Utilization of the thermal energy can be achieved by using geothermal brine directly, such as bathing in a hot spring, or using a radiator or heat exchanger to heat another fluid.

Dating back to prehistoric times, heating of pools for bathing is the oldest form of geothermal energy [5]. In present day, many countries, such as Iceland, Hungary, and Turkey, employ geothermal energy for the majority of their swimming pool heating and balneology heating needs. Installed capacity of geothermal heating for bathing and swimming is the second-highest worldwide application [5]. Space heating (and cooling) accounts for the majority of direct-use geothermal energy, either directly (10.7%) or through the use of heat pumps (70.9%). Iceland, for example, relies on geothermal energy for 89% of its space heating needs [6]. Beyond space heating and bathing, direct-use geothermal energy is widely used for a variety of industrial applications; greenhouse heating, food processing/drying, hardwood drying, and aquaculture being some of the more prevalent direct-use
1.1. GEOTHERMAL ENERGY

applications. Figure 1.2 depicts major direct-use applications as a proportion of the worldwide installed capacity of 70,885 MWth in 2015 [5].

Geothermal heat pumps (GHPs) are primarily used to supply heating and/or cooling for commercial or residential spaces. The systems rely on the relatively constant temperature of the ground at shallow depths and can therefore be installed and utilized wherever shallow drilling is possible. Common residential GHPs send water through vertical or horizontal piping loops in the ground where thermal energy is either gained or rejected, depending on what service the heat pump requires, and exchanges heat with a working fluid in a refrigeration cycle.

GHP systems are typically sized to compensate for the annual temperature extremes of their specific environment. The majority of GHPs are therefore not required to operate at full capacity for much of the year and have a low capacity factor compared to other applications which have more constant and predictable loads, such as the heating of greenhouse heating and swimming pool heating. While GHPs account for nearly three quarters of the installed capacity of direct-use applications, their energy utilization represents just over half of the worldwide usage, as shown in Figure 1.3.

Direct-use geothermal has seen tremendous growth over the past two decades. From 1995 to 2015, the worldwide installed capacity of geothermal direct-use applications has
increased sevenfold while the utilization has increased just fourfold [5]. The utilization has lagged behind the capacity due to the increased installation of GHPs which, as mentioned, have characteristically low capacity factors. In just the five-year period between 2010 and 2015, the capacity and utilization grew by 46.2% and 39.8%, respectively [5]. The growth of geothermal direct-use energy for the 20-year period is depicted in Figure 1.4.

![Figure 1.4: Growth of direct-use geothermal energy, installed capacity and annual utilization bases, for 1995 to 2015. Recreated from Lund & Boyd. [5]](image)

1.1.2 Indirect-Use

Indirect-use of geothermal energy refers to the generation of electricity from geothermal sources. A geothermal power plant works on the same principles, and has largely the same components, as any other thermal power plant operating on a Rankine cycle. The type of plant is defined by the source of the input thermal energy of the power cycle. In a fossil fuel power plant, coal, natural gas, or oil is combusted in a boiler to vaporize a working fluid, typically distilled water. The steam produced in the boiler drives a turbine-generator to produce power, then is condensed back to a liquid, pressure is increased through a pump and the cycle is repeated. In a geothermal flash power plant, high-pressure, high-temperature geothermal brine replaces the function of the boiler in the Rankine power cycle. The brine is extracted from wells, flashed to steam (if liquid), and used directly as the working fluid in the power cycle. Geothermal flash power plants can generate large amounts of base load power, as evidenced by The Geysers (1517 MWe) in California and Hellisheidi (303 MWe, 133 MWth) in Iceland.

Lower temperature geothermal fluids can also be used to generate power through the use of a binary plant. In this case, a heat exchanger is employed to transfer thermal energy from the geothermal fluid to a working fluid with a lower boiling point which then drives the power cycle. Common working fluids, such as refrigerants or hydrocarbons, often have a higher molecular mass than water and are referred to as organic working fluids. A Rankine cycle using such a fluid is commonly referred to as an Organic Rankine Cycle (ORC).

Power generation from geothermal power plants worldwide amounted to 73.5 TWh, or 265 PJ, in 2015 [7]. Geothermal power is integral for electricity production in several countries, such as Iceland, Kenya, and Costa Rica, however, it represents less than 1% of global electricity generation [8]. Wind and solar generation have outpaced geothermal in the past decade, largely due to the high risk and capital costs associated with geothermal projects, but also because of the limited geographical distribution of high-temperature reservoirs. Figure 1.5 displays the total worldwide installed capacity and annual utilization of geothermal
power generation from 1995 to 2015. A forecast of installed capacity in 2020, based on projects expected to be online, is also shown in the figure. The period from 2010 to 2015 experienced an annual growth rate of 3%, however, an increase in the growth rate is projected, the largest of which in the USA, Indonesia, and the Philippines [7].

Figure 1.5: Growth of installed capacity and annual utilization of geothermal power generation for 1995 to 2015. Projection of installed capacity in 2020 included. [7]

1.2 Geothermal Energy in Canada

1.2.1 Potential

Canada has tremendous potential to employ geothermal energy to reduce its dependence on fossil fuels. The commercial-scale geothermal resources are available in two forms: high-temperature volcanic regions and a low- to moderate-temperature sedimentary basin. Both types of resources are limited to the western part of the country. See Figure 1.6 for a summary of the country’s geothermal resources.

Figure 1.6: Map of Canada’s potential geothermal resources by geology and type of use. Source: Grasby et al. (2012). [9]
The Garibaldi volcanic belt is located in southeastern British Columbia (BC) and contains several volcanoes that may be suitable for high-temperature geothermal exploitation; the most promising of which being the Mount Meager system [9]. The northern Cordillera volcanic province constitutes a second volcanic system and is found in northern BC and throughout the Yukon (YK). These two systems are responsible for the majority of the highest down-hole temperatures found in Canada, but neither have been extensively characterized due to a lack of provincial regulatory framework related to geothermal leases as well as economic barriers to entry in local energy markets [10].

The Western Canadian Sedimentary Basin (WCSB) represents Canada’s most accessible geothermal resource and contains the area of interest for this study [11]. The basin extends from southern Manitoba (MB) to northeastern BC, underlying the majority of AB. The sedimentary cover begins at the eastern edge of the basin and slopes downwards in a southwesterly direction to a depth of over 6 km near the Rocky Mountains at the AB/BC border. Figure 1.7 displays the sedimentary depth and the main stratigraphy of the WCSB. Due to oil and gas exploration in Alberta, existing wells number in the hundreds of thousands, resulting in an extensive database of down-hole data. Bottom hole temperatures are used to map temperature gradients and determine the temperature of the basin at the bottom of the sedimentary cover, known as the pre-Cambrian basement. As expected, the temperature at the basement increases with depth as the sedimentary cover thickens from northeast to southwest [12]. The porous WCSB of western AB then represents a large geothermal resource suitable for binary power generation and/or commercial-scale direct-use heating.

1.2.2 Direct-Use

The geothermal energy industry in Canada currently consists solely of direct-use applications, namely space heating and bathing. Over 140 naturally-occurring thermal springs have been identified in the country, 12 of which have been commercially developed to some degree [14]. These developed springs, located in Western Canada, are used as a hot water source for swimming or bathing and amount to a modest total thermal power of 8.8 MWth [14]. Heat pumps are responsible for the vast majority of Canada geothermal energy production, totaling 1449 MWth in 2013 [14].

Direct-use on a commercial level in Canada is exceedingly rare and consists of a few large heat pump systems. The most novel of these installations is in abandoned mines which have been flooded. The flooded water is used in open or closed-loop heat pumps, acting as the heat sink or heat source depending on the season. The oldest such application is found in Springhill, Nova Scotia where an abandoned coal mine is used for space heating and cooling for several commercial users. According to one study, the mine provides 2440 GJ for heating and cooling on an annual basis, with the first commercial user reporting a payback period of less than one year when compared with the capital and operating costs of a conventional oil furnace [11]. The success at Springhill has spurred similar projects and feasibility studies at decommissioned mines around the country.

The heat pump applications in Canada utilize the nominal heat available in the ground at shallow depths. Geothermal, or ground-source heat pumps are often excluded from the category of low-temperature geothermal because exploitation of a geothermal resource implies a localized occurrence of a reservoir with sufficiently high temperature to be of use without a heat pump. The use geothermal energy that does not require the use of a heat pump as an intermediary is non-existent in Canada, aside from the minor contribution of the commercial hot springs.
Figure 1.7: Depth and main features of the Western Canada a) Isopach (m) of the sedimentary cover through the WCSB and designation of cross-section b) W-E layered cross-section of the basin. Source: Bachu (1993). [13]
1.2.3 Power Generation

Despite the demonstrated potential of the volcanic systems and the WCSB, Canada currently has no geothermal power generation. This is due in large part to relatively cheap fuel and electricity costs in Western Canada, where much of the geothermal potential lies. BC produces close to 90% of its electricity from hydroelectric sources and Alberta leverages its abundant fossil fuel resources, namely coal and natural gas, to produce most of its power [15]. Another major roadblock to geothermal development is the lack of governmental support and regulatory framework [10]. BC and Saskatchewan are currently the only provinces to offer a pathway to leasing a geothermal resource.

There are a handful of proposed power projects that have obtained permits over the past few decades, however, almost all have stalled due to a combination of the regulatory hurdles and high upfront capital costs. As of the writing of this study, the lone power projects that appear to be progressing are the DEEP (Deep Earth Energy Production) project in southeastern Saskatchewan and Borealis’s Canoe Reach project in BC. DEEP has recently secured a power purchase agreement from SaskPower, the provincial utility, to continue feasibility studies on proof-of-concept power generation [16]. Canoe Reach is a geopark concept wherein geothermal heating of a new community hot pool, along with industry heating with several commercial partners, is proposed, eventually followed by the development of commercial power generation [17].

1.3 Study Overview

This study aims to evaluate different utilizations of a geothermal reservoir in the service of a town in Western Alberta. As mentioned previously, there is significant geothermal potential in the WCSB with its relatively high permeability and steady increase of temperature with depth. Direct and indirect utilizations are modelled in service of the town of Hinton and are evaluated on their effectiveness in efficiently replacing current fossil fuel sources of energy consumed by the town’s population for heating and power. The calculation of the amount of displaced fossil fuel for each utilization and the relative comparison of those values is the ultimate objective of the study.

1.3.1 Political and Social Motivations

Influenced by the development of vast oil and gas reserves, Alberta had had a long history of conservative provincial governments. However, the New Democratic Party (NDP), a party with more liberal views, was elected to power in 2015. This shift coincided with a similar change at the federal level with the Liberal Party winning leadership from the Conservative party.

The change in political power, along with increasing public awareness and pressure, triggered a shift in energy policies in response to global climate change. The country now has a target to reduce greenhouse gas emissions by 30% below 2005 levels by 2030 as per the Paris Agreement [18]. The emissions target relative to the current projected rate is depicted in Figure 1.8. To this end, the federal government imposed a requirement for the provinces to implement a cap-and-trade emissions market or direct carbon pricing, through a levy or tax, by January 1, 2018 [18]. Provinces without a solution in place by 2018 are required to adopt a $10/tonne carbon tax, rising in $10 increments annually to $50/tonne by the year 2022.
1.3. STUDY OVERVIEW

Figure 1.8: Current and projected greenhouse gas emissions in Canada relative to targets set out in the Paris agreement. [18] [15]

1.3.1.1 Provincial Carbon Tax

Once elected, the NDP leadership of Alberta put forth a Climate Leadership Plan to curtail emissions and encourage renewable development. The plan included an emissions tax and carbon levy on fossil fuels with the more aggressive price tag, compared to the federal plan, of $20/tonne in 2017 and $30/tonne in 2018. The carbon levy translates into the rate increases of several common fuels, as shown in Table 1.1. The impact of the carbon tax and levy on businesses and residents is muted by subsidies, business tax reduction, and residential rebates [19]. Figure 1.9 shows a short-term increase in fuel prices upon introduction of the carbon tax, but other market forces appear to nullify this effect in a relatively short period of time.

Table 1.1: Rate increase of fossil fuels due to carbon levy instituted by Alberta government. [19]

<table>
<thead>
<tr>
<th>Type of Fuel</th>
<th>January 1, 2017 $20/tonne</th>
<th>January 1, 2018 $30/tonne</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel</td>
<td>+5.35 ¢/L</td>
<td>+2.68 ¢/L</td>
</tr>
<tr>
<td>Gasoline</td>
<td>+4.49 ¢/L</td>
<td>+2.24 ¢/L</td>
</tr>
<tr>
<td>Natural Gas</td>
<td>+1.011 $/GJ</td>
<td>+0.506 $/GJ</td>
</tr>
<tr>
<td>Propane</td>
<td>+3.08 ¢/L</td>
<td>+1.54 ¢/L</td>
</tr>
</tbody>
</table>

1.3.2 Energy in Alberta

1.3.2.1 Electricity

While renewable generation is responsible for 80\%\(^1\) of electricity generation in Canada, Alberta relies heavily on fossil fuels to generate power within the province. As shown in Figure 1.10, 90\% of the electricity consumed in the province was generated by either coal or natural gas [22].

One of the primary objectives of the Climate Leadership Plan formulated by the provincial government is the phaseout of coal power by 2030. This goal was echoed by the federal

\(^1\)Includes 16\% nuclear generation
CHAPTER 1. INTRODUCTION

Figure 1.9: Prices of gasoline, diesel, and natural gas shortly before and after the introduction of the carbon levy of $20/tonne on Jan. 1, 2017. [20] [21]

Figure 1.10: Source of electricity generation in Alberta, 2016 [22]. a) Installed capacity b) Total electricity generated.

government a year later in its declaration of a coal-free Canada by the same deadline. Gen-
1.3. STUDY OVERVIEW

Generation from coal combustion emits approximately 1 tonne CO$_2$ equivalent per kWh – more than double the emissions from natural gas plants$^2$ [23].

Of the 18 coal power plants in operation in the province, 6 were planned to operate beyond 2030. Part of the revenue generated from the carbon tax will be paid out to these power companies as remuneration for their early closure [24]. Where possible, some operators are adapting to this policy change by converting their existing coal plants to natural gas [25].

Two-thirds of the lost generation due to the coal phase-out is expected to be filled by additional natural gas generation, with the remainder being filled by renewables. To this end, the provincial government has put forth a goal of 30% renewable generation by the year 2030 [26]. This translates to an additional 5 GW$^3$ of renewable electricity generation in the next 14 years if there is no growth in demand, and almost 6.5 GW$^3$ of additional installed capacity if the average annual demand growth from the previous decade continues. The required increases in installed capacity are detailed in Table 1.2. To attempt to reach this goal, the government has constructed a Renewable Electricity Program to encourage investment in renewable power plants [26].

Table 1.2: Increase in natural gas and renewable power generation due to the planned coal phase-out in Alberta by 2030. [22]

<table>
<thead>
<tr>
<th>Year</th>
<th>Annual Growth in Demand</th>
<th>Coal (MW)</th>
<th>Natural Gas (MW)</th>
<th>Renewables (MW)</th>
<th>Total (MW)</th>
<th>Increase in Renewables (MW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2016</td>
<td>-</td>
<td>6,273</td>
<td>7,323</td>
<td>2,831</td>
<td>16,427</td>
<td>-</td>
</tr>
<tr>
<td>2030</td>
<td>0%</td>
<td>0</td>
<td>13,536</td>
<td>7,841</td>
<td>21,378</td>
<td>5,011</td>
</tr>
<tr>
<td>2030</td>
<td>1.8%</td>
<td>0</td>
<td>15,299</td>
<td>9,263</td>
<td>24,562</td>
<td>6,432</td>
</tr>
</tbody>
</table>

1.3.2.2 Heating

Canada’s colder climate requires that full-time residences be heated for much of the year. Water and space heating are responsible for over 80% of residential energy consumption, with almost two-thirds accounted for space heating alone, as shown in Figure 1.11. Space heating in Alberta is largely achieved through natural gas combustion, with forced-air furnaces as the chosen heating system for 94% of residences in the province [27]. The high heating demand results in an annual natural gas consumption of 119 GJ compared with an electricity consumption of 26 GJ (7200 kWh) [28]. The comparison between residential electricity and heating consumption is depicted in Figure 1.12.

1.3.3 University of Alberta Research

With such heavy fossil fuel consumption in both heating and electricity, there remains ample opportunity in Alberta to reduce its carbon footprint and increase its energy security by the introduction of renewable energy. The porous lithology along with the extensive down-hole data from the oil and gas industry provides tremendous opportunity for potential geothermal developers. The largest obstacle to a successful geothermal project is the uncertainty in the initial stages of the project which introduces high financial risk. The inherent risk in the development of a geothermal project is depicted in Figure 1.13. Leveraging the down-hole data...
With an increasing public awareness and advocacy for sustainable energy practices, the Canadian government has recently increased its investment in renewable energy research...
1.3. STUDY OVERVIEW

on both a provincial and federal level. In step with this development, the University of Alberta has developed a Future Energy Systems administration to support ongoing research related to renewable energy and energy storage. The Geothermal Energy group, headed by hydrogeologist Dr. Jonathan Banks, is focused on the accurate evaluation of geothermal potential in Alberta and the technologies associated with the development of direct and indirect geothermal systems.

The Geothermal Energy group completed a Deep Dive study in 2016 which aimed to characterize geothermal potential of the Devonian formations, some of the deepest in the WCSB. Potential geothermal power, thermal and electrical, was estimated for regions surrounding each of the six participating municipalities using data from wells drilled by the oil and gas industry. The study concluded that the area (50 km radius) surrounding the town of Hinton has the potential to produce 2500 MWth or 600 MWe over a 30-year lifespan - the highest of the municipalities in the study (see Figure 1.14).

![Figure 1.14: Estimated geothermal energy potential in the deep WCSB found in close vicinity to various counties in Alberta. [31]](image)

The results of the Deep Dive have spurred further collaboration with the town administration of Hinton. Ongoing studies include the evaluation of the potential for geothermal heating of several of the city-owned buildings, such as the RCMP, hospital, and town hall. To this end, Epoch Energy, a private company from Calgary, AB, delivered a pre-FEED (Front-End Engineering Design) feasibility study in early 2017 and has recently received funding to proceed with a complete FEED study to be delivered in 2018.

