Techno-Economic Design of a 100% Renewable House

A Case Study for St. John's

ENGI-990B (MESE Project Course)

Project report

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Abstract

Newfoundland and Labrador has agreed to a zero net GHG emissions by 2050 target which if strived for would entail a large transition in the province's energy consumption. This study aimed at the techno-economic design of a solar thermal combi system to replace the existing oil heating system for a residence in St. John's followed by an energy efficiency retrofit. First, eleven fundamental solar thermal equations were combined with load and weather data to derive relevant insights. It was found that a combi system can be described referencing two boundary conditions based on whether the water in the system is used or consumed with the system converging on the lower boundary as the tank size increases. This was confirmed using f-chart calculations and polysun simulation. Economic evaluation showed that the designed system was profitable and more viable than an electric heating system using PV-Wind technologies. The system was able to reduce the house's emissions by over 5.2 tonnes/year. A PVT system was also designed which was superior to the combi and PV systems in terms of area utilization and energy generation but worse in terms of economic performance. Then a south facing SolarWall was evaluated in RETScreen. The results show that by including the amount of building heat recaptured, the SolarWall produces 51% more energy than a roof mounted solar air collector of similar size. Finally, an energy retrofit was investigated which replaces old appliances and building elements with energy efficient ones. The results were evaluated based on energy usage change, emissions reduction and economic merit. It was concluded that air leakage reduction is able to save the most energy while exhibiting the most favorable economics. The final combination of included measures yielded a 52% reduction in heating oil consumption, 8% decrease in electricity consumption and over $1900 in annual savings which resulted in $11,000 in profit over the project's lifetime.

Key words- Solar combi system, PVT, SolarWall, Energy efficiency, BEOPT, RETScreen, Polysun, Air leakage, Newfoundland, Zero net energy, Technoeconomic design, Renewable energy.

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List of Abbreviations

A2L: Air-to-Liquid
AC: Air Conditioner
AET: Alternate energy technology
AFUE: Annual Fuel Utilization Efficiency
ASHPWH: Air Sourced Heat Pump Water Heater
ASHRAE: the American Society of Heating, Refrigerating and Air-Conditioning Engineers
BC: British Coloumbia
BE: Building Element
BEOPT: Building Energy Optimization Tool
BIPVT: Building Integrated Photovoltaic Thermal panels
CAD: Canadian Dollars
CanSIA: Canadian Solar Industry Association
CDC: Center for Disease Control
CDD: Cooling Degree Day
CEF: Combined Energy Factor
CFL: Compact Florescent Lightening
CO_{2e}: Carbon Dioxide Equivalent Emissions
COP: Coefficient of Performance
CTS: Clean Technology Scenario
DHW: Domestic Hot Water
DIY: Do It Yourself
DOE: U.S Department of Energy
DR: Demand Response
DSM: Demand Side Management
EA: Electrical Appliances
EIA: U.S. Energy Information Agency
EM: Efficiency Measure
EPA: U.S. Environment Protection Agency
ETC: Evacuated-Tube Collector
FTL: Freeze Tolerance Limit
GDP: Gross Domestic Product
GHG: Greenhouse Gases
GTC: Glazed Transpired Collector
HDD: Heating Degree Day
HIR: Heating input ratio
HOMER: Hybrid Optimization Model for Electric Renewables
HTF: Heat Transfer Fluid
HWB: Hottel–Whillier–Bliss equation
HX: Heat Exchanger
ICS: Integral Collector Storage
IMEF: Integrated Modified Energy Factor
IWF: Integrated Water Factor
L2L: Liquid-to-Liquid
LCOE: Levelized Cost of Energy
LCOH: Levelized Cost of Heat
LED: Light Emitting Diode
LHS: Latent Heat Storage
NL: Newfoundland and Labrador
NPC: Net Present Cost
NRCan: Natural Resources Canada
NRC: National Research Council of Canada
NREL: National Renewable Energy Laboratory
NPV: Net Present Value
NZ: New Zealand
OGO: Oil Gallons Offset
OSB: Oriented Strand Board insulation
PBP: Payback Period
PBSS: Packed Bed Storage System
PCM: Phase Change Material
PEF: Primary Energy Factor
PH: Power of Hydrogen
PV: Photovoltaic
PVT: Photovoltaic Thermal Collectors
RECS: Residential Energy Consumption Survey
ROI: Return On Investment
SHG: Solar Heat Gain
SHTES: Sensible Heat Thermal Energy Storage
SRCC: Solar Rating and Certification Corporation
TES: Thermal Energy Storage
TOU: Time of Use pricing
TRNSYS: Transient System Simulation Tool
WT: Wind Turbine
XPS: Extruded Polystyrene insulation

Research Highlights

- A novel approach to calculating combi systems solar fraction is introduced, justified and supported by f-chart calculations and Polysun simulation results.
- A solar thermal combi system is found to be economically superior to a wind-PV hybrid system used for electric heating.
- PVT system has better area density than PV only system and combi solar thermal system. It also has a better value of energy generation but higher LCOE.
- A south facing SolarWall produces less energy than a roof mounted system of similar size but produces more energy if the added insulation (heat recapture) provided as the result of SolarWall is taken into account. The amount of heat recapture is found superior to an R10 insulation upgrade in the walls' sheathing.
- Air leakage reduction is the most economical residential energy efficiency measure while wall insulation upgrade is the least.
- A smart appliance with demand response is found less economical than an energy efficient device (under TOU pricing) despite offering greater consumption reduction due to the higher price point.

I. Overview

Project A carried out the load profiling and design of Photovoltaic (PV) and wind energy systems for a typical household in St. John's to counteract the financial burdens of Muskrat Falls. Four scenarios were generated and analyzed. Scenarios A, B and D dealt with the electrical load while scenario C dealt with the thermal load. It was found that all electric load scenarios were feasible while the thermal scenario was not, mainly due to the affordability of heating oil. However, Newfoundland's recent commitment to zero net emissions by 2050 suggests heating oil's long-term unsuitability and unsustainability as one of the province's thermal energy sources due to high Greenhouse gas (GHG) emissions. Therefore, Project B will deal with the design of a solar thermal system coupled with thermal energy storage (TES). The resulting scenarios will be compared against scenario C and insights regarding the most feasible and reliable thermal energy system will be derived. Next a much-needed energy retrofit will be carried out to examine the extent of load reduction given the upgrade to efficient modern technology. The main Software that will be used is BEOPT, Excel, RETScreen and Polysun. Optional software includes Mathcad, Swift, SAM, MATLAB, Tsol and Google Earth.

TABLE I
PROJECT OVERVIEW

Project Scope	Project Scope Type
• Design of solar thermal and thermal energy storage systems	• Detailed analysis of a challenging engineering problem with proposed solutions and recommendations.
Project Deliverables	**Assessment of Resources Needed for the Project**
• "Manual" calculations for solar thermal system in Mathcad or Excel. • Design of thermal solar system and storage in Polysun. • Economic analysis of various energy generation scenarios. • Recommendations as to the most feasible options regarding household energy generation and storage.	• Polysun: Trial version is available for 30 days (will require renewal). • HOMER Pro: Accessible through university computers (but not active). • BEOPT: Free software. • Excel or Mathcad: Already obtained. • Swift, MATLAB (optional): Obtainable. • RETScreen: Free software but can't save files unless a subscription is purchased.

A. Project Gantt Chart

This section shows the division of tasks to be conducted over this project in order to complete it successfully. It clearly shows task name, allotted time, milestones, task level, task progress and important dates. There are 7 main tasks, 8 sub tasks and 7 sub-sub tasks.

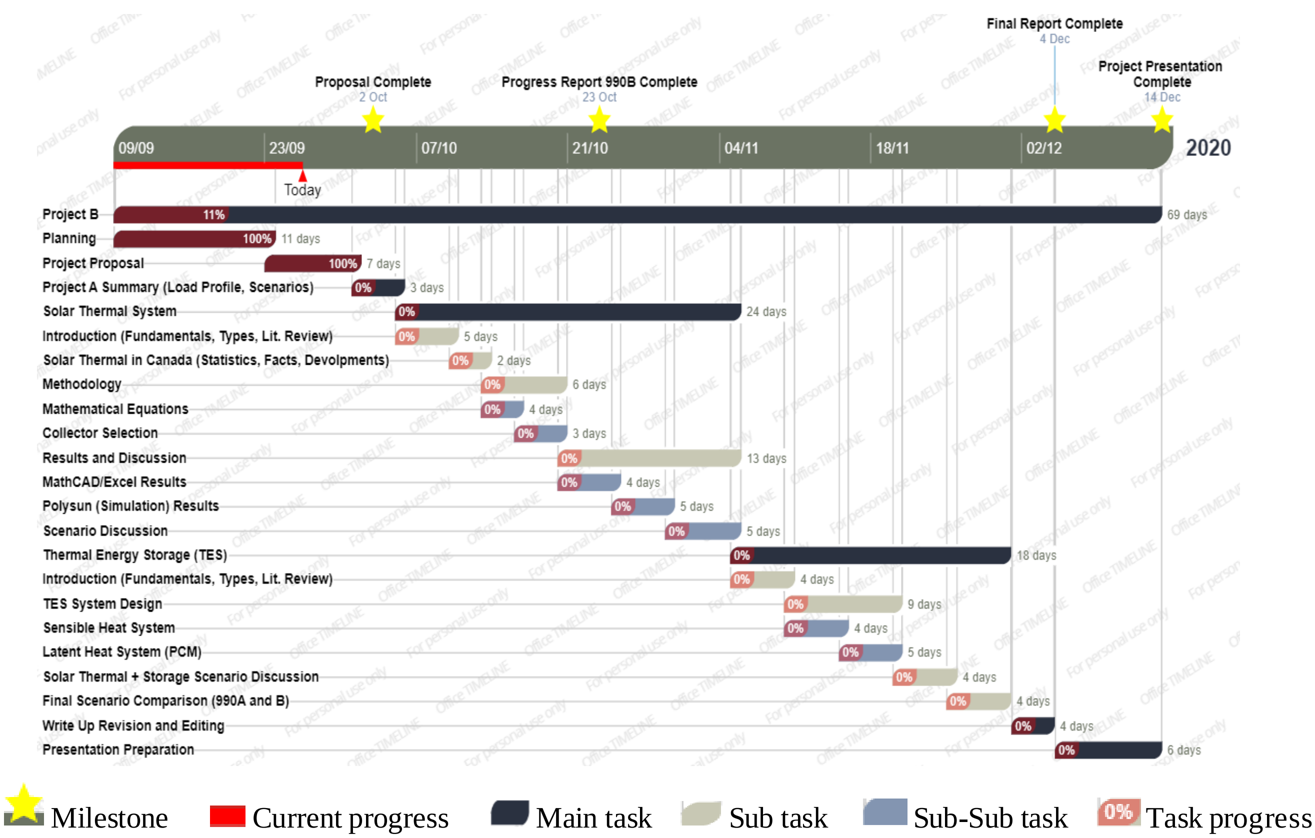


Fig. 1. Project Gantt Chart.

B. Problem Statement

1) Overall Project

As a consequence of the Muskrat Falls project, Nalcor energy needs to raise $725.9 million annually in order to stabilize the electricity price in Newfoundland at 13.5 cents/kWh, otherwise, the price is forecasted to increase to 22.9 cents/kWh which is almost double the current rate of 12.3 cents/kWh [1,2]. Therefore, this project's motivation is, in case the government's price stabilization plan fails and the price rises, some St. John's residents might want to go off-grid or at least to produce a portion of their energy from residential renewable sources. Therefore, this project will examine the design of such a system. Multiple scenarios will be created one of them being the highest energy diversity scenario which offers greater energy security than simply relying on a single source and another being the least-cost scenario which is compared against the worst-case scenario electricity rate (22.9 cents/kWh) and a conclusion can be made whether going off-grid at that time or generating a portion of energy from household renewable sources is feasible or not.

Newfoundlanders are switching back to oil-based heating since the island's residents are worried about the price of electricity due to Muskrat Falls [3]. According to the government of Newfoundland, The consumption of heating oil in the province in 2015 was 98 GJ/household which is an approximately 10 GJ increase from the 2013 figure while household electricity consumption decreased from 65.5 GJ/household in 2013 to 64.3 GJ/household in 2015 highlighting the popularity of heating oil for water and space heating [4]. Heating oil is a petroleum product and thus is environmentally damaging and will eventually be depleted (fossil fuel's bell curve). 20% of all environmentally damaging oil spills in Newfoundland are from domestic heating oil which contaminates the soil and is hazardous to humans [5]. Therefore, this project will study the replacement of oil-based house heating system with a solar thermal system(s).

2) Project-B Update

On the 21st of May 2020, Premier Dwight Ball wrote a letter to PM Justin Trudeau in which he announced Newfoundland's commitment to reaching net zero carbon emissions by 2050 [6]. The 2050 commitment, set by various governments, was catalyzed by a 2018 report from the United Nations Intergovernmental Panel on Climate Change, which said CO_2 emissions had to be brought down to zero by 2050 if the world was going to try to limit global warming to 1.5 °C and avoid catastrophic impacts [7]. Both Newfoundland and Labrador Environmental Industry Association (NEIA) [8] and Newfoundland and Labrador Oil and Gas Industries Association (NOIA) [9] have voiced their approval.

In the letter, Ball expressed the effect of COVID-19 on the oil and gas industry, which in the 7-year period between 2010 and 2017 accounted for 30% of the province's gross domestic product (GDP), was ubiquitous. During the COVID-19 pandemic, the Premier highlighted, foreign oil and gas supply went into over production while global demand dwindled. This poses a serious economic problem for Newfoundland to the tune of $61 billion from now to 2038 in terms of provincial GDP. For these reasons, all new oil and gas projects and exploratory activities are now halted [7]. While these losses are not insignificant, they present an opportunity for the province to incorporate more renewables and divest from fossil fuel production as the province strives to achieve its new targets.

Newfoundland and Labrador is in the process of developing its "Climate Change Action Plan" [11]. This five-year plan is aimed at reducing GHG emissions and simulating clean energy innovation and growth. The plan involves an $89.4 million investment in the federal low carbon economy leadership fund and $300 million investment for green infrastructure through the federal investing in Canada plan. On the microscale, the province has announced 5 programs to promote energy efficiency and switching off of heating oil for domestic water and space heating through retrofits (fuel switching). It is expected that by 2030 these programs would deliver 882,000 tonnes of cumulative GHG reductions [11,12]. On a regional level, Atlantic Canada is jointly attempting to reduce its GHG levels and increase deployment of renewables as evidenced by the collective letter written to PM Trudeau . The letter states that New Brunswick is investing in small modular nuclear technology, Nova Scotia is developing its marine renewables, Prince Edward Island is investing in energy efficiency and conservation programs and Newfoundland and Labrador has committed to achieving net-zero emissions by 2050 [10].

Therefore, this project will design a solar thermal and energy storage system(s) attempting to replace the existing heating oil system at a typical St. John's residence (assuming the success of/ adherence to NL net zero commitment). It is expected that useful insights will be derived as a result of this work that can contribute towards provincial plans.

3) Research Questions

In the last project it was found that wind and solar PV are not able to compete economically with heating oil for providing the thermal load. But the new information presented here alters the original research question

- **From**: Can renewables supply the thermal load at a price that is competitive with heating oil?
- **To**: Which renewable energy/combination of renewable energies is the most suitable for feasibly and reliably providing the thermal load?

II. Project-A summary

A. Load profile

In Project A, the load was studied using BEOPT. In this section some important information that is relevant to this study will be represented.

From project A, the following facts were reported. The house spends 2799 CAD on heating oil annually while 1266 CAD is spent on electricity. It consumes 999.6 gallons of oil per year which is equivalent to 165.4 MMBtu/year or 48,474 kWh/year of source energy used for heating. On the other hand, 10,727 kWh/year of electricity were consumed. Scaled daily electrical load is 29.4 kWh/day. CO_2 equivalent emissions are around 12.4 metric tonnes per year from the heating system. Energy intensity (114.8 kWh/m^{2}year) is close to the desired range set by Natural Resources Canada (NRCan) and Clean Technology Scenario (CTS) [12]. Table II and figure 2 show the results of the simulation. Table III shows the options values inputted into BEOPT which were provided by the owner of the design house.

TABLE II
LOAD INFORMATION

Parameter	Value
Electricity consumption	10,727 kWh/year
Source energy use (total)	280.7 MMBtu/year = 82,265.05 kWh/year
Source energy use (heating)	165.4 MMBtu/year = 48,473.96 kWh/year
Source energy use (space heating)	134.8 MMBtu/year = 39,505.98 kWh/year
Site energy use	175.5 MMBtu/year = 51,433.97 kWh/year
Utility bills	4161 CAD/year
Oil use	999.6 gal/year
Delivered energy	117.1 MMBTU/year = 34,318.62 kWh/year
Heating capacity	41.6 kBtu/hr
Area	1852 ft^2/level = 172 sqm/level = 344.12 m^2
Energy intensity (Source energy use)	239.059 kWh/m^2 year = 0.86 GJ/m^2 year
Energy intensity (Site energy use)	149.46 kWh/m^2 year = 0.538 GJ/m^2 year
Energy intensity (Heating energy)	140.86 kWh/m^2 year = 0.5071 GJ/m^2 year
Energy intensity (Space heating energy)	114.8 kWh/m^2 year = 0.41328 GJ/m^2 year
Energy intensity (Delivered energy)	728 h/sqm year

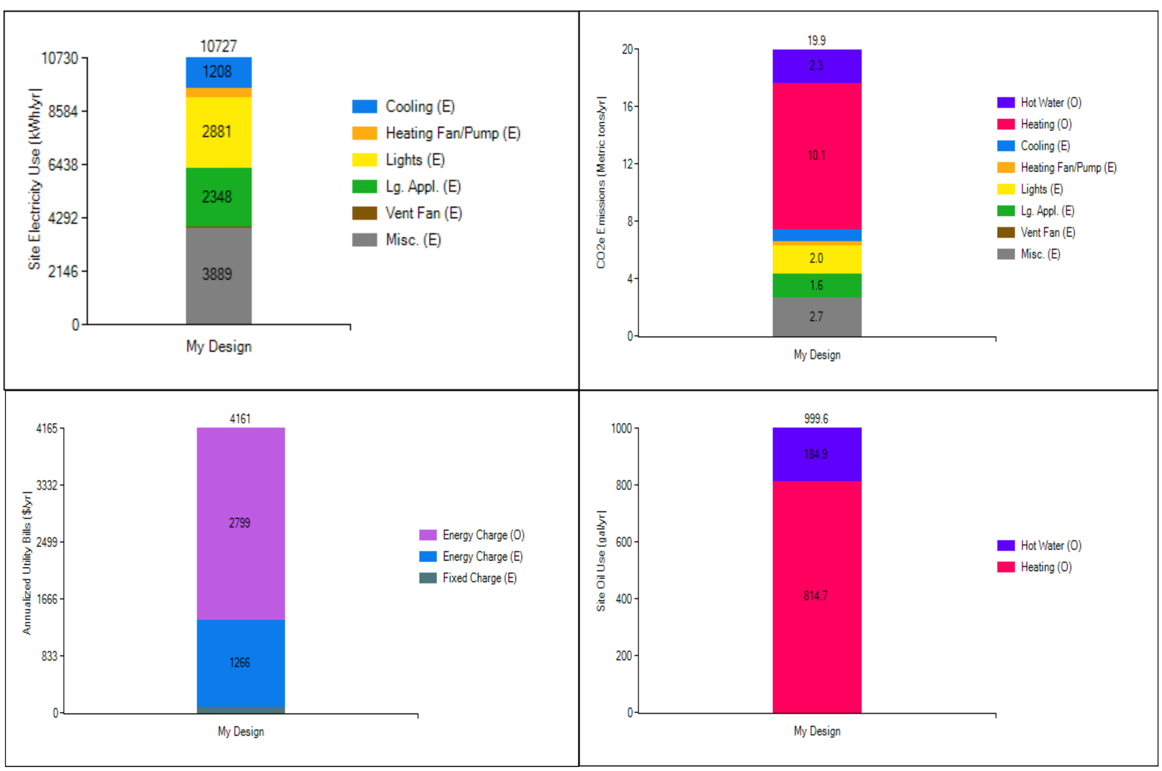


Fig. 2. BEOPT results.

TABLE III
BEOPT OPTIONS

Option	Value	Option	Value
Wood Stud	2x4 fiberglass r-13	Central Air Conditioner	None
Wall Sheathing	R-10 XPS	Boiler	Oil 80%
Exterior Finish	Vinyl, light	Ducts	15% Leakage Uninsulated
Unfinished Attic	R-30 fiberglass vented	Ceiling Fan	None
Roof Material	Asphalt dark	Dehumidifier	Standalone
Radiant Barrier	None	Cooling Set Point	None
Finished Basement	Whole Wall R-10 XPS	Humidity Set Point	45%
Carpet	0%	Heating Set Point	71 F setback 65 F
Floor Mass	Wood Surface	Cooling Set Point	None
Exterior Wall Mass	5/8 Drywall	Water Heater	50 gal Oil Standard
Partition Wall	None	Solar Water heating	None
Ceiling Mass	5/8-inch Drywall	Refrigerator	Top freezer
Windows	Double non-metal	Clothes Washer	EnergyStar
Door Area	20 ft^2	Clothes Dryer	Electric
Doors	Wood	Extra Refrigerator	Top Freezer
Eaves	3 ft	Freezer	Chest EF=24
Overhangs	None	PV system	None
Mechanical Ventilation	None	Natural ventilation	3 days/week

1) Thermal load

From D-View of BEOPT, the hourly site and source thermal energy consumption can be obtained. However, while source energy is in BTU and therefore easily convertible to kWh, site energy is in gallons of oil. To convert site energy to kWh, the BTU content of a gallon of heating oil was obtained from the North American Combustion Handbook [39] and the U.S. Energy Information Agency (EIA) [40] which show that heating oil produces 138,500 British thermal units per US gallon. Then, conversion from BTU to kWh is done where 1 BTU is equivalent to 0.000293071 kWh. The annual thermal energy consumption is 40.515 MWh with a scaled annual average of 111 kWh/day, peak power of 27.1 kW and average power of 4.63 kW (obtained by inputting thermal load time series into HOMER).

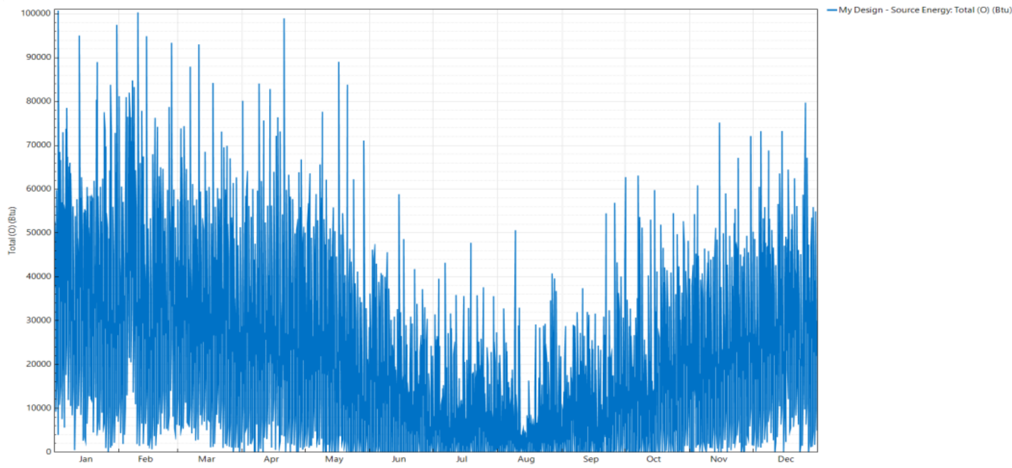


Fig. 3. Hourly thermal load.

B. **Results**

Table IV summarizes the results of the 4 generation scenarios explored in project A.

TABLE IV
PROJECT A RESULTS

Scenario	Technology Used	Load Covered	Energy Generated (kWh/year)	Excess Electricity and Unmet load	LCOE (CAD/kWh)	Payback Period (years)	Return on Investment
Scenario A	31 PV modules	Electrical	10,614	0% and 0%	0.189	10.7	125%
Scenario B	1 WT* and 8 PV modules	Electrical	10,613	0.06% and 0%	0.186	10.44	139%
Scenario C	1 WT** and 14 PV panels	Thermal	40,684	0% and 0%	0.162	25.8	0%
Scenario D	2 WTs*** and 8 PV panels	Electrical	10,853	1.14% and 0%	0.194	10.68	134%

Note: * Skystream 3.7 WT, ** Bergery E10 WT, *** DS3000 WT, PV modules = CS6K-305MS PV modules

III. Literature review

A. Introduction

Visible light (insolation) is the primary energy source utilized by solar thermal and PV systems for space and water heating and electricity. The amount of solar insolation incident on the earth varies throughout the year due to earth's axial tilt and rotation. It also varies daily from sunrise and sunset (altitude angle) and due to atmospheric conditions (Air Mass and clouds). Seasonal variations are marked by the spring and fall equinoxes (March and September 21st) and summer and winter solstices (June and December 21st for locations north of the equator). Equinoxes occur when the sun crosses the equator and solstices when the sun reaches its highest/lowest latitude. During summer solstice, daylight hours are the longest and during winter solstice they are the shortest [51]. For this project, hourly solar insolation time series is obtained from HOMER software for the location of the building studied (Latitude = 47.551, longitude = -52.712). Solar resource data can alternatively be obtained from PVWatts.

Solar water heating systems use both direct and diffuse (reflected) solar radiation. Generally, if the site of the installation is unshaded from 9 am to 3 pm and south facing it is suitable for solar thermal systems siting. From project A, it was found that the best tilt angle for solar PV is 40° which is 7.5° less that the latitude angle representing 0.79% increase in energy over the system angled at tilt equals latitude. This is in agreement with a 2017 review paper which reported that the optimal tilt for Ottawa is latitude angle – 7° and for Toronto latitude angle – 12° [50]. The optimum orientation for a solar collector in the northern hemisphere is facing true south or azimuth = 0° or 180° depending on convention. Optimum azimuth angle for Toronto was also reported as between 1° west of south and 2° east of south (mostly south facing). However, according to the U.S department of energy (DOE), the collector can face up to 90° east or west of true south without suffering from any significant decrease in performance [51,52]. Since the tilt of the roof in the current project is not easily obtainable the roof is assumed to be flat (similar to project A) also no objections to aesthetic value is assumed and no shading from nearby obstructions is assumed.

To introduce information related to solar thermal systems, first solar thermal knowledge is extracted from a variety of sources using two organisation devices. One is system classification and the other is system components. Note: since collector is covered adequately in the classification discussion it is not repeated in the component section.

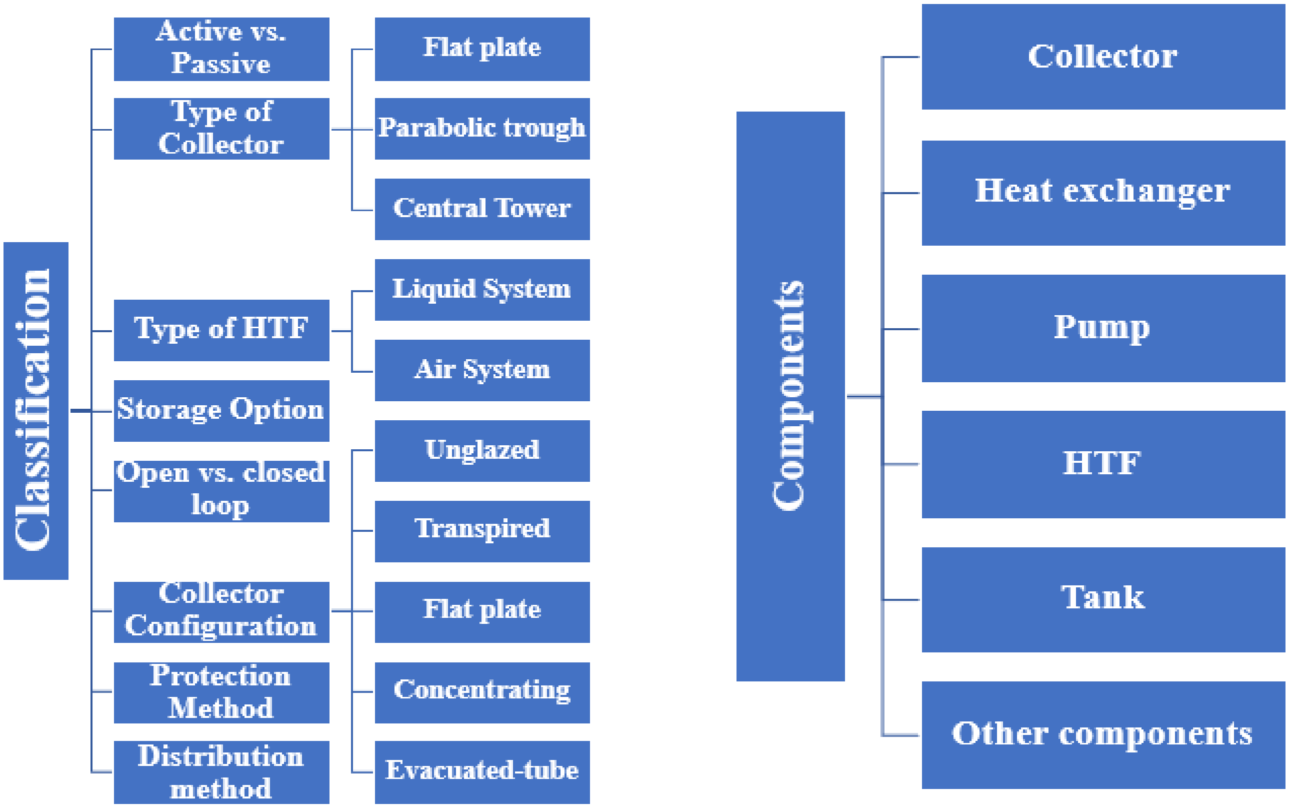


Fig. 4. Classification and component organizational structure.

1) *Classification*

i. *Active vs Passive*

Active solar systems (the main focus of this work): involve the harvesting of thermal energy by using solar collectors and employing active mechanical components, such as pumps or fans, to circulate the heat transfer fluid (HTF) (which can be liquid or air). If the solar system cannot provide the thermal load, then an auxiliary system is used to provide the remaining heat. If storage is included, then liquid systems are mostly used. Active solar systems are most cost effective in cold climates with good solar resources. It greatly reduces electric bills in the winter and is best used for both space and water heating. It is most economical to design an active system to provide 40% to 80% of the home's heating needs. A solar system that supplies enough heat 100% of the time is not practical or cost effective. [25,29].

Passive solar systems are usually associated with the built environment such as natural lighting or a thermal mass that stored heat such as Trombe wall. These systems do not use any active mechanical devices and instead rely on the distribution of warmth from natural heat flow. The three main aspects of passive solar heating systems are a south facing window, a thermal storage mass (walls or floors) and good insulation. They are usually categorized as direct and indirect gain (depending on utilization of a thermal mass between the sun and the interior space) and isolated gain (if the heated space is separate from the main house such as sunspaces). A well-designed and insulated home that incorporates passive solar heating techniques will require a smaller and less costly heating system of any type and may need very little supplemental heat other than solar. [25,28,29].

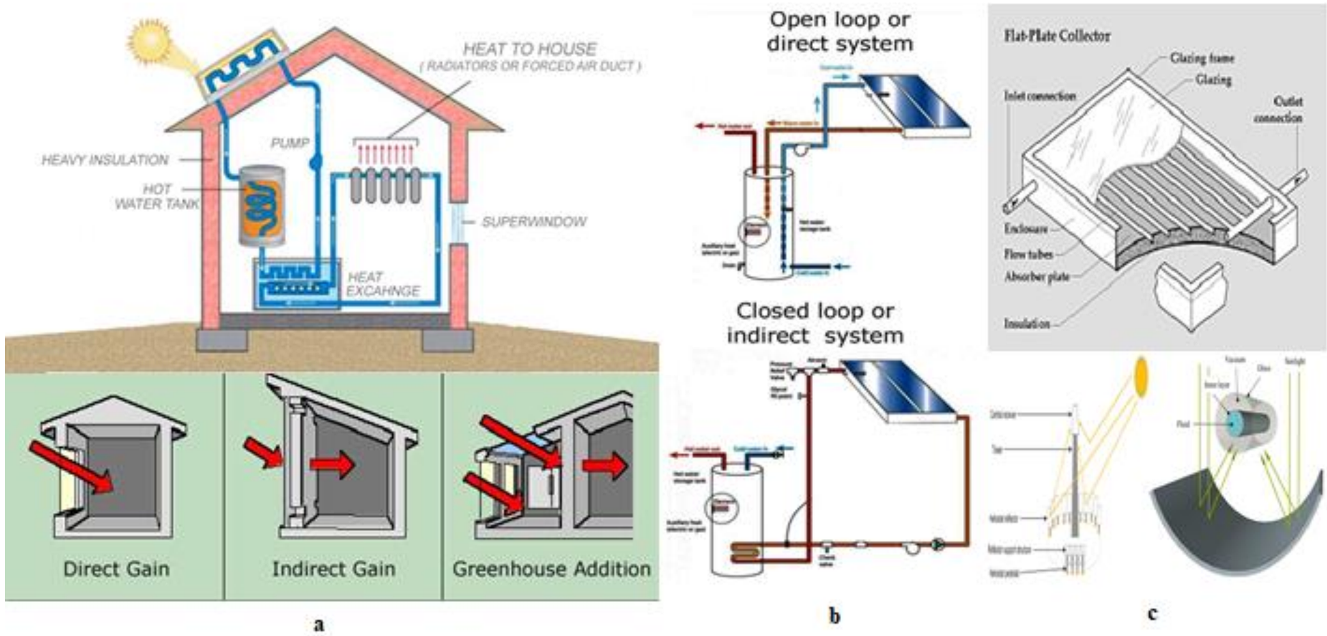


Fig. 5. Classification. a) active vs passive, b) open vs closed loop, c) collector type [25,28].

Passive solar water heaters are less expensive than active ones but not as efficient. They rely on gravity and the natural circulation of water when heated. As they contain no electrical components, they are generally more reliable, have longer lifespans and require less maintenance. There are two main types of passive solar heaters integral collector storage system and thermosyphon systems [30].

In integral collector storage system (ICS), the collector and storage tank are combined where one or more storage tanks are placed in an insulated box with the collector facing the sun. They are similar to flat plate collectors but with heat tubes of much bigger diameters (100-200 mm). They are open loop domestic hot water (DHW) systems. They use natural convection and do not require any mechanical components. These systems are best suited for places where freezing temperatures rarely occur. They do not work well if the thermal load is primarily in the morning as they lose most of their accumulated energy overnight [30,41]. In a 2020 study an ICS system for DHW production was designed using phase change materials (PCM) as energy storage (latent heat). The PCM maximum temperature was 79.3 °C, volume 0.02 m^3 and had a storage capacity of 24.57 kWh of thermal energy per month. The house's DHW load of 200 L/day was supplied at a solar fraction of 56% [43].

