An Integrated Wind-Powered Energy System for Residential Applications in Cold Regions

by

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The above committee determined that the thesis is acceptable in form and content and that a satisfactory knowledge of the field covered by the thesis was demonstrated by the candidate duringan oral examination. A signed copy of the Certificate of Approval is available from the School ofGraduate and Postdoctoral Studies.

Abstract

This thesis proposed an integrated direct wind-powered energy system that can provide drinking water, domestic hot water, space heating as well as electricity for residential applications of a remote community in a cold region. The novelty of the system is that it uses a wind turbine to directly power the heat pump, with the sea water as the evaporator that provides a constant temperature, without the need to use the wind turbine to convert the wind energy to electricity to power the heat pump. Environmental data from the selected location and the Engineering Equation Solver were used to perform the thermodynamic analyses of the system. The results indicate that the proposed system is promising. It is shown that in addition to being more environmentally benign, purchasing a wind turbine is more cost-effective over its lifetime than purchasing electricity from the local grid.

Keywords: Energy system; wind turbine; heat pump; greenhouse gases; desalination; PEM electrolyzer

AUTHOR'S DECLARATION

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STATEMENT OF CONTRIBUTIONS

I hereby certify that I am the sole author of this thesis and that no part of this thesis has been published or submitted for publication. I have used standard referencing practices to acknowledge ideas, research techniques, or other materials that belong to others. Furthermore, I hereby certify that I am the sole source of the creative works and/or inventive knowledge described in this thesis.

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List of Abbreviations and Symbols

C	Specific heat	kJ $(kg.K)^{-1}$
$\dot{E}x$	exergy destruction rate	kW
h	Specific enthalpy	kJ kg^{-1}
$\dot{m}$	Mass flow rate	kg s^{-1}
P	Pressure	kPa
$\dot{q}$	Heating capacity	kW
$\dot{Q}$	Heat rate	kW
s	Entropy	kJ $(kg.K)^{-1}$
t	Temperature	^0C
T	time	s
$\dot{S}$	Entropy rate	kW $(kg.K)^{-1}$
$\dot{W}$	Wind power	kW

Acronyms

BL	Building load
Ca	Calcium
$Ca(OH)_2$	Calcium hydroxide
CO_2	carbon dioxide
COP	Coefficient of performance
eq	Equivalence
Et	Exajoule
GDP	Gross domestic product

GHG	Greenhouse gas
Gt	Gigatonne
H_2O	Water
HP	Heat pump
HVAC	Heating,ventilaton and air condition
IEA	International energy agency
kL	Kiloliters
LCOE	lowest Levelized cost of energy
LHV	Low heating value
MEE	Multiple effect evaporation
PCM	Phase change materials
PEM	Proton-Exchange Membrane
PHES	Pumped heat energy storage
PJ	Petajoule
SHS	Sensible Heat Storage
TCS	Thermo-Chemical Storage
TES	Thermal Energy Storage
WEC	Wind energy converter
WTES	Wind-powered thermal energy storage

Greek letters

Symbol	Name	unit
Δ	Change in quantity	
ρ	Density	kg m^{-3}
η_c	Isentropic efficiency	%
ψ	Overall exergy efficiency	

Subscripts

Symbol	Name
C	Compressor
e	evaporator
HWHE	hot water heat exchanger
in	in
j	Outdoor bin temperature
OD	outdoor
out	out
r	Refrigerant
z	Slope factor
zi	Zero-load

Chapter 1: Introduction

1.1 Background

Global energy demand has increased due to the increase in the human population and the increased per capita energy consumption due to development. According to the United Nations, the world population will increase by 47% by 2050 [1]. To meet the high global energy demand, fossil fuels like gasoline, coal, kerosene and natural gas are employed at high economic and environmental costs. About 85% of the world's primary energy is from fossil fuels [2]. The biggest problem with fossil fuels is the greenhouse gas (GHG) emission they create, which causes climate change - the environmental crisis of our time.

Further, it is projected that the world population would reach 8.9 billion by the year 2050 [3]. This value represents a 47% increased as compared to that of 2000. Energy supply will confront substantial hurdles in terms of sustainability and parity, given that fossil fuels produced most of the primary energy [4].

According to the International Energy Agency [5], global primary energy demand is projected to increase by 45% by 2040 under the current policy scenario. In terms of the energy mix, fossil fuels will constitute 81% of global energy use, and CO_2 emissions will hit 44.1 Gt by 2040 under current policies. Even though fossil fuel's share of global main energy reliance will be reduced to 79 percent by 2040, this upward trend will continue. Given these projections, there is the need for low-carbon renewable energy systems to emerge, particularly as gas and oil extraction costs continue to rise. Renewable energy is, therefore, the most reliable and gainful option to mitigate the current menace of CO_2 on the environment.

Energy use for heating is projected to continue to grow. Renewable energy has, however, played a negligible role in heating, not considering the conventional use of biomass for space heating and cooking. Some commercially available renewable energy technologies can provide green, sustainable, and economical heating energy and can compete with fossil fuel heat generation sources with the right incentives [6]. These include heat pumps, electric cars and solar panels.

The government of Canada, under the Climate Protection Act, signed an agreement to reduce CO_2 emissions by 80% in 2050 [7]. Buildings produce 17% of GHG emissions in Canada, while 40% of electricity generated is used for buildings [8]. In frigid areas, cooling and heating account for over 70% of the total energy required for buildings [9]. Furnaces that burn fossil fuels or use electrical resistance elements are the most common means of heating homes in these areas [10], [11]. The two predominant types of furnaces that account for almost three-quarters of all heating systems use either natural gas or oil as their main source of energy. These fuels are also responsible for over 90% of all residential greenhouse gas (GHG) emissions [11]. To reduce GHG emissions, there is the need to phase out gas and oil furnaces and deploy those that can run on renewable energy [12]. These include renewable sources of energy such as hydro, solar, biomass, wind and tidal energy [13]. Electric furnaces and heat pumps can use renewable sources of energy, however, the former is one-third as efficient as heat pumps [14],[15].

Wind energy is an established technology and is cheaper than solar energy due to its high efficiency [16], but the initial investment and maintenance for wind energy may be higher than solar. Several studies have examined the conversion of wind energy into electrical energy [17], [18]. However, far less research is done on directly converting the kinetic energy of the wind to thermal energy for space heating and hot water supply. This study will explore the application of wind energy for space heating directly in cold regions.

Nunavut is one of the coldest provinces in Canada. They do not have a natural gas pipeline or large-scale liquefied natural gas facilities. Also, they do not have a provincial electricity grid because most communities are far away from each other. All the electricity is usually generated and distributed at the community level. This prevents communities from sharing excess electricity generated [19]. In 2017 Nunavut used 6.7 PJ of energy. Refined petroleum products account for 87% of fuel type while electricity accounts for the rest. Around 34% of it was used for heating. In addition, Nunavut has the highest electricity rate in Canada. As a result, it is more economical to use wind turbines to create energy.

1.2 Motivation

Before 1900 a vessel travelling from Shanghai, China to New York, USA, would have to go around South America which is about a 30,000 km trip. In 1914, the Panama Canal was opened, which cut the trip to 21,000 km [20]. In the past years, more than 13,000 vessels a year on average pass through the Panama Canal generating over $2.6 billion in toll fees a year, as shown in Table 1.1.

Table. 1.1: Panama canal traffic [21]

Year	Number of transits	Toll fees ($ millions)
2018	13,795	2,485
2019	13,785	2,593
2020	13,369	2,663

However, the Panama canal has limitations; only vessels that are 168 feet wide and 1200 long can pass through it [22].

More recently, MS Nordic Orion [23] passed through the northwest passage in 2013, which is located in Nunavut and the Northwest Territories [24]. The vessel was carrying 73,500 tons of goods which is 20% over the weight of goods allowed through the Panama canal [25]. The vessel was going from the port metro Vancouver, Canada, to the Port of Pori, Finland, cutting down the trip by 2,000 km and saving over $80,000 in fuel. There is, therefore, a huge potential of establishing a major port in Canada's north. In August 2007, the Canadian Government started plans to build the Nanisivik Naval facility in Baffin Island, Nunavut [26]. The facility is to house between 50 to 60 persons [27]. The base is to be primarily used for arctic refuelling patrol, ice breakers station and a military presence up north. The facility will also have a helicopter landing area and 162,000-litre diesel storage tanks. The whole cost of the facility is estimated to be around $258 million [28]. The challenge of the project is the extreme cold and the provision of efficient, cost-effective and sustainable energy systems to power such an extremely cold and remote community.

1.3 Objectives

The main objective of this thesis is to develop an integrated direct wind-powered energy system that can provide drinking water, domestic hot water, space heating as well as electricity for residential applications of a remote community in a cold region. Space heating consumes 70% of residential energy, as mentioned earlier, and it is also challenging to develop a space heating system that operates efficiently, cost-effectively and sustainably in cold climates like Canada's north. The study will focus on incorporating an innovative heat pump system into the integrated system to achieve the objective. The study will use a residential community of the Canadian Navy and Coast Guard that houses 60 officers as a case study. The community is located at Resolute Bay, Nunavut. The novelty of the present study is that the subsystem for heating and cooling will employ direct wind-powered heat pumps, eliminating the need to convert wind energy to electricity. The integrated system is thermodynamically, economically and environmentally studied and compared to current options available in Resolute Bay, Nunavut. The specific objectives are as follows:

1. To develop a clean energy system that can provide sufficient energy for electricity, heating, drinking water and domestic hot for a small residential community in a remote and extremely cold region of Canada.
2. To incorporate a high efficiency and cost-effective space heating system in the proposed energy system to achieve the objective.
3. To conduct detailed thermodynamic analyses of the proposed system and assess its performance

1.3 Thesis Structure

The following are the outline of the thesis:

- Chapter 1 is the introduction of the thesis, which highlights the environmental problems associated with energy production and utilization. It also shows the motivation and objective of the thesis.
- Chapter 2 summarizes the literature and state of the art in this field. It also identifies the gaps in the literature.

- Chapter 3 presents the methodology employed in this study.
- Chapter 4 is the discussion of the results.
- Chapter 5 provides the conclusions and recommendations for future studies.

Chapter 2: Literature Review

A number of studies have been undertaken on the cost of electricity production from renewable energy sources. Three renewable energy systems are currently directly in competition with fossil fuels, including hydropower, biomass, and geothermal power generation. Among these technologies, onshore wind energy is considered the most viable because it has the lowest Levelized cost of energy (LCOE) [29]. Although economically competitive, wind energy is still hindered by weather conditions when considering large-scale deployment. The use of various storage systems as the first solution thus arises. However, many commercially available large storage systems, such as pumped storage and batteries, have relatively high investment costs [30]. Instead of being viewed as a means to an end, storage could be viewed as a means of creating a secure, efficient, cost-effective, and environmentally friendly energy industry by improving renewable energy deployment and integration.

In this section, the review is broken down into subsections focusing on the different aspects of energy systems.

2.1 Space Heating in Cold Regions

The building sector currently accounts for about 30% of the world's energy consumption [31] and 47% of that used in heating [32]. As population and GDP continue to rise, energy demand for space heating, especially in cold regions, will increase accordingly.

Currently, a large percentage of the heating load in cold climates is met by fossil fuels in furnaces and this contributes directly to a large proportion of emissions. In the annals of the international energy agency (IEA), about 37% of the world's CO_2 emissions comes from the building sector when the construction and use stages are taken into account [31]. Thus, space heating contributes significantly to the prevalent GHG emissions.

Energy usage for space heating is driven by the size of the floor area and the constant increase in the number of households. To reduce the emissions coming from the building sector, various energy efficiency techniques and technologies are currently being employed. For example, Ordonez and Modi [33] studied the optimization of the building materials and thermal insulation. Also, automation of the building and HVAC systems has also been studied [34–36].

In an attempt to reduce the energy consumed in space heating, several studies have been carried out with the aim of improving the existing methods or technologies of space heating. For example, Zhang et al. [37] proposed a ground source heat pump that is driven by heat extracted from the duct network of a traditional heating system and which then extracts heat from a geothermal energy source. The system is reported to be capable of increasing the efficiency of the conventional system by 54% and also reduce yearly operating costs by almost 30%.

Furthermore, Hamada et al. [38] undertook a field study on the actual performance of a space heating system that uses the pile foundation of a building as heat exchangers for space air-conditioning. They reported an overall COP of 3.2 and energy reduction during the heating period as 23.2%. In a similar study, the authors in [39] investigated the performance of a geothermal energy pile technique on the University of Manitoba campus. The system harvests energy from the heat loss in the piles and rejects it into the HVAC system in the building. Though geothermal piles are mostly used for heating during the winter period in cold regions, the fact they are very susceptible to thermal imbalance limits their use.

Additionally, the work presented by [40] investigated the performance of four heating methods based on results obtained through field measurement. They looked at the operation of a solar collector system, ground source heat pump, gas boiler and a biomass boiler located in a residential building. Based on their studies, it was shown that the solar collector has a performance range of 12.5%-7.4%, the heat pump system has an annual COP of 3.5 - 3.8, the gas boiler operated within the range of 77.5% - 82.2% and the biomass boiler performance ranged from 30% - 50%. The results obtained revealed that it is possible for renewable energy sources to contribute the largest share of the residential heat requirements.

Moreover, research done by [41] investigated the feasibility of recovering heat generated in a data center using an AC with a thermosyphon. The results obtained showed the recovered heat was enough to significantly reduce the heating and cooling energy consumption of the data center as well as the building itself. A schematic of the proposed heat recovery system is shown in Figure 2.1.