While Epoch’s study will investigate the feasibility of re-purposing one or two wells to shoulder some of the heating load in the larger town buildings, this thesis project posits the potential development of much larger geothermal reservoirs in the Upper Cretaceous zone of the WCSB near Hinton. In this respect, this study aims to provide guidance on future development of the available geothermal resources by the town relative to its needs and current sources of energy.

1.3.4 Town of Hinton

Hinton is located approximately 300 km west of the provincial capital, Edmonton, and has a population of just under 10,000 [32]. As shown in Figure 1.15, the town is situated at the western front of the WCSB where the basin is its deepest. The main industries supporting the economy of Hinton are coal mining, oil and gas, and a pulp and paper mill.

According to a 2016 census, there are 3,670 occupied households in Hinton. It is assumed, for the purposes of this study, that households in Hinton follow the heating statistics
previously described by Figures 1.11 and 1.12 for average households in Alberta. A household is assumed to use 119 GJ for water heating and space heating combined, with space heating representing 77% (from Figure 1.11: 64% / (64% + 19%)), or 91.8 GJ, of the energy. As of 2009, all furnaces installed in Canada were required to have a minimum efficiency of 90% [33]. With this regulation and the provincial heating statistics, it is concluded that the average Hinton household requires 82.6 GJ of thermal energy for space heating, as shown in Table 1.3.

<table>
<thead>
<tr>
<th>Table 1.3: Alberta household heating by natural gas.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Household Usage</td>
</tr>
<tr>
<td>Water Heating</td>
</tr>
<tr>
<td>Space Heating</td>
</tr>
<tr>
<td>Furnace Efficiency</td>
</tr>
<tr>
<td>Space Heating Thermal Energy</td>
</tr>
</tbody>
</table>

Electricity consumption for Hinton was received from the utility provider for this study. The consumption data was divided into categories, one being residential, and was provided for 2013 through 2015 as shown in Table 1.4. For the purposes of this study, it is assumed that residential and total electricity demand of the town is equivalent to that of 2015, where residential consumption was 27.9 GWh (average of 3.2 MW).

<table>
<thead>
<tr>
<th>Table 1.4: Hinton Electricity Consumption for 2013-2015</th>
</tr>
</thead>
<tbody>
<tr>
<td>Category</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>Residential</td>
</tr>
<tr>
<td>Town Total</td>
</tr>
</tbody>
</table>
1.3.5 Resource Assessment

This study begins by defining and assessing the potential of a target geothermal resource. Data from the previous U of A studies are leveraged to characterize a potential resource. The arbitrary geothermal volume is therefore defined and bounded by those wells for which data was provided and which met spatial criteria. While the U of A research was focused on estimating potential in deep formations (5+ km depth), this study evaluates the geothermal potential in the Upper Cretaceous period, found between 3 and 4 km depth.

1.3.6 Utilization Scenarios

As described in the previous sections, geothermal energy can be utilized in direct-use heating or for power generation. This study evaluates both utilization scenarios and compares their outcomes to inform on which option has the higher energy payback with respect to non-renewable energy. Specifically, for the power generation scenario, a basic geothermal binary power plant with air-cooled condenser is modelled and optimized to calculate its design net power output. For direct-use, a residential closed-loop district heating system is designed to provide heating to family households. The heating system is designed to provide heat to the maximum amount of houses with a minimum ambient temperature of -20°C. Both models are used to calculate their total lifetime energy output.

Each scenario is evaluated on how well it could potentially serve the nearby town of Hinton, and, more importantly, which scenario results in the highest amount of displaced fossil fuels over its respective lifetime. Quantifying the displaced non-renewable energy of the heating scenario relative to the power scenario will be the most valuable output of the study.
Chapter 2

Methodology

2.1 Geothermal Reservoir

As mentioned in the introduction, this study leverages the well log data gathered during the previous studies by the U of A Geothermal Energy group, but focuses on shallower Cretaceous formations as the prospective geothermal reservoir. The reservoir is therefore spatially bound by the wells selected for the study. To be eligible for the study, wells were required to meet the following criteria: be within 50 km of the town of Hinton, east of the deformation belt at the western edge of the WCSB, penetrate to the Cretaceous formations, and have temperature measurements. After the criteria was applied, the remaining wells numbered 70. Figure 2.1 provides the well locations in context to the town of Hinton and the regional geography.

![Figure 2.1: Study location shown spatially within the regional geography of the area; Bottom-hole locations of wells are shown as are the corrected bottom-hole temperatures and interpolated temperatures between the well bottoms.](image)

2.2 Resource Assessment

The available energy in geothermal reservoirs is commonly estimated using the USGS volumetric “heat in-place” estimation method [34]. The method and the extent to which it applied in this study is described in the following section. In addition, the values used for each
variable in the assessment along with their rationale are explained in the sections following
the volumetric method description. In general, characterization of the variables is performed
through well log data in conjunction with literature review of comparable projects.

2.2.1 In-place Volumetric Method

The energy stored in the geothermal reservoir is calculated by in Equation 2.1 using mea-
sured or estimated values of the reservoir volume, temperature, and thermal properties.

\[ q_{th} = \rho_c V (T_R - T_r) \] (2.1)

where:

- \( q_{th} \) = Thermal energy stored in reservoir
- \( \rho_c \) = Volumetric heat capacity of fluid-saturated rock \((= \phi \rho_w c_w + (1 - \phi) \rho_r c_r)\)
- \( \rho_w (\rho_r) \) = Density of water (rock)
- \( c_w (c_r) \) = Specific heat of water (rock)
- \( \phi \) = Rock porosity
- \( V \) = Reservoir volume
- \( T_R \) = Reservoir Temperature
- \( T_r \) = Reference Temperature

These reservoir characteristics are typically estimated through geological and geophys-
ical surveys of the area and proven quantities from previous comparable project. Uncertainty
in these parameters can be narrowed down if well log data from nearby resource extraction
drilling is available. If not, latter stages of a geothermal project would include the drilling
of slim test wells to get direct measurements to confirm those estimated by the geophysical
and geochemical methods.

The potential energy recoverable at the well head is calculated by

\[ R = q_{wh} / q_{th} \] (2.2)

where \( q_{wh} \) is the recovered thermal energy, \( q_{th} \) is the total thermal energy, and \( R \) is the
recovery factor, the nature and value of which is described in a latter section.

Next, the enthalpy of the geothermal fluid at the wellhead can be calculated by

\[ h_{wh} = h_R - D g \] (2.3)

where \( h_R \) is the enthalpy of the geothermal fluid at the reservoir temperature, \( D \) is the
average reservoir depth, and \( g \) is gravity.

Mass flow rate at the wellhead can then be determined from the wellhead power by

\[ q_{wh} = \dot{m}_{wh} (h_{wh} - h_r) \] (2.4)

where \( \dot{m}_{wh} \) is the brine flow rate, \( h_{wh} \) is the wellhead enthalpy from Equation 2.3, \( h_r \) is
the enthalpy of the geothermal fluid at the reference temperature.

Following the USGS method would dictate a calculation of the available energy, or exer-
ergy, of the geothermal fluid at this point. A measure of the anticipated power plant exergetic
efficiency, known as the utilization factor, would then be applied to determine the potential for total electricity production. Amortization of the electricity over a project lifetime, typically 30 to 50 years, provides the size of power plant recommended by the method.

These final steps are omitted from the resource estimate for this study. Modelling of a basic power plant and heating system is used in their place to provide a more detailed estimate of exergy efficiency for both options. Therefore, the output of the resource assessment in this case is the annual thermal power at the wellhead and the corresponding geothermal mass flow rate.

2.2.1.1 Monte Carlo Simulation

Due to the uncertainty inherent in many of the required parameters, a Monte Carlo simulation is often used in the application of the heat in-place method. This approach allows the parameters of Equation 2.1 to be represented as a unique distribution (e.g. beta, triangle, uniform) of values across a plausible range. A depiction of the difference between a beta and triangle distribution with the same mean and range is given in Figure 2.2. Each iteration of the simulation employs random number generation to select a value for each parameter relative to its user-defined distribution. The wellhead power is then calculated using the equations defined in the previous section. The simulation must run a statistically significant amount of iterations - typically in the thousands for geothermal applications - and the result is a probability distribution of possible outcomes.

In this study, the Monte Carlo simulation is used to provide the distribution of potential wellhead thermal power. To account for mean differences in temperature and porosity between geologic formations, a simulation is run for each of the four formations that comprise the potential geothermal reservoir. A sample of the simulation input for one formation (Viking) is given in Table 2.1. The simulation for each formation was given the same input parameters except for thickness, reservoir temperature, and porosity which were formation-dependent. The estimated wellhead power at various cumulative probabilities can then be summed across the simulations to arrive at the total reservoir wellhead power. An appropriate cumulative probability is selected and the corresponding wellhead power is used to calculate the mass flow rate at the wellhead. The enthalpy of the geothermal fluid used in the calculation will correspond to an average temperature of the formation, weighted by the formation’s contribution to the total wellhead power.
Table 2.1: Input parameters for Monte Carlo simulation for one of four formations of interest.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Minimum</th>
<th>Most likely</th>
<th>Maximum</th>
<th>Distribution</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reservoir Area</td>
<td>km²</td>
<td>-</td>
<td>702.6</td>
<td>-</td>
<td>Fixed</td>
</tr>
<tr>
<td>Reservoir Thickness</td>
<td>m</td>
<td>-</td>
<td>7.06</td>
<td>-</td>
<td>Fixed</td>
</tr>
<tr>
<td>Reservoir Temperature</td>
<td>°C</td>
<td>100</td>
<td>115</td>
<td>130</td>
<td>Triangular</td>
</tr>
<tr>
<td>Recovery Factor</td>
<td>%</td>
<td>0.0</td>
<td>-</td>
<td>10</td>
<td>Even</td>
</tr>
<tr>
<td>Porosity</td>
<td>%</td>
<td>0.0</td>
<td>2.5</td>
<td>6.5</td>
<td>Beta</td>
</tr>
<tr>
<td>Specific Heat of Rock</td>
<td>kJ/m³ °C</td>
<td>2125</td>
<td>2277.5</td>
<td>2430</td>
<td>Beta</td>
</tr>
<tr>
<td>Ambient Temperature</td>
<td>°C</td>
<td>-</td>
<td>2.4</td>
<td>-</td>
<td>Fixed</td>
</tr>
<tr>
<td>Plant Capacity Factor</td>
<td>%</td>
<td>-</td>
<td>90</td>
<td>-</td>
<td>Fixed</td>
</tr>
<tr>
<td>Project Lifetime</td>
<td>Years</td>
<td>-</td>
<td>50</td>
<td>-</td>
<td>Fixed</td>
</tr>
</tbody>
</table>

2.2.2 Reservoir Volume

The WCSB is one of the largest basins in the world, with individual formations spanning hundreds of kilometers. While discovering water-producing regions in the stratigraphy remains the objective in geothermal exploration of this geology, those production zones are typically less defined than those found in fracture-dominated systems. A large porous basin with non-specific production zones may potentially lead to defining an unrealistically large reservoir volume. For this reason, this study considers the reservoir to be bounded by the wells used in the study.

The Geothermal Energy group used the well logs to generate a geologic model which was then used to calculate the volume of each of the formations of interest encircled by the group of wells. Due to this artificial bounding of the reservoir, there is no volume uncertainty carried through the resource assessment calculations. As shown in Table 2.2, the Mannville sections represent a majority of the total reservoir volume.

Table 2.2: Reservoir volumes of Cretaceous formations within well footprint.

<table>
<thead>
<tr>
<th>Formation</th>
<th>Volume (km³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Viking</td>
<td>4,960</td>
</tr>
<tr>
<td>Upper Mannville</td>
<td>168,456</td>
</tr>
<tr>
<td>Middle Mannville</td>
<td>60,760</td>
</tr>
<tr>
<td>Cadomin</td>
<td>8,240</td>
</tr>
</tbody>
</table>

2.2.3 Reservoir Temperature

The conductive heating mechanism for sedimentary reservoirs results in relatively constant thermal gradients at depth. The temperature of the WCSB increases at a rate of 25-35°C/km in central Alberta. Based on measured bottom-hole temperatures and total vertical depth (TVD), the wells from this study lie at the higher end of the gradient range at an average of 34°C/km. The gradient of each well was used to calculate the temperature at the top of each of the four Cretaceous formations depending on the depth of the well. The sample size of well log measurements for the Viking, Upper Mannville, Middle Mannville, and Cadomin was 69, 69, 60, and 55 respectively.
The temperature data was plotted in separate histograms for each formation. A representative temperature distribution size and shape was then selected for each formation to match its temperature data as closely as possible. The histogram and chosen triangular distribution for one of the formations is shown in Figure 2.3. The selected distributions of the formations are summarized in Table 2.3. Note that the distributions selected for the Viking and Upper Mannville are the same. This is unsurprising as the temperature data was corrected to the top of each formation and the Viking layer is very thin.

![Middle Mannville Temperature Distribution](image)

**Figure 2.3:** Distribution of top-of-formation well log temperatures in the Upper Mannville overlaid with the chosen distribution for Monte Carlo simulation.

**Table 2.3:** Temperature distributions chosen for Monte Carlo simulations for each formation.

<table>
<thead>
<tr>
<th>Formation</th>
<th>Temperature Range (°C)</th>
<th>Shape</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Minimum</td>
<td>Most likely</td>
</tr>
<tr>
<td>Viking</td>
<td>100</td>
<td>115</td>
</tr>
<tr>
<td>Upper Mannville</td>
<td>100</td>
<td>115</td>
</tr>
<tr>
<td>Middle Mannville</td>
<td>106</td>
<td>125</td>
</tr>
<tr>
<td>Cadomin</td>
<td>112</td>
<td>129</td>
</tr>
</tbody>
</table>

**2.2.4 Formation Porosity**

Porosity data from each well - measured from core samples - were averaged over the depth of the formation layers. Average porosities were then compiled for each formation and analyzed in the same manner as the temperature data. Histograms were generated and a distribution size and shape were selected to represent each formation. All distributions included a minimum porosity of 0% with maximum porosities ranging from 6.5% to 10%. A summary of the chosen distributions for the porosity data is given in Table 2.4.

**2.2.5 Volumetric Heat Capacity**

Density and specific heat capacity ranges for the applicable formation rock types at were used to generate a minimum and maximum expected value for volumetric heat capacity. Density and specific heat capacity were taken from the USGS Thermal Properties of Rocks [35]. See Table 2.5 for the values used in the calculation.
Table 2.4: Porosity distributions chosen for Monte Carlo simulations for each formation.

<table>
<thead>
<tr>
<th>Formation</th>
<th>Minimum</th>
<th>Porosity (%)</th>
<th>Maximum</th>
<th>Shape</th>
</tr>
</thead>
<tbody>
<tr>
<td>Viking</td>
<td>0</td>
<td>2.5</td>
<td>6.5</td>
<td>Beta</td>
</tr>
<tr>
<td>Upper Mannville</td>
<td>0</td>
<td>2.5</td>
<td>9.5</td>
<td>Beta</td>
</tr>
<tr>
<td>Middle Mannville</td>
<td>0</td>
<td>2.0</td>
<td>7.0</td>
<td>Beta</td>
</tr>
<tr>
<td>Cadomin</td>
<td>0</td>
<td>3.0</td>
<td>10.0</td>
<td>Triangular</td>
</tr>
</tbody>
</table>

Table 2.5: Volumetric specific heat capacity range calculated using a range of density and mass-specific heat capacity for rock type. [35]

<table>
<thead>
<tr>
<th>Variable</th>
<th>Unit</th>
<th>Minimum</th>
<th>Maximum</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>kg/m³</td>
<td>2500</td>
<td>2700</td>
</tr>
<tr>
<td>Specific Heat Capacity</td>
<td>kJ/kg °C</td>
<td>0.85</td>
<td>0.9</td>
</tr>
<tr>
<td>Volumetric Heat Capacity</td>
<td>kJ/m³ °C</td>
<td>2125</td>
<td>2430</td>
</tr>
</tbody>
</table>

2.2.6 Reference Temperature

The choice of reference, or rejection, temperature has a substantial effect on the calculated thermal power. The reference temperature is commonly chosen to be either the regional ambient temperature or the estimated condenser temperature [36]. Less common is the use of the fluid separation temperature as the reference temperature; this is reserved for high temperature, flash power plant projects. The decision of reference temperature is dependent on the magnitude of regional climate fluctuations, type of power plant planned (flash, binary, etc), and the fluid available for use in the condenser fluid temperature. Additionally, the choice of reference temperature impacts the recovery factor as well as the value for utilization (exergetic) efficiency, if applicable.

As this study involves the modelling of a power plant and heating system, appropriate reference temperatures can be selected for each step, and a utilization factor is not required, as explained previously. The choice of reference temperature in the resource assessment of this study will not greatly affect the outcome, i.e. the estimated geothermal flow rate, as long as the same reference temperature is used for both the estimation of thermal power and the conversion to mass flow rate, namely Equations 2.1 and 2.4.

The basis of this study is the comparison of heating and power system models. It is therefore most appropriate to use the ambient, rather than the condenser, temperature for the reference temperature. Average monthly temperatures from the Hinton area for the past five years (January 2013 to December 2017) were used to determine an annual average ambient temperature of 2.4°C [37].