Thermosyphon systems rely on the warm water's natural convection as it rises to circulate through the collectors and to the tank (which is placed above the collector). As water in the collector heats up it becomes less heavy and rises up naturally to the above placed tank. While the cooler water flows downwards to the bottom of the collector. This improves circulation. In freeze prone locations a form of closed loop (indirect gain) thermosyphon systems can be installed that utilize glycol in the collector loop [30]. In a recent study, a thermosyphon system that uses CO_2 refrigerant was designed for sub-zero temperature locations. For ambient temperatures around 30 °C CO_2 reaches a temperature of 75 °C at a collector efficiency of 85%. The CO_2 then passes through a heat exchanger (HX) and increases the inlet water temperature from 26 °C to 55 °C [42].

ii. ***Open Vs closed loop***

Open loop active systems (direct systems) circulate water directly from the tank through the collectors. This design is efficient and has lower operating costs but is not appropriate if the water supply is hard, because calcium deposits quickly build up in the collector, or acidic. It cannot be used if outside temperature drops below zero. These systems are not approved by the Solar Rating and Certification Corporation (SRCC) if they use recirculation freeze protection (circulation of warm tank water during freezing conditions) as this requires electrical power to be effective [30].

In a closed loop system (indirect systems), the heat-transfer fluid is pumped through the collectors and a heat exchanger is used to transfer heat from the collector loop to the water in the tank. Closed loop glycol systems are popular in areas subject to extended freezing temperatures as they offer good freeze protection. Some of these systems offer overheat protection. Overheating occurs when solar intensity is high, but the load is low resulting in superheating of the collector or the glycol [30].

iii. ***Type of collector***

The two most common solar collectors are flat plat collectors and parabolic trough collectors. Flat plate collectors are generally utilised for space and water heating in residential and commercial buildings and are usually mounted at a fixed angle. In contrast, parabolic trough collectors often possess the ability to track the altitude angle of the sun thereby focusing the sun's radiation along a pipe that carries the heat transfer fluid and is able to achieve much higher temperature (400 ℃) than flat plat collectors. They are usually used in industrial and commercial facilities and not by residences. Because of the lack of a tracking system, flat plate collectors are usually cheaper and require no maintenance. Tracking is available but it increases cost. Fixed angle collectors are usually set at zero azimuth angle and altitude equal latitude of location angle. Another type of active solar systems is the central receiver tower system. Which uses a field of heliostats to focus the suns rays on a single point usually positioned atop of a centrally placed tower. This system is able to achieve much higher temperatures than the previous two types (565 ℃) and is mainly used for utility power generation purposes [25].

The last type is dish sterling solar system. Here a dish shaped condenser focuses light onto the receiver of a sterling engine at a concentration ratio of over 2000 reaching temperatures up to 750 ℃ [49]. In [44], an altitude-azimuth concentrated tracking system that is self powered was developed. The system uses Fresnel lens to concentrate sunlight on a sterling engine (which can reach 900 ℃), while PV panels were used to provide power for the tracking motors.

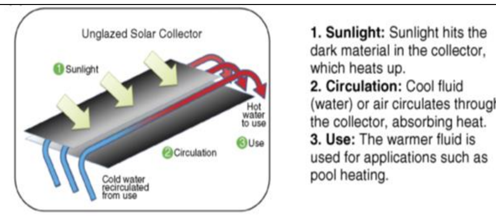
iv. ***Collector configuration***

There are 5 main collector configurations: unglazed, transpired, flat plate, evacuated tube and concentrating collectors. The following table explains the mechanism of each type [25].

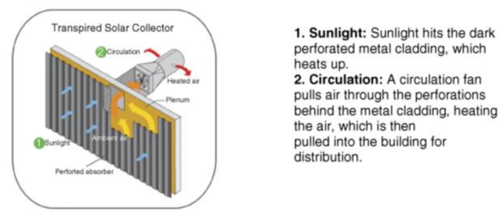
TABLE V
COLLECTOR CONFIGURATION

Type and mechanism	Diagram (explanation)

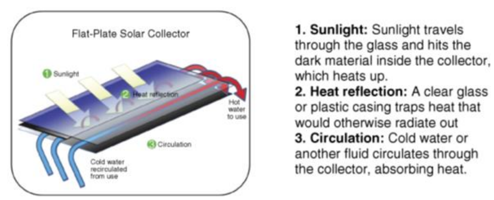
Unglazed collector: simplest collector. Has a receiver plate with high solar absorptivity through which water flows to act as the HTF (it can be an air or liquid system).



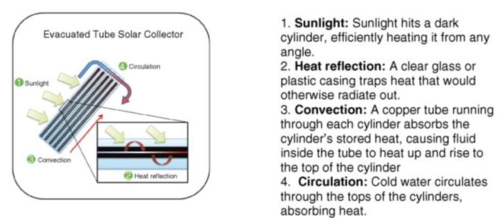
Transpired collector: simple collector where a receiver plate absorbs solar energy, and a fan is used to circulate air through a plenum in the back of the receiver. Air is warmed and used for space heating (air system). In a 2019 study, transpired collector was used to preheat the air for an air-to-air heat pump. It was found that combining a 2.5 kW heat pump with a 2.5 m^2 collector resulted in a 10% decrease in the pump's electricity consumption thus increasing the coefficient of performance (COP) of the heat pump especially during winter [45].



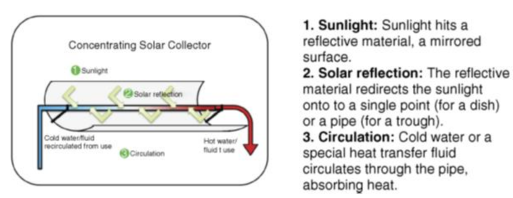
Flat plate collector: similar to unglazed collector but it has glazing. Glazing is a transparent cover with high solar transmissivity and low transmissivity of long wavelength heat waves to keep thermal radiation trapped inside the collector. (can be an air or liquid)



Evacuated-tube collector (ETC): an evacuated cylinder with high solar absorptivity is used to absorb solar radiation. Inside the cylinder there is a copper tube through which water flows (liquid system). A 2020 study combined an ETC with nano enhanced PCM by immersing it in a copper PCM composite which increased the ETC's efficiency by 32%, and provided 50 °C DHW at a mass flow rate of 0.08 L/min for 2 hours longer than a typical ETC can [46].



Concentrating collector: trough or parabolic or dish shaped mirrors are used to focus light on a central pipe or point where the HTF flows. (liquid system)



From figure 6, Unglazed and transpired air collectors are used for temperature applications of 100 °F or less such as pool and space heating. Flat plate collectors are suitable for space, domestic water and pool heating where temperatures for this collector can reach 150 °F. Evacuated tube and Concentrating collectors can be used for pool, space, water heating and industrial process heat as temperatures can reach 400 °F.

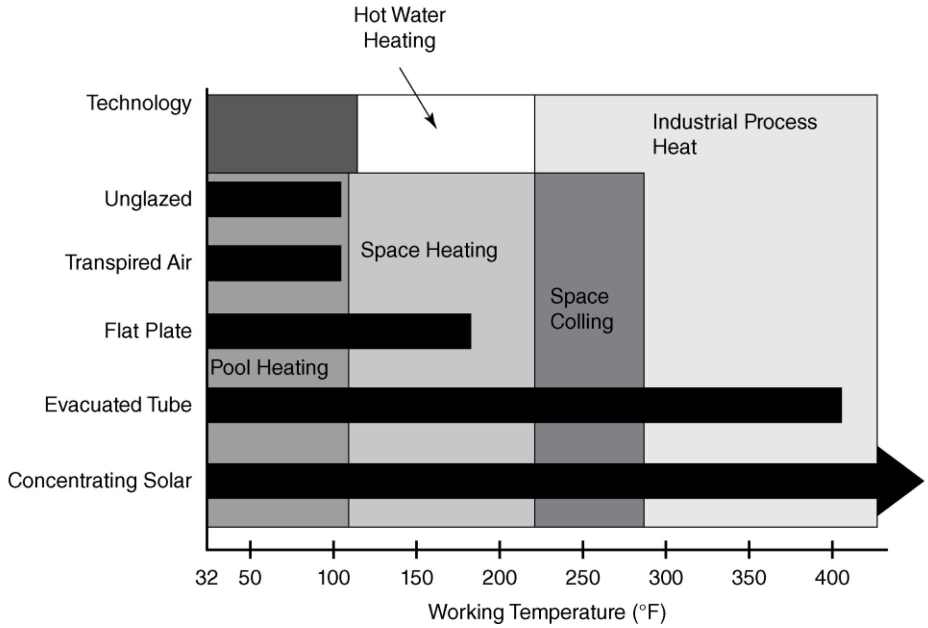


Fig. 6. Collector configuration applications [25].

v. ***Type of heat transfer fluid***

a) Liquid based collectors

Liquid based active collectors are mostly used for central heating and domestic water heating. Flat plate collectors are mostly used and to a lesser extent evacuated tube and concentrating collectors are also used. In the collector, a HTF such as water or antifreeze (propylene glycol) absorbs the solar heat. It is circulated through the collector by a controller operated circulating pump. As the liquid's flow is fast, its temperature only increases by 5.6 ℃ to 11 ℃ as it traverses the collector. This is preferred to heating of a small volume of fluid to a higher temperature which increases heat losses from the collector and reduces overall efficiency. The heated liquid flows to either a heat exchanger for immediate use or to a storage tank. System components of liquid systems include collector, HTF, piping, pumps, valves, expansion tank, heat exchanger, storage tank and a control system [29].

b) Air based collectors

Solar air collectors can directly heat individual rooms or preheat air passing into a heat recovery ventilator or through the air coil of an air source heat pump. Air collectors produce heat earlier and later in the day than a liquid system so they might produce more usable energy over a heating period. Air systems also do not freeze and are more leakage resistant. However, air is less efficient in heat transfer than liquid resulting in generally lower efficiencies. Rock beds can be used as energy storage, but this is not recommended due to the formation of mold and moisture and its effect on indoor air quality. Air systems can be divided into room air heaters or transpired air collectors [29].

In room air heaters, air collectors can be installed on a roof or an exterior of a south facing wall. The collector has an insulated airtight metal frame and a black metal plate for absorbing heat with glazing in front of it. Solar radiation heats the plate which in turn heats the air in the collector. An electric fan then pulls air from the room through the collector and blows it back into the room. Roof mounted collectors require ducts to carry the air between the room and collector while wall mounted collectors do not. Alternatively, window box collectors can be used which fit into an existing window opening. They are either active (with a fan) or passive. In the passive system colder air enters the bottom of the collector and rises as it gets heated and enters the room. A damper or baffle keeps the room air from flowing back into the panel (reverse thermosyphon). These systems however only provide a small amount of heat due to smaller collector area [29].

In transpired air collector, the collectors consist of dark perforated metal plates installed over a building's south facing wall. A gap is created between the old wall and the new facade. The black outer façade absorbs solar energy and rapidly heats up even when outside air is cold. A fan is used to draw ventilation air into the building. The collector warms the air by as much as 4.4 ℃ and glazing is not required. Transpired collectors are more suited for large buildings with high ventilation loads and less so for tightly sealed homes [29].

vi. ***Storage option (for liquid systems)***

Solar heat is stored in water tanks or in a masonry mass of a radiant slab system (sensible heat) or PCMs (latent heat). In the tank type, heat from the working fluid transfers to a distribution fluid in a heat exchanger within or outside of the tank. Tanks can be pressurized or unpressurized depending on overall design. Factors that affect choice of storage tank are cost, size durability and placement considerations. Tanks must also meet local regulatory codes. Insulation is also needed to prevent excessive heat loss while protective coating and sealing is needed to avoid corrosion and leaks. Tanks are usually made from stainless steel, fiber glass or high temperature plastic. It is best to use standard domestic water heaters as they meet all the above requirements [29].

vii. ***Distribution method (liquid systems)***

To distribute the solar heat, radiant floors, hot water radiators or central forced air system can be used. In radiant floor systems, the hot fluid circulates through pipes embedded in a thick concrete slab floor which then radiates heat to the room. It is best suited for solar heating as it can operate at low temperatures. A conventional boiler or standard domestic water heater can supply the back up heat required. Radiant slab systems, on the other hand, take a longer time to heat the room from a cold start but provide consistent heat once they are operating. Hot-water baseboards or radiators can require water temperatures between 71 ℃ and 82 ℃ to effectively heat a room which is generally incompatible with a solar collector which heat the HTF between 32 ℃ and 49 ℃. Therefore coupling a radiator with solar heating system requires the surface area of the radiator to be larger, the temperature of solar heated liquid to be increased by an auxiliary system or an evacuated tube collector be used instead of a flat plate collector as it offers higher temperatures. To use a central forced air heating system when the HTF is liquid then a liquid to air heat exchanger needs to be used where air is heated before it reaches the room as it passes over the solar heated liquid in the heat exchanger. Additional heating is done by the furnace as necessary [29].

viii. ***Protection method***

Freeze protection methods [45] prevent damage to a solar water heating system due to the expansion of freezing water. SRCC publishes a Freeze Tolerance Limit (FTL) for each OG-300 certified system which is specified by the manufacturer/supplier of the system. This is derived from the temperature at which the system is anticipated to withstand 18 hours of constant exposure. This value, however, is not validated through independent testing and does not assure total freeze protection under all conditions. The freeze protection methods commonly used in OG-300 certified systems include:

- Antifreeze Fluid: Usually propylene glycol with inhibitor and buffer chemicals added. The fluid must be checked periodically to verify that it still provides protection, since some fluids can break down over time.
- Drainback Tank: Piping from collectors in unconditioned space is sloped toward a drainback tank installed within conditioned space. When the pump stops running (e.g. at sunset or when the tank has reached a high temperature limit), the fluid in the collector drains into a drainback tank protecting the fluid from freezing or overheating. Water or a water/glycol mixture can be used in the collector loop.
- Direct Forced Recirculation: When the collector temperature approaches freezing, the system controller turns the pump on to circulate warm water from other parts of the system to the collector. Viability depends on availability of power, system responsiveness and the quality of the potable water. Such a system will repeatedly circulate warm water to the collector as long as freezing temperatures are detected.
- Freeze Valves: Temperature-actuated automatic flush valves, known as freeze valves may be used to flush cold fluid in the collector from the system whenever near-freezing conditions exist. Dribbling water from the system through the freeze valve causes warmer water to flow through the collector. Viability depends on water quality, valve maintenance and correct installation. Water drained must be routed and disposed of appropriately.
- Thermal Mass: The large volume of water in an integrated collector storage (ICS) collector takes a long time to freeze. This protection type is effective down to a specified FTL, which should be compared with local climate conditions. Note that piping to and from the collector are still subject to freezing. Freeze valves may be added to such a system to further extend the freeze resistance.
- Frost Plugs: are devices that can be installed on ports in a collector that are expelled in whole or in part when pressures rise in the collector as a result of freezing. The resulting pressure relief can prevent collector and piping damage. However, after the water thaws, considerable water loss can occur. The frost plug may need to be reset or replaced periodically.

System overheating can degrade heat transfer fluids, accelerate scaling, cause premature component failure, and reduce system performance. The maximum design temperature for each system will vary depending on the materials, overheat protection methods and operational modes selected by the manufacturer. OG-300 certified systems have been reviewed by ICC-SRCC to ensure they can operate within the design pressure and temperature limits specified by the manufacturer/supplier. Some system designs rely upon consistent hot water usage, grid power, storage size or user intervention to prevent damage due to system overheating. Overheat protection methods include: Drainback Tanks, Hot Dump Radiators, Vented Collector, Steam-Back Inhibitor, Pump Cycling and Pressure Stagnation [45].

2) Component

i. Heat exchanger

In closed loop systems, heat exchangers are used to transfer the solar energy harvested by the HTF to the air or liquid responsible for space or water heating. Solar heat exchangers are usually made from copper as it is a good thermal conductor while being resistant to corrosion. There are 2 types of heat exchangers liquid-to-liquid (L2L) and air-to-liquid systems (A2L) [30].

In liquid-to-liquid heat exchangers, HTF absorbs heat from collector then flows through a heat exchanger in the storage unit and transfers its heat to water. HTF can have antifreeze mixed in to protect the collector from freezing in the winter. L2L heat exchangers have either a single wall or a double wall between HTF and water. Single wall heat exchanger is a pipe surrounded by a fluid. HTF can be inside the pipe or the surrounding fluid while the other fluid is the usable water. In double wall heat exchanger, two walls exist between the 2 fluids. This is often used when the HTF is toxic such as ethylene glycol. This is also required in case of leakage prone systems so that the antifreeze does not mix with the usable water. Some local codes require solar water heaters to use double wall systems due to higher safety however they are less efficient due to heat transfer taking place between two surfaces. In air-to-liquid heat exchanger, solar heaters with air as the HTF usually don't require a heat exchanger between the collector and the air distribution system. If the solar air system is designed to heat water once the space heating requirement is satisfied, then an A2L heat exchanger is required. Examples of heat exchanger designs are coil in tank HX, shell and tube HX and tube and tube HX [30].

ii. Heat transfer fluid

The following criteria are used for HTF selection: Coefficient of expansion is the fractional change in length or volume of a material for a unit change in temperature, Viscosity is the resistance of a liquid to sheer forces, Thermal capacity is the ability of HTF to store heat, Freezing point is the temperature when liquid becomes solid, Boiling point is the temperature where liquid boils and Flash point is the lowest temperature at which the vapor can be ignited in air. In cold climates, the HTF should have a low freezing point and in desert climate a high boiling point. Viscosity and thermal capacity determine the amount of pumping energy required. Low viscosity and high thermal capacity mean easier pumping [30].

a) Types of Heat transfer fluids

The four main HTF types for solar collectors are air, water, glycol/water mix and refrigerants. Air does not freeze or boil and is noncorrosive, but it has a very low heat capacity and tends to leak. Water is nontoxic and cheap with high specific heat and low viscosity (so easy to pump) but low boiling point and high freezing point. It can become corrosive if the pH levels are not maintained. "hard" water can cause mineral deposits in the collector tubing and plumbing. Glycol/water are mixed at 50/50 or 60/40 glycol to water ratios. Glycol is an antifreeze with propylene glycol being the most commonly used HTF in closed loop solar water heaters. Ethylene glycol is highly toxic and must only be used with double wall closed loop systems. Propylene glycol on the other hand can be safely used with single walled HX and is nontoxic. Annual inspection of the HTF pH levels, freezing point and concentration of inhibitors is required. Refrigerants/phase change fluids are commonly used HTFs in refrigerators and Air conditioners (ACs). They have low boiling point and high heat capacity. This means that a small amount of refrigerant can transfer a high amount of heat efficiently. Since refrigerants respond quickly to solar heat, they are more effective on cloudy days than other HTFs. When heat absorption in the collector occurs the refrigerant boils (liquid to gas phase change). Release of heat occurs when the gaseous refrigerant condenses to liquid in the HX. Evacuated tube collectors are used with this kind of HTF. Refrigerants are however restricted due to their effect on the ozone layer [30].

iii. ***Pump***

The most common pump used in solar water heating systems is the centrifugal circulating pump. These pumps usually have low power consumption, low maintenance and high reliability. For closed loop systems pumps are made from low cost cast iron while for open loop bronze is required. Stainless steel pumps are used for pool heating and other applications that involve chemicals. Solar system head and flow requirements are also used to select the appropriate pump. Head is the pressure the pump must develop in order to create the desired flow. There are two types of head: static and dynamic heads. Static head is the pressure resulting from the vertical height which corresponds to the weight of the fluid. i.e. the higher the fluid must be lifted against gravity the higher the static head. Dynamic head includes the friction of the fluid against the pipes and fittings. Dynamic head varies with the size and length of the pipe, number of bends and flow rate and viscosity of the fluid. Circulation pumps are generally categorized as low, medium, or high head. Where low head pumps range from 0.9 to 3 m head, medium head 3 to 6 m and high head 6+ m [30].

iv. ***Storage tank and valves***

Solar water heating systems are typically used to preheat water before it enters into the conventional water heater. A storage tank is required to store the water heated by the solar system. The addition of solar storage tank increases the system efficiency and keeps the solar collector from over heating. During summer, when hot water produced by the solar system is hot enough that it does not require the back up water heater, a bypass valve can be installed between the solar storage tank and the backup water heater such that solar heated water can be used directly. This valve assembly is usually two 3-way valves. When the temperature of the solar heated water is hotter than needed a tempering valve should be added to the hot water line feeding the home to control the desired maximum water temperature. Hot water enters one side, cold another and mixed water exists the home’s hot water plumbing. Check valve is a valve that permits fluid to flow in one direction only used to prevent heat losses at night due to the flow of warm storage tank water to the cold collectors. Finally, a pressure relief valve is required in every hydronic heating system to provide protection against high pressures due to high temperatures. A pressure relief valve of 50 psi is usually adequate to protect closed-loop plumbing systems from excessive pressures [30].

v. ***Other components***

Other components important to solar thermal systems are the expansion tank, pressure gauge, controller and sensors. The expansion tank allows the fluid in a closed-loop system to expand and contract depending on the temperature of the fluid. Without the expansion tank, the plumbing would easily burst when the fluid is heated. The pressure gauge shows if the closed loop system is within an acceptable range of pressure. A typical system pressure is on the order of 10 to 15 psi. A pressure gauge is used as a diagnostic tool to monitor the state of the glycol charge. A loss of pressure indicates a leak in the system that needs to be located and repaired. The differential controller tells the pump when to turn on and off. The controller, via sensors connected to the collector and the storage tank, determines whether the collector outlet is sufficiently warmer than the bottom of the tank to turn the circulating pump on. Sensors are located at the collector outlet, and at the bottom of the solar storage tank. These sensors are thermistors that change their resistance with temperature. The differential controller compares the resistances of the two sensors. It turns the pump on when the collectors are warmer (usually 20 °F higher) than the bottom of the solar storage tank to collect useful heat and shuts the pump down otherwise.

B. **Solar thermal in Canada**

1) Overview

In Canada, 70% of the energy used in the residential and commercial/institutional buildings sector is used for heating. Since 2007, there are an estimated 544,000 m^2 of solar collectors operating in Canada. They are primarily unglazed solar collectors for pool heating (71%) and unglazed perforated solar air collectors for commercial building air heating (26%) delivering about 627,000 GJ of energy and displacing 38,000 tonnes of CO_2 annually [13].

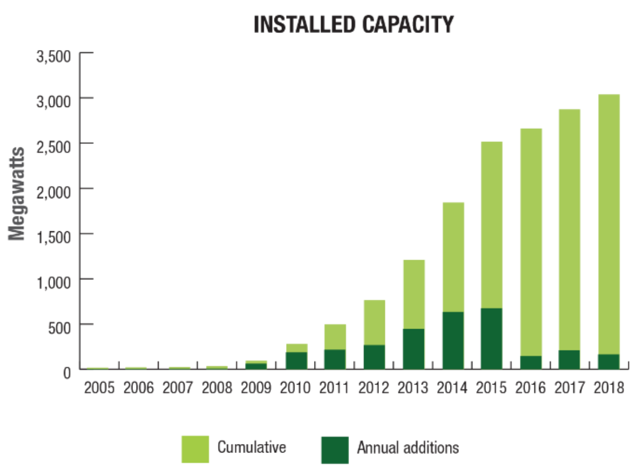


Fig. 7. Installed Canadian Solar thermal capacity [13].

2) CanmetENERGY

CanmetENERGY is Canada's leading research organisation in the field of clean energy. It advances solar thermal developments focusing on: 1- Low temperature (<60 °C) applications. 2- Development and testing of new products. 3- Supporting ecoENERGY for renewable heat program which started in 2007 and invested 36 million CAD in micro (distributed) thermal generation especially in solar thermal for space heating and cooling and water heating. 4- Supporting Canada's National Solar Test facility in Ontario which provides solar equipment testing and certification using a 200 kW indoor solar simulator and studying the feasibility of CSP in western Canada [13,14].

Applications of CanmetENERGY research include: Improving the cost effectiveness of residential solar heaters, increasing the utilization of solar thermal for space heating through development of seasonal thermal storage, developing solar cooling systems and initiating the Drake Landing Solar Community project, and providing software for modeling solar collectors for pool, space and water heating (such as Swift for SolarWall Design). Examples of CanmetENERGY's R&D innovations include development of both transpired solar air collector (SolarWall) and low flow high efficiency residential solar heater systems [16].

The Drake Landing Solar Community project (based in Alberta Canada) was the 1st system in North America capable of supplying 90% of the space heating requirements for each home in the community from solar energy. It involves the use of borehole seasonal storage of summer solar energy underground and distributing it back during winter achieving 92% solar fraction in the colder winter of 2013-2014. It has been operational for 12 years and achieved 100% solar fraction in the 2015-2016 heating season with an average solar fraction of 96% in the period of 2012-2016. It has a COP of 30 in terms of electricity usage relative to heating output [162].

3) Canadian Solar Industry Association

"By 2025, solar industry is widely deployed throughout Canada, having already achieved market competitiveness that removes the need for government incentives, and is recognized as an established component of Canada's energy mix. The Solar industry will be supporting more than 35,000 jobs in the economy and displacing 15–31 million tonnes of greenhouse gas emissions per year, providing a safer, cleaner environment for generations to come". These ambitious goals were formulated by the Canadian Solar Industry Association (CanSIA) in its vision statement. In the same source CanSIA stated that solar thermal is regarded as a mature competitive technology but is affected by natural gas demand and supply (or oil in Newfoundland), It can be used for hot water and space heating in residential, commercial, industrial and institutional buildings and works in conjunction with electric or natural gas heating to reduce reliance on conventional sources. Solar thermal is projected to grow to at least 8,000 MW of cumulative capacity in 2025. However, this study was produced in 2015 and as can be seen from upcoming sections solar thermal in Canada started declining after 2010 and has not recovered yet [15].

4) Market statistics

According to CanSIA, Overall, Canadian Domestic Solar thermal sales peaked in 2010 after which they continued to decline across all collector types. Canadian solar thermal industry continued to decline in 2017, after previous steep declines in 2016. Total (domestic + export) industry revenue decreased from $10.6 million in 2016 to $6.2 million in 2017. Total area of collectors sold decreased from 69,645 m^2 to 45,083 m^2. Revenue from domestic sales decreased from $6.5 million to $4.1 million. Collector area from domestic sales decreased from 36,173 m^2 to 24,953 m^2. Revenue from export sales decreased from $4.1 million to $2.2 million. Collector area from export sales decreased from 33,472 m^2 to 20,129 m^2. Overall, liquid unglazed collectors and air unglazed collectors (such as SolarWalls) are the best-selling types. Liquid evacuated tube is higher in sales than liquid glazed flat plate [24].

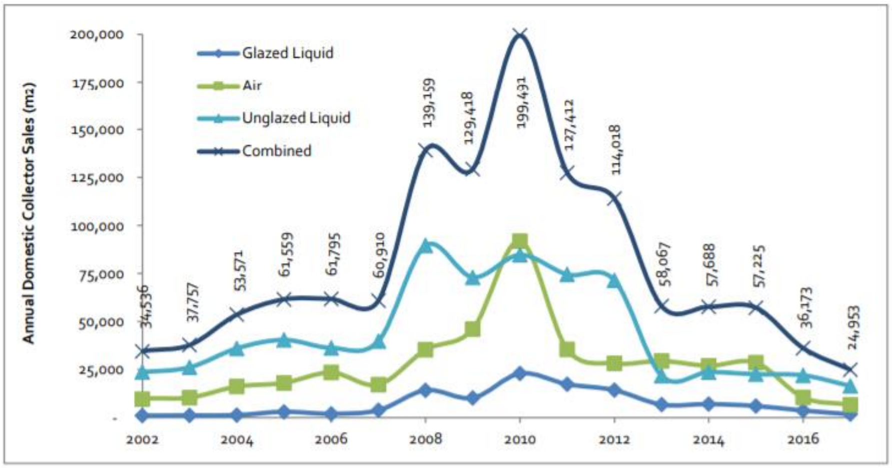


Fig. 8. Solar thermal market statistics in Canada [24].

There are various reasons why solar thermal is less competitive in Canada. Solar thermal seems to suffer from a market barrier due to less favorable economics as the initial capital cost for parts and installation can be substantially high. This is further exacerbated by the low price of natural gas and heating oil as thermal energy sources which increases solar thermal payback periods. Customers seem to dismiss solar thermal after finding out that the technology will not provide all of their heating needs and that they will still need to use a backup heating system. There is also a shortage of skilled labour who can capably install solar thermal systems. Finally, other renewable sources also create a market pressure on solar thermal. Photovoltaics and heat pumps have slowed down the use of solar thermal heating as they offer improved economics which net metered PV seems to offer the most [90].

5) Renewable energy incentives

Figure 9 infographic, obtained from [19], ranks each province from A to F in terms of 3 incentive categories: Energy efficiency incentives, renewable energy incentives and other clean energy incentives. Newfoundland and Labrador is ranked C, F and B respectively. In Canada, there is at least 77 clean energy incentive programs 285 energy efficiency rebates, 27 renewable energy rebates and 12 clean transportation rebates. Table VI shows some of the incentives available in Canada with a focus on solar thermal.

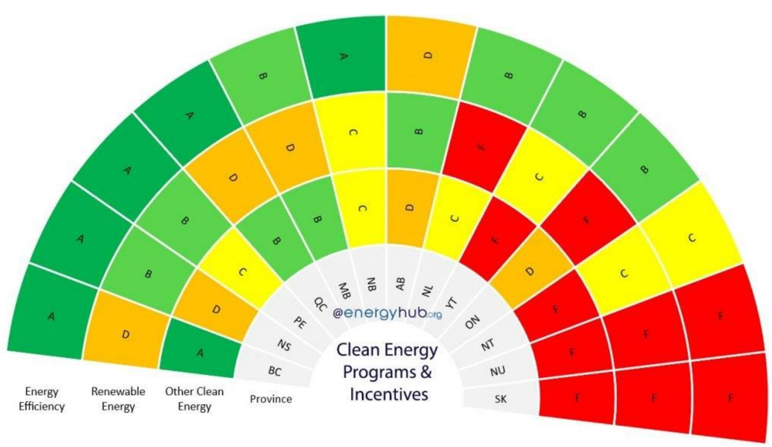


Fig. 9. Clean energy programs evaluation by province [19].

TABLE VI
RENEWABLE ENERGY INCENTIVES BY PROVINCE SELECTION

Province	Type	Details
Federal	1- GST/HST housing rebate [20] 2- Clean Energy Equipment Tax incentive [21]	1-Recover GST or HST paid for energy efficiency upgrades. 2-Accelerated CCA, fully deduct first year capital costs, 10% Atlantic investment tax credit of the cost of energy generation and conservation properties.
Alberta	1- Banff Solar Incentive 2- Edmonton Solar program 3- Medicine Hat smart program	1- $0.75/W rebate on solar energy systems in Banff 2- $0.40/W rebate on solar energy systems in Edmonton 3- $1.00W rebate on solar systems in Medicine Hat
British Columbia (BC)	1- PST Tax Exemption 2- RDN Renewable Energy System Program [23].	1- 7% PST exemption for alternative energy generation and energy conservation equipment. 2- Districts of Nanaimo and Lantzville offer a $250 incentive and $400 rebate for solar PV, solar thermal, and other systems.
Manitoba	Green Energy Equipment Tax Credit	Manitoba currently offers a 10% tax credit for solar thermal energy systems.
New Brunswick	Total Home Energy Savings	NB Power offers $0.20/W to $0.30/W on solar systems
Newfoundland	1- TakeCHARGE Rebate Program 2- Homer Energy Savings Program	1- NL Hydro offers rebates on thermostats, insulation, HVRs, heating equipment, low-flow shower heads, and more 2- The Newfoundland and Labrador Housing Corporation offers up to $5,000 for energy efficiency upgrades to residents whose annual household income equals $32,500 or less. Homeowners must currently have diesel-generated electricity or use 1000+ liters of oil annually.
Northwest Territories	Alternative Energy Technologies Program	50% rebate on the total cost of renewable generation projects (solar, wind, wood pellet heating, etc) for property owners in non-hydro communities.
Nova Scotia	1- Home Energy Assessment Program. 2- Heating System Rebate Program.	1- Home energy efficiency upgrades including insulation, solar thermal systems, windows, and more. 2- Home air and water heaters including solar thermal systems.
Nunavut	No programs	——
Ontario	1- Provincial Programs such as Save ON Energy 2- Enbridge Home Efficiency Rebate	1- Canceled by the conservative government (similar to Alberta) [22]. 2- Enbridge Gas offers rebates (up to $5000) for air sealing, high-efficiency furnaces, windows, doors, water heaters, and more.

Prince Edward Island	1- Energy Efficient Equipment Rebates 2- Solar Electric Rebate Program	1-Rebates on various home air and water heating equipment 2- $1.00 per watt incentive on solar photovoltaic systems.
Quebec	Energy Efficiency rebate programs	Four energy efficiency rebate programs that include water heaters
Saskatchewan	No programs	——
Yukon Territory	Good Energy Renewable Energy Rebate	The Government of Yukon offers a rebate of $800/kW for off-grid residents who installing solar or other generation systems.

C. Innovations in Solar thermal

1) SolarWall

Transpired Solar Collector (SolarWall) was developed by National Renewable Energy Laboratory (NREL) and Conserval Systems inc. in the late 1980s. It is a reliable, low cost technology used for space heating in commercial and residential buildings. It has a payback period of 3 to 12 years and 30-year life span and requires no maintenance. It is best installed on south facing walls. It is externally made from dark colored perforated metal sheeting which performs the role of a large solar collector. It should be mounted on the wall or the roof of the residence leaving a 4 to 6 inch gap in between. As outside air is drawn into the collector by ventilation fans its temperature increases by as much as 22 °C. The heated air flows to the top of the wall where it is circulated into the house. This system can convert as much as 80% of available solar radiation to heat and costs $6/ft^2 for new constructions and $10/ft^2 for retrofits. If roof mounted, the collector angle can be set up to maximize thermal energy gains during winter as that's when space heating is needed. Collector angle should be at least 45° to allow for snow to slide off the panels. Installing this system on the house walls leaves space on the roof for PV panels [17,18].

In a study in northeast China, glazed transpired collector (GTC) or SolarWall was used to provide space heating and improve the air quality of rural dwelling which uses Chinese biomass burning kang system for heating (which is used in 85% of rural houses but associated with poor indoor air quality). The SolarWall was able to increase average indoor temperatures by 4-6 °C and reduce CO, CO_2, PM2.5 and PM10 by up to 73.9%, 42.7%, 56.2% and 58.1% respectively [47].