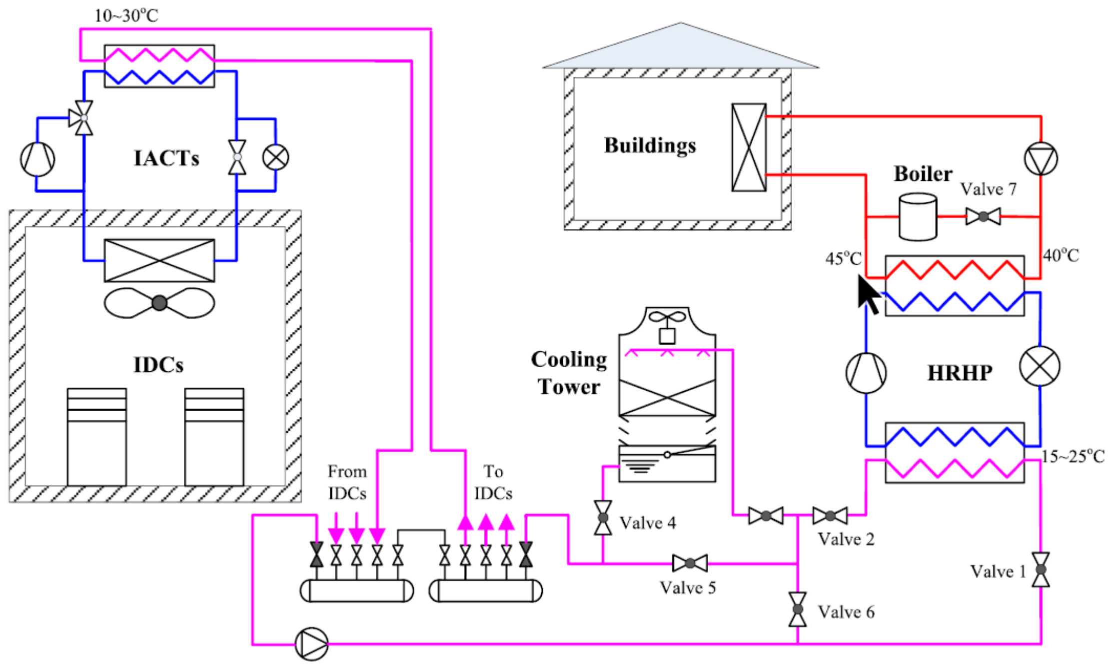

Figure 2.1: Schematic of a heat recovery system using a combined AC with thermosyphon [41].

The effectiveness of a solar-aided heat pump based on the indirect expansion of carbon dioxide as the refrigerant is investigated in [42]. In order to assess the effectiveness and to fully comprehend the system, a model was developed using a one-stage trans-critical CO_2 cycle, a two-stage trans-critical CO_2 cycle and the conventional single-stage R410a for space heat in Toronto. The operational changes of the three systems were simulated, compared, and presented using these models with the solar thermal system, reservoir, and heat pump compressor capacity. The two-stage system was reported to perform better than the other cold regions.

Additionally, the authors in [43] proposed a thermal energy system consisting of a wind turbine, heat pump and a pond. Three different configurations of the proposed system were analyzed: heat pump powered by wind integrated with a pond as the evaporator, wind turbine and heat pump without pond and heat pump powered by electricity from the grid. Based on the thermodynamic analysis, the proposed system of a wind-powered heat pump integrated with the pond was reported to outperform the other configurations in terms of overall coefficient of performance, exergy, and heat capacity. A conceptual design of the system is shown in Figure 2.2.

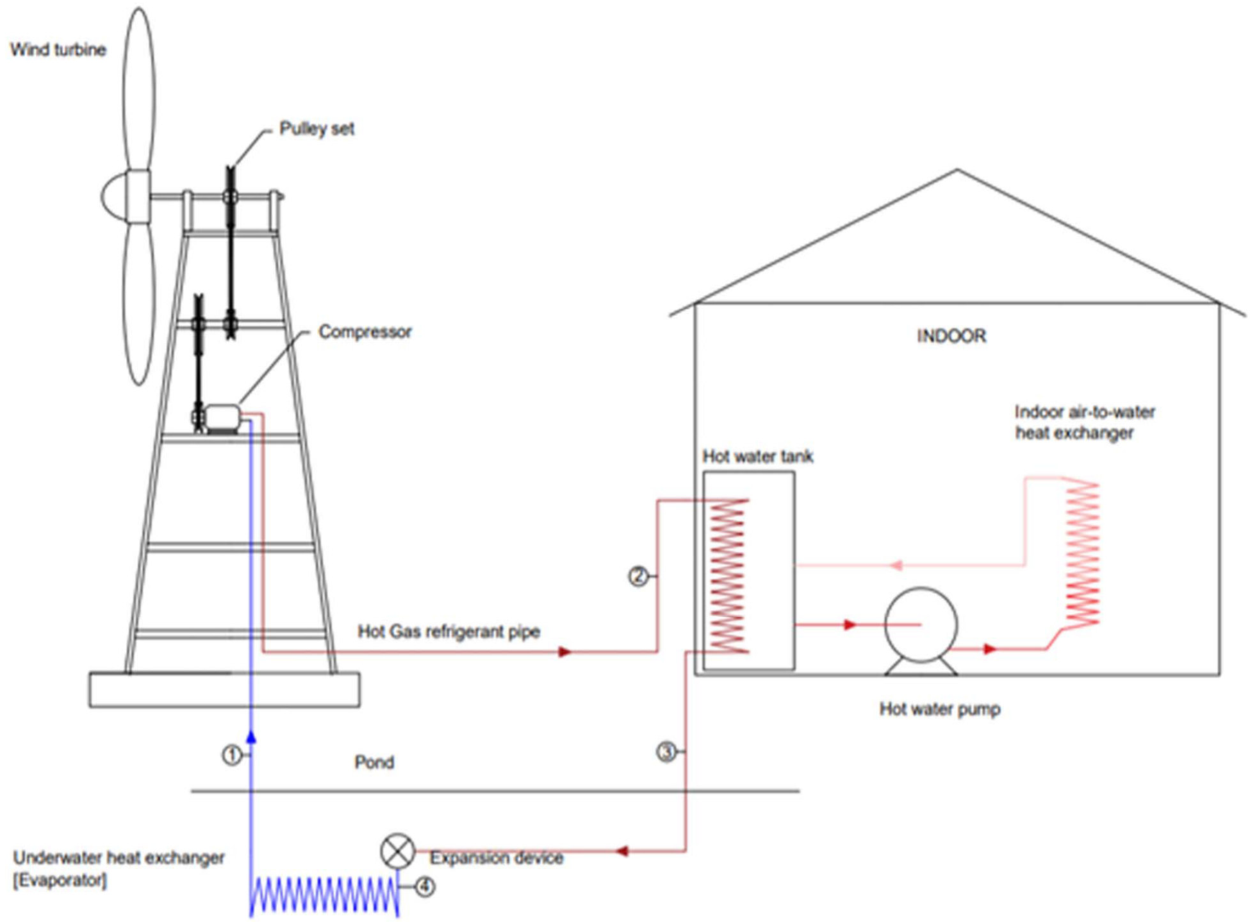

Figure 2.2: A direct wind-powered thermal energy system for cold regions [43].

2.2 Thermal Energy Storage

Primary energy consumption continues to rise in geometric proportion. This, coupled with the negative effects of fossil resources on the environment, the shift towards sustainable sources of energy has also seen a significant increase. To improve the natural equilibrium and meet the needs of increasing population demand, self-sustaining energy sources such as solar radiation, ocean currents, wind, and biogas continue to play a significant role. However, the means to store these intermittent types of energy have become urgently needed. This necessitates the development of effective and efficient energy storage methods.

Thermal energy storage encompasses several technologies, and its operation is based on stocking of the energy in a storage medium by cooling or heating for domestic and industrial heating needs and also for power generation. Thermal energy storage is mostly used in buildings and industrial processes. Heat can be stored at between the temperatures of -40℃ to 400 ℃ [44]. The benefits of incorporating TES into the energy mix include increased overall efficiency, increased reliability and improved economic efficiency, lower cost of investment and running, and lower environmental pollution, which means lower emissions of carbon dioxide [45].

There are three different TES techniques. These are thermo-chemical, sensitive heat and latent storage [44]. They have different principles of work. Sensitive heat storage systems are currently available in commercial quantities and are in a much more advanced development stage, while latent heat and thermochemical storage are still in their infancy, undergoing demonstration. Table 2.1 shows a comparison of the different TES, while Figure 2.3 shows the different materials used to store the energy for the different technologies.

Table 2.1: Features and comparison of TES [46]

Parameters	Sensible heat	Latent heat	Thermo-chemical
Energy density	Small:$\sim 50\ \mathrm{kWhm^{-3}}$	Medium:$\sim 100\ \mathrm{kWhm^{-3}}$	High:$\sim 500\ kWhm^{-3}$
Maturity	Industrial-scale	Testing stage	Laboratory scale

Transport	Short distance	Short distance	Theoretically unlimited distance
Technology	Simple	Medium	Complex
Period of storage	Due to thermal losses, there is a limit.	Due to thermal losses, there is a limit.	Unlimited
Temperature	Charging step storage temperature	Charging step storage temperature	Ambient temperature

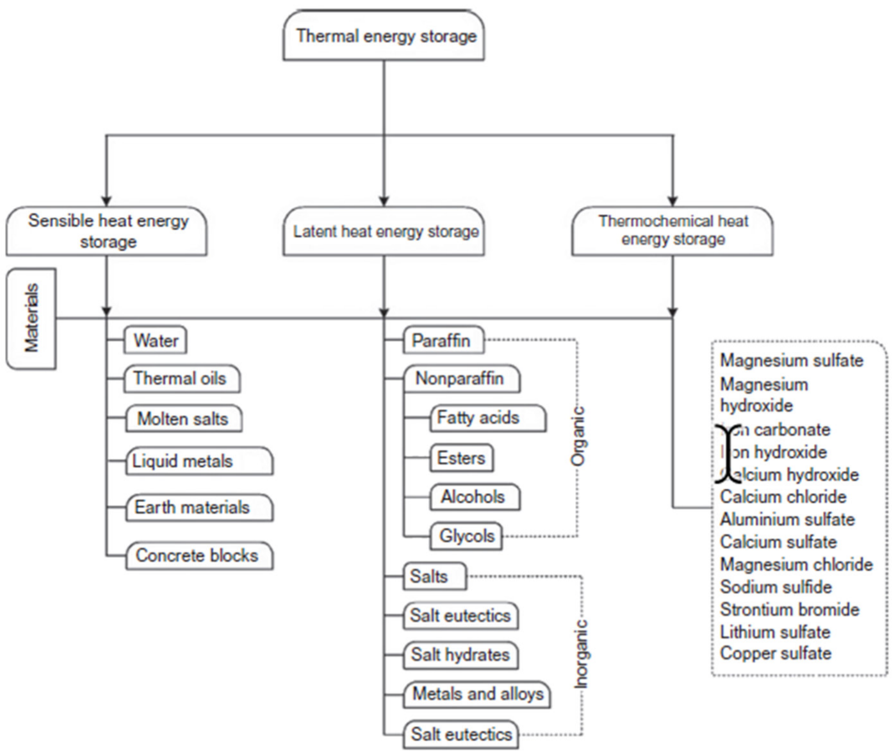

Figure 2.3: Different materials used for energy storage [52].

2.2.1 Sensible heat storage

This type of heat energy storage utilizes the variation in temperature of the process or material during storing and discharging. This method of heat storage is very advantageous as the process is completely reversible. This makes it easily applicable to all forms of heating, such as domestic systems, industrial processes and district heating. The quantity of the stored energy depends on the heat capacity of the material, and the thermal insulation method employed is very critical when using this technology. Sensible heat storage systems use large storage space because of their low energy density (thus, they have about 3-5 times lower energy density than PCM and TCS technologies) [47]. Equation 1 represents the sensible heat, Q stored by a material of mass, m:

$$Q = m \int_{T_1}^{T_2} C_p . dT \quad (1)$$

Where T_1 and T_2 are the charging and discharging temperatures of the process, respectively and C_p is the specific heating capacity of a material or medium.

2.2.2 Latent heat storage.

If higher energy densities and more steady temperatures to be released are needed, sensible heating is no longer an economical way of storage, and hence thermal energy storage using phase change materials (PCMs). PCMs provide a huge heat output over a narrow temperature range and can act as a near-isothermal heat storage tank. PCMs, change the phase from solid to liquid when the temperature rises. As it is endothermic, heat is absorbed. PCMs also can reversibly change from liquid to solid when the temperature is lowered, releasing heat. PCMs are generally poor conductors of energy which makes them take so long to charge and discharge. Equation 2 represents the heat energy stored by PCM [48]:

$$Q = m\left[\int_{T_1}^{T_m} C_{ps} . dT + \Delta h + \int_{T_m}^{T_2} C_{pl} . dT\right] \quad (2)$$

Where T_m, C_{ps}, Δh, and C_{pl} are the melting point temperature, the specific heating capacity of a material or medium in the solid phase, enthalpy, and specific heat capacity in the liquid phase, respectively. T_1 and T_2 are the same as defined in Equation 1.

2.2.3Thermo-chemical storage

This type of energy storage is based on energy storage using chemical reactions. The reactions that characterize this system are reversible. TES offers higher storage capacities than both PCM and SHS. Though TCS is largely still in the development stage, it has the potential to lead thermal energy storage technologies when it is fully implemented. To demonstrate this technology, Schmidt and Linder [49] proposed a novel long-term thermal energy storage system using chemical, thermal decomposition of Calcium hydroxide. With this concept, the $Ca(OH)_2$ decomposes into CaO and water vapour based on the reversible reaction:

$Ca(OH)2 + energy \leftrightharpoons CaO + H_2O$

If excess energy is available, the hydroxide will be decomposed into the oxide (charging phase). Condensation can easily separate the released water vapour from the solid. The energy in the state of the chemical potential of calcium oxide can be conserved for an infinite time. Once the energy is needed, liquid water or water vapour comes back into reaction with the calcium oxide, and the exothermic response that takes place releases the energy for use (discharging phase). They developed a conceptualized system for decentralized long-term energy storage suitable for the built environment. The conceptualized is shown in Figure 2.4:

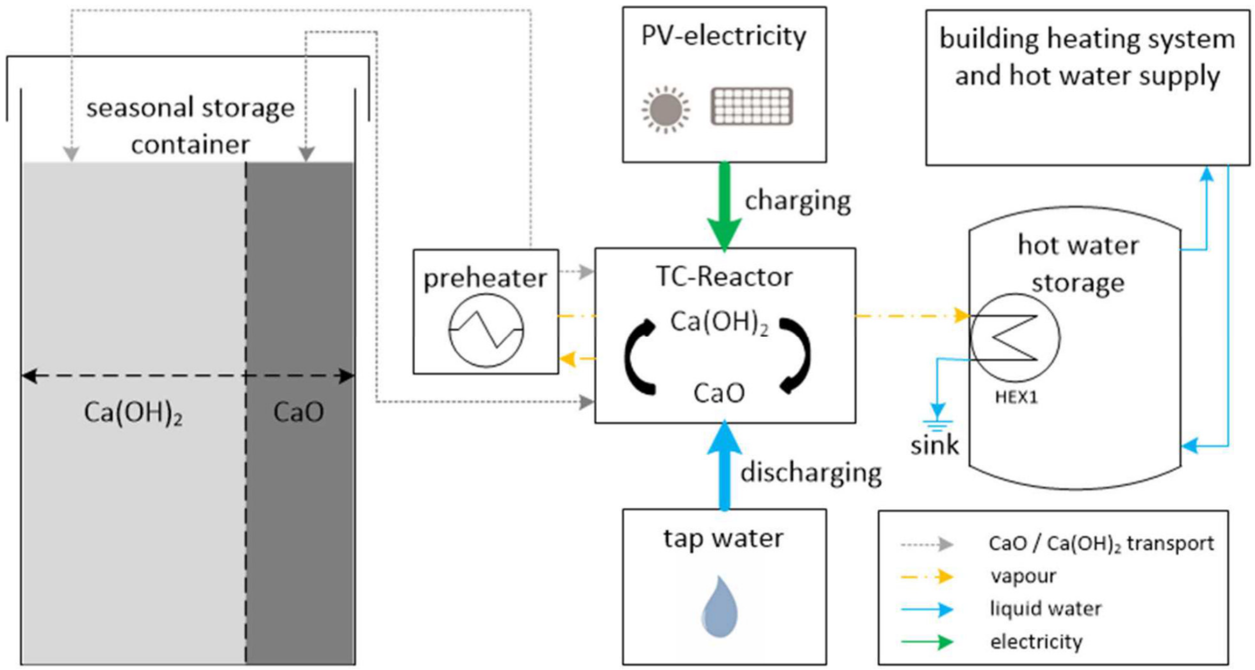

Figure 2.4: Conceptual process design for a decentralized thermochemical storage system using calcium hydroxide [49].