2.2.7 Availability and Capacity Factor

One measure of a power plant’s performance is how often the plant is down for scheduled or unscheduled maintenance over the course of a year. The percentage of hours that a plant is available for operation (not necessarily operating) over the course of a year relative to the number of hours in a year, is its availability. The reliability of geothermal plants is demonstrated by their high availability, often operating more than 95% of the time [30].
2.2. RESOURCE ASSESSMENT

The amount of power generated by a power plant relative to the maximum possible generation, typically measured on an annual basis, is known as the capacity factor. For power plants, the maximum generation is most often calculated by assuming the nameplate power generation for the entire year, as shown in Equation 2.5. However, others, such as the US Energy Information Administration (EIA), use the maximum generation possible by the plant under summer conditions as the benchmark to evaluate the plant’s performance with available exergy in mind [38]. In either scenario, the capacity factor incorporates the operational availability of a plant as well as its performance relative to ideal ambient conditions.

\[
CF = \frac{MWh_{generated}}{MW_{rated} \times 8760 \text{ h}} \tag{2.5}
\]

The assessment of the reservoir considered for this study is used as the input thermal power for both the power plant and heating models. Therefore, the capacity factor used in the resource assessment will represent only operational availability. The variation in performance of both systems relative to a change in ambient conditions, i.e. summer to winter, is calculated on the system level. The ESMAP Geothermal Handbook (2012) cites typical geothermal availability at 95% or higher, but uses a conservative value of 90% [30]. The resource assessment for this study assumes a capacity factor of 90% for the estimation of thermal power.

2.2.8 Recovery Factor

While there may a substantial amount of energy stored underground in a geothermal reservoir, a fraction of that energy is able to be recovered and utilized at the surface. Due to a myriad of variables, such as permeability structure, flow pathways, maintaining reservoir pressure, and drilling economics, much of the geothermal energy in a reservoir will remain unrecoverable. These variables and their effect on the recoverability of geothermal energy are represented in a multiplier defined as the (thermal) recovery factor (see Equation 2.2).

The recovery factor is both the most influential variable in the assessment and the most difficult to define. Due to the multitude of variables affecting the recoverability and the fact that most geothermal plants operate well beyond their original project lifetime, it is difficult to assess the original recoverability factors and establish a database.

Garg and Combs (2010) warn against using a non-zero minimum recovery rate until geothermal test wells have proven recoverability; they provide a wide range of 0 to 0.20 for potential rates [39]. A USGS report detailing a new assessment of geothermal resources in the US estimates the factor to be between 0.08 and 0.2 for fracture-dominated reservoirs and 0.1 to 0.25 for sediment-hosted reservoirs [34].

The reservoir considered in this study is a sedimentary basin with an extremely large footprint. The most likely scenario of development is the retrofitting of existing oil and gas wells. These wells were drilled into the hydrocarbon-producing zones, specifically avoiding water-producing zones. While many are likely to produce water and wells can potentially be deepened for additional costs, this study considers zero recovery as a potential scenario. The reservoir is relatively low temperature and there is a possibility of extremely muted circulation. For these reasons, a conservative range of 0 to 0.1, with uniform probability, is used for the recovery factor.
2.2.9 Reservoir Fluid

Geothermal fluid, often referred to as brine, is characteristic of local geology and fluid residence time. The fluid deep in the WCSB is relatively immobile and therefore absorbs more minerals than one might find in a geothermal system with high circulation. Accordingly, brine samples from the WCSB have been found to be high in dissolved solids and highly saline. Connolly et. al (1990) performed an extensive analysis on the brines found in the many formations of the WCSB [40]. The chemical analysis of the lower Cretaceous samples provided the basis for the geothermal brine posited for this study. The concentration of \( \text{Na}^+ \) and \( \text{Cl}^- \) ions in each sample, along with the measured density, was used to calculate the mass concentration of the brines. A summary of the ten samples is given in Table 2.6. The physical and thermodynamic properties of a NaCl brine with the average of 6.4% concentration by weight is used in the modelling exercises of this study.

Table 2.6: Salt concentration of brine samples from formations of interest.

<table>
<thead>
<tr>
<th>Stratigraphic Unit</th>
<th>TDS (g/L)</th>
<th>Density (g/L)</th>
<th>Na (g/L)</th>
<th>Cl (g/L)</th>
<th>NaCl (g/L)</th>
<th>Concentration by Mass</th>
</tr>
</thead>
<tbody>
<tr>
<td>Viking 55</td>
<td>1036</td>
<td>20.8</td>
<td>33.3</td>
<td>54.1</td>
<td>6.4%</td>
<td></td>
</tr>
<tr>
<td>Viking 60</td>
<td>1040</td>
<td>22.0</td>
<td>36.1</td>
<td>58.1</td>
<td>6.6%</td>
<td></td>
</tr>
<tr>
<td>Viking 74</td>
<td>1050</td>
<td>25.1</td>
<td>44.7</td>
<td>69.8</td>
<td>6.6%</td>
<td></td>
</tr>
<tr>
<td>Viking 74</td>
<td>1049</td>
<td>25.2</td>
<td>44.5</td>
<td>69.7</td>
<td>6.6%</td>
<td></td>
</tr>
<tr>
<td>Glauconitic 67</td>
<td>1044</td>
<td>24.7</td>
<td>39.7</td>
<td>64.4</td>
<td>6.2%</td>
<td></td>
</tr>
<tr>
<td>Glauconitic 65</td>
<td>1043</td>
<td>24.8</td>
<td>38.9</td>
<td>63.7</td>
<td>6.1%</td>
<td></td>
</tr>
<tr>
<td>Glauconitic 96</td>
<td>1063</td>
<td>31.9</td>
<td>58.3</td>
<td>90.2</td>
<td>8.5%</td>
<td></td>
</tr>
<tr>
<td>Ostracod 72</td>
<td>1048</td>
<td>25.9</td>
<td>42.7</td>
<td>68.6</td>
<td>6.5%</td>
<td></td>
</tr>
<tr>
<td>Ostracod 62</td>
<td>1041</td>
<td>22.6</td>
<td>37.2</td>
<td>59.8</td>
<td>5.7%</td>
<td></td>
</tr>
<tr>
<td><strong>Average</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td><strong>6.4%</strong></td>
<td></td>
</tr>
</tbody>
</table>

2.2.10 Project Lifetime

Geothermal power plants are often financed assuming a 20 to 30 year project lifespan, however, most operate well beyond their economic life [41]. Often safety-critical, heating systems are designed for much longer lifetimes and typically have a backup system. A project lifetime of 50 years is used in the resource assessment for this study.

2.3 Power Scenario

The completion of the resource assessment results in a defined geothermal energy flow which can then be considered for utilization. The first utilization scenario presented in this study is the development of a geothermal binary power plant.

A basic sub-critical organic Rankine cycle (ORC) was modelled in Engineering Equation Solver (EES) to estimate the power generation potential of the geothermal resource in this study. The power cycle processes and corresponding components are described below. A depiction of the power cycle on a temperature-entropy diagram is shown in Figure 2.4 with states corresponding to those used in this study. States 1 through 5 represent the ORC, while states 11 to 13 and 21 to 23 represent the heat source and sink fluids respectively. The fluid states and major power plant components are shown by Figure 2.5.
2.3. POWER SCENARIO

Figure 2.4: ORC geothermal binary power cycle represented on temperature-entropy diagram along with geothermal and cooling fluid states.

Figure 2.5: Process flow and components of geothermal binary power plant model.

2.3.1 Process 1 to 2 - Pump

At the inlet of the pump, the working fluid is assumed to be saturated liquid (quality of 0) with temperature and pressure equal to that of the condenser. The pump acts to increase the pressure to match that of the evaporator. The working fluid therefore exits the pump as a compressed liquid. Given the ORC operating pressures, the enthalpy of state 2 can be determined by

\[ h_2 = h_1 + v_1 \frac{dP}{\eta_{pump}} \]  

where \( h_1 \) is the enthalpy of the saturated liquid working fluid in state 1, \( v_1 \) is the specific volume in state 1, \( dP \) is the difference in evaporator and condenser pressures, and \( \eta_{pump} \) is the isentropic efficiency of the pump. The required pumping power of the feed pump is calculated by

\[ \dot{W}_{pump} = \dot{m}_{wf} \left( h_2 - h_1 \right) / \eta_{motor} \]  

where \( \dot{m}_{wf} \) is the mass flow rate of the working fluid and \( \eta_{motor} \) is the efficiency of the motor driving the pump.
2.3.2 Process 2 to 3 - Preheater

After exiting the pump, the compressed working fluid enters a heat exchanger in which the working fluid gains sensible heat from the geothermal brine. Adequate thermal energy is transferred such that the working fluid is brought up to its saturation temperature and exits the preheater as a saturated liquid (quality of 0).

The difference between the temperatures of the working fluid and the brine at the exit of the preheater is known as the pinch point, given by Equation 2.8. This is the point at which the temperatures of the two fluids are closest to each other.

$$T_{\text{pinch}} = T_{12} - T_3$$  \hspace{1cm} (2.8)

where $T_{12}$ is the temperature of the geothermal fluid entering the preheater and $T_3$ in the temperature of the working fluid at the exit of the preheater.

More heat exchanger surface area will allow for additional heat transfer and therefore a smaller pinch point, assuming other parameters remain the same. There exists then a balance, represented by the pinch point, between the cost per unit area of the heat exchanger and the increased power output of a higher evaporator temperature. The energy balance of the preheater is given by

$$\dot{m}_g c_g (T_{12} - T_{13}) = \dot{m}_{wf} c_{wf} (T_3 - T_2)$$  \hspace{1cm} (2.9)

where $\dot{m}_g$ is the mass flow rate of the geothermal brine, $c_g$ and $c_{wf}$ are the specific heat capacities of the brine and working fluid respectively, $(T_{12} - T_{13})$ is the temperature drop of the geothermal fluid across the preheater, and $(T_3 - T_2)$ is the temperature increase of the working fluid across the preheater.

2.3.3 Process 3 to 4 - Evaporator

Working fluid enters the evaporator as saturated liquid and absorbs latent heat from the geothermal brine. The heat addition is isothermal, with the working fluid being converted from a saturated liquid to a saturated vapour at the exit. Energy transferred in the evaporator is represented by

$$\dot{m}_g c_g (T_{11} - T_{12}) = \dot{m}_{wf} (h_4 - h_3)$$  \hspace{1cm} (2.10)

where $h_3$ and $h_4$ are the enthalpies of the working fluid at the inlet and outlet of the evaporator. The temperature changes associated with the preheater and evaporator are depicted in Figure 2.6.

2.3.4 Process 4 to 5 - Turbine

As all working fluids evaluated in this study are dry fluids (see Section 2.3.9), the model assumes no superheating prior to the turbine. It has been found that system efficiency does not increase with superheating at the turbine inlet [42]. Adiabatic expansion of the vapour through the turbine generates work which is converted to electricity by the generator. The isentropic efficiency of the turbine, $\eta_{\text{turb}}$, is determined from literature and is then used to calculate the outlet enthalpy of the fluid by

$$\eta_{\text{turb}} = \frac{h_4 - h_5}{h_4 - h_{5s}}$$  \hspace{1cm} (2.11)
2.3. POWER SCENARIO

Figure 2.6: Temperature change of geothermal brine and working fluid through the preheater and evaporator in a sub-critical ORC.

where \( h_{5s} \) is the enthalpy of the exit state with isentropic expansion and \( h_5 \) is the actual enthalpy.

2.3.5 Process 5 to 1 - Condenser

The shape of the temperature-entropy vapour saturation curves of dry fluids dictates that the fluid exiting the turbine is superheated vapour. The vapour enters the condenser where a cooling fluid is used to condense the working fluid to its saturated liquid state. Energy balance across an adiabatic, isobaric condenser is provided by

\[
\dot{m}_c c_c (T_{23} - T_{21}) = \dot{m}_{wf} (h_5 - h_1)
\]

where \( \dot{m}_c \) is the mass flow rate of the cooling fluid, \( c_c \) is the specific heat capacity of the cooling fluid, \( (T_{23} - T_{21}) \) is the temperature increase of the cooling fluid across the condenser, and \( (h_5 - h_1) \) is the enthalpy decrease of the working fluid across the condenser.

There are two general categories of condensers, defined by the medium used for cooling: water or air. Water-cooled systems can be once-through, meaning that the water gains heat from the working fluid through a heat exchanger and then exits the system, or they can be set up as a closed loop wherein the exit water is cooled in a cooling tower and then re-enters the heat exchanger. Closed-loop water-cooled systems and air-cooled systems both require fan-driven cooling towers, increasing the parasitic load on the system and decreasing the cycle efficiency [43]. However, very few binary plants have access to the unlimited water supply of a once-through system. The decision between cooling systems is largely determined by regional environmental conditions, such as average and seasonal ambient air temperatures, humidity, and availability of water supply.

Due to the low regional ambient temperature and Alberta’s seasonal demand fluctuations, an air-cooled condenser was selected for the power plant model. As mentioned previously, the average ambient temperature in the Hinton area is 2.4°C, with fluctuations between -10.0°C in the winter and 13.7°C in the summer as shown in Figure 2.7. The northern climate and corresponding low air temperatures year-round are conducive to the use of an air-cooled condenser. In addition, the relatively large seasonal changes in temperature would allow a prospective plant to produce more power in the winter when demand is higher and produce slightly less in the summer months when there is less demand for power [44].
CHAPTER 2. METHODOLOGY

2.3.6 Heat Exchanger Area

Owing to their high surface area to volume ratio, shell and tube exchangers are often the favoured choice for industrial processing applications [45]. Their long history of implementation in various industries offers a dependable and cost-effective solution for a binary geothermal power plant. Another option for the preheater and/or evaporator is a plate-type heat exchanger. Plate exchangers, while much less prevalent in industry, are generally more compact and may offer higher heat transfer rates than those of shell and tube exchangers [46]. The relative ease of maintenance makes plate-type and shell and tube exchangers particularly attractive to geothermal applications. Plates and tubes can be removed and cleaned of deposits, such as carbonate, sulphate, and salts, that are often present in geothermal brines.

The model in this study does not include heat exchanger design, however, heat transfer coefficients based on a shell and tube design are assumed from literature and used to estimate the required heat exchanger area for various power outputs. Area required in the preheater, evaporator, and condenser are calculated using their respective heat flow, heat transfer coefficients, and log mean difference in Equation 2.13. All heat transfer processes are assumed to be isobaric and adiabatic. Table 2.7 lists the heat transfer coefficients and the assumed pressure differential through the heat exchangers.

\[
\dot{Q}_{HX} = A_{HX} U_{HX} \Delta T_{LM,HX}
\]

(2.13)

where

\[
\dot{Q}_{HX} = \text{Rate of thermal energy transfer between fluids}
\]

\[
U_{HX} = \text{Heat transfer coefficient of heat exchanger}
\]

\[
A_{HX} = \text{Required area of heat exchanger}
\]

\[
\Delta T_{LM,HX} = \text{Log mean temperature difference of heat exchanger} \quad (\frac{\Delta T_a - \Delta T_b}{\ln(\frac{\Delta T_a}{\Delta T_b})})
\]

\[
\Delta T_a = \text{Difference in temperature of the two fluid streams on side a of the heat exchanger}
\]

\[
\Delta T_b = \text{Difference in temperature of the two fluid streams on side b}
\]

2.3.7 Power

Gross power output of the cycle is calculated using the enthalpy drop across the turbine, along with the generator efficiency as per
Table 2.7: Overall heat transfer coefficients assumed for heat exchangers.

<table>
<thead>
<tr>
<th>Heat Exchanger</th>
<th>Heat Transfer Coefficient [kW/m² °C]</th>
<th>Source/Sink dP (bar)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Preheater</td>
<td>1</td>
<td>0.5</td>
</tr>
<tr>
<td>Evaporator</td>
<td>1.6</td>
<td>0.5</td>
</tr>
<tr>
<td>Air-cooled Condenser</td>
<td>0.8</td>
<td>0.1 [48]</td>
</tr>
</tbody>
</table>

\[
\dot{W}_{gen} = \dot{m}_{wf} (h_4 - h_5) \times \eta_{gen} \tag{2.14}
\]

Parasitic power of the pumps and condenser fan(s) are given respectively by

\[
\dot{W}_{pump} = \left( \dot{m}_{wf} (h_2 - h_1) + \dot{m}_g v_g (dP_{PH} + dP_{EV}) \right) / \eta_{motor} \tag{2.15}
\]

\[
\dot{W}_{fan} = \frac{\dot{m}_c v_c dP_{fan}}{\eta_{fan} \eta_{motor}} \tag{2.16}
\]

where \( v_c \) and is the specific volume of the cooling fluid (air) entering the condenser, \( v_g \) is the specific volume of the brine entering the evaporator, and \( \eta_{motor} \) is the efficiency of the pump and fan motors.

Net power output of the cycle can then be calculated by

\[
P_{net} = \dot{W}_{gen} - \dot{W}_{pump} - \dot{W}_{fan} \tag{2.17}
\]

To get some measure of the relative cost of the optimizations, specific power is also calculated relative to the heat transfer area. The total area includes the preheater, evaporator, and condenser. Specific power is calculated by

\[
P_{spec} = \frac{P_{net}}{A_{HX}} \tag{2.18}
\]

A range of academic studies of similar binary power generation systems were surveyed to estimate reasonable component efficiencies and heat exchanger pinch points for this study. The findings from this research survey are displayed in Appendix A. The efficiencies along with the other key input variables to the power plant model are presented in Table 2.8.

