Employing Heat Pumps to Recover Low Grade Industrial Thermal Resources for Space Heating and Cooling

By

Ian Michiel Ross

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Thesis Examiners/Examinateurs de thèse:

Dr. John Scott
(Supervisor/Directeur(trice) de thèse)

Dr. Helen Shang
(Committee member/Membre du comité)

Dr. Markus Timusk
(Committee member/Membre du comité)

Dr. Baoqiang Liao
(External Examiner/Examinateur externe)

Approved for the Faculty of Graduate Studies
Approuvé pour la Faculté des études supérieures
Dr. David Lesbarrères
Monsieur David Lesbarrères
Dean, Faculty of Graduate Studies
Doyen, Faculté des études supérieures

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I Abstract

The use of heat pumps to upgrade and recover low grade industrial thermal resources is an opportunity for industry to deliver low cost space heating at a reduced carbon footprint. Heat pumps also offer the potential to provide space heating and space cooling from a single unit. To facilitate rapid determination of the potential for low grade heat recovery using ground source heat pumps, a rapid scoping method was developed capable of establishing critical temperatures to help elucidate which resources are most likely to be recovered economically. To demonstrate the applicability of the rapid scoping method, an analysis of various process cooling waters present at a smelter site was undertaken. The implications for cost and carbon dioxide emissions, were both analyzed for this facility, although the concepts developed can be employed to any site that generates large quantities of thermally low-grade heat.

Keywords: Heat pump, Heat recovery, Mineral processing, Pyrometallurgy, Smelting, Space heating, Waste heat

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VII List of Abbreviations

ADQ	Alcoa Deschambault Quebec
ASHP	Air Source Heat Pump
CAD	Canadian Dollar
COP	Coefficient of Performance
DH	District Heating
FDM	Finite Difference Method
GHG	Greenhouse Gas
GSHP	Ground Source Heat Pump
MWSHP	Mine Water Source Heat Pump
OECD	Organization for Economic Development & Cooperation
TMS	Minerals, Metals & Materials Society
TSHP	Terrain Source Heat Pump
USD	US Dollars
WSHP	Water Source Heat Pump
WWSHP	Waste Water Source Heat Pump

1 Introduction

Heat recovery is an opportunity for industry to reduce costs and emissions, while increasing process efficiency and sustainability; goals which all industries aspire to. As the temperature of a waste heat source decreases however, the potential applications it is suited for decrease as well (Fox, Sutter, & Tester, 2011; Kwak, Oh, & Kim, 2012). At some point, the resource will need to be upgraded (i.e., increased in temperature) before it can be usefully applied, such as to space heating. A heat pump is one technology that is ideally suited to upgrading waste heat (Braham, 1984; Urdaneta-B & Schmidt, 1980; Wright & Steward, 1985). They occupy a unique role in the field of heat recovery, because they offer the possibility of recovering waste heat only marginally above ambient temperatures.

At the Sudbury Integrated Nickel Operations (Sudbury INO) smelter site, there are numerous process streams in the low temperature range that contain otherwise waste heat suitable for recovery via heat pumps. One purpose of this thesis is to analyze the potential for recovered heat from these streams to be employed for space heating of on-site facilities and provide environmental benefit. Specifically: can heat pump systems be designed to recover waste heat in a cost effective manner whilst simultaneously reducing carbon dioxide (CO_2) emissions? The second purpose is to develop a rapid scoping model applicable to heat recovery opportunities where commercial, off the shelf geothermal heat pumps can be employed. The model will establish a simple metric to identify the potential value of different thermal resources and determine if it is worthwhile for further resources to be allocated to a full feasibility study.

It is intended that this thesis will help bring awareness of the potential for low cost, low carbon emissions space heating by employing heat pumps to recover low grade waste heat. The value of this research is in using an actual case study, which will help to motivate industry to be mindful of these

opportunities. Furthermore, the rapid scoping models will allow engineers to quickly and easily access the thermal resources available to them.

Novel contributions within this thesis include the rapid scoping model for space conditioning, which extends the existing concept of critical coefficient of performance (COP) to the concept of critical temperature by employing commercial heat pump data.

1.1 Background

In the most general sense, heat recovery is the field of engineering concerned with employing waste heat in a manner that is materially beneficial to society. Waste heat can be defined as any thermal energy at a temperature above ambient conditions not currently employed in a beneficial manner. All thermal resources present value, and it is the need of heat recovery engineering to devise the technology and methods so that the cost of recovery is less than the value recovered (Bhattacharjee, 2010).

Heat recovery schemes comprise three essential components: a source of waste heat, a technology or method to recover this heat, and a sink/application for the waste heat. Heat pumps occupy a unique niche as a recovery method, because they can make use of extremely low grade resources which are too cool for direct use applications (Law, Harvey, & Reay, 2013).

The highest grade heat is suited to electricity production using various Rankine cycles. As the grade drops, many direct use drying applications become applicable, as well as direct use space heating, hot water production and snow melting. When the grade is too low for direct use applications, one of the only options left is to employ a heat pump to upgrade the heat and employ it in space heating.

Furthermore, heat recovery has been identified as a “first fuel” by the International Energy Agency (IEA, 2013). This concept reflects the fact that efficiency improvements which reduce primary energy consumption are often the most effective way at reducing energy associated greenhouse gas (GHG) emissions. For example, from the production of a single kWh of electricity sourced from a coal

fired generation plant, there will be significant GHG emissions as well as other pollutants. If that kWh of electricity is instead generated by a solar cell, then direct GHG emissions are theoretically zero, although solar cells still produce indirect GHG emissions during their manufacture and ongoing maintenance. The first fuel concept states that the cleanest, least carbon intensive kWh that can ever be produced is the kWh that never needs to be produced in the first place because the demand for that energy was eliminated by efficiency improvements. In this way, waste heat can be viewed as a particularly valuable fuel, a first fuel, because it is a fuel that presents a truly unique opportunity to source energy with low carbon intensity. This concept assumes that efficiency improvements come with zero manufacturing or maintenance associated GHG (which is not true for the efficiency improvement proposed herein, since heat pump manufacturing is not carbon neutral), so a proper comparison between renewable energy generation and efficiency improvements should proceed on a case by case basis. Nevertheless, the first fuel concept illustrates the potential for efficiency improvements to assist in offsetting GHG emissions. Furthermore, efficiency improvements have saved IEA member countries 5.7 trillion US dollars (USD) since 1990 (IEA, 2016), with USD 550 billion saved in 2014 (Figure 1-1).

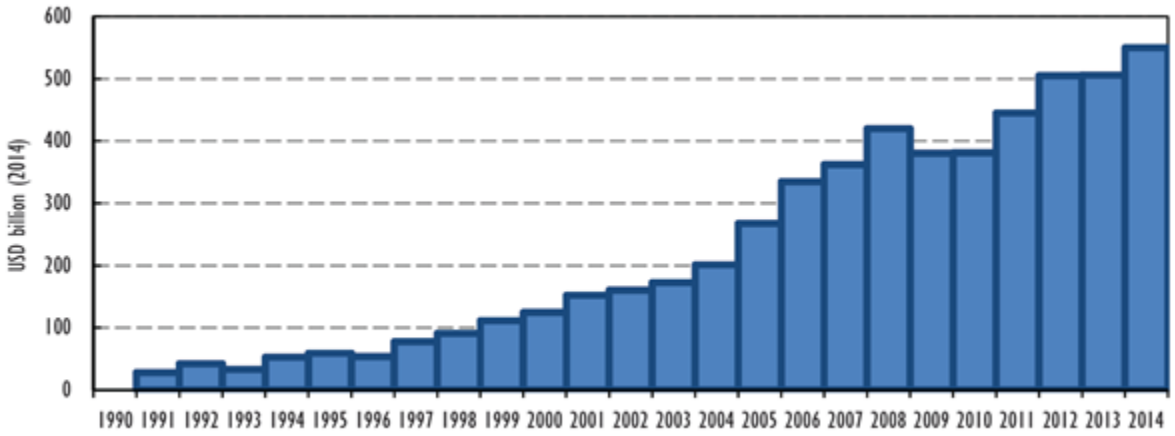

Figure 1-1: Annual savings from investment in energy efficiency (IEA, 2016)

Space heating is an important part of the energy system of humanity, as many of us live in climates where our homes, offices, factories and schools will require some degree of space heating. In fact, the single largest end-use consumer of energy in the residential sector of 19 OECD (Organization for Economic Cooperation and Development) countries is space heating, as seen in Figure 1-2 (IEA, 2014).

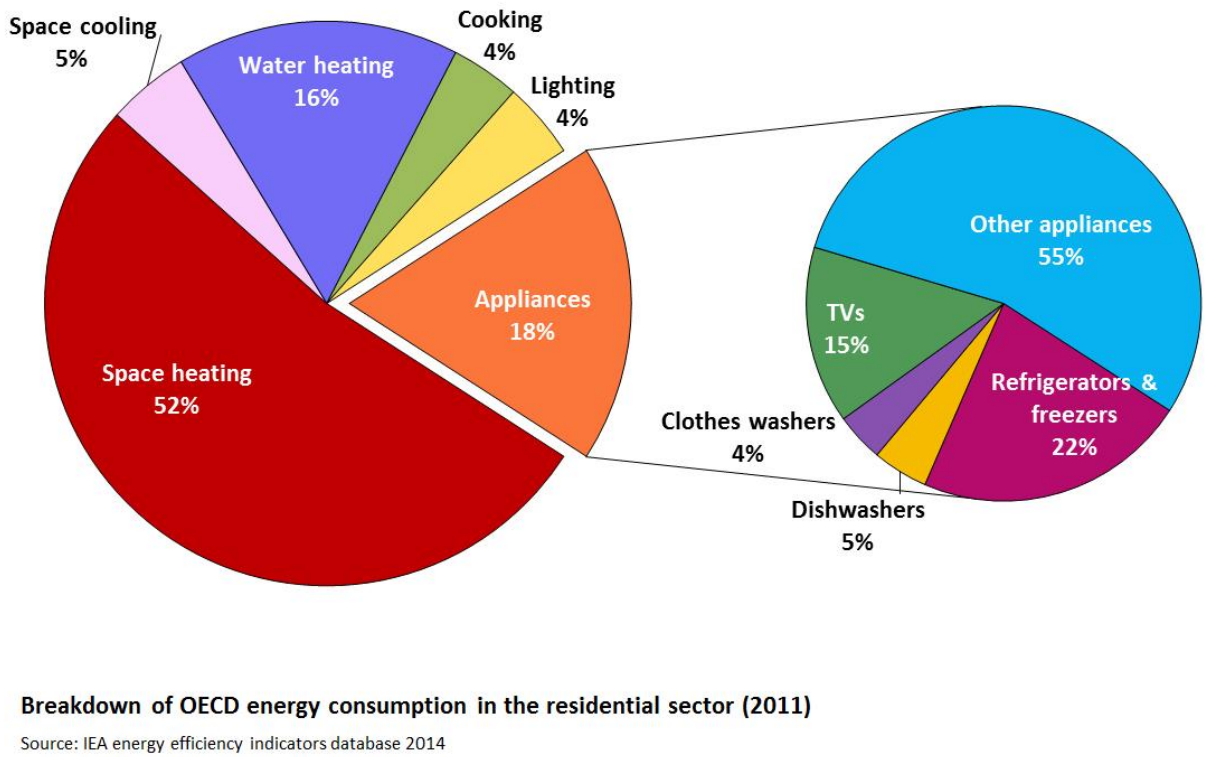

Figure 1-2: Residential energy consumption (IEA, 2014)

This implies that space conditioning (space heating and space cooling together) accounts for 57% of total residential energy consumption. For the OECD, residential energy consumption accounts for approximately 25% of final energy consumption nationally, so space conditioning of residences alone can account for roughly 14% of final energy consumption. Considering that the bulk of this energy is sourced directly from fossil fuels (natural gas, heating oil, propane) and electricity, with fairly limited

renewables (wood and other biomass, solar) it becomes clear that changing the way humanity delivers space conditioning is integral to reducing the carbon intensity of our energy system. The potential for heat pumps to deliver space heating more efficiently than competing technologies (which can achieve at most 100% conversion of input energy to heat) implies moving from alternative methods to heat pumps can reduce energy consumption. Furthermore, since the majority of heat pumps are driven by electricity, they have the ability to deliver space heating as renewably as the generation method used to produce the electricity they consume. This coupling between space conditioning and electricity generation means that as electrical utilities move towards low carbon, renewable methods of generation, space conditioning would become more low carbon and renewable as well (Chua, Chou, & Yang, 2010). To achieve a future of zero net CO_2 emissions, heat pumps are a critical technology because they move space heating away from combustion and consume less energy than electrical resistance.

1.2 Thesis Overview

In Chapter 2, a literature review will be presented summarizing recent research in the field of heat pump heat recovery. After this review chapter, the remainder of the thesis will comprise three chapters of original research and a final chapter for conclusions and potential further work. At the time of writing this thesis, some of this work has also been included in a paper which has been accepted for publication (see Appendix A)

Chapter 3 presents a review of the concept of critical COP for heat pump systems, and extends this concept to the idea of critical temperature and how it can be used to assess waste heat recovery opportunities. The concept of critical temperature is developed not only for space heating, but for space conditioning as well. It is believed that practicing engineers will be able to apply the concept of critical temperature with greater ease than critical COP, since critical temperature innately incorporates heat pump performance by employing data from a variety of commercial manufacturers of high capacity

heat pumps. In Chapter 4, an analysis of the potential for Sudbury INO to recover a variety of on-site waste heat streams is presented. A comparison of the results of Chapter 4 with the results from the rapid scoping models of Chapter 3 is also presented. In Chapter 5, the heat pump system design identified in Chapter 4 will be used to supply space conditioning (as opposed to solely space heating), demonstrating the unique potential for heat pumps within the field of heat recovery and offering an opportunity to employ the concept of critical space conditioning temperature. The conclusions of the thesis, as well as opportunities for further work along this research path, are presented in the sixth and final chapter.

2 Literature Review

This chapter will present the academic literature relevant to heat pumps and their connection with the wider field of heat recovery. General information on heat pumps is presented, along with specific literature on heat pump assisted heat recovery with regards to mining and mineral processing facilities. With respect to pyrometallurgical facilities, the present work is comparatively novel as the resources considered are of an exceptionally low grade compared to resources common in this field.

2.1 Heat Pumps Employed for Heat Recovery

As highlighted in Chapter 1, opportunities for heat recovery in 21st century industry are ubiquitous, and these opportunities have driven the development of technologies and techniques to recover the economic value inherent in waste heat (Ammar, Joyce, Norman, Wang, & Roskilly, 2012; Hammond & Norman, 2014). As the grade of a thermal resource decreases, the economic value of the resource and the number of technologies that can be employed for its recovery also decrease (Ammar et al., 2012). Many heat recovery schemes, termed direct use, apply waste heat directly to some beneficial end using heat exchangers. To operate, these direct use schemes require the waste heat to be at a temperature greater than the required temperature of the application. If the grade of a resource is exceptionally low, then it becomes impossible to employ a heat exchanger and the resource will require upgrading (Law et al., 2013). An example of a technology that can be used to upgrade a low grade resource is a heat pump (Seck, Guerassimoff, & Maizi, 2013).

2.1.1 Review of Heat Pumps

In the most general sense, a heat pump is any device capable of moving thermal energy against a thermal gradient when supplied with work (for an introduction to heat pumps, see (Moran, Shapiro, Boettner, & Bailey, 2011)). The figure of merit for a heat pump is called the coefficient of performance (COP). It is a measure of how effectively a heat pump can move heat when supplied with power. For

example, a heat pump which is able to move 3 kW of heat, Q_{in}, when supplied with 1 kW of input power, W, has a COP of 3, as calculated using Equation 1.

$$COP = \frac{Q_{in}}{W} \qquad 1$$

Heat pumps are one of the only technologies with a Sankey Diagram like the one seen in Figure 2-1, where the heat output is larger than the energy input, since much of the delivered energy has been drawn from the ambient environment.

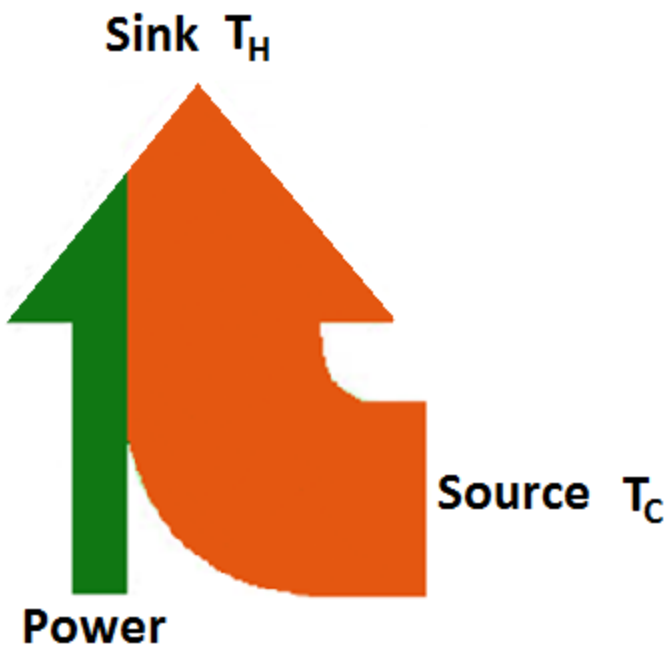

Figure 2-1: The Sankey Diagram of a heat pump

It is noteworthy that COP is the inverse of the thermodynamic efficiency of a conventional heat engine (the efficiency of a heat engine being the quantity of work derived from a given input of heat). This relationship also holds in the ideal case, for just as the efficiency of any heat engine is bounded above by the Carnot efficiency, the COP of any heat pump is also bounded above by the Carnot COP (COP_{carnot}). The COP_{carnot} of a heat pump is only dependent on the temperature of the reserves of heat, T_C for the source and T_H for the sink, between which it operates, as seen in Equation 2.

$$COP_{carnot} = \frac{T_H}{T_H - T_C} \quad 2$$

From examining Equation 2, it is apparent that as the temperature difference between the source and sink (called the lift) diminishes, the COP_{carnot} will tend towards infinity. No physical heat pump can ever attain such a COP, as any physically realisable system will have mechanical and isentropic inefficiencies that differ from the ideal Carnot cycle and limit the COP that can be practically achieved. However, the COP of a physical heat pump system is still proportional to $COP_{carnot,}$ so a diminishing lift will still result in an improving COP (Spoelstra, Haije, & Dijkstra, 2002). This relationship illustrates the opportunity for heat pumps in a heat recovery context. Consider a system designed to heat some process unit, but local sources of recoverable heat are too cool to employ heat exchangers. In this instance, a heat pump is necessary. Given that heat pumps can operate off the thermal energy of the ambient environment, there arises the question of why bother to use waste heat over ambient heat at all? The reason is that operating the heat pump on waste heat means that the elevated temperature improves the COP of the heat pump system, so recovering waste heat proceeds more efficiently than coupling to the ambient environment.

The heat pump is a well-developed and proven technology which has attained global use in space condition and industrial heat recovery applications (Chua et al., 2010), see for example early optimization work on heat pumps for industrial processes by (Wallin & Berntsson, 1994; Wallin, Franck, & Berntsson, 1990; Zaheer-Uddin, 1992). Heat pumps have been previously introduced in their most general form: as non-descript devices characterized only by their ability to move heat against the thermal gradient at the expense of work. This was done to differentiate between the general principle of a heat pump and the many technical embodiments which achieve this general principle, because while all heat pumps have similar functionality, their different technical embodiments vary significantly. The three largest categories of heat pump technology are vapour compression (also called Reverse Rankine), absorption and thermoelectric (Moran et al., 2011).

Vapour compression heat pumps are the most common from a commercial perspective, and are used in many industrial applications (Chua et al., 2010). The majority of all refrigerators, automotive cooling systems and household air conditioners are all vapour compression units. The basic operating principle of the vapour compression cycle is that a working fluid (also called refrigerant) will release heat when condensing from gas to liquid, and absorb heat when vaporizing from liquid to gas. By cycling these opposite actions, a device which absorbs heat at a low temperature and expels heat at a higher temperature can be created. Since the commercial heat pumps examined later in this thesis are vapour compression, an extended introduction into the operating principle and technology of vapour compression systems will presented later in this subsection.

The absorption heat pump has a similar operating principle, in that a gaseous fluid will release thermal energy when dissolved into solution, and will likewise uptake thermal energy when removed from solution (Wu, Wang, Shi, & Li, 2014). By cycling a working fluid between its gaseous and aqueous state, a continual process of collecting and releasing heat will occur. Arranging such a cycle so that absorption of gaseous working fluid occurs at the sink and desorption occurs at the source completes the absorption heat pump (Demir, Mobedi, & Ülkü, 2008).

The thermoelectric heat pump has the most limited commercial application and scope due to comparatively low COP and limited capacity, which makes it poorly suited for the majority of applications (Zebarjadi, Esfarjani, Dresselhaus, Ren, & Chen, 2012). The operating principle of a thermoelectric heat pump is based on the Peltier effect (also called the thermoelectric effect or Peltier-Seebeck effect) which is significantly different from vapour compression or absorption methods. The Peltier effect is the direct conversion of electrical energy into a temperature difference obtained by applying a voltage across a two dissimilar, conducting metals (or a semi-conductor), arranged in such a manner so that one face of the metal composite grows hotter while the other grows cooler.

Returning to the vapour compression cycle, it can be seen that the cycle steps are reciprocal to those of the more familiar Rankine cycle (and hence the vapour compression cycle is sometimes referred to as reverse Rankine). The steps are given below, with the corresponding states illustrated in the T-s diagram of Figure 2-2.

- 1-2: The working fluid undergoes an isentropic compression to a pressure above the saturation pressure of the working fluid at the prevailing sink temperature.
- 2-3: The working fluid condenses at constant pressure.
- 3-4: The working fluid is throttled to a lower pressure at constant enthalpy, this lower pressure being below the saturation pressure of the working fluid at the prevailing source temperature.
- 4-1: The working fluid evaporates at constant pressure.

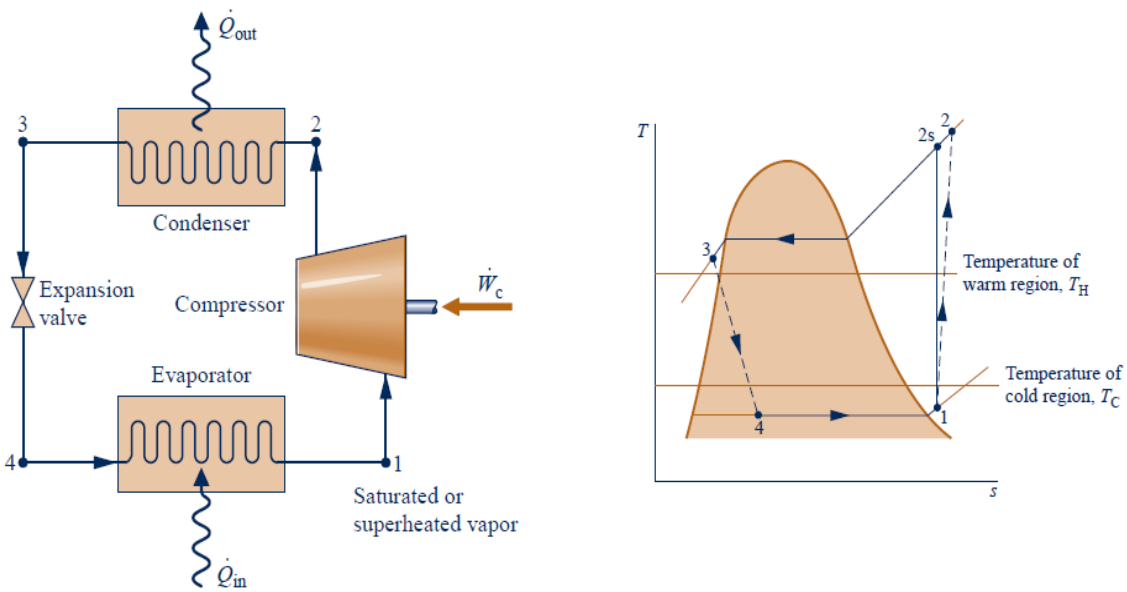

Figure 2-2: T-s diagram and vapour compression cycle schematic (Moran et al., 2011)

The ideal vapour compression cycle can be realized with four essential components: the compressor, the condenser, the throttling valve and the evaporator (Figure 2-2). The compressor is responsible for raising the pressure of working fluid to above the saturation pressure at the inlet of the

condenser, the condenser being proximal to and thermally connected with the sink. The condenser is a heat exchanger designed so that thermal energy released by the condensing working fluid is transferred to the sink. The throttling valve is a control valve connecting the condenser and evaporator, and is responsible for managing both pressure drop (necessary to ensure evaporation) and mass flow rate (controls the output of the heat pump). The pressure drop is great enough for the pressure of the working fluid to fall below the saturation pressure at the evaporator and so evaporation begins. Like the condenser, the evaporator is simply a heat exchanger designed to enable sufficient heat transfer between the working fluid and the source to which the evaporator is connected.

The preceding cycle can have the direction of heat transfer reversed by switching the direction in which working fluid flows, so that the physical orientations of the compressor and evaporator are also reversed. One option to achieve this goal would be to run the compressor in reverse. In practice however, reversing the flow direction of the compressor is complicated and requires sacrifice in compressor design that reduces efficiency and increases cost. Instead, the compressor is designed for unidirectional flow, and a fifth component called a reversing valve is responsible for alternating the flow of working fluid. Ultimately, the only difference between a true heat pump with a cooling and heating mode and the more common, exclusively cooling devices like refrigerators is the existence of this reversing valve (Staffell, Brett, Brandon, & Hawkes, 2012).

2.1.2 Classifying Heat Pumps

The classification scheme for heat pumps is derived from their commercial use for space conditioning in residential and commercial facilities, see for example (Kew, 1985; Sarbu & Sebarchievici, 2014). In addition to classifying heat pumps by their operating principle, heat pumps are further classified by the material with which they exchange heat (i.e., the material that absorbs heat in cooling mode or supplies heat in heating mode, such as air, water and soil). In commercial literature, heat pumps are generally divided into two categories: air-source heat pumps (ASHPs) and ground source heat

pumps (GSHPs) (Staffell et al., 2012). As the name implies, ASHPs exchange thermal energy between the interior of a building and the surrounding ambient air (F. Wang et al., 2015). Most residential air conditioning systems sold in North America are essentially ASHPs restricted to only operate in cooling mode. In contrast, GSHPs include all heat pumps which do not operate using air as their medium of exchange (Mustafa Omer, 2008; Self, Reddy, & Rosen, 2013; Wu, You, Wang, Shi, & Li, 2014). In practice, this means that GSHPs exchange heat with water or the physical mass of the earth (such as rock and soil), although they can also be integrated with ice storage, solar collectors and phase change materials (Zhu, Hu, Xu, Jiang, & Lei, 2014). Since the GSHP category is quite broad, it is necessary to further subdivide the category of GSHPs into water source heat pumps (WSHPs) and terrain source heat pumps (TSHPs). WSHPs are coupled to lakes, rivers or ponds in conventional implementations, while novel WSHP systems can make use of shower waste water, residential waste water or water from hydroelectric dams (see Section 2.1.3). TSHPs are traditionally coupled to rock within the earth's crust or shallow soil using a ground coupled heat exchanger called a ground loop (Soni, Pandey, & Bartaria, 2015; Yang, Cui, & Fang, 2010). It is important to note that the classification system of a heat pump is based upon the final material with which the heat pump exchanges its heat. For example, many TSHPs use a ground loop charged with water as the heat exchange fluid, so the process of heat exchange is from rock to water then to heat pump. It would be incorrect to classify such a system as a WSHP however, because while the heat pump operates using water as a medium of exchange, the thermal reserve of the system is ultimately the earth.