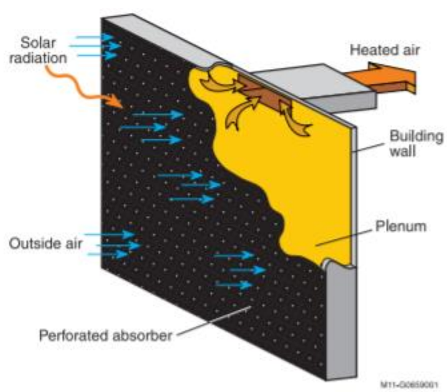


Fig. 10. SolarWall [17,18].

2) Combi systems

Solar combi systems combine solar DHW heating with space heating and or pool heating. Since DHW load is year-round it serves as a very good base load for the solar thermal system. Combi systems are usually custom-built systems whose success often depends on the designer. Solar glycol utilizing closed loop systems or drainback systems can be used to protect against freezing conditions with the former option being more popular. Drainback systems use water as the HTF in the solar loop but all the water must drain back into the building whenever the pump is not operating. This increases design and installation complexity. As solar combi systems usually utilize large array sizes, they are prone to overheating in the summer months where increasingly hot collector loops can lead to collector stagnation, steam formation and glycol damage. Europe utilizes small and simple combi systems designs (450-750 L storage and 10-15 m^2 collector area) as they are simpler to design, install and upkeep. It is best to use combi systems with loads that have a low return temperature (radiant heating) as that increases the efficiency of the solar systems. Solar contribution is lower in winter due to shorter days and lower radiation. One way to enhance winter output while reducing summer overheating is to vertically mount the panels at a slope of 70° or more [90].

In [91], the authors investigated directly storing the thermal energy generated in the building envelope instead of using a storage tank for a solar combi system by connecting the system directly to the space heating circuit. The result showed substantial reductions in energy demand and rise in solar yield due to no storage tank thermal losses thus increasing the solar fraction. However, the solar fraction increase was tied to the usage of space heating elements with low operational temperatures. This method does not eliminate energy storage, but it does reduce its size. For example, a solar fraction of 50% can be reached for a collector area of 30 m^2 (for the load provided in the study) with a 3000L tank or a 1000L tank if direct heating is utilized.

In [92] a solar combi-biomass hybrid system was designed to provide thermal energy demand for a the Pancretan National Stadium in Crete, Greece. The solar combi system contributed 59.5% of the thermal load (solar fraction) while the biomass system, which relied on the by products of the olive oil industry totalling 31 tonnes of biomass pellets, provided the rest.

3) PVT systems

Photovoltaic thermal (PVT) systems consist of a PV module, a channel, a coolant, a fan or pump and a collector. The coolant is used to remove heat from the PVT module and use it to provide space and water heating. PVT systems can be classified based on fluid circulation as natural or forced flow. Performance of a PVT system depends on thermal and electrical efficiency and exergy efficiencies. Thermal efficiency depends on fluid flow rate, type of solar collector and modifications such as the addition of fins to the coolant channel. When integrated in the building envelope (BI-PVT), these systems can generate electricity, heat and light simultaneously. However, this technology is still in the development phase as it has a few limitations, such as, higher cost compared to PV and solar thermal. The concept of PVT stems from the fact that 50% of sunlight incident on a PV cell is converted to heat which can cause damage to the cell if sustained for a long time. This heat can alternatively be recovered and used for various applications such as crop drying or space and water heating. Water flows beneath the PV panel in various flow patterns depending on design. While glazed collectors offer higher energy output unglazed modules offer higher exergy [93].

D. Related concepts

1) Desired temperatures

In recent solar thermal research, Hot water temperature was set to 50 °C while space heating set point temperature was 24 °C for a house in Newfoundland [31,32]. Whereas hot water temperature was set to 50 °C and space heating set point temperature was 20 °C for Athens and Florina, Greece [33]. BC plumbing code requires electric water heaters to be set to 60 °C but this doesn't apply to other water heater types. For safety, downstream of the tank (faucets') water temperature should not exceed 49 °C. Storing hot water below 60 °C (for electric heaters) leads to growth of legionella bacteria which is a health risk [34].

According to the Center for Disease Control (CDC), Legionella, a bacterium that causes legionnaires disease, occurs naturally in freshwater environments. Legionella grows best at temperatures between 25 °C and 42 °C. It is recommended by CDC to keep cold water cold (below 25 °C) and hot water hot (above 42 °C) and prevent stagnation to limit growth of this microorganism. Other germs such as pseudomonas and nontuberculous mycobacteria also flourish in similar environments [35]. According to the National Research Council Canada (NRC), legionnaire's disease is serious respiratory illness that results in sever pneumonia and can result in death. In 2012, there was an outbreak in one of Quebec's buildings caused by a cooling tower which resulted in 13 deaths and 170 infections. However, NRC estimates the number of infections per year in Canada is underreported as the disease is seldomly tested for [36].

According to Newfoundland Power, comfortable space temperatures are 18 °C to 21 °C for main living areas, 16 °C to 18 °C for bedrooms and 15 °C for basements and unoccupied spaces [37]. BC hydro gave similar figures and suggested that heating costs rise about 5% for every degree above 20 °C the thermostat is set to [38].

According to the owner of 16 hennebury place (design house) the thermostat is set for 71 °F (21.6 °C) from 6 am to 11 pm (daytime) and to 65 °F (18.3 °C) from 11 pm to 6 am (nighttime). These values are in line with Newfoundland Power's recommendation and therefore were inserted into BEOPT to calculate the thermal load required to sustain such a thermostat setting. The hot water set point according to the owner is 125 °F (51.6 °C). which was inputted into BEOPT.

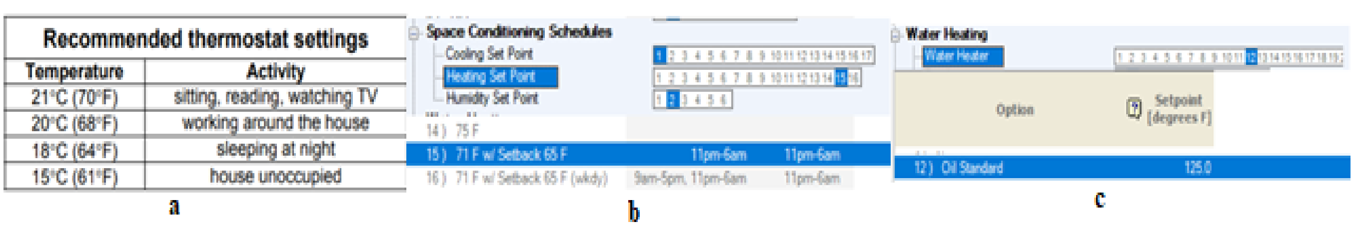


Fig. 11. a) Newfoundland Power recommended thermostat setting, b) Space heating setpoint, c) Water heater setpoint [37].

2) Sizing of solar water heating system

For proper sizing of a solar water heating system, total collector area and storage volume is needed in order to supply 90%-100% of the household's hot water needs in the summer. The use of RETScreen is recommended for this purpose. A rule of thumb proposed is to size the system so that the 1st two inhibitors of the house account for 2 m^2 of collector area each and 1.2 to 1.4 m^2 of collector area for each additional inhibitor. It is also recommended to use a small 50-60 gallon tank if 2 people live in the premise, medium 80 gallon tank for 3-4 people and a large 120 gallon tank for 4-6 people. For active solar water heating systems, the size of the storage tank increases by 1.5 gallons per ft^2 of collector used. This helps the system from overheating when the demand of hot water is low [51].

According to NRCan's solar water heating systems buyer's guide. The size of the water tank depends on the daily hot water use which depends on the number of occupants. For 5+ people the consumption is 300 litres per day and the suggested solar water heater size is large (greater than 6 m^2 of collector area) and with a preheat tank. The suggested storage water volume is 270 litres or 60 gallons (high power output). In the same study by NRCan it was estimated that a solar water heating system can provide 43% of hot water energy required by a house in St. John's using 6 m^2 of flat plate single glazed collector area and two 270 litre (60 gallon) hot water tanks [61].

In John Duffie's book solar engineering of thermal process [60], 1500 kg water mass and 11.1 W/°C heat loss coefficient-area product were used when calculating the temperature and losses of water storage tank. In the f-chart section of the book, it was explained that for economic reasons 50 to 200 liters of water per m^2 of collector area was optimum. While the f-chart method assumes 75 L/m^2 be default. In [62] a 900L tank was used with 4% storage tank losses and 1% pipe losses for solar water heating load in Newfoundland.

Authors of [63] underwent two designs for a household with a thermal energy consumption of 7887 kWh for space heating and 4689 kWh for DHW. In design 1 (space heating only), 16 m^2 of collector area, a 5 kW heat pump (COP = 3.5) were used for generation, 5000 W/K external heat exchanger was used between the collector fluid loop and the storage fluid loop, a 2.5 m^2 1000 W/element radiator was used (T_{in} = 45 °C & T_{out} = 35 °C) and 47,000 liters of water were used as storage volume corresponding to a tank volume of 47 m^3. The output of the collectors was 797 kWh/m^2. Heat pump consumed 1623 kWh and provided 4812 kWh of energy. Room temperature was set at 18 °C in winter months and reached 23 °C in summer months. Overall solar fraction for design 1 was 61.4%. In Design 2, both space heating and DHW were considered. The total collector area was 18 m^2 and total tank volume was 31 m^3. A 4 kW propane gas boiler was used. Solar fraction for space heating was 32% while for water heating it was 60% with an electrical heater covering the remaining 40%.

3) Alternate Energy Technology collectors

Alternate Energy Technology (AET) is a US company based in Jacksonville, Florida that has been manufacturing solar collectors since 1975. Alternate energy (AE) collector series can be used for space and water heating, radiant floor heating and industrial processes. They have the following characteristics:

- The absorber is a "thermafin" type absorber developed by AET in 1996 and used in all AE models. It included fins which are high frequency forge welded resulting in high heat conduction. It is 100% copper.
- "Crystal clear" selective coating (absorber coating) made from thousands of nanocrystalline growth projections which increases the surface area of the absorber resulting in 3% additional energy harvested.
- The finished coating has absorptivity > 0.96 and emissivity < 0.08.
- Glazing is low iron tempered glass with a solar transmissivity of over 90%.
- Polyisocyanurate foam board insulation.
- "Quick lock" mounting hardware which simplifies installation and can withstand wind loads of 195 mph. mounting options include: pitched roof, flat roof, ground, balcony and façade mounting.
- AE21, 24, 26, 28, 32, 40 have a recommended flow rate = 0.5 to 1.18 GPM and a working pressure 165 PSI.

TABLE VII
AE COLLECTOR COMPARISON

Collector		Glazing		Absorber		Gross Area (m^2)	Thermal Performance	
Manufacturer	Model	No. of Layers	Type	Material	Coating		kWh/Day	kWh/m^2
Alternative Energy Technologies	AE-21	1	Glass	Copper tubes and fins	Selective	1.93	5.16	2.68
	AE-26					2.35	6.36	2.7
	AE-32					2.96	8.06	2.72
	AE-40					3.7	10.8	2.73
	AE-32-E				Moderately Selective	2.96	6.54	2.21
	AE-40-E					3.69	8.18	2.22
	ST-32E					2.87	6.71	2.34
	ST-40E					3.59	8.32	2.32
	MSC-21				Selective	2	5.1	2.56
	MSC-32					3.04	7.97	2.63
	MSC-40					3.92	10.29	2.63

4) Solar Rating and Certification Corporation

Certification data on specific thermal solar collectors and solar hot water heating systems are available from SRCC. Types of collectors covered are flat plate, tubular, ICS, thermosyphon, concentrating and transpired solar thermal collectors as well as solar PV water heating collectors. SRCC also provides convenient summary sheets of various collectors which include: company name, model number, collector area, absorber coating, slope, intercept and average daily energy collected under standard conditions. The "Collector Thermal Performance Rating" section consists of performance metrics in SI and Inch-Pound (English) units for: three irradiance conditions (clear, mildly cloudy, and cloudy), five temperature categories (A–E), slope, intercept as well as quadratic efficiency equations. Incident-angle modifier expressions are also given [25].

OG-100 ICC-SRCC™ CERTIFIED SOLAR COLLECTOR # 10001940

ICC-SRCC
SOLAR RATING & CERTIFICATION CORPORATION

Supplier:
Alternate Energy Technologies
1345 Energy Cove Court
Green Cove Springs, FL 32043 USA
www.aetsolar.com

Reference Standard:
ICC 901/SRCC Standard 100-2015

Brand:	Alternate Energy
Model:	AE-21
Collector Type:	Glazed Flat Plate
Certification #:	10001940
Original Certification:	December 12, 2014
Renewal Expiration Date:	November 30, 2020

Certifications must be renewed annually.

This solar collector has been evaluated and certified by the Solar Rating & Certification Corporation™ (ICC-SRCC™), an ISO/IEC 17065 accredited and EPA recognized Certification Body, in accordance with the SRCC OG-100 Certification Program, as defined in SRCC OG-100, Operating Guidelines and Minimum Standards for Certifying Solar Collectors. This award of certification is subject to all terms and conditions of the current OG-100 Program Agreement and the documents incorporated therein by reference. This document must be reproduced in its entirety.

COLLECTOR THERMAL PERFORMANCE RATINGS

Kilowatt-hours (thermal) Per Panel Per Day				**Thousands of Btu Per Panel Per Day**			
Climate -> Category (Ti-Ta)	High Radiation (6.3 kWh/m².day)	Medium Radiation (4.7 kWh/m².day)	Low Radiation (3.1 kWh/m².day)	Climate -> Category (Ti-Ta)	High Radiation (2000 Btu/ft².day)	Medium Radiation (1500 Btu/ft².day)	Low Radiation (1000 Btu/ft².day)
A (-5 °C)	8.8	6.7	4.6	A (-9 °F)	30.1	22.9	15.6
B (5 °C)	7.7	5.6	3.5	B (9 °F)	26.4	19.1	11.9
C (20 °C)	6.1	4.0	2.0	C (36 °F)	20.8	13.7	6.7
D (50 °C)	3.1	1.3	0.0	D (90 °F)	10.5	4.3	0.0
E (80 °C)	0.5	0.0	0.0	E (144 °F)	1.8	0.0	0.0

A- Pool Heating (Warm Climate) **B**- Pool Heating (Cool Climate) **C**- Water Heating (Warm Climate) **D**- Space & Water Heating (Cool Climate) **E**- Commercial Hot Water & Cooling

TECHNICAL RESULTS				**Tested in accordance with:** ISO 9806:1994			
ISO Efficiency Equation [NOTE: Based on gross area and (P)=Ti-Ta]							
SI UNITS:	η= 0.740 - 5.18920(P/G) - 0.01910(P²/G)			**Y Intercept:**	0.749	**Slope:**	-6.159 W/m².°C
IP UNITS:	η= 0.740 - 0.91456(P/G) - 0.00187(P²/G)			**Y Intercept:**	0.749	**Slope:**	-1.086 Btu/hr.ft².°F
Incident Angle Modifier							
θ	10°	20°	30°	40°	50°	60°	70°
Kτα	1.00	1.00	0.99	0.97	0.93	0.79	
Impact Safety Rating: 11							

COLLECTOR SPECIFICATIONS					
Gross Area:	1.932 m²	20.80 ft²	**Dry Weight:**	33.6 kg	74.0 lb
Net Aperture Area:	1.783 m²	19.19 ft²	**Fluid Capacity:**	2.7 liter	0.7 gal
REMARKS:					

Fig. 12. SRCC certification example [25].

TABLE VIII
CATEGORY EXPLANATION

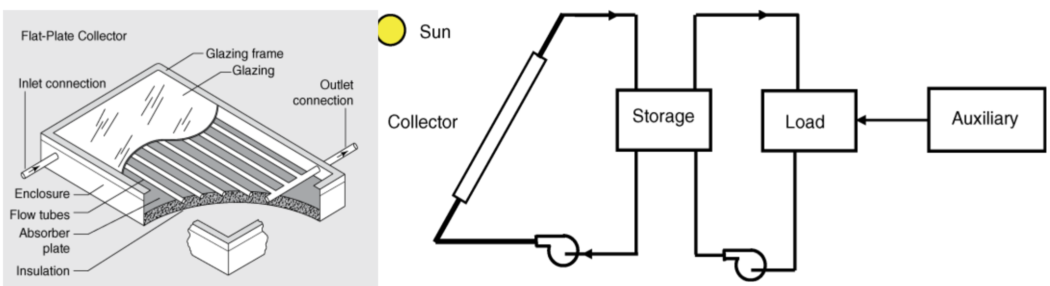
Category	Temp. difference	Application
A	-5 ℃	Pool Heating
B	5 ℃	Space Heating, Hot Water (air Systems)
C	20 ℃	Space Heating, Hot Water (air Systems)
D	50 ℃	Space Heating, Hot Water (liquid Systems), Air Conditioning
E	80 ℃	Space Heating, (liquid Systems) Air Conditioning (process heat)

5) Simple Flat plate system

Flat plate collectors are the most commonly used solar collector type for domestic space and water heating. According to [51] flat plate collectors are the best option for solar thermal generation in many northern U.S. states. Figure 13 (a) shows a cut away view of a flat plate collector with the following components:

- Glazing which are transparent cover sheets which have high solar transmissivity and low thermal transmissivity passing most of solar radiation and trapping the reflected heat.
- Absorber plate is made from a material with high solar absorptivity and low thermal emissivity.
- Flow tubes through which HTF flows to absorb heat from the absorber plate.
- Collector bottom which is directly below the absorber and is insulated to minimize heat losses.
- Inlet and outlet connections and casing.

Figure 13 (b) shows a simple flat plate collector system schematic. In it we observe two fluid loops one from the flat plat collector to the storage system and one from the storage system to the load. In the 1st loop, water is circulated through the collector (using a pump) where it is heated and then stored in the storage tank. In the 2nd loop hot water to serve the thermal load is drawn from the storage tank, supplies thermal energy to the load and is then returned to the tank at a lower temperature. The auxiliary heater is used when the solar system is unable to supply hot enough water to meet the thermal load [25].



a b

Fig. 13. Flat plate collector a) schematic b) simple system [25].

E. **Thermal Energy Storage**

1) Introduction

The three main types of thermal energy storage are sensible heat, latent heat and thermochemical storage. In general, the term thermal energy storage includes both heat and cold storage. Heat storage absorbs thermal energy during charging while cold storage releases heat during charging. Also, heat storage usually happens at a temperature above the ambient, while cold storage happens at a temperature below ambient temperature. Sensible heat (Q) is the thermal energy involved in changing the temperature of a material or system given by $Q = mc_p(T_2-T_1)$. Where m is the mass of the system. T_1 and T_2 are the initial and end temperatures and c_p is the specific heat of the material. Latent heat (L), on the other hand, causes heat transfer but without temperature change. This happens during phase change processes where heat is absorbed or released at a constant temperature. The heat required to melt a solid substance is the latent heat of fusion (h_{sl}) and to vaporize a liquid the latent heat of vaporization (h_{lv}). For water h_{sl} = 335 kJ/kg and h_{lv} = 2251 kJ/kg. For paraffin wax the melting temperature is 50 °C and h_{sl} = 200 kJ/kg [53,54].

During phase change/transition there is no change in the chemical composition of the material as it is a physical process and not a chemical reaction. PCM mostly involve solid to liquid phase change. Phase transitions can occur where both phases are solid (α-titanium to β-titanium at 882 °C) here only the crystal structure changes. There are 18 crystalline phases of water. Until 1758, it was believed that phase change, which is accompanied by a small change in temperature, needs only a small amount of heat to be added or extracted. This was challenged by Dr. Joseph Black (a professor of medicine) who conducted experiments measuring the amount of heat transferred during freezing, condensation and evaporation of water and noted that the amount of heat transferred during phase change was very large despite little change in temperature this could not be explained as sensible heat which led to the invention of the term latent heat to describe the phase change process. This inspired Dr. Black's assistant James Watt to apply this theory and improve the efficiency of the steam engine by 500% [53,54].

In a real thermal energy storage system both sensible and latent heat are present. The total amount of energy stored can then be given by $Q = m[c_1(T_m - T_2) + L + c_2(T_2 - T_m)$. Where c_1 is the specific heat of the 1st phase while c_2 is the specific heat of the 2nd phase. T_m is the phase change temperature, T_1 is the initial lower temperature while T_2 is the final high temperature. Applications of TES vary by scale and temperature range. Small scale applications include the thermal management of electronics using PCMs and the temperature regulation in biomedical fields. These involve low temperature ranges (5 °C to 18 °C). Medium applications involve residential and commercial building space and water heating, solar thermal energy storage and waste heat recovery. These involve temperatures from 20 °C to 90 °C. Industrial processes involve temperatures in the 100 °C to 250 °C range and large-scale storage is usually used with CSP plants (such as molten salt) and involves temperatures in the 300 °C to 1400 °C range [53,54].

The last type of TES is thermochemical energy storage in which energy is stored in the chemically reversible reactions where the reaction products can be separated and stored over a long period of time. During discharging, an exothermic reaction releases the stored energy. Thermochemical energy storage systems permit very high energy storage densities but have been rarely implemented in practice. The technology is largely still in the research and development phase. Thermal Insulation is important in both sensible and latent heat TES to decrease heat loss to the environment. A material with low thermal conductivity is desired to prevent heat conduction such as aerogel insulation which is 99.8% air [53,54].

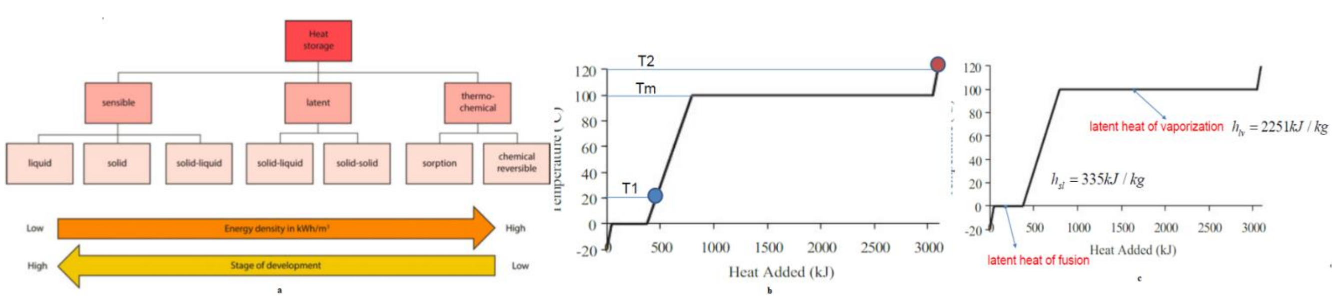


Fig. 14. Thermal storage classification, b- and c- latent and sensible heat curves [54].

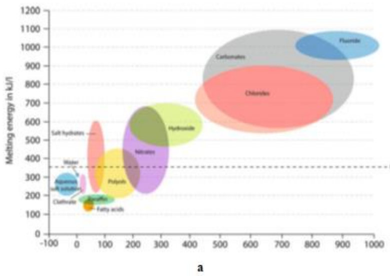
2) Sensible heat TES systems

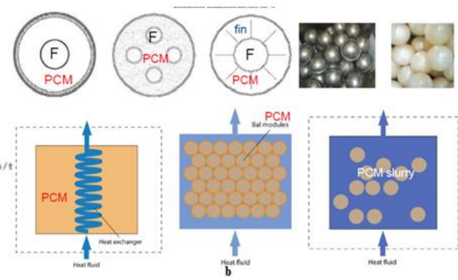
From the sensible heat equation provided earlier, it is clear that higher specific heat results in higher energy storage capacity. The mass specific energy density can be derived as $Q_m = \frac{Q}{m} = c(T_2 - T_1)$ while the volume specific energy density as $Q_V = \frac{Q}{V} = \frac{mc(T_2 - T_1)}{\frac{m}{\rho}} = \rho c(T_2 - T_1)$. This shows that for higher volumetric specific energy density the product of density (ρ) and specific heat (c) should be as high as possible. The fluid with the highest ρc product is actually water however water is limited in its sensible heat applications due to temperature range limit of 5-95 ℃ before/after which it freezes/evaporates which is suitable for building heating/cooling. Thermal conductivity of materials determines the rate at which heat is transferred during charging and discharging processes. Higher thermal conductivity generally leads to higher conductive and convective heat transfer rate. Conductive heat transfer is given by $q_x = -Ak\frac{dT}{dx}$ where qx [W] is the amount of heat conduction in the x direction, A is the area of heat conduction , $\frac{dT}{dx}$ is the temperature gradient and k is the thermal conductivity of the material. Convective heat transfer between a solid surface and the HTF is given by $q = h_c A(T_s - T_f)$ where q [W] is the amount of heat convection, h_c is the convective heat transfer coefficient [W/m^{2}K] which is related to k, A is the area of heat transfer and T_s and T_f are the temperatures of the solid surface and the HTF respectively. Examples of sensible heat TES systems include Geothermal energy storage, Aquifer thermal energy storage and molten salt energy storage [53,54].

A 2020 study reviewed packed bed storage systems (PBSS) and highlighted its suitability for low thermal applications where air is the HTF which is different from liquid storage in the fact that charging and discharging can not occur simultaneously. In PBSS convection between the packing element and the HTF is the dominant mode of heat transfer [55]. In [56] a flat plat collector was coupled with a sensible heat TES tank for domestic water and under floor space heating. It was found that increasing the flow rate of the HTF increases the mean temperature and reduces charging time, in other words, heat transfer increases. Exergy efficiency of the charging process and the collector efficiency also increase. The best performance was found for the highest mass flow rate of 0.4 kg/s and small tank height of 0.8 m. The storage medium temperature varied from 44 ℃ to 59 ℃.

3) Latent heat storage

PCM is a substance with a high heat of fusion (or vaporization) which when undergoing phase change at a certain temperature can store/release large amounts of thermal energy. PCMs can be organic such as paraffin wax or inorganic such as salts. To maximize energy storage capacity and charging and discharging rates in latent heat storage (LHS) system heat transfer process needs to be optimized such as adding internal heat conduction structures, for instance fins, to increase heat transfer rate. The usage of encapsulated PCM modules over bulk PCM bodies increases the contact area between the HTF and the PCM and therefore increases heat transfer. Another development is direct contact storage where the HTF comes in direct contact with the PCM and no HX is required. A form of this is the so called PCM slurry or PCS which has the advantage of being pumpable even in its solid state. It consists of small encapsulated PCM particles suspended in a carrier fluid with a low solidification temperature. In which case it can serve as both the heat transfer fluid and the storage medium [53,54]. In [57], microencapsulated PCMs (MPCMs) were added to an ethanol/water mixture multi walled carbon nanotube nanofluid (MWCNT) at 15 wt%. It was found that for the hybrid MPCM-MWCNT slurry, the temperature rise was lower than the base fluid, but the discharge temperatures were 4.6 °C higher.





Material	Melting temperature (°C)	Heat of fusion (kJ kg^{-1})
Paraffin wax	64	173.6
Polyglycol E 400	8	99.6
Polyglycol E 600	22	127.2

Material	Melting temperature (°C)	Heat of fusion (kJ kg^{-1})
Stearic acid	69	202.5
Palmitic acid	64	185.4
Capric acid	32	152.7
Caprylic acid	16	148.5

c

Fig. 15. a) Comparison between different PCM materials, b) PCM topologies, c) PCM material heat of fusion [54].

4) Energy storage in solar thermal applications

Mismatch between supply of solar thermal and demand by the load is the reason storage is needed. In most applications it is not practical to meet all of the load "L" with energy from solar energy over long periods of time, and an auxiliary energy source must be used. The total load L is met by a combination of solar energy "L_S" (which in practice will be somewhat less than Q_u because of losses) and "L_A" (the auxiliary energy supplied). For water heating with liquid collectors, Sensible heat thermal energy storage (SHTES) using water is logical. For air heating with air collectors SHTES (pebble bed) or latent heat thermal energy storage (LHTES) can be used. Characteristics of TES are: Capacity per unit volume, Temperature range over which it operates (Temperature at which heat is added and removed), Means of addition/removal of heat, Temperature stratification, Power requirements for heat addition/removal, Containers/tanks, Means of controlling thermal losses, Cost [60].

Losses include transport from collector to storage, into storage (HX) losses, storage losses, out of storage (HX) losses, transport from storage to application losses, into application (HX) losses. If open loop system is used some of these losses (HX related) can be neglected. Objective is to minimize losses subject to cost constraint. Storage design affects the temperature at the collector. For many solar systems, water is the ideal material in which to store usable heat. Energy is added to and removed from this type of storage unit by transport of the storage medium itself, thus eliminating the temperature drop between transport fluid and storage medium. 1-hour time scale is used due to availability of data at that scale [60].

In PBSS, heat addition can't happen at the same time as heat removal (unlike water storage). The heat transfer coefficient between air and the solid storage material is high which promotes thermal stratification while the costs of the storage material and container are low. The conductivity of the bed is low when there is no airflow. The high heat transfer coefficient–area product between the air and pebbles means that high-temperature air entering the bed quickly loses its energy to the pebbles. The pebbles near the entrance are heated, but the temperature of the pebbles near the exit remains unchanged and the exit air temperature remains very close to the initial bed temperature. As time progresses a temperature front passes through the bed. By the fifth hour, the front reaches the end of the bed and the exit air temperature begins to rise. When the bed is fully charged, its temperature is uniform. Reversing the flow with a new reduced inlet temperature results in a constant outlet temperature at the original inlet temperature for 5 h and then a steadily decreasing temperature until the bed is fully discharged. If the heat transfer coefficient between air and pebbles were infinitely large, the temperature front during charging or discharging would be square. The finite heat transfer coefficient produces a ‘‘smeared’’ front that becomes less distinct as time progresses. A packed bed in a solar heating system does not normally operate with constant inlet temperature. During the day the variable solar radiation, ambient temperature, collector inlet temperature, load requirements, and other time-dependent conditions result in a variable collector outlet temperature [60].

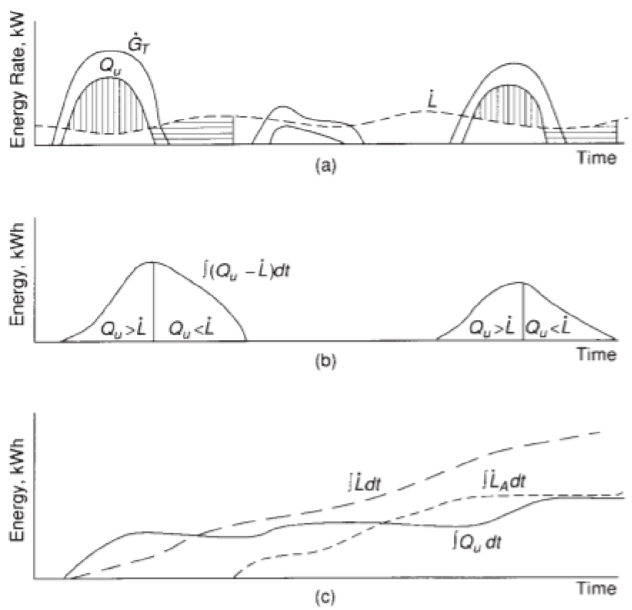


Fig. 16. Solar process with storage. Q = useful energy, L = load, G = irradiance, L_s = load supplied by solar, L_A= load supplied by auxiliary unit. (a) vertical shaded areas = excess, horizontal shaded - deficit, (b) energy added or removed from storage, (c) integration of Q, L_s and L_A [60].

F. Software used

1) BEOPT

BEOPT is short for Building Energy Optimization Tool. It is a software developed by NREL which is affiliated with the U.S department of energy (DOE). It can be used to identify the most cost optimal efficiency designs and their resulting energy savings. It can be used to analyze new constructions or to design retrofits. BEOPT provides detailed results based on house characteristics, size, architecture, occupancy, location and utility rates. It utilizes the U.S DOE developed simulation engine EnergyPlus.

In 2016, a study presented the sizing of a solar thermal energy storage system for domestic water heating in a detached house in St. John’s. The proposed model used a combination of BEOPT and Simulink to determine the temperature of a tank and the heat loss of the system. System design depends on temperature, time and flow rate.

The proposed model was used to determine storage water and house temperatures. The maximum temperature of the storage tank was 82.4 °C and the house space temperature was found to be 18 to 25.11 °C. Space heating for the studied house required 12,268 kWh/yr [62].

In a relevant study, two houses in St. John's were simulated using BEOPT and the result showed that the annual energy consumption using the software was almost the same as the actual energy consumption logged with a 2-minute sample time. The paper notes that using BEOPT is important for the design of a renewable energy system and to start on the path of zero net energy. Houses in NL are heated for more than 6 months due to long winters. The utility company logs electricity consumption once per month, but a much faster rate of logging is required for energy analysis of the house. This is why software like BEOPT is recommended [94].

2) Polysun

Polysun was developed in 1992 by the Institute for Solar Technology (SPF) at the University of Applied Sciences in Rapperswil, Switzerland. The software includes more than 1000 built-in templates that are based on actual/practical project layouts. The program provides economic and GHG emission analysis. the software was validated to be 90% to 95% accurate by [96,97].

In a 2018 study, an hourly model based on energy balance equations was used to simulate a DHW solar thermal system in Mozambique and compared against polysun results. While the comparison showed similar results, polysun's use of auxiliary heating was deemed by the authors as being either undersized or oversized depending on the control layer (89.5% error). This was seen as being due to polysun's usage of a temperature stratification model for the storage tank [95]. In [96], the authors compared between three widely used solar thermal software namely, polysun, TRNSYS and energyPRO. The results showed that Polysun results were closer to TRNSYS results and therefore more reliable. However multiple sources of error occurred such as 5.2% higher solar thermal output, 24.4% higher auxiliary heater output, 2% higher tank losses and 42% higher pipe losses (for a DHW system).

3) RETScreen

Developed by the government of Canada, RETScreen expert is used by renewable energy professionals and federal government bodies in Canada required to report emissions [98]. RETScreen was featured in a 2020 study as a tool for measuring the effects of energy efficiency measures in a house retrofit which included passive solar elements [99]. In [100], a French church was refurbished with a solar thermal system which was designed using RETScreen. The study examined both a flat plate water collector and a vacuum solar air collector in terms of the building integrability. Using the flat plate water collector, DHW load decreased by 45%. On the other hand, the air collector delivered 1615 kWh more heat (69% solar fraction) even as the total collector area decreased. Air heaters where mounted as window shutters and provided 47% of the space heating load.

IV. Methodology (Manual Calculations)

In this section, a study using eleven well established equations are used in excel to give an initial estimate to the values pertaining to a solar space and water heating combi system design and to elaborate on the implications of these equations.

Hottel–Whillier–Bliss (HWB) equation for flat plate collectors (eq 2) can be used to calculate the useful amount of energy output from the collector and the collector efficiency [25]. First, efficiency can be defined as the ratio of useful energy to the input solar energy such that:

$$\eta_c = \frac{Q_{useful}}{Q_{in}} = \frac{A_c[I_T\tau\alpha - U_L(T_{avg} - T_a)]}{I_T A_c} = \tau\alpha - U_L\frac{T_{avg} - T_a}{I_T} \tag{1}$$

Replacing average collector temperature T_{avg} with inlet fluid temperature T_{in} yields

$$\eta_c = F_R\tau\alpha - F_R U_L\frac{T_{in} - T_a}{I_T} \tag{2}$$

Where:

η_c = Collector efficiency (unitless)

F_R = collector heat-removal factor (unitless), τ = cover glass transmissivity (unitless), α = absorber plate absorptivity (unitless), U_L = Overall conductance.