2.3 Wind-Powered Thermal Energy Storage

Wind thermal energy storage has not yet been widely researched, and its economic and technical viability is unclear. This study focuses on them, and consequently, this chapter gives an insight into the thermal systems that are powered by wind to illustrate the principle and the different methodologies of heat generation. Five key WTES ideas will then be discussed and evaluated, together with their advantages and disadvantages and their different operating principles. A review of wind and heat production will then be presented in the last section.

The WTES is focused on wind transformation into heat that can be stored economically and almost irrespective of the location. This technology offers the prospect to sporadically convert wind energy into a dispersible renewable energy source that is available on request. As a result, WTES can lead to a reliable capacity that can meet strategic energy goals relating to supply security and environmental livability.

Various WTES options can be obtained by combining separate technologies currently available, but these have not currently been examined together in a complete system: high-temperature heat generators, heat mechanisms and high-temperature heat storage. However, they have never been discussed together as a whole system. This will link the systemic advantages of steam power plants to the use of wind energy, thereby reducing the need for investment in dispatchable electricity capacity.

To develop an economic wind conversion system, currently available heat generators such as heat pumps, electric boilers, and retarders can be used. Further, the correct technical selection takes into consideration all the main elements, and some energy sources and energy usage are other important factors for ideal heat storage. Lastly, traditional heat engines such as those used at traditional steam plants could be used for the transformation from heat to electricity.

Heat generation using wind energy can be done indirectly or directly. Direct heat production involves the direct conversion of wind power to thermal energy without the need for an electric generator. The heat generation occurs at the wind energy converter's nacelle. In contrast to direct production, an electrical generator needs to convert the wind to electricity and then use power to produce heat indirectly.

2.3.1 Direct wind thermal systems

This concept of heat generation involves converting the rotational motion of the wind into heat. Ancillary heat creation has the benefit that the heat generator must not be located near a primary heat storage facility that transports electric power. High-temperature generation and pumped heat energy storage (PHES) can be obtained by electric heat pumps to achieve the most efficient power-heat-electric conversion. This type of heat generation mostly uses hydrodynamic/ induction retarders or mechanically powered heat pumps [44].

2.3.2 Electrically driven heat pump configuration.

This configuration replaces the boiler with a heat pump which is also powered by electricity.

The main characteristic of this concept is the use of HP. Like electric boilers, electrical-driven heat pumps can almost instantaneously be adjusted between standstill and full load, a beneficial feature for WTES. Heat pumps use 2 to 5 times less electricity to produce the same amount of heat as the electric boiler [50]. Though, in general, the literature available relating to wind thermal energy storage is very scanty, few authors have examined this technique of wind energy storage. Li et al., for example, investigated the adaptive output of a separate wind turbine connected to a heat pump and an electric storage system for heat supply in single-family homes [51]. They demonstrated that, due to the intermittent existence of wind energy, the heat pump's electricity demands cannot always be met. This emphasizes the value of a storage unit as well as a cautious wind turbine configuration to minimize supply and demand differences. The proposed system is shown in Figure 2.5.

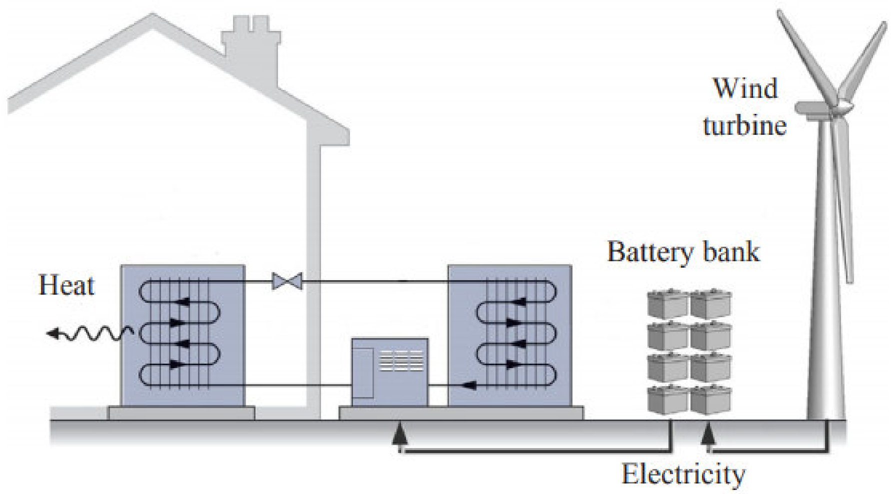

Figure 2.5: Wind power-driven standalone heat pump [51].

Other power-to-heat studies in the Scandinavian area, which concern district heating supply, primarily concentrate on the use of heat pumps. Münster et al., for example, investigated the role of district heating in Denmark's renewable energy transition. They discovered that expanding district heating is financially feasible because it provides cost-effective ways to incorporate more wind power using massive heat pumps with thermal energy storage to utilize the excess power from wind energy generators when wind abounds [52]. Additionally, Hedegaard and Münster [53] investigated the economic viability of integrating a large amount of wind-powered heat pumps into the building sector. They demonstrated that heat pumps could help fund wind energy investments while also lowering device costs.

Rieck et al. [54] carried a feasibility study of a heat pump powered by a wind energy generator and solar photovoltaic system for single homes in Germany. A heat pump coupled to a storage tank was used. They used simulation to depict strategies for integrating renewable energies in various parts of Germany while reducing the need to shut down wind turbines or feed electricity into the grid. A single house's energy consumption is modelled. As an energy source, various wind turbines and photovoltaic systems are implemented. These systems' profitability is measured and compared to traditional gas or oil systems. Under the current circumstances, small wind turbines are a realistic option for meeting electricity demand, according to the results obtained. PV plants, on the other hand, are currently unsuitable for meeting heat demand because their basic costs are higher than those of conventional systems.

Furthermore, Chemekov and Kharchenko [55] discussed water-type heat pumps in residential houses powered by an electrically driven wind energy storage system for resort zones in the black sea coast in Russia. The studied system was found to be more cost-effective and more efficient than a geothermal system. The schematic of the proposed system is shown in Figure 2.6.

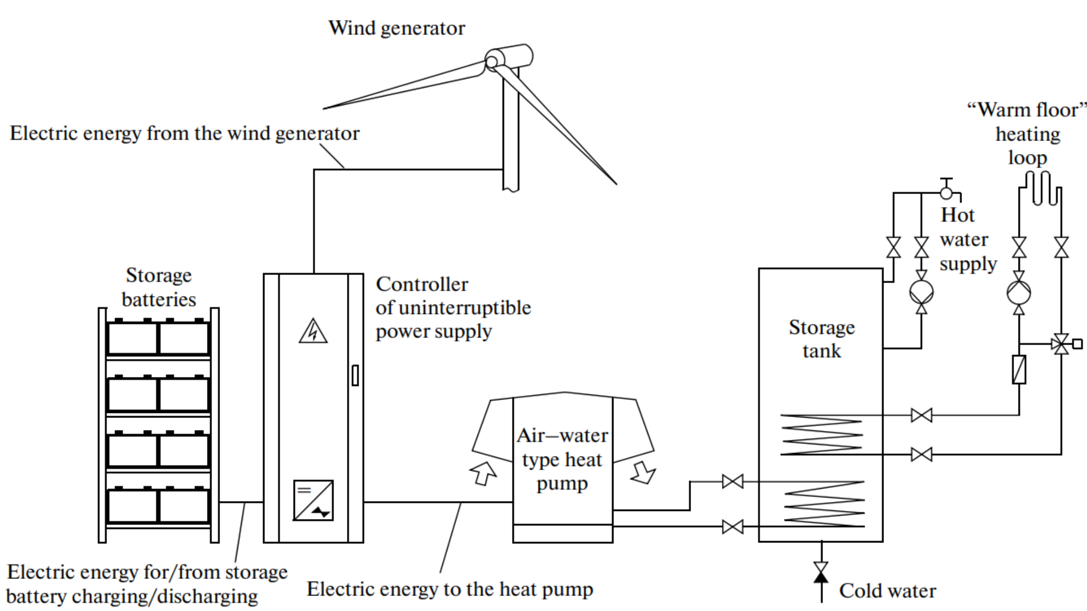

Figure 2.6: Schematic diagram of the investigated system [55]

2.3.3 Mechanical driven heat pump configuration

This is direct wind thermal energy storage technique that is based on the conversion of wind energy into thermal energy without employing an electric generator. Here, the wind energy is at the turbine blades is converted directly into rotational energy at the energy converter's shaft and used to power the heat pump directly through the compressor shaft. This system, however, requires a gearbox since the rotational energy of a wind energy converter is generally lower.

To demonstrate this technique, Klueter and Liljedahl [56] carried feasibility studies of a heat pump for a dairy that is mechanically driven. The system was a wind turbine mechanically coupled to drive a refrigeration compressor used at a dairy farm. The system refrigerates an ice-builder big enough to supply ice to cool the milk to last at least three days. The refrigeration compressor is mounted to the wind turbine through a mechanical drive system. The refrigerant from the compressor is cooled by cold water as it moves through a water-cooled condenser. The hot water tank receives the warmed water. The condenser's refrigerant is routed to a receiver, then through an appropriate expansion valve to ice-builder piping, and finally back to the compressor. The ice water from the ice maker circulates into a counter-flow, parallel-plate heat exchanger, chilling warm milk from the milking machine. The cooled milk is then stored in an enclosed bulk milk tank. The conceptual plan of the proposed system is demonstrated in Figure 2.7.

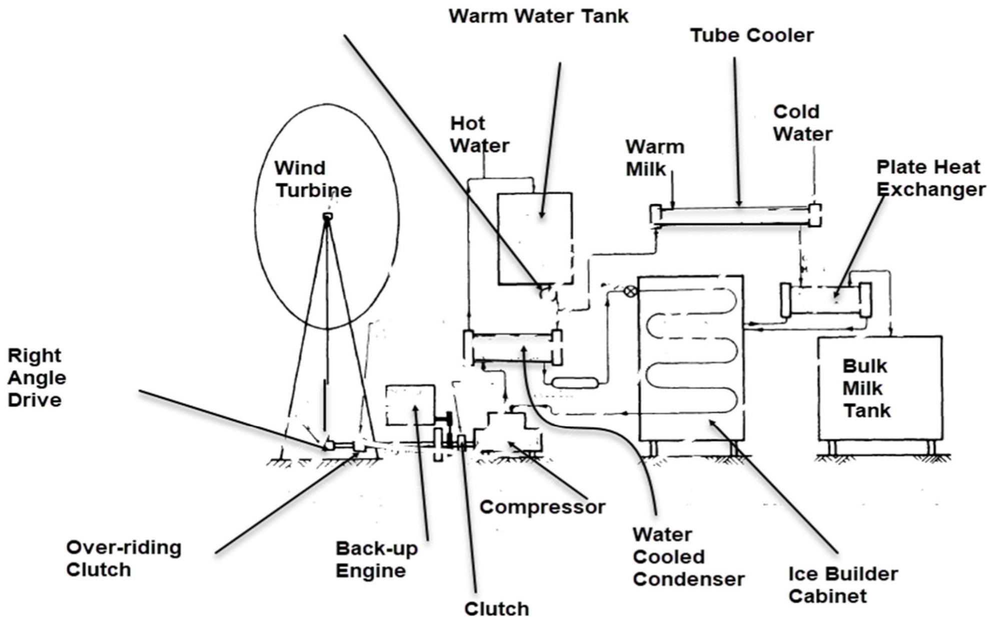

Figure 2.7: General system schematic of a wind-driven heat pump for a diary [56]

Jwo et al. [57] developed a wind thermal energy system directly connected to a forced heat pump and analyzed its efficiency. The system uses a four-way valve to control the operation of the heat pump mechanism, which switches between a chiller and a heat pump. Based on theoretical and experimental studies, they reported that the conventional system, in which the heat pump is directly powered by forced wind, has a theoretical efficiency of 42.19 percent. The standard value for cooling water generating efficiency in the indoor temperature of 25 °C and outside temperature of 35 °C was 54.38 percent, and for hot water generating efficiency in the indoor temperature and the outside temperature both of 10 °C was 52.25 percent. The proposed system was able to improve by a factor of 10 percent for cooling as well as heating as shown in Figure 2.8.

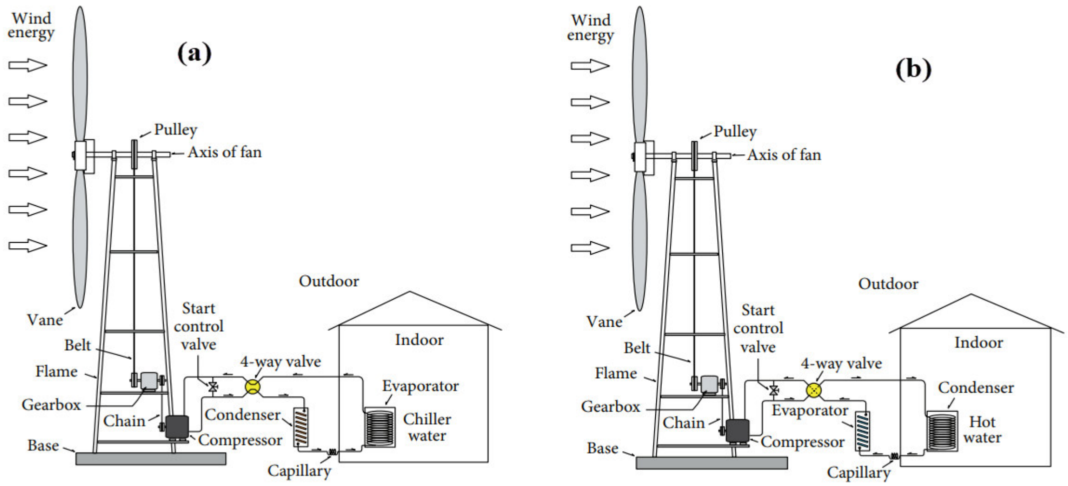

Figure 2.8: Wind directly forced heat pump:(a) chiller for the summer, (b) heat pump for the winter [57].