An overview of the many methods in which a GHSP system can be coupled to a reserve of thermal energy is presented in Figure 2-3. The best method for a given situation will depend on local availability, soil and rock types, accessibility for heavy machinery, water quality, expected system life and the operating characteristics of the heat pump.

A further categorization for GSHP systems is based on the method in which they exchange energy with their thermal reserve (Self et al., 2013; Staffell et al., 2012). A system which moves the physical material of the reserve to the heat pump, separating the material from the refrigerant of the heat pump with a single heat exchanger stage is termed open loop. In Figure 2-3, the second row and bottom right panel are open loop designs as the water comprising their thermal reserve is directly transported to the heat pump. Alternatively, the first row and bottom left panel show closed loop designs.

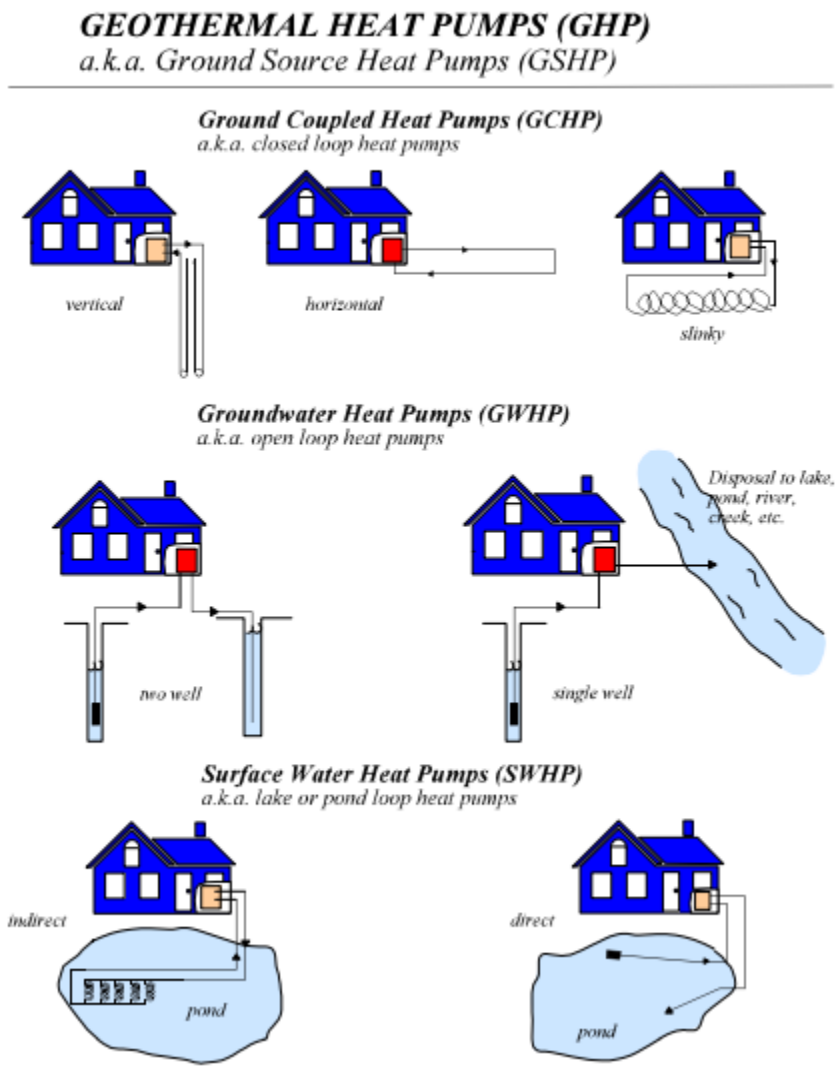

Figure 2-3: Design options for closed and open loop GSHPs (Rafferty, 2001)

In addition to the evaporator and condenser heat exchangers intrinsic to a vapour compression heat pump, a closed loop employs a third heat exchanger in the form of the ground loop. In the top left panel of Figure 2-3, the ground loop is installed in vertical bore holes 100-150 m deep (see Yang et al. (2010) for a review of vertical borehole system), while the second and third panels of the first row shows the ground loop installed in a shallow trench that is usually between 1 – 2 meters deep (Staffell et al., 2012). Figure 2-4 demonstrates what installing such ground loops actually looks like in situ.

Figure 2-4: A slinky type ground loop and a vertical borehole (Staffell et al., 2012)

2.1.3 Application of Heat Pumps to Recovery

Heat pumps have an important role within the field of heat recovery, particularity because they act as the critical enabling technology for recovery of very low grade resources which are impossible to recover by other means (Bruckner et al., 2015); this is not to say they are only useful in low temperature

applications however, see for example(Follinger & Merker, 1992; Gorianec, Pozeb, Tomi, & Trop, 2014; Spoelstra et al., 2002).

In the field of power generation, waste heat and local buildings with space heating and hot water needs are abundant, so heat pump enabled heat recovery has been regarded as a natural fit (Gorianec et al., 2014; Hebenstreit et al., 2014; Qiu, Zhang, & Liu, 2013). Hebenstreit et al. (2014) aimed to recover flue gas enthalpy from wood-fired biomass boilers by coupling a heat pump system with the boiler unit. They examined 10 kW wood pellet up to 10 MW wood chip fired systems, and discovered that operating costs could be decreased by 2-13% accompanied by an increase in primary energy efficiency of 3-21%. Qiu et al. (2013) looked at the application of a WSHP to recover thermal energy presented by the condenser of a traditional Rankine cycle power plant, concluding that the proposal was feasible and should be investigated further. The feasibility of recovering waste heat from the generator cooling water of a hydroelectric plant was studied by Gorianec et al. (2014). The primary energy efficiency of the generators can be increased by as much as 1% in this way. For a 500 kW capacity heat pump system, they identified a payback period of 2 years if recovered heat is applied to on-site buildings, while a payback period of 7 years would apply if recovered heat is used for an urban district heating (DH) system. For a review of DH systems see Rezaie & Rosen (2012), and for a Canadian perspective on DH see (Dalla Rosa, Boulter, Church, & Svendsen, 2012).

One interesting opportunity for sourcing novel space heating is to employ a waste water source heat pump (WWSHP), namely a WSHP configured to exchange energy with residential or commercial waste water (Chao, Yiqiang, Yang, Shiming, & Xinlei, 2013; Deng & Gu, 2012; Gu & Wang, 2013; Kahraman & Çelebi, 2009; Meggers & Leibundgut, 2011; Xiao, Luo, & Li, 2014). While the source of waste heat is different, the characteristics of waste heat from residential wastewater are similar to the waste heat streams examined later in this thesis in terms of temperature and stability. The low temperature of waste water makes it well suited for application to both space heating and cooling (an

application examined in this thesis as well) as highlighted in (Deng & Gu, 2012; Gu & Wang, 2013). Chao et al. (2013) present the first experimental study for a bath water WWSHP by examining the potential of bath water from a spa. A significant issue of using spa bath waste water was fouling: the heat pump suffered a gradual decrease in COP, and the supply pipe experienced a 20% capacity drop over five months as scaling occurred.

The possibility of employing a WWSHP for a single family residential home has been examined (Liu, Lau, & Li, 2014; Ni et al., 2012). Liu et al. (2014) employed a hybrid ASHP/WWSHP, capable of operating using either air or residential grey water (or both simultaneously), to compensate for the fact that residential buildings produce insufficient grey water to meet their heating demand. Ni et al. (2012) characterized the benefits of their proposed residential grey water WWSHP against a conventional space conditioning and hot water solution (a gas furnace + air conditioner + electric water heater) for a family home in New York City. They concluded decreases of 34% and 27% could be attained for primary energy and potable water consumption, respectively.

An experimental study with very low temperature lifts was undertaken by Li et al. (2014) when they examined heat recovery from waste water for heating flatfish culture water. With waste water at temperatures no greater than 12°C and flatfish desiring temperatures in the 10-15°C range, the temperature lifts are exceptionally low, leading to COPs as high as 5.1 in the experimental test system.

2.2 Heat Recovery in the Mining and Mineral Processing Industry

This section will present literature on heat recovery within the mining and mineral processing industry. Emphasis will be placed on those studies which highlight the use of heat pumps or the application of waste heat to space heating.

2.2.1 Active and Abandoned Mines

Heat pumps are ideally suited to heat recovery from active and abandoned mines, as many resources (dewatering flows, mine air, sump water) are low grade, and non-space heating applications

for recovered heat are rare (An, 2012; Smith & Arthur, 1996). For abandoned mines in particular, where recoverable heat is geothermal in nature, resources may be only a few degrees above ambient temperatures; in a study of 1600 abandoned mines, 80 sites were identified as producing warm water outflows, the greatest of which was only 7.3°C above average annual ambient conditions (Lawson & Sonderegger, 1978).

For active mines, heat pumps which are coupled to the dewatering flow of the mine can be termed a mine water source heat pump (MWSHP), see (Du, Dou, & Qi, 2013; Jinggang, Meixia, Chuanchuan, & Xiaoxia, 2010; J. Raymond, Therrien, & Gosselin, 2010; Jasmin Raymond & Therrien, 2007). The Moyska and Doyon mines in Abitibi, Quebec, were identified as being possible candidates for MWSHPs, as the installation costs for a traditional ground loop in a standard GSHP system are significantly reduced by making use of the mines pre-existing dewatering infrastructure (J. Raymond et al., 2010). A unique advantage MWSHPs have over traditional GSHPs is that exothermic oxidization of waste rock within the mine structure effectively adds to the thermal potential of the site (J. Raymond et al., 2010).

An extensive review of geothermal energy recovery from abandoned underground mines has been provided in (Hall, Scott, & Shang, 2011). For an earlier review, see (Banks, Skarphagen, Wiltshire, & Jessop, 2003). In addition to elevated temperatures courtesy of geothermal energy, the thermal inertia of surrounding rock causes the variation in annual temperatures for mine sourced water and air to be considerably less than ambient air or surface water. This stability enables heat pump systems to achieve higher heating COP in the winter and higher cooling COP in the summer (Watzlaf & Ackman, 2006).

Recently, a DH system for Murdochville industrial park in Quebec, Canada was proposed, analyzed and optimized in Raymond & Therrien (2014). This system aims to extract energy from the flooded mines of Gaspé in Murdochville. Coal mines are particularly suited to recovery of this kind

because they are fairly common, shallow and easy to access (Banks, Fraga Pumar, & Watson, 2009; Richardson & Lopez, 2014). For more recent examples of energy recovery from flooded mine water after Hall et al. (2011), see (Karu, Valgma, & Kolats, 2013; Kelso & Johnson, 2015).

2.2.2 Operating Mineral Processing Sites

The literature most relevant to this thesis pertains to heat recovery from mineral processing sites, particularly smelters. At the 2008 Minerals, Metals and Materials Society (TMS) annual meeting, Johnson, Choate, & Dillich (2008) estimated that 20-50% of the primary energy employed in metal and non-metallic mineral processing is dissipated as waste heat.

Heat recovery opportunities at a new-build magnesium siliothermic reduction plant planned by Nevada Clean Magnesium Inc. is overviewed in (Sever, 2013). It was concluded that 43 MW of power generating capability is possible using various recovery and cogeneration technologies, including Organic Rankine cycles (Hung, Shai, & Wang, 1997; Lecompte, Huisseune, van den Broek, Vanslambrouck, & De Paepe, 2015), with the majority of recoverable heat contained in the ferrosilicon furnace exhaust and gas turbine exhaust.

Many studies examine the potential for heat recovery from aluminum smelters, as aluminium production is both energy intensive and high in production volume (primary aluminum produced by the Hall-Héroult process requires 12-15 MWh of electricity per ton (Nowicki & Gosselin, 2012; Wedde & Sorhuus, 2012).

Nowicki & Gosselin (2012) examined the primary aluminum industry as a whole and the potential for waste heat and thermal integration therein. They estimate that half of all energy employed in primary aluminum production is ultimately lost as waste heat. As a case study, they analyze the Alcoa Deschambault Quebec (ADQ) smelter (production of 260,000 tons/yr), located in Quebec, Canada. With all sources considered it was estimated that the ADQ smelter presented on average 205 MW of waste heat, 34% of which was easily extractable in the opinion of the authors. Of

this total heat loss, the two greatest shares were potline exhaust gas at 88 MW and heat flux across cell walls at 89 MW. Furthermore, there is 40 MW more extractable heat than on-site demand, indicating that off-site roles for recovered heat should be considered to capture the greatest economic value. Echoing the focus of this thesis, Nowicki & Gosselin (2012) identified cooling water at 40°C carrying 7.4 MW, which they deemed to be well suited for space heating on-site facilities with a peak demand of 8.6 MW. After assuming a COP of 4 for a hypothetical heat pump recovery system (a reasonable assumption given the results of this thesis), annual savings were estimated to be in the range of $300,000 with an additional 6,000 tons of CO_2 emissions being off-set.

Fanisalek, Bashiri and Kamali (2012) studied the possibility of using waste heat recovered from the potroom exhaust gases of a primary aluminum smelter for employment in a local desalination plant. Potroom exhaust gases are corrosive and abrasive, and the main challenge identified by Fanisalek et al. (2012) in implementing a heat exchanger for recovery of exhaust gas thermal energy was damage to the exchangers leading to a drop in heat transfer rate. A related challenge was the large surface area required by the heat exchangers, since the potroom exhaust is comparatively cool and additional surface area is needed to compensate for failing performance. The potential for recovering heat from the side walls of the aluminum reduction cells which comprise the potline was examined by Barzi, Assadi, & Arvesen (2014), who choose to employ a simpler heat pipe technology. The possibility of employing recovered heat from an aluminum smelter for direct use district heating is examined by Fleer, Lorentsen, Harvey, Palsson, & Saevarsdottir (2010) as well as in Fleer (2010). A total of 55 MW_{th} of recoverable heat is presented by the potline exhaust at the Nordural aluminum smelter in Iceland, a facility producing 270,000 tonnes/yr at the time. The exhaust gases from the potline, at approximately 100°C, were identified as a good fit for coupling with conventional Icelandic district heating systems, which typically operate with 80°C supply and 40°C return. A district heating system based on the Nordural smelter would be capable of providing base load heat and hot water to 16,000 residents of

nearby Akranes. As with Fanisalek et al. (2012), fouling of heat exchangers inserted into the exhaust gas is the major concern blocking implementation. The heat transfer coefficient of a fouling probe inserted into the exhaust stream was found to have dropped by 16% over 11 days.

In Chifeng, China, Fang, Xia, & Jiang (2015) looked to employ waste heat from a copper smelter for a district heating scheme by recovering multiple low grade resources in the range of 20 - 90°C. The completion of many heat recovery devices in September 2013 enabled data collection for the heating season of October 2013 to April 2014. Averaged over a week, the resources considered represented 22.5 MW of waste heat (ranging from a minimum of 11 MW to a maximum of 30 MW), with 80% recovery efficiency. However, production disruptions proved to have a significant impact on the ability of the smelter to supply waste heat (e.g., in a later week the smelter provided an average of only 17 MW due to a plant shutdown). To summarize the benefits of this 2013 to 2014 heating season, Fang et al. (2015) calculated that 108 GWh of heat had been recovered, offsetting 35 thousand tonnes of CO_2 emissions. The thermal efficiency of the smelter experienced a 20 percentage point improvement, rising from 30% to 50%. As a final note, it was predicted by Fang et al. (2015) that the investment in recovery equipment would have a payback period of approximately 4 years.

2.2.3 Sudbury INO Smelter Site

The information in this section is derived from (Loken, 2013) and personal correspondence with employees of the Sudbury INO smelter site. The Sudbury INO smelter site (hereinafter referred to as the smelter site) is an active pyrometallurgical smelting facility located in Sudbury, Ontario, Canada.

Figure 2-5: The Sudbury INO smelter site

The smelter site has a long history within the Sudbury basin mining community, first beginning operations in 1929. Today, the smelter site is a polymetallic operation which processes primarily nickel and copper custom feeds and concentrates, in addition to smaller quantities of cobalt, gold, silver, platinum and palladium. Annual production is in the range of 75,000 tonnes of nickel and 23,000 tonnes of copper, both in a matte form. The matte, a black, crystalline, granular powder comprising many metals, is then transported for further refinement in Quebec and Norway.

An overview of the smelter site process path is as follows: the primary feedstock of the site is nickel and copper bearing sulphide ores in the form of flotation concentrates, as well as a variety of custom feeds including batteries, oil refinery catalysts and reverts. The flotation concentrates are transported to a pair of fluidized bed roasters, where oxidation and de-sulphurization occur. The

product of this roasting process is called calcine. This calcine is transported to an electric arc furnace for separation into slag and matte; the furnace is among the largest electric arc furnaces in the world, and contributes to the smelter site being the 3rd largest consumer of electricity in Ontario. Off-gas from the fluidized bed roasters and some of the off-gas from the furnace are delivered to the Acid Plant, which employs the SO_2 laden off-gas as a feedstock in the production of sulphuric acid. Remaining off-gas from the furnace and processed off-gas from the acid plant are ultimately released to the environment after multiple cleaning stages, including cyclones and electrostatic precipitators. After being tapped from the furnace, the matte is poured into ladles for transport to the converter aisle. In the converter aisle, air is blasted onto the surface of the molten matte to oxidize remaining iron and other impurities. The matte is then cooled and granulated under a water deluge, now comprising 55-60% nickel, 15-20% copper, 2% cobalt, 2% iron and 21% sulphur.

In Loken (2013), an analysis of the potential for employing waste heat from the smelter site to maintain algal growth tanks at an optimal temperature even in winter was undertaken. The waste heat can enable year round algae growth with the goal of employing algae in biodiesel production. Biodiesel is particularly well suited for use underground, and so has applications to Sudbury INO's other facilities. Five major sources of waste heat were identified: the fluidized bed roaster off-gas, the furnace off-gas, the furnace cooling water, the matte granulation cooling water and acid plant cooling water. Together these process units represent approximately 61 MW of waste heat dissipating to atmosphere during smelter operation.

2.3 Summary

Heat recovery is a critical avenue for industrial organizations to pursue, since it provides a clear opportunity to decrease the operating costs, pollution and GHG emissions associated with energy consumption. Within the mining and mineral processing industry, there are many opportunities for low grade recovery from both abandoned and active mine sites. For mineral processing sites, studies

examining heat recovery opportunities highlighted the potential that waste heat offers to the industry. The literature also demonstrates that GSHP based systems employing cooling water can circumvent the rapid fouling associated with heat exchangers directly inserted into exhaust streams. In Fang et al. (2015), heat pumps are employed for recovery of thermal resources at a copper smelter which averaged between 20 - 90 °C. With this in mind, this thesis is based on research that looks at extending the lower temperature range by examining pyrometallurgical process streams of a very low grade (12 - 23°C).

3 Rapid Scoping for Heat Pump Assisted Recovery

Table 3-1: Nomenclature for Chapter 3

Variable	Meaning	Units
b	y-intercept	-
C	capital cost	$
Cap	rated capacity	kW
CC	capital cost	$
COP	coefficient of performance	-
m	Slope	-/°C
O	operating cost	$/yr
P	price	$
PBP	simple payback period	yr
Q	cumulative heating demand	kWh
T	temperature	°C
η	Efficiency	-

Table 3-2: Subscripts for Chapter 3

Subscript	Meaning
a	alternative
c	cooling
e	electricity
h	heating
o	zeroth order

Within the specific research question of this thesis, there is an opportunity to generalize: if process water from a smelter site can be studied for application to heat pump assisted heat recovery, then it is natural to consider the question of any process water from any industrial facility (process water here referring to any site specific water stream). In the literature, the application of heat pumps to different process water streams tends to be examined on a case-by-case basis, with an in depth analysis of the characteristics of the resource, available technology and local pricing. Therefore, the goal of this chapter is to establish a simple method to determine if a thermal resource warrants the

additional expenditure of time, effort and money needed to perform a more thorough engineering analysis, or if the resource is too poor to justify such an analysis. In this way, the simplified method aims to provide a straightforward yes or no answer: if "yes", the resource warrants additional analysis into the potential for recovery via heat pumps, or if "no", the resource is not well suited to recovery of this kind.

One method to arrive at the answer is the critical COP method (Feng & Berntsson,1997). The critical COP method establishes a COP which, if the system were capable of averaging over a full season, would enable the heat pump system to have lower operating costs relative to the current system in the case of a retrofit or the current best contender in the case of a new build analysis. This chapter will expand upon the critical COP method by developing the concept of critical temperature for space conditioning, and then employ these methods to make some general conclusions regarding the applicability of heat pump heat recovery at the smelter site.

3.1 Critical Temperature for Space Heating

The task of establishing a critical COP for space heating applications is tackled in Feng & Berntsson (1997), and their process is presented here. To establish a zeroth order model for critical COP, they reasoned that at a minimum, a heat pump system must attain lower operating cost than the alternative system, since more conventional heating systems tend to have lower capital costs (and in the case of a retrofit the alternative system has zero capital cost). Feng & Berntsson (1997) concluded that the critical COP is that which enables heat to be delivered at the same cost as heat from the alternative system. A system COP above this critical value means cheaper operation. This zeroth order model is expressed mathematically as in Equation 3, where the operating cost of the alternative system, O_a, is equal to the operating cost of the heat pump system, O_{hp}. Substituting the expressions for operating cost and rearranging yields the zeroth order heating model of Equation 5, where the critical COP value,

$COP_{h,o}$, is equal to the price of electricity multiplied by the efficiency of the alternative system divided by the fuel price of the alternative system (P_e,η_a,P_a respectively).

$$O_a = O_{hp} \tag{3}$$

$$\frac{Q_h P_a}{\eta_a} = \frac{Q_h P_e}{COP_{h,o}} \tag{4}$$

$$COP_{h,o} = \frac{P_e \eta_a}{P_a} \tag{5}$$

The assumptions inherent in this model are:

- Maintenance costs are equivalent between the heat pump system and the alternative system; see Section 4.2.2 for an extended discussion of this assumption.
- Air handling costs are equivalent between systems.
- Parasitic operating costs for the heat pump system are negligible.

The expression of Equation 5 establishes a quick check for cost effectiveness, but its applicability to an engineer in the field is fairly limited because it is not immediately obvious what COP an engineer should expect from an arbitrary thermal resource. In other words, while the method of critical COP establishes a minimum performance level, it offers no insight into which resources offer such a performance level. To improve the ease of employing this method, it is possible to relate COP to resource temperature by identifying a function, $COP_h(T)$, valid for a range of sample heat pumps, so that Equation 5 can be expressed as critical temperature, rather than critical COP (Figure 3-1). A set of 13 large capacity GSHPs was employed in establishing this relationship (Table 3-3). In the case of space heating, the critical temperature separates temperatures too cool from those temperatures hot enough to attain economic recovery.

Table 3-3: A summary of GSHPs presented in Figure 3-1

Manufacturer & Line	Model Numbers	Min COP @ 4.44°C	Max COP @ 4.44°C
Waterfurnace Envision Commercial	NLV:160,180,240,300	3.34	5.5
Geosmart Premium G	Single Speed PSC: 60,70	3.73	3.79
Carrier Large Capacity WSHP	HQP: 180,242 VQP: 181,210,240,300,360	4	4.3

Both the Waterfurnace and Geosmart heat pumps are not recommended for operation at temperatures greater than 32.2°C, while the Carrier Line is not recommended for operation at temperatures above 26.6°C. As such, resource temperatures approaching this limit are not well suited to off the shelf GSHP recovery, and alternative methods such a high temperature heat pumps or direct use space heating should be considered.

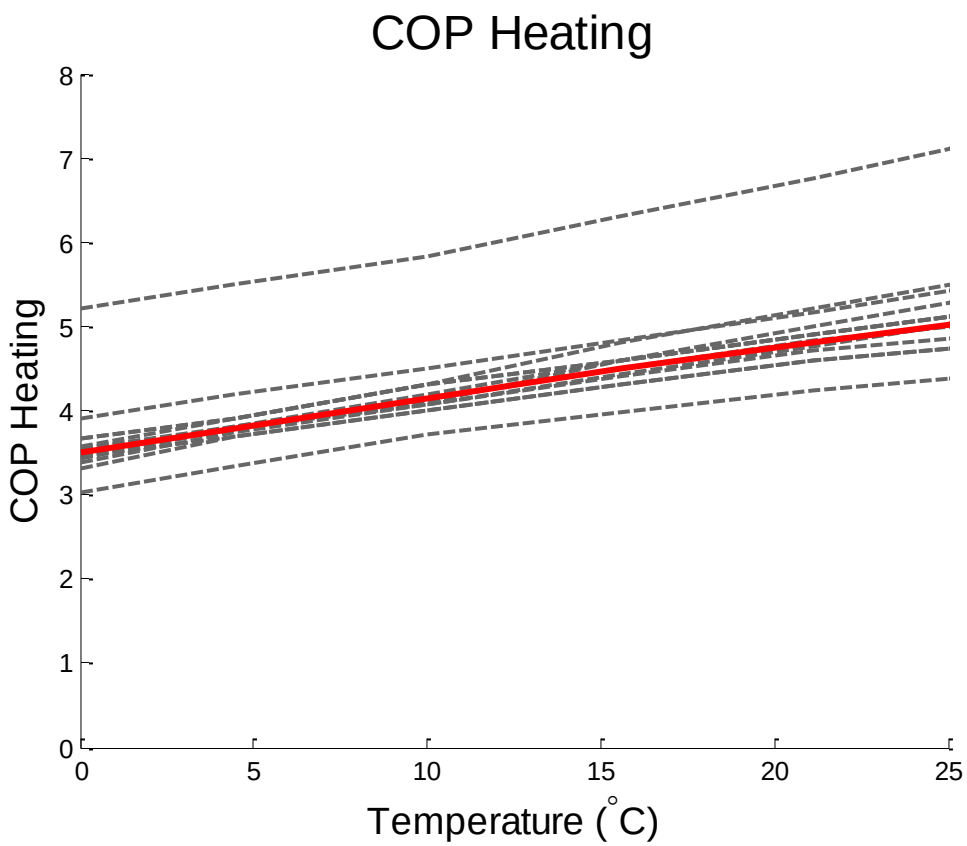

Figure 3-1: COP_h as a function of Temperature

Assuming a linear curve fit for COP_h ($COP_h=m_hT+b_h$), we arrive at Equation 7. The average COP_h of the 13 heat pump models is given by the red line (realizing that the very top grey line for the Waterfurnace Envision NLV240 is an outlier and therefore excluded from the average). The results of the linear curve fit indicate values of m_h=0.06 and b_h=3.52 with R^2=0.9986.

$$m_h T_{h,o} + b_h = \frac{P_e \eta_a}{P_a} \tag{6}$$

$$T_{h,o} = \left(\frac{P_e \eta_a}{P_a} - b_h\right)\frac{1}{m_h} \tag{7}$$

While the zeroth order critical temperature method is easy to employ, it ignores many important factors in accessing cost effectiveness of heat pump systems, the most obvious of which are capital costs. To bring capital costs into their critical COP model, Feng and Berntsson (1996) started with Equation 8, reasoning that the critical COP is that which enables capital costs, *C*, to equal the relative operating savings multiplied by the simple payback period, *PBP*, of the organization performing the analysis. In the event of a retrofit, *C* is equal to the capital costs of the heat pump system. For a new build however, *C* should be equal to the capital costs of the heat pump system minus the capital costs of the next best alternative.

$$PBP = \frac{C}{O_a - O_{hp}} \tag{8}$$

$$PBP = \frac{C}{\frac{Q_h P_a}{\eta_a} - \frac{Q_h P_e}{COP_h}} \tag{9}$$

$$COP_h = \frac{P_e}{\frac{P_a}{\eta_a} - \frac{C}{PBPQ_h}} \tag{10}$$

$$T_h = \frac{1}{m_h}\left(\frac{P_e}{\frac{P_a}{\eta_a} - \frac{C}{PBPQ_h}} - b_h\right) \tag{11}$$

Equation 10 is essentially a slightly modified version of Feng & Berntsson's full critical COP model. As before, the linear relationship between COP and temperature is employed to express the critical COP

as a critical temperature, arriving at the critical temperature model of Equation 11. Additional assumptions of this model are:

- The entire heating demand should be met with a heat pump.
- Thermal resources present enough energy to meet peak heating demand without supplementary heating.