2.3.8 Optimization

Ideally, the power plant design would be optimized based on a cost model, incorporating estimations of capital costs, O&M costs, and revenue. Due to the current political climate and the lack of industry experience in geothermal power, the uncertainties in a cost model would not provide a particularly useful cost-optimization. As this study is primarily an energy and exergy comparison, the net power was chosen to be the optimization variable. The power plant model was optimized by varying the condenser and evaporator temperatures to obtain a maximum. The variable metric method, incorporated in the EES program, was used for all optimizations [49].
Table 2.8: Input parameters for EES power plant model.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Brine Temperature</td>
<td>118</td>
<td>°C</td>
</tr>
<tr>
<td>Brine NaCl Conc’n</td>
<td>6.4</td>
<td>%</td>
</tr>
<tr>
<td>Ambient Temperature</td>
<td>2.4</td>
<td>°C</td>
</tr>
<tr>
<td>Evaporator Pinch</td>
<td>7.5</td>
<td>°C</td>
</tr>
<tr>
<td>Condenser Pinch</td>
<td>5</td>
<td>°C</td>
</tr>
<tr>
<td>Turbine Efficiency</td>
<td>0.85</td>
<td></td>
</tr>
<tr>
<td>Pump Efficiency</td>
<td>0.80</td>
<td></td>
</tr>
<tr>
<td>Fan Efficiency</td>
<td>0.70</td>
<td></td>
</tr>
<tr>
<td>Motor Efficiency</td>
<td>0.95</td>
<td></td>
</tr>
<tr>
<td>Generator Efficiency</td>
<td>0.96</td>
<td></td>
</tr>
<tr>
<td>Fan dP</td>
<td>0.1</td>
<td>bar</td>
</tr>
<tr>
<td>Preheater dP</td>
<td>0.5</td>
<td>bar</td>
</tr>
<tr>
<td>Evaporator dP</td>
<td>0.5</td>
<td>bar</td>
</tr>
<tr>
<td>Piping Network dP</td>
<td>1.0</td>
<td>bar</td>
</tr>
</tbody>
</table>

2.3.9 Working Fluids

Flash steam geothermal plants, along with fossil fuel plants, utilize water as the working fluid in the power generation cycle. Its high critical temperature and pressure allows power plants of this type to generate massive amounts of power while still operating on sub-critical Rankine cycles. Binary geothermal plants operate with much lower resource temperatures and therefore must employ organic fluids, which vaporize and condense at lower temperature points, in their power cycles. Selection of the right working fluid for an ORC requires consideration of physical and thermodynamic properties, environmental factors, and health and safety hazards.

Working fluids are classified as wet, dry, or isentropic depending on their temperature-entropy (T-s) vapor saturation curves, as shown in Figure 2.8. Wet fluids, such as water or R22, have a negative sloping saturation curve (dT/ds < 0), dry fluids, such as pentane or R245fa, have positive sloping curves (dT/ds > 0), and isentropic fluids, such as R11, have near-vertical vapor saturation curves. Dry and isentropic fluids are then capable of entering the turbine in saturated vapor state and are in no danger of condensation during expansion, which can lead to damage of turbine components, unlike wet fluids which require some degree of superheating to ensure turbine integrity. This study follows the vast majority of academic studies and real-world binary power plants and limits its working fluid evaluation to dry and isentropic fluids.

Density of the working fluid is a crucial component of working fluid selection for a proposed plant. The specific volume, along with the flow rate, has a direct impact on the sizing of the turbine, condenser, and largest piping in the plant (turbine outlet to condenser inlet). This is an important practical consideration as turbine size is, at first approximation, relative to its cost [51]. This factor is addressed upon determination of the optimized operating pressures of each selected fluid.

Organic fluids historically used in binary power plants can be categorized by their chemical composition as chlorofluorocarbons (CFCs), hydrochlorofluorocarbons (HCFCs), hydroflurocarbons (HFCs), and hydrocarbons (HCs). Due to their chlorine content, CFCs and HCFCs released into the atmosphere cause damage to the Earth’s ozone layer. For this reason, the international community agreed, through the Montreal Protocol, to ban the use of
CFCs by 1996 [52]. The treaty also called for a phase-out of HCFCs by the year 2030. HFCs were used commonly as a substitute for HCFC systems as they have zero ozone depletion potential (ODP). However, increasing climate change concerns have led international community to initiate a reduction in HFCs due to their high global warming potential (GWP). The Kigali Amendment (2016) to the Montreal Protocol dictates a progressive reduction in HFCs beginning in 2019 for developed countries with an aim of 85% reduction by 2030 [52].

In addition to potential environmental dangers, the health and safety concerns of working fluids must be evaluated. The American Society of Heating, Refrigeration and Air-Conditioning Engineers (ASHRAE) have issued standards relating to the hazards, namely toxicity and flammability, of refrigerants. Table 2.9 provides a summary of the classification characterized in ANSI/ASHRAE 34-2016 [53].

<table>
<thead>
<tr>
<th>Toxicity</th>
<th>Flammability</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low</td>
<td>A1</td>
</tr>
<tr>
<td>High</td>
<td>B1</td>
</tr>
</tbody>
</table>

Commonly used working fluids are listed in Table 2.10. As the modelling program EES is used for this study, the working fluid selection is further limited by the program’s thermodynamic library. Therefore, novel, inert fluids, such as PF5050, were not evaluated in the working fluid selection [46]. The considerations described in this section lead to a selection of five working fluids for this study: butane (R600), isobutane (R600a), pentane (R601), isopentane (R601a), and R245fa.

### 2.4 Heating Scenario

As mentioned in the introduction, forced-air furnaces make up the vast majority of the space heating infrastructure in Alberta, with natural gas as their fuel source. The heating scenario consists of a district heating design that provides heating to as many homes as possible. The design is built upon statistics regarding household energy use in Alberta as it relates
### Table 2.10: Physical and safety-related characteristics of common ORC working fluids.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>R600</td>
<td>Dry</td>
<td>58.12</td>
<td>A3</td>
<td>0</td>
<td>4</td>
</tr>
<tr>
<td>R600a</td>
<td>Dry</td>
<td>58.12</td>
<td>A3</td>
<td>0</td>
<td>3</td>
</tr>
<tr>
<td>R601</td>
<td>Dry</td>
<td>72.15</td>
<td>A3</td>
<td>0</td>
<td>~4</td>
</tr>
<tr>
<td>R601a</td>
<td>Dry</td>
<td>72.15</td>
<td>A3</td>
<td>0</td>
<td>~4</td>
</tr>
<tr>
<td>R11</td>
<td>Isentropic</td>
<td>134.7</td>
<td>A1</td>
<td>0.05</td>
<td>4660</td>
</tr>
<tr>
<td>R22</td>
<td>Wet</td>
<td>86.47</td>
<td>A1</td>
<td>0.05</td>
<td>1760</td>
</tr>
<tr>
<td>R123</td>
<td>Dry</td>
<td>152.93</td>
<td>B1</td>
<td>0.012</td>
<td>79</td>
</tr>
<tr>
<td>R134a</td>
<td>Wet</td>
<td>102.03</td>
<td>A1</td>
<td>0</td>
<td>1300</td>
</tr>
<tr>
<td>R141b</td>
<td>Isentropic</td>
<td>116.95</td>
<td>A2</td>
<td>0.12</td>
<td>782</td>
</tr>
<tr>
<td>R142b</td>
<td>Isentropic</td>
<td>100.49</td>
<td>A2</td>
<td>0.07</td>
<td>1980</td>
</tr>
<tr>
<td>R245fa</td>
<td>Dry</td>
<td>134.05</td>
<td>B1</td>
<td>0</td>
<td>858</td>
</tr>
</tbody>
</table>

to heating and regional climate patterns, as well as design parameters from district heating systems around the world.

Geothermal district heating systems can be separated into two categories: flow-through and closed loop. Flow-through systems are those in which the geothermal fluid is pumped from the well directly to residential radiators and/or hot water distribution networks. Some flow-through systems require mixing with radiator return water or cold groundwater to ensure the district hot water is cool enough (< 90°C) for safe direct use [56]. If the geothermal fluid chemistry is unsuitable for direct use, a closed loop system is used. In a closed-loop system, heat is transferred from the geothermal fluid to a secondary fluid that delivers energy through the distribution network [56]. The secondary fluid is sanitary water in the case of the hot water system, and usually water or a water-glycol mixture for radiator supply fluid.

#### 2.4.1 Design

A closed-loop central district heating network is assumed in this study. Due to the high dissolved solids and saline content of the expected geothermal fluid, a flow-through system is not desirable. A closed-loop system will ensure that any fouling is contained to the central heat exchanger, therefore allowing for simpler, more dependable design of consumer radiators. Typically, heating systems located in cold climates use a water-glycol (propylene or ethylene) solution as the working or radiator fluid in order to protect against pipes freezing and/or bursting during scheduled and unscheduled outages. However, literature review revealed no empirical relationships for calculating entropy values of a glycol-water solution - a value required for exergy calculations. Therefore, the radiator fluid in this study is assumed to be water.

The design of a district heating network involves a series of energy balances to ensure supply matches network demand. Energy lost by residences depends on the overall effectiveness of their exterior construction. This lost energy must be replaced by thermal energy transferred from the radiator to the living spaces, which is a function of the room temperature, radiator fluid temperature, and radiator design. Thermal energy gained by the interior spaces is equivalent to the energy lost by the working fluid in the heating network. Energy flow of the working fluid must be supplied by geothermal fluid in the central heat exchanger. Finally, the heat transfer between fluids is a function of the central heat exchanger design.
and the inlet and outlet temperatures of the fluids. The energy balance sequence is represented by Equation 2.19, and a basic schematic of the heating system, with numbered states corresponding to this study, is depicted in Figure 2.9.

\[
\dot{Q}_{\text{Loss}} = \dot{Q}_{\text{Radiator}} = \dot{Q}_{\text{Rad Fluid}} = \dot{Q}_{\text{Geo Fluid}} = \dot{Q}_{\text{HX}}
\]  

(2.19)

Figure 2.9: Diagram of closed-loop geothermal district heating system.

Individual household energy demand for the proposed district network is determined based on the climate and energy consumption data available for the province of Alberta. While there exists ample data on the insulation values required for household construction in Alberta (exterior walls, roofs, basements, etc.), there are no equivalent data regarding the evaluation of the overall heat transfer coefficient for a house a whole. Using data for the energy consumption of households in Alberta, along with temperature data from the study region, a value for the overall household heat transfer coefficient can be determined by

\[
\dot{Q}_{\text{Loss},i} = K_L (T_{\text{RM}} - T_{\text{amb},i})
\]  

(2.20)

Where \( \dot{Q}_{\text{Loss},i} \) is the thermal demand of a single home on a specific day, \( T_{\text{amb},i} \) is the average ambient temperature on that day, \( T_{\text{RM}} \) is the interior room temperature, and \( K_L \) is the overall household heat transfer coefficient in units of kW/°C. The energy lost to the environment is equated to the amount of thermal energy provided by the combustion of natural gas in furnaces of existing households. Given average household consumption of natural gas, proportion of that which provides space heating, and an average furnace efficiency, thermal energy lost by the average Albertan household was determined to be 82.6 GJ (see Table 1.3). Average daily temperatures near Hinton were then used to calculate the daily temperature difference with \( T_R = 20^\circ \text{C} \). Energy lost per day can then be determined and the summation over a year results in

\[
\sum_{i=1}^{365} (\dot{Q}_{\text{Loss},i} \times 86400 \text{ [s/day]}) = K_L \sum_{i=1}^{365} ((T_R - T_i) \times 86400 \text{ [s/day]})
\]

(2.21)

\[
82.6 \text{ [GJ/year]} = 86400 K_L \sum_{i=1}^{365} (T_R - T_i)
\]

The overall heat transfer coefficient calculation was performed using average daily temperatures for the five most recent years and summarized in Table 2.11. Demand of the district heating network will be based on the average household heat transfer coefficient of 0.165 kW/°C.
Table 2.11: Overall household heat transfer coefficients calculated based on average household consumption and regional daily temperatures.

<table>
<thead>
<tr>
<th>Year</th>
<th>K_L Value (kW/°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2013</td>
<td>0.166</td>
</tr>
<tr>
<td>2014</td>
<td>0.159</td>
</tr>
<tr>
<td>2015</td>
<td>0.173</td>
</tr>
<tr>
<td>2016</td>
<td>0.170</td>
</tr>
<tr>
<td>2017</td>
<td>0.159</td>
</tr>
<tr>
<td>Average</td>
<td>0.165</td>
</tr>
</tbody>
</table>

The rate of heat loss, as a function of ambient temperature, must be matched by the heat supplied by a household’s radiator(s), given by

\[ \dot{Q}_{\text{Loss}} = \dot{Q}_{rad} = U_{rad} A_{rad} \Delta T_{LM,rad} \]  \hspace{1cm} (2.22)

where:

- \( U_{rad} \) = Heat transfer coefficient of household radiators
- \( A_{rad} \) = Heat transfer area of household radiators
- \( \Delta T_{LM,rad} \) = Log mean temperature difference across the radiator

\[ \Delta T_{LM,rad} = \log \frac{T_s - T_{ret}}{T_{ret} - T_{RM}} \]

- \( T_s \) = Supply temperature of radiator fluid
- \( T_{ret} \) = Return temperature of radiator fluid
- \( T_{RM} \) = Room temperature of houses (Set to 20°C)

The supply and return temperatures correspond to \( T_2 \) and \( T_3 \) in Figure 2.9. Heat transfer coefficients were calculated for a popular line of household convection radiators and their average value of 5.6 W/m²°C was used in the radiator energy calculations [57].

To determine a suitable radiator design scenario and its relative supply and return temperatures, design methodology from Iceland was investigated. Geothermal energy in Iceland accounts for over 90% of household space heating [5]. Systems in Iceland are designed to provide the majority of base-load heating, down to an ambient temperature of -15°C. The design scenario is described, in terms of the household radiator temperatures (°C), as 80/40/-15/20 (Radiator supply / return / ambient / room temperatures) [56]. This study takes a similar approach to the design scenario, but accounts for a slightly lower ambient temperature due to the colder Canadian prairie winters. The flow rates and heat exchanger sizes are designed to a standard of 80/40/-20/20 for this study. Using the supply and return design temperatures of 80°C and 40°C respectively, the mass flow rate of the radiator fluid (subscript "rf") can be calculated by

\[ \dot{Q}_{rad} = \dot{Q}_{rf} = \dot{m}_{rf} c_{p_{rf}} (T_s - T_{ret}) \]  \hspace{1cm} (2.23)

where \( \dot{m}_{rf} \) is the mass flow rate of the radiator fluid and \( c_{p_{rf}} \) is its specific heat capacity. Note that all specific heat capacities in this analysis are calculated at the midpoint between the two temperatures in their respective energy flow equations.
2.4. HEATING SCENARIO

Though piping is assumed to be installed underground with adequate insulation, as is common in colder climates, there will still be heat losses from the radiator fluid to the ground surrounding the pipelines. Karlsson (1982) puts forth an estimate of 5-10% losses in heating networks, concurrent with the 5.9% losses calculated for the Afyon heating system in Turkey [58] [59]. Losses in the system theorized in this study are expected to be on the lower side due to extremely high flow rates in the main lines of the system. A thermal energy loss of 5% is estimated for the system, with the supply side incurring slightly higher losses (3%) due to its higher temperature. These losses are added to the radiator supply and return energy flows and used to determine the inlet and exit temperatures of the radiator fluid in the central heat exchanger ($T_4$ and $T_1$ in Figure 2.9), while maintaining supply and return temperatures ($T_2$ and $T_3$ in Figure 2.9) of 80°C and 40°C at the household radiators.

Once the inlet and outlet temperatures of the radiator fluid are known, an energy balance can be performed between the radiator fluid and the geothermal brine, as shown in

\[ \dot{Q}_{HX} = \dot{m}_{rf} c_{pf} (T_1 - T_4) \]  

\[ \dot{Q}_{HX} = \dot{m}_g c_{pg} (T_{11} - T_{12}) \]  

The energy flow and fluid temperatures in Equations 2.24 and 2.25 can then be used to determine the area required in the central heat exchanger to allow for the transmission of this energy flow. To encourage comparisons between the heating and power scenarios, the heat transfer coefficient is taken to be 1 kW/m²°C – the same value used for the preheater in the power plant model. The heat transfer across the central heat exchanger is given by

\[ \dot{Q}_{HX} = U_{HX} A_{HX} \Delta T_{LM,HX} \]  

where:

$U_{HX}$ = Heat transfer coefficient of the central heat exchanger

$A_{HX}$ = Heat transfer area of the heat exchanger

$\Delta T_{LM,HX}$ = Log mean temperature difference of heat exchanger ($= \frac{\Delta T_a - \Delta T_b}{\ln(\frac{\Delta T_a}{\Delta T_b})}$)

$\Delta T_a$ = Difference in temperature of the two fluid streams on side a of the heat exchanger ($= T_{11} - T_1$)

$\Delta T_b$ = Difference in temperature of the two fluid streams on side b ($= T_{12} - T_4$)

A second parameter, in addition to the input geothermal energy, must be fixed in order to finalize the heating system design. To facilitate the scenario comparison, the temperature drop of the geothermal fluid in the heating system is set to equal the temperature drop in the power scenario, such that the energy provided by the geothermal fluid is the same for both scenarios.

Finally, as the electricity consumed by the working fluid pumps of a large heating system can be significant, its affects on the overall exergy balance should be considered. For this study, the pumping power consumption is assumed to be equal to or less than the power consumed by household furnaces. The predicted annual electricity consumption of the most efficient furnaces with an adequate annual thermal output (82.6 GJ) were gathered and an average power consumption was determined to be 43 W [60]. With the introduction of a
district heating network, this electricity would be “freed” and available to pump the radiator fluid through the system without drastically changing the exergy balance. That is to say the heating system would occur no exergy deficits as long as the power required to pump the radiator fluid was less than the cumulative power consumption of the absent furnaces. The freed power is denoted as power available for pumping and is calculated by multiplying 43 W by N, the number of households served by the heating network. As the pumping power is a linear function of number of households, just as the radiator mass flow rate is, we can determine the maximum pressure drop that can be overcome with the available power for pumping by

\[ \dot{W}_\text{Pump,rf} = \dot{m}_{rf} v_{rf} \Delta P_{rf} \]  

(2.27)

where \( \dot{W}_\text{Pump,rf} \) is the power available ("freed") for pumping the radiator fluid, \( \dot{m}_{rf} \) is the mass flow rate, \( v_{rf} \) is the radiator fluid specific volume, and \( \Delta P_{rf} \) is maximum pressure drop allowable in the radiator fluid system. Just as the \( \dot{m}_{rf} \) per household can be determined from Equations 2.20, 2.22, and 2.23, so too can the pressure drop. Maximum differential pressure in the design case, before the scenario experiences an exergy deficit due to net power, is determined to be 9.15 bar or 915 kPa. This analysis serves in place of undetermined detailed pressure loss calculations of thousands of metres of piping in a district heating network. Upon a future full-scale design of a heating network, the pressure loss and subsequent pumping power would be calculated and compared to the "freed" electricity. If the pressure drop exceeds 9.15 bar, the additional pumping power would need to be factored into the utilization comparison.