In a similar study, Ting et al. [58] proposed the development of a chiller powered by wind energy. This technique of direct conversion of wind energy to power a chiller is meant to circumvent the losses due to the traditional mechanical to electrical wind energy conversion process. The proposed system was said to increase the efficiency in comparison with the indirect refrigerating system from a wind energy generator. They reported a 21.28 % chiller efficiency.

2.3.4 Retarder

All the features listed in the concept of the mechanically driven heat pump configuration regarding the wind energy converter (WEC) assumptions are identical in this concept. This method uses the rotational energy of the WEC, and it is converted directly into thermal energy, in this case by using a retarder that is directly attached to the shaft of the WEC. To compare different nominal powers, all the necessary retarders are assumed to be connected in sequence.

A gearbox is needed to run the unit around the retarder's optimum working condition, as it provides nominal power between 700 and 1,500 rpm. An example of such a system that employs a braking system to convert the wind into thermal energy is reported in [59]. The heat produced in this process is carried by a heat transfer fluid to a storage tank which could be used for electricity generation later or for space heating.

In addition, Chakirov and Vagapov [60] developed a system that operates as a Joule machine to convert wind energy directly into heat. This heat generator is a mixer installed into a tank full of heat transfer fluid. The kinetic energy from the wind is used to rotate the shaft of the mixer, and fluid is mixed by an impeller. The frictional energy generated in the process of mixing is converted into heat energy which is further transferred to a heating system.

2.4 Gaps in the Literature

The literature shows that among the different sources of renewable energy for heating, the wind is generally considered the one with the most potential. Despite the enormous potential of wind for space heating, the literature review paints a somewhat oblique picture of its ability to supplant conventional furnaces, especially in cold regions like Canada, due to the limited research. Only a few researchers have investigated the potential of wind thermal energy systems. However, these have centered their studies on indirect wind thermal systems which employ an intermediary (electric generator), which contributes to more losses. Direct wind thermal systems have barely been studied despite their superiority over indirect systems, especially their integration with a heat pump system. This study will investigate the application of direct thermal systems.

Chapter 3: Methodology

In this chapter, the proposed system and its operations are described. Also, the cost of energy, GHG and thermodynamic analyses are performed, including mass balance, entropy energy and exergy.

3.1 Environmental factors

It is important to mention the temperature and wind speed conditions of the surrounding environment of the chosen location for this study, which is Resolute, Nunavut, Canada. Figures 3.1 and 3.2 presents the average air and sea temperatures, as well as the average wind speeds of this location. One interesting thing to notice here is that the sea temperature remains the same at about 10°C, year-round. This is a good thing because it provides thermal stability for the operation of the integrated system and a reliable source of heat from the surrounding environment, especially during the winter months. This is in contrast with the air temperature that fluctuates dramatically across the months of the year from as low as -33 in February to as high as 4°C in July. Such fluctuations require the system to operate in a wide range of operating conditions for the same location. The wind speed ranges from a low of 5.6 m/s to a high 6.7 m/s year-round. This is a good wind speed for a wind turbine, since cut-in speed of most wind turbines including the selected wind turbines is 3m/s [61].

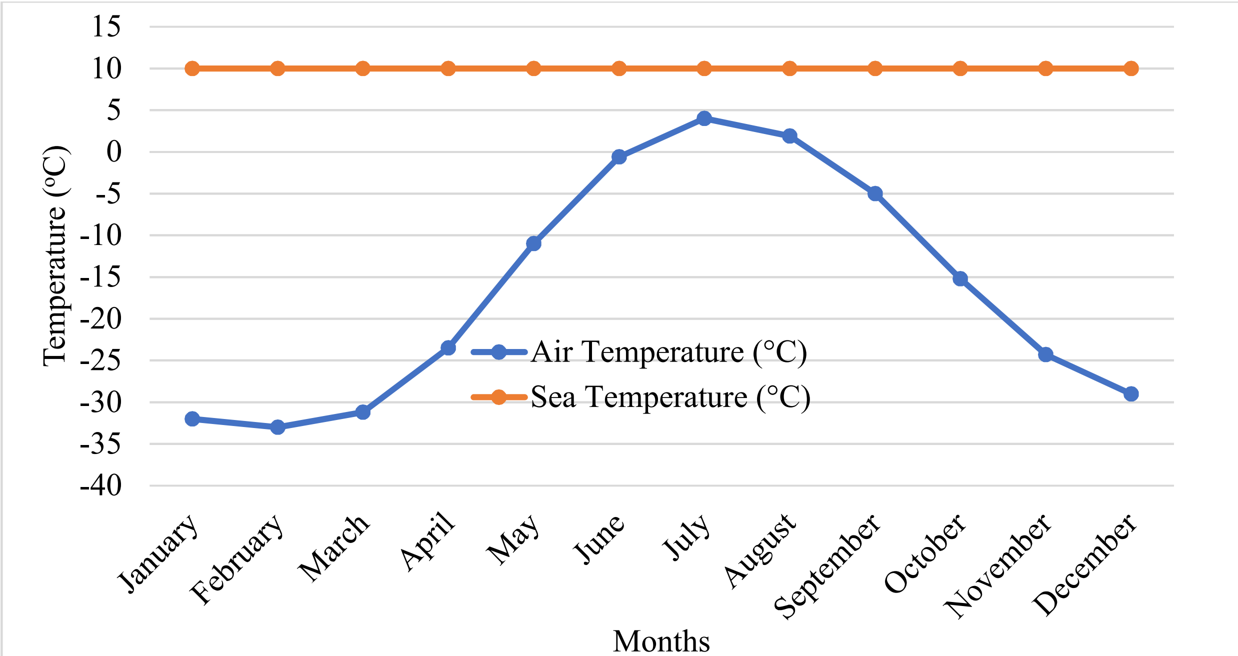

Figure 3.1: Average monthly sea and air temperature for Resolute, Nunavut, Canada [62]

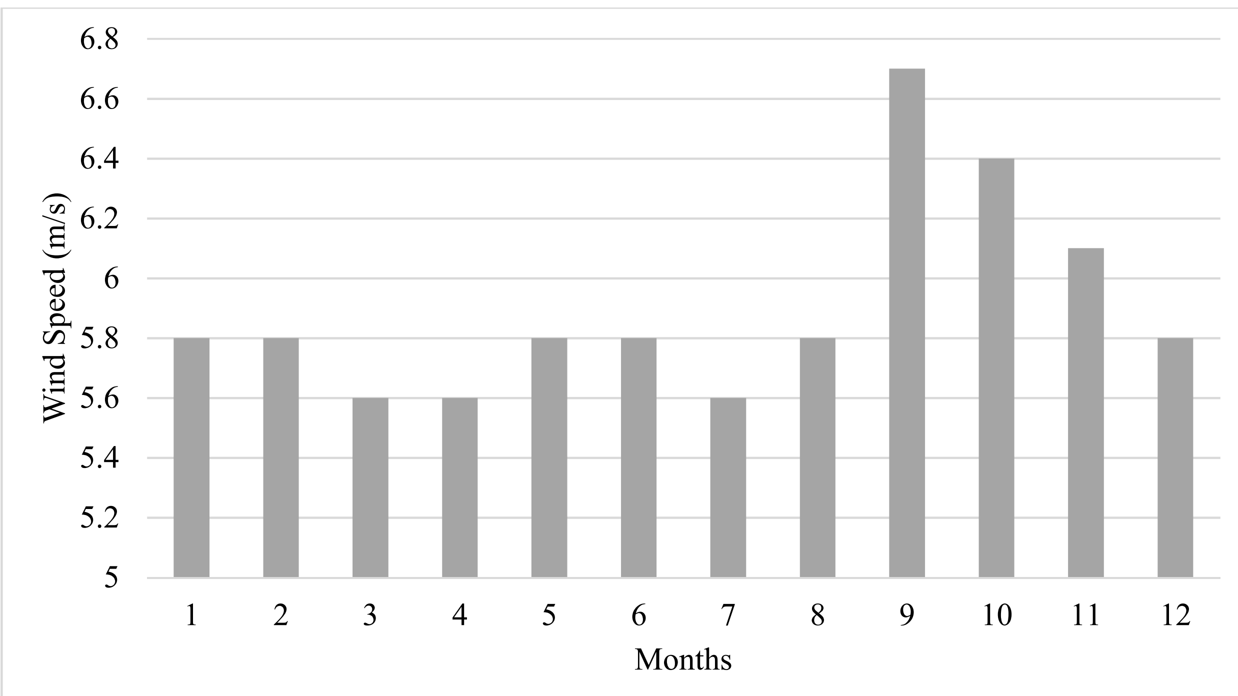

Figure 3.2: Average monthly wind speed for Resolute, Nunavut, Canada [62]

3.2 Energy Profile of the Target Community

The proposed system must meet the requirements of a small residential community in Resolute Bay, Nunavut, as indicated earlier. The community consists of a six-story complex with ten apartments on each floor, for a total of 60 units. It is assumed that each unit houses a single person with frequently owned domestic items that generate sensible heat, such as ovens, microwaves, computers, televisions, and refrigerators. The floor space of each unit is 3 m × 3.7 m × 15 m. The room has a single double-glazed slider window with an aluminum thermal break. Table 3.1 shows the internal heat gain of materials involved in the study. The dimensions of each window are considered to be 1m × 1 m, resulting in a total area of 1 m^2 per window. On the building's façade, there are two doors, which are double vestibule windward doors with dimensions of 2.06 × 1.52 meters each.

It was found that the needed volumetric flow rate of desalinated water into the community for its 60 persons was 18 m^3/day. This number was obtained using the assumption that one person in a household setting requires 0.303 m^3/day water from a desalination system to be comfortable[63]. The goal was to achieve a minimal power requirement of 280 kW and a heating load of 460 kW to meet the unique system specifications base on a calculation using the air-conditioning, heating, & refrigeration institute (AHRI) standard 210/240 [64].

Table 3.1: Heat rate of units [65]

Medium	Heat gain (W)
Sedentary person (latent)	45
Sedentary person (sensible)	72
Television	80

Medium appliances for light per m_2	35
Oven	710
Computer	65
Microwave	690
Refrigerator	690

3.3 System Design Consideration

Different forms of energy were considered for the proposed system. Since the location chosen is Resolute Bay, Nunavut, which is along the Northwest Pathway, only the available energy sources at that location can be explored. There is no commercial crude oil production in Nunavut, nor are there any refineries. Further, Nunavut has no production facilities for natural gas or natural gas liquids at the moment. Despite the fact that Nunavut's natural gas reserves are estimated to be 181.4 trillion cubic feet [66], almost all of Nunavut's electricity comes from diesel fuel imported during the summer and stored for the rest of the year. Due to low levels of sunlight, solar is not suitable for the community because its average solar radiation-horizontal ranges as low as 0 kWh/m^2/d from November to January which means there is no sunlight. Its yearly average solar radiation-horizontal is 2.37 kWh/m^2/d [67] . This means a solar system cannot be the main source of energy for the selected location. A key consideration is that the system must be designed to be as green as possible to be in line with the Pan-Canadian Framework, which is to reduce GHG by 40% by 2030.

3.3 Proposed System Description

The multi-output system shown in Figure 3.3 depicts the proposed system. Four products will be produced by the system, including electricity directly from the wind turbines, hydrogen for chemical energy storage, space heating, and freshwater. The system is powered by a wind turbine which generates electricity for the community and the two compressors. The first compressor is an open compressor that uses the kinetic energy of the wind from the wind turbine for the heat pump to provide space heating to the community, and the second compressor supplies heat to the multi-effect desalination unit to produce clean drinking water to the community. The excess electricity is stored in a chemical form using a Proton-Exchange Membrane (PEM) electrolyzer to make hydrogen through an electrochemical reaction referred to as water electrolysis. This is the energy storage subsystem of the proposed system.

The heat pump system starts operating when a refrigerant entering Compressor 1 (Comp 1) at State 1 is compressed to State 2. The superheated gas refrigerant delivered by the compressor flows into the condenser (hot water heat exchanger), releasing the heat to the water in the tank while condensing to State 3. Flowing through the expansion device, the liquid refrigerant expands to State 4. The refrigerant liquid-gas mixture continues to flow inside the evaporator (i.e., the sea) where it gains heat from the surrounding environment resulting in the refrigerant evaporation back to State 1. The gas refrigerant then enters the suction port of the compressor and thus closes the cycle. It should be reiterated that the main purpose of sea water is to maintain the ambient temperature at 10 ^{o}C irrespective of how low the outside air temperature falls. The thermal energy is stored in the hot water tank, where it is used for space heating when required.

The multi-effect desalination unit is described next. The seawater is pumped to high pressure and ambient temperature, which is represented here as State 21. The refrigerant entering compressor two at State 19 is compressed to State 16. The superheated gas refrigerant delivered by the compressor flows into the condenser (desalination unit), releasing the heat energy to the seawater while condensing to State 17. Flowing through the expansion device, the liquid refrigerant expands to State 18. The refrigerant liquid-gas mixture continues to flow inside the evaporator (i.e., the sea) where it gains heat from the surrounding environment resulting in the refrigerant evaporation back to State 19. Compressor 2 is powered by the wind turbine, and it supplies heat to the desalination unit. The seawater stream goes through three stages of desalination, and two products are made: water and brine. The brine is returned back to the sea. The freshwater is split into two streams; one for domestic use and the other is electrochemically split into hydrogen and oxygen at their respective electrodes in the PEM water electrolysis, with hydrogen at the cathode and oxygen at the anode. PEM water electrolysis occurs when water is pumped to the anode and split into oxygen , protons , and electrons. These protons travel to the cathode side via the proton conducting membrane. The electrons leave the anode via the external power circuit, which provides the reaction with a driving force. Protons and electrons recombine at the cathode to produce hydrogen. The hydrogen is stored to be used as backup power by PEM fuel cells. The electrolyte of PEM fuel cells, also known as proton exchange membrane fuel cells, is a proton-conducting polymer membrane. These cells can function at low temperatures and change their output fast to meet changing power demands. This will be utilized to generate electricity and also to power vehicles if there is no wind.