3.2 Critical Temperature for Space Conditioning

While the model of Equation 11 is a fairly capable model, one obvious issue is that it only considers the potential of supplying space heating, whereas one of the key capabilities of a heat pump system is its ability to supply both space heating and space cooling (i.e., space conditioning). Additional assumptions necessary to extend the concept of critical temperature to space conditioning are:

- The conventional system employs an electrically driven air conditioning unit.
- Annual variations in resource temperature are random, so no benefit is realized from higher temperatures in winter and cooler temperatures in summer.

As before, a linear curve fit will be used to convert from a critical COP value to a critical temperature value. Using the same heat pump set as given Table 3-3, it is possible to solve for average values for a linear curve fit of the form $COP_c=m_cT+b_c$, as displayed in Figure 3-2. Values of m_c= -0.11 and b_c= 7.51 were found, with an R^2 of 0.9920. It is important to note that only the Geosmart models were recommended for operation as low as -1.11 °C, while the Waterfurnace models became operable at 4.44 °C and the Carrier models at 10°C. As such, it seems that critical temperatures below around 10°C may be only valid for some heat pump units if the resource is to be used for space cooling. All three manufacturers had operability up to 48.88 °C for cooling, but since the heating functionality of the heat pump can only go up to the 25°C mark, these results are not displayed in Figure 3-2.

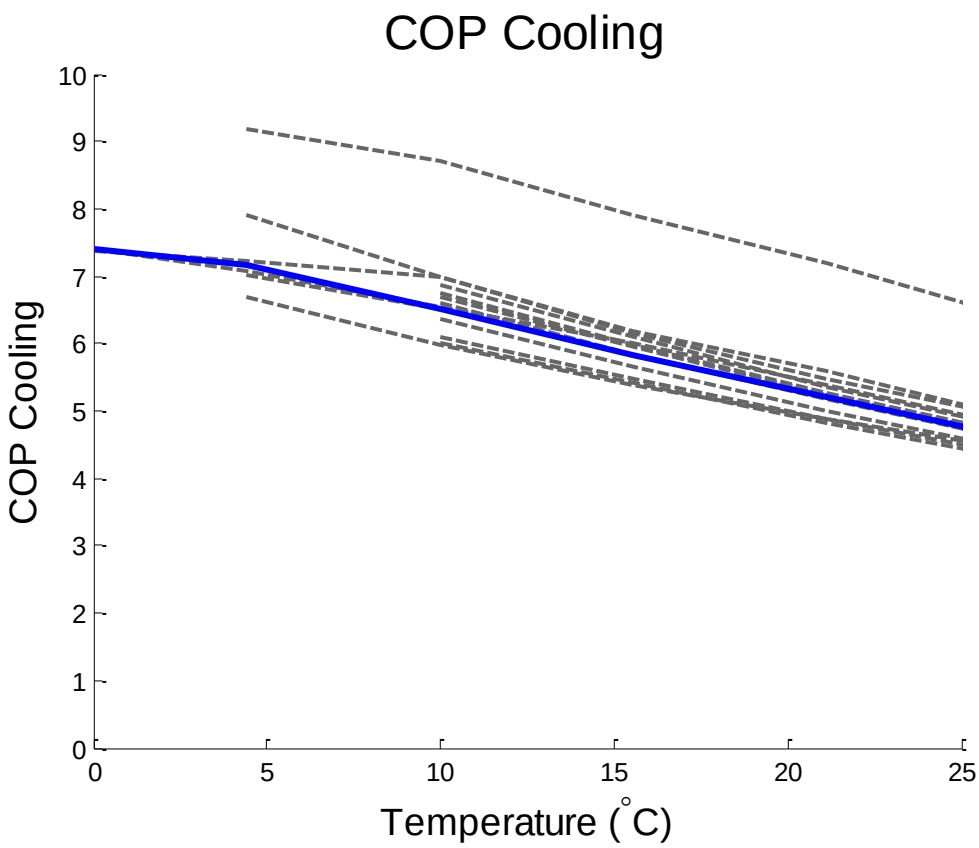

Figure 3-2: COP_c as a function of temperature

Including space cooling in the model yields new equations for the annual operating costs as given below:

$$O_c = \frac{Q_h P_a}{\eta_a} + \frac{Q_c P_e}{COP_a} \tag{12}$$

$$O_{hp} = \frac{Q_h P_e}{COP_h} + \frac{Q_c P_e}{COP_c} \tag{13}$$

Substituting these values into Equation 8 gives:

$$PBP = \frac{C}{\frac{Q_h P_a}{\eta_a} + \frac{Q_c P_e}{COP_a} - \frac{Q_h P_e}{COP_h} - \frac{Q_c P_e}{COP_c}} \tag{14}$$

Substituting in the curve fits and rearranging yields:

$$\frac{C}{PBP} = \frac{Q_h P_a}{\eta_a} + \frac{Q_c P_e}{COP_a} - \frac{Q_h P_e}{m_h T + b_h} - \frac{Q_c P_e}{m_c T + b_c} \quad 15$$

$$\frac{-C}{PBP} + \frac{Q_h P_a}{\eta_a} + \frac{Q_c P_e}{COP_a} = \frac{Q_h P_e}{m_h T + b_h} + \frac{Q_c P_e}{m_c T + b_c} \quad 16$$

We now set the left hand side of Equation 16 as being equal to alpha (α). We then rearrange:

$$\alpha = \frac{Q_h P_e (m_c T + b_c)}{(m_h T + b_h)(m_c T + b_c)} + \frac{Q_c P_e (m_h T + b_h)}{(m_c T + b_c)(m_h T + b_h)} \quad 17$$

$$\alpha = \frac{Q_h P_e (m_c T + b_c) + Q_c P_e (m_h T + b_h)}{m_c m_h T^2 + (m_c b_h + m_h b_c) T + b_c b_h} \quad 18$$

$$\alpha m_c m_h T^2 + [\alpha (m_c b_h + m_h b_c) - P_e (Q_h m_c + Q_c m_c)] T + [\alpha b_c b_h - P_e (Q_h b_c + Q_c b_h)] = 0 \quad 19$$

Equation 19 is now in the canonical form of a second order polynomial ($a_2T^2+a_1T+a_0=0$). The quadratic formula is then employed (Equation 20), where the lesser of the two solutions is the critical temperature for the case that $Q_h > Q_c$ (the greater of the two solutions representing the critical temperature when $Q_c > Q_h$).

$$T = \frac{-a_1 \pm \sqrt{{a_1}^2 - 4 a_2 a_0}}{2 a_2} \quad 20$$

Where we have all the inputs into this equation expressed as:

$$\alpha = \frac{-C}{PBP} + \frac{Q_h P_a}{\eta_a} + \frac{Q_c P_e}{COP_a} \quad 21$$

$$a_2 = \alpha m_h m_c \quad 22$$

$$a_1 = \alpha (m_c b_h + m_h b_c) - P_e (Q_h m_c + Q_c m_c) \quad 23$$

$$a_0 = \alpha b_c b_h - P_e (Q_h b_c + Q_c b_h) \quad 24$$

In the critical temperature model for space heating, the interpretation of critical temperature was simple: resources that present higher temperatures can be recovered economically (within the assumptions of the model) and resources which present a temperature lower than the critical

temperature cannot. In the model for space conditioning however, the situation is slightly more complicated. The interpretation of critical temperature is contingent on the climatic conditions in which the facility to be heated is located, because while COP_h improves with increasing resource temperature, COP_c decreases. This means that for climates where space cooling dominates over space heating ($Q_c > Q_h$), it would be advantageous to increase COP_c at the expense of COP_h by looking for cooler resources. As such, the interpretation of critical temperature is flipped. That is, for climates where demand for space cooling is greater than space heating, critical temperature is that temperature below which "recovery" proceeds economically, while resources above the critical temperature are not cost effective to recover.

To validate the three proposed models (critical temperature for zeroth order space heating, space heating, space conditioning), we can compare the derived critical resource temperatures with the sources under analysis in Chapters 3 and 4. A summary of the parameters employed for these models is given below in Table 3-4, all financial figures in this thesis being given in 2015 American dollars by employing an exchange rate of 0.82 USD per CAD (Bank of Canada, 2015b).

Table 3-4: Summary of values employed

Parameter	Value	Units	Comments
P_e	0.0685	$/kWh	(IESO, 2015)
η_a	0.85	-	Assumed, see (Mei & Nephew, 1987; Self et al., 2013; Watzlaf & Ackman, 2006)
P_a	0.0224	$/kWh	(Union Gas, 2015)
PBP	5	yrs	From Sudbury INO
C	289,000	$	See below
Q_h	4.81	10^6 kWh	See Section 4.1.1
Q_c	0.73	10^6 kWh	See Chapter 5
COP_a	3	-	Assumed, see (Mei & Nephew, 1987; Self et al., 2013)
m_h	0.06	-/°C	
b_h	3.52	-	
m_c	-0.11	-/°C	
b_c	7.51	-	

Establishing an estimate for the capital cost of the system is likely be the most difficult parameter to estimate if wishing to employ the methods given above. While it is difficult to get pricing data for the high-capacity heat pumps presented in Table 3-3, a specific capital cost approximation curve as a function of capacity provided by Staffel et al. (2012) can be used instead (Figure 3-3). The approximation of Staffel et al. (2012) was made applicable to 2015 USD by employing the average exchange rate between British pounds and Canadian dollars for 2012 of 1.58 (Oanda, 2015), a total core inflation from 2012 to 2015 of 3.8% (Bank of Canada, 2015a), and then the exchange rate between CAD and USD.

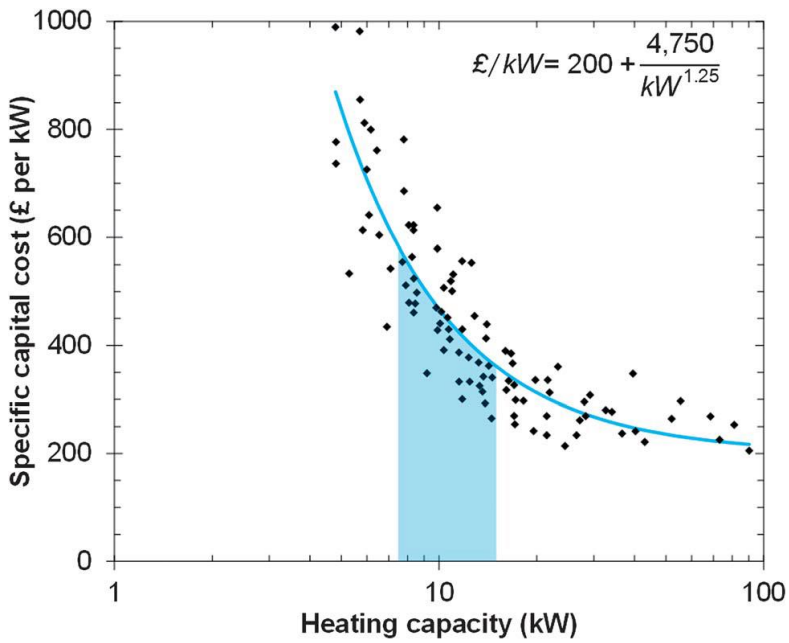

Figure 3-3: Specific capital cost versus heating capacity (Staffel et al.,2012)

$$CC = 1.343 \cdot (200 + \frac{4750}{Cap^{1.25}}) \qquad 25$$

With an expression for specific capital cost at hand, the total capital cost can be calculated as the specific capital cost multiplied by the total capacity to be installed. The peak heating demand of the facility in question is 1000 kW so the capacity of the heat pump system will be set as 1000 kW. It will be

assumed that this capacity will be met with ten 100 kW units so that Equation 25 can be employed without extrapolation. This yields a specific capital cost of 289 CAD/kW for a total capital cost of 289,000 CAD. Equations 5, 10, 7, 11, and 20 can now be employed to arrive at the critical values from each of the five models as in Table 3-5.

Table 3-5: Summary of Critical Values

Model	Critical Value (T in °C)
$COP_{h,o}$	2.6
COP_h	5.9
$T_{h,o}$	-15.3
T_h	39.0
T	19.5

For the zeroth order heating model, only a very low COP_h of greater than 2.6 is required to achieve operating costs lower than the current natural gas system. That corresponds to a necessary resource temperature of -15.3°C, easily attained by all resources at the smelter site. In contrast, the full heating model indicates that a COP_h greater than 5.9 is necessary to achieve payback in five years, which is exceptionally high (although still attainable by the Waterfurnace NLV240 unit). The associated critical temperature is 39.0°C, too hot for any off the shelf GSHP to utilize, and hot enough that any resource attaining that temperature is potentially better suited for direct use space heating. Examining the space conditioning model, we see that the critical temperature has fallen considerably to just 19.5°C, a temperature attained by one of the cooling waters examined later. This decrease in temperature reflects the fact that the GSHPs easily achieve a greater COP than the COP_a of 3, so including space cooling gives the GSHP system an opportunity to earn relative operating savings against their significant capital costs.

4 Recovering Smelter Resources for On-Site Space Heating

Table 4-1: Nomenclature for Chapter 4

Variable	Meaning	Units
AV	annual value	$/yr
CEF_{elec}	carbon emission factor for Ontario electricity	tonnes CO_2/kWh
CEF_{natgas}	carbon emission factor for natural gas	tonnes CO_2/m^3
C_{elec}	consumption of electricity	kWh
C_{natgas}	consumption of natural gas	m^3
COP	coefficient of performance	
c_p	specific heat capacity	J/(kg K)
D	inner diameter	m
d	depreciation rate	%
E	emissions	tonnes CO_2
HC	temperature specific capacity	W
i	interest rate/IRR	%
k	thermal conductivity	W/(m K)
l	section length	m
n	compounding periods	-
Nu	nusselt number	-
P	power demand	W
Pr	prandtl number	-
PV	present value	$
q	heat transfer	W
q_r	recoverable heat	W
Q	volumetric flow rate	m^3/s
R	thermal resistance	K/W
Re	reynolds number	-
t	thickness	m
T	temperature	K
T_r	resource temperature	K
T_a	ambient temperature	K
tr	tax rate	%
W	work	W
ΔP	pressure drop	Pa
ΔT	temperature change	K
η	efficiency	-
ρ	density	kg/m^3

In this chapter, an analysis of the potential for Sudbury INO process heat to be recovered and utilized for space heating applications is presented. The analysis is split into three categories: technical, financial and environmental (specifically CO_2 emissions). The technical analysis is concerned with the engineering performance of the system in terms of work and heat, while the financial analysis will employ market pricing to determine if the engineering performance of the system is commensurate with cost savings compared to the currently installed system. The environmental analysis will examine the CO_2 implications of moving space heating capacity from the current system to the proposed heat pump system.

4.1 Design Methodology

In any heat recovery scheme, it is necessary to properly couple the source of waste heat, the method of recovery, and the end use. For the purposes of this thesis, the method of recovery will be a heat pump system and the end use space heating, so the necessary design steps are:

- Identify sources of waste heat at the smelter site in the right temperature range for heat pumps.
- Identify on-site facilities that present a demand for space heating.
- Determine the best method of enabling a heat pump system to recover this waste heat.

4.1.1 Sources & Sinks

The smelter site, as a pyrometallurgical facility, is replete with opportunities for waste heat recovery. Furthermore, since the original facility was designed and constructed in 1929, heat recovery opportunities were not capitalized upon during initial plant construction.

In addition to the five sources identified by Loken (2013) and summarized in Section 2.2.3, there is another source of waste heat which is well suited to heat pump based recovery, namely the calciner cooling water. The calciner is a process unit that lies at the beginning of the process path for certain custom feeds, such as nickel and cobalt based oil refinery catalysts and batteries, too laden with organic

contaminates to be processed along the regular path. The calciner is essentially a rotary kiln where the organic contaminates are oxidized, and the cooling water used in this process is an additional source of waste heat that can be considered.

Examining the average annual characteristics of the six waste heat streams given above, it can be seen that the temperatures of the roaster and furnace off-gas (both in the range of 350°C) are far greater than appropriate for space heating, so they can be excluded as potential sources (Loken, 2013). The acid plant cooling water at 33°C is certainly low grade, but is above the temperature which can be recovered using commercial GSHP systems as discussed in Section 3.1. As such, while the acid plant does present a valuable stream of thermal energy, it would be better analyzed for use in direct use heating schemes or using a custom, high temperature heat pump solution.

Therefore, the waste heat proposed for recovery is contained in the process cooling waters emanating from the calciner, the matte granulation process and the furnace. The cooling system of the calciner is an open loop using water sourced from a high capacity underground well and then discharged to a nearby water body (Boucher Lake). Coming from an underground well results in relatively high temperatures (10.6°C annual average) and strong annual temperature stability for the calciner cooling water, despite the fact that comparatively little thermal energy is contributed to the cooling water from the actual cooling process (the average temperature at the calciner inlet is 10.6°C while the average temperature at the outlet is 12.3°C). In this way, the calciner blends aspects of geothermal energy recovery and traditional heat recovery, since the majority of the value of the resource is geothermal in nature. In contrast, both cooling systems of the furnace and matte granulation process are closed loops with dedicated cooling towers, and the entirety of their recoverable heat is directly removed from the process units they cool. A full year of data was collected for the furnace and matte granulation processes from June 2013 to May 2014 and for the calciner from January to December of 2012. A summary of the annual characteristics of these sources can be seen in Table 4-2. Recoverable heat flow

was calculated on a monthly basis using Equation 26, where T_a was taken as the ambient temperature of Sudbury (Figure 4-1).

$$q_r = \rho Q c_p (T_r - T_a) \quad (26)$$

Table 4-2: Annual average temperature and flow rate for sources

	Calciner	Furnace	Matte Gran
Temperature (°C)	12.3 ± 0.7	18 ± 2	23 ± 3
Flow Rate (L/s)	19 ± 3	23.6 ± 0.9	$(5 \pm 1)\times 10^2$
Recoverable heat flow (kW)	$(7 \pm 9)\times 10^2$	$(1 \pm 1)\times 10^3$	$(4 \pm 2)\times 10^4$

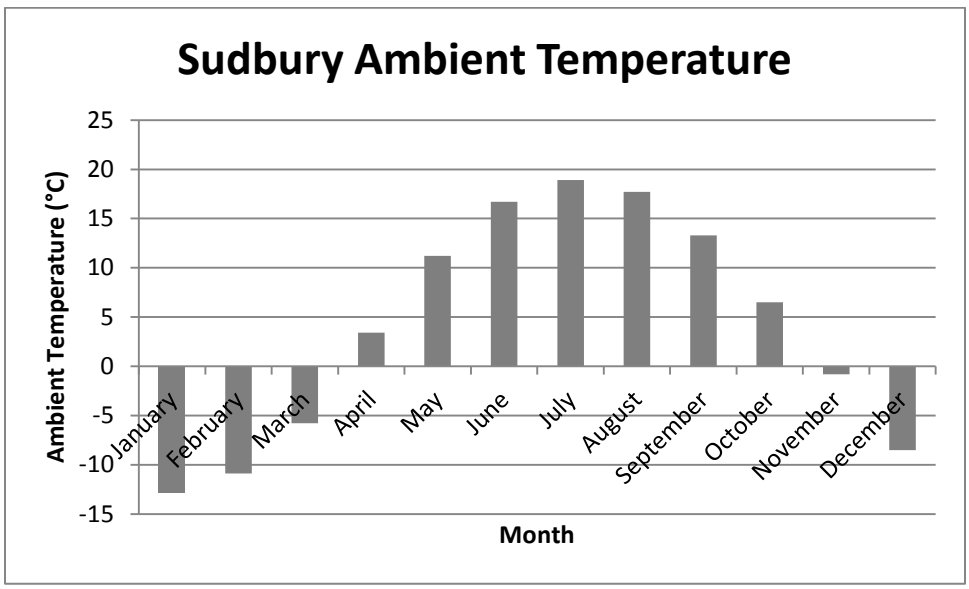

Figure 4-1: Monthly ambient temperature of Sudbury (NASA, 2016)

Now that potential sources have been identified, it is necessary to identify facilities with space heating needs that are well suited to coupling with the identified sources (i.e., total heating demand is similar to total heat available, the source is in close proximity, etc.). There are many such facilities on and adjacent to the smelter site that meet the requirement, such as the administrative building, the shop floor, the XPS facility and, in a hypothetical scenario, a district heating system for the town of

Falconbridge. While there are many opportunities, the XPS facility in particular was deemed well suited for the recovery of the identified sources.

The XPS facility is a two story office and laboratory complex that acts as a mineralogical and chemical testing facility to meet the assaying needs of the larger Glencore Corporation.

Figure 4-2: The XPS facility (XPS, 2016)

It was estimated that the XPS facility has a floor space of 10,000 m^2. The facility currently employs a natural gas furnace with a forced air (ducted) distribution system to meet their space heating needs. This furnace was installed in 1994, and as such is beginning to near the end of its expected economic life. The average annual space heating demand of the XPS facility is 510 kW, with a peak heating demand in December of 1000 kW (Figure 4-3).

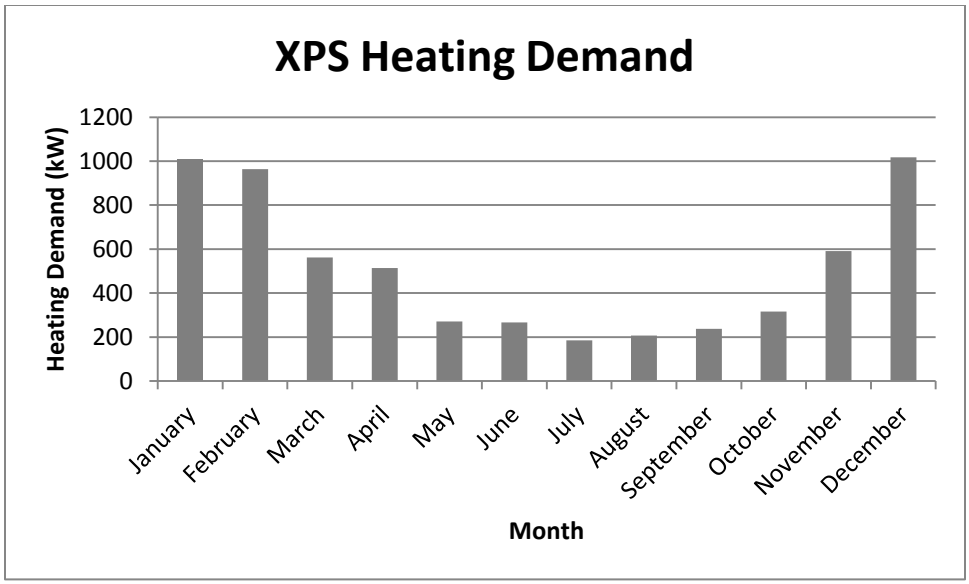

Figure 4-3: Monthly heating demand of the XPS facility.

4.1.2 Design Concept

With the sources and a potential sink identified, it is possible to design a system by which heat can be recovered at the source and delivered to the sink. A formal design objective to help guide the design concept was established: "design a heat recovery system employing heat pumps as the means of recovery which results in both minimum cost and minimum annual CO_2 emissions".

Ideally of course, the design concept will present costs and annual emissions which are less than the currently installed natural gas system as well. In addition to this primary design objective, there are secondary design objectives which should be met by this system:

- The system must be minimally invasive to pre-existing processes during operations.
- The system must be minimally invasive to the operation of the site during construction.
- The heat pumps employed should be water-to-air systems, as the cost of retrofitting the XPS facility with a hydronic distribution system is far too great.

Multiple system designs can now be proposed which can reasonably be expected to meet the design requirements identified above. Two key design parameters to consider are:

- Open vs. Closed Loop: an open loop system is simpler, cheaper to construct and more efficient, but only possible if water quality is high enough to ensure limited fouling and corrosion of the heat exchangers, a common problem identified in Section 2.2.2 . It is also likely that open loop designs will be less invasive than closed loop, depending on the form of closed loop heat exchanger chosen.
- Piping network above vs. below ground: burying the network is more expensive, invasive to install and more difficult to service in the event of pipe failure. Above ground networks however can be more invasive over their entire life to surface activity, and experience greater heat losses.

The heat pump database of Table 3-3 contains a variety of water-to-air GSHPs which are suitable for consideration. The key requirement here was a comparatively large capacity which could be expected to meet the heating demand of the XPS facility with only a few units, as too many units becomes ultimately untenable in terms of occupied floor space within the XPS boiler room. It will be seen later that some of the heat pump units have too low a capacity and lead to an unrealistically large number of units; it was deemed that these units should be included in the analysis to demonstrate the potential for the performance level of off the shelf GSHPs, even though the some units have too low a capacity to be realistically employed at XPS. The total number of heat pumps in the bank will be left as a decision variable, and it will be seen later on that this decision is critical to the overall performance of the heat pump system.

After considering the possible design parameters, it was concluded the best system design is an open loop, above ground piping network that directly removes “warm” cooling water from a given source and delivers it to the sink (XPS facility). On arrival at the sink, the cooling water flow is split into

equal portions for delivery to each of the individual heat pumps within the bank, which are arranged in parallel. A parallel arrangement is necessary and desirable: necessary in that the flow capacity of the heat pumps interior heat exchangers is much less than the flow delivered and also desirable in that in the parallel arrangement, each heat pump is exposed to water at its highest temperature (thus enabling a greater COP), whereas in the series arrangement, the temperature drop through each consecutive unit results in a reduced average COP for the bank.

After the process water has passed through the bank, it is delivered back to the point of origin from where it was removed. While this is theoretically unnecessary for the calciner cooling water as it is dumped into the environment, there is no location for disposal in the vicinity of the XPS facility, so it will be returned it to its regular process path.

The assumptions which have been made in choosing the proposed design concept are:

- Water quality is high enough to operate the heat pumps without undo maintenance; fouling of the piping network due to corrosion, scaling or deposition is minimal over system lifetime. This assumption is validated, despite the common corrosion issues identified in Section 2.2.2, because the cooling waters are separated from direct contact with contaminates by the walls of their cooling loop heat exchangers.
- Piping routes are realizable; there are no regulations, governmental or internal to Sudbury INO, which would disallow the proposed piping network.
- The boiler room of the XPS facility is accessible to the piping network.
- The furnace room is large enough in size to support multiple heat pump units.
- The existing forced air distribution is compatible, or can be made compatible, with the heat pumps.
- Supplementary heating can be delivered by the existing natural gas system.

- The piping network will never freeze and burst. The temperature drop through the heat pump system is too low to cause freezing even on the coldest days, but machinery failure and subsequent stagnation of the flow could present a freezing risk. It is assumed that redundancy in the number of heat pumps and circulation pumps renders this risk negligible.

4.2 Analysis Methodology

In this section, the mathematical framework for determining the performance of the proposed heat pump system is presented, the framework having been implemented in MATLAB (MathWorks, 2016). It is interesting to note the generality of the following framework; the only data from Sudbury INO is source and sink characteristics and the specific form the piping route needs to take. As such it is easy to adapt this framework for analysis outside of mineral processing, such as mining, power generation or manufacturing.