2.5 Energy and Exergy Analysis

Evaluations of power plants, or any energy transfer system for that matter, are based on the first and second laws of thermodynamics. The first law states that energy cannot be destroyed, only converted from one state to another. Its working equation is

\[ \dot{Q} - \dot{W} = - \sum_{i=1}^{n} m_i (h_i + \frac{v_i^2}{2} + gz_i) \]  

(2.28)

where \( \dot{Q} \) is the energy flow across the system boundary, \( \dot{W} \) is the rate of work done by the system, \( m_i \) is a mass flow into or out of the system, \( h_i \) is the enthalpy of the respective mass flow, \( v_i \) is its respective velocity, and \( z_i \) is the elevation of each inlet or outlet relative to a chosen reference point.

In analyses of power plants, velocity and gravity effects are often assumed to be negligible. An energy balance of a system with one inlet and one outlet then becomes

\[ \dot{Q} - \dot{W} = \dot{m} (h_2 - h_1) \]  

(2.29)

where \( h_1 \) and \( h_2 \) are the enthalpy of the mass flow at the inlet and outlet respectively.

First law, or thermal, efficiency of a power plant is therefore determined by the ratio of the usable output energy to the energy input. For a binary geothermal plant, this is the ratio of the net power output to the energy extracted from the geothermal fluid in the heat exchangers. For the model developed in this study, the equation is as follows

\[ \eta^I = \frac{P_{\text{net}}}{Q_{\text{in}}} = \frac{\dot{W}_{\text{gen}} - \dot{W}_{\text{pump}} - \dot{W}_{\text{fan}}}{m_{\text{geo}} (h_{11} - h_{13})} \]  

(2.30)
where \( W_{\text{gen}} \) is the gross power output of the turbine-generator, \( W_{\text{pump}} \) and \( W_{\text{fan}} \) are the parasitic power loads, \( \dot{m}_{\text{geo}} \) is the mass flow rate of the geothermal brine, and \( h_{11} \) and \( h_{13} \) are the inlet and outlet enthalpies of the brine.

While the first law of thermodynamics is concerned primarily with quantities of energy, the second law allows one to evaluate energy quality. The second law states that the entropy of a system must remain constant or increase over time. The Carnot heat engine is one such theoretical thermodynamic cycle in which entropy remains constant. Its thermal efficiency therefore represents the maximum efficiency attainable of any engine operating between the same heat source and sink. Due to their relatively low-temperature heat sources, binary geothermal power plants have characteristically low Carnot efficiencies. This reality demonstrates the difficulty of using thermal efficiency as the sole performance metric of a plant. The Carnot efficiency is given by

\[
\eta_C = 1 - \frac{T_L}{T_H} \tag{2.31}
\]

where \( T_H \) and \( T_L \) are the absolute temperatures (K) of the heat source and sink respectively. As shown in Table 2.12, even in the colder ambient temperatures of the winter months, the maximum Carnot thermal efficiency of the binary plant is less than 33%.

Table 2.12: Carnot efficiency of prospective power plant in study area with 118°C geothermal brine.

<table>
<thead>
<tr>
<th>Ambient Temperature (°C)</th>
<th>( \eta_C^J )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Annual Average</td>
<td>2.4</td>
</tr>
<tr>
<td>Summer</td>
<td>13.7</td>
</tr>
<tr>
<td>Winter</td>
<td>-10</td>
</tr>
</tbody>
</table>

Continuing with the second law analysis, the steady-state working equation of an isolated system with one inlet and one outlet is given by

\[
\frac{\dot{Q}_0}{T_0} = \dot{m} \left( s_2 - s_1 \right) \tag{2.32}
\]

where \( \dot{Q}_0 \) is the heat transfer from the dead state, \( T_0 \) is the temperature of the dead state, and \( s_1 \) and \( s_2 \) are the entropy values of the mass flow rate at the inlet and outlet.

The concept of exergy can be arrived at by combining the first and second laws. Exergy is described as the maximum amount of useful work a system can perform relative to its environment. Matter and energy can perform work until its thermodynamic quantities are equivalent to that of its surroundings. This means that two systems of the same energy content can have different exergy values if their respective surroundings are of different thermodynamic states. Exergy is therefore often a better quantity to compare the performance of multiple plants as it takes their relative environments into account. The surroundings are represented as a reference, or dead, state, denoted by state 0. By combining Equation 2.29 and Equation 2.32, values for exergy and mass-specific exergy can be derived and are expressed respectively by

\[
\dot{E}_1 = \dot{m}_1 \left( h_1 - h_0 - T_0 \left( s_1 - s_0 \right) \right) \tag{2.33}
\]
Two methods of calculating exergetic efficiency for a component are detailed by DiPippo (2012). The so-called brute-force efficiency is defined as the ratio of the output exergy to the input exergy [51]. The more commonly-used method of calculating exergy efficiency is the ratio of desired output exergy to the required input exergy, known as the functional efficiency [51].

Calculation of the overall plant exergy efficiency can be performed in two ways, depending on whether the geothermal fluid is considered of any use once it exits the plant. If the fluid is planned to be used for a secondary application, the functional efficiency is calculated by the exergy difference between the input and output streams of the brine. The outgoing fluid stream may also be considered useful if the reservoir is characterized as an underground heat exchanger, and the remaining exergy is transferred back to the reservoir. However, if the fluid is considered to be of no use once it exits the plant, then the entirety of the input exergy stream is used in the ratio, denoted as the overall system efficiency. Both methods are calculated in this study, and are denoted as overall and functional as per

\[
\eta_{\text{overall}}^I = \frac{P_{\text{net}}}{E_{\text{in}}} \\
\eta_{\text{func}}^I = \frac{P_{\text{net}}}{E_{\text{in}} - E_{\text{out}}}
\]

\[ \text{(2.35)} \]

\[ \text{(2.36)} \]

### 2.6 Non-Renewable Energy Displacement

The heating and power scenarios are judged against each other largely in comparison of their ability to displace existing fossil fuel consumption. Models for each scenario provide the calculation of their respective annual and lifetime energy outputs, then the equivalent non-renewable energy is calculated by using the efficiencies characteristic of the equivalent fossil fuel systems.

As explained in the introduction, both Alberta and Canada have pledged to phase-out power generation by coal in the near future. Therefore, power generation by natural gas would be the likely fossil fuel displaced by potential future geothermal power generation. Combined-cycle natural gas plants operate at significantly higher efficiencies (60%) as compared to single-cycle plants (35% - 42%) [61]. The four largest natural gas power plants in Alberta utilize combined cycle gas turbines as will any future natural gas plants [62] [63] [64] [65]. Natural gas is assumed to have a power generation efficiency of 60%.

Heating for the vast majority of households in Alberta is provided by forced-air natural gas furnaces. As described previously, all furnaces installed after 2009 were required to have a thermal efficiency of 90%. Natural gas efficiency with regards to providing space heating is therefore considered to be 90%.

Non-renewable energy savings of the systems are calculated in Joules and also equivalent volume of natural gas. Natural Resources Canada lists natural gas with an energy content of 37.3 MJ/m$^3$ [66]. Note that both power and heating efficiencies are end-user efficiencies and do not include the energy used in the current extraction and transportation of natural gas for use in space heating and power generation. These additional inefficiencies are assumed to be equivalent across the scenarios.
Chapter 3

Results

3.1 Resource Assessment

The volumetric assessment was calculated by performing a Monte Carlo simulation for each of the four stratigraphic formations to more accurately capture their geothermal potential relative to their individual temperature and porosity distributions. Each simulation was comprised of 10,000 iterations and produced the estimated thermal power available from each formation at cumulative probabilities from 0% to 100%.

The distribution of thermal power results from the simulation for the Viking formation is shown in Figure 3.1. The simulation does not produce results in an expected normal distribution, but rather a relatively flat, or even, distribution. This results from many of the input variables being fixed or having an even distribution, such as reservoir volume and recovery factor. As the reservoir-dependent variables – temperature, porosity – have less of an impact on the calculation of thermal power than many of the independent variables, the flat distribution shape is common through the simulations for all four formations.

Of the variables that were represented by a distribution rather than a fixed value, the recovery factor is by far the most impactful on the resulting thermal power, as shown by the tornado chart in Figure 3.2. This is further demonstrated by performing the simulation a second time, increasing the minimum recovery factor from 0% to 2.5% - a reasonable assumption given the well log data and the sedimentary nature of the reservoir. The simulation yielded thermal power increases from 226 MW to 1318 MW for 95% cumulative probability and 455 MW to 1485 MW for 90% probability.

A resource estimate for the entire Lower Cretaceous system is achieved by combining all four Monte Carlo simulations. The thermal power estimates are added together at each cumulative probability interval resulting in the distribution in Table 3.1 shown below. As expected, the flat simulation distributions lead to a substantial spread of 226 MW to 4394 MW for 95% to 5% cumulative probability for the thermal power production of the prospective reservoir.

A conservative cumulative probability of 95% is used for this study. Along with the conservative choices for recovery factor, capacity factor, and project lifetime, the thermal power of 226 MW represents the absolute minimum estimated production should this reservoir volume be developed.

Using the equation for wellhead thermal power (Equation 2.4), the estimated thermal power can then be converted into a geothermal brine flow rate. The values used for temperature and depth were generated by averaging the formation mean values, weighted by their proportional contribution to thermal power in the Monte Carlo simulation. The enthalpies
CHAPTER 3. RESULTS

Figure 3.1: Distribution of thermal power outcomes from Monte Carlo simulation of Viking formation.

Figure 3.2: Sensitivity analysis (tornado chart) of variables in simulation of Viking formation.

Table 3.1: Results of thermal power Monte Carlo simulation for all formations.

<table>
<thead>
<tr>
<th>Cumulative Probability</th>
<th>Viking</th>
<th>Upper Mannville</th>
<th>Middle Mannville</th>
<th>Cadomin</th>
<th>Combined</th>
</tr>
</thead>
<tbody>
<tr>
<td>100%</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>95%</td>
<td>4</td>
<td>155</td>
<td>58</td>
<td>8</td>
<td>226</td>
</tr>
<tr>
<td>90%</td>
<td>9</td>
<td>309</td>
<td>121</td>
<td>17</td>
<td>455</td>
</tr>
<tr>
<td>75%</td>
<td>22</td>
<td>786</td>
<td>302</td>
<td>43</td>
<td>1153</td>
</tr>
<tr>
<td>50%</td>
<td>45</td>
<td>1558</td>
<td>606</td>
<td>86</td>
<td>2295</td>
</tr>
<tr>
<td>25%</td>
<td>69</td>
<td>2329</td>
<td>904</td>
<td>129</td>
<td>3431</td>
</tr>
<tr>
<td>10%</td>
<td>83</td>
<td>2802</td>
<td>1084</td>
<td>155</td>
<td>4123</td>
</tr>
<tr>
<td>5%</td>
<td>88</td>
<td>2984</td>
<td>1158</td>
<td>164</td>
<td>4394</td>
</tr>
<tr>
<td>0%</td>
<td>106</td>
<td>3619</td>
<td>1408</td>
<td>196</td>
<td>5329</td>
</tr>
</tbody>
</table>

used in the calculation assumed a geothermal brine of 6.4% NaCl concentration as explained previously. The resulting wellhead flow rate of 118°C brine was determined to be 540 kg/s, or 2.4 kg/s per MW.
3.2 Power Scenario

3.2.1 Model Selection

The EES power plant model was optimized to maximize net power output for all five working fluids. Condenser and evaporator temperatures were identified as the independent variables and adjusted by the optimization until a global maximum within the constraints was found. Key parameters from the optimization results are listed in Table 3.2.

Table 3.2: System parameters of optimized power plant model for selected working fluids.

<table>
<thead>
<tr>
<th>Working Fluid</th>
<th>Condenser Temperature (°C)</th>
<th>Evaporator Temperature (°C)</th>
<th>Flow Rate (kg/s)</th>
<th>Gross Power (kW)</th>
<th>Net Power (kW)</th>
<th>ηI</th>
</tr>
</thead>
<tbody>
<tr>
<td>Isopentane</td>
<td>14.0</td>
<td>66.7</td>
<td>302</td>
<td>14343</td>
<td>11988</td>
<td>9.17%</td>
</tr>
<tr>
<td>n-pentane</td>
<td>14.0</td>
<td>66.4</td>
<td>284</td>
<td>14137</td>
<td>11845</td>
<td>9.15%</td>
</tr>
<tr>
<td>Isobutane</td>
<td>14.0</td>
<td>68.2</td>
<td>332</td>
<td>15240</td>
<td>12360</td>
<td>9.10%</td>
</tr>
<tr>
<td>n-butane</td>
<td>14.0</td>
<td>67.3</td>
<td>296</td>
<td>14731</td>
<td>12130</td>
<td>9.20%</td>
</tr>
<tr>
<td>R245fa</td>
<td>14.0</td>
<td>67.5</td>
<td>565</td>
<td>14711</td>
<td>12237</td>
<td>9.22%</td>
</tr>
</tbody>
</table>

It is shown that, with the given model assumptions, the working fluids do not differ greatly on performance for an optimized power plant between the provided heat source and sink. All five fluids have roughly the same first law efficiency and produce power within a ± 4.3% band from the median of 12.1 MW.

While overall plant performance is similar from fluid to fluid, there are some significant differences between the cycles that play a deciding role in selecting a working fluid for further design. These factors, namely heat exchanger area, operating pressures, and mass flow rates, are those which have a substantial effect on the capital cost, O&M costs, and operational logistics of a prospective plant. This study does not include a specific component-level cost estimate; however, general cost considerations can be factored into the choice of working fluid.

Heat exchangers are often the most expensive components of a plant, with their cost relative to their surface area. Preference is given, therefore, to fluids requiring less total heat exchanger area per unit power output, i.e. specific power.

Cost criteria related to piping size is also taken into account. The largest piping in the system is required from the turbine exit to the condenser inlet and is determined by the volumetric flow rate of the working fluid at the condenser pressure. Required piping size is calculated as a ratio with respect to the diameter of the smallest volumetric flow rate of the fluids. Again, preference is given to the smaller diameter piping as it represents a lower overall plant piping cost.

Finally, the logistics inherent in the operating pressures of each working fluid cycle is considered. Higher pressures have some effect on piping and component cost, however, a larger concern is that of pressures lower than atmospheric. Plants operating with a condenser pressure lower than atmospheric need to incur additional capital costs, and potentially O&M costs, due to the inclusion of a condenser system capable of handling potential air in-leakage. Therefore, preference was given to fluids with lower evaporator pressures while maintaining a condenser pressure above atmospheric.

These considerations and the resulting working fluid ranking is provided in Table 3.3. The power plant model using n-butane is chosen for further analysis.
Table 3.3: Secondary evaluation parameters for optimized power plants.

<table>
<thead>
<tr>
<th>Working Fluid</th>
<th>Condenser Pressure (bar)</th>
<th>Evaporator Pressure (bar)</th>
<th>Specific Power (kW/m³)</th>
<th>Volumetric Flow Rate (m³/s)</th>
<th>Piping Diameter Ratio</th>
<th>Final Ranking</th>
</tr>
</thead>
<tbody>
<tr>
<td>Isopentane</td>
<td>0.61</td>
<td>3.26</td>
<td>0.625</td>
<td>169.6</td>
<td>1.79</td>
<td>4</td>
</tr>
<tr>
<td>n-pentane</td>
<td>0.45</td>
<td>2.57</td>
<td>0.625</td>
<td>216.8</td>
<td>2.02</td>
<td>5</td>
</tr>
<tr>
<td>Isobutane</td>
<td>2.51</td>
<td>10.45</td>
<td>0.610</td>
<td>52.9</td>
<td>1.00</td>
<td>2</td>
</tr>
<tr>
<td>n-butane</td>
<td>1.71</td>
<td>7.61</td>
<td>0.622</td>
<td>70.8</td>
<td>1.16</td>
<td>1</td>
</tr>
<tr>
<td>R245fa</td>
<td>0.97</td>
<td>5.70</td>
<td>0.622</td>
<td>104.5</td>
<td>1.41</td>
<td>3</td>
</tr>
</tbody>
</table>

3.2.2 n-Butane Model

A visual representation of the net power optimization of the n-butane model, using the condenser and evaporator temperatures as the independent variables, is depicted in Figure 3.3. The optimization maximum is a net power output of 12.1 MW, found at a condenser temperature of 14.0°C (P_{sat} = 1.71 bar) and evaporator temperature of 67.3°C (P_{sat} = 7.61 bar).

![Net Power Optimization for n-Butane](image)

With the evaporator and condenser pinch temperature assumptions, the required cost-normalized heat exchanger area is quite large at over 19,000 m². It is therefore valuable to perform a second optimization to maximize net power output per m² of heat exchanger area, or specific power. As shown in Figure 3.4, the specific power increases with higher condenser and evaporator temperatures until a maximum is reached at 32.1°C (P_{sat} = 3.02 bar) and 100.5°C (P_{sat} = 15.4 bar) respectively. The highest specific power was found to be 1.22 kW/m², at a net power output of 4.0 MW. This is in contrast to the 0.622 kW/m² specific power of the net power optimization.

A cost optimization of plant design will weigh the increased revenue of a higher power rating against the increased capital cost of the plant. While a cost optimization is not performed in this study, it can be concluded that the solution to the optimization will lie on the spectrum between the net power and specific power optimizations. That is, the plant would not be sized larger than the P_{net} design or smaller than the P_{spec} optimization. If the price for power is relatively low and heat exchanger cost relatively high, the power plant would be designed closer to the specific power optimization, whereas higher power prices and lower heat...
3.2. POWER SCENARIO

Figure 3.4: Specific power optimization of n-butane power plant model.

Exchanger costs will shift the design point towards net power optimization. The solution space for a potential cost optimization is shown on the net and specific power optimization charts in Figures 3.5 and 3.6. The graphs show the net and specific power outputs as a function of condenser temperature, and are shown for evaporator temperatures of 67.3°C and 100°C which are the optimum for net and specific power respectively. The results of the net and specific power optimizations are then shown by the intersection of their respective optimum condenser and evaporator temperatures. All of the potential condenser and evaporator temperature combinations that lie between the two optimizations on the graphs then represent the potential power plant designs. The location of the final power plant design point would lie somewhere in this solution space, determined by regional economic factors through the cost optimization.