I_T = irradiance (W/m^2) and A_c = total collector area (m^2)

$F_R\tau\alpha$ = intercept (unitless) and $F_R U_L$= slope (W/m^2.°C)

T_{in} is the inlet fluid temperature (°C) and T_a is the ambient temperature (°C)

Q_{useful} can then be calculated using equation 3:

$$\text{Quseful} = \eta_c * I_T * A_c \tag{3}$$

Due to the 1st law of thermodynamics, Quseful can also be given by the sensible heat equation as

$$\text{Quseful} = \dot{m}c_p(T_{out} - T_{in}) \tag{4}$$

Where:

$\dot{m}$ = mass flow rate of the fluid.

c_p= Specific heat of HTF (4.186 kJ/kg °C for water).

$T_{out} - T_{in}$= Temperature difference of fluid at inlet and outlet of the collector.

Note: for this project SI units will be used whenever possible.

A. Efficiency Calculation

The efficiency calculation was done for every hour of the year using excel. The inputs to the equations are highlighted in this section.

1) Inlet fluid temperature

Inlet fluid temperature (T_{in}) was initially 15 °C assumed value which was obtained from B. Hodge's book [25]. This value makes sense for southern U.S state such as Mississippi but is rather large for a town like St. John's (temperature of a river in Mississippi varied from 0 to 30 °C throughout the year [73]). Therefore, later actual hourly water main temperatures were obtained from BEOPT for the design house location and were used in efficiency calculation to establish the upper boundary (where all water was consumed) followed by the use of ambient temperatures to establish the lower boundary (where water was used but not consumed). The St. John's water main's maximum temperature was 11.75 °C, minimum 4.3 °C and average 8 °C. The effect of the three inputs on the final system and explanation of the boundaries will be discussed shortly.

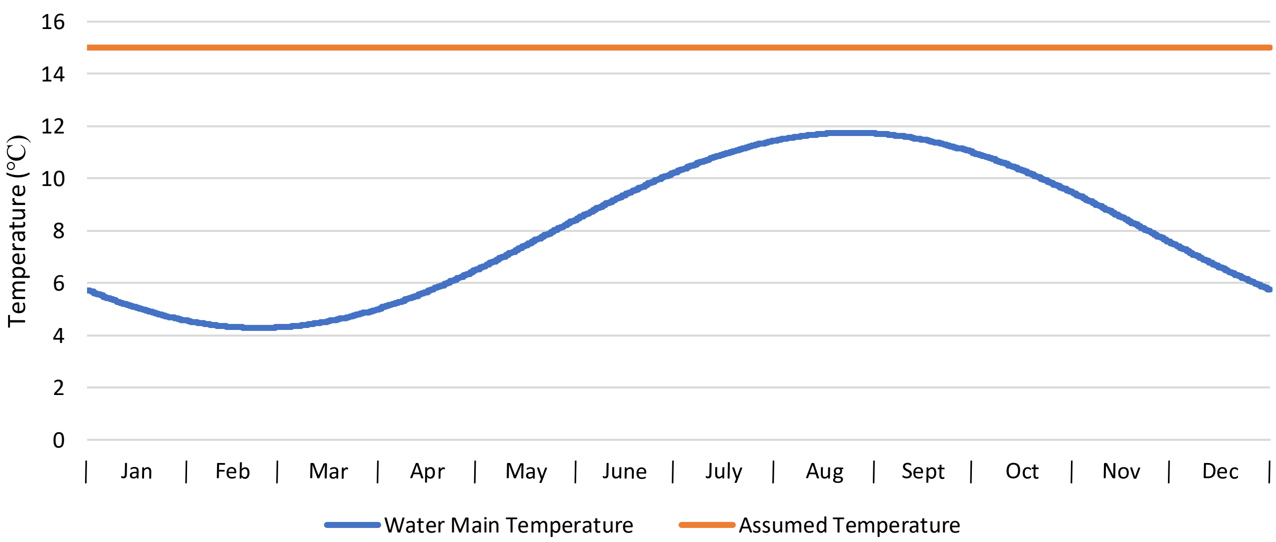


Fig. 17. Water main temperatures vs. assumed water temperature.

2) Ambient temperature

Hourly ambient temperatures (T_{amb}) for St. John's were obtained from Canada's meteorological data website [73]. Monthly hourly data was download for Jan-Dec 2019 and then compiled in one file and inserted into excel. The maximum ambient temperature was 27.9 °C, minimum -18.3 °C and average 4.86 °C. Figure 18 shows the hourly distribution while table IX shows the yearly and monthly daily averages (calculated in excel). These will be used later for f-chart calculations.

It should be noted that the term ambient is different with respect to the collector than it is with respect to the storage tank where the former is usually installed outside and the latter in the basement. Therefore, when discussing storage, the ambient temperature will be the basement temperature while for the collector it is the outside temperature and for the fluid inlet it is the basement/living space temperature.

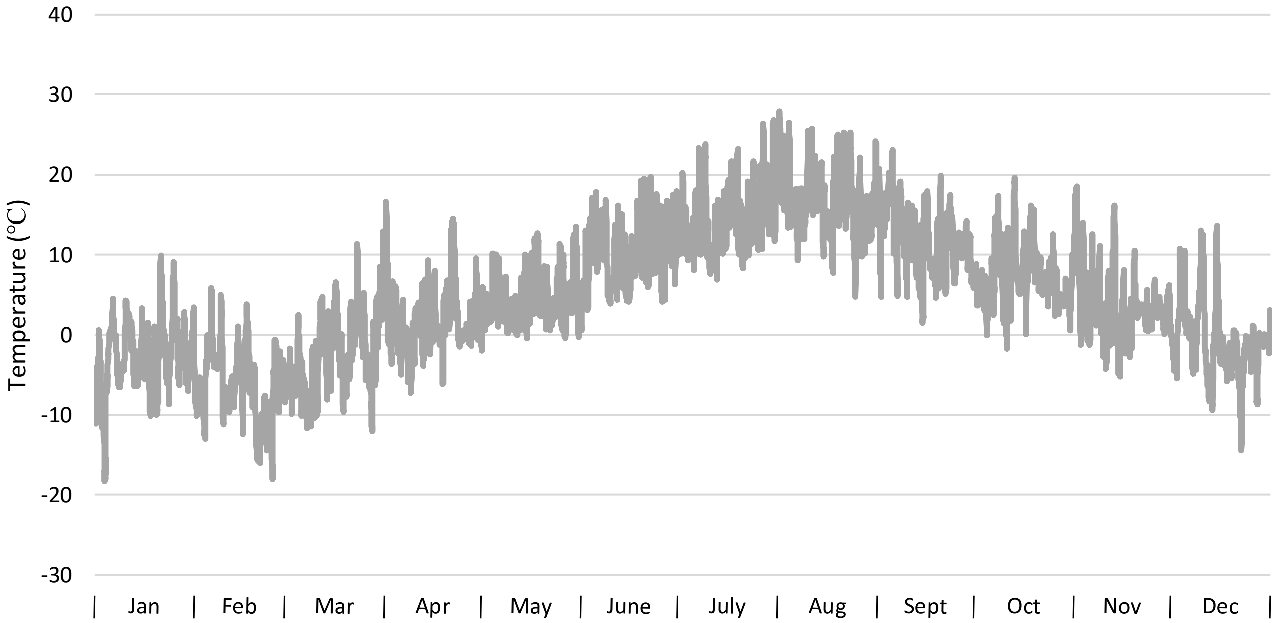


Fig. 18. Ambient (outside) temperature for the design location.

TABLE IX
AMBIENT TEMPERATURE (MONTHLY)

Time span	Temperature (°C)
Yearly	4.86
January	-2.67
February	-6.23
March	-2.40
April	1.66
May	3.89
June	10.14
July	14.46
August	16.80
September	11.80
October	7.23
November	3.72
December	-0.84

3) Collector slope and intercept

Considering AE-21 solar collector, $F_R U_L$ was obtained from AE-21 SRCC certification as -6.159 W/m^2°C while $F_R \tau\alpha$ was 0.749. According to the data sheet AE-21 transmissivity is over 90% with a selective absorber plate coating with an absorptivity α = 0.96 [25,26,66]. Therefore, the $\tau\alpha$ product is 0.864 making F_R = 0.867 and U_L = -7.1 W/m^2°C. According to B. Hodge [25], $\tau\alpha$ product for single glazed selective collector is 0.85. According to RETScreen $F_R \tau\alpha$ is the optical efficiency of the collector and ranges from 0.5 to 0.9, while $F_R U_L$, a measure of collector losses, ranges from 3.5 to 6 W/m^2°C for glazed collectors making AE-21 somewhat efficient but with slightly high losses [77].

4) Irradiance

Hourly irradiance (I_T) data is somewhat rare as most sources provide monthly averages. However, HOMER was used to obtain hourly irradiance data which can be found by exporting data from the DMap plot of the solar resource. The hourly data was then averaged to obtain table X. The yearly irradiance sum is 1160.7 kWh/m^2. Irradiance peaked in June at 5.26 kWh/m^{2}day and was lowest in December at less than 1 kWh/m^{2}day daily average.

The last variable, the collector's area, was inputted as 1.932 m^2 as given in the datasheet.

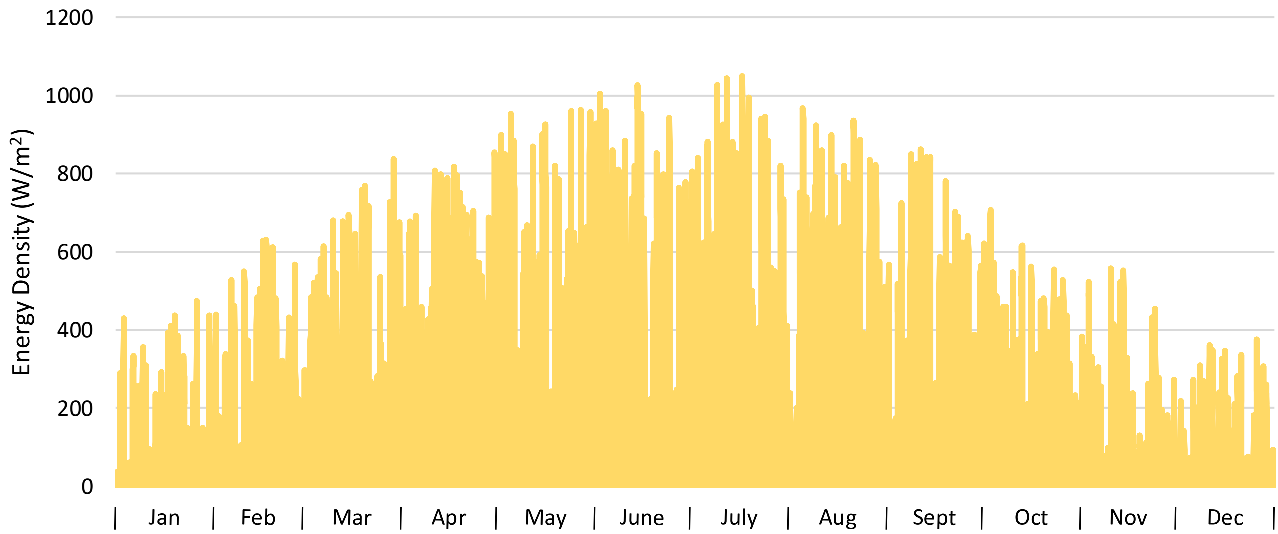


Fig. 19. Hourly Irradiance for the design location.

TABLE X
IRRADIANCE CALCULATED VALUES

Time Span	Irradiance Daily Average (kWh/m^{2}day)
January	1.16
February	1.91
March	3.13
April	4.01
May	4.93
June	5.26
July	5.08
August	4.57
September	3.50
October	2.28
November	1.31
December	0.94

Note: In efficiency and energy calculations no negative efficiency is assumed meaning that if the conditions are as such that the inlet fluid will lose energy if it traverses the collector then the controller will simply shut the system down during this time (such as low irradiance hours like early morning). This occurs due to irradiance being negatively proportional to the negative term ($F_R U_L \frac{T_{in}-T_a}{I_T}$) in the efficiency equation (eq 2). So as irradiance decreases the negative term increases up to a point where it can overtake the efficiency term and make it negative. Therefore, during this time efficiency will be assumed to be 0 i.e. system is turned off. This is accomplished by an if condition in excel. Another if condition is used to remove divide by 0 errors which occur when the irradiance is 0 W/m^2 (at night).

B. Thermal Load

The thermal load obtained from BEOPT (as reported earlier) was inputted into excel. The total load is 40.515 MWh. Figure 20 shows the hourly thermal load for both space and water heating since this "manual" calculation is for a combi system where both space heating and water heating are supplied by the same array of liquid flat plate collectors and utilize the same storage tank. Space heating in this case is utilizing the baseboard radiator technology where hot water passes through pipes (with fins) installed near the floor and the heat exchange from the water to the air results in space heating. The amount of energy required to heat the water to the required temperature is the space heating thermal load i.e., the heat transfer efficiency between water and air is accounted for.

It should be noted that when BEOPT (developed by NREL) generates the thermal load it is taking into account the house's existing storage (50 gallon water tank) and natural energy storage (walls and carpets) since insulation, window material, carpets, floor mass, natural ventilation, doors and other factors that influence natural energy storage were all inputted into BEOPT. Space and water temperature set points were also inputted (51.6 °C for water heating and 18.3 to 21.6 °C for space heating as provided by the owner). This means that BEOPT is calculating the amount of thermal energy required to maintain the house at the set point temperatures given the house's properties.

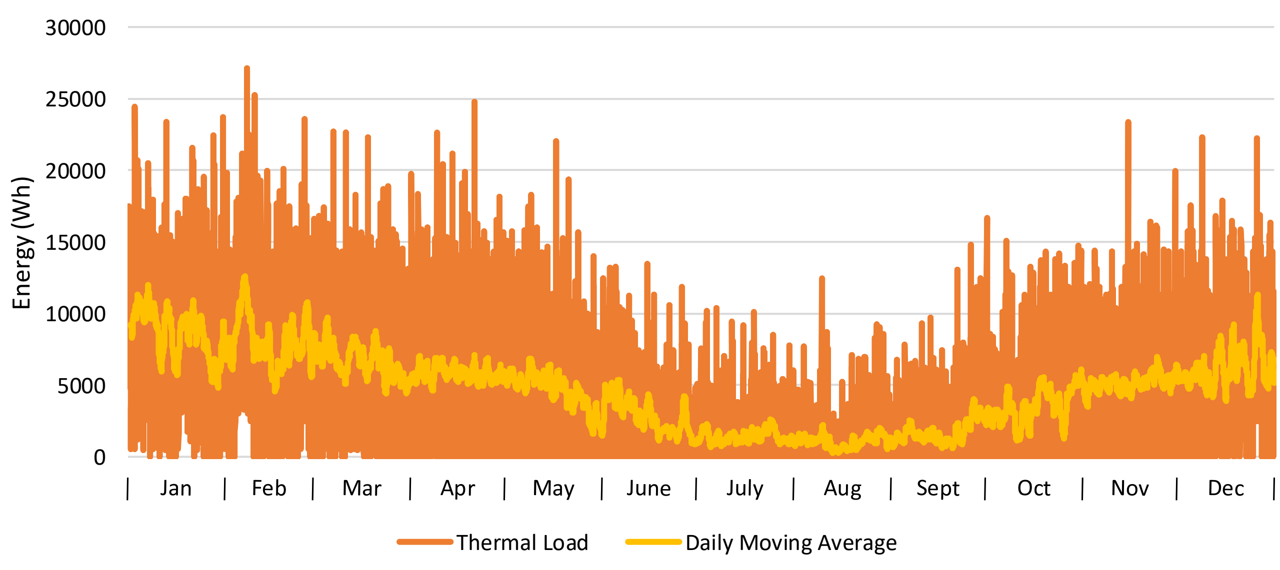


Fig. 20. Hourly thermal load.

C. Q_{useful} calculation

Given the previous inputs, hourly efficiency was calculated. Efficiency calculations were then used to calculate Q_{useful} using equation 3 for three scenarios; one where T_{in} is assumed at 15 °C, the 2nd where the temperature of the water main was used (dubbed actual-1) and the 3rd where the ambient temperature was used (dubbed actual-2).

The question is what is the temperature of the fluid at the inlet to the collector? This depends on whether the water is *used* or *consumed*. For space heating the water is used. i.e. water heated to 30 °C, for example, will go to the baseboard radiator and warm the air, dropping down to a temperature near ambient say 20 °C. then this near ambient temperature water will go back to the collector. Therefore, for space heating it is logical to use ambient temperature as the inlet tempeteráture. In this case we define the space heating load as the energy needed to raise the ambient water temperature to the temperature needed for space heating.

However, this is different for water heating where the water is consumed. i.e. water warmed from water main temperature to 30 °C will be consumed entirely and all of its energy will be useful in this case new water from water main will replace the consumed water and will go to the collector. Therefore in this case $T_{in} = T_{water\ main}$. Here we define the load as the energy needed to raise the temperature of water coming from the water main to the temperature needed for DHW. The water consumption of a 6 person household is between 300 and 360 Liters per day so it is not a small amount (according to RETScreen, BEOPT and NRCan [61]) . And the water has to be raised to a high temperature through the solar collector and auxillary oil burner.

The actual case is somewhere in between (a combination of) these two cases so it is imperative to calculate for both cases to provide an upper and a lower boundary. It should be mentioned, however, that the actual case will be closer to the lower boundary (actual-2) than the upper one (actual-1) since the rate at which water is used for space heating is higher than the rate at which it is used for hot water heating. In a stratified tank model, the temperature the collector fluid is in contact with (wether it is open loop or closed loop) is the temperature of the fluid at the bottom of the tank. i.e. the lowest temperature. In case of space heating only, the stratifcation of the tank will change so that ambient temperature water is at the bottom of the tank while for DHW only water-main temperature water will be at the bottom for any given hour. For a combi system some combination of these two cases will exist at any given hour.

1) Assumed inlet temperature

Q_{useful} of this scenario is shown in figure 21 and table XI. Useful energy output peaked in the summer reaching 232.6 kWh in July while December's output was only 14.22 kWh. The yearly output of 1 AE-21 collector is 1367 kWh/year or 3.75 kWh/day. This can be compared against AE-21 collector testing in Florida reported earlier which showed a daily output of 5.16 kWh/day. This higher figure is due to Florida exhibiting higher irradiance than St. John's but also due to the assumed inlet water temperature being too high (7 °C above the water main average temperature).

The percentage of solar energy captured (efficiency) is 60.97% which is well above that of PV systems (this is not the solar fraction which will be discussed later). This is assuming an open loop system. i.e. no heat transfer losses between collector fluid and water. This might be impractical for Canada's wintery climate, but this calculation is more conceptual and less practical. All minute details will be included in the simulations.

Since the annual thermal load stands at 40.515 MWh then 30 collectors are needed to satisfy the load (in toto disregarding mismatch between supply and demand). Since 30 collectors are needed and each collector is 1.932 m^2 in gross area, therefore, 57.25 m^2 of roof area is needed which is less than the 158.95 m^2 of rooftop space available as can be seen in figure 23 (obtained via google earth). Even if 3 m^2 mounting area per collector is used (as suggested by [61]), the array placement is feasible. A comparison between the output of 30 collectors and the thermal load is shown in figure 24.

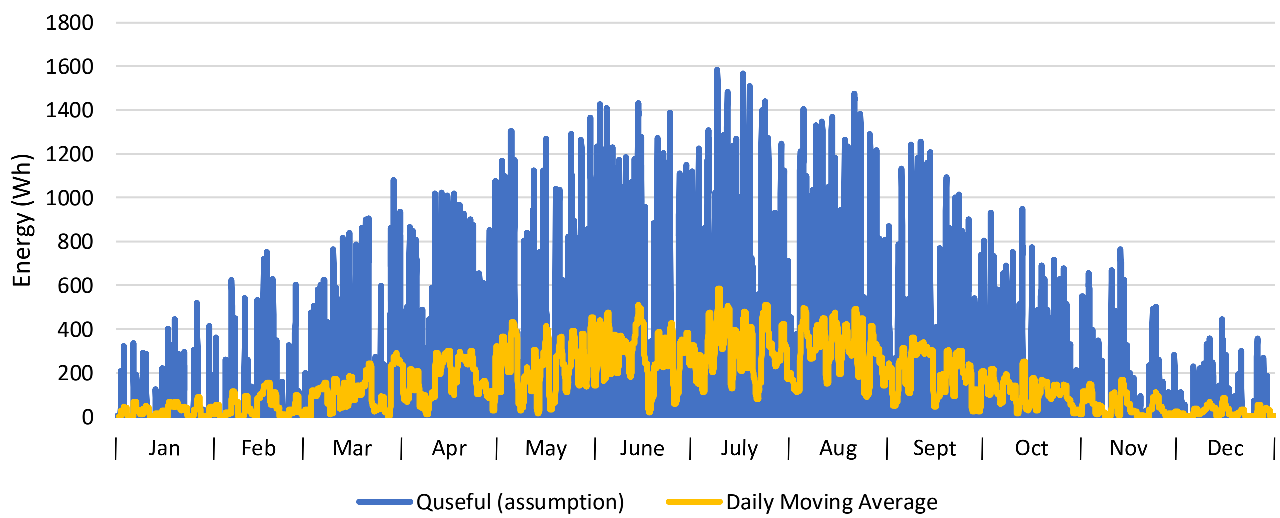


Fig. 21. Energy output of 1 collector for the assumed temperature.

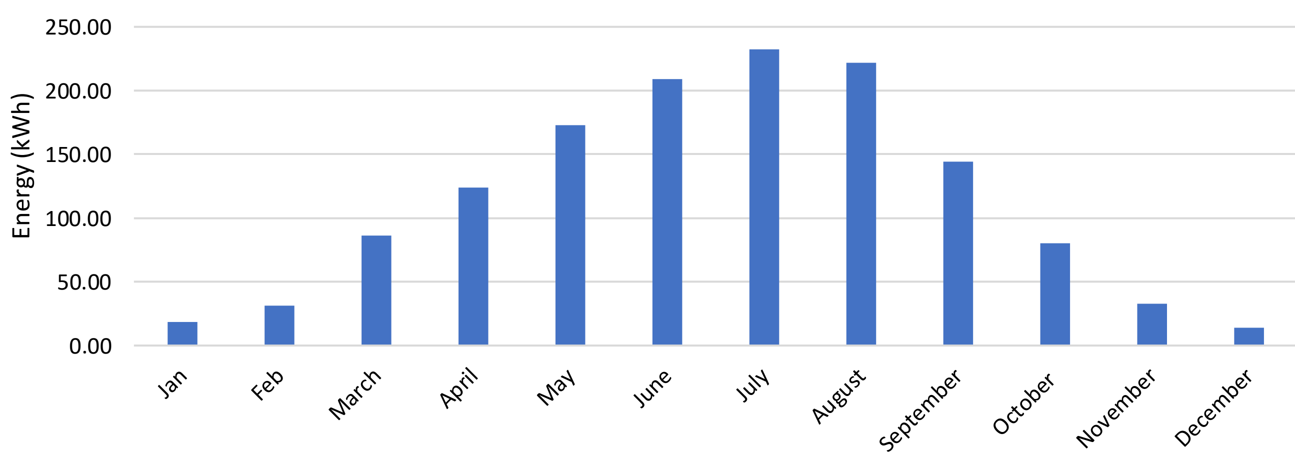


Fig. 22. Monthly cumulative energy output (1 collector) (assumed temperature).

TABLE XI
ASSUMED INLET TEMPERATURE CALCULATION RESULTS

Variable	Value
Qyearly (Wh)	1,367,343
Qyearly (kWh)	1367.34
Qdaily (kWh) (average)	3.75
% captured	60.97
Qjanuary (kWh)	18.58
Qfeb (kWh)	31.17
Qmarch (kWh)	86.05
Qapril (kWh)	123.77
Qmay (kWh)	172.74
Qjune (kWh)	209.26
Qjuly (kWh)	232.57
Qaugust (kWh)	221.78
Qsept (kWh)	144.36
Qoct (kWh)	80.18
Qnov (kWh)	32.66
Qdec (kWh)	14.22
Annual thermal load (Wh)	40,515,000
No. of panels required	29.63
Area required (m^2)	57.25
Area available (m^2)	158.95

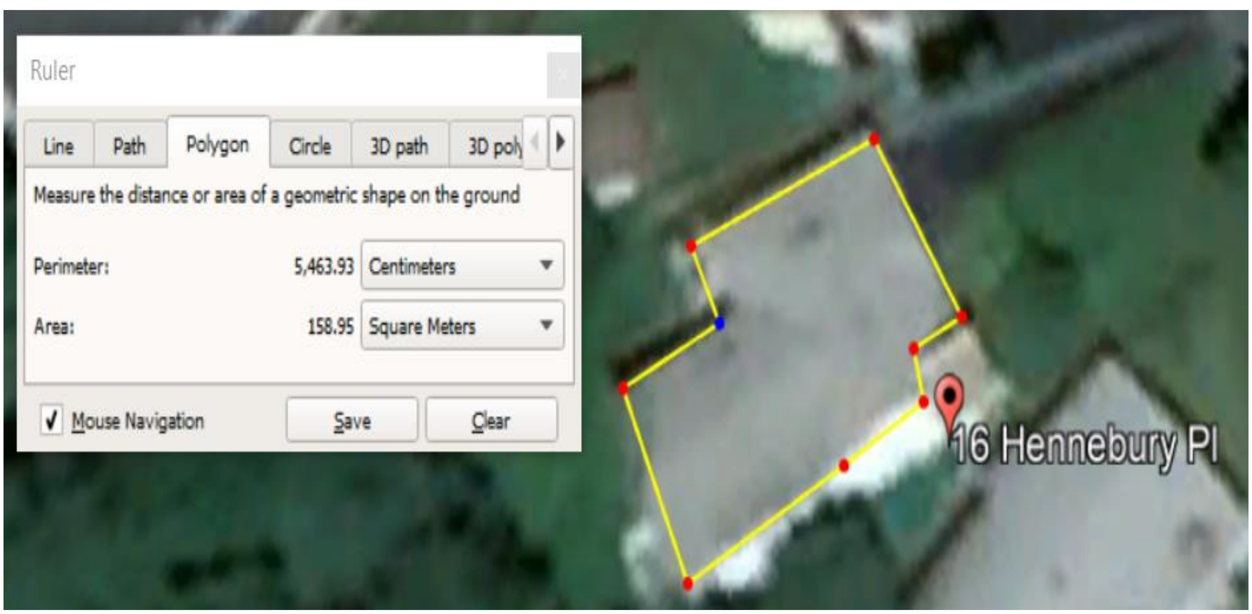


Fig. 23. Google earth view and calculation of the rooftop of the design house.

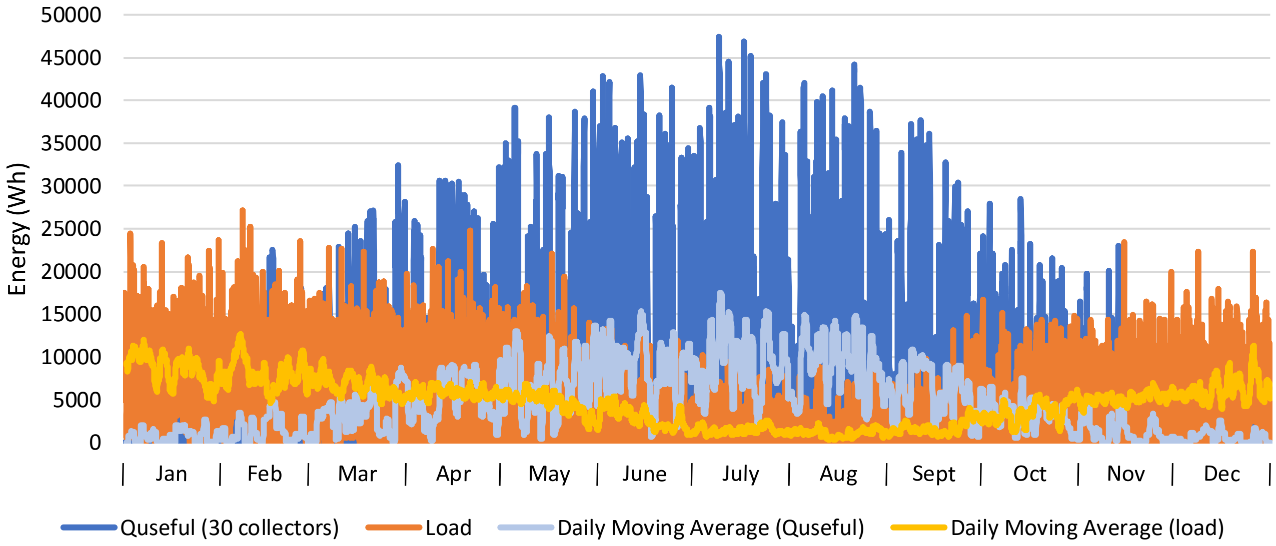


Fig. 24. Energy output of 30 collectors vs the thermal load.

i. *Energy Surplus and Deficit*

By subtracting the hourly thermal load from the hourly energy output of the 30 collector system (for Tin =15 °C), an energy surplus/deficit curve (figure 25) was obtained which shows the hours of the day where the output was higher than the load and the hours were it was lower. The figure shows that during winter the load is generally in deficit while during summer a portion of the energy generated is more than what the load needs (surplus). Concepts like Excess energy and Unmet load (used by HOMER) are therefore reasonable to examine. Where unmet load is the integration (summation) of all the hourly deficit in a year (1 - solar fraction) and excess energy is the summation of surplus. The value of the unmet load is a function of the number of collectors, useful collector energy and the load. The point where excess energy is equal to the unmet load is the same point where the overall energy generated by the solar system is equal to the overall load (shown in the coming figures.).

Unmet load is also directly related to the amount of heating oil the auxiliary heating unit will use. Simply, as the unmet load decreases (such as through storage or increasing the number of collectors) then the amount of oil needed will also reduce. By adding the energy deficit/surplus points we get an *overall excess* of 438 kWh (1.1 % of the overall load) because 30 collectors is higher than the 29.6 collectors needed. Note, this is not to say that unmet load and excess energy are not high it simply means that, overall, the energy generated has been sized to be 1% higher than the total load as the number of collectors was rounded up. Two count-if functions are used to separate surplus from deficit.

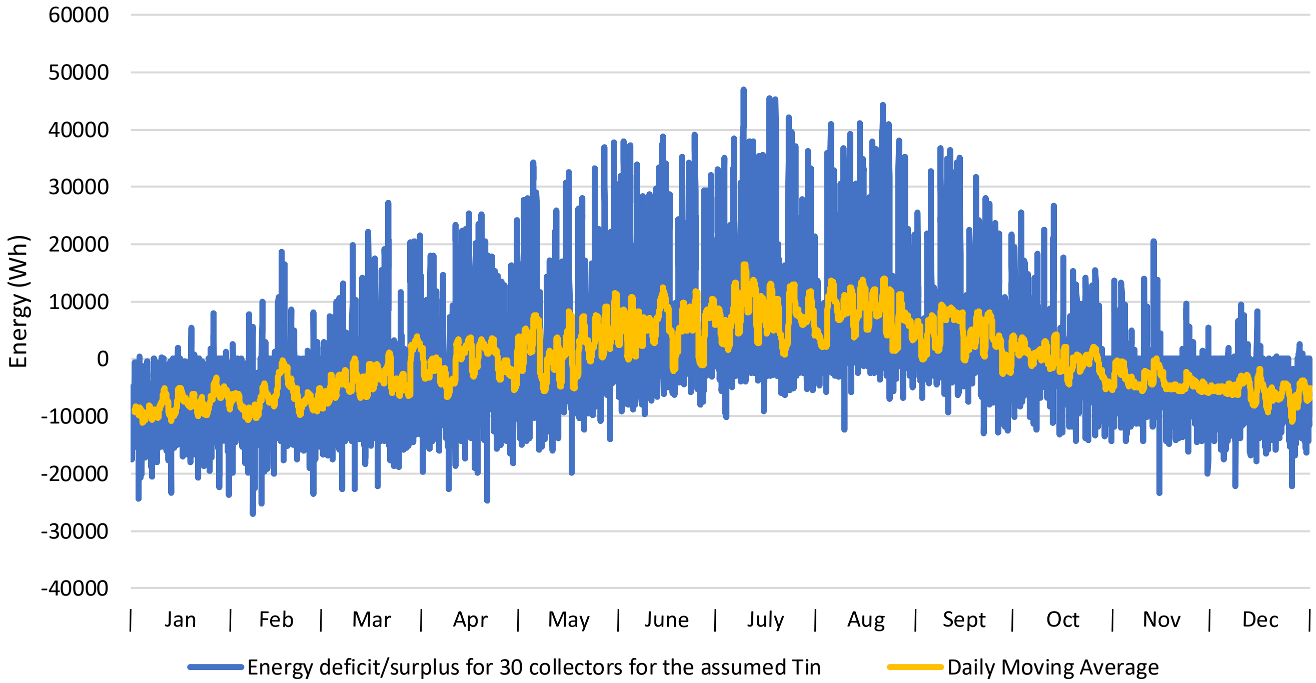


Fig. 25. Energy deficit/surplus for 30 collectors.

Due to the mismatch between supply from the solar thermal generation and consumption of the thermal load there is a lot of excess energy and unmet load. To quantify these, for 30 collectors (for Tin =15 °C), the unmet load is 29,015 kWh (71.6% of the total load) which if supplied by an auxiliary oil unit would lead to the consumption of 714.8 gallons of oil per year while the excess energy is almost equal at 29,462 kWh which represents 72% excess energy that would be unused if the set temperatures provided by the residence's owner are adhered to. Therefore, additional energy storage is valuable in trying to reduce these figures. However, how much energy storage is capable of improving these metrics within economic and spatial constraints will be examined later.

Table XII shows how excess energy increases and unmet load decreases as the number of collectors increases from 0 to 30. This is also shown in figure 26 where it can be observed that the point where the unmet load and excess energy curves meet is the point at which the number of collectors is the number needed to generate in toto the same amount of energy as the load requires (i.e. 40.515 MWh). In figure 26, unmet load, excess energy and relative abatement are to be read using the secondary y-axis (on the right) while oil needed using the primary y-axis.