4.2.1 Technical Analysis

The first stage of the technical analysis is to determine the temperature change and pressure drop which takes place along the piping connection between the process water source and the XPS facility. This modeling was performed using a finite difference method (FDM) scheme, in which the full piping run was divided into many discrete, equal lengths of pipe. Convergence tests are used to validate that each individual length is short enough for the process water within the pipe to be approximated as being at a constant temperature.

Heat transfer across the pipe boundary is taken to be steady state, one dimensional, constant in thermo-physical properties and the process water has a uniform temperature distribution with respect to the cross sectional area of the flow. The last assumption of uniform temperature is validated by a Reynolds number within the turbulent regime. The piping route characteristics for the three sources under consideration are given below in Table 4-3. Standard 90° elbows and standard tees are taken to have an effective Length to diameter ratio of 30 and 20, respectively (Munson, Young, Okiishi, &

Huebsch, 2009). It was found that minor losses account for between 15 and 22 percent of total pressure loss in the piping route.

Table 4-3: Summary of Piping route characterisitics

	Calciner	Furnace	Matte Gran
Estimated Length (m)	200	250	250
# of standard 90° elbows	10	18	22
# of standard tees	2	2	2

Piping connections are assumed to to be fabricated from American standard schedule 40 4" diameter stainless steel pipe. This diameter was chosen because it placed fluid velocities in the range of 1-3 m/s, which is recommended for long piping runs by Holland & Bragg (2002). Any decrease in temperature that occurs over the supply side of the piping run results in a decrease in COP, so the supply pipes are to be wrapped in 3" thick fibreglass insulation.

With this piping design, the equivalent thermal resistance network between the process water and ambient air is given in Equation 27, where the total thermal resistance is simply the sum of the resistance to internal convection, resistance to conduction through the steel pipe (k=50 W m^{-1} K^{-1}) and fibreglass insulation (k=0.038 W m^{-1} K^{-1}) and resistance to external convection. These resistances are denoted by $R_{interior}$, R_{pipe}, $R_{insulation}$, $R_{exterior}$, respectively.

$$R_{total} = \left(R_{interior} + R_{pipe} + R_{insulation} + R_{exterior}\right) \quad 27$$

The internal convection coefficient of the pipe is found using the Nusselt number, $Nu_{interior}$, correlation provided by (Incropera, De Witt, Bergman, & Lavine, 2006), which employs the Reynolds number of the pipe flow, $Re_{interior}$, and the Prandtl number of water, Pr_{water}. Equation 28 allows the equivalent heat transfer coefficient $h_{interior}$ to be calculated using the thermal conductivity of water, k_{water}, and the internal diameter of the pipe, D:

$$Nu_{interior} = 0.023(Re_{interior})^{0.8}(Pr_{water})^{0.4} \quad 28$$

$$h_{interior} = \frac{(Nu_{interior})(k_{water})}{D} \quad 29$$

The interior convection coefficient is then expressed as an equivalent thermal resistance for a single section of the FDM analysis, where l denotes the length of the section:

$$R_{interior} = \frac{1}{h_{interior}\pi D(l)} \quad 30$$

For the total thermal resistance expressed in Equation 27, it is necessary to calculate the resistance to convection between the insulation and ambient air. The exterior Nusselt number $Nu_{exterior}$, is calculated using the correlation of Equations 31 and 32 with the exterior Reynolds number $Re_{exterior}$ and the Prandtl number of air Pr_{air} (Incropera et al., 2006). Equation 33 is then employed as before to obtain the convection coefficient at the exterior of the pipe, $h_{exterior}$, where k_{air} denotes the thermal conductivity of air and t_{pipe}, $t_{insulation}$ the thickness of the pipe wall and insulation, respectively:

$$Nu_{exterior} = C({Re_{exterior}}^{m})({Pr_{air}}^{0.33}) \quad 31$$

$$(C, m) = \begin{cases} (0.989, 0.330) \ \ if\ Re < 4 \\ (0.911, 0.385) \ \ if\ Re < 40 \\ (0.683, 0.466) \ \ if\ Re < 4000 \\ (0.193, 0.618)\ if\ Re < 40000 \\ (0.027, 0.805)\ if\ Re > 40000) \end{cases} \quad 32$$

$$h_{exterior} = \frac{(Nu_{exterior})(k_{air})}{D + 2(t_{pipe}) + 2(t_{insulation})} \quad 33$$

Once the total thermal resistance across the pipe has been determined, Equation 34 is used to determine the heat transfer across a single length of pipe, q, given the temperature of ambient air surrounding the pipe T_{out} and the temperature of process water within, T_{in}:

$$q = \frac{T_{out} - T_{in}}{(R_{interior} + R_{pipe} + R_{insulation} + R_{exterior})} \quad 34$$

Over each length of pipe, T_{in} is assumed constant, allowing the heat transfer between the process water and ambient air to be determined. Employing Equation 35 allows the temperature of the process water as it moves from one section of pipe, $T_{in,n,}$ to the next, $T_{in,n+1}$, to be calculated. Q denotes

the volumetric flow rate of the process water, $c_{p,water}$ the specific heat capacity of water and ρ_{water} the density of water:

$$T_{in,n+1} = \frac{q}{Q c_{p,water} \rho_{water}} + T_{in,n} \qquad 35$$

Repeating this process across the total length of piping allows a discrete temperature profile to be determined. This temperature profile allows the total temperature loss from source to sink to be determined. Whilst different systems produced different temperature drops along the supply piping, averaged over a year most systems tended to produce a drop on the order of 0.05 to 0.1 °C/km.

In addition to the temperature loss that occurs in the piping, pressure losses also need to be determined to help assess the power demand of pumping the process water. The equations employed in this calculation are well known and well described elsewhere (Munson et al., 2009). The pumping work, W_{pump}, as a function of pressure loss, ΔP, volumetric flow rate and pump efficiency, η_{pump}, can be then be approximated by:

$$W_{pump} = \frac{Q \Delta P}{\eta_{pump}} \qquad 36$$

The operation of individual heat pumps can be analyzed given the temperature loss has been accounted for. In Chapter 3, curve fits for COP_h were employed. In this chapter, because each heat pump is considered individually, linear interpolation between each heat pumps data points is used. Using the temperature of the cooling water in conjunction with manufacturer's specification data allows the temperature dependent *COP* and power demand, *P*, to be calculated.

The product of these two numbers is the capacity of the heat pump at the given temperature, denoted *HC*:

$$HC(T,Q) = COP(T,Q) \cdot P(T,Q) \qquad 37$$

With the total number of heat pumps within the bank left as a variable, it is necessary to employ a supplemental heating system to meet demand in excess of what can be met by the chosen number of

heat pumps. The supplemental heating system in this case is the natural gas furnace currently installed at XPS. With the number of heat pumps variable, it is possible that the majority of demand will in fact be met by the supposed "supplemental" system. In this sense, supplemental heating is more than just a peak shaving method for the coldest months of the year, but rather a complementary heat source that can work with the heat pump system to yield a cost minimized design, as we shall see later.

The total monthly heating delivered by the bank is calculated as the product of the heating capacity and the total number of heat pumps. The pressure drop though the entire heat pump bank is equal to the pressure drop through a single heat pump at the given flow rate and temperature due to the parallel arrangement of the heat pumps. Subsequent power requirements necessary to overcome this internal pressure drop are calculated by reapplying Equation 36. Applying Equation 38 allows the temperature drop through the heat pump bank to be calculated, $\Delta T_{heat\ pump}$ where the heat removed from the process water is equal to the heating capacity minus the power requirement of a single heat pump:

$$\Delta T_{heat\ pump} = \frac{(HC - P)}{QC_{p,water}\rho_{water}} \qquad 38$$

The process water, upon leaving the heat pump bank, is returned to its source through a piping run that lies immediately adjacent (although there is no mutual interaction effects) to the supply side piping run. Pressure loss and pumping work associated with this return piping is calculated using the same methods as outlined above.

4.2.2 Financial Analysis

Financial results are determined by applying cost and pricing data to relevant technical results. The financial analysis was performed as a replacement analysis, where the two options under consideration were the current natural gas system and the proposed heat pump system. The costs for both systems are expressed as an annuity by amortizing capital costs over expected system life. The

choice to express costs as an annuity was made because annuities are a conceptually natural tool to use when considering space heating, whose costs are always expressed on a per unit time basis.

The capital costs of both the heat pump systems are annuitized by employing Equation 39, where *AV* is the annuity of the asset, *PV* is the present value of the asset, *i* is the hurdle rate and *n* is the number of compounding periods:

$$AV = PV\frac{i(i+1)^n}{(i+1)^n - 1} \qquad 39$$

In this instance, the number of compounding periods is the expected system lifetime of the capital asset in question. A review of parameters used in this section is given in Table 4-4.

Table 4-4: Summary of Financial Parameters

Parameter	**Value**	**Units**	**Comments**
Price of electricity	6.85	$/kWh	(IESO, 2015)
Price of nat. gas	2.24	$/kWh	(Union Gas, 2015)
efficiency of nat. gas furnace	85	%	Assumed, see (Mei & Nephew, 1987; Self et al., 2013; Watzlaf & Ackman, 2006)
hurdle rate	15	%	From Sudbury INO
combined pump efficiency (η)	70	%	Assumed, see (Munson et al., 2009)
Life of Heat Pump	20	yrs	Commonly cited in literature for GSHP
Unit capital cost for pipe	100	$/m	(Lindley & Floyd, 1993)
Unit capital cost for insulation	30	$/m	(Buy Insulation Products, 2016)
depreciation rate (*d*)	30	%	(Canada Revenue Agency, 2015a)
tax rate (*tr*)	26.5	%	(Canada Revenue Agency, 2015b)

The annual cost of the heat pump system is broken down into two categories; the capital costs associated with purchasing the heat pumps, piping and circulation pumps, and the operating costs associated with pumping the cooling water, powering the heat pumps and providing maintenance. Expected maintenance costs for both systems are difficult to predict with accuracy, especially considering the natural gas system is nearing the end of its expected system life. Whilst removing maintenance costs from the analysis is a source of inaccuracy in the results, the purpose of this chapter

is to determine if space heating can be economically applied by a heat recovery heat pump system. This purpose is unaffected by ignoring maintenance costs, since natural gas systems tend to present greater maintenance costs (Cane, Morrison, & Ireland, 1998; Cane & Garnet, 2000), so the analysis is made more conservative by shifting the economics in favor of a natural gas system.

Capital costs of the individual heat pumps are determined using the correlation of Equation 25. The capital cost of the heat pump system is not born entirely by the smelter, because the depreciation of these assets can be claimed as a tax deduction. These tax savings can be calculated as a present worth using Equation 40 knowing the depreciation rate, *d,* and the tax rate*, tr*.

$$PV_{depreciation} = \sum_{n=1}^{life\ of\ asset} (tr)(Cap)[(1-d)^{n-1}d]\,(1+i)^{-n} \qquad 40$$

4.2.3 Environmental Analysis

Switching from natural gas to a heat pump system is expected to result in significantly lower annual CO_2 emissions, *E*, if consumption of natural gas is replaced by consumption of electricity, which as in Ontario, is heavily sourced from low carbon alternatives. To examine the impact of switching, annual CO_2 emissions for both systems are characterized using emissions factors. An emission factor is a number that quantifies how many tonnes of CO_2 are released given unit consumption of some material. A carbon emission factor, *CEF*, of 0.001879 tonnes of CO_2/m^3 of natural gas consumed at STP (Environment Canada, 2013) is employed along with the annual consumption of natural gas, C_{natgas}, to quantify the total annual emissions for the existing natural gas system, as in Equation 41:

$$E_{natgas} = CEF_{natgas}C_{natgas} \qquad 41$$

In Ontario, the opportunity for reducing emissions via heat pump usage is significant due to the low carbon intensity of Ontario's electricity thanks to significant usage of hydroelectricity, nuclear and wind electricity production. As mentioned in Chapter 1, heat pump emissions are as renewable as the method of electricity generation used to source their power. For Ontario, electricity produced is treated

as releasing emissions of 0.04 kilograms of CO_2 per kWh (Aegent Energy Advisors, 2015). Equation 42 sums the emissions generated by natural gas fired supplemental heat and from heat pump electricity consumption, where C_{elec} denotes the total consumption of electricity in kWh:

$$E_{heat\ pump} = CEF_{elec}C_{elec} + CEF_{natgas}C_{natgas} \quad 42$$

For this analysis, it is assumed that the hidden emissions associated to the heat pump system from the manufacture and transportation of its components is negligible in comparison to the operating emissions. It is also assumed that any greenhouse gas effects from leaked refrigerant are negligible in comparison to the operating emissions.

4.3 Results & Discussion

With three sources under examination and thirteen heat pump options, a total of 39 heat pump systems were available for analysis. While all possible systems were analyzed, only the heat pump model from each brand that resulted in the lowest system cost for a given source is presented to make results more concise. To discover the lowest system cost possible, the preceding model was optimized by exhaustive enumeration with respect to the number of heat pumps in the heat pump bank, increasing by integer values from one to the maximum number of heat pumps necessary to fully meet peak demand. This result is displayed Figure 4-4. The total annual cost is presented as the sum of four cost categories: powering the heat pumps, powering the circulation pump, annuitized capital costs and cost of supplemental natural gas. In all of the following figures, Cal., F. and MG. are calciner, furnace and matte granulation respectively, whilst WF, G and C denote Waterfurnace, Geosmart and Carrier respectively.

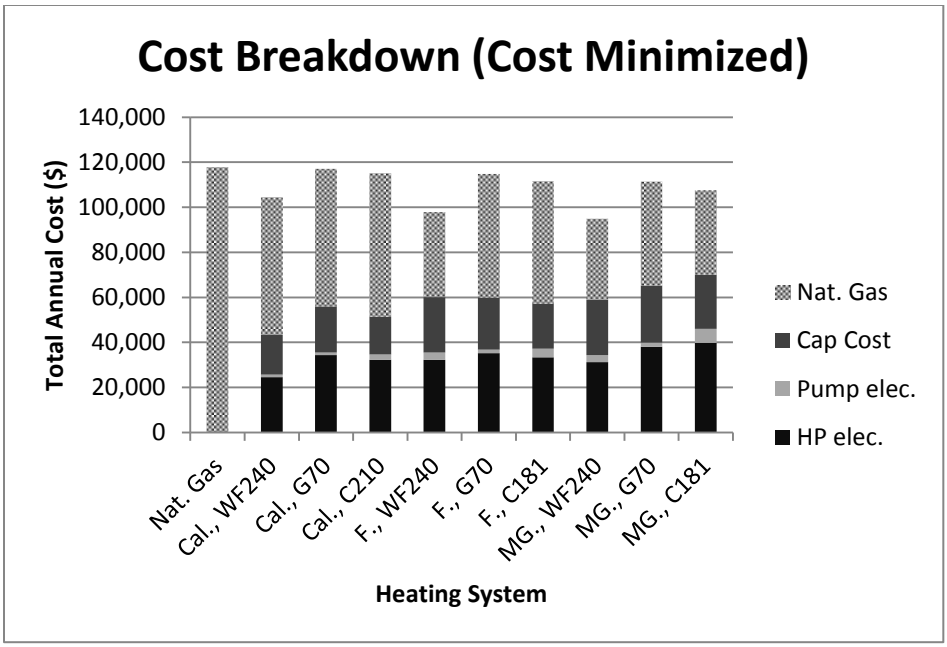

Figure 4-4: Total annual cost for cost minimized systems

For all three sources, the heat pump system with the lowest annual cost utilized the Waterfurnace Envision NLV 240 heat pump, which displays an exceptionally high COP (Table 4-5). Since both the matte granulation and furnace process water is sourced from cooling towers, it is possible to quantify the total cooling demand of the towers met by the temperature drop engendered by the heat pump system. The percent reduction is displayed in Table 4-5. This reduction has the potential to generate additional savings for the heat pump systems by reducing pumping costs associated with the cooling towers, or improving process efficiency as a result of a decreased coolant temperature.

Table 4-5: Number of heat pumps, average COP and percent of tower cooling load extracted for the systems of Figure 4-4

Source and Model	# of heat pumps in bank	Average COP	% Cooling Load
Calciner; Waterfurnace Envision NLV 240	4	6.0	-
Calciner; Geosmart Premium G 70	12	4.3	-
Calciner; Carrier VQP 210	4	4.4	-
Furnace; Waterfurnace Envision NLV 240	6	6.5	14.5
Furnace: Geosmart Premium G 70	13	4.6	10.6
Furnace; Carrier VQP 181	6	5.0	10.9
Matte Gran.; Waterfurnace Envision NLV 240	6	6.9	6.4
Matte Gran.; Geosmart Premium G 70	15	4.9	5.2
Matte Gran.; Carrier VQP 181	8	5.3	5.9

For some of the cost minimized systems the largest share of the annual cost is natural gas. This outcome is caused by the high initial cost of a heat pump and the subsequent need for high usage to recover this initial cost. As a single heat pump has lower operating costs than the equivalent capacity of natural gas, there is a minimum time of operation at which point the heat pump will pay back its higher initial costs and begin to generate relative savings. However, the marginal heat pump, installed to meet peak heating demand during only the coldest months of the year will often fail to attain the necessary usage over a reasonable payback period. This dynamic leads to cost optimized heat pump systems that rely on significant supplemental heat.

With so many heat pumps in the bank, the validity of assuming equal maintenance cost between systems becomes questionable, as annual maintenance will increase as more heat pumps require servicing. Therefore, systems with fewer heat pumps in the bank should be favored in the case of real world implementation, as they will present lower maintenance costs.

In Chapter 3, the critical temperature for space heating was determined as 39.0°C, a temperature which was not achieved by any of the resources under examination. However, the critical temperature model relied on the assumption that the entirety of the heating demand would be met by

heat pumps. As such, the results of the preceding analysis suggest that it would be prudent to develop critical temperature models with the capacity to reflect only partial heating supplied by heat pumps.

In Figure 4-5, the annual CO_2 emissions associated to the systems of Figure 4-4 are displayed. It can be seen that all heat pump systems offer a reduction in emissions, with the same system that achieved the lowest system cost simultaneously achieving the highest reduction (62%).

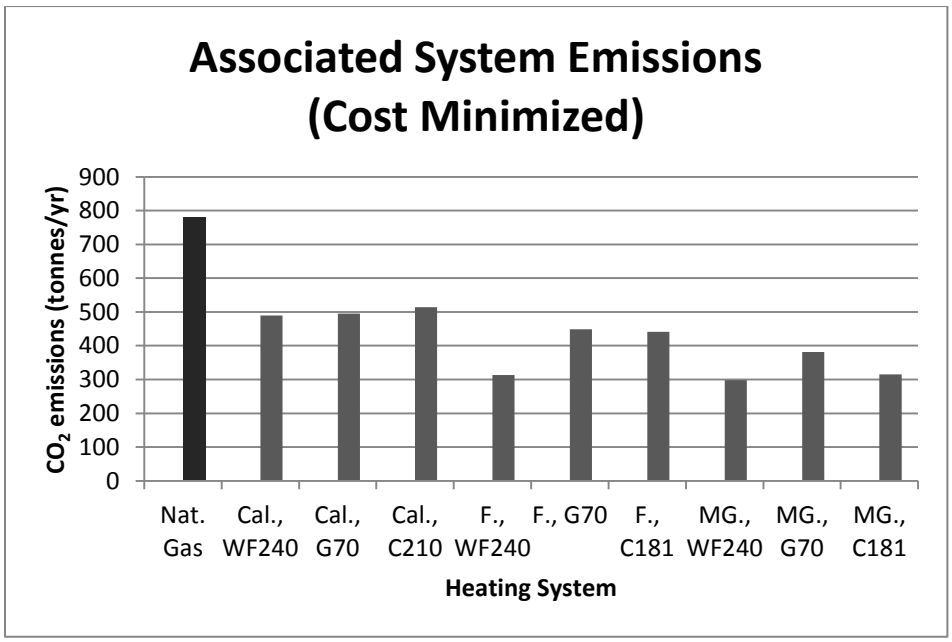

Figure 4-5: CO_2 emissions generated by the heat pump systems of Figure 4-4

If minimizing yearly CO_2 emissions is prioritized over a reduction in costs, then the heat pump system will maximize the amount of space heating supplied by the heat pumps. A unit of heating supplied from the heat pumps will have a lower associated CO_2 emissions than a unit from natural gas. This can lead to an increase in costs that exceed the natural gas system, but the reduction in CO_2 emissions is significantly greater than the comparative cost minimized systems. The ability of the heat pump system to meet heating demand is limited by the thermal energy contained in the process cooling water, so even a CO_2 minimized system may still rely on significant supplemental heating. This is demonstrated in Figure 4-6, which shows results from those systems that generated the lowest CO_2 emissions.

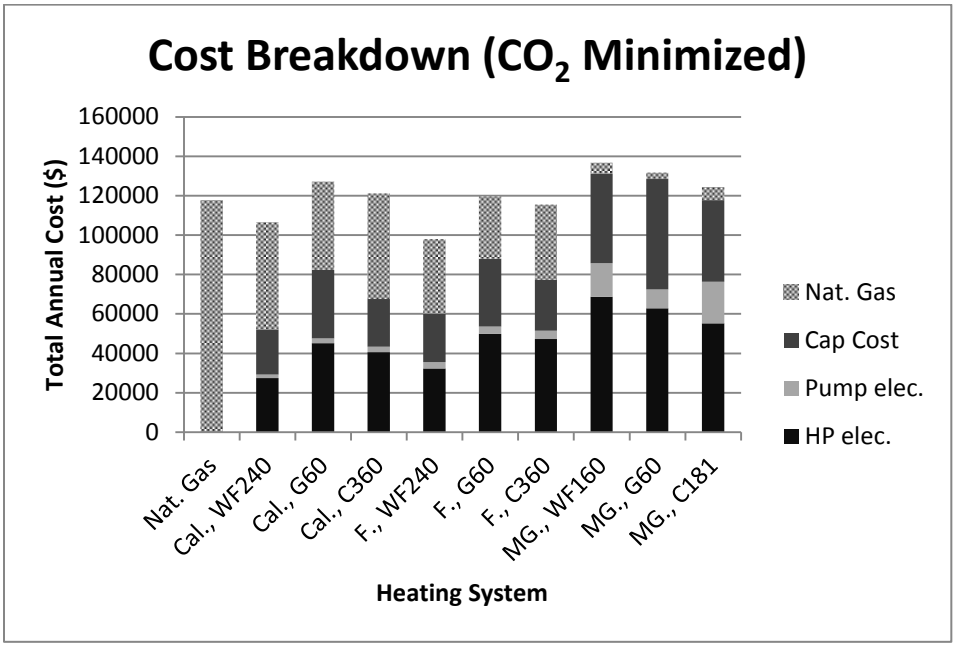

Figure 4-6: Total annual cost for CO_2 minimized systems

Compared to Figure 4-4, consumption of natural gas for supplemental heating as a share of total costs has been significantly reduced for matte granulation sourced systems. This change is less pronounced for calciner and furnace sourced systems due to the fact that the thermal energy delivered by these process streams is insufficient to meet the heating demand of the facility in the colder months of the year. This leads to a greater need for supplemental heat and a subsequent decrease in total costs. The total number of heat pumps within the bank and average COP across the year is given in Table 4-6.

Table 4-6: Number of heat pumps, average COP and percent of tower cooling load extracted for the systems of Figure 4-6

Source and Model	# of heat pumps in bank	Average COP	% Cooling Load
Calciner; Waterfurnace Envision NLV 240	6	6.0	-
Calciner; Geosmart Premium G 60	27	4.2	-
Calciner; Carrier VQP 360	4	4.1	-
Furnace; Waterfurnace Envision NLV 240	6	6.5	14.5
Furnace: Geosmart Premium G 60	25	4.5	14.4
Furnace; Carrier VQP 360	4	4.4	13.3
Matte Gran.; Waterfurnace Envision NLV 160	24	4.3	7.9
Matte Gran.; Geosmart Premium G 60	46	4.8	8.3
Matte Gran.; Carrier VQP 181	17	5.3	8.2

The emissions associated to the heat pump systems of Figure 4-6 are presented in Figure 4-7. For matte granulation sourced systems, CO_2 emission reductions as great as 97% are possible. The only CO_2 minimized system to achieve lower total annual costs, the calciner sourced Waterfurnace Envision NLV 240 system, generated an emissions reduction of 44%.

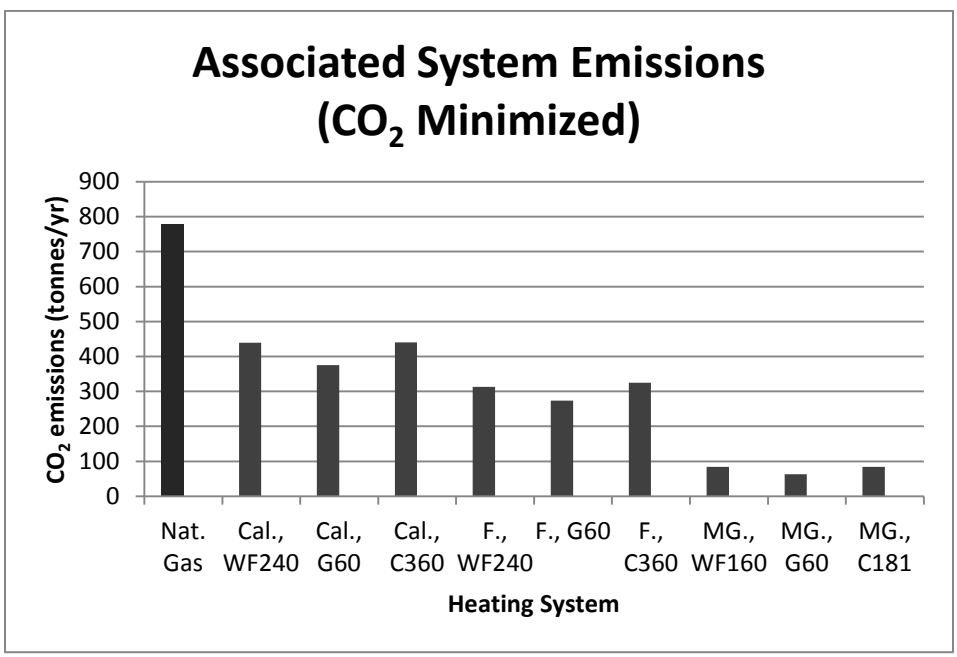

Figure 4-7: CO_2 emissions generated by the systems of Figure 4-6

Taking the lowest cost heat pump system, namely the Waterfurnace Envision NLV 240 based system sourcing process water from matte granulation as a base case ($94,977 annuitized cost), a sensitivity analysis was performed (Figure 4-8). The cost of electricity, the price of natural gas, the efficiency of the natural gas furnace, the capital costs of purchasing and installing the heat pumps and the average temperature of the process water are varied from -20% to 20% of the base case. These parameters were chosen because they had the greatest influence on total annual costs.