Figure 3.5: Range of potential power plant designs of n-butane model shown on net power optimization chart.

At the optimized evaporator temperature of 67.3°C, the overall and functional system exergy efficiencies (Equations 2.35 and 2.36 respectively) were calculated for various condenser temperatures and plotted in Figure 3.7. Both efficiencies follow a trend similar to the net power optimization (see Figure 3.5 for comparison) the net power trend when plotted
CHAPTER 3. RESULTS

Figure 3.6: Range of potential power plant designs n-butane model shown on specific optimization chart.

against condenser temperature. Efficiencies increase uniformly with evaporator temperature, whereas the system has a local maximum for exergetic efficiency at the net-power-optimized condenser temperature of 14°C. The overall and functional model efficiencies of 36.4% and 20.0% are within range of similarly sized geothermal plants but are on the lower end of the range due to the low ambient temperature and, therefore, high exergy value of the incoming geothermal brine [67].

Figure 3.7: Overall and functional exergy efficiency of n-butane model plotted against condenser temperature.

3.2.3 Seasonal Fluctuations and Annual Output

The power plant was modelled with an air-cooled condenser rather than a water-cooled one. An air-cooled condenser, while requiring more heat transfer area, allows the plant to take advantage of the relatively cool ambient temperatures of central Alberta, and avoids the need to secure a stable water source. The nature of an air-cooled design dictates that it will experience significant fluctuations in power output with seasonal temperature changes. As the temperatures cool, the condenser is able to migrate to a lower value, increasing the
attainable pressure drop across the turbine, with the converse occurring at higher ambient temperatures.

The highest and lowest average monthly temperatures in Hinton, 13.7°C and -10.0°C respectively, were used as ambient temperatures in the seasonal model scenarios. The power plant model parameters were adjusted to produce optimization scenarios at these summer and winter conditions while maintaining similar design parameters, such as heat exchanger sizes. The seasonal variations of the model produced average monthly net power outputs of 9.47 MW in the summer scenario, and 16.1 MW in winter. The net power output of the seasonal models are depicted in Figure 3.8 as a comparison to the design case of 12.1 MW. Seasonal fluctuation in condenser and evaporator conditions are summarized in Table 3.4. It should be noted that even in the winter scenario, the lower pressure of the condenser is still positive relative to atmospheric, avoiding any air in-leakage concerns.

![Seasonal Power Plant Performance](image)

Figure 3.8: Seasonal fluctuations in net power output of power plant model. Net power at design conditions shown with dotted line.

<table>
<thead>
<tr>
<th>Ambient Temperature (°C)</th>
<th>Condenser Temp (°C)</th>
<th>Condenser Pres (bar)</th>
<th>Evaporator Temp (°C)</th>
<th>Evaporator Pres (bar)</th>
<th>Flow Rate (kg/s)</th>
<th>Net Power Output (MW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design</td>
<td>2.4</td>
<td>14.0</td>
<td>1.71</td>
<td>67.3</td>
<td>7.61</td>
<td>296</td>
</tr>
<tr>
<td>Summer</td>
<td>13.7</td>
<td>24.1</td>
<td>2.37</td>
<td>73.0</td>
<td>8.68</td>
<td>271</td>
</tr>
<tr>
<td>Winter</td>
<td>-10.0</td>
<td>1.0</td>
<td>1.07</td>
<td>71.8</td>
<td>8.44</td>
<td>271</td>
</tr>
</tbody>
</table>

Table 3.4: Change in heat exchanger conditions and power output of winter and summer power plant configurations.

Ideally, the power model would be tested for its power output on a daily basis using temperature data from an entire year. Automated optimization for daily temperatures, while maintaining design parameters, was not able to be achieved in this study. An approximation of monthly power output based average monthly temperatures (see Figure 2.7) was calculated by generating a polynomial expression for net power output as a function of ambient temperature using the summer, winter, and design models. Monthly average power outputs by this approximation are shown in Figure 3.9 along with their calculated monthly energy output.
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Figure 3.9: Average thermal power and total monthly energy of district heating system.

The total annual energy output of the power plant model can then be approximated by combining the output from each month. The power plant is estimated to provide 390 TJ (108 GWh) of energy annually from the geothermal reservoir over the project lifetime of 50 years. Geothermal brine then provides 720 GJ annually per kg/s of flow.

3.3 Heating Scenario

3.3.1 Design

The district heating network was designed to serve the maximum number of typical Alber- tan households at the coldest expected daily average ambient temperature. The system was designed from the household radiator side, using an 80/40/-20/20 (supply/return/ambient/room) standard.

With the geothermal energy input set and fixed supply and return temperatures, the number of households served was controlled by the heat transfer area of the central heat exchanger. The more heat transfer area, the higher the flow rate of radiator fluid could be supported (with the specified temperature increase), and the more houses can be supported. Figure 3.10 depicts this relationship.

As explained previously, the design of the district heating system is determined by the amount of energy transfer from the geothermal brine. The output temperature of the geothermal fluid was set to match the outlet temperature of the preheater in the power plant model to ensure the energy input to each system was equal. This input allowed the model to be solved for its design configuration. The central heat exchanger area was established at 5258 m², resulting in a radiator fluid flow of 728 kg/s, as shown in Table 3.5. Note that the low outlet temperature of the geothermal fluid warrants consideration of potential mineral precipitation. This is not a concern for this study as the chemistry of the brine in the target formations does not contain a significant amount of minerals, namely silica, expected to precipitate at low temperatures.

Using Equation 2.20, the heating load at the design temperature of -20°C was calculated at 6.6 kW per household, assuming an interior temperature of 20°C. The radiator fluid flow rate required to match the heating load could then be determined. With the inputs used in this study, the heating network model was able to provide over 18,000 households with space heating at design conditions. The radiator heat transfer details are summarized in Table 3.6. Under this design, each household is required to have 32.37 m² of radiator heat transfer area.
Figure 3.10: Effect of increasing heat exchanger area on number of households served by heating system and the decrease on geothermal fluid exit temperature. The selected design is shown by the dotted vertical line.

Table 3.5: Fluid type and stream temperatures of central exchanger and district heating system.

<table>
<thead>
<tr>
<th>Heat Exchanger Side</th>
<th>Fluid</th>
<th>Flow Rate (kg/s)</th>
<th>Inlet Temp (°C)</th>
<th>Outlet Temp (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Geothermal</td>
<td>NaCl Brine</td>
<td>540</td>
<td>118</td>
<td>56.1</td>
</tr>
<tr>
<td>Radiator</td>
<td>Water</td>
<td>728</td>
<td>39.2</td>
<td>82.6</td>
</tr>
</tbody>
</table>

Based on the radiators surveyed for the building heat transfer coefficient, this translates into a household radiator footprint of 7.3 m² [57].

Table 3.6: Summary of household radiator parameters for heating network design.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. of Households</td>
<td>18447</td>
<td></td>
</tr>
<tr>
<td>Radiator Area per Household</td>
<td>32.37</td>
<td>m²</td>
</tr>
<tr>
<td>Supply Temperature</td>
<td>80</td>
<td>°C</td>
</tr>
<tr>
<td>Return Temperature</td>
<td>40</td>
<td>°C</td>
</tr>
<tr>
<td>Flow Rate per Household</td>
<td>0.0395</td>
<td>kg/s</td>
</tr>
</tbody>
</table>

3.3.2 Available Energy

Rather than provide a small portion of base-load heating to a great number of homes, the district heating system was designed to provide all thermal energy required at the lowest expected daily average ambient temperature. It is clear then, that at any ambient temperature above -20°C, the heating system will not be extracting the maximum amount of energy from the geothermal fluid. The required flow rate per household decreases as ambient temperature increases, as shown in Figure 3.11. Therefore, maintaining the geothermal inputs results in a proportion of radiator fluid that becomes freed up for other potential customers.
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Figure 3.11: Thermal power used for households as a function of ambient temperature.

The district heating provider would have the opportunity to partner with heat-intensive industries, such as timber drying, food processing, balneology, to receive thermal energy in the form of the excess radiator fluid at times when the ambient temperature is higher than -20°C. As the thermal energy would otherwise go unused, it could be sold at levels competitive with low fossil fuel costs to form a mutually beneficial partnership.

Acquiring a low-cost source of thermal energy, even if intermittent, would likely be attractive to heat-intensive industries. The potential for available thermal energy can be calculated using temperature data to determine the heating load of the housing system. The household overall heat transfer coefficient calculated for 2013 most closely matched the average, which was the value used for the model design. Therefore, the daily temperature data from 2013 is used as a trial to view the fluctuation between heating load and available energy for industry. Figure 3.12 shows the household and available energy as an average daily output with trendlines showing the smoothed data (30 days).

Figure 3.12: Household heating and excess available thermal power based on daily average temperatures from 2013.

It can be seen that the so-called available energy for potential industry partners is significant, especially over the summer months where the thermal power rarely drops below 90 MW. Revenue from an average of 73.8 MW provided to industry could provide significant incentives for housing developers to invest in a geothermal district heating system. The total thermal energy provided for household space heating is shown by month in Figure 3.13 using the monthly temperature averages from the 2013-2017 period.
3.3. **HEATING SCENARIO**

Note that maintaining the geothermal and radiator flow rate regardless of ambient temperature is not typical for district networks. If a network services a fixed number of homes, the geothermal and radiator flow rates would be controlled relative to the ambient temperature. The scenario assuming local industries would utilize any excess power based on constant geothermal and radiator flow rates is constructed to evaluate total project energy output for the heating utilization and allow for comparison with the power utilization.

![Monthly Energy Delivery by Heating System](image)

Figure 3.13: Total monthly energy provided by the district heating system for heating and potential available energy for consumption by industry partners.

### 3.3.3 Annual Energy

Using the calculations displayed in Figure 3.13, the annual thermal energy provided for space heating of households can be determined. Table 3.7 lists the household heating data from the figure, along with the average annual output of 1512 TJ of energy. This total represents the thermal energy required to heat 18,447 homes for 50 years. However, if restricted to this project time scale, there is an even greater amount of 2328 TJ that would potentially go unused if not for auxiliary industry consumption. The heating system, as modelled, is able to provide up to 3840 TJ of thermal energy if there is sufficient demand. This is equivalent to an annual specific energy output of 7.1 TJ per kg/s of geothermal brine.
Table 3.7: Total monthly energy required for household heating using 2013-2017 temperature data.

<table>
<thead>
<tr>
<th></th>
<th>Household Heating Energy (TJ)</th>
<th></th>
<th></th>
<th></th>
<th></th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>2013</td>
<td>2014</td>
<td>2015</td>
<td>2016</td>
<td>2017</td>
<td></td>
</tr>
<tr>
<td>Jan</td>
<td>201</td>
<td>180</td>
<td>191</td>
<td>211</td>
<td>218</td>
<td>200</td>
</tr>
<tr>
<td>Feb</td>
<td>151</td>
<td>238</td>
<td>187</td>
<td>142</td>
<td>196</td>
<td>183</td>
</tr>
<tr>
<td>Mar</td>
<td>193</td>
<td>205</td>
<td>141</td>
<td>151</td>
<td>192</td>
<td>176</td>
</tr>
<tr>
<td>Apr</td>
<td>145</td>
<td>128</td>
<td>123</td>
<td>98.2</td>
<td>137</td>
<td>126</td>
</tr>
<tr>
<td>May</td>
<td>74.3</td>
<td>97.9</td>
<td>87.3</td>
<td>86.4</td>
<td>77.1</td>
<td>85</td>
</tr>
<tr>
<td>Jun</td>
<td>57.6</td>
<td>59.1</td>
<td>43.2</td>
<td>47.6</td>
<td>49.3</td>
<td>51</td>
</tr>
<tr>
<td>Jul</td>
<td>43.2</td>
<td>20.6</td>
<td>29.7</td>
<td>36.3</td>
<td>29.8</td>
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<td>215</td>
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<tr>
<td></td>
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</table>
Chapter 4

Discussion

4.1 Resource Assessment

Due to the nature and objective of this study, the proposed reservoir and its subsequent assessment are much larger than a typical project. Where most projects have a degree of uncertainty in the footprint of their reservoir, the reservoir area for this study was defined by the area encircled by the wells that met the criteria set out for the project, resulting in a fixed footprint.

As a result of the fixed reservoir volume, and even distribution of recovery factor, the thermal power distribution of the assessment simulation was quite flat, as opposed to a normal distribution generated by typical geothermal projects. The shape of the output distribution relays that the outcome is extremely dependent on the input distribution shape and values given to the recovery factor, as shown in the tornado chart (Figure 3.2), and results in a larger spread between cumulative probabilities than is common in geothermal Monte Carlo simulations. For example, a change from 95% cumulative probability to 90%, for example, results in a doubling of the estimated thermal power - 226 MW to 455 MW - over the 50-year project lifetime.

The selection for recovery factor, the most influential simulation parameter, and the selection of cumulative probability from the simulation results were both conservative in nature. Compared to the literature review, the selected range of 0% to 10% for recovery factor was extremely conservative for sedimentary reservoirs. The cumulative probability of 95% was the most conservative choice possible and is made even more so given the shape of the distribution and large spreads between probability levels.

While the assessment of this study can scarcely be seen as the potential production for a singular project, given its large proposed reservoir volumes, it nonetheless provides a measure of the geothermal energy consistent throughout the WCSB. In fact, it is likely that this assessment underestimates the geothermal energy per unit volume in the Upper Cretaceous lithology in the area near Hinton, due to the selection of conservative parameters in the Monte Carlo simulation.

Future Monte Carlo simulations for individual projects in the study area will encompass specific water-producing zones. With reservoir volumes narrowed in this way, reservoir parameters will likely be more well-defined as they will be specific to one well or a handful of wells, rather than 70. This should result in a more typical simulation distribution and provide a thermal power estimate with a higher confidence level.
4.2 Total Energy Delivered

For the purposes of evaluating total project energy and fossil fuel displacement, it is assumed that no output energy will go unused. The economics of a geothermal power plant are likely not feasible, but it is assumed that incentives would be instituted to encourage investment and ensure that all power produced by such a plant would be purchased by the transmission operator. Similarly, it is unlikely that any thermal power would be wasted. If industry partners are unable to consume all of the excess energy of maximum output, the geothermal flow rate would be controlled to match the demand of the network extending the project lifetime accordingly.

Total energy provided by the power and heating scenarios over the project lifetime is calculated to be 19.5 PJ and 192 PJ respectively. This disparity is to be expected by noting their thermal efficiencies of 9.2% and 92.4% and the fact that the input geothermal energy was equal in both scenarios. Figure 4.1 provides a visual comparison of the total energy delivered by the scenarios.

![Figure 4.1: Total energy delivered by space heating and power plant models. Energy proportional to circle area.](image)

4.3 Hinton-specific Project Lifetimes

The development scenarios for this study are evaluated on two criteria. One criterion is the magnitude of displaced fossil fuel energy based on the maximum energy output of each scenario – net power and number of houses respectively – given an equivalent amount of input geothermal energy. Another criterion is how effectively each scenario could serve the residences of the nearby town of Hinton.

As designed, the power plant provides 108 GWh annually with an output of 12.1 MW at the average annual ambient temperature. However, the residential power consumption in 2015 was just 27.9 GWh. Assuming the demand follows roughly the same seasonal fluctuation as the maximum plant output, that is higher in winter and lower in summer, which a typical Albertan household would, the plant could be theoretically downsized to serve only the residential demand of the town [44]. In this scenario, if demand remains at 2015 levels, the power plant could serve the residential community for 193 years.
4.4. NON-RENEWABLE ENERGY SAVINGS

The same approach can be taken with the heating system. Households in Hinton number 3,670, far less than the maximum number of houses served by the design heating network. If the design network was modified to match the heating demand of residential Hinton alone, it would output 301 TJ annually, extending the lifetime of the heating network to 638 years. This timeline would surely qualify the heating network as a "sustainable" enterprise.

The adjusted project lifetimes as related to the demands of the town of Hinton are depicted in Figure 4.2. The total geothermal input energy for both scenarios remains equivalent. This approach may not necessarily be pragmatic but does provide a sense of the sustainability of both options as they relate to a typical Albertan town. In this case, since the maximum output of the heating and power systems exceed the town’s demand for both, the reservoir could be exploited using cogeneration to serve both needs. A cogeneration system would consist of a power plant that generates a smaller amount of power to match the town’s demands. The exit temperature from this plant would therefore be higher than in the 12 MW plant described in this project, and be suitable for use in a district heating network. A cogeneration system designed specifically for Hinton would have a thermal higher efficiency that is much higher than that of the power plant described in this project, but still lower than that of the district heating network. The resulting non-renewable energy displacement would therefore fall between the values calculated for each system, detailed in the upcoming sections. A cogeneration system was not designed and evaluated as part of this study as the objective was to maximize energy output from each of the utilization methods on their own and contrast the results.

![Figure 4.2: Revised project lifetimes for power and heating scenarios when serving residential Hinton only.](image)

4.4 Non-renewable Energy Savings

While the heating scenario provides almost ten times the energy of the power generation scenario, this does not tell the whole story. Thermal energy is considered low-grade energy, whereas electricity is more valuable as it is higher-grade, organized energy which can be used for a multitude of applications. Fossil fuel energy displacement is used to evaluate the scenarios on equal footing.

Energy efficiency of displaced non-renewable energy systems was determined to be 60% and 90% for power and heating respectively, as detailed in Section 2.6. Using these system efficiencies, the non-renewable energy savings of the power scenario was determined to be 32.4 PJ, compared with 213 PJ in the heating scenario. The heating scenario provides over 6 times the fossil fuel savings of the power plant scenario, translating to an additional 4.85B m$^3$ of natural gas saved by the heating scenario, as summarized in Figures 4.3 and 4.4.
CHAPTER 4. DISCUSSION

Figure 4.3: Total energy provided by heating and power scenarios and their equivalent non-renewable energy displacements.

Figure 4.4: Total natural gas volume displaced by development options.