Relative abatement is the oil gallons offset per 100 dollars invested (OGO) normalized using the maximum value of OGO which occurs when 1 collector is used (i.e., 3.91 gallons/100$). It is for plotting purposes as it is simply an indicator of the change in OGO. Oil needed reduces from 999.8 gallons needed for 0 collectors to 714.8 gallons needed for 30 collectors. However, the rate at which the price increases is higher than the rate at which oil is abated (since a lot of the energy generated doesn't serve the load) and so gallons of oil offset per 100$ invested (or relative abatement) decreases as the number of collectors increases. Note: abatement in this context refers to the amount of oil abated and therefore increasing it is a positive.

To calculate OGO the following relationship is used

$$OGO = \frac{Oil\ consumption_{old} - Oil\ consumption_{new}}{Price\ of\ system} = \frac{gallons}{1\ \$} * 100 = \frac{gallons}{100\ \$}$$

Since AE21 costs $745 and has a 30-year lifetime [48], payback period (PBP) can be calculated using the following equation

$$PBP = \frac{Price\ of\ system\ (\$)}{(solar\ fraction * load)\left(\frac{kWh}{year}\right) * \left(0.0246\ \frac{gallons}{kWh}\right) * price\ of\ oil\ gallon\ (\frac{\$}{gallon})}$$

In the denominator, the (solar fraction * load) is the amount of energy (in kWh) supplied successfully by the solar system per year (where solar fraction is 100% - Unmet load). This figure is converted from kWh to gallons of oil equivalent (here oil is used as a reference since the default system is oil based and not electricity based). In the last term, number of gallons is converted to their dollar equivalent making the final value of the denominator in $/year, which means that by dividing the system price by it, the number of years required to recover the investment (payback period) can be obtained (neglecting discount rate). As can be seen from Table XII the payback period ranges from 10 years to 28 years which are all feasible given the 30-year lifespan of the collector.

TABLE XII
RESULTS FOR 30 COLLECTORS FOR TIN = 15 °C

Variable	Value							
# of collectors	0	1	5	10	15	20	25	30
% excess energy	0%	13%	32%	46%	56%	63%	68%	72%
% of unmet load	100%	97%	89%	82%	78%	75%	73%	72%
Gallons of oil needed	999.8	970.6	885.1	818.2	777.9	750.4	730.3	714.8
Price of collectors ($)	0	745	3725	7450	11175	14900	18625	22350
Oil gallons offset per 100 dollars invested	0.00	3.91	3.08	2.44	1.99	1.67	1.45	1.27
Annual sum paid on oil ($)	2799	2718	2478	2291	2178	2101	2045	2002
Payback period (years)	-	9.67	11.77	14.79	18.12	21.48	24.84	28.19
Relative abatement	0%	100%	79%	62%	51%	43%	37%	32%

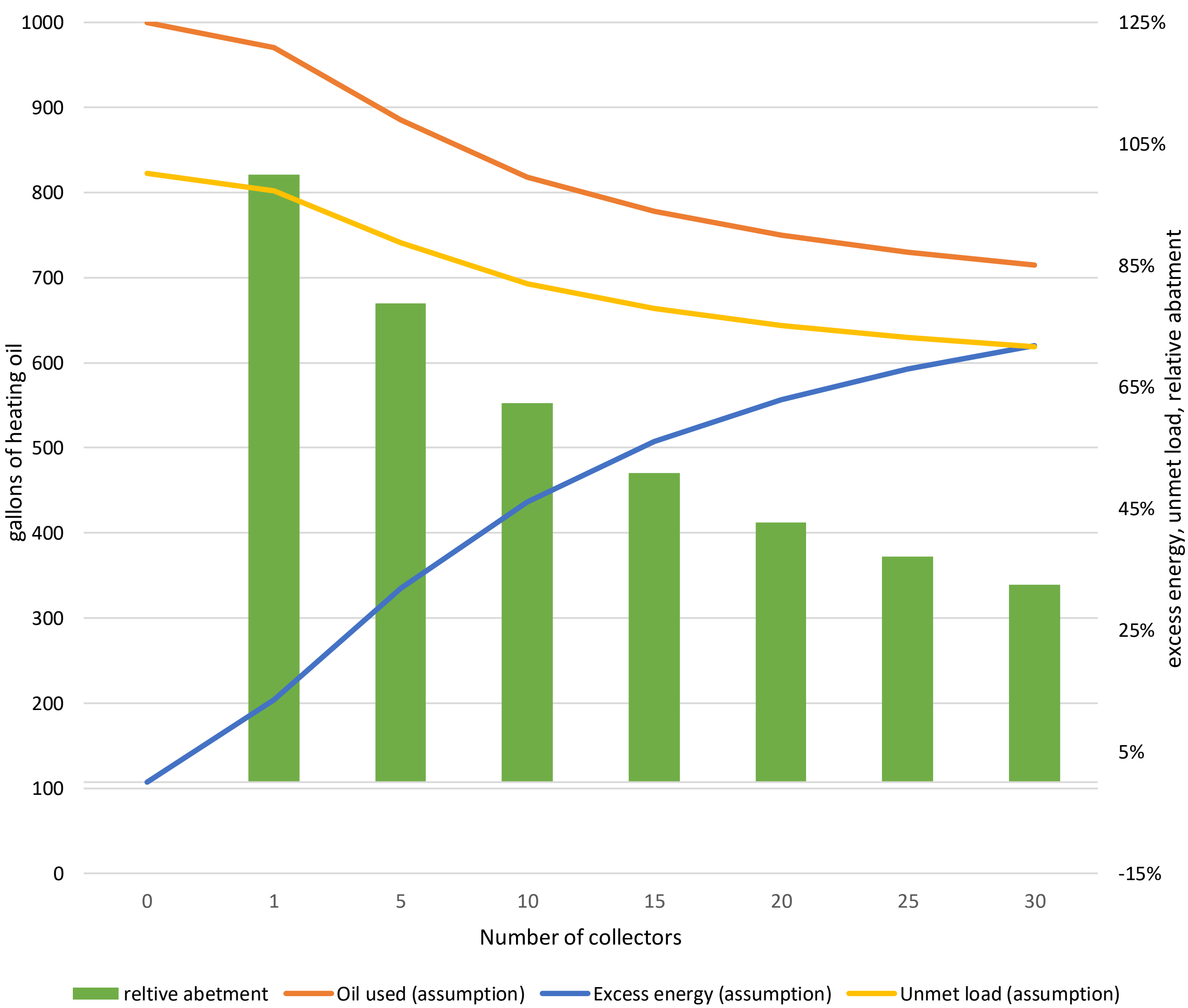


Fig. 26. Results for Tin =15 °C (assumption).

2) Actual-1 inlet temperatures

In this section, actual-1 inlet temperatures will be used instead of assumed temperature to provide the upper boundary; the results are shown in figure 25 and tables XIII and XIV. Initially, T_{in} was assumed as 15 °C but actual-1 temperatures (which are water main temperatures) were obtained from BEOPT (ranging from 4 to 12 °C) and inputted into excel and the result is an increase from 1367 kWh per collector per year to 1636 kWh. This means that the number of collectors needed to supply the thermal load is reduced from 30 to 25 collectors and the daily energy average of the collector increases from 3.75 to 4.48 kWh/day which is closer to the 5.16 kWh/day figure reported for AE-21 collector testing in Florida [51].

This implies that the lower irradiance in St. John's decreases the efficiency term (in eq 2) since it is indirectly proportional to the negative term which means decreasing irradiance increases the negative term which in turn decreases the efficiency parameter. However, inlet water temperature is directly proportional to the negative term which means that decreasing the inlet water temperature decreases the negative term which increases efficiency. Therefore, lower irradiance values in St. John's decreases the efficiency but lower water mains temperature increases it. The result is that yearly energy harvested from AE-21 collector in Florida is 1883 kWh while in St. John's it is 1636 kWh.

The increase in efficiency at colder inlet water temperatures is one of the reasons solar heaters are used to preheat the water before it goes into the auxiliary heater instead of the other way around (water entering the collector is colder).

The percentage of solar irradiance captured also increased from 61% to 72% (60-80% is the expected efficiency of solar collectors [72]) while the energy output of coldest months, January and December, increased by 90% and 88% respectively, energy output for the hottest month, July, increased by only 9.8%. This is very important as it illustrates that the use of actual-1 water inlet temperature (which varies seasonally) reduces the unmet load and excess energy by improving the energy output of the colder months more than the hotter ones which evens things out a bit as can be seen from figure 27.

Table XIV shows that all parameters have improved. A noticeable change is the amount of oil consumed by the auxiliary unit is reduced from 715 gallons to 665 gallons. As previously explained, this is due to the water main temperature being lower in winter which increases the efficiency during that season so that the difference between low irradiance low temperature seasons and high irradiance high temperature seasons for the calculation including actual-1 inlet water temperatures is less pronounced and therefore the difference between the load and supply (surplus and deficit) is less. The surplus energy was also reduced from 72% to 67% all the while the number of collectors decreases from 30 to 25 which reduces the price while increasing the number gallons of oil offset per dollar invested. The lowest payback period has also decreased to under 20 years and annual sum paid on oil reached as low as $1860 for the 25-collector system.

Figure 28 shows the calculation results at actual-1 vs assumed temperatures. The unmet load, excess energy and oil used curves shifted down (reduced) and to the left (for a fewer number of collectors) whereas the relative abatement curves increased since the amount of oil offset per 100$ invested increased. Here the relative abatement is defined in terms of the old abatement to find the percentage increase in value.

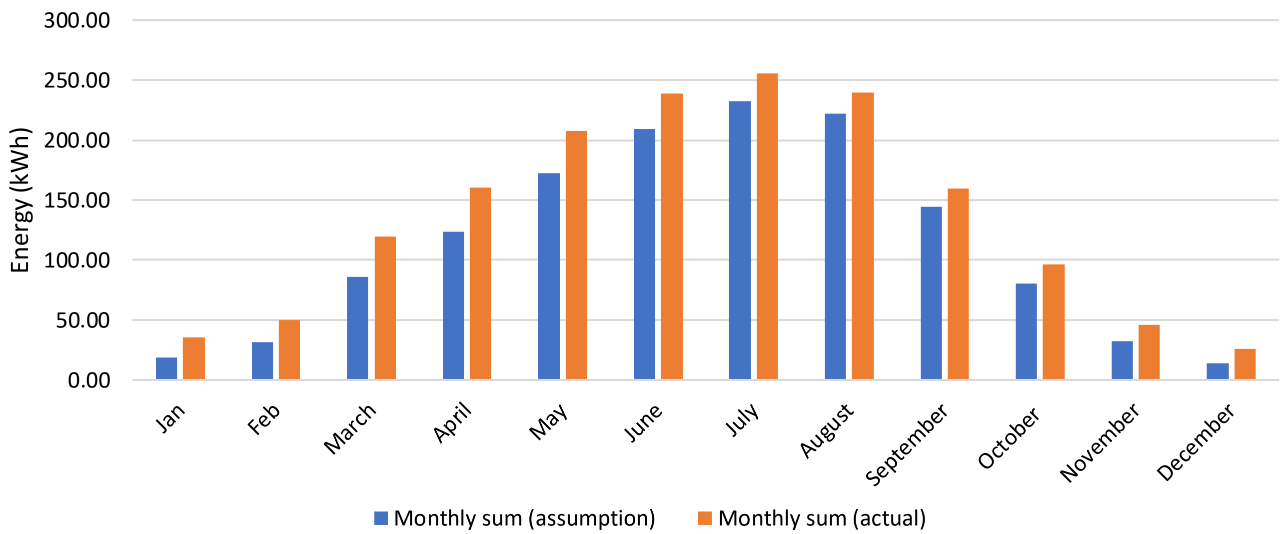


Fig. 27. Quseful Monthly sum assumption vs actual-1.

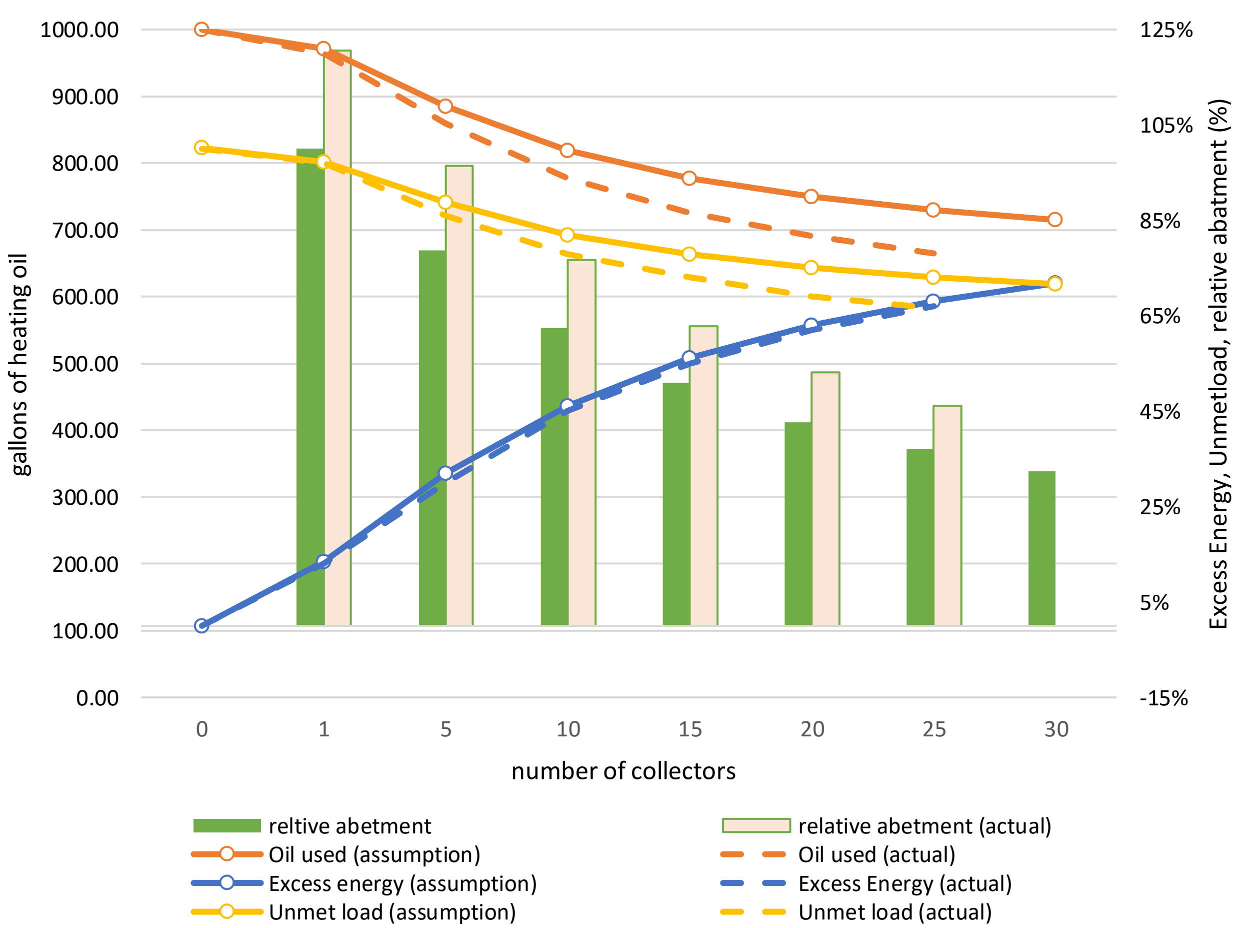


Fig. 28. Results for assumed vs actual-1 inlet temperatures.

TABLE XIII
ACTUAL-1 TEMPERATURE CALCULATION RESULTS

Variable	Value
Qyearly (Wh)	1,635,898
Qyearly (kWh)	1635.89
Qdaily (kWh) (average)	4.48
% captured	72.95
Qjanuary (kWh)	35.28
Qfeb (kWh)	50.05
Qmarch (kWh)	119.78
Qapril (kWh)	160.42
Qmay (kWh)	207.78
Qjune (kWh)	238.70
Qjuly (kWh)	255.53
Qaugust (kWh)	239.79
Qsept (kWh)	159.72
Qoct (kWh)	96.23
Qnov (kWh)	46.30
Qdec (kWh)	26.27
Annual thermal load (Wh)	40,515,000
No. of panels required	24.76
Area required (m^2)	47.84
Area available (m^2)	158.95

TABLE XIV
RESULTS FOR ACTUAL-1 TEMPERATURES

Variable	Value						
# of collectors	0	1	5	10	15	20	25
% excess energy	0%	13%	30%	45%	55%	62%	67%
% of unmet load	100%	97%	86%	78%	73%	69%	67%
Gallons of oil needed	999.78	964.64	859.3	776.6	725.5	690.5	664.77
Price of collectors ($)	0	745	3725	7450	11175	14900	18625
Oil gallons offset per 100 dollars invested	0	4.72	3.77	2.996	2.45	2.076	1.799
Annual sum paid on oil ($)	2799.4	2701.0	2406.0	2174.5	2031.4	1933.4	1861.4
Payback period (years)	-	8.89	9.52	12.12	14.81	17.20	19.95
Relative abatement	0%	121%	96%	77%	63%	53%	46%

By changing the value of the assumed temperature from 15 ℃ to 8 ℃ (water main temperature average) the results on the different metrics (unmet load, excess energy, energy output…etc) are 1-2% higher than the ones provided in this section. So, the usage of averages in simplified solar thermal calculations is fairly accurate.

i. *Collector temperature*

By going back to eq (1) and plugging in the value of the efficiency and the other parameters T_{avg} which is the collector's average temperature for each hour can be obtained. The results are shown in figure 29. The minimum collector temperature is -18.3 °C, maximum 29.5 °C and average 7.4 °C. Therefore, the use of solar glycol mixture is important as it can withstand temperatures as low as -35 °C [67].

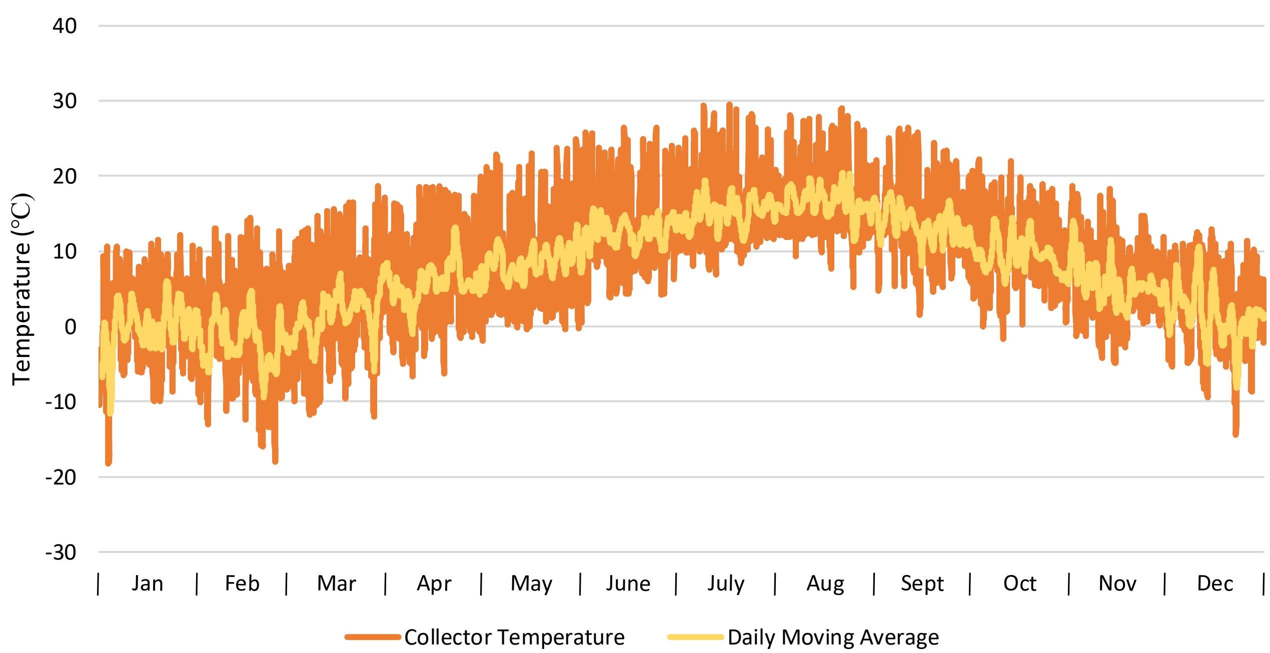


Fig. 29. Average collector Temperature.

3) *Actual-2 inlet temperatures*

Actual-1 water temperatures were used to establish the upper boundary a combi system can achieve. This was done assuming all water is consumed so inlet fluid temperatures are the water main temperatures (fluid replenishment) however the lower boundary need also be established. In this case it is assumed that only space heating takes place (no fluid replenishment) and therefore the temperature of the fluid entering the collector is near the temperature of the fluid returning from the radiator which is near ambient.

In this extreme, Yearly energy yield is 1125 kWh which is over 240 kWh less than the output of the collector at the assumed temperature and 511 kWh less than actual-1 yearly energy yield. The percentage of solar radiation captured is 50.2%. In this scenario, 36 collectors are required to (in toto) match the load. The area required also increases to 70 m^2 (which is still feasible). When compared with the assumed temperature all the months saw a decrease in energy output for example January saw a 40% (-7.3 kWh) decrease while July saw a 15% (-33 kWh) decrease in energy output (figure 30). Figure 31 shows a contrast between the assumed and actual-1 and actual-2 inlet temperatures. Figure 32 shows the collector temperature for actual-2 which increased to 11.67 °C average temperature (4.3 °C increase over actual-1 collector average temperature).

Table XV and figure 33 show how for actual-2 boundary case excess energy, unmet load, gallons of oil needed, payback period and relative abatement change as the number of collectors increase from 0 to 36. It can be seen that opposite of actual-1 the curve has shifted up and to the right relative to the assumed case. It is therefore generating more excess energy and unmet load for a higher number of collectors (thereby inciting higher oil consumption). The relative abatement bar chart also shows lower values compared to the base (assumed) case. The payback period for this case is also much higher crossing the line of feasibility for systems with 25 or more collectors. Figure 34 compares actual-1 (upper boundary) with actual 2 (lower boundary) the actual solution is expected to be within these boundaries but more likely to be closer to the lower boundary as space heating rate is higher. Both relative abatement curves are referencing the assumed temperature relative abatement value as a baseline value.

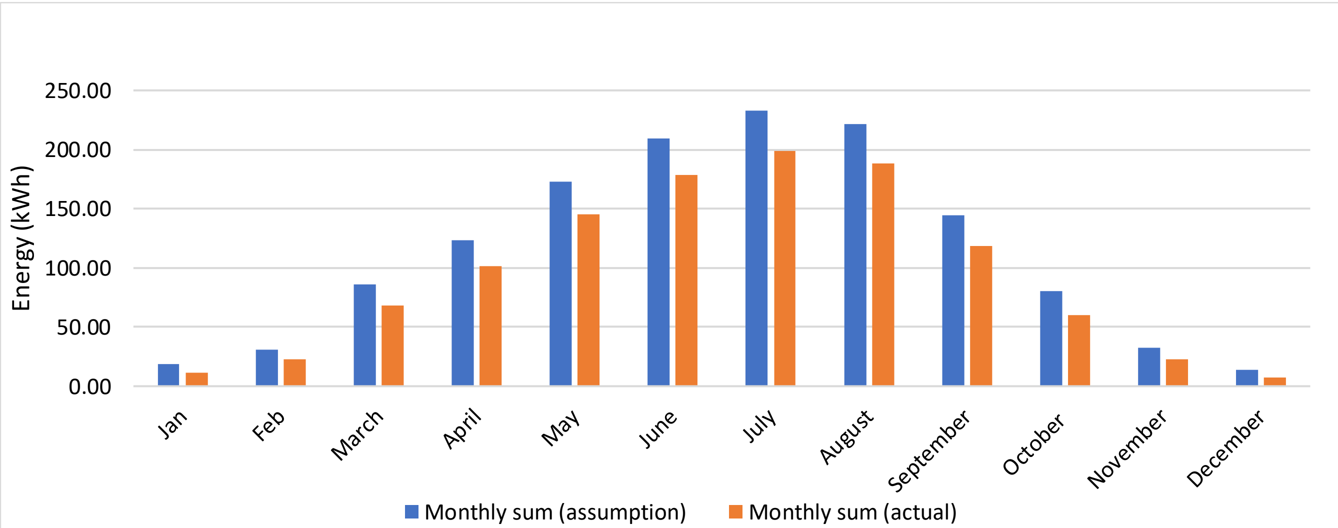


Fig. 30. Quseful (kWh) (monthly sum) assumption vs actual-2.

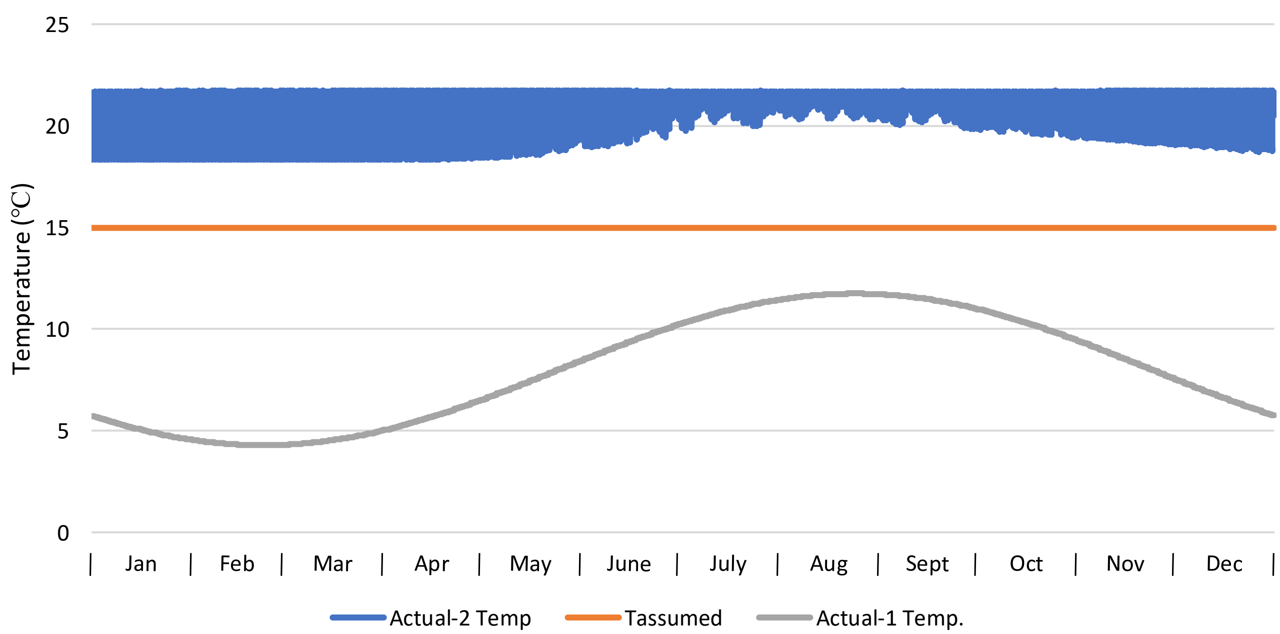


Fig. 31. Inlet fluid temperature.

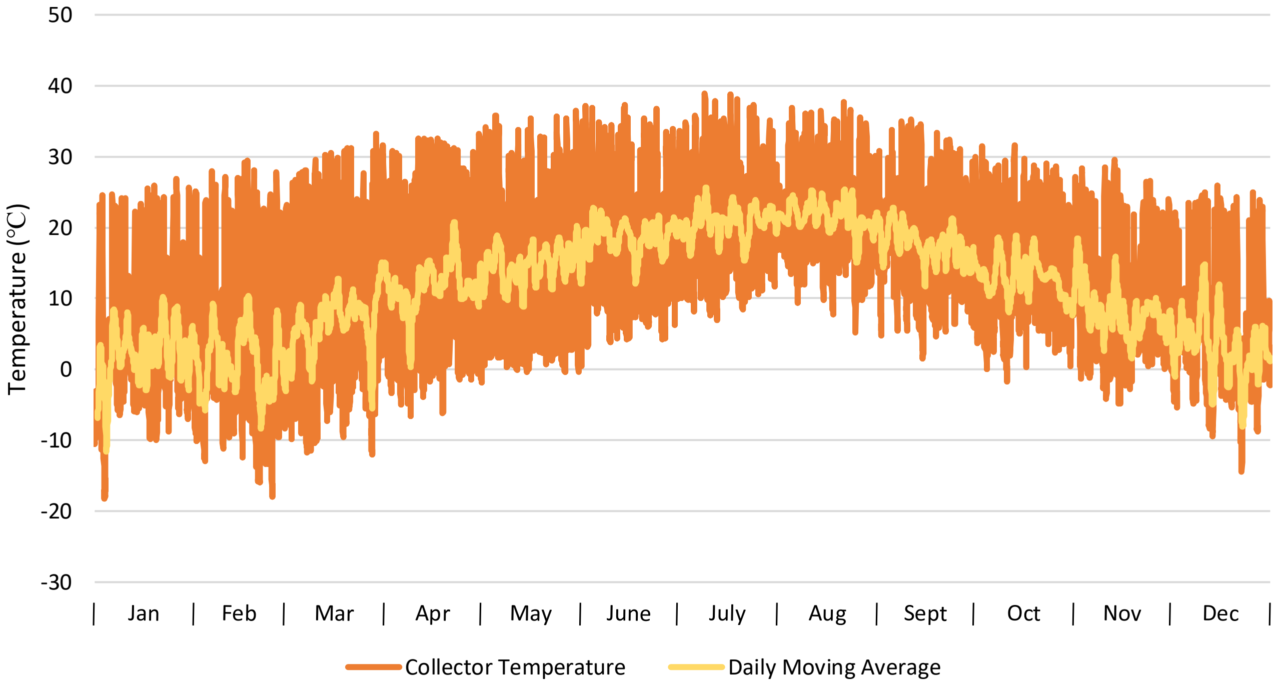


Fig. 32. Average collector temperature (actual-2 case).

TABLE XV
RESULTS FOR ACTUAL-2 TEMPERATURES

Variable	Value								
# of collectors	0.00	1.00	5.00	10.00	15.00	20.00	25.00	30.00	36.00
% excess energy	0%	13%	32%	46%	56%	63%	68%	72%	75%
% of unmet load	100%	97%	91%	85%	82%	80%	78%	77%	75%
Gallons of oil needed	999.8	975.8	905.4	849.8	816.4	794.0	777.8	765.2	753.6
Price	0	745	3725	7450	11175	14900	18625	22350	26820
Annual sum paid on oil	2799	2732	2535	2380	2286	2223	2178	2143	2110
Payback period	-	9.67	14.34	17.94	21.96	26.07	30.18	34.27	39.16
Oil gallons offset per 100 dollars invested	0.00	3.22	2.53	2.01	1.64	1.38	1.19	1.05	0.92
Relative abatement	0%	82%	65%	51%	42%	35%	30%	27%	23%

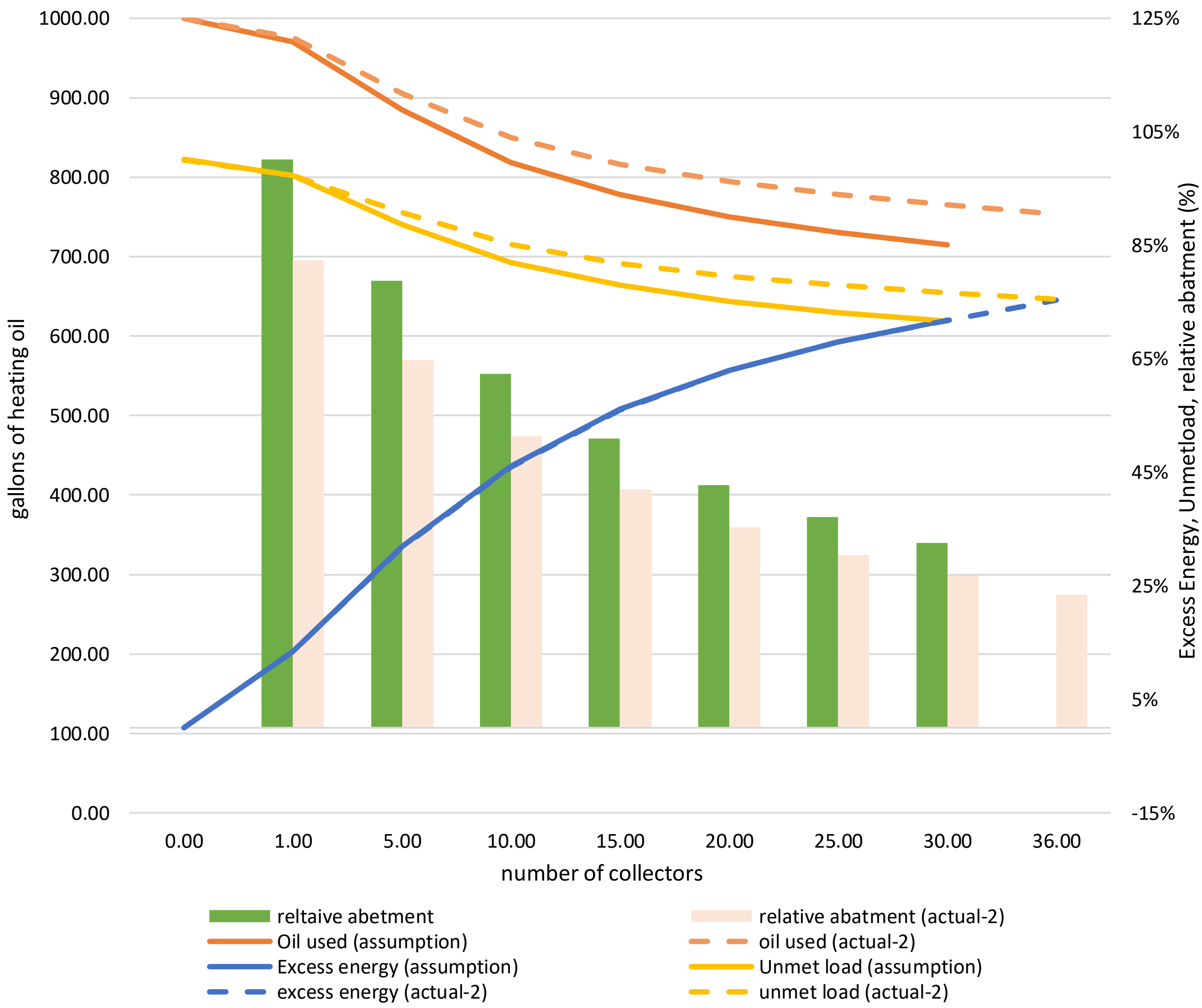


Fig. 33. Results for assumed vs actual-2 inlet temperatures.