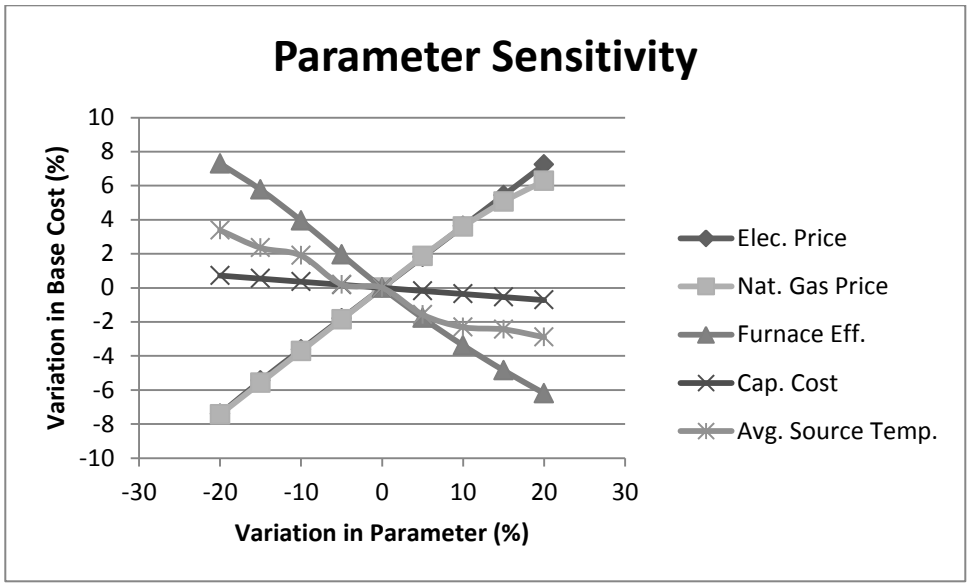

Figure 4-8: Sensitivity Analysis of five key parameters for the Waterfurnace Envision NLV 240 system sourcing water from matte granulation

5 Recovering Smelter Resources for On-Site Space Heating and Cooling

In the preceding chapter, recovery of smelter site resources for space heating alone was considered. In general, however, there is no need to restrict the analysis to only supplying space heating. That is, as seen in the critical temperatures of Table 3-5, considering space cooling can be a means to further improve the business case of a heat pump system. While space heating may seem more intuitive in the context of heat recovery, and the domination of space heating in Sudbury's climate certainly prioritizes it, restricting only to space heating misses the full opportunity of a heat pump unit. In fact, it is interesting to note that in June in Sudbury, the ambient temperature will achieve an average of 19°C. This means that a traditional air source air conditioning unit will face a greater temperature lift than that faced by a heat pumps unit installed to operate off either the calciner or furnace at approximately 12°C and 18°C respectively. In this way, while both of the sources are notionally valuable for their elevated temperature, they simultaneously provide value as a cooling resource since they are colder than ambient conditions when demand for space cooling is high.

With this in mind, it is interesting to consider the possibility of replacing the current XPS space conditioning solution with a single heat recovery heat pump, operating off the same resource during heating and cooling mode, globally benefiting from elevated temperatures of the thermal resource, whilst simultaneously benefiting from the relative cool of the resources during the summer months. The current space conditioning solution of the XPS facility is a natural gas furnace and an air source air conditioner unit, assumed to have an average annual COP of 3. The peak cooling demand of the XPS facility is 290 kW in July, with an average cooling demand over the year of just 80 kW (Figure 5-1). This compares to an average heating demand of 510 kW, implying space cooling has only 16% the relative importance of space heating. It is important to note that the cooling demand presented does not take into account heat production by the heat pumps and circulation pumps housed in the XPS facility

machine room, which implies cooling demand may be underrepresented. A drafted the machine room would help to mitigate this concern.

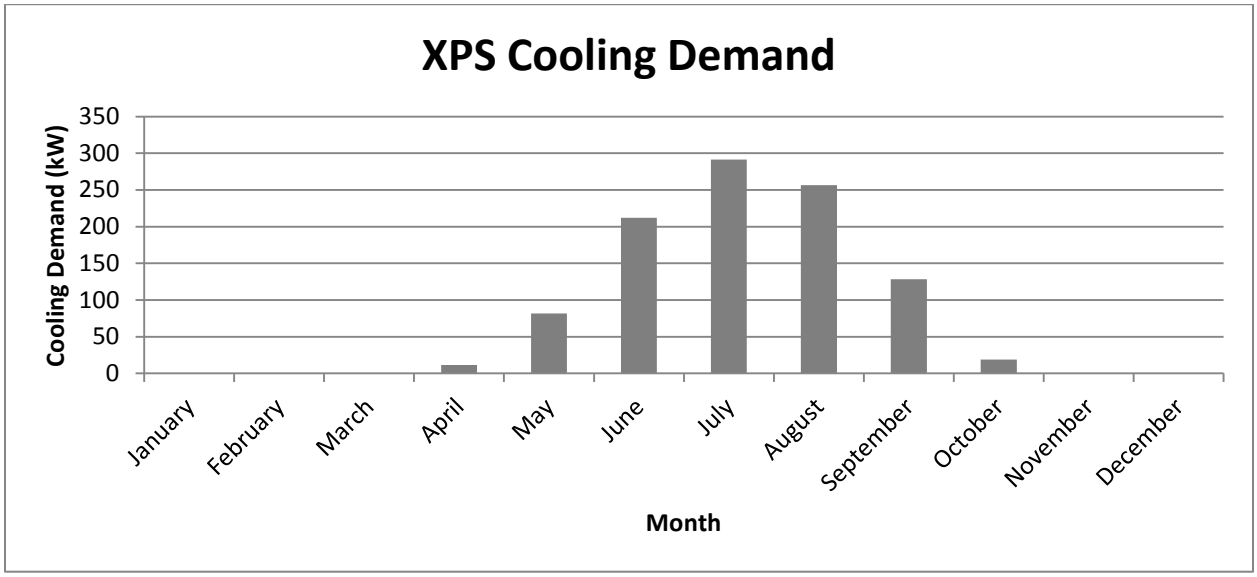

Figure 5-1: Monthly cooling demand of XPS

For the proposed heat pump system, the design concept and analysis methodology are identical to those presented in Chapter 4. In this sense, this chapter effectively extends the system design of Chapter 4 to include space cooling as well. The technical, financial, and environmental analysis will proceed as presented before, with additional terms added to reflect the electricity consumption of the current space conditioning system.

5.1 Results & Discussion

As before, the 39 different heat pumps and the different capacity heat pump systems they enable depending on the number of units installed in the bank are analyzed, but only those results for a given source and manufacturer combination are presented. The result of performing a cost minimization with respect to the number of heat pumps in the bank is presented Figure 5-2. As before, the total annual cost for each system is divided into four categories: electrical power consumed by the heat pump bank,

electrical power consumed by the circulation pumps, annuitized capital costs and the costs associated with supplying supplementary natural gas heating capacity.

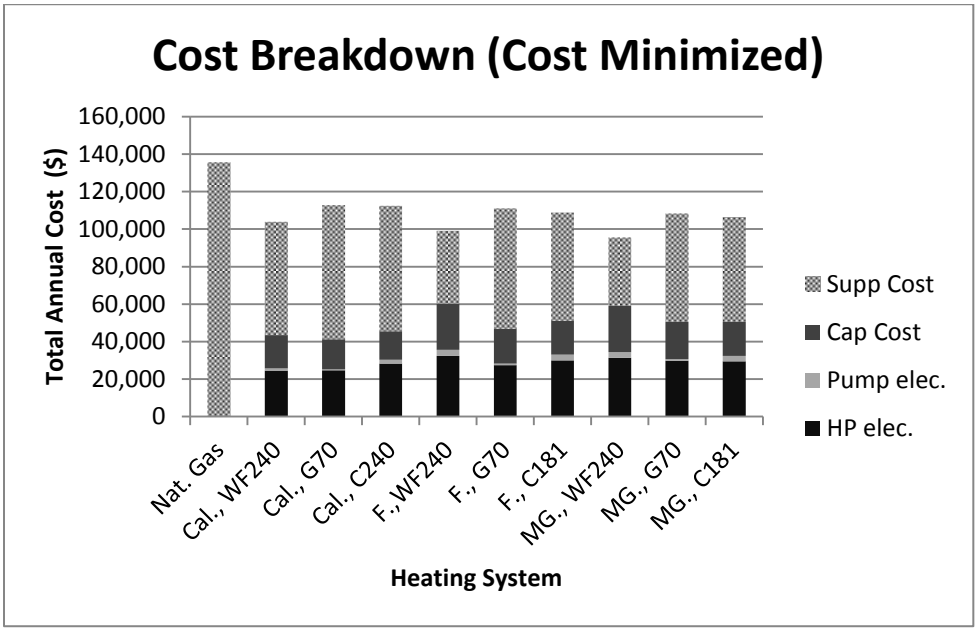

Figure 5-2: Total Annual cost for cost minimized systems

In Chapter 3, the critical temperature for space conditioning was calculated as 19.5°C, which is exceeded by the average temperature of matte granulation (22.7°C). As before, the significant usage of supplement heating conflicts with the assumption of fully meeting space conditioning demand with heat pumps alone. It is interesting to note that the higher COP for the heat pump systems was actually obtained during cooling mode (Table 5-1), further highlighting the seemingly unintuitive idea that heat pumps can be employed to recover a waste heat stream, but can still deliver highly efficient cooling should the waste heat stream be at a sufficiently low grade.

Table 5-1: Number of heat pumps, average COP for heating and cooling for the systems of Figure 5-2

Source and Model	# of heat pumps in bank	Average COP_h	Average COP_c
Cal., WF240	4	6.0	8.4
Cal., G70	8	4.3	6.7
Cal., C240	3	4.4	5.0
F.,WF240	6	6.5	7.6
F., G70	9	4.6	6.0
F., C181	5	5.0	4.4
MG., WF240	6	6.9	7.0
MG., G70	10	4.9	5.4
MG., C181	5	5.3	4.1

Table 5-1 reflects the average temperature of the three sources and how they impact COP through Equation 2. The heating COP increases from calciner to furnace to matte granulation as before, but there is the reverse trend in the cooling COP. This is to be expected from Equation 2, as rising source temperature increases the efficiency with which thermal energy is extracted, but decreases the efficiency with which it can be injected into the source. The dominance of heating relative to cooling means higher heating COPs are preferable to higher cooling COPs, so the higher temperature sources still produce the lower cost systems.

As discussed in Chapter 5, systems with fewer heat pump units in the bank will better conform to the assumption of equal maintenance cost between the proposed heat pump system and the currently installed system. As such, systems with fewer heat pumps will present more realistic results under the constraints of the stated assumption.

As before, a key result of cost minimization is that a heat pump system will rely on a significant amount of supplementary natural gas which is often greater than the total heating provided by the heat pumps. The annual CO_2 emissions associated to the above systems is presented in Figure 5-3.

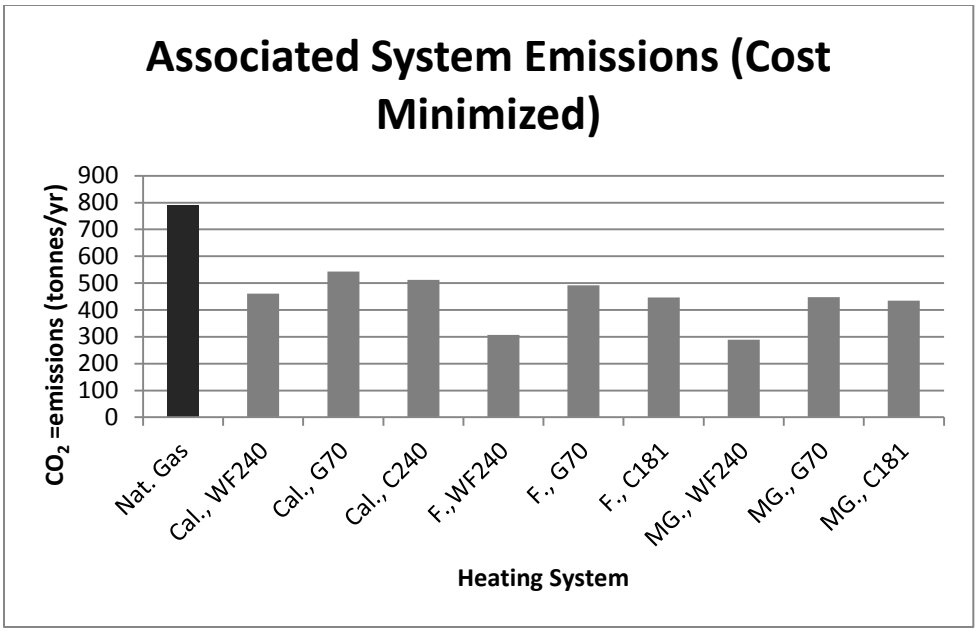

Figure 5-3: CO_2 emissions associated to the systems of Figure 5-2

There is comparatively little change to be found between Figure 5-3 and Figure 4-5, despite the fact that the CO_2 emissions associated to electrically powered space cooling have be included. This is due to the minimal demand for space cooling and the comparatively low emissions intensity of electricity compared to natural gas. In fact, almost the entirety of the emissions associated to the heat pump systems are from the supplementary natural gas. That said, all heat pump systems significantly lowered annual CO_2 emissions when compared to the current natural gas heating and air conditioning system. The most expensive heat pump system, a Carrier model 240 used with the calciner, resulted in a decrease in annual CO_2 emissions of 276 tonnes. The system that resulted in the lowest annual cost, a matte granulation sourced Waterfurnace Envision NLV series model 240 used with matte granulation, had the greatest reduction in emissions at 500 tonnes per annum.

The benefit of further cooling the process water streams is presented in Figure 5-4, which uses the minimum cost matte granulation sourced Waterfurnace 240 system as an example. As can be seen,

monthly temperature drops associated with heating are far more numerous than the slight increase during June and July that result from net cooling. Averaged over an entire year, the drop in process water temperature through the heat pump bank is approximately 3.7°C. This net temperature drop will improve the overall efficiency of the process waters in their intended use as coolants.

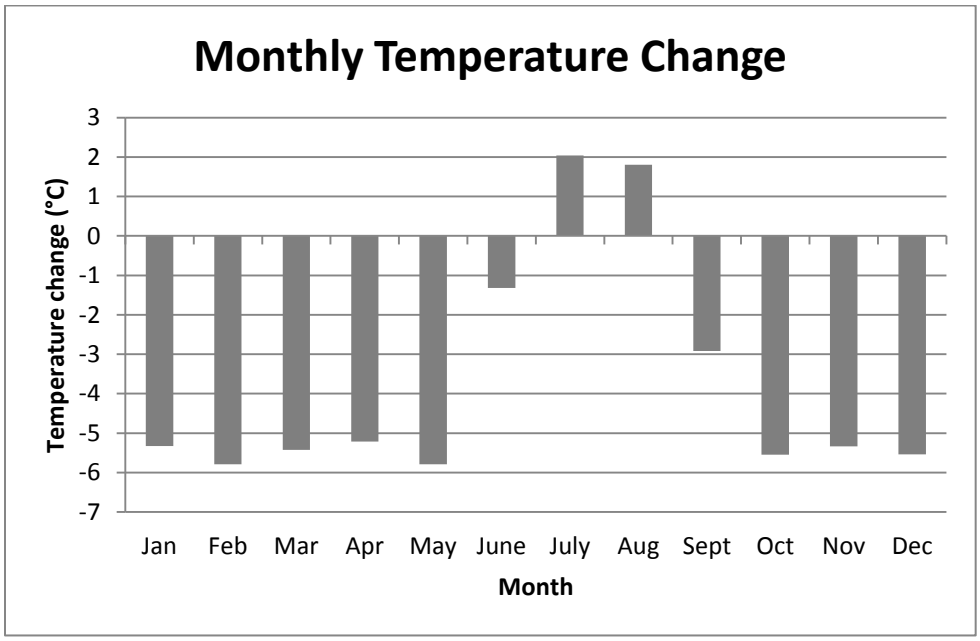

Figure 5-4: The change in temperature of the process water through the heat pump bank

To further demonstrate the power of the proposed heat pump systems, a second CO_2 minimization if presented considering space conditioning. The results of this analysis are presented in Figure 5-5.

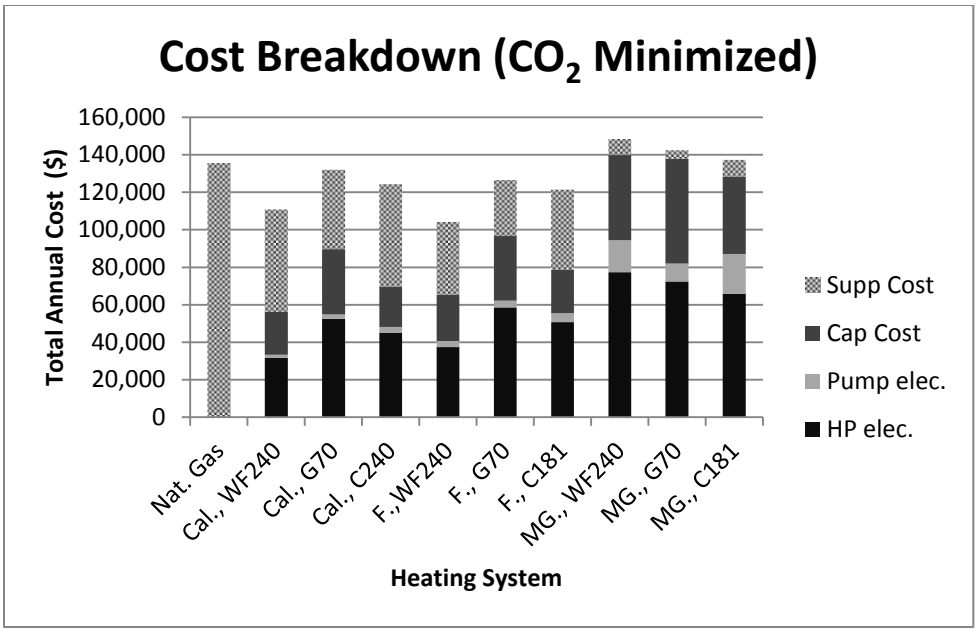

Figure 5-5: Total Annual cost for CO_2 minimized systems

A summary of system parameters is presented in Table 5-2, and the associated CO_2 emissions are given in Figure 5-6.

Table 5-2: Number of heat pumps, average COP for heating and cooling for the systems of Figure 5-5

Source and Model	# of heat pumps in bank	Average COP_h	Average COP_c
Cal., WF240	6	6.0	8.4
Cal., G60	27	4.2	6.3
Cal., C210	6	4.4	4.7
F.,WF240	6	6.5	7.6
F., G60	25	4.5	5.7
F., C210	6	4.7	4.3
MG., WF160	24	4.3	5.1
MG., G60	46	4.8	5.2
MG., C181	17	5.3	4.1

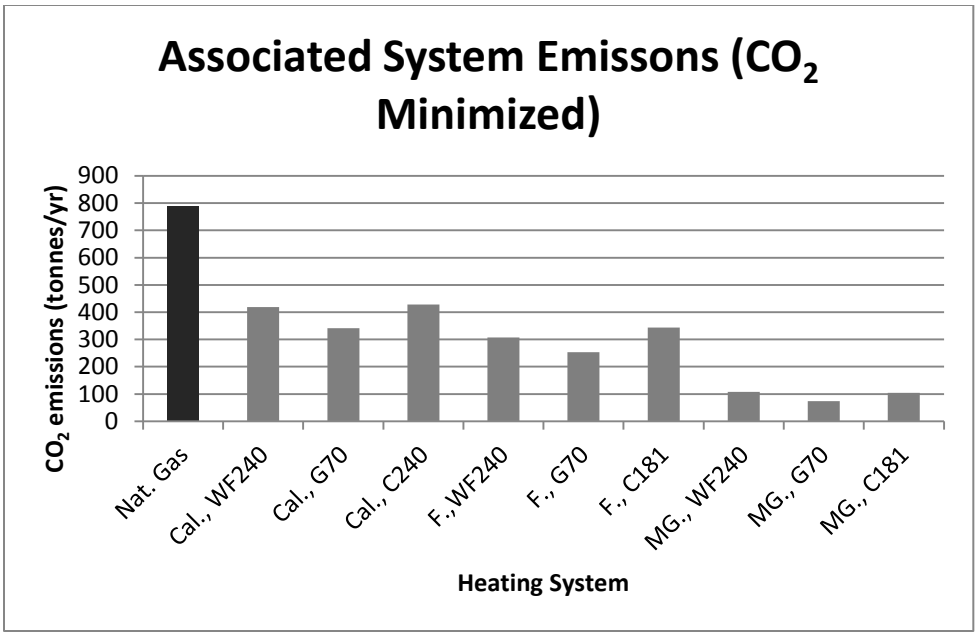

Figure 5-6: CO_2 emissions associated to the systems of Figure 5-5

6 Conclusions

In this thesis, the potential for heat pumps to recovery exceptionally low grade resources only marginally above ambient conditions was presented. In the introduction, a specific research question was posed in regards to the Sudbury INO portion of this work, namely: can heat pump systems be designed to recover waste heat in a cost effective manner while simultaneously reducing CO_2 emissions? The results presented in Chapters 4 and 5 indicate that yes, within the scope of the analysis performed, a heat pump system could be used to lower the annual space heating (or space conditioning) costs of an on-site building relative to the current system employed. Furthermore, the reduction in annual cost comes with an even greater drop in annual associated CO_2 emissions. As is common in the case of retrofits, new builds with heat recovery heat pump systems incorporated at the outset can further improve the cost effectiveness of such systems relative to more traditional space conditioning. Many industries, such as mineral processing, power generation and manufacturing should be particularity mindful of the potential for heat pump heat recovery systems to be incorporated at the beginning of the design process.

The rapid scoping models of Chapter 3 are not limited to a smelter and are intended to aid engineers in making rapid, low cost assessments of recovering on-site thermal resources with the use of off the shelf GSHP. A key limitation of these models, established by examining the results of Chapters 4 and 5, is that they fail to take into account that heat pumps need not meet the entirety of given space heating demand; they can be combined with different supplementary heating systems that can lower the annual costs of the combined system below the costs of a strictly conventional or strictly heat pump system.

6.1 Future Work

There are multiple avenues to extend the current work, such as:

- Updating the scoping models of Chapter 3 to reflect the value of combining GSHP systems with complementary heating in terms of minimizing annual costs.
- A more detailed analysis of the proposed piping route needs to be undertaken to ensure compliance with various governmental and internal regulations.
- Additional information regarding the water quality of all sources should be collected to ensure that the heat pumps will be able to attain their expected economic lives without undue maintenance or fouling.
- Process engineers should establish the probability that a given thermal resources will continue to be available over the full economic life of the heat pump system, as any changes to the process path which lead to a reduction in temperature or flow rate are detrimental to the efficacy of the heat pump system.
- The economic analysis has assumed process downtime can be treated as having zero monetary cost. While this is true for a new build installation, it may not necessarily be true for a retrofit unless the installation can be accomplished during scheduled downtime (The smelter has up to a full month of scheduled downtime every summer). A process engineer familiar with Sudbury INO's scheduling should ensure that the heat pump system can be installed without forcing process downtime.

7 Bibliography

Aegent Energy Advisors. (2015). Ontario Electricity Carbon Intensity: How Does It Measure Up? Retrieved from http://www.aegent.ca/newsletters/OntarioCarbonIntensity.html

Ammar, Y., Joyce, S., Norman, R., Wang, Y., & Roskilly, A. P. (2012). Low grade thermal energy sources and uses from the process industry in the UK. *Applied Energy*, *89*(1), 3–20. http://doi.org/10.1016/j.apenergy.2011.06.003

An, Q. (2012). Application of air-source heat pump water heater with exhaust air heat recovery in mining. *China Water and Wastewater*, *28*(23), 101–103.

Bank of Canada. (2015a). Historical Data. Retrieved from http://www.bankofcanada.ca/rates/indicators/capacity-and-inflation-pressures/inflation/historical-data/

Bank of Canada. (2015b, January). Daily Currency Converter. Retrieved from http://www.bankofcanada.ca/rates/exchange/daily-converter/

Banks, D., Fraga Pumar, A., & Watson, I. (2009). The operational performance of Scottish minewater-based ground source heat pump systems. *Quarterly Journal of Engineering Geology and Hydrogeology*, *42*(Part 3), 347–357. Retrieved from http://dx.doi.org/10.1144/1470-9236/08-081

Banks, D., Skarphagen, H., Wiltshire, R., & Jessop, C. (2003). Mine water as a resource: Space heating and cooling via use of heat pumps. *Land Contamination and Reclamation*, *11*(2), 191–198.

Barzi, Y. M., Assadi, M., & Arvesen, H. M. (2014). A novel heat recovery technology from an aluminum reduction cell side walls: Experimental and theoretical investigations. In *Light Metals 2014 - TMS 2014 Annual Meeting and Exhibition, February 16, 2014 - February 20* (pp. 733–738). Faculty of Science and Technology, University of Stavanger, 4036 Stavanger, NorwayGoodtech Recovery Technology AS, Per Krohgs Vei 4, 1065 Oslo, Norway: Minerals, Metals and Materials Society.

Bhattacharjee, K. (2010). Energy conservation opportunities in industrial waste heat recovery systems. *Energy Engineering: Journal of the Association of Energy Engineering*, *107*(6), 7–13.

Braham, D. (1984). Looking to the future use of heat pumps. *Building Services & Environmental Engineer*, *6*(6), 13–15.

Bruckner, S., Liu, S., Miro, L., Radspieler, M., Cabeza, L. F., & Lavemann, E. (2015). Industrial waste heat recovery technologies: An economic analysis of heat transformation technologies. *Applied Energy*, *151*, 157–167. http://doi.org/10.1016/j.apenergy.2015.01.147

Buy Insulation Products. (2016). Fiberglass pipe Insulation. Retrieved April 8, 2016, from http://www.buyinsulationproductstore.com/Fiberglass-Pipe-Insulation-SSL-ASJ/

Canada Revenue Agency. (2015a). Classes of Depreciable Property. Retrieved from http://www.cra-arc.gc.ca/tx/bsnss/tpcs/slprtnr/rprtng/cptl/dprcbl-eng.html#class46

Canada Revenue Agency. (2015b). Corporation Tax Rates. Retrieved from http://www.cra-arc.gc.ca/tx/bsnss/tpcs/crprtns/rts-eng.html

Cane, D., & Garnet, J. (2000). Update on maintenance and service costs of commercial building ground-source heat pump systems. In *ASHRAE Transactions* (Vol. 106, p. PA/ –). Dallas, TX, USA.

Cane, D., Morrison, A., & Ireland, C. (1998). Maintenace and service costs of commercial building ground-source heat pump systems. In *ASHRAE Transactions* (pp. 699–706).

Chao, S., Yiqiang, J., Yang, Y., Shiming, D., & Xinlei, W. (2013). A field study of a wastewater source heat pump for domestic hot water heating. *Building Services Engineering Research and Technology*, *34*(4), 433–448. http://doi.org/10.1177/0143624412463571

Chen, C., Sun, F.-L., Feng, L., & Liu, M. (2005). Underground water-source loop heat-pump air-conditioning system applied in a residential building in Beijing. *Applied Energy*, *82*(4), 331–344. http://doi.org/10.1016/j.apenergy.2004.12.002

Chua, K. J., Chou, S. K., & Yang, W. M. (2010). Advances in heat pump systems: A review. *Applied Energy*, *87*(12), 3611–3624. http://doi.org/10.1016/j.apenergy.2010.06.014

Dalla Rosa, A., Boulter, R., Church, K., & Svendsen, S. (2012). District heating (DH) network design and operation toward a system-wide methodology for optimizing renewable energy solutions (SMORES) in Canada: A case study. *Energy*, *45*(1), 960–974. http://doi.org/10.1016/j.energy.2012.06.062

Demir, H., Mobedi, M., & Ülkü, S. (2008). A review on adsorption heat pump: Problems and solutions. *Renewable and Sustainable Energy Reviews*, *12*(9), 2381–2403. http://doi.org/10.1016/j.rser.2007.06.005

Deng, H., & Gu, Y. (2012). The feasibility analysis of wastewater source heat pump using the urban wastewater heat. *Research Journal of Applied Sciences, Engineering and Technology*, *4*(18), 3501–3504.