Natural gas displaced by the heating scenario amounts to 56% of the annual non-industrial\(^1\) consumption of the province in 2016 [68]. The volume of natural gas saved by the prospective heating network over its lifetime would be equivalent to the entire residential consumption of the province for over a year. If utilized by the existing natural gas power generation infrastructure, the saved natural gas could be used to produce over 21 TWh of electricity, compared to the 5.4 TWh generated by the geothermal binary power plant [68]. Table 4.1 lists the additional days that could be provided to a variety of sectors in the provincial market based on their daily consumption in 2016 [68].

The greenhouse gas emissions avoided as a result of the non-renewable energy savings can also be calculated. Approximately 50.3 kg of CO\(_2\) is emitted per GJ of natural gas [61]. Following the non-renewable energy and natural gas savings, the heating network results in 6.6 times more avoided emissions at a total of 10.8 megatonnes of CO\(_2\) - a significant savings considering the current carbon tax of $30/tonne. The comparison of emissions avoided in both scenarios is depicted in Figure 4.5.

\(^1\)Includes residential, commercial, and transportation consumption
Table 4.1: Equivalent consumption days provided by both utilization scenarios based on provincial consumption in 2016. Data from AER (2018). [68]

<table>
<thead>
<tr>
<th>Sector</th>
<th>Consumption (10^6 m³/d)</th>
<th>Equivalent Days</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Heating</td>
</tr>
<tr>
<td>Residential</td>
<td>14.1</td>
<td>406</td>
</tr>
<tr>
<td>Oil Sands Production</td>
<td>45.3</td>
<td>126</td>
</tr>
<tr>
<td>Power Generation</td>
<td>24.6</td>
<td>233</td>
</tr>
<tr>
<td>Provincial Total</td>
<td>149.8</td>
<td>38</td>
</tr>
</tbody>
</table>

Figure 4.5: Total CO\textsubscript{2} emissions avoided for geothermal development options.

It is this > 6 to 1 (6.6:1) ratio regarding displaced fossil fuel that is the most instructive comparison of the study. Knowledge of the difference in energy payback, relative to regional markets, will allow for resource owners (governments, communities, companies, etc) to focus on the most productive development of geothermal resources.

4.5 Ongoing and Future Studies

The Geothermal Energy group at the University of Alberta has been working on geothermal energy in the Hinton region for three years. Current activities are focused in several areas: geological modelling, oil and gas well re-purposing, and community outreach and education. While this group has been focused on geological research, they also liaise with engineering research at the university involving geothermal power generation utilizing a novel Stirling engine. The Geothermal Energy group is also heavily involved in a partnership with the town council of Hinton and Epoch Energy, a private company based in Calgary, AB. Epoch has received funding the University of Alberta as well as the provincial government to complete a pre-FEED study (delivered in 2017) to investigate the re-purposing of oil and gas wells near the town to provide geothermal heating for select commercial buildings. The company has now begun work on a full FEED study, again supported by the Geothermal Energy group at the U of A, to be delivered later in 2018.
There is obvious research and industry value in advancing geothermal engines to generate power at lower and lower temperatures. As part of an international effort to further this technology, these advances could have significant impact on those communities with insecure energy sources, unstable electrical grid access, and/or extremely high electricity costs. Remote settlements in northern Canadian territories are an example of communities which could potentially benefit from such research.

That being said, implementation of geothermal energy needs to remain focused on supplanting the current fossil-fuel-reliant space heating infrastructure of the province. The objective of replacing as much non-renewable energy as possible is best achieved by the utilization of existing oil and gas infrastructure to provide space heating. The ongoing studies, in cooperation with the town of Hinton, are in concert with this notion.

Partnerships with oil and gas companies will be important for geothermal development as they have world-leading expertise in drilling and have existing empirical well log data that can be used to pinpoint, and further research, water-producing reservoirs. There are countless anecdotal stories of high water production from individual wells during oil exploration and extraction, providing more proof of the potential for geothermal utilization.

In addition, knowledge transfer with international academic and industry partners should be a priority for Alberta research parties. Experience in addressing the challenges associated with using geothermal fluid in district heating systems should be leveraged from countries such as Iceland and Turkey which have implemented many large-scale district heating systems over several.

Heat transfer research into heat exchangers and radiators (geometry, 3D-printing, etc.) and working fluids (zeotropic fluids, nanofluids, etc.) should continue to progress and be adapted for regional conditions. Drilling and pipe construction techniques are likely more than sufficient due to the history of oil and gas, however, the re-purposing of existing wells for use in geothermal production should take center stage of research in the near future. This is the province’s greatest asset as it allows for prospective developers to potentially bypass the large initial investment characteristic of most geothermal projects. Therefore, it represents an opportunity to encourage and fast-track investment in Alberta’s budding geothermal industry.

4.6 Conclusion

The aim of this study was to provide confirmation of the magnitude of geothermal energy available in a sample region in Alberta and quantify, at a high level, the energy payback of two different development options based on displaced non-renewable energy.

The resource assessment with largely conservative parameter choices proves there is significant geothermal energy available throughout the WCSB, even at shallow depths. With reasonable alternative selections of recovery factor and cumulative probability, a thermal power estimate of an order of magnitude higher than the study value of 226 MW could be justified. Finding regions of high permeability will be the major factor in specific project development in the future. This assessment adds to numerous analyses in previous studies which have purported the WCSB as a massive source of geothermal energy.

A basic EES-modelled geothermal binary power plant produced a net power output of 12.1 MW with n-butane as its working fluid. The plant operated at a 9.2% thermal efficiency with overall and functional second law efficiencies of 36% and 20% respectively. Designed with an air-cooled condenser, the model power output dropped to 9.5 MW at average summer temperatures and increased to 16.1 MW in winter. The power plant, over its lifetime,
was found to produce 19.5 PJ (5.4 TWh) of energy and displace a potential 0.87B m³ of natural gas.

Next, a district heating network was designed to provide heating to as many houses as possible at a design condition of 80/40/-20 (supply/return/ambient temperature in °C). It was found that over 18,000 houses could be serviced by the network for a 50-year lifetime. Alternatively, if the network was restricted to only the households in Hinton, it could provide heating for the residences for over 600 years.

Under the constraints of the study, the heating network provided 192 PJ of energy representing 5.72B m³ of displaced natural gas, and most notably, outperformed the power scenario on a higher than 6:1 basis with respect to non-renewable energy displacement. While one may have guessed that geothermal heating would provide the higher energy payback, due to the disparity between the thermal efficiencies of these types of systems, there was a desire to test the scenarios against a regional environment and to quantify the displaced energy comparisons.

With its high energy density and relatively low emissions, natural gas is crucial in the projected transition to a cleaner energy economy. It is poised to become even more prevalent in Alberta’s electricity infrastructure as plans move forward with the provincial and federal phase-out of coal power by 2030. The void left by coal will be filled in part by increasing wind and solar power, but the provincial grid will need a backbone of natural gas for the foreseeable future. Wind and solar cannot currently match the scale and dependability inherent in natural gas power plants. The drive to transition to a more renewable energy economy should go hand-in-hand with a recognition of the value and versatility of fossil fuels – natural gas in particular. To lessen the dependence on natural gas supply and prices, that are sure to increase in the next few decades, geothermal energy should be employed to displace natural gas space heating where possible. While there remains justification for small-scale geothermal power generation, heating is most useful application of geothermal in Alberta at this time.

The comparisons performed in this study were completed on purely an energy basis, with no evaluation of economics. However, some general comments with regards to the economics of the options can be made. Due to the machinery involved in a geothermal binary power plant, the cost of the power generation option is sure to be at least one order of magnitude greater than that of a district heating network. Also, a geothermal plant in Alberta is not likely to provide the magnitude and cost of power to attract power-intensive industries, or to even compete against historical low natural gas power prices without heavy subsidies. On the other hand, developing abundant and competitively-priced geothermal heating does have the potential to attract many industries such as greenhouses, lumber drying, spas, and aquaculture. Also, Iceland has proved that geothermal in and of itself can be an attraction, even if it is just in the form of a mineral spa.

The origin of this study was to employ the novel approach of fossil fuel displacement to evaluate a greenfield reservoir. The objective was not to provide specifics of development scenarios, but rather the overall recommended direction of development. The conclusion, in general, is that as long as fossil fuels make up the bulk of an energy market, all efforts to implement geothermal energy should be focused on displacing fossil fuels in space heating prior to entertaining large-scale power generation projects.
Bibliography


Appendix A

ORC Literature Survey
<table>
<thead>
<tr>
<th>Reference</th>
<th>Efficiencies</th>
<th>Fan(\text{dP}) (Pa)</th>
<th>Evaporator Pinch (°C)</th>
<th>Condenser Pinch (°C)</th>
</tr>
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<tbody>
<tr>
<td>Ahangar (2012) [47]</td>
<td>85%</td>
<td>75%</td>
<td></td>
<td></td>
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<tr>
<td>Budisulistyo et. al (2015) [69]</td>
<td>85% 98% 80%</td>
<td>5</td>
<td></td>
<td></td>
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<tr>
<td>Dai et. al (2009) [70]</td>
<td>85% 60%</td>
<td>8</td>
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<td></td>
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<tr>
<td>Deethayat et. al (2015) [71]</td>
<td>85% 80%</td>
<td>6 3</td>
<td></td>
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<tr>
<td>Dickson &amp; Fanelli (2005) [72]</td>
<td>81-85%</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>El-Emam &amp; Dincer (2013) [73]</td>
<td>89% 97% 95%</td>
<td>5.3-12</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gitobu (2016) [48]</td>
<td>85% 75% 65% 100</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>He et. al (2012) [74]</td>
<td>80% 96% 75%</td>
<td>5 5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Hettiarachchi et. al (2007) [46]</td>
<td>85% 96% 75-80%</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Khennich &amp; Galanis (2012) [75]</td>
<td>80%</td>
<td>80%</td>
<td>5</td>
<td></td>
</tr>
<tr>
<td>Lakew &amp; Bolland (2010) [76]</td>
<td>80% 90% 80%</td>
<td>10 5</td>
<td></td>
<td></td>
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<tr>
<td>Lukawski (2009) [77]</td>
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<td></td>
<td></td>
</tr>
<tr>
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<td></td>
<td></td>
<td></td>
</tr>
<tr>
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<td>75% 80%</td>
<td>10 5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Saleh et. al (2007) [79]</td>
<td>85%</td>
<td>65%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Shengjun et. al (2011) [80]</td>
<td>80% 96% 75%</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Zarrouk &amp; Moon (2014) [81]</td>
<td>96-99%</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Appendix B

EES Output - Power Plant Diagram
Appendix C

EES Output - Heating Network Diagram
APPENDIX C. EES OUTPUT - HEATING NETWORK DIAGRAM
Appendix D

EES Code - Power Plant

"INFORMATION"
Name: Casey Lavigne
Term: Fall 2017 to Spring 2018
Purpose: This geothermal binary power plant model was developed as a part of a Masters thesis project to evaluate the utilization of an unttapped geothermal resource near Hinton, AB

The inputs therefore represent reservoir conditions that have been determined from measurement data and/or assumed properties from comparable reservoirs

"INPUTS"
"ASSUMPTIONS"
dP_PH = 0.5 [bar] {Pressure drop of geothermal fluid through Preheater}
dP_EV = 0.5 [bar] {Pressure drop of geothermal fluid through Evaporator}
dP_DH = 1 [bar] {Downhole pressure req’d to pump brine to facility}
T_sub = 0 [C] {Amount of condenser sub-cooling}
T_sup = 0 [C] {Amount of evaporator super-heating}
T[0] = 2.4 [C] {Temperature of ambient air}
P[0] = 1 [bar]
Pinch_EV = 7.5 [C] {Pinch point in evaporator}
Pinch_C = 5 [C] {Pinch point in condenser}
dP_fan = 100 [Pa] {Pressure drop of air across fan}
"OPTIMIZATION PARAMETERS"
T_cond = 14.0 [C]
T_evap = 67.3 [C]
P_evap = p_sat(WF$,T=T_evap)
P_cond = p_sat(WF$,T=T_cond)
"EFFICIENCIES"
\eta_turb=0.85 \{Turbine Efficiency\}
\eta_gen=0.96 \{Generator Efficiency\}
\eta_pump = 0.8 \{Pump Efficiency\}
\eta_fan = 0.70 \{Fan Efficiency\}
\eta_motor = 0.95 \{Motor Efficiency\}
"HEAT TRANSFER COEFFICIENTS"
U_PH = 1 [kW/m2-C]
U_EV = 1.6 [kW/m2-C]
U_C = 0.8 [kW/m2-C]
"FLUIDS"

hfluid$ = 'NaCl'
C_geo = 6.4
WF$ = 'n-butane'
cfluid$ = 'Air_ha'

"GEOTHERMAL FLUID"
m_dot_geo = 540 [kg/s]
T_geo_in = 118 [C]
T_geo_out = 80 [C] {Exit temp just for initial model testing}
P_geo = 2 [bar]

"GEOTHERMAL FLUID"

{STATE 11 - Evaporator Inlet}
T[11] = T_geo_in
m[11] = m_dot_geo

{STATE 12 - Evaporator outlet/Preheater Inlet}
h[12] = enthalpy(hfluid$,C=C_geo,P=P[12],T=T[12])

{STATE 13 - Preheater Outlet}
T[13] = T_geo_out
P[13] = P[12] - dP_PH {Assume majority of pressure drop occurs in evaporator; negligible dP in PH}
m[13] = m[12]
h[13] = enthalpy(hfluid$,C=C_geo,P=P[13],T=T[13])

"BINARY LOOP"

{State 1 - Condenser Outlet}
T[1] = T_cond - T_sub
P[1] = P_cond
x[1] = 0 {Saturated liquid approximation}
h[1] = enthalpy(WF$,T=T[1],x=x[1])
s[1] = entropy(WF$,T=T[1],x=x[1])
v[1] = volume(WF$,T=T[1],x=x[1])

{State 2 - Pump Outlet/Preheater Inlet}
P[2] = P_evap
T[2] = temperature(WF$,h=h[2],P=P[2])
s[2] = entropy(WF$,h=h[2],P=P[2])

{State 3 - Preheater Outlet/Evaporator inlet}
x[3] = 0
T[3] = T_evap
P[3] = P_evap
h[3] = enthalpy(WF$,x=x[3],T=T[3])
s[3] = entropy(WF$,x=x[3],T=T[3])
\[
\text{(Preheater Energy Balance)}
\]
\[
Q_{\text{PH}} = m_{\text{dot WF}} \cdot (h[3] - h[2])
\]
\[
Q_{\text{PH}} = m_{\text{dot geo}} \cdot (h[12] - h[13])
\]

\[
\text{(Evaporator Energy Balance)}
\]
\[
Q_{\text{EV}} = m_{\text{dot WF}} \cdot (h[4] - h[3])
\]
\[
Q_{\text{EV}} = m_{\text{dot geo}} \cdot (h[11] - h[12])
\]

\[
\text{(State 4 - Evaporator Outlet/Turbine Inlet)}
\]
\[
P[4] = P_{\text{evap}}
\]
\[
x[4] = 1
\]
\[
\]
\[
"h[4] = \text{enthalpy(WFS,P=P[4],T=T[4])} \{\text{change P to x if T_{sup}=0}\}
\]
\[
s[4] = \text{entropy(WFS,P=P[4],T=T[4])}"
\]
\[
h[4] = \text{enthalpy(WFS,x=x[4],T=T[4])}
\]
\[
s[4] = \text{entropy(WFS,x=x[4],T=T[4])}
\]
\[
\{\text{Calculate h[5] using turbine eff}\}
\]
\[
h_s = \text{enthalpy(WFS,P=P[5],s=s[4])}
\]
\[
dh_s = h[4] - h_s
\]
\[
\]
\[
\eta_{\text{turb}} = \frac{dh_a}{dh_s}
\]

\[
\text{(State 5 - Turbine Exit/Condenser Inlet)}
\]
\[
P[5] = P_{\text{cond}}
\]
\[
s[5] = \text{entropy(WFS,P=P[5],h=h[5])}
\]
\[
T[5] = \text{temperature(WFS,P=P[5],h=h[5])}
\]
\[
"x[5] = \text{quality(WFS,P=P[5],h=h[5])}\"
\]

\[
\text{(State 6 - Sat Vap in Condenser)}
\]
\[
P[6] = P_{\text{cond}}
\]
\[
x[6] = 1
\]
\[
T[6] = t_{\text{sat(WFS,P=P[6])}}
\]
\[
h[6] = \text{enthalpy(WFS,P=P[6],x=x[6])}
\]
\[
s[6] = \text{entropy(WFS,P=P[6],x=x[6])}
\]

\[
\text{"CONDENSER - AIR-COOLED"}
\]
\[
\text{"CONDENSER INLET - COOLING FLUID"}
\]
\[
T[21] = T[0]
\]
\[
h[21] = \text{enthalpy(cfluid$\text{,T=T[21],P=P[0]}\})
\]
\[
v[21] = \text{volume(cfluid$\text{,T=T[21],P=P[0]}\})
\]
\[
s[21] = \text{entropy(cfluid$\text{,T=T[21],P=P[0]}\})
\]
\[
"MID-CONDENSER" \{\text{Where working fluid is saturated vapour}\}
\]
\[
h[22] = \text{enthalpy(cfluid$\text{,T=T[22],P=P[0]}\})
\]
\[
s[22] = \text{entropy(cfluid$\text{,T=T[22],P=P[0]}\})
\]
\[
"CONDENSER OUTLET - COOLING FLUID"
\]
\[
\eta_{\text{cond}} = T_{\text{cond}} - T[23] \{\text{Change this somehow}\}
\]
\[
h[23] = \text{enthalpy(cfluid$\text{,T=T[23],P=P[0]}\})
\]
\[
s[23] = \text{entropy(cfluid$\text{,T=T[23],P=P[0]}\})
\]
\[
\{\text{Energy Balance}\}
\]
\[
Q_{\text{C1}} = m_{\text{dot c}} \ast (h[22] - h[21]) \{\text{Energy for first half of condenser}\}
\]
\[
Q_{\text{C1}} = m_{\text{dot WF}} \ast (h[5] - h[6])
\]
Q_C2 = m_dot_c * (h[23] - h[22]) {Energy for second half of condenser}
Q_C2 = m_dot_WF * (h[6] - h[1])
Q_C = Q_C1 + Q_C2