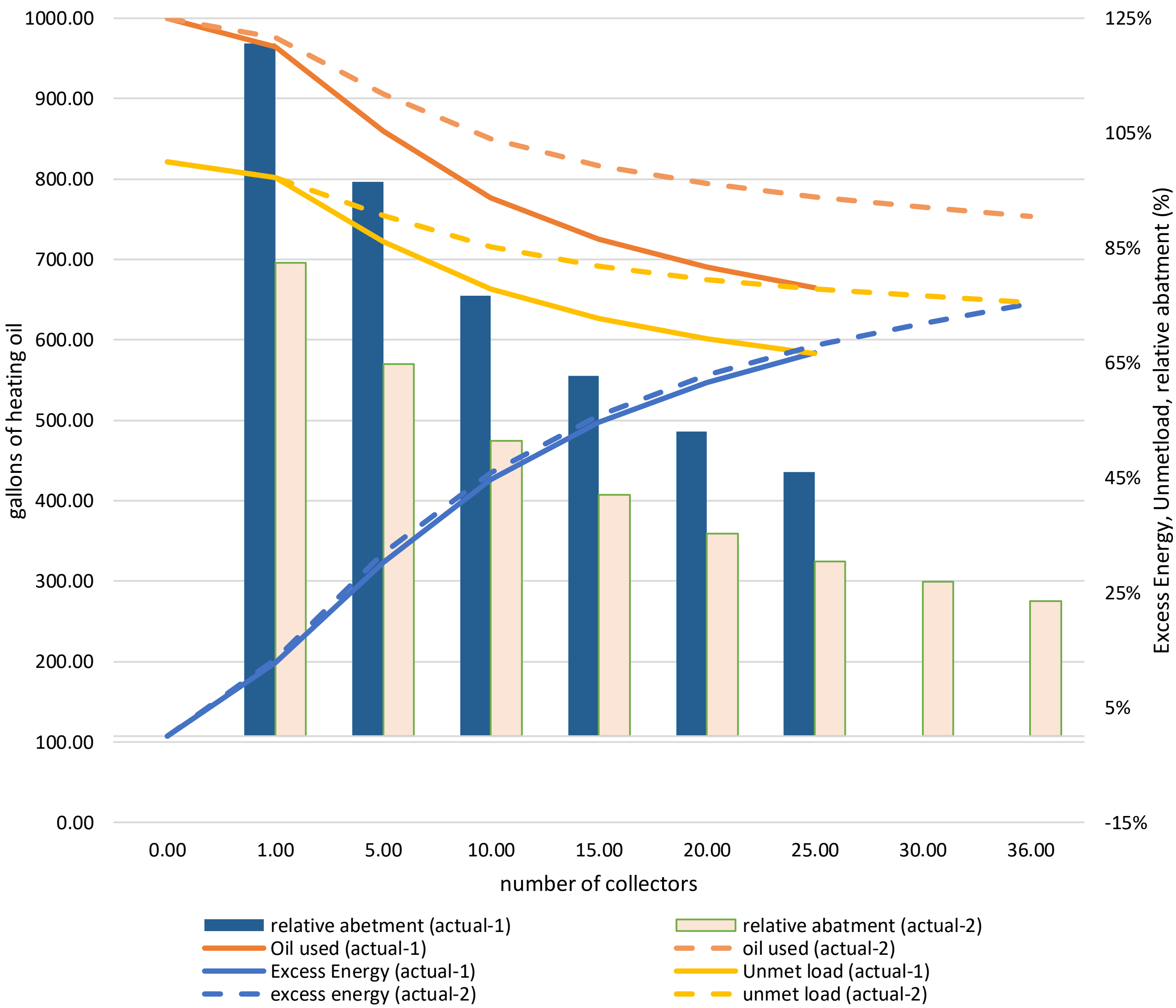


Fig. 34. Results for actual-1 vs actual-2 inlet temperatures.

D. Energy storage

1) Actual-1

As explained earlier, there is room for the contribution of energy storage since the unmet load and excess energy parameters are quite high. In this section a sensible heat unstratified water storage is examined. This section will be reported for actual-1 boundary case. The equation used to calculate the storage temperature; the energy delivered by the storage as well as the losses was obtained from John Duffie's Solar engineering of thermal processes [60]

$$T_s{}^+ = T_s + \frac{\Delta t}{mc_p} * \left(Q_u - L_s - (UA)(T_s - T_b)\right) \tag{5}$$

Where

$T_s{}^+$ is the storage temperature at the end of the time period (1 hour) (℃)

T_s is the storage temperature at the beginning of the storage period (1 hour) (℃)

Δt is the time period which in this case is 1 hour

m is the mass of the storage water (kg)

c_p is the specific heat of the storage medium (4190 J/kg ℃)

Q_u is the rate of addition of energy to storage (collector output) (J)

L_s is the rate of removal of energy from storage (load) (J)

UA is the product of heat loss coefficient and area product

T_b is the ambient temperature of the basement where the storage is placed.

$$Tank\ losses = \left(Q_u - L_s - (UA)(T_s - T_b)\right) \tag{6}$$

Initial assumed temperature is 60 ℃ but it could be any value as it has negligible impact over overall results. UA product is assumed at 11.1 W/℃ as given by [60]. Different values of water mass (presented earlier) are evaluated. Initially, 1500 kg mass was assumed based on [60]. Two nested if conditions are used to limit the temperature of the water to between 5 ℃ and 95 ℃ since the phase change temperature of water limit its application [53]. Practically this means that for example if the temperature of the water in the tank reaches 95 ℃ then no more energy should be added to it as this would lead to evaporation. So, the surplus energy in this scenario would have to be utilized differently (such as fed to a dump load). This is just an assumption as more practical water min/max temperature can be adopted. The last term in the brackets $\left(Q_u - L_s - (UA)(T_s - T_b)\right)$ can be used to calculate tank losses [60].

Before applying equation 5, hourly energy storage and deficit values were integrated to produce figure 35. The figure shows that the ideal storage for this application is seasonal. In the blue curve we can see that overall, the 1st quarter of the year runs at a deficit reaching -12.24 MWh around March (speaking conceptually of course as no energy can be withdrawn from a system that doesn't have it in store). From March to June an overall surplus can be observed which compensates for the earlier accumulated deficit reaching 0 kWh at the middle of the year. In the 3rd quarter the surplus continues reaching its peak of 7.42 MWh at the end of fall. As winter begins the surplus stored energy starts to be withdrawn until the stored capacity reaches 0 by the end of the year.

This shows an overall pattern, daily or weekly fluctuations in energy exist however relatively small they may be. If the initial storage was 12.24 MWh instead of 0 the curve will shift up, start at 12.24 MWh and end at it as well making the system's power rating 19.66 MW and the curve would never cross 0 on the y-axis meaning the storage would be self-sustaining. If we assume a water temperature difference of 75℃ then the mass required to sustain this operation is over 225,000 kg which if used to calculate the solar fraction results in 2% unmet load.

The authors of [63] calculated energy storage mass by basing it on excess energy. The total amount of excess energy in the *summer* was used to determine the mass of the system through the sensible heat equation where the temperature difference used was 75°C. the study considered May to September as summer. if a similar approach is taken for this project then the mass of the storage is 239,800 kg (a 6.5% difference). Testing this value in excel produced a solar fraction of 98%.

This is assuming a very large lossless form of storage which is most likely not practically obtainable so this demonstration only serves to show what an ideal storage would look like. More practical sizes will be discussed next.

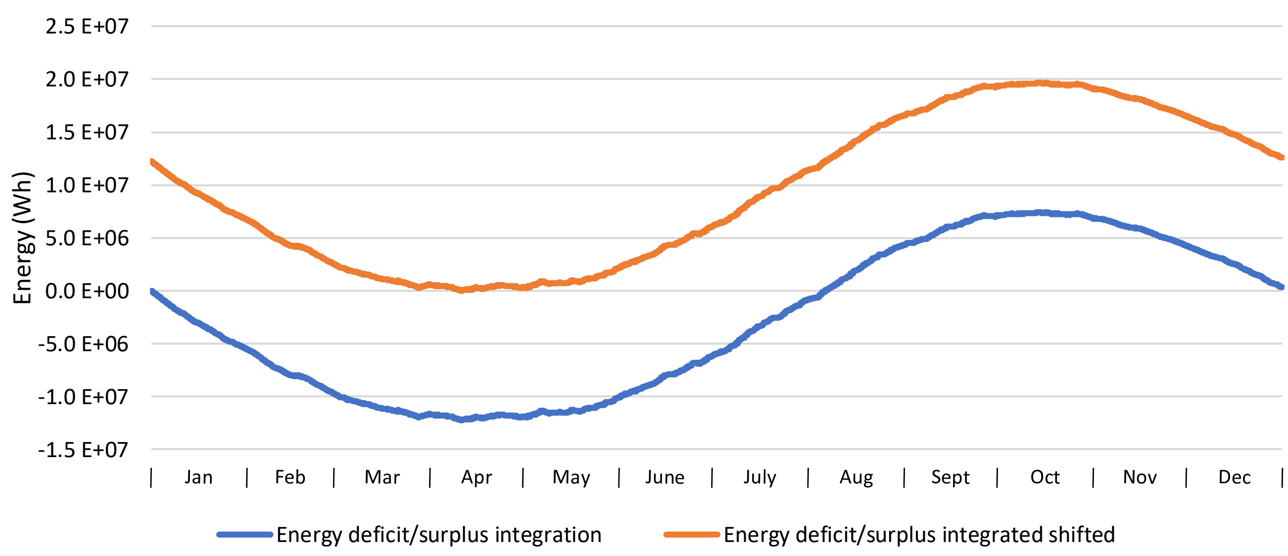


Fig. 35. Energy deficit/surplus integration.

By applying equation 5 to the data, figure 36 was obtained which shows the storage temperature for every hour in the year. The upper temperature limit condition can clearly be seen. In figure 37 the hourly tank losses for a 1500 kg are shown which can be calculated using equation 6 or using equation 4. As the temperature of the storage tank increases so do the losses reaching a peak of -814 W in summer.

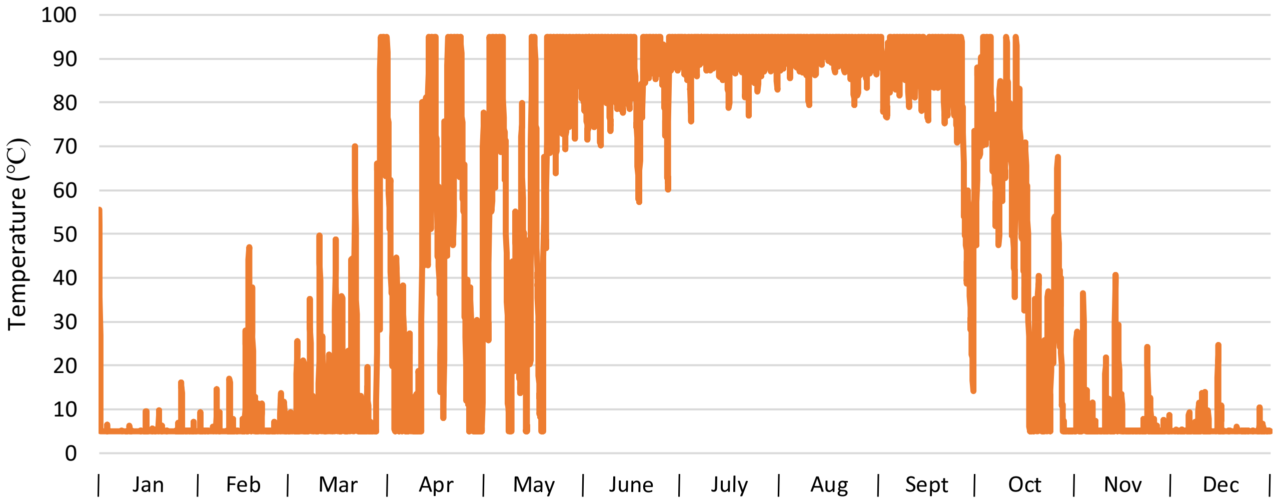


Fig. 36. Storage temperatures.

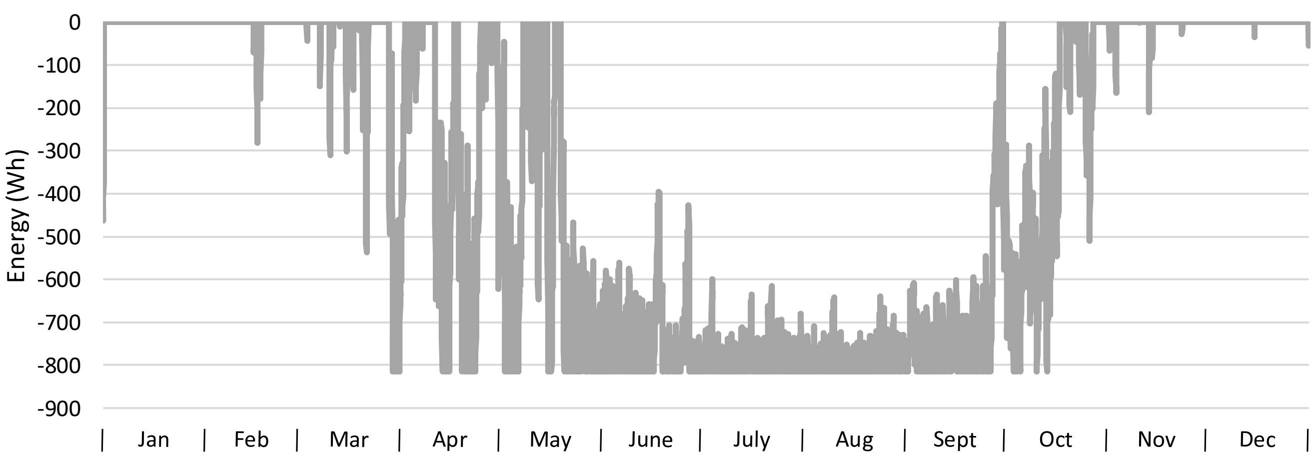


Fig. 37. Tank losses.

Based on the tank sizing review provided earlier in this report, 5 tank sizes were justified by different sources for different reasons. These tank sizes and their associated solar fractions are shown in table XVI and figure 38. The results show that with storage the solar fraction can at most reach 58% given economic constraints. However, given spatial constraints it is most likely that the 270-liter (60-gallon) tank is the reasonable choice. Which agrees with the solar fraction range of 30-60% given by [65].

TABLE XVI
STORAGE RESULTS (ACTUAL-1)

Variable	Value					
Tank size (kg)	0.1	1	270	1500	3622.5	9660
Solar fraction	33%	34%	46%	54%	56%	58%
Oil needed	664	661.6	542	462.8	442	418
Source	0	1	60-gal tank	Duffie problem	Duffie 75 L/m^2	Duffie 200L/m^2

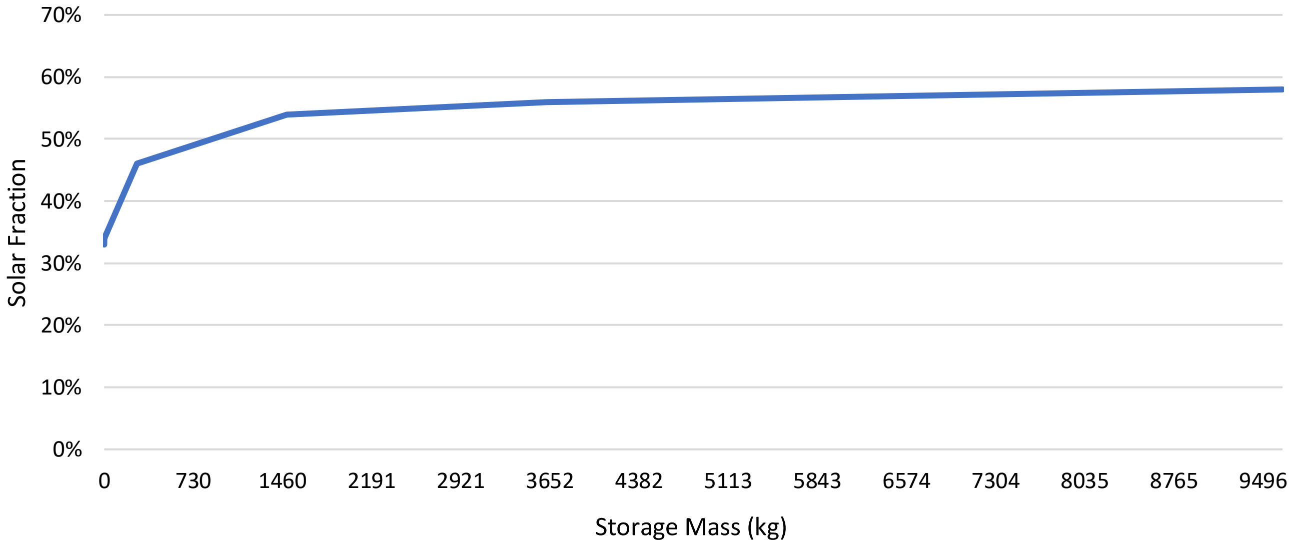


Fig. 38. Solar fraction vs storage mass (actual-1).

As a proof of concept, we can assume a fluid with water's specific heat and density but without its evaporation phase change temperature. For this imaginary fluid, the upper temperature can be 1000 ℃. In this case for 1500 kg storage mass the unmet load becomes 31% which is a 23% reduction from the actual case. The maximum temperature the fluid will reach is 821 ℃ and the maximum tank loss is 8875 Wh which is over 1000% increase.

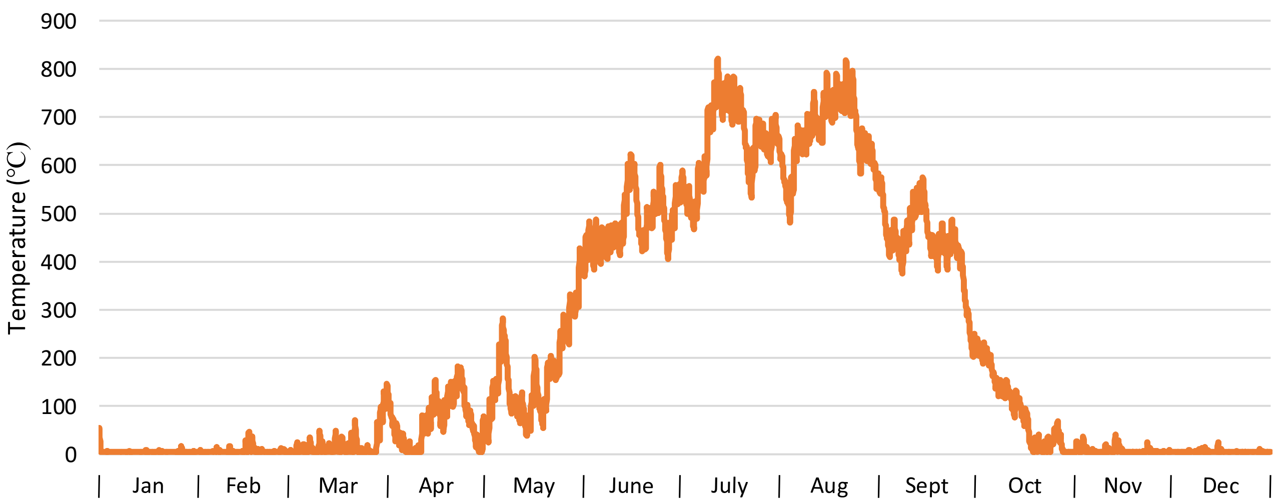


Fig. 39. Storage temperature (imaginary fluid).

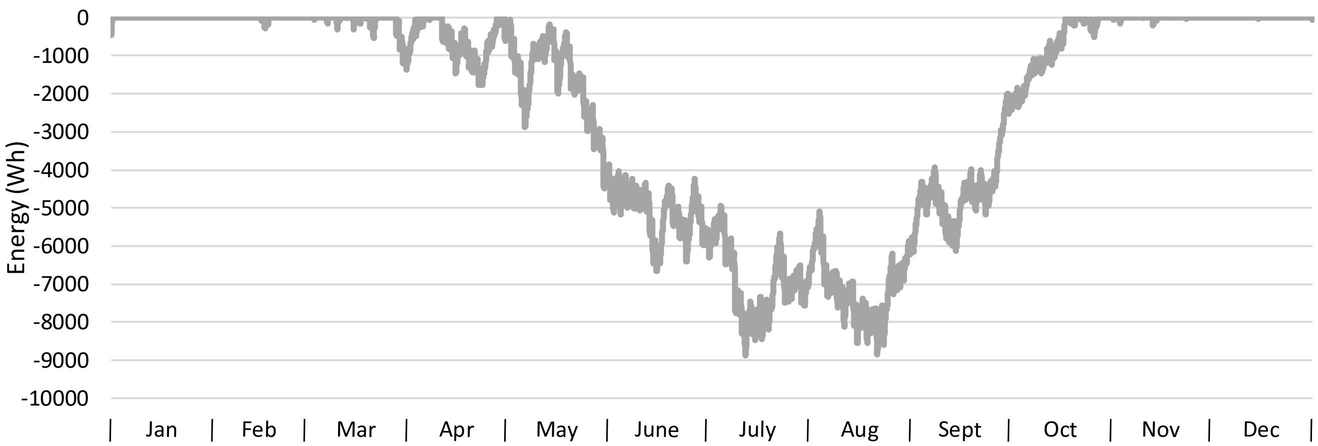


Fig. 40. Tank losses imaginary fluid.

2) Actual-2

The previous section was obtained for actual-1 scenario. It can easily be repeated for actual-2 scenario which results in figure 41 and table XVII. The maximum value of sola fraction that can be obtained within economic constraints is 53%. The amount of storage based on excess energy method [63] is 289,802 L and from integration 313,488 L both of which result in 98% solar fraction. All other details can be replicated but would take up too much space.

TABLE XVII
STORAGE RESULTS (ACTUAL-2)

Variable	Value					
Tank size (L)	0.1	1	270	1500	5216.4	13910.4
Solar fraction	25%	25%	36%	46%	50%	53%
Source	0	1	60-gal tank	Duffie problem	Duffie 75 L/m^2	Duffie 200L/m^2
Oil needed (gal)	753	750	634	538	499	466

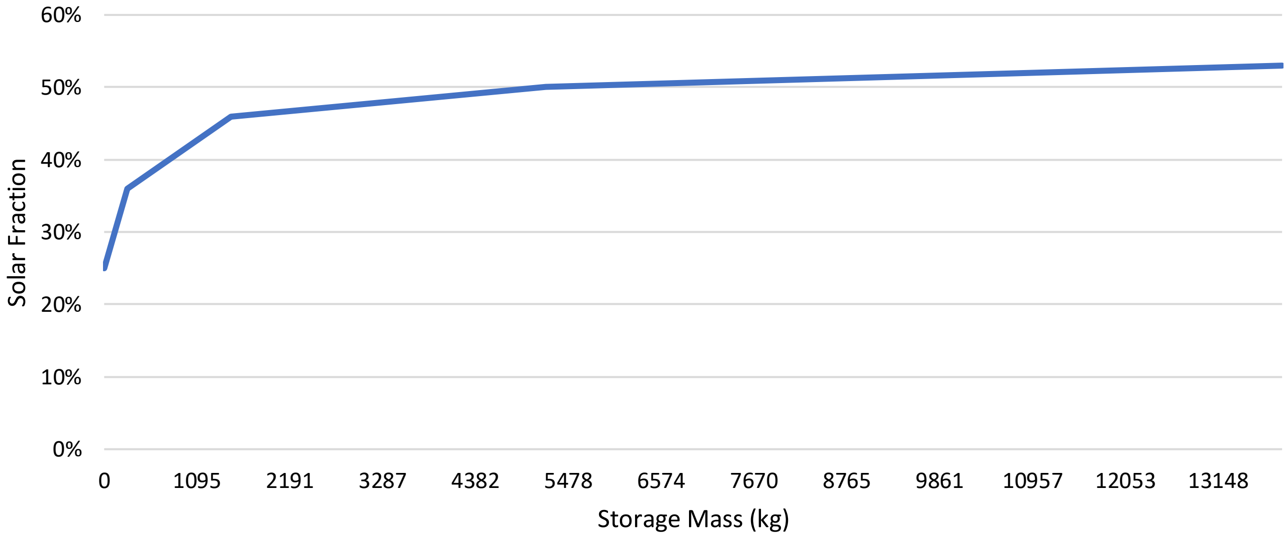


Fig. 41. Solar fraction vs storage mass (actual-2).

E. **F-chart**

F-chart was developed by Beckman et al in 1976. It is used to characterize the long-term performance of residential solar heating systems in a quick and robust way despite having some uncertainties [59].

F-chart is a procedure for estimating the fraction f of the total monthly heating load (space and water heating) that can be supplied by an active solar thermal system. It is based on the correlation of f as a function of X and Y. where X is the ratio of collector losses to heating loads and Y is the ratio of absorbed solar irradiation to heating loads. The correlation was derived from the results of a large number of hour by hour solar simulations in TRNSYS. F-chart was developed for 3 system types: 1-space and water heating using water 2-space and water heating using air and 3-process hot water. For this project the 1st type will be examined. A schematic of the system this calculation describes is presented in figure 42 [25,60].

$$X = \frac{A_c \dot{F_R} U_L \Delta t (100 - \overline{T_a})}{L} \tag{7}$$

Where: X is the ratio of collector losses to the heating load, A_c is the collector area (m^2), $\dot{F_R}$ is the corrected heat removal factor usually taken as 0.97 F_R, U_L is the collector loss coefficient, $\overline{T_a}$ is the monthly average ambient temperature (°C), Δt is the number of seconds in a month and L is the monthly space and hot water heating load (J)

$$Y = \frac{A_c \dot{F_R} \overline{\tau\alpha} H_T N}{L} \tag{8}$$

Where: Y is the ratio of the absorbed solar irradiation to the total heating load, ($\overline{\tau\alpha}$) is the monthly average τα product for the collector (taken as 0.96 τα [25]), H_T is the monthly average daily radiation on the collector surface (J/m^2) and N is the number of days in the month.

.

For liquid system

$$f = 1.029Y - 0.065X - 0.245Y^2 + 0.0018X^2 + 0.0215Y^3 \quad (9)$$

For air systems

$$f = 1.040Y - 0.065X - 0.159Y^2 + 0.00187X^2 - 0.095Y^3 \quad (10)$$

f, X and Y are dimensionless. For a list of assumptions inherent in the f-chart method see [60]. Initially f-chart was used as presented by B. Hodge [25]. However upon examination of John Duffie's Solar engineering of thermal processes [60], additional information came to light such as the f-chart assumes 75 L/m^2 of collector area energy storage as well as that the X&Y values are only applicable for X<15 and Y<3. Outside of those limits the correlation does not hold but figure 43 can be extrapolated. In [25] the author used X and Y values for his example that were outside of those limits. He also neglected to mention the assumed energy storage.

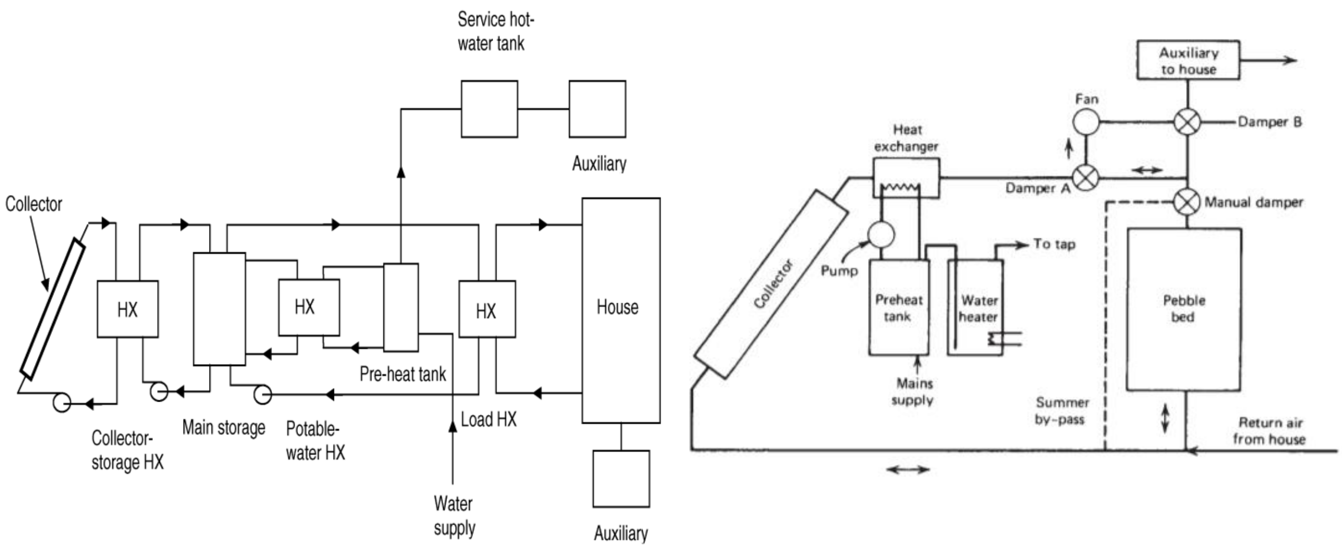


Fig. 42. F-chart liquid solar system schematic (left) and air solar system schematic (right) [60].

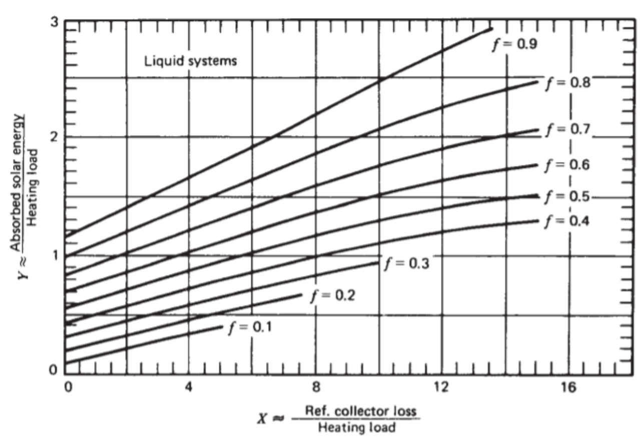


Fig. 43. The f -Chart for systems using liquid heat transfer and storage media [60].

Solar thermal systems used for both space heating and DHW (combi systems) are becoming popular for residential applications; however, when using only diurnal storage these systems typically are unable to achieve solar fractions greater than 50%. To increase the solar fraction of these systems seasonal thermal storage system is required. When seasonal thermal storage systems are successfully implemented an annual solar fraction approaching 100% is obtainable [64]. According to [65] solar fraction for residential systems is typically in the 30-60% range.

1) F-chart vs boundaries

From previous sections the upper and lower boundaries were established. In this section the f-chart solution will be compared against the boundaries. Duffie and Beckman [60] assume 75 L of storage per m^2 of collector area used. In [60] an equation was provided in case one needs to alter this assumption which is:

$$\frac{X_c}{X} = \left(\frac{actual\ sotrage\ capacity}{standard\ storage\ capacity}\right)^{-0.25} \tag{11}$$

Where X is the old X and X_c is the corrected X factor. i.e. $\frac{X_c}{X}$ can be multiplied by the original X to replace it with the correct X. Equation 11 can be used for storage values between 37.5 and 300 L/m^2 [60]. For the purpose of this section, this equation will be modified by the addition of a term representing the existing storage (60 gallons/270 liters tank) in terms of collector area for different number of collectors.

$$existing\ storage = \frac{270\ L}{Collector\ Area\ *\ Number\ of\ collectors}$$

To use the f-chart, average monthly daily values of irradiance, load and ambient temperatures had to be tabulated and are given in table XVIII. These values were used to calculate X, Y and f for the selected system. The result of the calculation is shown in table XIX. The results show that the solar fraction for a 36 collector liquid system using the f-chart is 50% while from earlier calculations it was (1 – unmet load) = 50% (for actual-2 system). This initially confirmed the earlier calculation but due to f-chart limitation explained earlier (X<15 & Y<3) the confirmation is less certain. Also, the f-chat is likely to be considering more system losses than earlier calculations. The results for the air system were also calculated and were within 4% of the liquid system. (Note: f-chart does not rely on inlet fluid temperature)

TABLE XVIII
F-CHART INPUTS

Time span	Monthly Average Daily radiation	Value ((kWh/m2*day))	Monthly Thermal Load	Value (kWh)	Monthly Average Ambient temperature	Value (℃)
Yearly	I_{yearly} (kWh/m2)	1160.70	L_{yearly} (kWh)	40,581	T_{yearly}	4.86
January	I_{jan}	1.16	L_{jan}	6420	T_{jan}	-2.67
February	I_{feb}	1.91	L_{feb}	5456	T_{feb}	-6.23
March	I_{march}	3.13	L_{march}	4924	T_{march}	-2.40
April	I_{april}	4.01	L_{april}	4228	T_{april}	1.66

May	I_{may}	4.93	L_{may}	3392	T_{may}	3.89
June	I_{june}	5.26	L_{june}	2034	T_{june}	10.14
July	I_{july}	5.08	L_{july}	1050	T_{july}	14.46
August	I_{august}	4.57	L_{august}	805	T_{august}	16.80
September	I_{sept}	3.50	L_{sept}	1256	T_{sept}	11.80
October	I_{oct}	2.28	L_{oct}	2680	T_{oct}	7.23
November	I_{nov}	1.31	L_{nov}	3729	T_{nov}	3.73
December	I_{dec}	0.94	L_{dec}	4607	T_{dec}	-0.84

TABLE XIX
F-CHART RESULTS

X	Y	f (liquid systems)	f*Load	% of load supplied	f(air system)	f*Load	% of load supplied
2.49	0.27	0.1	706.2	50%	0.1	766.3	46%
2.74	0.52	0.3	1694.8		0.3	1836.5	
3.24	0.95	0.6	2877.0		0.6	3188.8	
3.50	1.42	0.8	3479.5		0.9	3913.0	
4.41	2.18	1.0	3391.8		1.0	3391.8	
6.65	3.87	1.0	2033.8		0.7	1517.5	
12.67	7.24	1.0	1050.3		0.0	0.0	
16.08	8.49	1.0	805.4		0.0	0.0	
10.57	4.17	1.0	1256.2		0.4	513.4	
5.39	1.27	0.7	1767.5		0.8	2016.1	
3.89	0.53	0.3	935.9		0.3	1035.2	
3.41	0.31	0.1	421.5		0.1	475.1	

Next a modifier representing the number of collectors used was introduced for ease of calculation. The modifier changes the energy surplus/deficit, energy deficit only and storage mass equations resulting in a change in the solar fraction. 75 L/m^2 of collector area plus the existing 270 L storage tank were examined for both the f-chart and manual calculations. The result of the comparison is shown in table XXI and figure 44. The calculation was repeated for both actual-1 and actual-2 scenarios. The results show that the f-chart results are within the upper and lower boundaries. The f-chart curve seems to converge to the lower boundary as the number of collectors increases. This could be due to the increase in collector number increasing collector area which increases storage size since the storage is a function of collector area. Physically what this could imply is that as the size of the storage tank increases the effect of water consumption (DHW load) becomes less and less pronounced and the effect of space heating becomes dominant this could be due to the size of water extracted/replenished by the DHW load becoming relatively smaller as the size of the tank increases.