Du, H., Dou, Y., & Qi, C. (2013). Application of mine water for water-source heat pump system. *Applied Mechanics and Materials*, *291-294*, 1701–1707. http://doi.org/10.4028/www.scientific.net/AMM.291-294.1701

Environment Canada. (2013). Fuel Combustion. Retrieved from http://www.ec.gc.ca/ges-ghg/default.asp?lang=En&n=AC2B7641-1

Fang, H., Xia, J., & Jiang, Y. (2015). Key issues and solutions in a district heating system using low-grade industrial waste heat. *Energy*, *86*, 589–602. http://doi.org/10.1016/j.energy.2015.04.052

Fanisalek, H., Bashiri, M., & Kamali, R. (2012). Aluminum smelter waste heat recovery plant (heat exchangers fouling and corrosion-a detailed investigation). In *Energy Technology 2012: Carbon Dioxide Management and Other Technologies - TMS 2012 Annual Meeting and Exhibition, March 11, 2012 - March 15* (pp. 203–214). Payam-e-Noor University, Bandar Abbas, IranRazi University, Kermanshah, Iran: Minerals, Metals and Materials Society.

Feng, X., & Berntsson, T. (1997). Critical COP for an economically feasible industrial heat-pump application. *Applied Thermal Engineering*, *17*(1), 93–101. http://doi.org/10.1016/1359-4311(96)00010-5

Fleer, M. (2010). *Heat recovery from the exhaust gas of aluminum reduction cells*. Reykjavik University.

Fleer, M., Lorentsen, O.-A., Harvey, W., Palsson, H., & Saevarsdottir, G. (2010). Heat recovery from the exhaust gas of aluminum reduction cells. In *Light Metals 2010 - TMS 2010 Annual Meeting and Exhibition, February 14, 2010 - February 18* (pp. 243–248). Reykjavik Energy Graduate School of Sustainable Systems, Reykjavik, IcelandHydro Aluminum, Primary Metal Technology, Porsgrunn, NorwayReykjavik University, Reykjavik, IcelandUniversity of Iceland, Reykjavik, Iceland: Minerals, Metals and Materials Society.

Follinger, T., & Merker, G. P. (1992). Periodic high-temperature heat pumps for industrial heat recovery. II. Example of application and design of apparatus. *Heat Recovery Systems & CHP*, *12*(3), 257–269. http://doi.org/10.1016/0890-4332(92)90054-L

Fox, D. B., Sutter, D., & Tester, J. W. (2011). The thermal spectrum of low-temperature energy use in the United States. *Energy and Environmental Science*, *4*(10), 3731–3740. http://doi.org/10.1039/c1ee01722e

Gorianec, D., Pozeb, V., Tomi, L., & Trop, P. (2014). Exploitation of the waste-heat from hydro power plants. *Energy*, *77*, 220–225. http://doi.org/10.1016/j.energy.2014.06.106

Gu, Y., & Wang, S. (2013). Research on the energy-saving and economic analysis of wastewater source heat pump. *Energy Education Science and Technology Part A: Energy Science and Research*, *31*(2), 1167–1170.

Hall, A., Scott, J. A., & Shang, H. (2011). Geothermal energy recovery from underground mines. *Renewable and Sustainable Energy Reviews*, *15*(2), 916–924. http://doi.org/10.1016/j.rser.2010.11.007

Hammond, G. P., & Norman, J. B. (2014). Heat recovery opportunities in UK industry. *Applied Energy*, *116*, 387–397. http://doi.org/10.1016/j.apenergy.2013.11.008

Hebenstreit, B., Schnetzinger, R., Ohnmacht, R., Höftberger, E., Lundgren, J., Haslinger, W., & Toffolo, A. (2014). Techno-economic study of a heat pump enhanced flue gas heat recovery for biomass boilers. *Biomass and Bioenergy*, *71*, 12–22. http://doi.org/10.1016/j.biombioe.2014.01.048

Holland, F., & Bragg, R. (2002). *Fluid Flow for Chemical and Process Engineers*. Butterworth-Heinemann.

Hung, T. C., Shai, T. Y., & Wang, S. K. (1997). A review of organic rankine cycles (ORCs) for the recovery of low-grade waste heat. *Energy*, *22*(7), 661–667. http://doi.org/10.1016/S0360-5442(96)00165-X

IEA. (2013). *Energy Efficiency Market Report 2013*. Retrieved from http://www.iea.org/publications/freepublications/publication/EEMR2013_free.pdf

IEA. (2014). Energy Snapshot of the Week. Retrieved April 5, 2016, from https://www.iea.org/newsroomandevents/graphics/2014-07-22-what-drives-energy-consumption-in-households.html

IEA. (2016). Energy Snapshot of the Week. Retrieved April 6, 2016, from https://www.iea.org/newsroomandevents/graphics/2015-10-20-savings-from-energy-efficiency-investments-made-since-1990.html

IESO. (2015). Power Data. Retrieved from http://www.ieso.ca/Pages/Power-Data/default.aspx#download

Incropera, F. P., De Witt, D. P., Bergman, T. L., & Lavine, A. S. (2006). *Fundamentals of heat and mass transfer*. *Applied Mechanics Reviews* (Vol. 44). John Wiley & Sons.

Jinggang, W., Meixia, D., Chuanchuan, D., & Xiaoxia, G. (2010). Exergy analysis of mine water source heat pump system. *Advanced Materials Research*, *121-122*, 980–985. http://doi.org/10.4028/www.scientific.net/AMR.121-122.980

Johnson, I., Choate, B., & Dillich, S. (2008). Waste heat recovery: Opportunities and challenges. In *TMS Annual Meeting* (pp. 47–52). New Orleans, LA, United states.

Kahraman, A., & Çelebi, A. (2009). Investigation of the Performance of a Heat Pump Using Waste Water as a Heat Source. *Energies*, *2*(3), 697–713. http://doi.org/10.3390/en20300697

Karu, V., Valgma, I., & Kolats, M. (2013). Mine water as a potential source of energy from underground mined areas in Estonian oil shale deposit. *Oil Shale*, *30*(2 SUPPL.), 336–362. Retrieved from http://dx.doi.org/10.3176/oil.2013.2S.12

Kelso, R. M., & Johnson, W. S. (2015). Flooded mine water-to-air heat pump system. *ASHRAE Journal*, *57*(8), 34–38.

Kew, J. (1985). HEAT PUMPS FOR BUILDING SERVICES. *Technical Note TN - BSRIA (Building Services Research and Information Association)*, (8).

Kwak, D.-H., Oh, S.-Y., & Kim, J.-K. (2012). Process integration study for the use of industrial low grade heat (Vol. 29, pp. 1591–1596). Department of Chemical Engineering, Hanyang University, 222 Wangsimni-ro, Seongdong-gu, Seoul, 133-791, Korea, Republic of: Italian Association of Chemical Engineering - AIDIC. http://doi.org/10.3303/CET1229266

Law, R., Harvey, A., & Reay, D. (2013). Opportunities for low-grade heat recovery in the UK food processing industry. *Applied Thermal Engineering*, *53*(2), 188–196. http://doi.org/10.1016/j.applthermaleng.2012.03.024

Lawson, D., & Sonderegger, J. (1978). *Report (geothermal data-base study)*. Butte (USA).

Lecompte, S., Huisseune, H., van den Broek, M., Vanslambrouck, B., & De Paepe, M. (2015). Review of organic Rankine cycle (ORC) architectures for waste heat recovery. *Renewable and Sustainable Energy Reviews*, *47*, 448–461. http://doi.org/10.1016/j.rser.2015.03.089

Li, X., Mu, G., Zhang, G., Sun, C., Zhang, Y., & Wang, Y. (2014). Application effect of seawater source heat pump for water heating and waste heat recovery in flatfish culture system. *Nongye Gongcheng Xuebao/Transactions of the Chinese Society of Agricultural Engineering*, *30*(3), 142–148.

http://doi.org/10.3969/j.issn.1002-6819.2014.03.019
Lindley, N. L., & Floyd, J. C. (1993). Piping systems: How installation costs stack up. *Chemical Engineering*. Retrieved from http://search.proquest.com/docview/194424789?accountid=11956
Liu, X., Lau, S.-K., & Li, H. (2014). Optimization and analysis of a multi-functional heat pump system with air source and gray water source in heating mode. *Energy and Buildings*, *69*, 1–13. http://doi.org/10.1016/j.enbuild.2013.09.039
Loken, M. (2013). *Recovery of Waste Heat from Pyrometallurgical Facilites for use in Microalgae production*. Laurentian University.
MathWorks. (2016). MATLAB. Retrieved April 2, 2016, from http://www.mathworks.com/products/matlab/
Meggers, F., & Leibundgut, H. (2011). The potential of wastewater heat and exergy: Decentralized high-temperature recovery with a heat pump. *Energy and Buildings*, *43*(4), 879–886. http://doi.org/10.1016/j.enbuild.2010.12.008
Mei, V. C., & Nephew, E. A. (1987). *Life-cycle cost analysis of residential heat pumps and alternative HVAC systems*. Oak Ridge National Laboratory.
Moran, M. J., Shapiro, H. N., Boettner, D., & Bailey, M. B. (2011). *Fundamentals of Engineering Thermodynamics* (Vol. 6). John Wiley & Sons.
Munson, B. R., Young, D. F., Okiishi, T. H., & Huebsch, W. W. (2009). *Fundamentals of Fluid Mechanics* (Vol. 6th). USA: John Wiley & Sons, Inc.
Mustafa Omer, A. (2008). Ground-source heat pumps systems and applications. *Renewable and Sustainable Energy Reviews*, *12*(2), 344–371. http://doi.org/10.1016/j.rser.2006.10.003
NASA. (2016). NASA surface meteorology and solar energy. Retrieved April 8, 2016, from https://eosweb.larc.nasa.gov/cgi-bin/sse/retscreen.cgi?email=&step=1&lat=46.52&lon=-80.95&submit=Submit
Ni, L., Lau, S. K., Li, H., Zhang, T., Stansbury, J. S., Shi, J., & Neal, J. (2012). Feasibility study of a localized residential grey water energy-recovery system. *Applied Thermal Engineering*, *39*, 53–62. http://doi.org/10.1016/j.applthermaleng.2012.01.031
Nowicki, C., & Gosselin, L. (2012). An overview of opportunities for waste heat recovery and thermal integration in the primary aluminum industry. *JOM*, *64*(8), 990–996. http://doi.org/10.1007/s11837-012-0367-4
Oanda. (2015). Historical Exchange Rates. Retrieved from http://www.oanda.com/currency/historical-rates/
Qiu, W., Zhang, L., & Liu, Q. (2013). Research on the heating plan of recovering condensing heat of power plant by water source heat pump unit. *Advanced Materials Research*, *608-609*, 1241–1245. http://doi.org/10.4028/www.scientific.net/AMR.608-609.1241
Rafferty, K. (2001). *An Information Survival Kit for the Prospective Geothermal Heat Pump Owner*. Retrieved from http://www.builditsolar.com/References/Calculators/Fuels/heat pump survival.pdf
Raymond, J., & Therrien, R. (2007). Geothermal Energy Resources on Active Mines. In *CIM Montreal 2007*.
Raymond, J., & Therrien, R. (2014). Optimizing the design of a geothermal district heating and cooling system located at a flooded mine in Canada. *Hydrogeology Journal*, *22*(1), 217–231. http://doi.org/10.1007/s10040-013-1063-3
Raymond, J., Therrien, R., & Gosselin, L. (2010). Low-temperature geothermal energy in mining environments. *CIM Journal*, *1*(2).
Rezaie, B., & Rosen, M. A. (2012). District heating and cooling: Review of technology and potential enhancements. *Applied Energy*, *93*, 2–10. http://doi.org/10.1016/j.apenergy.2011.04.020
Richardson, J. J., & Lopez, D. L. (2014). Investigating heat flow through a flooded coal mine complex for heat pump applications. *Geological Society of America, 2014 Annual Meeting & Exposition*, *46*(6),

119.
Sarbu, I., & Sebarchievici, C. (2014). General review of ground-source heat pump systems for heating and cooling of buildings. *Energy and Buildings*, *70*, 441–454. http://doi.org/10.1016/j.enbuild.2013.11.068
Seck, G. S., Guerassimoff, G., & Maizi, N. (2013). Heat recovery with heat pumps in non-energy intensive industry: A detailed bottom-up model analysis in the French food drink industry. *Applied Energy*, *111*, 489–504. http://doi.org/10.1016/j.apenergy.2013.05.035
Self, S. J., Reddy, B. V., & Rosen, M. A. (2013). Geothermal heat pump systems: Status review and comparison with other heating options. *Applied Energy*, *101*, 341–348. http://doi.org/10.1016/j.apenergy.2012.01.048
Sever, J. C. (2013). Waste heat recovery opportunities in a magnesium silicothermic reduction plant. In *TMS Annual Meeting* (pp. 69–75). San Antonio, TX, United states.
Smith, L. H., & Arthur, D. C. (1996). Mine ventilation: Waste heat recovery. *CIM Bulletin*, *89*(998), 126–130.
Soni, S. K., Pandey, M., & Bartaria, V. N. (2015). Ground coupled heat exchangers: A review and applications. *Renewable and Sustainable Energy Reviews*, *47*, 83–92. http://doi.org/10.1016/j.rser.2015.03.014
Spoelstra, S., Haije, W. G., & Dijkstra, J. W. (2002). Techno-economic feasibility of high-temperature high-lift chemical heat pumps for upgrading industrial waste heat. *Applied Thermal Engineering*, *22*(14), 1619–1630. http://doi.org/10.1016/S1359-4311(02)00077-7
Staffell, I., Brett, D., Brandon, N., & Hawkes, A. (2012). A review of domestic heat pumps. *Energy and Environmental Science*, *5*(11), 9291–9306. http://doi.org/10.1039/c2ee22653g
Union Gas. (2015). Important Information About Your Rates. Retrieved from https://www.uniongas.com/~/media/aboutus/rates/residential/rate310.pdf?la=en
Urdaneta-B, A., & Schmidt, P. S. (1980). Thermodynamic and economic analysis of heat pumps for energy recovery in industrial processes. *Transactions of the ASME.Journal of Energy Resources Technology*, *102*(3), 173–180. http://doi.org/10.1115/1.3227869
Wallin, E., & Berntsson, T. (1994). Integration of Heat Pumps in Industrial Processes. *Heat Recovery Systems & CHP*, *14*(3), 287–296.
Wallin, E., Franck, P. A., & Berntsson, T. (1990). Heat pumps in industrial processes. An optimization methodology. *Heat Recovery Systems & CHP*, *10*(4), 437–446.
Wang, F., Wang, Z., Zheng, Y., Lin, Z., Hao, P., Huan, C., & Wang, T. (2015). Performance investigation of a novel frost-free air-source heat pump water heater combined with energy storage and dehumidification. *Applied Energy*, *139*, 212–219. http://doi.org/10.1016/j.apenergy.2014.11.018
Wang, Y., Zhao, F.-Y., Kuckelkorn, J., Spliethoff, H., & Rank, E. (2014). School building energy performance and classroom air environment implemented with the heat recovery heat pump and displacement ventilation system. *Applied Energy*, *114*, 58–68. http://doi.org/10.1016/j.apenergy.2013.09.020
Watzlaf, G. R., & Ackman, T. E. (2006). Underground Mine Water for Heating and Cooling using Geothermal Heat Pump Systems. *Mine Water and the Environment*, *25*(1), 1–14. http://doi.org/10.1007/s10230-006-0103-9
Wedde, G., & Sorhuus, A. (2012). Waste heat recovery from industrial smelting exhaust gas. In *International Smelting Technology Symposium, Incorporating the 6th Advances in Sulfide Smelting Symposium - TMS 2012 Annual Meeting and Exhibition, March 11, 2012 - March 15* (pp. 31–38). Alstom Norway, Drammensveien 165, 0277 Oslo, Norway: Minerals, Metals and Materials Society.
Wright, J. R., & Steward, F. R. (1985). Three Industrial Heat Pump Installations Operating in Canada. *Journal of Heat Recovery Systems*, *5*(2), 81–88.
Wu, W., Wang, B., Shi, W., & Li, X. (2014). Absorption heating technologies: A review and perspective.

Applied Energy, *130*, 51–71. http://doi.org/10.1016/j.apenergy.2014.05.027

Wu, W., You, T., Wang, B., Shi, W., & Li, X. (2014). Simulation of a combined heating, cooling and domestic hot water system based on ground source absorption heat pump. *Applied Energy*, *126*, 113–122. http://doi.org/10.1016/j.apenergy.2014.04.006

Xiao, S. H., Luo, Q. H., & Li, G. F. (2014). Utilizing thermoelectric heat pump to heat recovery of shower Waste water. In *2013 2nd International Conference on Sustainable Energy and Environmental Engineering, ICSEEE 2013, December 28, 2013 - December 29* (Vol. 521, pp. 757–761). School of Urban Construction, University of South China, Hengyang, Hunan, 421000, China: Trans Tech Publications. http://doi.org/10.4028/www.scientific.net/AMM.521.757

XPS. (2016). XstrataXWPBranding. Retrieved April 8, 2016, from http://www.xps.ca/XstrataXWPBranding/Images/XPS6.jpg

Yang, H., Cui, P., & Fang, Z. (2010). Vertical-borehole ground-coupled heat pumps: A review of models and systems. *Applied Energy*, *87*(1), 16–27. http://doi.org/10.1016/j.apenergy.2009.04.038

Zaheer-Uddin, M. (1992). Multiple setpoint control of a heat recovery system. *Heat Recovery Systems & CHP*, *12*(1), 23–37. http://doi.org/10.1016/0890-4332(92)90004-2

Zebarjadi, M., Esfarjani, K., Dresselhaus, M. S., Ren, Z. F., & Chen, G. (2012). Perspectives on thermoelectrics: From fundamentals to device applications. *Energy and Environmental Science*, *5*(1), 5147–5162. http://doi.org/10.1039/c1ee02497c

Zhu, N., Hu, P., Xu, L., Jiang, Z., & Lei, F. (2014). Recent research and applications of ground source heat pump integrated with thermal energy storage systems: A review. *Applied Thermal Engineering*, *71*(1), 142–151. http://doi.org/10.1016/j.applthermaleng.2014.06.040

RECOVERY OF LOW GRADE WASTE HEAT FROM SMELTER PROCESS WATER STREAMS UTILIZING HEAT PUMPS

I. M. Ross
Bharit School of Engineering, Laurentian University
Sudbury, Canada

G. M. Ross
Northern Ontario School of Medicine, Laurentian University
Sudbury, Canada

*J. A. Scott
Bharti School of Engineering,
& Northern Ontario School of Medicine
Laurentian University, Sudbury, Canada

*(*Corresponding author: jascott@laurentian.ca)*

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ABSTRACT

The otherwise waste heat in many water streams in the mining industry, can be tapped into to provide space heating of adjacent infrastructure including laboratories, workshops and offices. This can be achieved by utilizing heat pump systems which can work as a replacement or in conjunction with existing space heating. In the example investigated, process cooling streams at an ore smelter, the analysis provided illustrates that annual heating costs can be 19% lower and a simultaneous 62% reduction in CO_2 emissions achieved.

KEYWORDS

Heat pump, Heat recovery, Mineral processing, Pyrometallurgy, Smelting, Space heating, Waste heat

INTRODUCTION

Within the mining and mineral processing industries, improving heat recovery is an important avenue for reducing costs and minimizing emissions. Thermal energy is often allowed to dissipate into the environment as waste heat, despite having economic value (Bhattacharjee, 2010; Ammar, Joyce, Norman, Wang, & Roskilly, 2012; Seck, Guerassimoff, & Maizi, 2013). This value is inversely proportional to the temperature of the waste heat; as the temperature decreases its applicability to direct use applications (including drying or hot water production) also decreases. In general, a thermal resource that is too cool for direct use applications is considered low-grade and will need to be upgraded before it can be effectively employed (Law et al., 2013). One technique for upgrading a low-grade resource for space heating is to use a heat pump (Urdaneta-B & Schmidt, 1980; Chen, Sun, Feng, & Liu, 2005) . Heat pumps are a proven technology that emerged from the work of Kelvin and Carnot in the mid-19th Century and became widely available in the 20th Century (Staffell et al., 2012). A heat pump is a device that moves thermal energy from a low temperature source to a higher temperature sink at the expense of work. Using a heat pump to upgrade a low grade heat source, rather than producing heating from a traditional system such as natural gas or electrical resistance, creates an opportunity for energy efficient and cost effective space heating and cooling (Wright & Steward, 1985; Bruckner et al., 2015). The applicability of heat pumps to the mining environment, particularly the possibility of coupling heat pump systems to ubiquitous geothermal resources like mine dewatering flows, is examined by Raymond & Therrien (2007) and Raymond, Therrien and Gosselin (2010).

This paper examines opportunities for low grade heat recovery from various nickel-copper smelter process water streams utilizing heat pumps for on-site space heating. Research into heat recovery opportunities in the smelting industry have largely focused on aluminum processing. Fleer et al. and Fanisalek et al. examine the potential for heat recovery from aluminum smelter exhaust gases to space heating and desalination respectively, see (Fleer, Lorentsen, Harvey, Palsson, & Saevarsdottir, 2010; Fanisalek, Bashiri, & Kamali, 2012). The Nordural Aluminium smelter in Iceland examined by Fleer et al. is estimated to present 55 MW_{th} of recoverable heat for use in supplying district heating to a community of 6,000 inhabitants. In both papers, fouling of heat exchangers are a main design challenge to be overcome; a test probe installed into the exhaust gas off the pot line at the Nordural smelter found a drop of 7% in overall heat transfer coefficient over 11 days.

Nowicki and Gosselin (2012) identified a variety of thermal sources and sinks at the Alcoa Deschambault aluminium smelter in Quebec, Canada. Cooling water carrying a total of 7.4 MW_{th} at a temperature of 40°C was well suited to space heating of on-site facilities with a peak demand of 8.6 MW. Assuming a coefficient of performance (COP) of 4 for heat pumps, savings were estimated at $0.3 million a year with a reduction of 6,000 tons of associated CO_2. Fang, Xia, and Jiang (2015) examined a copper smelter in Chifeng, Northern China, which

supplies multi-grade waste heat in the range of 25–90°C to a local district heating system. Up to 85 MW_{th} was deemed recoverable from the sulphuric acid plant, furnace and slag flushing water. A payback period of 4 years was estimated for the installation after operations began in winter of 2013/2014.

This work looks at new possible opportunities by examining recovery of exceptionally low-grade thermal resources (between 10 and 25°C) in mineral processing. Heat recovery in this range is often ignored due to the difficulties of attaining cost effective recovery and the perceived cost of retrofitting equipment. We show there are significant opportunities for heat pumps to recover very low-grade heat from otherwise wasted thermal resources.

The smelter investigated is located in Ontario, Canada. Beginning operations in 1929, the smelter is a polymetallic operation that processes nickel and copper (as well as cobalt, gold, silver, platinum and palladium) custom feeds and ores. Annual production is in the range of 75,000 tonnes of nickel in matte and 23,000 tonnes of copper in matte. Three sources of low-grade waste heat in the form of process cooling waters were examined for their potential application to a heat pump system: matte granulation; the furnace; and the calciner. The matte granulation process solidifies molten matte from finishing converter vessels using a spray of water with a flow rate of up to 1 m^3/s. The furnace process water is used to cool the exterior of the electric arc furnace by removing heat from copper cooling fins inserted into the refractory brick of the furnace walls. The furnace is among the largest electric arc furnaces in the world and contributes to the smelter site being the third largest consumer of electricity in Ontario. The calciner is a custom feedstock recycling facility designed to process scrap material, like oil refinery catalysts, with organic contaminants too high for the standard process path.

Recovered heat will be utilized for space heating a two story, 10,500 m^2 laboratory and office block adjacent to the smelter. This building has yearly heating load of 4,500 MWh_{th} with a peak demand of 1 MW_{th} in December. Whilst heat pumps can be utilized for both heating and cooling, the focus is on space heating opportunities due to the relatively cool local climate (heating degree days dominate cooling degree days 11:2).

BACKGROUND

Heat pumps move thermal energy against a thermal gradient by using work supplied to the unit (Moran et al., 2011). While there are different methods of heat pump operation, such as absorption and thermoelectric methods (Zebarjadi, Esfarjani, Dresselhaus, Ren, & Chen, 2012; Wu, Wang, Shi, & Li, 2014; Wu, You, Wang, Shi, & Li, 2014), this work examines the vapour compression method only as it is the most readily available commercial heat pump solution. A schematic of a vapour compression heat pump recovering waste heat from a process water flow can be seen in Figure 1.

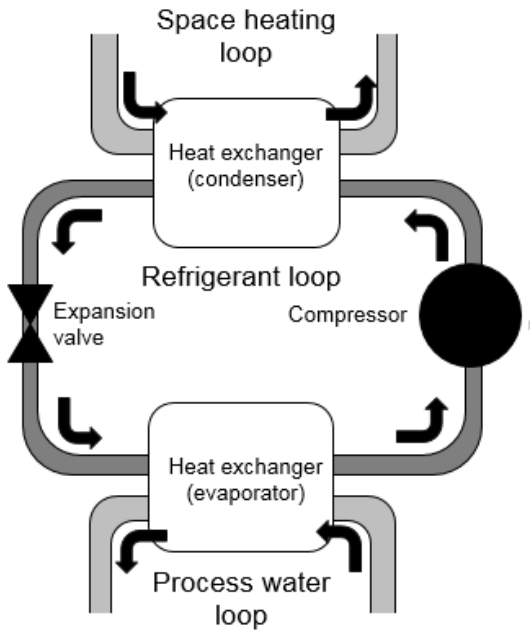

Figure 1. Schematic of heat recovery setup

The primary metric for evaluating the performance of a heat pump is COP, which indicates how many units of thermal energy can be delivered for a given unit of input energy. The maximum is the Carnot COP, which depends on the temperatures of the source, T_C, and sink, T_H, between which the heat pump operates, as seen in equation 1 (please see nomenclature for units for all equation terms):

$$COP_{Carnot} = \frac{T_H}{T_H - T_C} \tag{1}$$

This equation reveals an important property of heat pumps; their COP decreases as the difference between the temperatures of the source and sink grows. An actual heat pump will have a COP that is significantly lower than the Carnot COP due to entropy production in the compressor and expansion valve as well pressure losses in the condenser and evaporator (Spoelstra et al., 2002). Nevertheless, convergence of source and sink temperature still results in an improved COP.

In space heating applications, the temperature of the sink is set by the preferences of the building's users (typically approximately 20ºC) and the capacity of the heat distribution system (Y. Wang, Zhao, Kuckelkorn, Spliethoff, & Rank, 2014). The heat pump will therefore have a nearly constant sink temperature. To achieve cost effective heat recovery, a low-grade heat source must present a high enough temperature to ensure a COP that lowers the total costs of a heat pump system below the total costs of a traditional fossil fuel system (Feng & Berntsson, 1997). This illustrates the value inherent in low-grade thermal resources in the 10–20ºC range to provide an improved COP.

Heat pumps employed for space heating are divided into three categories based on the material from which heat is sourced: air source heat pumps (ASHPs), water source heat pumps (WSHPs) and terrain source heat pumps (TSHPs). Both WSHPs and TSHPs belong to the more general category of ground source heat pumps (GSHPs) which remove thermal energy from either water or earth, although water is employed as the heat transfer fluid in both instances. Since the waste heat to be recovered in this work is contained in process cooling water, only GSHPs will be considered (Qiu et al., 2013).

Heat sources

The waste heat proposed for recovery at the smelter is contained in process cooling waters leaving the calciner, matte granulation process and furnace. The cooling water produced by the calciner is sent to a settling tank to await discharge back to the environment, whilst the process water used by matte granulation and the furnace circulates in dedicated cooling towers dissipating 2.0 ± 0.2 and 4.8 ± 1.6 MW_{th}, respectively.