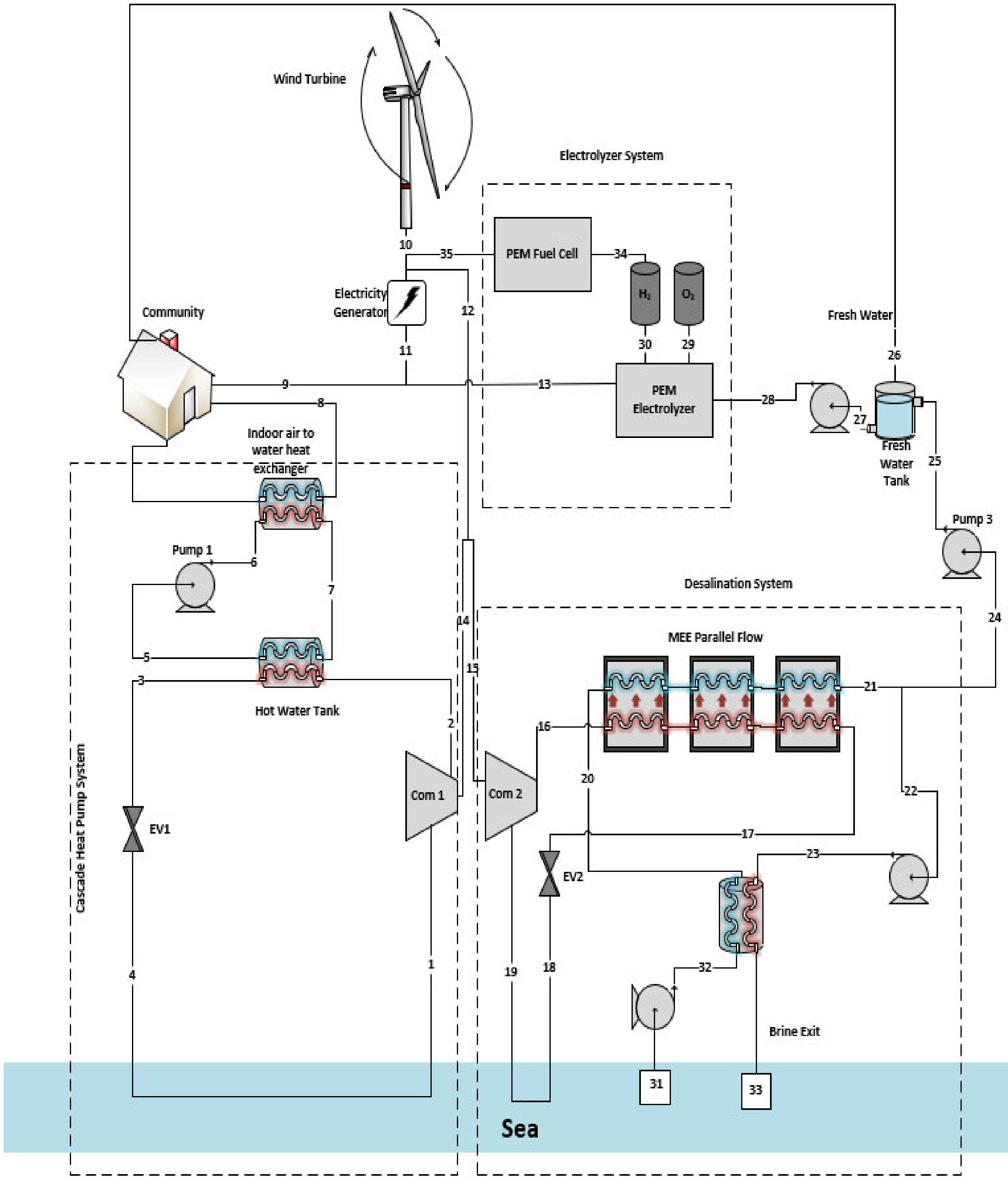

Figure 3.3: System diagram of the multi-output wind turbine-based system.

3.4 Thermodynamic Analyses

The thermodynamic analysis based on the first and second laws of thermodynamics is presented in this subsection. Before going to the mathematical forms of this analysis, it is important to mention the general assumptions employed to study this multi-output system. The first main assumption is that all the components of this system are working under steady-state conditions. The second assumption is that all the components have negligible heat losses except the PEM electrolyzer. Since the system analysis is focused on thermal and work processes, it is reasonable to assume that the changes in kinetic and potential energies across the system components are small compared to heat transfers, work, and enthalpy changes [68]. Another assumption to be mentioned is the pressure losses across heat exchangers, and connecting pipes are too small to be accounted for in the analysis. The only exception is the desalination unit heat exchangers. Regarding the compressors and pumps, they are operating under adiabatic conditions.

Next, the conservation laws of mass, energy are outlined for each component of the system, along with their specific assumptions. In addition, the second law of thermodynamics is presented for each component in terms of entropy to find the entropy generation rate for the major components of the system as well as the exergy destruction rates.

3.4.1 Compressor 1

To start with, compressor 1 has a mass balance equation that can be written as:

$$\dot{m}_1 = \dot{m}_2 \tag{3.1}$$

and the energy balance equation is:

$$\dot{m}_1 \, h_1 + \dot{W}_{14} = \dot{m}_2 \, h_2 \tag{3.2}$$

where $\dot{W}_{14}$ is the part of the electric power input from the wind turbines. This compressor is assumed to operate under a constant isentropic efficiency of 85%. The isentropic efficiency of this compressor is mathematically expressed as:

$$\eta_{C1} = \frac{h_{2,s} - h_1}{h_2 - h_1} \tag{3.3}$$

where $h_{2,s}$ is the enthalpy if the compressor was operating under ideal conditions. Another assumption for this compressor is that it is adiabatic. Next, the entropy balance equation is:

$$\dot{m}_1 \, s_1 + \dot{S}_{gen,C1} = \dot{m}_2 \, s_2 \tag{3.4}$$

From knowing the entropy generation rate of compressor 1 and the reference temperature, it is possible to find the exergy destruction rate of this component by using the following expression:

$$\dot{E}x_{D,C1} = T_o \, \dot{S}_{gen,C1} \tag{3.5}$$

3.4.2 Hot water heat exchanger

After compressor 1, the fluid goes to the hot water heat exchanger, and the thermodynamic balance equations are discussed here. There are two mass balance equations, one for the hot fluid and the other for the cold fluid. For the hot fluid, it is:

$$\dot{m}_2 = \dot{m}_3 \tag{3.6}$$

and for the cold fluid:

$$\dot{m}_7 = \dot{m}_5 \tag{3.7}$$

The energy balance equation is expressed as follows when the heat exchanger heat losses are ignored:

$$\dot{m}_2 \, h_2 + \dot{m}_7 \, h_7 = \dot{m}_3 \, h_3 + \dot{m}_5 \, h_5 \tag{3.8}$$

In addition, the entropy balance equation is:

$$\dot{m}_2\, s_2 + \dot{m}_7\, s_7 + \dot{S}_{gen,HWHE} \; = \; \dot{m}_3\, s_3 + \dot{m}_5\, s_5 \tag{3.9}$$

where the subscript HWHE stands for hot water heat exchanger. To calculate the exergy destruction rate, we use a similar expression:

$$\dot{Ex}_{D,HWHE} \; = \; T_o\, \dot{S}_{gen,HWHE} \tag{3.10}$$

3.4.3 Expansion valves

There are two expansion valves in the studied system, and both are operated under isenthalpic conditions, meaning the enthalpy is constant for the flowing fluid. For EV1, the mass balance and enthalpy balance are expressed respectively as:

$$\dot{m}_3 = \dot{m}_4 \tag{3.11}$$

and

$$h_3 = h_4 \tag{3.12}$$

For EV2, they are

$$\dot{m}_{17} = \dot{m}_{18} \tag{3.13}$$

and

$$h_{17} = h_{18} \tag{3.14}$$

3.4.4 Pump 1

The next component to be modelled is pump 1. The mass balance equation is:

$$\dot{m}_5 = \dot{m}_6 \tag{3.15}$$

and the energy balance is:

$$\dot{m}_5\, h_5 + \dot{W}_{Pump1} = \dot{m}_6\, h_6 \tag{3.16}$$

The power input to this pump is calculated using the constant specific volume model which is described as:

$$\dot{W}_{Pump1} = \frac{\dot{m}_5 v_5}{\eta_{Pump}} (P_6 - P_5) \tag{3.17}$$

where v_5 is the specific volume in units of m^3 kg^{-1}, P is the pressure in kPa, and η_{Pump} is the pump efficiency which is assumed to be 90%. The second law analysis can also be conducted for this pump. The entropy balance is:

$$\dot{m}_5 \, s_5 \; + \dot{S}_{gen,Pump1} \; = \; \dot{m}_6 \, s_6 \tag{3.18}$$

and the exergy destruction rate is:

$$\dot{Ex}_{D,Pump1} \; = \; T_o \, \dot{S}_{gen,Pump1} \tag{3.19}$$

3.4.5 Air to water heat exchanger

The last component in the heat pump is the indoor air to water heat exchanger, abbreviated as AWHE, and its thermodynamic modelling is presented next. Similar to the previous heat exchanger, there are two mass balance equations. For the hot fluid, which is water, it is:

$$\dot{m}_6 = \dot{m}_7 \tag{3.20}$$

and for the cold fluid, which is the indoor air:

$$\dot{m}_o = \dot{m}_8 \tag{3.21}$$

After assuming no heat losses in this heat exchanger, the energy balance equation becomes:

$$\dot{m}_o \, h_o + \; \dot{m}_6 \, h_6 = \dot{m}_7 \, h_7 + \dot{m}_8 \, h_8 \tag{3.22}$$

The entropy balance equation is:

$$\dot{m}_o \, s_o + \; \dot{m}_6 \, s_6 + \dot{S}_{gen,AWHE} \; = \; \dot{m}_7 \, s_7 + \dot{m}_8 \, s_8 \tag{3.23}$$

The exergy destruction rate simple is:

$$\dot{E}x_{D,AWHE} = T_o \, \dot{S}_{gen,AWHE} \tag{3.24}$$

3.4.6 Compressor 2

The next subsystem to be discussed in a similar way is the desalination subsystem. The first component in this subsystem is compressor 2. Sea water is sucked from the sea at a constant and steady mass flow rate. The mass balance equation is:

$$\dot{m}_{19} = \dot{m}_{16} \tag{3.25}$$

and the energy balance equation, which consumes electric power from the wind turbines to heat the sea water, is:

$$\dot{m}_{19} \, h_{19} + \dot{W}_{15} = \dot{m}_{16} \, h_{16} \tag{3.26}$$

This compressor is assumed to operate under a constant isentropic efficiency of 85%. The isentropic efficiency of this compressor is:

$$\eta_{C2} = \frac{h_{16,s} - h_{19}}{h_{16} - h_{19}} \tag{3.27}$$

This compressor has negligible heat losses. After that, the entropy balance equation is:

$$\dot{m}_{19} \, s_{19} + \dot{S}_{gen,C2} = \dot{m}_{16} \, s_{16} \tag{3.28}$$

Like the previous components:

$$\dot{E}x_{D,C2} = T_o \, \dot{S}_{gen,C2} \tag{3.29}$$

3.4.7 Pump 2

The next component to be modelled is pump 2. The mass balance equation is:

$$\dot{m}_{20} = \dot{m}_{21} \tag{3.30}$$

and the energy balance is:

$$\dot{m}_{20}\, h_{20} + \dot{W}_{Pump2} = \dot{m}_{21}\, h_{21} \tag{3.31}$$

The power input to this pump is found using the following expression, where the specific volume of the fluid is assumed constant:

$$\dot{W}_{Pump2} = \frac{\dot{m}_{20} v_{20}}{\eta_{Pump}} (P_{21} - P_{20}) \tag{3.32}$$

Similar to pump 1, η_{Pump} is has a value of 90%. The entropy balance is:

$$\dot{m}_{20}\, s_{20} + \dot{S}_{gen,Pump2} = \dot{m}_{21}\, s_{21} \tag{3.33}$$

and the exergy destruction rate is:

$$\dot{E}x_{D,Pump2} = T_o\, \dot{S}_{gen,Pump2} \tag{3.34}$$

3.4.8 Desalination unit

The thermodynamic model for the desalination unit is now discussed. The mass balance equation of the hot fluid which supplies the heat is:

$$\dot{m}_{16} = \dot{m}_{17} \tag{3.35}$$

and the water streams that undergo desalination have a mass balance equation expressed as:

$$\dot{m}_{21} = \dot{m}_{23} + \dot{m}_{24} \tag{3.36}$$

The combined energy balance of the desalination unit is:

$$\dot{m}_{16}\, h_{16} + \dot{m}_{21}\, h_{21} = \dot{m}_{17}\, h_{17} + \dot{m}_{23}\, h_{23} + \dot{m}_{24}\, h_{24} \tag{3.37}$$

The heat transfer rate supplied to this unit is calculated as the difference in enthalpies of the hot fluid as:

$$\dot{Q}_{Desalination} = \dot{m}_{16}(h_{16} - h_{17}) \tag{3.38}$$

Additionally, the overall entropy balance equation is:

$$\dot{m}_{16}\, s_{16} + \dot{m}_{21}\, s_{21} + \dot{S}_{gen,Desalination} = \dot{m}_{17}\, s_{17} + \dot{m}_{23}\, s_{23} + \dot{m}_{24}\, s_{24} \tag{3.39}$$

To find the exergy destruction rate for this unit, we use this expression:

$$\dot{E}x_{D,Desalination} = T_o\, \dot{S}_{gen,Desalination} \tag{3.40}$$

3.4.9 Pumps 3 and 4

Modelling pumps 3 and 4 is the same as the previous two pumps described earlier. Pump 3 has the flow of desalinated water going through it. The mass balance equation is:

$$\dot{m}_{24} = \dot{m}_{25} \tag{3.41}$$

and the energy balance is:

$$\dot{m}_{24}\, h_{24} + \dot{W}_{Pump3} = \dot{m}_{25}\, h_{25} \tag{3.42}$$

The power input to this pump is:

$$\dot{W}_{Pump3} = \frac{\dot{m}_{24} v_{24}}{\eta_{Pump}} (P_{25} - P_{24}) \tag{3.43}$$

The entropy balance is:

$$\dot{m}_{24}\, s_{24} + \dot{S}_{gen,Pump3} = \dot{m}_{25}\, s_{25} \tag{3.44}$$

and the exergy destruction rate is:

$$\dot{E}x_{D,Pump3} = T_o\, \dot{S}_{gen,Pump3} \tag{3.45}$$

Pump 4 has the exact balance equations, but the inlet stream is changed from state 24 to state 27, and the outlet state changes from 25 to 28. Since these are exactly the same, it is redundant to write the pump model for pump 4 here.