TABLE XX
F-CHART VS ACTUAL-1 VS ACTUAL-2

# of collectors	1	5	10	15	20	25	30	36
Storage volume (L)	145	725	1449	2174	2898	3623	4347	5216
Actual-2	3%	14%	23%	30%	36%	41%	45%	50%
f-chart	4%	16%	26%	34%	40%	43%	47%	50%
Actual-1	5%	19%	32%	41%	49%	56%	60%	65%

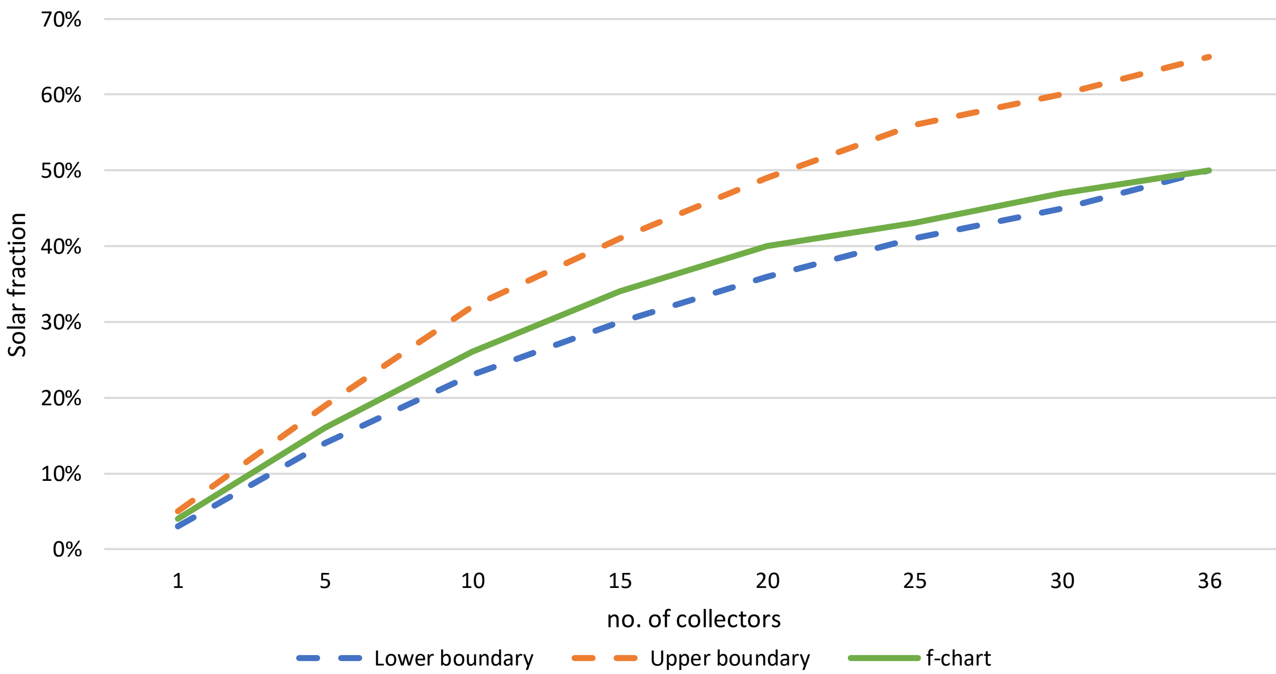


Fig. 44. F-chart vs boundary conditions.

F. Collector comparison

Table XXII shows a comparison between a few different solar thermal collectors, namely, the AE-series, ST-series and MSC-series from alternate technologies and finally the SC5830 evacuated tube collector from Dudu. It can be seen that AE-21 (according to its most recent SRCC certification [66]) has the best area utilization at 47.9 m^2 total area (5.5% less area than other AE-series models) but its solar fraction is 1% less than other AE-series solar collectors. AE-21 had higher loss coefficient (F_RU_L) than other AE-series models but 4% higher intercept (optical efficiency). AE-26,32 and 40 had the highest solar fractions and 2nd lowest areas. American energy (AE-E) models had the highest area amongst flat plat collectors and the lowest intercept and solar fractions making them the worst performing. Starfire (ST) models were 2.2% better than AE-E models in terms of area utilization but just as low in terms of solar fractions. Morning star series (MSC) had the same area utilization and solar fractions as the AE-series. Finally, the evacuated tube model SC5830 from Dudu had the lowest optical efficiency (intercept) but also the lowest losses coefficient (slope). Despite taking up the most area the tubular model did not produce the highest energy per collector (AE-40 is the highest) for this reason the total area taken by the 18+ modules required is 83.8 m^2 which is far higher than any of the flat plate collector. on the other hand, the solar fraction for SC5830 is also 4-

5% higher than the other models. Therefore, if area utilization is not a constraint evacuated tube models can provide slightly higher performance but if area consideration is included then AE-21 is the most compact despite offering slightly lower solar fraction.

TABLE XXI
COLLECTOR COMPARISON

Collector	AE-21	AE-26	AE-32	AE-40	AE-21-E	AE-32-E	AE-40-E	ST-32E	ST-40E	MSC-21	MSC-32	MSC-40	SC5830
Area (m^2)	1.93	2.35	2.96	3.70	1.93	2.96	3.69	2.87	3.59	2.00	3.04	3.92	4.57
Intercept	0.75	0.71	0.7!	0.71	0.66	0.66	0.66	0.67	0.67	0.71	0.71	0.7!	0.42
Slope (W/m^2°C)	-6.2	-4.91	-4.91	-4.91	-6.37	-6.37	-6.37	-6.02	-6.02	-4.91	-4.91	-4.91	-0.79
Annual Quseful (1 collector) (Wh)	1635	1879	2366	2958	1439	2188	2728	2185	2733	1599	2430	3134	2206
No. of collectors needed	24.8	21.6	17.1	13.7	28.1	18.5	14.8	18.5	14.8	25.3	16.6	12.9	18.4
Solar fraction	53%	54%	54%	54%	53%	53%	53%	53%	53%	54%	54%	54%	58%
Total area (m^2)	47.9	50.7	50.7	50.7	54.4	54.8	54.8	53.2	53.2	50.7	50.7	50.7	83.8
Reference	[66]						[69]						[70]

Note: 1- number of collectors needed were not rounded up to ensure comparative accuracy. In practical implementation, rounding up would be necessary. 2- 1500 kg water storage tank was assumed for all calculations, 3- HWB equation was used for evacuated tube since reference [69] used it in the same manner. [71] suggested a modification to the equation but the experimentally assessed coefficients required were not obtainable for the collector in question.

G. **Uncertainty**

In using BEOPT, the gallons of oil used for space heating show some values in the summer months which should normally not happen. This is due to the basement being underground so despite the hot ambient temperature above ground the basement is exposed to the ground temperature as its ambient which ranges from 45 °F to 50 °F so heating is required even during summer. The actual basement is not fully underground only partially. To model a partially elevated basement in BEOPT however was not doable. Therefore, the space heating load from BEOPT can be higher than the actual load used by the house.

In calculating the lower boundary, ambient temperatures for the basement level are not the same as the living space during the summer months where the living space is not heated but the basement is. For the sake of this report the basement temperature will be taken as ambient as it's the only level being heated year-round.

The temperature of the water after it completes its space heating objective is assumed to drop to ambient. In reality the fluid will drop to near ambient say 1 to 2 degrees higher than the ambient.

The temperature of the water at the lowest node of the stratified tank in case of DHW only will not be the same throughout the day as during daytime water consumption (dishes, shower, laundry) is a lot higher than during night time so it is likely that the tank temperatures will change at night.

Not all different types of losses were accounted for.

V. Results and Discussion (Simulation)

A. Polysun

1) Combi System

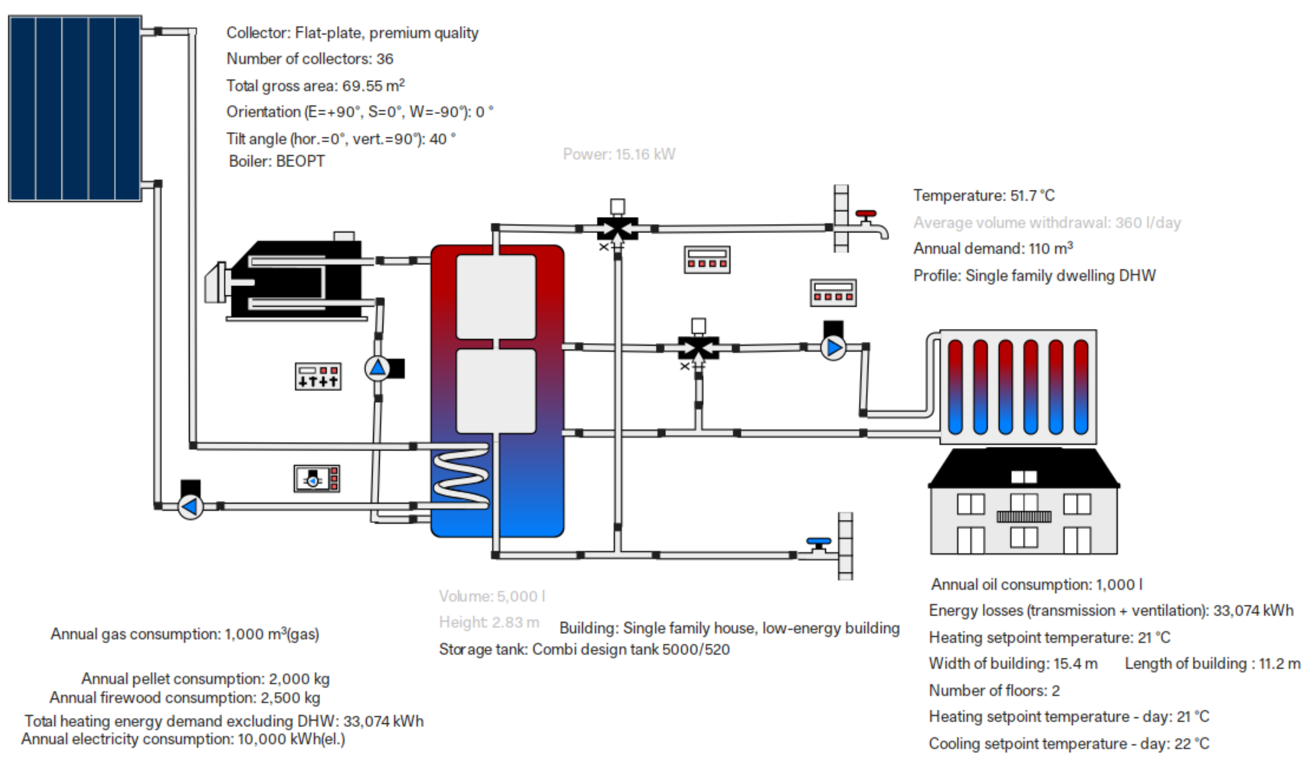
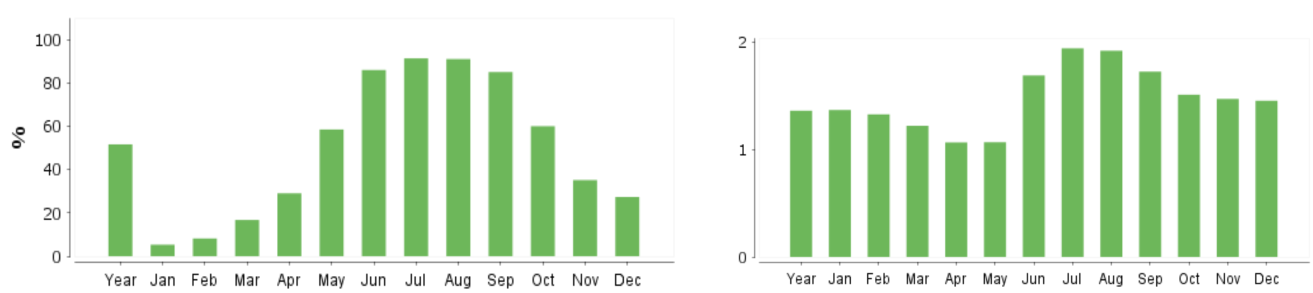


Fig. 45. Polysun Combi System.



a b

Fig. 46. Polysun combi system selected results, a) Solar fraction, b) Primary energy factor.

TABLE XXII
COMBI-SYSTEM POLYSUN RESULTS

Metric	Unit	Value
Solar fraction (total)	%	51.7
Solar fraction (hot water)	%	52.4
Solar fraction (space heating)	%	52.4
Solar thermal energy output	kWh	44,050
Total energy consumption	kWh	43,205
Energy deficit	kWh	69.6
Total electricity consumption	kWh	18.7
Primary energy factor	unitless	1.36

This section describes scenario E (shown in figure 45), which is the actualization of the combi system described earlier in the manual calculations stage. It involves the solar collector array, boiler, combi water tank, baseboard radiator, space and water loads, pumps and the interconnection piping and controls. It was based on a built-in model that was customized and experimented with to fit the design of the proposed system.

In the project wizard, the design house location was selected from the map latitude of 47.5, longitude of -52.7 and elevation of 82 m. Solar thermal and boiler were selected as the energy providers, DHW and space heating as the loads (no pool heating) and residential system as the system size. Heat pump, Chiller and Photovoltaics options were disabled. This narrowed down the template options from which 9u space heating and DHW system was then selected from the USA systems as the system diagram most closely resembled the system envisioned during the manual calculations stage. The results as can be seen from figure 46 and table XXII show a 51.7% solar fraction and 1.36 primary energy factor (PEF). The solar fraction is within the range provided in the methodology and close to the value provided by the f-chart (50%). More Polysun results figures can be seen in appendix A.

Under hot water options, 6 was selected as the number of occupants and 51.7 °C as the hot water temperature. Daily hot water demand was selected as 360 L and absences as never. The load profile was calculated automatically by Polysun as 24.5 kWh/day which is an “XXL” size profile according to the software.

Under building, “single family house normal building” option was selected. The length and width of the building was inputted as 11.2 m and width 15.4 m respectively (obtained from Google earth). Number of floors was set to two and so the heated living area then becomes 345 m (172 m per floor). The heating set point temperature was set to 21 °C. This resulted in a space heating load of 33 MWh (same as BEOPT). Under heating element, a radiator with nominal inlet temperature of 60 °C and nominal return temperature of 21 °C were selected (assuming that, ideally, temperature should drop to near ambient at the end of any given hour) however changing the radiator to 40 °C nominal inlet and 30 °C nominal return temperatures [63] yields 4% higher solar fraction and 0.14 lower primary energy factor. Low return temperatures are recommended for combi systems [90]. Due to uncertainty regarding the temperature parameters of the existing radiator a conservative choice was made to keep the earlier design.

Under Solar system, North America was selected as the test standard, AE-21 was selected as the collector module, orientation is set to 0 (south facing) and tilt is set 40 degrees. Solar fraction was selected as "medium" from low to high options. Collector number was set as 36 which makes the gross area 69.6 m^2 and assuming 75 L of storage per m^2 of collector area then the storage size should be 5216 L so a 5000 L combi tank was selected. The tank is 2.83 m tall and made from stainless steel with 160 mm thick polyurethane foam insulation. It is a tank in tank model which according to [149] offers the best efficiency, saves space and cost. Reducing the size of the tank to a more practical size (50 gallons) increases the primary energy factor by 0.9 and reduces the space heating solar fraction to 40.3% (12% reduction) while increasing energy deficit to 296 kWh (325% increase) and stagnation time by 20%. Increasing the size of the storage to 6000 L only led to a 0.3% increase in solar fraction, no change in the primary factor or energy deficit. This is supported by the results in figures 38 and 41 (and similar figures found in [60]) which show that beyond a certain point solar fraction is less responsive to increases in storage volume. Placing the 5000 L tank outdoor (in the backyard) yields a solar fraction of 50.5% and primary energy factor of 1.38 which might be more practical than placing it in the basement. The reason a 5000 L tank is studied is to confirm the results obtained earlier. A smaller tank can be accommodated at reduced solar fraction as reported earlier.

From the load data from BEOPT, the maximum heating load that occurred at any given hour (after installing the solar thermal system) was 27 kW, by accounting for the 80% boiler efficiency, the boiler should have a rated power of approximately 32 kW so under heat generator a 35 kW boiler that uses heating oil was selected. This was close to the recommended power by Polysun which was 38 kW. It should be noted that the boiler initially had an efficiency of 85% but was changed to 80% to match BEOPT. However, doing so had no effect on the solar fraction but it increased the primary energy factor by 0.8. The boiler has a minimum power of 5 kW and was placed indoor in a heated area (basement), not doing so reduces the solar fraction by 0.7% and increases the primary energy factor by 0.02.

The three pumps (solar loop, boiler loop and space heating loop) were initially selected as large with 30% heat transfer percentage. However, changing them to small leads to the same solar fraction and primary energy factor, 0.01 kWh higher energy deficit, but an 88% (136 kWh) lower electricity consumption. To choose a pump with 60% heat transfer such as Biral M12 for all three pumps leads to a 0.3% increase in solar fraction, no change in primary energy factor and a 42 kWh decrease in electricity consumption over the large pumps system. Therefore, small eco pumps will be used. Pipes were given by the built-in model as 22 mm diameter copper tubes. Replacing these with the much larger 107 mm PVC pipes yielded only a 0.2% increase in solar fraction even when coupled with larger pumps. It also did not reduce stagnation time (high stagnation is a common problem in solar combi systems [90]). Therefore, smaller pipes and pumps will be used.

The primary energy factor is a measure of efficiency (its inverse) which connects energy used with energy generated. It enumerates the value of the amount of primary energy used to generate one unit of electricity or useful thermal energy. In the European Union, the default PEF for electricity generation is 2.5 which is equivalent to 40% efficiency. This metric is mainly used in Europe (Polysun is a German software) and is criticized for not reflecting emissions associated with generation [147]. For this project, a PEF of 1.36 is equivalent to 73.5% efficiency which includes various sources of losses.

It should be noted that Polysun is using a slightly different set of weather data (obtained from Meteronorm 6) than the ones used in earlier calculations and so might offer marginally different results. attempts to manually input the earlier data were not successful.

i. ***Economics***

For the results of this section to be comparable to the results of project A (Scenario C), different economic metrics need to be calculated. This involves the selection of components and obtaining their prices. The price of a 5000L stainless steel solar water tank was $5000 [150], AE-21 solar collector $750 [48]. Polysun used 32 m of 22x1 mm fiber glass insulated copper pipes in total which can be purchased for $32.4/m for the pipes [151] and $3.6/m of insulation [152]. A small solar heater pump costs $142 [153]. According to [154], installation cost of a solar thermal system is around $1500 (assuming the owner doesn't install it himself which he is prone to do). A mixing valve from Home Depot costs $44 [160] and a controller is worth $32 [161]. The radiator and boiler are already in existence. According to [62] residential solar thermal has no associated O&M costs. Table XXIII and figure 47 summarize the associated costs.

This yields a total cost of $35,294 and since the solar fraction is 51.7% the annual useful energy provided by the system 20,946 kWh which offsets 26,182 kWh of oil (80% efficient boiler). Given the $2.8/gallon heating oil price, this results in a payback period of 19.7 years at an annual saving of $1,806. This results in $18,886 profit (for a 30-year project lifetime) corresponding to a 53.5% Return on investment (ROI) and 1.44% annualized ROI [155]. By using the Levelized cost of Energy (LCOE) calculator from NREL* [156], but modifying it to be compatible with the current system (appendix C), an LCOE of 0.123 USD/kWh or 0.16 CAD/kWh was obtained which is 1.3% better than thermal load scenario from project A (Scenario C) which utilized one 10 kW wind turbine and fourteen 0.305 kW PV panels to provide electricity that was used for electric heating. The payback period for scenario C was also higher at 25.8 years and had no positive ROI making it unfeasible. According to [163] the LCOE of a solar system in Europe is 0.139 €/kWh making the current system relatively cheap. Table XXIV summarizes the economics of the project.

The previous results are noteworthy since the electric system used net metering (grid acts as storage) and so had no excess or deficit energy (demand/supply mismatch was eliminated). This could be because the solar thermal system has a 73.5% efficiency (inverse of PEF) and according to [72], a solar collector can reach 80% efficiency (relative to sunlight input) while the PV panel had an efficiency of 18.6% (relative to sunlight input). Solar thermal is also a more mature technology making the technology more valuable despite one kWh of thermal energy being less valuable than a one kWh of electricity. The lifetime of the solar thermal system is also 5 years longer. The LCOE for the solar thermal system does not include the LCOE of the oil boiler which is responsible for almost half the load. As the boiler system is already paid for its LCOE is expected to be close to per kWh cost of fuel which will reduce the overall system LCOE further.

Note*: NREL is also the developer of HOMER pro which was used to obtain the different LCOEs in project A. a 6% discount rate was assumed similar to project A (HOMER's default). LCOE in this scenario is equivalent to LCOH.

TABLE XXIII
COMPONENT PRICE

Expenditure category	Number of units	Total cost per category ($)
Collector array	36	27,000
Tank	1	5000
Pumps	3	426
Piping	32 m	1152
Mixing valves	2	88
Controllers	4	128
Installation		1500
Total		35,294

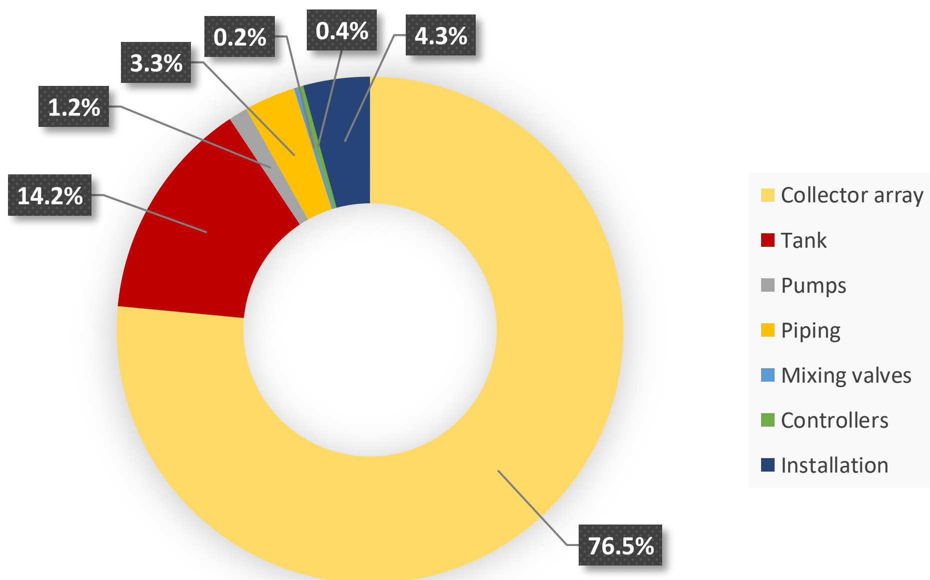


Fig. 47. Combi system cost pie chart

TABLE XXIV
ECONOMIC RESULTS COMPARISON

Economic Metric	Scenario C	Scenario E
Capital cost ($)	76,185	35,294
Project period (years)	25	30
Payback period (years)	25.8	19.7
Annual savings ($)	2,807	1806
Total savings ($)	70,180	54,180
Return on investment (%)	Negative	53.5
LCOE ($/kWh)	0.162	0.16

ii. *Emissions*

According to the EIA, home heating oil generates 10.16 kg of CO_2 per gallon burned [159], which means that by installing the solar system the house can reduce its annual emissions by 51.7% or 5.25 tonnes of CO_2. The average per capita annual emissions of buildings sector in NL is 1.8 tonnes per person [157]. After the solar system the house's CO_2 emissions are 0.82 tonnes of CO_2 per person (given 6 occupants).

2) PVT system

In this section a PVT system (scenario F) is simulated to see how it compares with the PV only system developed in project A and the thermal only system developed in this project. Figure 48 shows the system diagram.

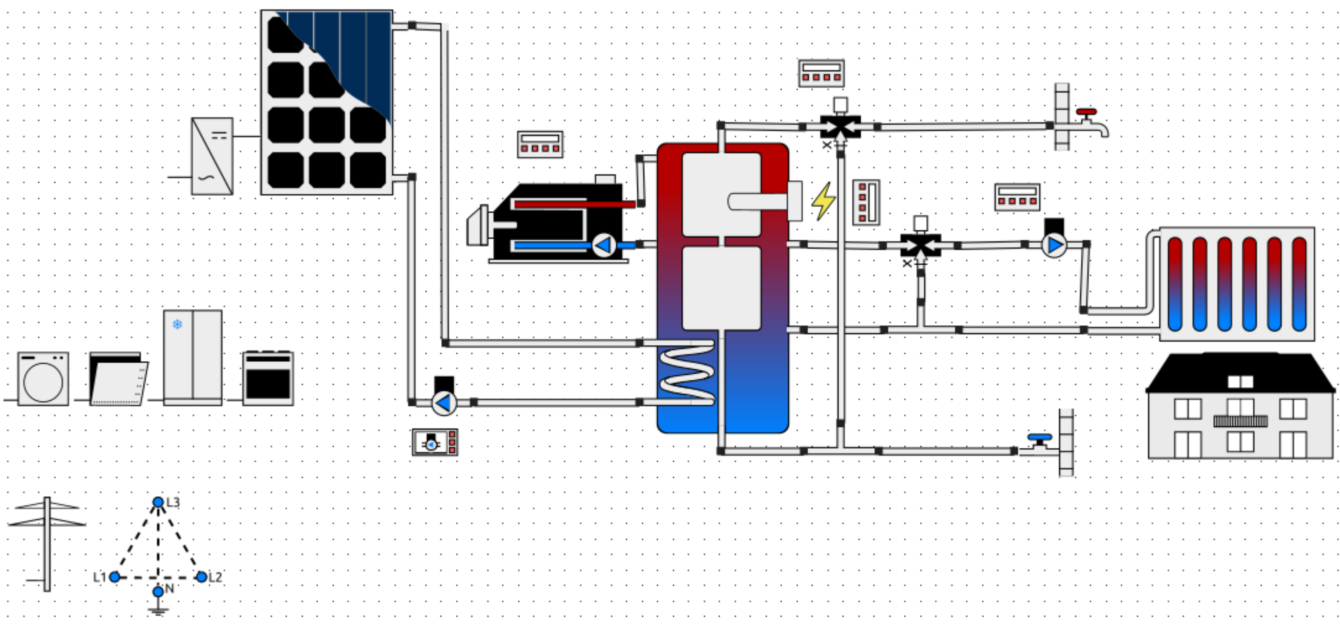
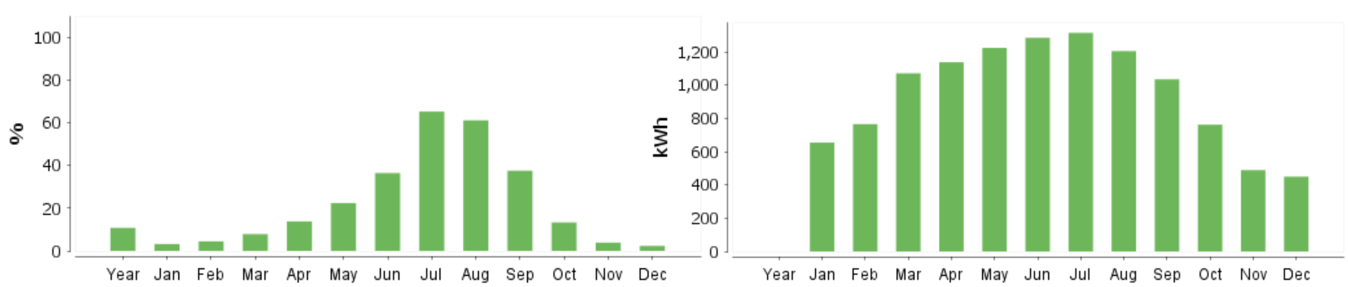


Fig. 48. Polysun PVT System.



a b

Fig. 49. Polysun PVT Selected Results; a) Solar fraction, b) Electricity generated.

SolarOne 290 W flat plate PVT module was selected. It is manufactured by Gasokol GmbH. It is a monocrystalline model with an efficiency of 17.1%. An array of 36 of said module has a gross area of 61 m^2. It is set up at a tilt angle of 40 degrees, non tracking option and 0.5 %/year degradation rate suggested by Polysun are used (supported by [148]). 0% cable losses, 2% soling losses and 3% mismatch losses are assumed (same as figure 26 in project A which is the helioscope results for the PV system). Sunny Boy 10 was the selected inverter (same as project A). 12 modules per string and 3 strings are used to make the results comparable with project A. The electric load is set to 10,731 kWh (BEOPT's electric load used in project A). The same boiler, hot water load, space heating load, space heating element and pumps used in the previous section are replicated here. The tank is however different since it is not justifiable to use a 5000 L tank as it did not lead to a higher thermal output/solar fraction so the existing 50-gallon tank is used instead. This removes the tank cost from calculation. The results (shown in figure 49) show a much lower solar fraction at only 12% (34.6% DHW and 9.5% space heating) equivalent to 5.38 MWh of heat which is 12% of the heating load. On the other hand, the system managed to generate 11.4 MWh of electricity which is enough to fully satisfy the electric load (106% of the load). The primary energy factor for this system is 1.3 which is equivalent to 77% overall efficiency.

Comparing these results with the PV only system from project A (scenario A) where the 50.74 m^2 of PV area generated 10.614 kWh making its energy density 209.2 kWh/m^2 while for the PVT system the electrical density is 187 kWh/m^2 and by including both electrical and thermal generation it becomes 275.6 kWh/m^2. The thermal energy from the combi system is 20.95 MWh for an area of 69.6 m^2 making its energy density 301 kWh/m^2. For better comparison electricity rate is currently 12.3 cents and the cost of heating oil 2.8 $/gallon which translates to $2.8 per 40.6 kWh of (source) thermal energy or 6.9 cents per kWh. Which means that electricity is 1.78 times more valuable than thermal energy. By considering this, the energy density of the PV only system becomes 372.4 kWh/m^2 and for PVT 421.7 kWh/m^2 while solar thermal stays at 301 kWh/m^2, which means that if the three technologies are competing over rooftop space, PVT should be given first priority followed by PV followed by solar thermal. However, if the fuel replaced by the thermal system (26.18 kWh) is considered instead, its energy density becomes 376 kWh/m^2 (80% efficient boiler) making it place 2nd instead of 3rd. The efficacy of a thermal system will depend in part on the efficiency of the heater it is replacing.

For a more accurate comparison two cases are proposed; one that compares PVT with the PV only case from project A and one that compares it with the solar thermal case. In the 1st case, the power of the PVT system is matched to the power of the PV system in scenario A and in the 2nd the area of the PVT system is matched to the area of the solar thermal system. As can be seen from table XXV, case 1 has 1.2% more power, 1.2% more area but 1.5% lower efficiency relative to scenario A. The PVT case is able to generate 8% more electricity than the PV system while generating an additional 5.2 MWh of heat energy which satisfies 11.6% of the thermal load. Comparing the combi system scenario (scenario E) with a PVT system of equal physical size we find that the PVT system is able to generate approximately 27% of the combi systems useful energy output but with an additional 13.3 MWh of electricity. By setting scenario A as the reference and given the fact that a kWh of thermal energy is worth 56% the value of a kWh of electricity ($\frac{1}{1.78}$), the PVT systems seem to outperform both Scenarios A and E in terms of value of generation. According to the literature review conducted by [93] electrical efficiency for nano fluid enhanced PVT modules ranges from 12.7% to 23.5% for the studies covered while thermal efficiency ranges from 33% to 85%. A previous study showed that practical PVT efficiencies are around 11.8% electrical efficiency and 24% thermal efficiency [190].

TABLE XXV
PVT VS PV VS SOLAR THERMAL

Metric	Scenario A	PVT Case 1	Scenario E	PVT Case 2
Number of collectors	31	33	36	41
Area (m^2)	50.84	51.45	69.6	69.5
Power (kW)	9.46	9.57	--	11.9
Module power (kW)	0.305	0.290	--	0.290
Module efficiency (%)	18.6	17.1	74.9 ($F_R\tau\alpha$)	17.1
Thermal energy output (kWh)	--	5221	20,946	5625
Electricity output (kWh)	10,614	11,460	--	13,333
Solar fraction (%)	--	11.6	51.7	12.5
Primary energy factor	--	1.31	1.36	1.29
Capacity factor (electricity only) (%)	13	13.7	--	12.8
Relative generation value	100%	136%	111%	155%
Capital cost ($)	26,103	39,930	35,294	47,153
Project period (years)	25	25	30	25
Payback period (years)	19.9	21.5	19.7	22.2
Annual savings ($)	1305	1859	1806	2125
Total savings ($)	32,625	46,497	54,180	53128
Return on investment (%)	25	16.5	53.5	12.7
LCOE ($/kWh)	0.19	0.22	0.16	0.22

In order for the analysis to be complete, economics have to be calculated. There is some difficulty in obtaining a representative price for PVT collectors. Assuming the same system as before but with different collectors and using the existing tank, the price of PVT collectors is assumed as 2000 €/kW as given by [164] this results in a total system cost of PVT case 1 of $39,930 and for the 2nd case, $47,153. The cost of the inverter is $4469 [165], which includes replacement in year 15. The breakdown of costs can be seen in figure 50. The capacity factor obtained for scenario A (13%) was lower than that calculated for PVT case 1. A 100 $/year O&M cost based on [165] and 25-year lifetime were assumed (same as scenario A). The LCOE for both PVT cases is 0.22 CAD/kWh which is 16% higher than the PV only system and 37.5% higher than scenario E. The payback period and ROI for the PVT system was also worse than the PV and combi system cases which is mainly due to the high collector price. In [191] the author stated that PVT collectors have to fall to less than half their 2014 prices in order to be competitive with solar thermal and solar PV while novel siloxane lamination panels need to reach prices as low as 290 €/m^2 to be competitive. In [192] the payback period for a PVT system installed in London was 22.7 years which is similar to the figure obtained here while is sunnier places like Athens the payback period was lower at 15.6 years. The Payback period for London is calculated to drop to less than 19 years if the price of the PVT collector is reduced by half.