Data were collected for the furnace and matte granulation processes from June 2013 to May 2014 and for the calciner from January to December 2012. The highest COP will be obtained from the heat source with the highest temperature, but the flow-rate must also be considered in terms of delivering enough thermal energy for the required heat pump usage. Hence, the most effective source is likely to be the matte granulation process, followed by the furnace and calciner (Table).

Table 1. Temperature and flow rate of sources averaged over the year of data collection

	Matte granulation	Furnace	Calciner
Temperature (°C)	22.7 ± 3.2	18.2 ± 2.4	12.3 ± 0.7
Flow rate (L/s)	487.2 ± 96.5	23.6 ± 0.9	18.7 ± 3.3

A potential heat sink

Any local facility currently employing an alternative space heating method offers an opportunity for employing recovered heat. The specific facility examined within this work is an office and laboratory complex that employs a natural gas furnace with forced air (ducted) distribution for space heating. The monthly heating demand for this facility from June 2013 to May 2014 is presented in Figure 2

Figure.

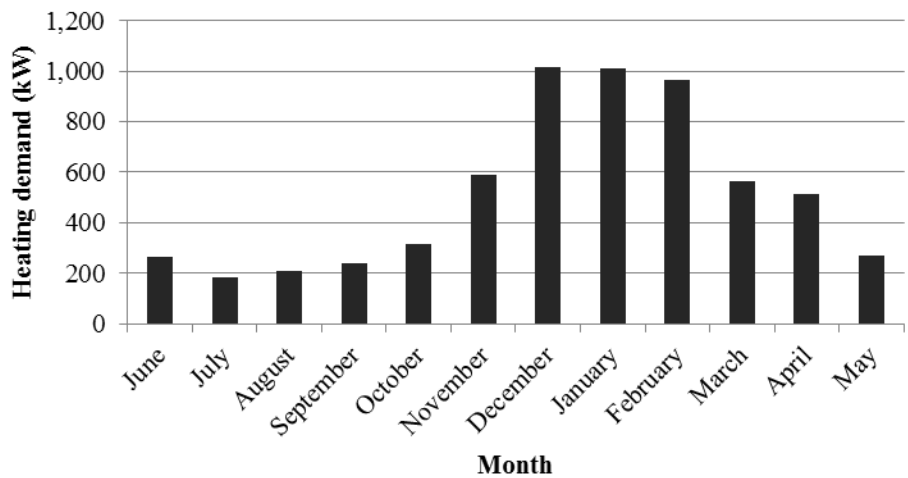

Figure 2. Monthly heating demand of the office and laboratory building

ANALYSIS METHODOLOGY

The process of supplying space heating to the building using heat recovered from the three potential sources of cooling water with a variety of commercially available heat pumps was modelled. The analysis was separated into three overarching categories: technical, financial and environmental. As the currently installed heat distribution system at the facility is forced air, the range of heat pumps is restricted to water-to-air GSHPs, as a water-to-water system would require the prohibitively costly and disruptive installation of a new heat distribution system.

To supply the required heating demand, it is necessary to utilize more than one heat pump and form a heat pump bank. A heat pump bank refers to a set of heat pumps designed to meet a given heating demand, whilst a heat pump system includes a heat pump bank and all additional components necessary for operation (e.g., piping, circulation pumps, a supplemental heating system). The heat pumps within the heat pump bank are arranged in

parallel so that the incoming flow of process water is divided into multiple streams. This is necessary when the individual heat pumps have permissible maximum flow-rates lower than the flow-rate of supplied process water.

The piping connection that supplies process water from the source to the heat pump banks analyzed as an open loop system; the process water will transported from its point of origin to contact the internal heat exchangers of the heat pumps. This is in contrast to a closed loop system, which uses a fluid circulating within an additional heat exchanger stage that removes the thermal energy from the process water before delivering that thermal energy to the heat pumps. An open loop system was chosen over a closed loop system because they are less expensive and invasive to install, thereby minimizing disruption and downtime.

A variety of commercially available high-capacity heat pumps were examined for their applicability to the design. These include the Waterfurnace Envision NLV 160,180, 240 and 300 models, the Geosmart Premium G 60 and 70 models, the Carrier 50 HQP 180 and 242 and also the Carrier 50 VQP 181, 210, 240, 300, and 360 models. These heat pumps are representative of commercially available models that have published specifications, including data on COP, pressure loss, and power requirements as a function of flowrate and temperature. Piping connections are assumed to be fabricated from American standard schedule 40 four-inch stainless steel pipe. This diameter was chosen because it placed fluid velocities in the range of 1–3 m/s recommended for long piping runs (Holland & Bragg, 2002). Since any decrease in temperature that occurs within the piping leads to a decrease in COP, the pipes were wrapped in three inch thick fibreglass insulation.

Where it was difficult to get pricing data for the high-capacity heat pumps examined within this work, a capital cost approximation curve provided by Staffell et al. (2012) as a function of capacity was used (equation 2), which has been made applicable to 2015 Canadian dollars by employing the average exchange rate for 2012 of 1.58 (Oanda, 2015), and a total core inflation of 3.8% (Bank of Canada, 2015b). *CC* is used to denote the capital cost of a given heat pump model, whilst *Cap* is the heat pump's rated capacity defined in the manufacturer's specification:

$$CC = 1.64 \cdot Cap \cdot (200 + \frac{4750}{Cap^{1.25}}) \tag{2}$$

Technical analysis

The temperature change along the piping connection between the process water source and the sink (facility to be heated), is modelled using the finite difference method (FDM), in which the full run of piping is split into many equal lengths of pipe, each short enough for a constant temperature approximation to be valid. Heat transfer across the pipe boundary is assumed to occur at a uniform temperature distribution within the piping (validated by a Reynolds number within the turbulent regime). The total length of piping necessary between the calciner and the sink is 200 m whilst the piping between both the furnace and matte granulation process and the sink is 250 m.

A total thermal resistance between the cooling water and the ambient air can be expressed as the sum of the resistance to internal convection, resistance to conduction through the steel pipe (k = 50 W/mK) and the fibreglass insulation (k = 0.038 W/mK) and resistance to external convection (Incropera et al., 2006).

The internal convection coefficient of the pipe can be found using the Gnielinski correlation (equation 3) for $Nu_{interior}$, with Reynolds number, $Re_{interior}$, Darcy friction factor f and Prandtl number of water, Pr_{water} (Incropera et al., 2006). Alternatively, the simpler Dittus-Boelter correlation can be used to reduce computer run time with minimal loss in accuracy; it was determined that system cost is only sensitive at the 0.0001% level to switching between Gnielinski or Dittus-Boelter correlations. Equation 4 allows the equivalent heat transfer coefficient $h_{interior}$ to be calculated using the thermal conductivity of water, k_{water}, and the internal diameter of the pipe, D:

$$Nu_{interior} = \frac{(f/8)(Re_{interior} - 1000)Pr_{water}}{1 + 12.7(f/8)^{1/2}\left(Pr_{water}^{2/3} - 1\right)} \tag{3}$$

$$h_{interior} = \frac{(Nu_{interior})(k_{water})}{D} \tag{4}$$

The interior convection coefficient is then expressed as an equivalent thermal resistance, $R_{interior}$, for a single section of the FDM analysis, where l denotes the length of the section (equation 5):

$$R_{interior} = \frac{1}{h_{interior}\pi D(l)} \tag{5}$$

An exterior Nusselt number $Nu_{exterior}$, is calculated using the correlation of equations 6 and 7, with the exterior Reynolds number $Re_{exterior}$ and the Prandtl number of air Pr_{ai} (Munson et al., 2009). Equation 8 is then employed as before to obtain the convection coefficient at the exterior of the pipe, $h_{exterior}$, where k_{air} denotes the thermal conductivity of air and t_{pipe}, $t_{insulation}$ the thickness of the pipe wall and insulation, respectively:

$$Nu_{exterior} = C({Re_{exterior}}^{m})({Pr_{air}}^{0.33}) \tag{6}$$

$$(C,m) = \begin{cases} (0.989,0.330)\ \ if\ Re < 4 \\ (0.911,0.385)\ \ if\ Re < 40 \\ (0.683,0.466)\ \ if\ Re < 4000 \\ (0.193,0.618)\ if\ Re < 40000 \\ (0.027,0.805)\ if\ Re > 40000) \end{cases} \tag{7}$$

$$h_{exterior} = \frac{(Nu_{exterior})(k_{air})}{D + 2(t_{pipe}) + 2(t_{insulation})} \tag{8}$$

Once each of the four thermal resistances across the pipe has been determined, equation 9 is used to determine the heat transfer across a single length of pipe, q, given the temperature of ambient air surrounding the pipe T_{out} and the temperature of process water within, T_{in}:

$$q = \frac{T_{out} - T_{in}}{(R_{interior} + R_{pipe} + R_{insulation} + R_{exterior})} \tag{9}$$

Employing equation 10 allows the temperature of the process water as it moves from one section of pipe, $T_{in,n,}$ to the next, $T_{in,n+1}$, to be calculated. Q denotes the volumetric flow rate of the process water, $c_{p,water}$ the specific heat capacity of water and ρ_{water} the density of water:

$$T_{in,n+1} = \frac{q}{Q c_{p,water}\rho_{water}} + T_{in,n} \tag{10}$$

Repeating this process across the total length of piping allows the temperature of the process water at the inlet of the heat pump bank to be determined. Whilst different systems produce different temperature drops, averaged over a year most systems tend to produce a temperature drop on the order of 0.05 to 0.1°C/km. The specific system of Figure 7 experienced a temperature drop of 0.081°C/km.

Pressure loss, ΔP, is determined using equation 11, where V denotes velocity, L the total length of straight pipe and L_e the equivalent length associated to minor losses (Munson et al., 2009).

$$\Delta P = f\frac{L}{D}\frac{\rho_{water}V^2}{2} + \sum f\frac{L_e}{D}\frac{\rho_{water}V^2}{2} \tag{11}$$

Minor losses associated with individual elbows ($L_e/D = 30$) and tees ($L_e/D = 20$) accounts for between 15 and 22% of total pressure loss. The friction factor is iteratively calculated to within a tolerance of 1% using equation 12, which employs a surface roughness, ε, of 0.025 mm (Munson et al., 2009).

$$\frac{1}{\sqrt{f}} = -2\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{Re_{interior}\sqrt{f}}\right) \tag{12}$$

The pumping work, W_{pump}, as a function of pressure loss, ΔP, volumetric flow rate and pump efficiency, η_{pump}, is approximated by equation 13:

$$W_{pump} = \frac{Q\Delta P}{\eta_{pump}} \tag{13}$$

The operation of individual heat pumps is analyzed using the temperature of the process cooling water in conjunction with manufacturer's data to calculate temperature dependent COP and power demand. The product of these two numbers is the capacity of the heat pump at the given temperature. The total number of heat pumps within the bank is left as a decision variable. Any heating demand not met by heat pump capacity is provided by supplemental natural gas heating. Since heat pump capacity can be chosen to be a small share of total heating demand, supplemental heating can in fact deliver the greater share of total heating.

The total monthly heating delivered by the bank can be calculated as the product of the heating capacity and the total number of heat pumps. The pressure drop though the heat pump bank is equal to the pressure drop through a single heat pump at the given flow rate and temperature, and subsequent pump power requirements are calculated using equation 13. Applying equation 14 allows the temperature drop through the heat pump bank to be calculated, $\Delta T_{heat\ pump}$ where the heat removed from the process water is equal to the heating capacity minus the power requirement of a single heat pump:

$$\Delta T_{heat\ pump} = \frac{(HC - P)}{QC_{p,water}\rho_{water}} \tag{14}$$

The process water, upon leaving the heat pump bank, is returned to its source. Pressure loss and subsequent pumping work associated with this return piping is calculated as above. Accounting for this temperature drop allows the total temperature drop due to the process water moving from its point of origin and back again to be calculated. This temperature drop is a desirable side-effect of the heat pump system, as the cooled water can now act more effectively in its intended role as a process coolant.

Financial analysis

Financial results are determined by applying cost data to relevant technical results. The financial analysis was performed as a replacement analysis, where the two options under consideration were the current natural gas system and the proposed heat pump system. The cost for both systems can be expressed as an annuity by amortizing capital costs over expected system lifetimes. This is done by employing equation 15, where *AV* is the annual value or annuity of an asset, *PV* is the present value of the asset, *i* is the hurdle rate (15% for the smelter site) and *n* is the life of the asset.

$$AV = PV\frac{i(i+1)^n}{(i+1)^n - 1} \tag{15}$$

Capital costs of the existing natural gas system are sunk, so the cost of the gas fired system is purely a fuel cost. The fuel cost can be determined using the smelter's natural gas provider's industrial pricing schedule (Union Gas, 2015), which averaged over a year of consumption was 2.24 cents per kWh. All financial figures presented in the report, excluding equation 2, are given in USD. An exchange rate between Canadian and US dollars of 0.8189 USD per CAD was used (Bank of Canada, 2015a).

The annual cost of the heat pump system can be broken down into two categories; capital costs such as purchasing the heat pumps, piping and circulation pumps, and operating costs associated with pumping the cooling water and powering the heat pumps. Expected maintenance costs for both systems are difficult to predict with accuracy, especially when, as in this case study, the existing heating system is nearing the end of its expected system life (the current natural gas system was installed in 1994). Whilst removing maintenance costs from the analysis is a source of uncertainty in the results, the purpose of this paper is to determine if space heating can be economically applied by a heat recovery heat pump system. This purpose is unaffected because natural gas systems tend to have

greater maintenance costs overall, so the analysis is made more conservative by shifting the economics in favor of natural gas (Cane, Morrison, & Ireland, 1998; Cane & Garnet, 2000).

The capital costs of individual heat pumps are determined using the correlation of equation 2. The total capital cost of the heat pump system is not born entirely by the smelter, because depreciation is claimed as a tax deduction in the year it occurs. These tax savings can be converted to a present worth using equation 16 and subtracted from the capital costs to determine a more accurate picture of costs born. The depreciation rate, *d*, of 30% was found using Canada's capital cost allowance schedule (Canada Revenue Agency, 2015a). The tax rate, *tr*, for large corporations in Ontario with greater than 15 million dollars in revenue is 26.5% (Canada Revenue Agency, 2015b).

$$PV_{depreciation} = \sum_{n=1}^{life\ of\ asset} (tr)(Cap)[(1-d)^{n-1}d]\,(1+i)^{-n} \tag{16}$$

The operating cost of the heat pump system comes from the electricity supplied to both the heat pumps and the circulation pumps. The cost of electricity was estimated using data provided by Ontario's Independent Electricity System Operator (2015) for large industrial electricity consumers. This produced an average electricity cost of 6.85 cents per kWh for the year of analysis.

Environmental analysis

Switching from natural gas to a heat pump system is expected to result in significantly lower associated CO_2 emissions, *E*. For this analysis, equation 17 can be employed, which expresses the system CO_2 emissions as a function of natural gas consumption in cubic meters, C_{natgas}, and electricity consumption in kWh, C_{elec}. A carbon emission factor, $CEF_{nat\ gas}$, of 0.001879 tonnes of CO_2/m^3 for natural gas was used (Environment Canada, 2013). Within this analysis, electricity produced in Ontario is treated as releasing emissions of 0.04 kilograms of CO_2 for every kWh of electricity (Aegent Energy Advisors, 2015).

$$E_{heat\ pump} = CEF_{elec}C_{elec} + CEF_{natgas}C_{natgas} \tag{17}$$

RESULTS AND DISCUSSION

With three sources under examination and thirteen heat pump options, a total of 39 heat pump systems were available for analysis. To make results presentation concise, those heat pump systems utilizing the furnace cooling water as their source are omitted, as these systems have costs between those of equivalent calciner and matte granulation sourced systems. Additionally, only the heat pump model from each brand that resulted in the lowest system cost for a given source of process cooling water is presented. Optimizing the model with cost minimization as the objective generates results displayed in Figure 3. The total annual cost of different heat pump systems is presented as a combination of the cost of powering the heat pumps and circulation pump with electricity, as well as annuitized capital costs and the cost of supplemental natural gas.

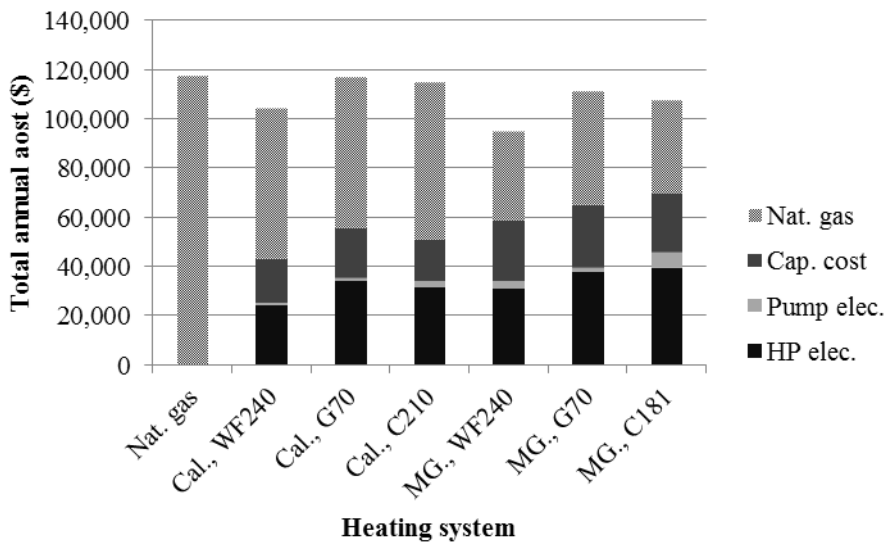

Figure 3. Total annual costs for those systems which presented the lowest total costs after cost minimization. Cal. and MG. are calciner and matte granulation respectively, whilst WF, G and C denote Waterfurnace, Geosmart and Carrier respectively.

For both calciner and matte granulation process cooling water, the heat pump system with the lowest annual cost was based on the Waterfurnace Envision NLV 240 heat pump, which has an exceptionally high coefficient of performance (Table 2). The average COP in table 2 is calculated over a full year of operation, and includes all losses internal to the heat pump, such as compressor and motor efficiency. For the matte granulation sourced systems, the percent of the total cooling demand of their towers that is effectively met by the temperature drop engendered by the heat pump systems is also displayed. This reduction in the total cooling demand placed on the cooling towers will generate additional savings in reduced pumping costs or improved process efficiency. Finally, the annual energy savings attained by sourcing space heating from waste heat is also provided

Table 2. Summary of heat pump systems examined in Figure 3

Source and model	# of heat pumps in bank	Average COP	% cooling load	Energy saved (GWh/yr)
Calciner; Waterfurnace Envision NLV 240	4	6.0		1.4
Calciner; Geosmart Premium G 70	12	4.3		1.3
Calciner; Carrier VQP 210	4	4.4		1.2
Matte Gran.; Waterfurnace Envision NLV 240	6	6.9	6.4	2.4
Matte Gran.; Geosmart Premium G 70	15	4.9	5.2	1.9
Matte Gran.; Carrier VQP 181	8	5.3	5.9	2.3

For some of the cost minimized systems the largest share of the annual cost is natural gas. This outcome is caused by the high initial cost of a heat pump and the subsequent need for high usage to recover this initial cost. As a single heat pump has lower operating costs than the equivalent capacity of natural gas, there is a minimum time of operation at which point the heat pump will pay back its higher initial costs and begin to generate relative savings. However, the marginal heat pump, installed to meet peak heating demand during only the coldest months of the year will often fail to attain the necessary usage over a reasonable payback period. This dynamic leads to cost optimized heat pump systems that rely on significant supplemental heat. For this case study, retrofitting the heat pump system further favors supplemental heat because the sunk capital cost means that supplemental heat is cheaper than a new installation.

In Figure 4, the annual CO_2 emissions associated to the systems of Figure 3 are displayed. It can be seen that all heat pump systems offer a reduction in emissions, with the same system that achieved the lowest system cost simultaneously achieving the highest reduction (62%).

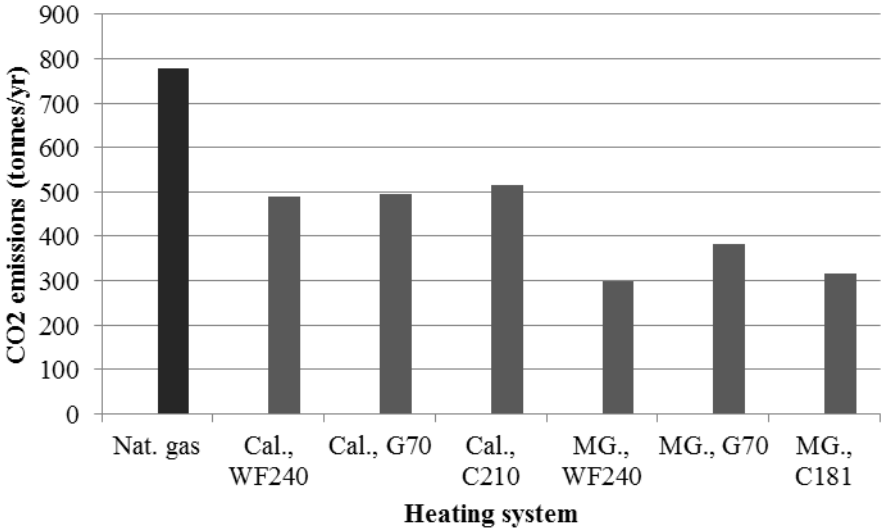

Figure 4. CO_2 emissions generated by the heat pump systems in Figure 3

If minimizing yearly CO_2 emissions is prioritized over a reduction in costs, then the heat pump system will maximize the amount of space heating supplied by the heat pumps. A unit of heating supplied from the heat pumps will have a lower associated CO_2 emissions than a unit from natural gas. This can lead to an increase in costs that exceed the natural gas system, but the reduction in CO_2 emissions is significantly greater than the comparative cost minimized systems. The ability of the heat pump system to meet heating demand is limited by the thermal energy contained in the process cooling water, so even a CO_2 minimized system may still rely on significant supplemental heating. This is demonstrated in Figure 5, which shows results from those systems that generated the lowest CO_2 emissions for the calciner and matte granulation process waters.

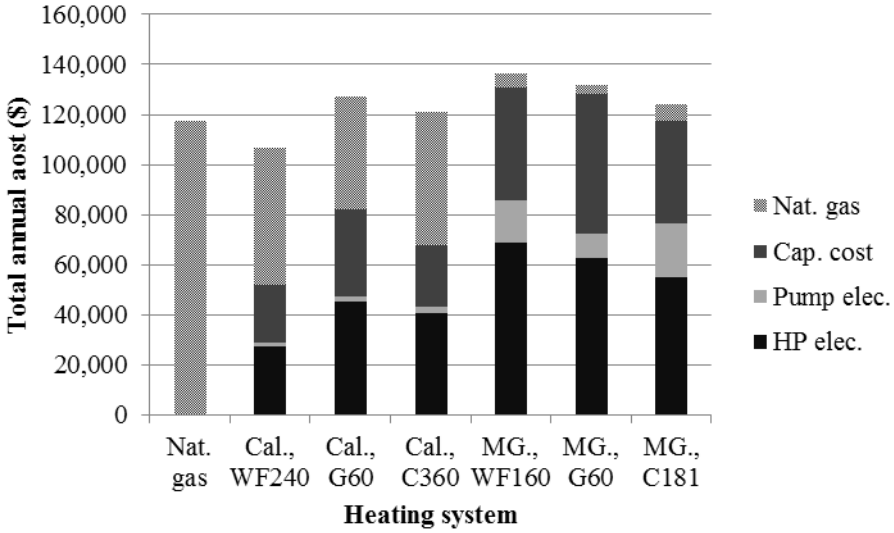

Figure 5. Total annual costs for those systems which presented the lowest CO_2 emissions when optimized for minimum emissions

Compared to Figure 3, consumption of natural gas for supplemental heating as a share of total costs has been significantly reduced for matte granulation sourced systems. This change is less pronounced for calciner sourced systems due to the fact that the thermal energy delivered by the calciner process stream is insufficient to meet the heating demand of the facility in the colder months of the year. This leads to a greater need for supplemental heat and a subsequent decrease in total costs. The total number of heat pumps within the bank and average COP across the year is given in Table 3.

Table 3. Summary of heat pump systems examined in Figure 5

Source and model	# of heat pumps in bank	Average COP	% cooling load	Energy saved (GWh/yr)
Calciner; Waterfurnace Envision NLV 240	6	6.0		1.7
Calciner; Geosmart Premium G 60	27	4.2		1.9
Calciner; Carrier VQP 360	4	4.1		1.6
Matte Gran.; Waterfurnace Envision NLV 160	24	4.3	7.9	3.2
Matte Gran.; Geosmart Premium G 60	46	4.8	8.3	3.4
Matte Gran.; Carrier VQP 181	17	5.3	8.2	3.4

The emissions associated to the heat pump systems of Figure 5 are presented in Figure 6. For matte granulation sourced systems, CO_2 emission reductions as great as 97% are possible. The only CO_2 minimized system to achieve lower total annual costs, the calciner sourced Waterfurnace Envision NLV 240 system, generated an emissions reduction of 44%.

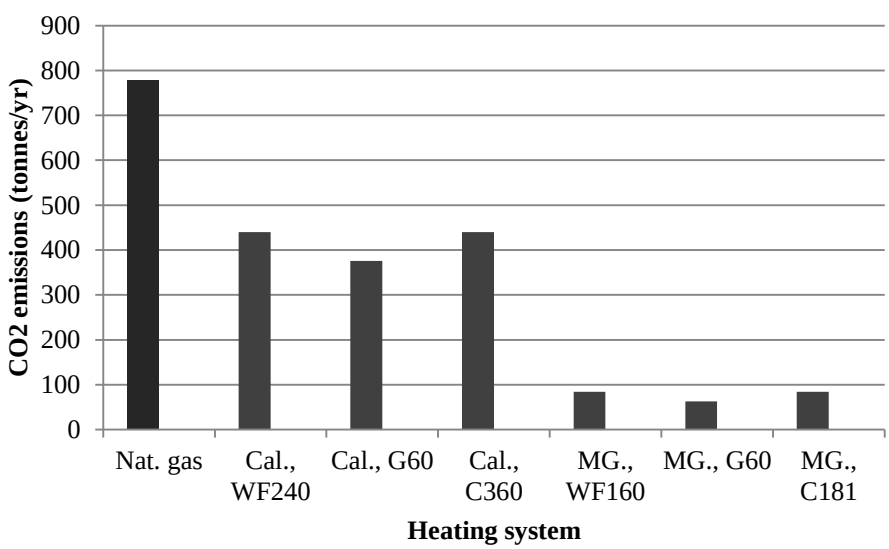

Figure 6. CO_2 emissions generated by the systems shown in Figure 4

Returning to cost optimized results, and taking the lowest cost heat pump system, namely the Waterfurnace Envision NLV 240 based system sourcing process water from matte granulation as a base case ($94,977 annual cost), a sensitivity analysis was performed (Figure 7). The cost of electricity, the price of natural gas, the efficiency of the natural gas furnace, the capital costs of purchasing and installing the heat pumps and the average temperature of the process water are varied from –20% to 20% of the base case. These parameters were chosen because they had the

greatest influence on total annual costs. Furthermore, as the annual cost of the natural gas system is 24% higher than the base cost, significant flex in parameters is possible while still maintaining relatively lower costs.