3.4.10 PEM electrolyzer

The last component in the studied system is the PEM electrolyzer. The thermodynamic model starts with the mass balance equation and it is:

$$\dot{m}_{28} = \dot{m}_{29} + \dot{m}_{30} \quad (3.46)$$

The electrolyzer is assumed to operate under steady temperature and pressure. The power consumption can be found from knowing the efficiency of the PEM which is assumed to be 77%. The mathematical definition for this efficiency is described as:

$$\eta_{PEM} = \frac{\dot{W}_{13}}{\dot{m}_{30}\, LHV_{H2}} \quad (3.47)$$

where LHV_{H2} is the lower heating value of hydrogen, and $\dot{m}_{30}$ is the hydrogen production rate. The heat losses associated with this process can be found from the energy balance equation which is:

$$\dot{m}_{28}h_{28} + \dot{W}_{13} = \dot{m}_{29}h_{29} + \dot{m}_{30}h_{30} + \dot{Q}_{PEM\ loss} \quad (3.48)$$

The overall chemical equation that describes this process is:

H_2O + electric energy $\longrightarrow$ $H_2 + 0.5O_2$

The second law analysis of this electrolyzer is described through the entropy balance equation as:

$$\dot{m}_{28}s_{28} + \dot{S}_{gen,PEM} = \dot{m}_{29}s_{29} + \dot{m}_{30}s_{30} + \frac{\dot{Q}_{PEM\ loss}}{T_o} \quad (3.49)$$

With knowing the entropy generation rate, the exergy destruction rate is:

$$\dot{Ex}_{D,PEM} = T_o\, \dot{S}_{gen,PEM} \quad (3.50)$$

3.4.11 Overall performance

The last part of the thermodynamic analysis of the considered system is to define the overall performance indicators. The first indicator is the Coefficient of Performance (COP) of the heat pump according to energy terms. This is defined in this study as:

$$COP_{energy,heat\ pump} = \frac{\dot{m}_6(h_6 - h_7)}{\dot{W}_{14} + \dot{W}_{Pump1}} \quad (3.51)$$

and the exergy form of this heat pump COP is:

$$COP_{exergy,cascaded\ heat\ pump} = \frac{\dot{m}_6(ex_6 - ex_7)}{\dot{W}_{14} + \dot{W}_{Pump1}} \tag{3.52}$$

Now, for the entire system, the system efficiency in energy terms is:

$$\eta_{energy,system} = \frac{\dot{W}_9 + \dot{m}_6(h_6 - h_7) + \dot{m}_{26}h_{26} + \dot{m}_{30}\ LHV_{H2}}{\dot{W}_{10}} \tag{3.53}$$

While the exergy form is:

$$\eta_{exergy,system} = \frac{\dot{W}_9 + \dot{m}_6(ex_6 - ex_7) + \dot{m}_{26}ex_{26} + \dot{m}_{30}\ ex_{30}}{\dot{W}_{10}} \tag{3.54}$$

The building heat load is calculated using :

$$BL_{tj} = \left(\frac{t_{zi}-t_j}{t_{zi}-t_{OD}}\right) C_z * \dot{q}_{AFull} \tag{3.55}$$

Were t_j is the outdoor bin temperature, t_{zi} as the zero-load temperature, t_{OD} as the outdoor design temperature, C_z as the slope factor and $\dot{q}_{AFull}$ as the heating capacity.

3.4.12 Cost and emissions of selected energies

Since the community will require a freshwater system regardless of which energy source we use the cost of the system will only focus on the energy source and the wind turbine. 70% of heating in Canada is by natural gas [69]. Nunavut does not have natural gas pipelines; therefore most heating is done by refined petroleum or electricity [70]. Table 3.2 contains the cost of energy in Nunavut.

Table 3.2: Cost of energy in Nunavut [71],[72]

Energy source	Unit	Cost ($)
Electricity	kWh	0.38
Heating Diesel	L	1.08
Heating Gasoline	L	1.07

As shown in Table 3.3, the top 5 different turbines were analyzed based on cold temperature operation and the E70E4 was chosen based on price and performance. It cost $4.2 million for a E70E4 wind turbine, transformer, blade heating and transportation from Enercon in Montreal to Nunavut [73].

Table 3.3: Wind turbines for cold climates [73]

Manufacturers	Siemens	Northern power	Enercon	Senvion	Nordex
Turbine model	SWT-2.3-101	NW100 Arctic	E70E4	3.2M114CCV	N117/3000
Power	2.3 MW	100 kW	2.3 MW	3.2 MW	3.0 MW
Cold temperature operation	-25 ^{O}C	-40 ^{O}C	-40 ^{O}C	-30 ^{O}C	-30 ^{O}C
Anti/de-icing capabilities	De-icing system	Hydrophobic polymer coating	Anit -icing: hot air blown from the start	Anti-icing and de-icing	Anti-icing system heats

			of the blade to the edge		hip of the blades
Overall cost ($ millions)	**4.5**	**4.3**	**4.2**	**4.3**	**4.2**

As shown in Table 3.4, *although wind energy is a renewable energy source, it still emits some* CO_2*. However, when compared to hydroelectricity, it emits the least amount of* CO_2 *equivalent emissions per kg. This amount is more than that of solar. When compared to fossil fuel energy resources such as natural gas, diesel, and gasoline, it emits significantly less.*

Table 3.4: Comparison of CO_2 equivalent emissions between energy sources [74].

Resource	CO_2 equivalent emissions (kg)
Natural gas	0.525
Hydroelectric	0
Nuclear	1.5×10^{-4}
Wind	7.4×10^{-4}
Solar	6.15×10^{-3}
Diesel	2.69
Gasoline	2.35

Chapter 4: Results and Discussion

This chapter discusses the results of the proposed system in two stages, namely the case study and parametric studies.

4.1 Case Study

The thermodynamic simulation work of the multi-output system has resulted in the state points shown in Table 4.1. The mass flow rate of the refrigerant R134a used in the first cycle of the heat pump is 3.127 kg s^{-1}, while the water flow rate is more than three times higher than this value, which is 11.25 kg s^{-1}. This higher mass flow rate for water is needed to absorb the high amount of thermal energy given by the refrigerant which comes from the seawater and Compressor 1. The same relation is found in the desalination unit where the incoming seawater absorbs heat from a high-pressure and high-temperature refrigerant stream named state 16. This heat helps the purifies the seawater to produce freshwater for domestic use and the PEM electrolyzer. The ratio of produced freshwater (state 24) to the incoming seawater (state 21) in the desalination unit is calculated to be 9.6% which is reasonable for typical multi-effect desalination units. The rest of the seawater is ejected back to the sea as brine fluid. The compressors' power inputs can be calculated from this table as well. For compressor 1, the power input is 98.2 kW, and for compressor 2, the power input is 13.7 kW. One of the benefits of needing lower mass flow rates for the refrigerant is lowering the power demands of the integrated system which are supplied by the wind turbines. The energy and exergy COP values of the heat pump are 5.08 and 0.71, respectively. Also, the overall energy and exergy efficiencies of the entire system are found to be 1.39 and 0.402, respectively.

The second part of this thermodynamic simulation work of this integrated system is the exergy analysis. This analysis compares the exergy destruction rates of the different components of the system to determine the locations for possible improvements. From Table 4.2, it is seen that the highest exergy destruction rate is found in the fresh water tank at a value of 286.6 kW. This value is much higher than the exergy destruction rates of all the other components in the system, because it is losing heat to the atmosphere, which is not a useful energy to the system. The second and third components in terms of exergy destruction rates are compressor 1 and compressor 2, at values of 12.46 kW, and 1.594 kW, respectively. The pumps have orders of magnitude lower exergy destruction rates than the compressors, which means that pumps are more exergetically efficient. One way to improve the overall efficiency of this system is to use refrigerants that can carry more thermal energy per one kilogram of refrigerant compared to water. This will lower the mass flow rate of the refrigerants needed to supply the same amount of heat to the system which will lower the exergy destruction rates of the compressors linearly according to the exergy model of this system mentioned before.

Table 4.1: Thermodynamic state points of the multi-output wind turbine-based system.

State #	Fluids	Temperature (°C)	Pressure (kPa)	Specific enthalpy (kJ kg^{-1})	Specific entropy (kJ kg^{-1} K^{-1})	Specific exergy (kJ kg^{-1})	Mass flow rate (kg s^{-1})
0	R134A	10	101.325	263.8	1.062	NA	NA

0	Water	10	101.325	42.12	8.151	NA	NA
1	R134A	10	414.9	256.2	0.9264	30.65	3.127
2	R134A	60	1511	287.6	0.9405	58.12	3.127
3	R134A	60	1683	139.4	0.4892	37.58	3.127
4	R134A	10	414.9	64.43	0.2529	30.53	3.127
5	Water	60	300	251.4	0.8311	303.5	11.25
6	Water	60.03	1000	251.6	0.831	17.55	11.25
7	Water	50	1000	230.7	2.12	11.69	11.25
8	Water	22	2.645	2541	8.622	101.8	NA
16	R134A	90	2757	301.6	0.9551	70.77	0.3021
17	R134A	10	414.9	65.43	0.2529	30.53	0.3021
18	R134A	10	414.9	65.43	0.2529	30.53	0.3021
19	R134A	10	414.9	256.2	0.9264	30.65	0.3021
20	Salt	10	550	44.65	NA	NA	2.55
21	Water	62.61	550	262.6	NA	NA	2.55
22	Salt	70	350	291.2	NA	NA	2.298
23	Salt	70	550	291.4	NA	NA	2.298
24	Water	70	350	293.3	0.9549	23.71	0.2448
25	Water	70	550	293.5	0.955	23.92	0.2448
27	Water	60	101.3	293.3	0.9549	23.71	0.2448
28	Water	60	550	293.3	0.9558	23.97	0.2448
31	Salt	10	101.3	42.74	NA	NA	2.55
32	Salt	20	350	84.28	NA	NA	2.298

Table 4.2: Exergy destruction rates of components of the multi-output wind turbine-based system.

Component	Exergy destruction rate (kW)
Compressor 1	12.46
Compressor 2	1.594
Fresh water tank	286.6
Pump 1	0.01403
Pump 3	0.002261
Pump 4	0.9564

4.2.1 Heat load of a community

The first parametric study presented in this section is the heat load of a community in the location mentioned earlier, as shown in Figure 4.1. The heat load changes linearly as the outside air temperature changes. As expected, the heat load drops linearly from 600 kW to 90 kW as the outside air temperature increases from -32 to 10°C. Since the city chosen is in northern Canada, there will always be a need for space heating every month of the year. Next, the wind turbine power input needed to supply enough heating as a function of outside air temperature is plotted in Figure 4.2. There exists some nonlinearity in the curve as the temperature increases from -32 to 10°C. Obviously, the wind turbine power input decreases as the outside air temperature increases, and this is directly due to the fact that the heating demands of the community are reduced. This shows that space heating is the main and driving commodity of this integrated system. This agrees with the results discussed earlier from the exergy destruction rate of compressor 1. Another point that can be made when comparing the heating load and the needed wind turbine power input is that the values of both of them are close to each other at every temperature level. To illustrate, when the outside air temperature is -32°C, the heating load is 600 kW, and the needed power input is only a little higher at a value of 750 kW. This indicates that this integrated system supplies heat to the community with appreciable and consistent efficiency across a wide range of temperatures.

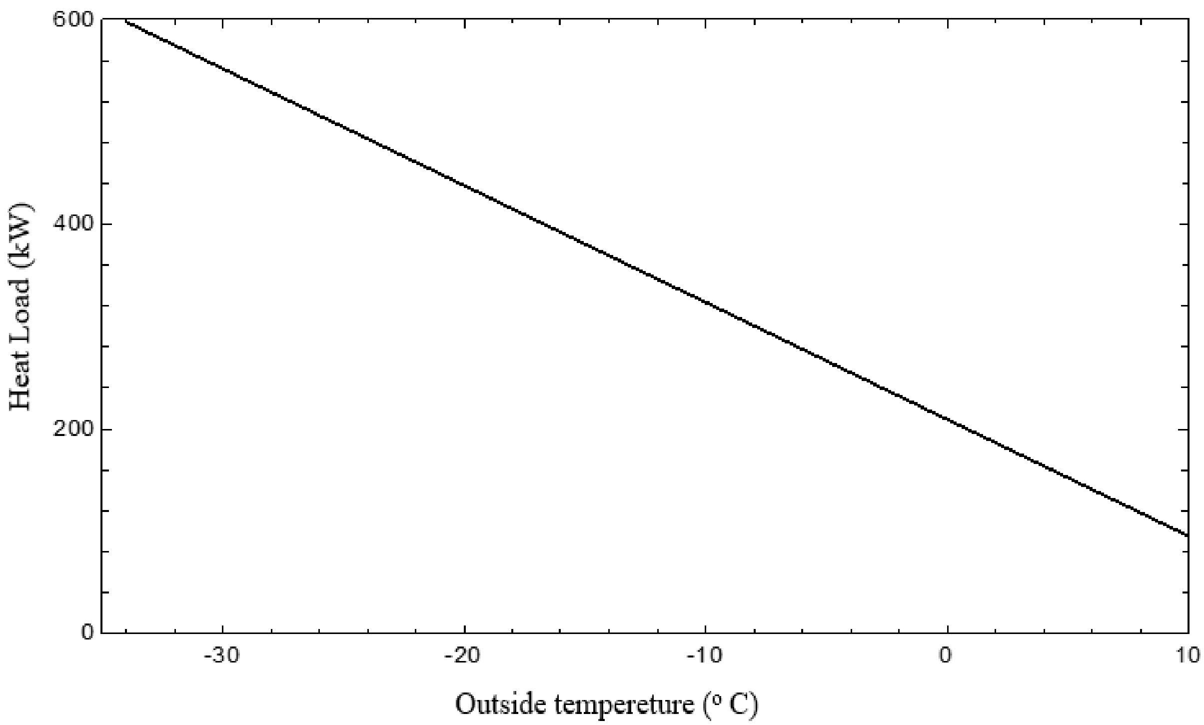

Figure 4.1 Heating load of a community in the chosen location as a function of outside air temperature.