"HEAT EXCHANGER AREAS"
{Log-mean Temp Difference: Preheater}
\[ dT_b_{PH} = T[12] - T[3] \]
LMTD_{PH} = (dT_a_{PH} - dT_b_{PH}) / ln(dT_a_{PH}/dT_b_{PH})
{Log-mean Temp Difference: Evaporator}
\[ dT_a_{EV} = dT_b_{PH} \]
\[ dT_b_{EV} = T[11] - T[3] \] {Need to add another section for superheating if it is > 0}
LMTD_{EV} = (dT_a_{EV} - dT_b_{EV}) / ln(dT_a_{EV}/dT_b_{EV})
{Log-mean Temp Difference: Condenser (split in two sections)}
\[ dT_a_{C1} = T[5] - T[23] \]
\[ dT_b_{C1} = T[6] - T[22] \]
LMTD_{C1} = (dT_a_{C1} - dT_b_{C1}) / ln(dT_a_{C1}/dT_b_{C1})
\[ dT_a_{C2} = dT_b_{C1} \]
\[ dT_b_{C2} = T[1] - T[21] \]
LMTD_{C2} = (dT_a_{C2} - dT_b_{C2}) / ln(dT_a_{C2}/dT_b_{C2})
A_PH = Q_PH / (U_PH * LMTD_{PH})
A_EV = Q_EV / (U_EV * LMTD_{EV})
A_C = Q_C1 / (U_C * LMTD_{C1}) + Q_C2 / (U_C * LMTD_{C2})

{Heat Exchanger Costs}
M_PH = 0.9 {Cost multiplier - estimated PH cost = $450}
M_EV = 1 {Cost multiplier - estimated EV cost = $500}
M_C = 1.2 {Cost multiplier - estimated Cond cost = $600}
A_HX = A_PH*M_PH + A_EV*M_EV + A_C*M_C {Cost-normalized HX area with EV cost estimate as the baseline}

"WORK AND POWER"
\[ W_{turb} = m_{dot_WF} * (h[4] - h[5]) * eta_gen \]
\[ W_{fan} = m_{dot_c} * v[21] * dP_{fan} * convert(W,kW) / eta_fan / eta_motor \]
\[ W_{FP} = m_{dot_WF} * (h[2] - h[1]) / eta_motor \]
\[ W_{DP} = m_{dot_geo} * v[11] * (dP_PH + dP_EV + dP_DH) * convert(bar,kPa) / eta_motor \]
W_pump = W_{FP} + W_{DP}
P_net = W_{turb} - W_{fan} - W_pump
P_spec = P_net / A_HX

"EXERGY"
{Dead state enthalpy and entropy for all fluids}
h_wf[0] = enthalpy(WF$,T=T[0],P=P[0]) {Working fluid}
s_wf[0] = entropy(WF$,T=T[0],P=P[0])
h_gf[0] = enthalpy(hfluid$,C=C_geo,T=T[0],P=P[0]) {Brine}
h_cf[0] = enthalpy(cfluid$,T=T[0],P=P[0]) {Air}
s_cf[0] = entropy(cfluid$,T=T[0],P=P[0])
T_K[0] = converttemp(C,K,T[0]) {convert ambient temperature from Celsius to Kelvin}
{Calculate entropy of geothermal brine using interpolation - Dittman (1977)}
\[ T_{\text{min}} = 48.9 \, [\text{C}] \]
\[ T_{\text{max}} = 126.7 \, [\text{C}] \]
\[ s_{\text{min}} = 0.1535 \times 4.1868 \] \{Entropy of 6.4\% brine at \( T_{\text{min}} \); 4.1868 is unit conversion from btu/lbmR to kJ/kgK\}
\[ s_{\text{max}} = 0.3521 \times 4.1868 \] \{Entropy of 6.4\% brine at \( T_{\text{max}} \)\}
\[ s_{\text{gf}[0]} = \text{entropy(Steam\_IAPWS,} T=T[0],P=P[0]) \] \{Interpolation of brine entropy\}
\[ \text{Duplicate } j = 11,13 \]
\[ s[j] = s_{\text{min}} + (T[j] - T_{\text{min}})/(T_{\text{max}} - T_{\text{min}}) \times (s_{\text{max}} - s_{\text{min}}) \]
\[ \text{End} \]

\[ s_{\text{gf}[0]} = \text{entropy(Steam\_IAPWS,} T=T[0],P=P[0]) \] \{Interpolation of brine entropy\}
\[ \text{Duplicate } j = 11,13 \]
\[ s[j] = s_{\text{min}} + (T[j] - T_{\text{min}})/(T_{\text{max}} - T_{\text{min}}) \times (s_{\text{max}} - s_{\text{min}}) \]
\[ \text{End} \]

\[ \text{Exergy of Working Fluid} \]
\[ \text{Duplicate } j=1,6 \]
\[ e[j] = (h[j] - h_{\text{wf}[0]}) - T_{\text{K}[0]} \times (s[j] - s_{\text{wf}[0]}) \]
\[ E_{\text{dot}[j]} = m_{\text{dot\_WF}} \times e[j] \]
\[ \text{End} \]

\[ \text{Exergy of Geothermal Fluid} \]
\[ \text{Duplicate } j=11,13 \]
\[ e[j] = (h[j] - h_{\text{gf}[0]}) - T_{\text{K}[0]} \times (s[j] - s_{\text{gf}[0]}) \]
\[ E_{\text{dot}[j]} = m_{\text{dot\_geo}} \times e[j] \]
\[ \text{End} \]

\[ \text{Exergy of Cooling Fluid} \]
\[ \text{Duplicate } j=21,23 \]
\[ e[j] = (h[j] - h_{\text{cf}[0]}) - T_{\text{K}[0]} \times (s[j] - s_{\text{cf}[0]}) \]
\[ E_{\text{dot}[j]} = m_{\text{dot\_c}} \times e[j] \]
\[ \text{End} \]

"SYSTEM EFFICIENCIES"

\[ \text{First Law Eff} \]
\[ \eta_{\text{thermal}} = P_{\text{net}} / (Q_{\text{PH}} + Q_{\text{EV}}) \]

\[ \text{Second Law Effs} \]

\[ \text{Turbine} \]
\[ \eta_{\text{turb\_fun}} = W_{\text{turb}} / (E_{\text{dot}[11]} - E_{\text{dot}[13]}) \]

\[ \text{Preheater} \]
\[ \eta_{\text{PH\_fun}} = (E_{\text{dot}[3]} - E_{\text{dot}[2]}) / (E_{\text{dot}[12]} - E_{\text{dot}[13]}) \]
\[ \eta_{\text{PH\_bf}} = (E_{\text{dot}[3]} + E_{\text{dot}[13]}) / (E_{\text{dot}[2]} + E_{\text{dot}[12]}) \]

\[ \text{Evaporator} \]
\[ \eta_{\text{EV\_fun}} = (E_{\text{dot}[4]} - E_{\text{dot}[3]}) / (E_{\text{dot}[11]} - E_{\text{dot}[12]}) \]
\[ \eta_{\text{EV\_bf}} = (E_{\text{dot}[4]} + E_{\text{dot}[12]}) / (E_{\text{dot}[3]} + E_{\text{dot}[11]}) \]

\[ \text{System} \]
\[ \eta_{\text{sys\_overall}} = P_{\text{net}} / E_{\text{dot}[11]} \] \{Decide which is right - 1 or 2?\}
\[ \eta_{\text{sys\_fun}} = P_{\text{net}} / (E_{\text{dot}[11]} - E_{\text{dot}[13]}) \] \{Should \( E_{[13]} \) be subtracted?\}
\[ \eta_{\text{sys\_bf}} = (W_{\text{turb}} + E_{\text{dot}[23]} + E_{\text{dot}[13]}) / (W_{\text{pump}} + W_{\text{fan}} + E_{\text{dot}[11]}) \]
Appendix E

EES Code - Heating Network

" INFORMATION
   Name: Casey Lavigne
   Term: Fall 2017 to Spring 2018 Purpose: This geothermal heating network model was
developed as a part of a Masters thesis project to evaluate the utilization of an unttapped
geothermal resource near Hinton, AB The inputs therefore represent reservoir conditions
that have been determined from measurement data and/or assumed properties from compara-
ble reservoirs

"INPUTS AND ASSUMPTIONS"

"Geothermal Fluid"
m_dot_g = 540 [kg/s]
T_g_in = 118 [C] {Geothermal wellhead temperature}
P_g = 2 [bar]
C_g = 6.4 [%]
GF$ = 'NaCl'
cp_g=specheat(GF$,T=(T[11]+T[12])/2,C=C_g)

"Radiator Fluid"
RF$ = 'Water'
P_RF = 3 [bar] {Assume pressure changes are negligible w.r.t. enthalpy and entropy
calcs (compressed liquid)}
"Equipment"
eta_motor = 0.95
U_r = 0.0056 [kW/m2-C] {Radiator heat transfer coefficient calculated from readily
available radiators on market} U_HX = 1 [kW/m2-C] {Used same value as Preheater in
Power Plant model}
"Household Energy"
K_Loss = 0.165 [kW/C] {Overall transfer coefficient of average house - see "K Value"
spreadsheet}

"Radiator Fluid Capacity"
{Subsc. "hx" = central heat exchanger}
{Subsc. "r" = household radiator}
cp_rf_hx[0]=specheat(RF$,T=(T_0[1]+T_0[4])/2,P=P_RF) {Design hx}
cp_rf_r[0]=specheat(RF$,T=(T_0[2]+T_0[3])/2,P=P_RF) {Design r}
cp_rf_hx=specheat(RF$,T=(T[1]+T[4])/2, P=P_RF)

"Design Temps"
T\_R = 20 [C] \{Room Temp\}
T\_0[11] = T\_g\_in

"DESIGN PARAMETER"
A\_HX[0] = 5258 \{Design parameter - Calculated by making T\_0[12] = exit temp from power plant\}
"T\_0[12] = 56.07"
A\_House = A\_r / N \{Radiator area per household\}

"DESIGN CONDITIONS"
"Design Temps"
T\_0[0] = -20 [C] \{Design Ambient\}
T\_0[2] = 80 [C] \{Consumer radiator inlet temperature\}
T\_0[3] = 40 [C] \{Radiator exit/return temp\}

"Household Heat Loss"
N = 10000 \{No. of homes - to be "enced out later\}
Q\_dot\_r[0] = K\_Loss * (T\_R-T\_0[0]) * N \{Heat lost to environment; N is number of households\}

"Radiator Energy Balances"
LMTD\_r[0] = (T\_0[2] - T\_0[3]) / ln((T\_0[2] - T\_R) / (T\_0[3] - T\_R))
Q\_dot\_r[0] = U\_r * A\_r * LMTD\_r[0] \{Heat given off by radiator\}
Q\_dot\_r[0] = m\_dot\_r\_0 * cp\_rf\_r[0] * (T\_0[2] - T\_0[3]) \{Heat provided by radiator\}

"Heat Losses in Pipes"
cp\_rf\_r[0] * T\_0[1] *0.97 = cp\_rf\_r[0] * T\_0[2] \{3\% thermal losses on supply side piping\}
cp\_rf\_r[0] * T\_0[3] *0.98 = cp\_rf\_r[0] * T\_0[4] \{2\% thermal losses on supply side piping\}

"Central HX Energy Balances"
Q\_dot\_HX\_0 = m\_dot\_r\_0 * cp\_rf\_hx[0] * (T\_0[1] - T\_0[4])
Q\_dot\_HX\_0 = m\_dot\_g * cp\_g\_0 * (T\_0[11] - T\_0[12])

"Central HX Area"
dT\_HX\_a[0] = T\_0[11] - T\_0[1] \{Temp difference between fluids on side a\}
dT\_HX\_b[0] = T\_0[12] - T\_0[4] \{Temp difference between fluids on side b\}
LMTD\_HX[0] = (dT\_HX\_a[0] - dT\_HX\_b[0]) / ln(dT\_HX\_a[0]/dT\_HX\_b[0])
Q\_dot\_HX\_0 = U\_HX * A\_HX[0] * LMTD\_HX[0]

"PUMPING POWER"
"Geothermal Pump"
dP\_g = 0.25 [bar] + A\_HX[0] / 1991 [m\^2] * 0.25 [bar] \{dP relative to dP assumed in power plant model - 0.5 bar for 1991 m\^2 in Preheater\}
v[11] = volume(GFS,C=C\_g,T=T\_0[11])
W\_P\_g = m\_dot\_g * v[11] * dP\_g *convert(bar,kPa) / eta\_motor

"Freed’ Power Available for Radiator Pumping (from the absence of furnace fans)"
W\_fan = 0.0426 [kW] \{Average (over the year) power used by furnace - see ‘Furnace Power’\}
W\_P\_Avail = W\_fan * N
v\_0[1] = volume(RFS,P=P\_RF,T=T\_0[1])
W\_P\_Avail = m\_dot\_r\_0 * v\_0[1] * dP\_r *convert(bar,kPa) / eta\_motor
"EXERGY CALCULATIONS"

\[ T_{0\_K[0]} = \text{converttemp('C', 'K', T_{0[0]})} \]

{Calculate entropy of geothermal brine using interpolation - Dittman (1977)}

\[ T_{\text{min}} = 48.9 \ [^\circ\text{C}] \]

\[ T_{\text{max}} = 126.7 \ [^\circ\text{C}] \]

\[ s_{\text{min}} = 0.1535 \times 4.1868 \ \{\text{Entropy of 6.4\% brine at } T_{\text{min}}; \ 4.1868 \text{ is unit conversion from btu/lbmR to kJ/kgK}\} \]

\[ s_{\text{max}} = 0.3521 \times 4.1868 \ \{\text{Entropy of 6.4\% brine at } T_{\text{max}}\} \]

\[ s_{\text{g}[0]} = 0.03554 \times 4.1868 \ \{\text{Value from linear extrapolation of brine entropy curve}\} \]

{Dead State Enthalpy/Entropy}

\[ h_{\text{g}[0]} = \text{enthalpy(GF$,C=C_{\text{g}},T=T_{0[0]}, P=P_{\text{g}})} \]

\[ h_{\text{r}[0]} = \text{enthalpy(RF$,T=T_{0[0]},P=P_{\text{RF}})} \]

\[ s_{\text{r}[0]} = \text{entropy(RF$,T=T_{0[0]},P=P_{\text{RF}})} \]

{Interpolation of brine entropy}

Duplicate \( j = 11,12 \)

\[ s_{\text{g}[j]} = s_{\text{min}} + (T_{0[j]} - T_{\text{min}})/(T_{\text{max}} - T_{\text{min}}) \times (s_{\text{max}} - s_{\text{min}}) \]

End

{Exergy of Geothermal Fluid}

Duplicate \( j=11,12 \)

\[ h_{\text{g}[j]} = \text{enthalpy(GF$,C=C_{\text{g}},T=T_{0[j]},P=P_{\text{g}})} \]

\[ e_{\text{g}[j]} = (h_{\text{g}[j]} - h_{\text{g}[0]} - T_{0\_K[0]} \times (s_{\text{g}[j]} - s_{\text{g}[0]})) \]

\[ E_{\text{dot}[j]} = m_{\text{dot}_{\text{g}}} \times e_{\text{g}[j]} \]

End

{Exergy of RadiatorFluid}

Duplicate \( j=1,4 \)

\[ s_{\text{r}[j]} = \text{entropy(RF$,T=T_{0[j]},P=P_{\text{RF}})} \]

\[ h_{\text{r}[j]} = \text{enthalpy(RF$,T=T_{0[j]},P=P_{\text{RF}})} \]

\[ e_{\text{r}[j]} = (h_{\text{r}[j]} - h_{\text{r}[0]} - T_{0\_K[0]} \times (s_{\text{r}[j]} - s_{\text{r}[0]})) \]

\[ E_{\text{dot}[j]} = m_{\text{dot}_{\text{r}_{0}}} \times e_{\text{r}[j]} \]

End

"SYSTEM EFFICIENCIES"

{First Law Eff}

\[ \eta_{\text{thermal}} = (Q_{\text{dot}_{\text{r}[0]} - W_{P_{\text{g}}}) / Q_{\text{dot}_{\text{HX}}} \]

{Second Law Effs}

{Central Heat Exchanger}

\[ \eta_{\text{HX\_fun}} = (E_{\text{dot}[1]} - E_{\text{dot}[4]}) / (E_{\text{dot}[11]} - E_{\text{dot}[12]}) \]

\[ \eta_{\text{HX\_bf}} = (E_{\text{dot}[1]} + E_{\text{dot}[12]}) / (E_{\text{dot}[4]} + E_{\text{dot}[11]}) \]

{System}

\[ \eta_{\text{sys\_overall}} = (E_{\text{dot}[2]} - E_{\text{dot}[3]} - W_{P_{\text{g}}}) / E_{\text{dot}[11]} \]

\[ \eta_{\text{sys\_fun}} = (E_{\text{dot}[2]} - E_{\text{dot}[3]} - W_{P_{\text{g}}}) / (E_{\text{dot}[11]} - E_{\text{dot}[12]}) \]

\[ \eta_{\text{sys\_bf}} = (E_{\text{dot}[2]} - E_{\text{dot}[3]}) / (W_{P_{\text{g}}} + E_{\text{dot}[11]}) \]

"DAILY ENERGY"

Life = 50 [year]

\[ m_{\text{g\_daily}} = 540 \times 86400 \ [\text{kg}] \]

\[ m_{\text{g\_total}} = m_{\text{g\_daily}} \times 365 \times 50 \]

\[ Q_{\text{g\_Daily}} = Q_{\text{dot}_{\text{HX\_0}}} \times 86400 \ [\text{s}] \]

\[ Q_{\text{r\_Daily}} = Q_{\text{dot}_{\text{r}[0]}} \times 86400 \ [\text{s}] \]