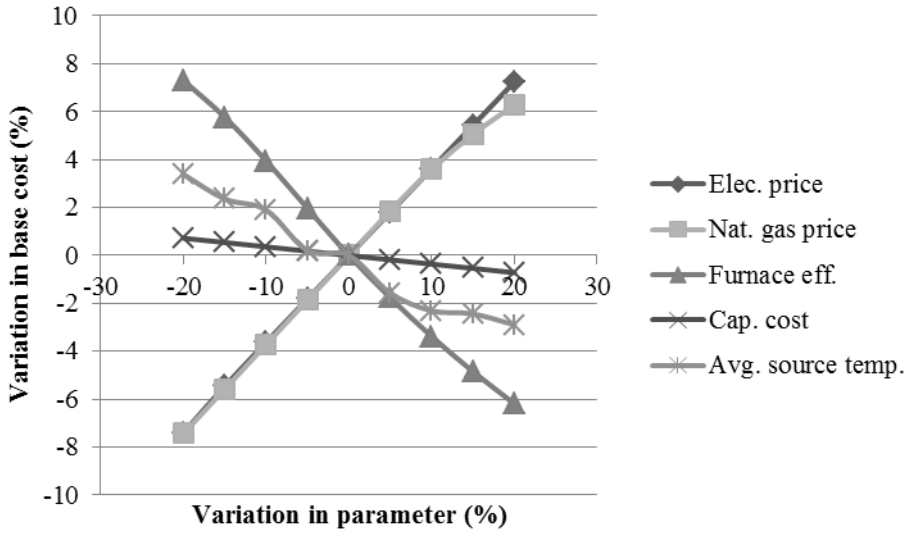

Figure 7. Sensitivity analysis of five key parameters for the Waterfurnace Envision NLV 240 system sourcing water from Matter Granulation

CONCLUSIONS

The process industries, including mining and metallurgical, contain many opportunities for heat recovery using heat pumps that could be capitalized upon to increase plant wide energy efficiency, decrease costs and reduce CO_2 emissions. Industrial facilities can rapidly test for feasibility of heat pump assisted recovery of cooling water waste heat. To demonstrate this, a variety of nickel/copper smelter process cooling water flows of very low grade (10–25°C), were examined for applicability to heat pump assisted space heating of an adjacent office and laboratory facility. The methodology presented is straightforward and robust, and suitable for use in any industrial facility with untapped cooling water flows.

Our analysis highlights that the onsite process cooling water can offer the potential for simultaneous cost and emissions reductions if recovered by an open loop heat pump system. Modelling of smelter process flows indicated that retrofitting a heat pump system could lead to a reduction in space heating costs by 19% and a decrease in CO_2 emissions of 62%. Furthermore, the removal of thermal energy in the process cooling water streams results in a cooler return flow that could further improve the efficiency of the process cooling.

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Ian Ross is a graduate in the Natural Resources Engineering program at Laurentian university and has an undergraduate degree in mechanical engineering from Queen's University. His research interests lie in identifying opportunity for low grade heat recovery and reuse within the mining and mineral processing industry.

Greg Ross is at the Northern Ontario School of Medicine, with research interests in algae sourced biochemicals and public health health, including developing novel rapid detection methods for toxic blue-green algae blooms with remote sensing technologies.

John Scott is cross appointed between the Bharti School of Engineering, Laurentian University and the Northern Ontario School of Medicine. His interests include public health and prospecting mine sites for valuable non-ore resources, such as waste energy, and microalgae for biodiesel and health beneficial compounds.

REFERENCES

Aegent Energy Advisors. (2015). Ontario Electricity Carbon Intensity: How Does It Measure Up? Retrieved from http://www.aegent.ca/newsletters/OntarioCarbonIntensity.html

Ammar, Y., Joyce, S., Norman, R., Wang, Y., & Roskilly, A. P. (2012). Low grade thermal energy sources and uses from the process industry in the UK. *Applied Energy*, *89*(1), 3–20. http://doi.org/10.1016/j.apenergy.2011.06.003

An, Q. (2012). Application of air-source heat pump water heater with exhaust air heat recovery in mining. *China Water and Wastewater*, *28*(23), 101–103.

Bank of Canada. (2015a). Historical Data. Retrieved from http://www.bankofcanada.ca/rates/indicators/capacity-and-inflation-pressures/inflation/historical-data/

Bank of Canada. (2015b, January). Daily Currency Converter. Retrieved from http://www.bankofcanada.ca/rates/exchange/daily-converter/

Banks, D., Fraga Pumar, A., & Watson, I. (2009). The operational performance of Scottish minewater-based ground source heat pump systems. *Quarterly Journal of Engineering Geology and Hydrogeology*, *42*(Part 3), 347–357. Retrieved from http://dx.doi.org/10.1144/1470-9236/08-081

Banks, D., Skarphagen, H., Wiltshire, R., & Jessop, C. (2003). Mine water as a resource: Space heating and cooling via use of heat pumps. *Land Contamination and Reclamation*, *11*(2), 191–198.

Barzi, Y. M., Assadi, M., & Arvesen, H. M. (2014). A novel heat recovery technology from an aluminum reduction cell side walls: Experimental and theoretical investigations. In *Light Metals 2014 - TMS 2014 Annual Meeting and Exhibition, February 16, 2014 - February 20* (pp. 733–738). Faculty of Science and Technology, University of Stavanger, 4036 Stavanger, NorwayGoodtech Recovery Technology AS, Per Krohgs Vei 4, 1065 Oslo, Norway: Minerals, Metals and Materials Society.

Bhattacharjee, K. (2010). Energy conservation opportunities in industrial waste heat recovery systems. *Energy Engineering: Journal of the Association of Energy Engineering*, *107*(6), 7–13.

Braham, D. (1984). Looking to the future use of heat pumps. *Building Services & Environmental Engineer*, *6*(6), 13–15.

Bruckner, S., Liu, S., Miro, L., Radspieler, M., Cabeza, L. F., & Lavemann, E. (2015). Industrial waste heat recovery technologies: An economic analysis of heat transformation technologies. *Applied Energy*, *151*, 157–167. http://doi.org/10.1016/j.apenergy.2015.01.147

Buy Insulation Products. (2016). Fiberglass pipe Insulation. Retrieved April 8, 2016, from http://www.buyinsulationproductstore.com/Fiberglass-Pipe-Insulation-SSL-ASJ/

Canada Revenue Agency. (2015a). Classes of Depreciable Property. Retrieved from http://www.cra-arc.gc.ca/tx/bsnss/tpcs/slprtnr/rprtng/cptl/dprcbl-eng.html#class46

Canada Revenue Agency. (2015b). Corporation Tax Rates. Retrieved from http://www.cra-arc.gc.ca/tx/bsnss/tpcs/crprtns/rts-eng.html

Cane, D., & Garnet, J. (2000). Update on maintenance and service costs of commercial building ground-source heat pump systems. In *ASHRAE Transactions* (Vol. 106, p. PA/ –). Dallas, TX, USA.

Cane, D., Morrison, A., & Ireland, C. (1998). Maintenace and service costs of commercial building ground-source heat pump systems. In *ASHRAE Transactions* (pp. 699–706).

Chao, S., Yiqiang, J., Yang, Y., Shiming, D., & Xinlei, W. (2013). A field study of a wastewater source heat pump for domestic hot water heating. *Building Services Engineering Research and Technology*, *34*(4), 433–448. http://doi.org/10.1177/0143624412463571

Chen, C., Sun, F.-L., Feng, L., & Liu, M. (2005). Underground water-source loop heat-pump air-conditioning system applied in a residential building in Beijing. *Applied Energy*, *82*(4), 331–344. http://doi.org/10.1016/j.apenergy.2004.12.002

Chua, K. J., Chou, S. K., & Yang, W. M. (2010). Advances in heat pump systems: A review. *Applied Energy*, *87*(12), 3611–3624. http://doi.org/10.1016/j.apenergy.2010.06.014

Dalla Rosa, A., Boulter, R., Church, K., & Svendsen, S. (2012). District heating (DH) network design and operation toward a system-wide methodology for optimizing renewable energy solutions (SMORES) in Canada: A case study. *Energy*, *45*(1), 960–974. http://doi.org/10.1016/j.energy.2012.06.062

Demir, H., Mobedi, M., & Ülkü, S. (2008). A review on adsorption heat pump: Problems and solutions. *Renewable and Sustainable Energy Reviews*, *12*(9), 2381–2403. http://doi.org/10.1016/j.rser.2007.06.005

Deng, H., & Gu, Y. (2012). The feasibility analysis of wastewater source heat pump using the urban wastewater heat. *Research Journal of Applied Sciences, Engineering and Technology*, *4*(18), 3501–3504.

Du, H., Dou, Y., & Qi, C. (2013). Application of mine water for water-source heat pump system. *Applied Mechanics and Materials*, *291-294*, 1701–1707. http://doi.org/10.4028/www.scientific.net/AMM.291-294.1701

Environment Canada. (2013). Fuel Combustion. Retrieved from http://www.ec.gc.ca/ges-ghg/default.asp?lang=En&n=AC2B7641-1

Fang, H., Xia, J., & Jiang, Y. (2015). Key issues and solutions in a district heating system using low-grade industrial waste heat. *Energy*, *86*, 589–602. http://doi.org/10.1016/j.energy.2015.04.052

Fanisalek, H., Bashiri, M., & Kamali, R. (2012). Aluminum smelter waste heat recovery plant (heat exchangers fouling and corrosion-a detailed investigation). In *Energy Technology 2012: Carbon Dioxide Management and Other Technologies - TMS 2012 Annual Meeting and Exhibition, March 11, 2012 - March 15* (pp. 203–214). Payam-e-Noor University, Bandar Abbas, IranRazi University, Kermanshah, Iran: Minerals, Metals and Materials Society.

Feng, X., & Berntsson, T. (1997). Critical COP for an economically feasible industrial heat-pump application. *Applied Thermal Engineering*, *17*(1), 93–101. http://doi.org/10.1016/1359-4311(96)00010-5

Fleer, M. (2010). *Heat recovery from the exhaust gas of aluminum reduction cells*. Reykjavik University.

Fleer, M., Lorentsen, O.-A., Harvey, W., Palsson, H., & Saevarsdottir, G. (2010). Heat recovery from the exhaust gas of aluminum reduction cells. In *Light Metals 2010 - TMS 2010 Annual Meeting and Exhibition, February 14, 2010 - February 18* (pp. 243–248). Reykjavik Energy Graduate School of Sustainable Systems, Reykjavik, IcelandHydro Aluminum, Primary Metal Technology, Porsgrunn, NorwayReykjavik University, Reykjavik, IcelandUniversity of Iceland, Reykjavik, Iceland: Minerals, Metals and Materials Society.

Follinger, T., & Merker, G. P. (1992). Periodic high-temperature heat pumps for industrial heat recovery. II. Example of application and design of apparatus. *Heat Recovery Systems & CHP*, *12*(3), 257–269. http://doi.org/10.1016/0890-4332(92)90054-L

Fox, D. B., Sutter, D., & Tester, J. W. (2011). The thermal spectrum of low-temperature energy use in the United

States. *Energy and Environmental Science*, *4*(10), 3731–3740. http://doi.org/10.1039/c1ee01722e

Gorianec, D., Pozeb, V., Tomi, L., & Trop, P. (2014). Exploitation of the waste-heat from hydro power plants. *Energy*, *77*, 220–225. http://doi.org/10.1016/j.energy.2014.06.106

Gu, Y., & Wang, S. (2013). Research on the energy-saving and economic analysis of wastewater source heat pump. *Energy Education Science and Technology Part A: Energy Science and Research*, *31*(2), 1167–1170.

Hall, A., Scott, J. A., & Shang, H. (2011). Geothermal energy recovery from underground mines. *Renewable and Sustainable Energy Reviews*, *15*(2), 916–924. http://doi.org/10.1016/j.rser.2010.11.007

Hammond, G. P., & Norman, J. B. (2014). Heat recovery opportunities in UK industry. *Applied Energy*, *116*, 387–397. http://doi.org/10.1016/j.apenergy.2013.11.008

Hebenstreit, B., Schnetzinger, R., Ohnmacht, R., Höftberger, E., Lundgren, J., Haslinger, W., & Toffolo, A. (2014). Techno-economic study of a heat pump enhanced flue gas heat recovery for biomass boilers. *Biomass and Bioenergy*, *71*, 12–22. http://doi.org/10.1016/j.biombioe.2014.01.048

Holland, F., & Bragg, R. (2002). *Fluid Flow for Chemical and Process Engineers*. Butterworth-Heinemann.

Hung, T. C., Shai, T. Y., & Wang, S. K. (1997). A review of organic rankine cycles (ORCs) for the recovery of low-grade waste heat. *Energy*, *22*(7), 661–667. http://doi.org/10.1016/S0360-5442(96)00165-X

IEA. (2013). *Energy Efficiency Market Report 2013*. Retrieved from http://www.iea.org/publications/freepublications/publication/EEMR2013_free.pdf

IEA. (2014). Energy Snapshot of the Week. Retrieved April 5, 2016, from https://www.iea.org/newsroomandevents/graphics/2014-07-22-what-drives-energy-consumption-in-households.html

IEA. (2016). Energy Snapshot of the Week. Retrieved April 6, 2016, from https://www.iea.org/newsroomandevents/graphics/2015-10-20-savings-from-energy-efficiency-investments-made-since-1990.html

IESO. (2015). Power Data. Retrieved from http://www.ieso.ca/Pages/Power-Data/default.aspx#download

Incropera, F. P., De Witt, D. P., Bergman, T. L., & Lavine, A. S. (2006). *Fundamentals of heat and mass transfer*. *Applied Mechanics Reviews* (Vol. 44). John Wiley & Sons.

Jinggang, W., Meixia, D., Chuanchuan, D., & Xiaoxia, G. (2010). Exergy analysis of mine water source heat pump system. *Advanced Materials Research*, *121-122*, 980–985. http://doi.org/10.4028/www.scientific.net/AMR.121-122.980

Johnson, I., Choate, B., & Dillich, S. (2008). Waste heat recovery: Opportunities and challenges. In *TMS Annual Meeting* (pp. 47–52). New Orleans, LA, United states.

Kahraman, A., & Çelebi, A. (2009). Investigation of the Performance of a Heat Pump Using Waste Water as a Heat Source. *Energies*, *2*(3), 697–713. http://doi.org/10.3390/en20300697

Karu, V., Valgma, I., & Kolats, M. (2013). Mine water as a potential source of energy from underground mined areas in Estonian oil shale deposit. *Oil Shale*, *30*(2 SUPPL.), 336–362. Retrieved from http://dx.doi.org/10.3176/oil.2013.2S.12

Kelso, R. M., & Johnson, W. S. (2015). Flooded mine water-to-air heat pump system. *ASHRAE Journal*, *57*(8), 34–38.

Kew, J. (1985). HEAT PUMPS FOR BUILDING SERVICES. *Technical Note TN - BSRIA (Building Services Research and Information Association)*, (8).

Kwak, D.-H., Oh, S.-Y., & Kim, J.-K. (2012). Process integration study for the use of industrial low grade heat (Vol. 29, pp. 1591–1596). Department of Chemical Engineering, Hanyang University, 222 Wangsimni-ro, Seongdong-gu, Seoul, 133-791, Korea, Republic of: Italian Association of Chemical Engineering - AIDIC. http://doi.org/10.3303/CET1229266

Law, R., Harvey, A., & Reay, D. (2013). Opportunities for low-grade heat recovery in the UK food processing industry. *Applied Thermal Engineering*, *53*(2), 188–196. http://doi.org/10.1016/j.applthermaleng.2012.03.024

Lawson, D., & Sonderegger, J. (1978). *Report (geothermal data-base study)*. Butte (USA).

Lecompte, S., Huisseune, H., van den Broek, M., Vanslambrouck, B., & De Paepe, M. (2015). Review of organic Rankine cycle (ORC) architectures for waste heat recovery. *Renewable and Sustainable Energy Reviews*, *47*, 448–461. http://doi.org/10.1016/j.rser.2015.03.089

Li, X., Mu, G., Zhang, G., Sun, C., Zhang, Y., & Wang, Y. (2014). Application effect of seawater source heat pump for water heating and waste heat recovery in flatfish culture system. *Nongye Gongcheng Xuebao/Transactions of the Chinese Society of Agricultural Engineering*, *30*(3), 142–148. http://doi.org/10.3969/j.issn.1002-6819.2014.03.019

Lindley, N. L., & Floyd, J. C. (1993). Piping systems: How installation costs stack up. *Chemical Engineering*. Retrieved from http://search.proquest.com/docview/194424789?accountid=11956

Liu, X., Lau, S.-K., & Li, H. (2014). Optimization and analysis of a multi-functional heat pump system with air source and gray water source in heating mode. *Energy and Buildings*, *69*, 1–13. http://doi.org/10.1016/j.enbuild.2013.09.039

Loken, M. (2013). *Recovery of Waste Heat from Pyrometallurgical Facilites for use in Microalgae production*. Laurentian University.

MathWorks. (2016). MATLAB. Retrieved April 2, 2016, from http://www.mathworks.com/products/matlab/

Meggers, F., & Leibundgut, H. (2011). The potential of wastewater heat and exergy: Decentralized high-temperature recovery with a heat pump. *Energy and Buildings*, *43*(4), 879–886. http://doi.org/10.1016/j.enbuild.2010.12.008

Mei, V. C., & Nephew, E. A. (1987). *Life-cycle cost analysis of residential heat pumps and alternative HVAC systems*. Oak Ridge National Laboratory.

Moran, M. J., Shapiro, H. N., Boettner, D., & Bailey, M. B. (2011). *Fundamentals of Engineering Thermodynamics* (Vol. 6). John Wiley & Sons.

Munson, B. R., Young, D. F., Okiishi, T. H., & Huebsch, W. W. (2009). *Fundamentals of Fluid Mechanics* (Vol. 6th). USA: John Wiley & Sons, Inc.

Mustafa Omer, A. (2008). Ground-source heat pumps systems and applications. *Renewable and Sustainable Energy Reviews*, *12*(2), 344–371. http://doi.org/10.1016/j.rser.2006.10.003

NASA. (2016). NASA surface meteorology and solar energy. Retrieved April 8, 2016, from https://eosweb.larc.nasa.gov/cgi-bin/sse/retscreen.cgi?email=&step=1&lat=46.52&lon=-80.95&submit=Submit

Ni, L., Lau, S. K., Li, H., Zhang, T., Stansbury, J. S., Shi, J., & Neal, J. (2012). Feasibility study of a localized residential grey water energy-recovery system. *Applied Thermal Engineering*, *39*, 53–62. http://doi.org/10.1016/j.applthermaleng.2012.01.031

Nowicki, C., & Gosselin, L. (2012). An overview of opportunities for waste heat recovery and thermal integration in the primary aluminum industry. *JOM*, *64*(8), 990–996. http://doi.org/10.1007/s11837-012-0367-4

Oanda. (2015). Historical Exchange Rates. Retrieved from http://www.oanda.com/currency/historical-rates/

Qiu, W., Zhang, L., & Liu, Q. (2013). Research on the heating plan of recovering condensing heat of power plant by water source heat pump unit. *Advanced Materials Research*, *608-609*, 1241–1245. http://doi.org/10.4028/www.scientific.net/AMR.608-609.1241

Rafferty, K. (2001). *An Information Survival Kit for the Prospective Geothermal Heat Pump Owner*. Retrieved from http://www.builditsolar.com/References/Calculators/Fuels/heat pump survival.pdf

Raymond, J., & Therrien, R. (2007). Geothermal Energy Resources on Active Mines. In *CIM Montreal 2007*.

Raymond, J., & Therrien, R. (2014). Optimizing the design of a geothermal district heating and cooling system located at a flooded mine in Canada. *Hydrogeology Journal*, *22*(1), 217–231. http://doi.org/10.1007/s10040-013-1063-3

Raymond, J., Therrien, R., & Gosselin, L. (2010). Low-temperature geothermal energy in mining environments. *CIM Journal*, *1*(2).

Rezaie, B., & Rosen, M. A. (2012). District heating and cooling: Review of technology and potential enhancements. *Applied Energy*, *93*, 2–10. http://doi.org/10.1016/j.apenergy.2011.04.020

Richardson, J. J., & Lopez, D. L. (2014). Investigating heat flow through a flooded coal mine complex for heat pump applications. *Geological Society of America, 2014 Annual Meeting & Exposition*, *46*(6), 119.

Sarbu, I., & Sebarchievici, C. (2014). General review of ground-source heat pump systems for heating and cooling of buildings. *Energy and Buildings*, *70*, 441–454. http://doi.org/10.1016/j.enbuild.2013.11.068

Seck, G. S., Guerassimoff, G., & Maizi, N. (2013). Heat recovery with heat pumps in non-energy intensive industry: A detailed bottom-up model analysis in the French food drink industry. *Applied Energy*, *111*, 489–504. http://doi.org/10.1016/j.apenergy.2013.05.035

Self, S. J., Reddy, B. V., & Rosen, M. A. (2013). Geothermal heat pump systems: Status review and comparison with other heating options. *Applied Energy*, *101*, 341–348. http://doi.org/10.1016/j.apenergy.2012.01.048

Sever, J. C. (2013). Waste heat recovery opportunities in a magnesium silicothermic reduction plant. In *TMS Annual Meeting* (pp. 69–75). San Antonio, TX, United states.

Smith, L. H., & Arthur, D. C. (1996). Mine ventilation: Waste heat recovery. *CIM Bulletin*, *89*(998), 126–130.

Soni, S. K., Pandey, M., & Bartaria, V. N. (2015). Ground coupled heat exchangers: A review and applications. *Renewable and Sustainable Energy Reviews*, *47*, 83–92. http://doi.org/10.1016/j.rser.2015.03.014

Spoelstra, S., Haije, W. G., & Dijkstra, J. W. (2002). Techno-economic feasibility of high-temperature high-lift chemical heat pumps for upgrading industrial waste heat. *Applied Thermal Engineering*, *22*(14), 1619–1630. http://doi.org/10.1016/S1359-4311(02)00077-7

Staffell, I., Brett, D., Brandon, N., & Hawkes, A. (2012). A review of domestic heat pumps. *Energy and Environmental Science*, *5*(11), 9291–9306. http://doi.org/10.1039/c2ee22653g

Union Gas. (2015). Important Information About Your Rates. Retrieved from https://www.uniongas.com/~/media/aboutus/rates/residential/rate310.pdf?la=en

Urdaneta-B, A., & Schmidt, P. S. (1980). Thermodynamic and economic analysis of heat pumps for energy recovery in industrial processes. *Transactions of the ASME.Journal of Energy Resources Technology*, *102*(3), 173–180. http://doi.org/10.1115/1.3227869

Wallin, E., & Berntsson, T. (1994). Integration of Heat Pumps in Industrial Processes. *Heat Recovery Systems & CHP*, *14*(3), 287–296.

Wallin, E., Franck, P. A., & Berntsson, T. (1990). Heat pumps in industrial processes. An optimization methodology. *Heat Recovery Systems & CHP*, *10*(4), 437–446.

Wang, F., Wang, Z., Zheng, Y., Lin, Z., Hao, P., Huan, C., & Wang, T. (2015). Performance investigation of a novel frost-free air-source heat pump water heater combined with energy storage and dehumidification. *Applied Energy*, *139*, 212–219. http://doi.org/10.1016/j.apenergy.2014.11.018

Wang, Y., Zhao, F.-Y., Kuckelkorn, J., Spliethoff, H., & Rank, E. (2014). School building energy performance and classroom air environment implemented with the heat recovery heat pump and displacement ventilation system. *Applied Energy*, *114*, 58–68. http://doi.org/10.1016/j.apenergy.2013.09.020

Watzlaf, G. R., & Ackman, T. E. (2006). Underground Mine Water for Heating and Cooling using Geothermal Heat Pump Systems. *Mine Water and the Environment*, *25*(1), 1–14. http://doi.org/10.1007/s10230-006-0103-9

Wedde, G., & Sorhuus, A. (2012). Waste heat recovery from industrial smelting exhaust gas. In *International Smelting Technology Symposium, Incorporating the 6th Advances in Sulfide Smelting Symposium - TMS 2012 Annual Meeting and Exhibition, March 11, 2012 - March 15* (pp. 31–38). Alstom Norway, Drammensveien 165, 0277 Oslo, Norway: Minerals, Metals and Materials Society.

Wright, J. R., & Steward, F. R. (1985). Three Industrial Heat Pump Installations Operating in Canada. *Journal of Heat Recovery Systems*, *5*(2), 81–88.

Wu, W., Wang, B., Shi, W., & Li, X. (2014). Absorption heating technologies: A review and perspective. *Applied Energy*, *130*, 51–71. http://doi.org/10.1016/j.apenergy.2014.05.027

Wu, W., You, T., Wang, B., Shi, W., & Li, X. (2014). Simulation of a combined heating, cooling and domestic hot water system based on ground source absorption heat pump. *Applied Energy*, *126*, 113–122. http://doi.org/10.1016/j.apenergy.2014.04.006

Xiao, S. H., Luo, Q. H., & Li, G. F. (2014). Utilizing thermoelectric heat pump to heat recovery of shower Waste water. In *2013 2nd International Conference on Sustainable Energy and Environmental Engineering, ICSEEE 2013, December 28, 2013 - December 29* (Vol. 521, pp. 757–761). School of Urban Construction, University of South China, Hengyang, Hunan, 421000, China: Trans Tech Publications. http://doi.org/10.4028/www.scientific.net/AMM.521.757

XPS. (2016). XstrataXWPBranding. Retrieved April 8, 2016, from http://www.xps.ca/XstrataXWPBranding/Images/XPS6.jpg

Yang, H., Cui, P., & Fang, Z. (2010). Vertical-borehole ground-coupled heat pumps: A review of models and systems. *Applied Energy*, *87*(1), 16–27. http://doi.org/10.1016/j.apenergy.2009.04.038

Zaheer-Uddin, M. (1992). Multiple setpoint control of a heat recovery system. *Heat Recovery Systems & CHP*, *12*(1), 23–37. http://doi.org/10.1016/0890-4332(92)90004-2

Zebarjadi, M., Esfarjani, K., Dresselhaus, M. S., Ren, Z. F., & Chen, G. (2012). Perspectives on thermoelectrics: From fundamentals to device applications. *Energy and Environmental Science*, *5*(1), 5147–5162. http://doi.org/10.1039/c1ee02497c

Zhu, N., Hu, P., Xu, L., Jiang, Z., & Lei, F. (2014). Recent research and applications of ground source heat pump integrated with thermal energy storage systems: A review. *Applied Thermal Engineering*, *71*(1), 142–151. http://doi.org/10.1016/j.applthermaleng.2014.06.040

NOMENCLATURE

AV	annual value ($/yr)
Cap	rated capacity (kW)
CC	capital cost ($)
CEF_{elec}	carbon emission factor for Ontario electricity (tonnes CO_2/kWh)
CEF_{natgas}	carbon emission factor for natural gas (tonnes CO_2/m^3)
C_{elec}	consumption of electricity (kWh)
C_{natgas}	consumption of natural gas (m^3)
COP	coefficient of performance
c_p	specific heat capacity (J/kg K)
D	inner diameter (m)
d	depreciation rate
E	emissions (tonnes CO_2)
HC	temperature specific capacity (W)
i	interest rate/IRR
k	thermal conductivity (W/m K)
L	pipe length
L_e	equivalent pipe length
l	section length (m)
n	compounding periods
Nu	nusselt number
P	power demand (W)
Pr	prandtl number
PV	present value ($)
q	heat transfer (W)
Q	volumetric flow rate (m^3/s)
R	thermal resistance (K/W)
Re	reynolds number
t	thickness (m)
T	temperature (K)
T_C	temperature of source (K)
T_H	temperature of sink (K)
tr	tax rate
V	fluid velcotiy (m/s)
W	work (W)
ΔP	pressure drop (Pa)
ΔT	temperature change (K)
ε	surface roughness (m)

η	efficiency
ρ	density (kg/m^3)