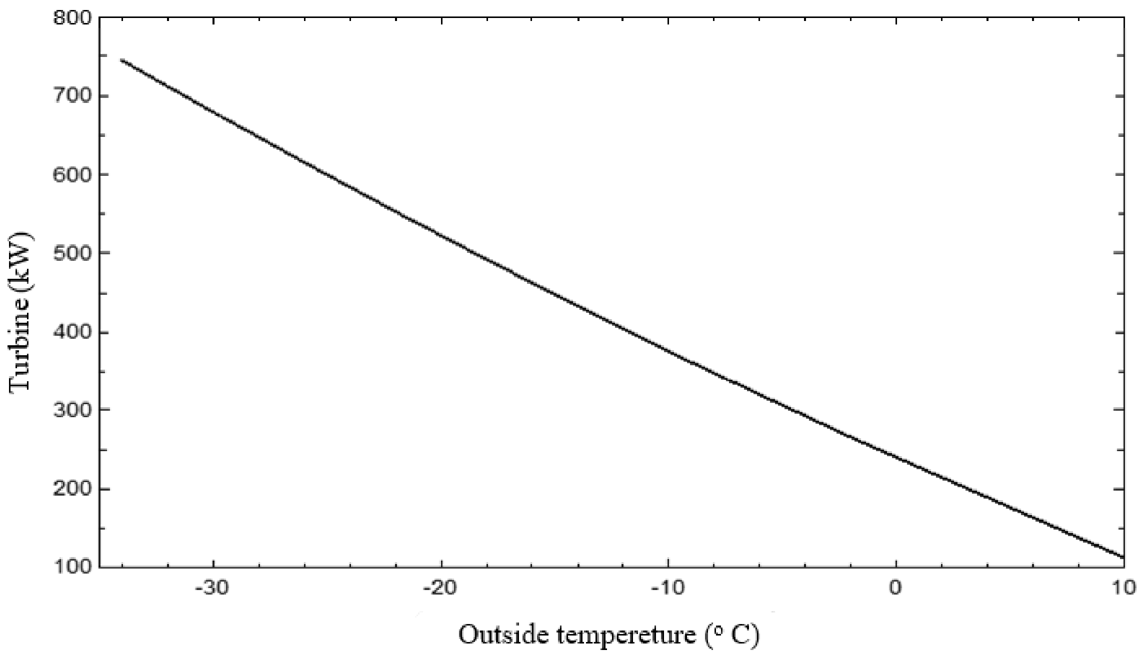

Figure 4.2: Turbine power input needed as a function of outside air temperature

Figure 4.3 presents a comparison between the heat pump of the present system and a typical air heat pump in terms of energy COP. As clearly shown, there is a huge difference between the two heating systems as the temperature increases from extremely low-temperature levels to a moderate outside air temperature. Here, the advantage of using the heat pump in the proposed system is presented clearly since the heat is withdrawn from a stable source which is the sea, at a consistent temperature of 10°C. The consistency of the sea temperature across seasons keeps the energy COP of the heat pump in the proposed system constant and stable at a value of 5.08. On the other hand, the air heat pump performance depends heavily on the outside air temperature since it uses ambient air as a source of its input heat. The effects of the outside air temperature are tremendous, and the energy COP changes from 2.6 to 5.08 as the temperature increases.

The same comparison between these two heat pumps, but now in terms of exergy COP is depicted in Figure 4.4. As anticipated, the exergy COP of the proposed system stays constant for the same reason, which is the stability of the sea temperature that is the source of heat for the proposed system. The value is 0.71. However, the exergy COP of the air heat pump changes rapidly from a low point of 0.36 to the same point as the proposed system at 0.71. This is because both sources of heat, namely sea and air have the same temperature. Again, the advantage of using the proposed system in terms of supplying heat at a consistent performance across a wide range of temperature levels is demonstrated exergetically.

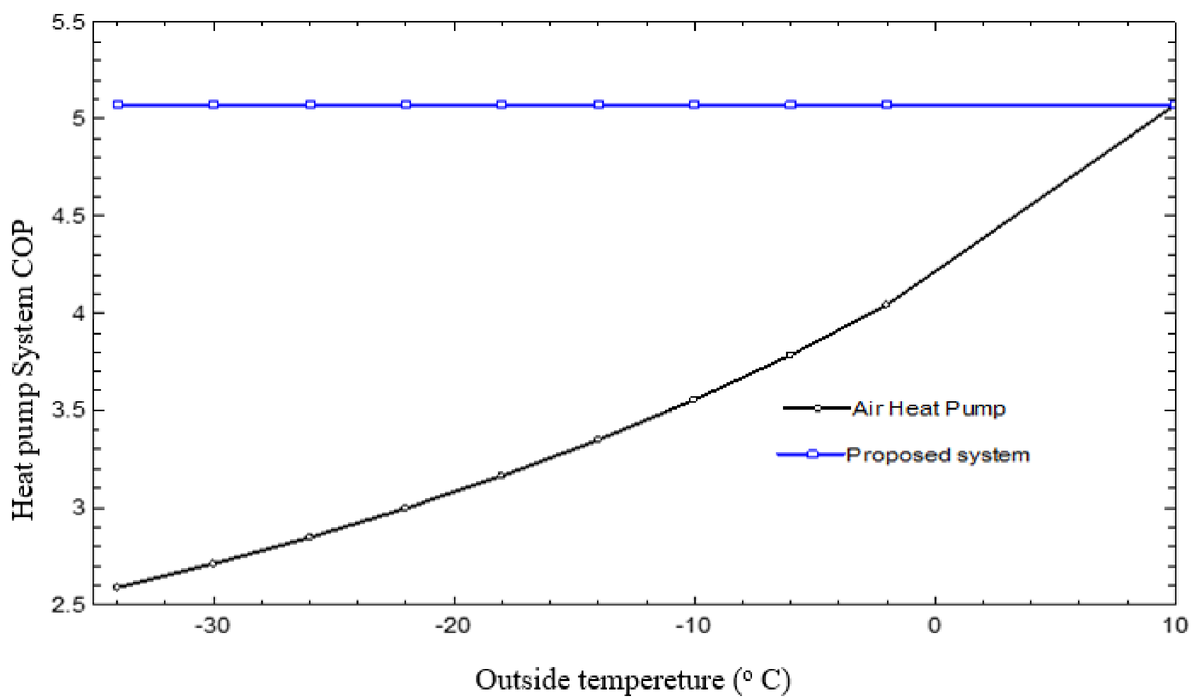

Figure 4.3: Energy COP of the proposed system and energy COP of a typical Air heat pump as functions of outside air temperature.

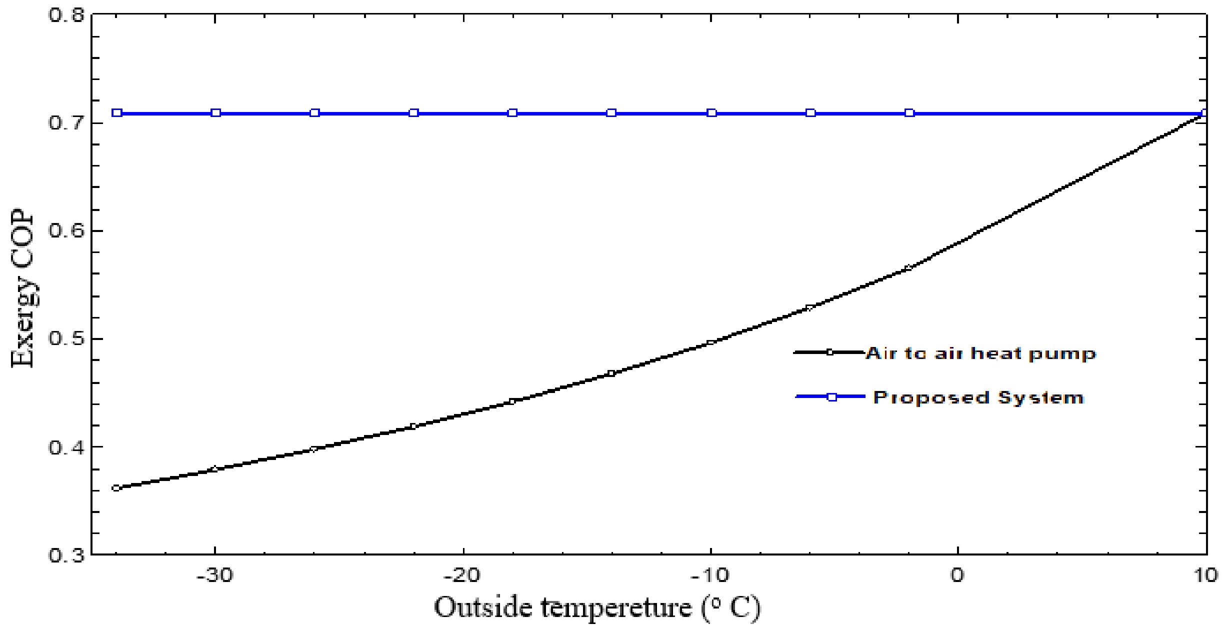

Figure 4.4: Exergy COP of the proposed system and exergy COP of a typical Air heat pump as functions of outside air temperature.

4.2.2 Overall energy and exergy efficiencies of the System

Next, it is important to compare the proposed system's overall performance to a typical air heat pump energetically. In Figure 4.5, the overall energy efficiency of the system is reported to have a constant value of around 1.39, while the air heat pump's overall energy efficiency varies dramatically from 1.18 to 1.39 as the temperature of the heat source, which is air, increases. The main reason these values are higher than unity is that heat pumps are the main subsystem of both systems in the comparison. Heat pumps by nature have higher than unity values. It is interesting to note that the proposed system's overall energy efficiency is almost 6 times lower than its energy COP and this is due to the deficiencies of the other components in the integrated system, such as the desalination unit and the PEM electrolyzer which produce fresh water and hydrogen, respectively, at low efficiencies.

Finally, a similar comparison of the overall exergy efficiencies of the integrated system and an air heat pump is presented in Figure 4.6. The value of the overall exergy efficiency of the proposed system remains the same across a wide range of outside air temperatures, which is 0.471. This is unlike the air heat pump system, which has a varying trend as the outside air temperature changes. As the temperature increases, the overall exergy efficiency goes up from around 0.359 to 0.471. This shows the heavy dependency on the outside temperature, while the proposed system is exergetically independent of the outside temperature because the heat source temperature, which is the sea, is always constant when the air temperature changes.

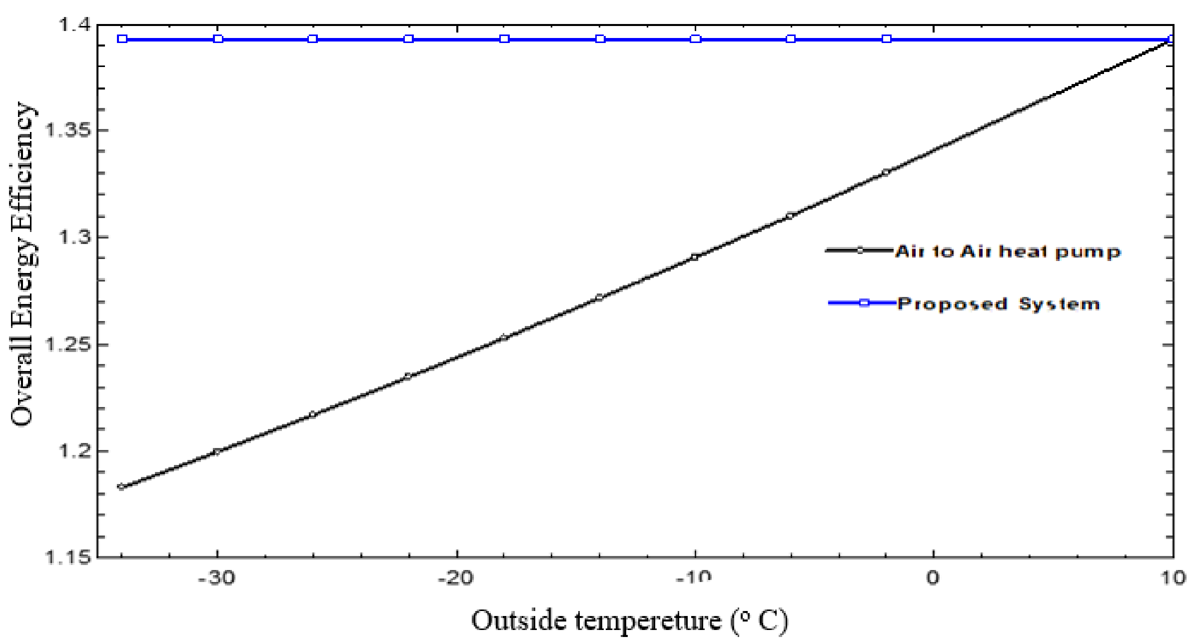

Figure 4.5: Overall energy efficiency of the proposed system and overall energy efficiency of a typical Air heat pump as functions of outside air temperature.

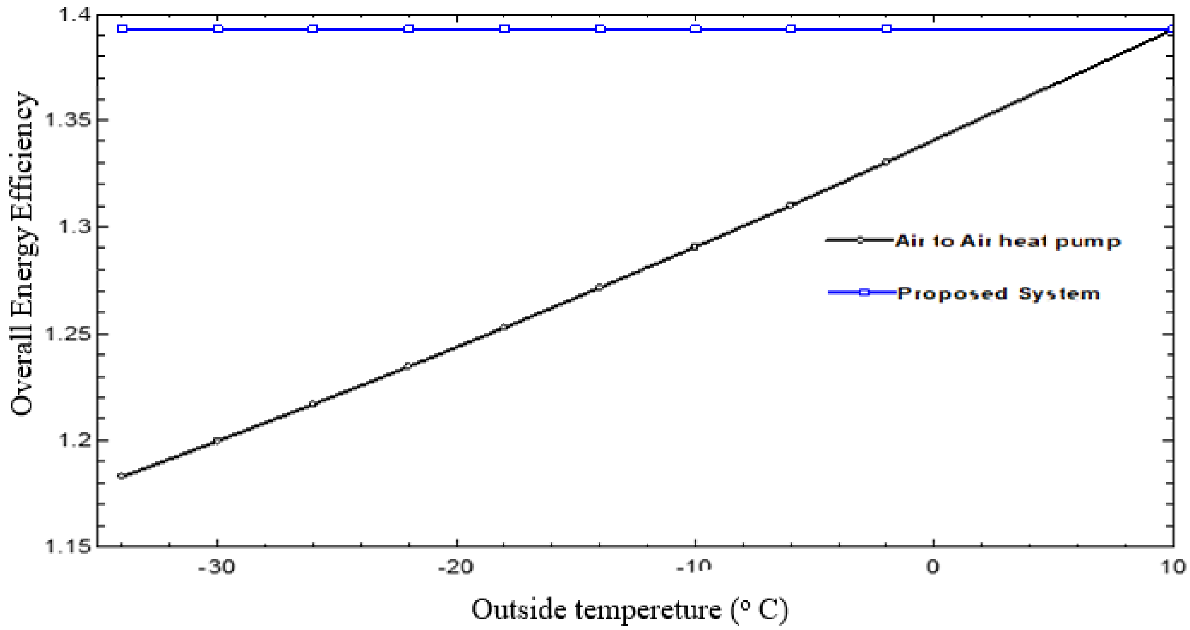

Figure 4.6: Overall exergy efficiency of the proposed system and overall exergy efficiency of a typical Air heat pump as functions of outside air temperature

4.2.3 Cost of energy

Canada has some wind energy projects in Nunavut, however, there is no history of cost in the literature to use as a benchmark to obtain the true cost estimates of the proposed wind energy project. The financial viability of the proposed system is determined by its economic conditions. The study was conducted using today's financial factors, which will change over time. The annual real interest rate, project lifetime, and system fixed capital can all be specified using RETScreen [67]. As a result, all the cost calculation was based on today's rates over a period of 20 years. They are a fixed cost that occurs regardless of the system's size, architecture, or energy source. These characteristics would have an impact on each system's total net present cost but by the same amount. As a result, they have no bearing on the cost differential in the system. The estimates in this thesis were developed using the wind turbine, Enercon E70 2.3 MW. This wind turbine has blade heating technology. The estimated cost of the wind turbine and installation is $4.3 million [73]. Table 4.3 shows cost estimate of heating with wind turbine. Since the wind energy resource is free, the energy cost of using a wind turbine is zero. Table 4.4 contains the cost of heating with electricity, year-round. Even though the temperature of February is lower, the cost of heating with electricity is higher in January because January has more days. Table 4.5 shows the cost of heating with gasoline year-round. Gasoline is the most expensive form of heating in Nunavut. Table 4.6 shows the cost of heating with diesel year-round. July is the lowest cost of heating. The life span of most wind turbines is 20 years. Based on calculations using rates for electricity, gasoline and diesel. The cost estimate of using electricity to heat the community is $5.5 million, while it cost $13.94 million and $12.31 million for gasoline and diesel. Since wind is free, the energy the wind turbine produces will be free. It cost less to use the wind turbine to run the whole system than to use electricity, gasoline and diesel for heating the community, as shown in Table 4.7.

Table 4.3: Cost estimate of heating with wind turbine

Month	Average outdoor temperature (°C)	Monthly heating (MW)	Wind energy (MW)
January	-32	460.3	90.3
February	-33	424.0	83.1
March	-31.2	453.0	88.8
April	-23.5	370.2	72.6
May	-11	268.3	52.6
June	-0.6	167.6	32.8
July	4	131.1	25.7

August	1.9	150.3	29.4
September	-5	206.4	40.4
October	-15.2	306.7	60.1
November	-24.3	377.3	74.0
December	-29	432.9	84.9

Table 4.4: Cost estimate of heating with electricity

Month	Average outdoor temperature (°C)	Monthly heating (MW)	Electricity (MW)	Electricity cost ($)
January	-32	460.3	90.2	34,299
February	-33	424.0	83.1	31,595
March	-31.2	453.0	88.8	33,754
April	-23.5	370.2	72.6	27,587
May	-11	268.3	52.6	19,988
June	-0.6	167.6	32.9	12,484
July	4	131.1	25.7	9,766
August	1.9	150.3	29.4	11,197
September	-5	206.4	40.4	15,386

October	-15.2	306.7	60.1	22,850
November	-24.3	377.3	74.0	28,114
December	-29	432.9	84.9	32,254

Table 4.5: Cost estimate of heating with gasoline

Month	Average outdoor temperature (°C)	Monthly heating (MW)	Gasoline (kL)	Gasoline cost ($)
January	-32	460.3	79.6	85,660
February	-33	424.0	73.3	78,907
March	-31.2	453.0	78.3	84,298
April	-23.5	370.2	64.0	68,897
May	-11	268.3	46.4	49,919
June	-0.6	167.6	29.0	31,179
July	4	131.1	22.7	24,389
August	1.9	150.3	26.0	27,963
September	-5	206.4	35.7	38,426
October	-15.2	306.7	530.1	57,067
November	-24.3	377.3	65.2	70,214

December	-29	432.9	74.8	80,554

Table 4.6: Cost estimate of heating with diesel

Month	Average outdoor temperature (°C)	Monthly heating (MW)	Diesel (kL)	Diesel cost ($)
January	-32	460.3	70.0	75,645
February	-33	424.0	64.5	69,682
March	-31.2	453.0	68.9	74,443
April	-23.5	370.2	56.3	60,842
May	-11	268.3	40.8	44,083
June	-0.6	167.6	25.4	27,534
July	4	131.1	19.9	21,538
August	1.9	150.3	22.9	24,694
September	-5	206.4	31.4	33,934
October	-15.2	306.7	46.7	50,395
November	-24.3	377.3	57.4	62,006
December	-29	432.9	65.9	71,136

Table 4.7: Cost estimate of heating for 20 years

Resource	Cost ($ million)
Wind turbine	4.3
Electricity	5.58
Gasoline	13.94
Diesel	12.31

4.2.3 Environmental factors

For a similar heating output, the environmental analysis of the system focused on the energy consumption of a wind turbine versus the electricity, gasoline, and diesel consumption of the heating system. This energy study was conducted with the expectation that the heat pump system would provide homes with more efficient heating. The materials utilized and the product's end of life aren't among the other environmental considerations considered in the examination. Standard materials had to be used in order to meet the design goals of manufacturability and ease of assembly. Using data from Ontario power generation and RETScreen wind energy project model, GHG's were calculated using monthly average temperatures for 20 years. The GHG for isolated gird, central -grid, off-grid wind power, gasoline and diesel heating systems were calculated. Tables 8 to Tables 11 shows the monthly average CO_2 emission equivalence per kg for wind turbine, electricity, gasoline and diesel respectively. While Table 12 shows 20 years of CO_2 emission equivalence per kg for , wind turbine, electricity, gasoline and diesel. The results show that even though the power consumption for wind energy and electricity are the same the GHG for wind energy is 2205 CO_2 eq kg and 39.5 million CO_2 eq kg for electricity. The same applied to gasoline and diesel. Even though 13 million liters of gasoline will be used while 11 million liters of diesel will be used to heat the community for 20 years. The GHG for diesel is higher than that of gasoline, it is 310 million and 271 million for gas and diesel respectively . This is evidence that wind energy is a more environmentally friendly alternative than using electricity, gasoline and diesel.

Table 4.8: GHG of heating with wind turbine

Month	Average outdoor temperature (°C)	Monthly heating (MW)	Wind energy (MW)	Wind energy CO_2 eq kg
January	-32	460.3	90.2	14
February	-33	424.0	83.1	12
March	-31.2	453.0	88.8	13
April	-23.5	370.2	72.6	11
May	-11	268.3	52.6	8
June	-0.6	167.6	32.9	5
July	4	131.1	25.7	4
August	1.9	150.3	29.4	4
September	-5	206.4	40.4	6
October	-15.2	306.7	60.1	9
November	-24.3	377.3	74.0	11
December	-29	432.9	84.9	13

Table 4.9: GHG of heating with electricity

Month	Average outdoor temperature (°C)	Monthly heating (MW)	Electricity (MW)	Electricity CO_2 eq kg
January	-32	460.3	90.2	242,800
February	-33	424.0	83.1	223,660
March	-31.2	453.0	88.8	238,941
April	-23.5	370.2	72.6	195,285
May	-11	268.3	52.6	141,493
June	-0.6	167.6	32.9	88,375
July	4	131.1	25.7	69,130
August	1.9	150.3	29.4	79,261
September	-5	206.4	40.4	108,917
October	-15.2	306.7	60.1	161,754
November	-24.3	377.3	74.0	199,020
December	-29	432.9	84.9	228,327

Table 4.10: GHG of heating with gasoline

Month	Average 0utdoor temperature (°C)	Monthly heating (MW)	Gasoline (L)	Gasoline CO_2 eq kg
January	-32	460.3	79.6	1,664,259
February	-33	424.0	73.3	1,533,068
March	-31.2	453.0	78.3	1,637,805
April	-23.5	370.2	64.0	1,338,571
May	-11	268.3	46.4	969,853
June	-0.6	167.6	29.0	605,764
July	4	131.1	22.7	473,849
August	1.9	150.3	26.0	543,289
September	-5	206.4	35.7	746,566
October	-15.2	306.7	530.1	1,108,734
November	-24.3	377.3	65.2	1,364,171
December	-29	432.9	74.8	1,565,058

Table 4.11: GHG of heating with diesel

Month	Average outdoor temperature (°C)	Monthly heating (MW)	Diesel (L)	Diesel CO_2 eq kg
January	-32	460.3	70.0	1,905,045
February	-33	424.0	64.5	1,754,874
March	-31.2	453.0	68.9	1,874,764
April	-23.5	370.2	56.3	1,532,236
May	-11	268.3	40.8	1,110,172
June	-0.6	167.6	25.4	693,407
July	4	131.1	19.9	542,406
August	1.9	150.3	22.9	621,893
September	-5	206.4	31.4	854,580
October	-15.2	306.7	46.7	1,269,147
November	-24.3	377.3	57.4	1,561,540
December	-29	432.9	65.9	1,791,492

Table 4.12: GHG of heating for 20 years

Resource	CO_2 eq kg
Wind turbine	2205

Electricity	39.5 million
Gasoline	271 million
Diesel	310 million

Chapter 5: Conclusion

This chapter consists of two main sections, including the concluding remarks, and recommendations for future work, as outlined below.

5.1 Concluding Remarks

This thesis proposed an integrated direct wind-powered energy system that can provide drinking water, domestic hot water, space heating as well as electricity for residential applications of a remote community in a cold region. The proposed system was optimized for space heating since it constitutes the largest portion of residential energy by employing an innovative heat pump suitable for cold climates. The novelty of the system is that it uses a wind turbine to directly power the heat pump, with the seawater as the evaporator that provides a constant temperature, without the need to use the wind turbine to convert the wind energy to electricity to power the heat pump. The study used a remote residential community of the Canadian Navy and Coast Guard located at Resolute Bay, Nunavut (Canada) that houses 60 officers as a case study. Employing the Engineering Equation Solver and the RETscreen software for calculations and environmental data, the study performed thermodynamic analyses to evaluate the performance, costs and environmental impact of the proposed system. The analyses revealed that there is enough wind power to keep the wind turbine running all year round. It also indicates that in addition to being more environmentally benign, purchasing a wind turbine is more cost-effective over its lifetime than purchasing electricity, gasoline, or diesel. Based on these preliminary results, the proposed system is promising.

The following are a summary of the results:

- The wind turbine produces sufficient energy to power all the subsystems even at the lowest monthly average wind speed at the selected location.
- The break-even period for buying electricity from the province of Nunavut for the system versus buying a wind turbine is 15 years.
- The break-even for buying gasoline for the system versus buying a wind turbine is 6 years.
- The break-even for buying diesel for the system versus buying a wind turbine is 6.5 years.
- The GHG produced by buying electricity from the province of Nunavut is 17,931 times more annually than that of the wind turbine.
- The GHG produced by gasoline is 122,911 times more annually than that of the wind turbine.
- The GHG produced by diesel is 140,694 times more annually than that of the wind turbine.
- The COP for heat pump system and overall system are 5.1 and 1.3, respectively.
- The exergy efficiencies for the heat pump and overall system are 0.7 and 0.45, respectively.
- At the lowest temperature of -35 ºC the wind turbine provides power of 750 kW for the system.
- The heat load for the system is 600 kW at the lowest temperature of -35 ºC.
- The system is able to produce 21 m^3 of drinking water in a day.

5.2 Future Work

The following are recommendations for future work:

- Perform a detailed cost analysis that includes every component cost and transportation cost.
- Investigate potential grid frequency and voltage fluctuation issues, as well as the requirement for battery energy storage or other stabilizing technologies.

- Conduct detailed life cycle analyses of the proposed system against the systems available in Nunavut
- Investigate the use of novel refrigerants, such as R410A

5.3 Publications

1. **B. Addo-Binney**, W. Besada, M. Agelin-Chaab, Analysis of an Integrated Thermal Energy System for Applications in Cold Regions, J. Energy Resour. Technol. 144 (2022) 1–14. https://doi.org/10.1115/1.4050887.
2. **B. Addo-Binney**, M. Agelin-Chaab, E. Bamfo, S. Koohi-Fayegh, A Comparative Life Cycle Assessment of a Cascade Heat Pump and a Natural Gas Furnace for Residential Heating Purposes, Integr. Environ. Assess. Manag. (2021) ieam.4494. https://doi.org/10.1002/IEAM.4494.

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Appendices

Appendix 1: A Sample Calculation of the Cost Estimate for Electricity in January

Using data from Figure 3.1, the January temperature of -32 ^{O}C can be found. From Figure 4.1 using the outside temperature of January of -32 ^{O}C, the heat load was found as 618.7 kW.

Since there are 31 days in a month and 24 hrs in a day. The number of hrs in January is 31 x 24 = 744.

744 Hrs in January was multiply by the heat load to get the total heat load for the month as 744 x 618.7= 460326.9 kWh.

The monthly heat load was divided by the COP of the turbine which is 5.1, 460326.9 kWh/5.1 = 90260. Then it was multiplied by the cost of electricity to obtain, 90260 x 0.38 = $34299, as shown in Table 4.4

Appendix 2: A Sample Calculation of the GHG for electricity in January

The January temperature of -32 ^{O}C can be calculated using data from Figure 3.1.

Using Figure 4.1 and the January outside temperature of -32 ^{O}C, the heat load was found to be 618.7 kW.

Since there are 31 days in a month and 24 hours in a day.

The number of hours in January is 31 × 24 = 744, which was multiplied by the heat load to get the total heat load for the month, which is 744 × 618.7= 460326.9 kWh.

The monthly heat load was divided by the turbine's COP, which is 5.1, for a total of 460326.9 kWh/5.1 = 90260.

The CO_2 equivalent per kg of electricity is 2.69, was multiplied by the monthly heat load of electricity, 90260 x 2.69, to yield a GHG of 232800, as shown in Table 4.9.