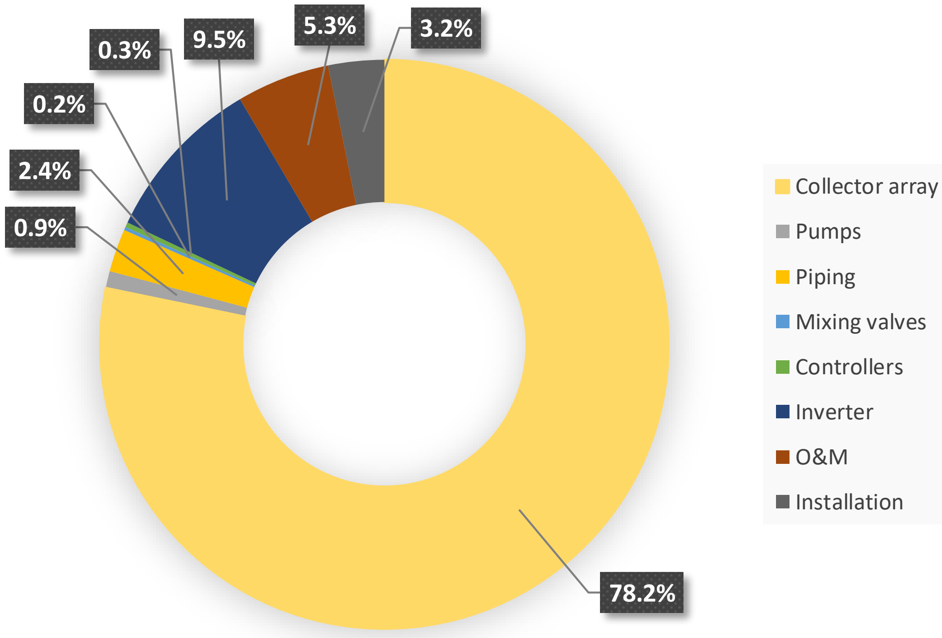


Fig. 50. PVT system cost pie chart

B. RETScreen

This section describes Scenario G which includes a solar water heater, SolarWall air heater and glazed flat plate solar air heater. Table XXVI highlights the chosen option values.

TABLE XXVI
RETSCREEN OPTION VALUES

Main Tab	Sub Tab	Option	Value
Location		Longitude, latitude	47.6, -52.7
		Climate zone	6b Cold-Dry
		House elevation	83 m
Facility	Level 1	Facility type	Residential detached dwelling
		Facility size	172 m^2
		Energy consumption (density)	299 Wh/m^2
		Total energy consumption	51,433.97 kWh/year
		Target	0%
	Level 2	Electricity consumption	10,727 kWh
		Oil consumption	999.6 gallons
Energy	Electricity and fuels	Fuel	Diesel (#2)
		Heating value	19,561 BTU/lb

		Fuel rate	2.8 $/gallon
		Electricity rate	12.3 ¢/kWh
	Schedules	Space heating temperature	21 ℃
		Unoccupied space heating temperature	15 ℃
		Occupancy	16 hours/day
		Heating/cooling changeover temperature	21 ℃
	Boiler	Fuel	Diesel (#2)
		Seasonal efficiency	80%
	Water heater	Fuel	Diesel (#2)
		Seasonal efficiency	62%
End use	Hot water	Number of occupants	6
		Hot water use	360 L/day
		Hot water setpoint temperature	51.7 ℃
		Operating hours	24 hours/day
Heating	Solar water heater	Tracking	fixed
		Slope	40 °
		Azimuth	0 °
		Collector model	AE-21
		Number of collectors	4
		Miscellaneous losses (snow)	3%
		Temperature coefficient	0
		Storage capacity	75 L/m^2
		Heat exchanger efficiency	80%
		Miscellaneous losses (tank and pipes)	6%
		Pump power (density)	12.4 W/m^2
		Collector price	$750
	SolarWall	Area	24.73 m^2
		R-Value (wall insulation)	10 $\frac{ft^2 - °F - h}{BTU}$
		Design air flow rate	216 m^3/h
		Heating load	16,788 kWh
		Tilt	90°
		Azimuth	0°
		Collector model	Conserval engineering Solarwall

	Colour	Black
	Performance factor	1.0
	Shading percentage	10%
	Wind sheltering	30%
	Fan power (density)	0.432 W/m^2
	Cost	$6256
	Collector model	Cansolair RA 240 Solar max
	Collector area	2.29 m^2
Solar air heater	Number of collectors	3
	Miscellaneous losses	3%
	Fan power	0.432 W/m^2
	Collector price	$2500

1) Location

In this section the characteristics of the location and its associated meteorological data is inserted into RETScreen. The design house location was inputted as 47.6 latitude and -52.7 longitude. St. John's airport was selected as the nearest climate data location. The climate zone is based on ASHRAE climate zones while the moisture zone can be either marine, dry or humid depending on average temperature, degree days and precipitation data. In [75], St. John's is listed as an ASHRAE climate zone 6 and in [76] the city is listed as mostly dry (based on a dew point definition). Therefore, 6B Cold-Dry is selected as the climate zone. According to google earth, the elevation of the design house is 83 m above sea level while the elevation of St. John's international airport meteorological station is 140.5 m above sea level [73].

The values of the air temperature, relative humidity, wind speed and solar radiation were changed to match the data used in earlier sections. The first three were obtained from Canada's meteorological data website [73] in hourly format for 2019 and then averaged on a monthly basis while the latter was obtained from HOMER. Earth temperature and atmospheric pressure were obtained from BEOPT and averaged for each month. The height of the wind speed measurement point and depth of earth temperature were left at the default value of 10m and 0m respectively. Table XXVII highlights the new values.

TABLE XXVII
RETSCREEN INPUTS

Month	Air Temperature (°C)	Solar Radiation (kWh/m2/d)	Relative Humidity (%)	Wind Speed (km/h)	Earth Temperature (°C)	Atmospheric Pressure (kPa)
January	-2.7	1.16	84.4%	15.4	1.02	99.65
February	-6.2	1.91	75.7%	19.9	-0.54	99.22
March	-2.4	3.13	75.1%	15.3	-0.67	99.08

April	1.7	4.01	79.9%	17.0	0.07	99.28
May	3.9	4.93	82.0%	15.3	3.05	99.7
June	10.1	5.26	84.0%	11.8	6.02	99.48
July	14.5	5.08	82.4%	10.3	8.59	100.02
August	16.8	4.57	80.5%	11.1	10.23	99.44
September	11.8	3.50	83.6%	13.8	10.37	99.93
October	7.2	2.28	86.4%	13.2	9.05	99.35
November	3.7	1.31	87.0%	17.3	6.54	99.48
December	-0.8	0.94	87.6%	17.1	3.67	99.98
Annual	4.9	3.18	82.4%	14.7	4.82	99.6

Precipitation, heating degree days (HDDs) and cooling degree days (CDDs) were left at their default value. Precipitation was obtained by RETScreen from NASA database and was 1273 mm per year. HDDs were 4818 ℃-d annual total defined as days where temperatures are below 18 ℃ while CDDs were 407 ℃-d defined as temperatures above 10℃ based on average air temperature.

Degree day is based around the concept of no-load/balance point temperature and average daily temperatures. The balance point ambient temperature is one at which no heating or cooling energy would be required for human comfort inside a building. It has been typically taken as 65 °F (18.3 ℃). Thus, heating would be required for temperatures below 65 °F and cooling for temperatures above it. In other words, degree days are calculated as the difference between the balance point temperature and the average daily temperature for a given day. Daily HDDs and CDDs are then summed for the entire year. Degree days are useful in estimating the amount of energy required to heat or cool a building with HDD energy estimates being more accurate as CDDs do not consider humidity effects [25]. Therefore, for RETScreen to define HDDs as temperatures below 18 ℃ is correct but to define CDDs as temperatures above 10 ℃ might not be.

2) *Facility*

In this section the facility's properties are inputted.

i. *Facility level 1*

Under facility 1, Residential detached dwelling was chosen as the facility type. Facility size is 1852 ft^2 or 172 m^2 area (according to BEOPT and google earth). 299 kWh/m^2 was selected so that the total energy consumption is 51,433.97 kWh/year site energy use (same as project A). Target was set at 0% as energy efficiency retrofitting is not currently a goal.

ii. *Facility level 2*

In level 2 of the facility tab, electricity consumption was set to 10,727 kWh and gallons of oil used to 999.6 gallons (BEOPT results for the design house). Energy intensity was calculated by RETScreen as 62.4 kWh/m^2 for electricity and 248 kWh/m^2 for heating. The amount of energy from oil consumption was 42.5 MWh which is higher than the figure from BEOPT since RETScreen seems to be using an oil with a higher thermal energy content than the one used in BEOPT which had an energy content of 138,500 BTU per gallon of oil which yields 40.5 MWh per 999.6 gallons. The oil from RETScreen, dubbed Oil (#6), seems to have a thermal value of 157,711 BTU per gallon. Diesel (oil #2) in RETScreen, on the other hand, seems to be closer, yielding 40.25 MWh of heat energy. Since diesel and heating oil should have the same energy content and emission rate [78][79], diesel (oil #2) is selected as the fuel for this section.

3) Energy

i. *Electricity and fuels*

In this section electricity and heating oil are defined as the house's energy sources.

Under electricity and fuels in the energy tab, oil was initially selected at a heating value of 18,529 Btu/lb which is 154,717 BTU/gallon. Diesel was then selected which has a heating value of 19,561 BTU/lb or 136,927 BTU/gallon. The reason the energy content of a gallon of diesel is less than a gallon of oil despite oil having a lower heating value (thermal energy content) is because diesel is less dense at a volumetric density of 7 lbs/gallon while heating oil is 8.2 lbs/gallon. Next, fuel rate (price) was set to 2.8 $/gal (BEOPT's price). The fuel rate relative to energy content is 0.07 $/kWh. The electricity rate was set to 12.3 cents/kWh (Newfoundland's current rate) [2].

ii. *Schedules*

Under Schedules, Space heating temperature was set to 21 ℃ and no space cooling was chosen. Unoccupied space heating temperature was set to 15 ℃ [37] and the house is occupied 16 hours/day on weekdays as occupants go to work/school (normally). The house is assumed occupied 24 hours during weekends. Total occupancy rate then comes to 6,674 hours per year which is 76.2%. The heating/cooling changeover temperature is the ambient temperature at which the house's HVAC system will switch from heating to cooling. By setting this temperature to 21℃ it can be seen that the length of the cooling season is 0 days. This is desirable since the house has no cooling load/system.

iii. *Heating*

In this section the existing heating equipment is defined.

a) Boiler

After boiler is added to the heating section, diesel is chosen as the fuel type and the price of a gallon of diesel is set to 2.8 $/gal. The seasonal efficiency of the boiler is set to 80% (obtained from BEOPT). According to RETScreen, seasonal efficiency is lower than steady state efficiency since steady state efficiency is for full load conditions while seasonal efficiency is taking into account lower efficiency of part load conditions. RETScreen suggests 55-65% seasonal efficiency for a standard boiler or furnace, 65 to 75% for a mid efficiency boiler and 75 to 85% for a high efficiency unit [77]. It should be noted that as the design house is old there is doubt regarding the efficiency of the boiler. The value used was the one obtained from surveying the owner of the house and used in project A and so will continue to be used.

b) Water heater

Next, a water heater was added which had a seasonal efficiency of 62% (from BEOPT/landlord) and used diesel (#2 oil) as fuel.

4) End use

In this section the hot water load is defined.

i. ***Hot water***

Here the facility is selected as house and the number of occupants is chosen as 6. The daily hot water use estimated by RETScreen is 360 L/day for 6 people which is based on a desired water temperature of 60 °C [77]. From NRCan's solar buyer's guide it was stated that DHW consumption of 5+ people homes is 300 L/day [61]. The RETScreen suggested value will be used as 300L/day does not differentiate between 5 and 6 occupants. The hot water setpoint temperature is set at 125 °F or 51.7 °C in both RETScreen and BEOPT. The water main temperature was calculated automatically by RETScreen as between 1 °C minimum and 9 °C maximum based on the local weather conditions provided earlier. "Operating hours" was selected as 24 hours/day. So, water is available all day even if it is practically mostly used during certain parts of the day. The DHW heating load is then shown as 7175 kWh which is slightly (4.5%) lower than the 7510 kWh of energy used for hot water heating from BEOPT.

5) Heating

In this section the different heating systems will be introduced and discussed.

i. ***Solar water heater***

a) Inputs

Under solar water heater, tracking was selected as fixed, slope 40 degrees (established in a project A, slope = 40° is superior to slope = latitude). Azimuth was set to 0 (south facing). Type of collector is glazed; manufacturer is Alternate energy technologies and collector model is AE-21. Values for gross area, aperture area were imported from RETScreen library but $F_R\tau\alpha$ and $F_R U_L$values were outdated and not in line with SRCC's latest certification (figure 12) so had to be adjusted. The suggested number of collectors was four. From excel, by dividing the DHW load by the annual generation of AE-21 at the assumed inlet temperature four collectors are found to be needed to satisfy the load. Temperature coefficient for $F_R U_L$ is the quadratic term of the efficiency equation which ranges from 0 to 0.01 W/m^2/°C when linear equations are used the quadratic term should be set to 0 [77]. Solar collector area was 7.7 m^2. Miscellaneous losses are defined as losses due to obstruction of the collector due to dirt or snow. RETScreen recommends a loss value of 2% to 5% for evacuated tube collectors and 3% to 10% for other collectors. This parameter is a function of collector tilt angle, climatic conditions and availability of upkeep (cleaning). Since the collector is mounted at 40 degrees tilt it should be good at snow shedding so 3% losses will be assumed.

Under balance of system, storage capacity was enabled and 75 L/m^2 was selected (same as f-chart). The resulting storage capacity was 535 L. Since this system is placed in areas where freezing temperatures occur, a closed system using glycol is recommended which means a heat exchanger is necessary. Heat exchanger efficiency is defined as the ability of the heat exchanger to transfer the same amount of heat from the solar loop to the service hot water but with a narrower temperature range. It typically ranges from 50% to 85% with 80% being recommended [77]. Miscellaneous losses (which account for pipe and tank losses) can be 1-2% for short distances between collector and the rest of the system and between 4-8% otherwise. For systems with storage an additional 5% to 10% should be added to account for tank losses [77]. For this design, 6% losses will be assumed since the storage tank is kept in a heated area and losses from the pipes can be seen as a form of space heating.

b) Pump Sizing

For pump sizing, the maximum value of the required mass flow rate can be obtained using equation 4 where the maximum value of Q_{useful} for T_{in} = water main temperature is 1634 W. This value is further corroborated by assuming an hour where the irradiance, ambient temperature and water inlet temperature are highest. This means I = 1049.61 W/m^2, T_{amb} = 27.9 °C and T_{in} = 11.75 °C which result in an efficiency of 84% and Q_{useful} of 1711 W which verifies the earlier figure. The maximum temperature difference (T_{out}-T_{in}) can be assumed as 75°C for a system utilizing glycol [62]. The specific heat of Propylene glycol is 2481 J/kg°C [81]. The maximum flow rate then becomes $\dot{m} = \frac{1634}{2481*75} = 0.0088\ \frac{kg}{sec}$. Next, equation 12 can be used.

$$P = \rho g H Q = \dot{m} g H \tag{12}$$

Where P = hydraulic power (W), g = gravitational constant (9.81 m/sec^2), H = head (m) and Q = volumetric flow rate (m^3/sec). ρ = density (kg/m^3) and $\dot{m} = \rho * Q$ = mass flow rate (kg/sec).

Assuming a head of 10 m [62], which is categorized as high head [30]. P = 0.0088 * 9.81 * 10 = $\frac{kg}{sec} * \frac{m}{sec^2} * m = 0.86\frac{kg*m^2}{sec^3} = 0.86\ W$/collector = 0.45 W/m^2, which is comparable to the 0.5 W used by [82] for 2.3 m^2 collector for DHW system. However, According to RETScreen specific pump power should be 3 to 20 W/m^2 of collector area for collectors with aperture areas of 2 to 6 m^2 [77]. So, the value obtained is outside the acceptable range.

In [62], 31.6 W pump was used for 4.4 m^2 of collector area for a DHW solar system which is 7.18 W/m^2 of collector area. This calculation assumed a flow rate of 0.5 m^3/h, a head of 10 m and a pump efficiency of 60% and motor efficiency of 80%.

In [32] the pump size was 59.5 W for the solar loop for 16 m^2 of collector area which is 3.71 W/m^2. This value was obtained from international energy agency solar heating and cooling program task 26 which gave the following relationship $P = 44.6 * e^{0.0181*area}$. By the using this relationship, the pump power for the present system becomes 51.3W for 4 collectors or 6.6 W/m^2 (which is within the recommended range).

An alternative method can be to use the maximum advised flow rate by SRCC from AE-21 collector testing which is 1.18 GPM [66], which is equivalent to 0.41 m^3/hour. The pump does not have to sustain this flow rate but it is the maximum it should be able to achieve. The density of propylene glycol is 1.032 kg/L [81], which is 1032 kg/m^3. From equation 12, the hydraulic power becomes $P = \rho g H Q = \frac{1032*9.81*10*0.41}{3600} = 11.53\ W$. $P_{electrical}$ then becomes $P_{elec} = \frac{11.53}{0.6*0.8} = 24\ W$ for 1 collector and 96 W for all four. In terms of power density, the value is $\frac{96}{1.932*4}$ =12.4 W/m^2 which is within the range given by RETScreen and so will be used.

The price of AE-21 collector is $750 [48] so four collectors cost $3000.

c) Results

The results show that 169 kWh of electricity is needed for the pump, 3481 kWh of heat energy was supplied by the system at a solar fraction of 48.5 %. The solar fraction found in [80] was 40% for south Sweden and 65% for Madison, Wisconsin, for a DHW system with a collector that has $F_R\tau\alpha$ of 0.7 and $F_R U_L$ of 5 using the f-chart method (which assumes 75 L/m^2 storage) .

ii. ***SolarWall***

a) South facing wall

Fig. 51. Design house south facing wall.

In this section, transpired solar collector (SolarWall) will be installed on the non window bearing area of the south facing wall. Figure 51 shows the south facing wall which has been estimated to be 5.8 m wide and 5 m tall. It has 7 windows with an average window size of 0.61 m^2 making the net area of the windowless portion of the wall 24.73 m^2.

b) Inputs

Under load characteristics, the indoor temperature (which is the thermostat setpoint) is chosen as 21℃ for both the base and proposed cases. The R-Value of the wall insulation is set to 10 $\frac{ft^2-°F-h}{BTU}$ which was provided by the landlord and entered into BEOPT. According to RETScreen the value of R ranges from 0.6 $\frac{ft^2-°F-h}{BTU}$ for uninsulated walls to 57 $\frac{ft^2-°F-h}{BTU}$ for super insulated wall systems. The design air flow rate was set to 36 m^3/h per person or 216 m^3/h for 6 people as recommended for residential buildings [77]. According to the United States Environmental Protection Agency (EPA), the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) recommends a minimum of 15 cfm of fresh air flow rate per person inorder to ensure appropriate air quality [83]. This translates to 25.5 m^3/hour per person or 153 m^3/hour for 6 people. Since the earlier figure exceeds the minimum flow rate then it will be used. Space heating operating hours/days is set to 24/7. The heating load is then shown as 10,175 kWh which is increased to 16,788 kWh by increasing its value by 65% for all months. This is done so that the heating load faced by the SolarWall is approximetly half the total space heating load since only half of the house (6/12 rooms) are aligned to the south facing wall. Natural convection between the south side of the house and the other side of the house can not be relied on as doors are mostly kept closed. The other half of the space heating load will be addressed using flat plate collectors in the next section.

Under resource assessment, the collector is selected as have a fixed tilt of 90° (wall mounted) and an azimuth of 0° (south facing), Under solar air heater properties, transpired plate is the type and standard operation is the design objective. There is an inverse relationship between inlet air temperature rise and system efficiency. High temperatures can be achived by reducing the flow rate but it leads to lower efficinecy. High efficiency is obtained by increasing flow rate and reducing temperature rise. This is dubbed “high air volume” systems. Standard operation is somewhere in between these two extremes [77, 85].

Conserval engineering inc. solarwall was selected [84]. Black was selected as the colour as it offers the highest solar absorptivity (0.94). Performance factor is a ratio of the collector efficiency to the efficiency of a rated collector that was tested at an irradiance of 900 W/m^2 with an airflow of 4 scfm/ft^2. Performance factors range from 0.51 to 1.29 [77]. The default value 1.0 was kept. Solar collector area is 24.73 m^2 and resulting capactiy is 17.3 kW.

Solar collector shading depends on the latitude of the location and the altitude angle the sun must be in order to clear obstructions. For a latitude of 40° and an altitude angle of 15° to 20° of adjacent buildings, the shading percentage is between 6 and 15% [77]. For this project 10% will be assumed (based on location and observation). Wind sheltering is a coefficient describing the loss in wind’s speed due to nearby obstancles. RETScreen reduces the windpseeds entered in location section by the value of wind sheltering. Speed of the wind hitting the collector has an effect on its energy output. In a recent study from the univeristy of Concordia, researchers found that there is a 21% difference in the thermal gain and efficiency of solar thermal collectors placed on the same roof but in leeward side vs the windward side [86]. In a 2020 article in springer, collector thermal efficiency dropped from 60% for 0.1 m/s wind speeds to 25% for 0.6 m/s wind speeds (measured by an onsite anemometer in Lublin, Poland) [87]. For this project, 30% wind sheltering will be used, which is what RETScreen recommends for low rise builidings.

In this system, fans will be needed to move pull/air through the SolarWall and into the house. simple sizing can be done using equation 13

$$P = \frac{dp\ q}{\mu_f \mu_b \mu_m} \tag{13}$$

Where: P is the power used by the fan, q is the air flow rate in m^3/sec, μ_f is the fan efficiency, μ_b is the belt efficiency and μ_m is the motor efficiency and dp is the total pressure increase.

For a small system $\mu_f = 0.9$, $\mu_b = 0.78$ and $\mu_m = 0.4$. For this design the air flow rate is 108 m^3/h (half the total flow rate as this is for the south facing half of the house) or 0.03 m^3/sec. Pressure increase can be assumed as 100 Pa for a small unit. This results in P = 10.7 W of total fan power or 0.432 W/m^2 fan power per m^2 of collector area installed [88].

According to OMFRA, the cost of 100 m^2 of SolarWall from Conserval engineering inc. is $25,300 installed cost, which is equivelant to $6256 for the current project (24.73 m^2 area) [89].

c) Results

The results of the simulation show 154 kWh of energy used by the fans. 2904 kWh of heating energy delivered and 2706 kWh of building heat losses recaptured. These figures are interesting to examine while the SolarWall is experiencing more shading than a rooftop installation, it is also experiencing lower wind speeds due to sheltering and it experiences an additional 93.2% of energy that is due to the recapturing of heat losses of the house by the SolarWall. This could point to the existing insulation being subpar. The collector fan flow rate is 8.7 m^3/h/m^2 and average air temperature rise is 6.6 ℃. Overall 5610 kWh of heating energy was delivered which represents 33.4% of the half load. Boiler was the auxillary heating unit. According to the collector manufacturer, SolarWall can heat the incoming air up to 24 ℃ above ambient temperatures and achieve 20-50% of the heating load (solar fraction) [84]. It should be noted that the solar fraction obtained in this section is similar the solar fraction reported in the methodology section for the case where there was no storage. RETScreen does not seem to have an option of adding storage for air solar systems.

iii. *Solar air heater*

Since 24.7 m^2 of SolarWall area was used in the previous section, the rooftop solar air heater designed in this section will also have approximately the same area for ease of comparison. It should be noted that design of liquid solar collector for space heating in RETScreen was not possible and there is a lack of online resources indicating that such design is possible.

The load characteristics for this collector array is the same as the SolarWall load characteristics. The system has a fixed tilt of 40° and the heating load is 16,788 kWh, which is half the space heating load. The collector is a glazed type Cansolair RA 240 Solar max. the collector aperture area, gross area, $F_R\tau\alpha$ and F_RU_L were obtained automatically from RETScreen's database as 2.29 m^2 2.51 m^2 ,0.58 and 5.83 respectively. The number of collectors was initially chosen as 10 so that total collector area is close to 24.73 m^2 (25.1 m^2). However, it seems that increasing the number of collectors more than 3 has no effect on the energy saved (solar fraction) and so 3 collectors were chosen to increase the economics to reasonable values. This could be due to the lack of storage placing a limit on the maximum amount of useful energy the solar air heater system can supply (i.e. extra energy is excess). The capacity of the collector array is 17.6 kW. The miscellaneous losses are set to 3% (same as solar water heater). The fan power is 0.432 (same as SolarWall) and the initial cost is $2500/collector however this is a 2014 price which might have dropped by now [157].

In the results, the electricity consumption of the fan is 157 kWh and the heating delivered is 3717 kWh, the solar air heater seasonal efficiency is 12.7% and the energy saved (solar fraction) is 22.1%. boiler was used as the auxiliary heater where the energy generated by the boiler reduces from 16,788 kWh before the solar system to 11,178 kWh after. Although it should be noted that to use this system a duct system will have to be installed which might take up space and incur high costs. Alternatively, an air to liquid heat exchanger can be used in order to be compatible with the current hydronic baseboard radiator system.

6) Overall results

In this section the results of the three systems are shown and discussed as can be seen from table XXVIII.

TABLE XXVIII
SUMMARY OF SCENARIO G RESULTS

Category	Metric	Solar Water Heater	SolarWall	Solar Air Heater	Total
	Solar fraction (%)	48.5	33.4	22.1	30.3
Energy	Thermal energy supplied (kWh)	3481	5610	3717	12,808
	Electricity used (kWh)	169	154	157	480
Emissions	CO_2 saved (tonnes)	0.87	1.40	0.93	3.2
	Payback period (yr)	8.1	13.4	18.9	20.6
Financial	Cost ($)	3000	6256	7500	16,756
	Profit ($)	5258	6423	2846	14,527

i. *Energy*

By comparing SolarWall with solar air heater, we can see that the air heater is producing 28% more energy with the same collector area. this could be because 0% shading losses were assumed due to the installation being on the rooftop clearing nearby obstacles but if we include the energy saved by the SolarWall in terms of building heat recaptured then the SolarWall is generating 51% more energy. So SolarWall could be a great technology to use for houses with subpar insulation.

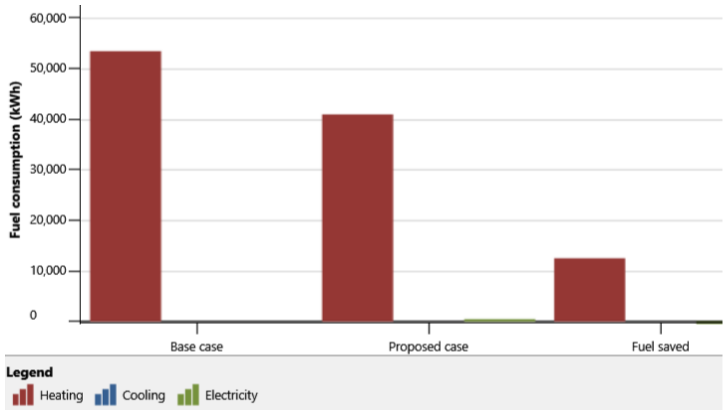


Fig. 52. Energy Savings.

Overall, the combined load was 40.75 MWh of heating energy (BEOPT's thermal load). The cost of the three systems is $16,757. Solar water heater had a payback period of 8.1 years, SolarWall 13.4 years (which is higher than the 12 years advertised [17]) and solar air heater 18.9 years making the total payback period 20.6 years. 311 gallons of fuel were saved each year by the system which is equivalent to $871. Figure 52 shows the kWh fuel savings. The solar fractions were 48.5 % for the water heater, 33.4% for the SolarWall and 22.1% for the solar air collector. The overall solar fraction is 30.3% which is low because of no storage for the air systems and this calculation considering several sources of losses such as heat exchanger, shading, snow covering and piping losses.

ii. *Emissions*

3.2 tonnes of CO_2 emissions were saved by the system per year which correspond to a 23% reduction. The average per capita emissions of buildings sector in NL is 1.8 tonnes per person [157]. Before the solar system, the emissions of the house were above the national average at 2.3 T/person but after it was closer at 1.785 T/person. So even with a low solar fraction the solar system can notably reduce house emissions. The generation of the proposed system has a GHG emission factor of 0.0195 $kgCO_2/kWh$. The fuel mix was 75.5% diesel (heating oil), 1% electricity (pumps and fans) and 23.5% solar energy.

iii. *Financial*

Assuming a 2% inflation rate, discount rate of 6%, 30-year project lifetime (collector design life) and 0% debt ratio (project is funded out of pocket) then the simple payback period is 20.6 years and equity payback period is 17.2 years. The project has a pre-tax internal rate of return of 4.7%. Although it doesn't seem that RETScreen is considering some costs such as the cost of installation, pipes, storage, etc. Compared with scenario E, the emissions reduction, payback period, profit and solar fraction were worse for this scenario.

C. Energy efficiency (BEOPT)

1) Background

i. *Related work*

In [101], the author attempted energy efficiency and conservation retrofits for a group of houses in New York using BEOPT, the results showed that of all energy conservation options considered (infiltration reduction, insulation improvement for the basement, attic and walls and window replacement) the most effective were infiltration reduction, basement insulation and ceiling insulation for an oil fueled house as they were deemed most significant in defining the building's envelope (heat losses). The paper used a database provided by NREL to give the cost for implementing conservation measures. Building orientation and window area had negligible effect on heating demand. The paper compared the results of BEOPT versus the average heating load provided in the residential energy consumption survey (RECS) by the EIA and found that the difference is in the single digits where BEOPT can slightly underestimate the heating energy needed. Though RECS asked many specific questions, it did not ask for technical values such as: "What is the R value of the attic insulation?" or "How many air changes per hour does the house experience?" Rather, it asked if the house was well insulated, adequately insulated, poorly insulated, or not insulated.

There are 2 cases defined, 1- initial conditions, 2- post retrofit house. For wall insulation R-15 batting was used for both cases, for attic insulation R-30 insulation was upgraded to R-60. For the basement, a lack of insulation was upgraded to R-18 whole wall insulation board. Window type was kept as 2 panes non metal frame. Infiltration was upgraded from 74ACH50 to 1ACH50. Basement insulation and significant infiltration reduction both proved to be very effective, though the increased need for ventilation was not considered in the infiltration analysis. Upgrading wall insulation and window type were not as cost effective. The annual amount of heating oil demanded reduced by 41% after the renovations. Assuming an inflation rate of 2%, real discount of 3%, 0% fuel escalation rate and 30 year analysis period (BEOPT's defaults), the payback period was 8.5 years the net present value (NPV) was $5,029 and net savings were $24,780 [101].

In [102], a combined optimization process was developed that uses both HOMER and BEOPT for residential renewable generation and energy conservation hybrid scenarios. The results showed 10% lower costs than the base case and 5% less than the standard process case which relies on standard efficiency improvement practices. Energy demand reduction was based on selecting energy efficient appliances as well as technologies that reduce demand such as higher levels of insulation. The study stated that BEOPT is the most commonly used tool for building optimization and simulation of residential dwellings.

The standard process consisted of energy efficiency measures (EEM) followed by optimization of the hybrid system (HS) represented by a single run. It starts with a set of building elements selected on best practice; these parameters are run through a building simulation tool to calculate the demand profile. This demand profile is then used as an input into a hybrid system optimization tool, which provides the least cost hybrid design. Finally, the costs of the building elements and energy generation supply are added to identify the total cost. The Combined optimization process starts by generating energy demand profiles considering all possible combinations of building elements. Then, each of these energy demand profiles is sent through the hybrid system optimization tool in order to identify the least-cost option for each profile. Consequently, the two costs of building elements and energy supply systems are added up for each combination and the smallest one is selected [102].

The study considered 5 scenarios for the standard process. There are 5 data inputs: house specification, economic inputs, energy sources, hybrid system components and energy efficiency measures. House specifications include house design, dimensions, location, building materials, appliances, orientation, and operational characteristics. Economic inputs included a 3.8% real discount rate and project lifetime of 25 years. Energy source inputs included solar, wind, wood and diesel sources. The hybrid energy system consisted of Wind, PV array, battery bank, diesel generator and converters. Energy efficiency measures were divided into building elements (BE) (such as increased insulation) and electrical appliances (EA) such as replacing old appliances with newer more energy efficient ones. R value was used for BE and energy star rating for EA. The minimum R-Value was defined by the New Zealand (NZ) building code. Since the 5 scenarios used more efficient EA their energy consumption was lower than the base scenario [102].

In scenario 1: cellulose insulation was applied in walls (R = 2.29 m^2/KW which is equivalent to R = 13 hr-ft^2-°F/Btu) and ceilings (R = 3.7 m^2/KW) while fiberglass batting insulation was applied in the ceiling of the crawlspace (R = 1.58 m^2/KW). Windows were upgraded to double pane air filled glass windows (R = 0.36 m^2/KW) and the old fridge was replaced with a 2.0 energy rating Mitsubishi fridge. In scenario 2: cellulose insulation was applied in walls (R = 2.29 m^2/KW) and in the ceiling (R = 3.7 m^2/KW) while fiberglass batting insulation was applied in the crawlspace ((R = 1.58 m^2/KW). Film R-2.04 windows (R = 0.36 m^2/KW), Mitsubishi bottom freezer fridge (2.0 energy rating) and Haier washing machine (1.5 energy rating) replaced the old components. Scenario 3 represented investment into EAs; uninsulated walls, cellulose vented insulation in the ceiling (R = 1.94 m^2/KW), uninsulated floor (R = 0.99 m^2/KW) and single pane windows (R = 0.26 m^2/KW) options were selected. Samsung bottom freezer fridge (3.5 Energy rating), 0.8 Benchmark dish washer and beko washing machine (4) appliances were employed. Scenario 4 considered top insulation with no investment in EAs. Scenario 5 considered top insulation and highly efficient EAs [102].

Energy demand was lowest for scenario 4 of the standard process and the combined optimization process. Net present cost (NPC) was calculated for the EEMs and HS and then summed up. NPC_{EEM} was lowest for the base case and scenario 3. NPC_{HS} was lowest for the combined optimization process and scenario 4. NPC_{Total} was lowest for the combined optimization process (-15%) and scenario 5 (-12%). Scenario 1 was more expensive than the base case and so the least recommended. From the combined optimization process, EEM package 14 yields the lowest total NPC despite not having the lowest energy demand. For EEM 14 the wall insulation was R 2.29 FG, the ceiling insulation was R 6.69 , crawlspace insulation was R 1.58, windows were Film, refrigerator was rated at 413 kWh/yr consumption, dish washer was benchmark ,clothes washer was rated as 490 kWh/yr, refrigerator 2 was rated as 382 kWh/yr. The EEM with the lowest demand solution did not guarantee the lowest cost solution because the size and characteristics of the hybrid system depend on both the total annual energy demand and the hourly demand profile [102].

In [103] the studied building was a new residential house located in Milan (Italy). The case study aims at the minimization of embodied energy (energy required to manufacture the materials and equipment) and investment costs, as well as maximization of electricity and gas savings through energy efficiency measures. Envelope (insulation), appliances and equipment were considered. Simulation was done in BEOPT and compared against the base case. For each option, the estimated overall and incremental costs were provided along with measure lifetime and physical characteristics. The simulated measure savings in electricity and natural gas for each option are given along with associated CO_2 emission reductions as well as the lifetime monetary energy savings and simple payback. Efficiency measures were able to reduce the operational energy required to heat, cool, heat water and run appliances in a building. This study took into account the energy needed to locate, refine, manufacture and install the different efficiency measures in addition to the energy saved through those measures.

ii. ***Energy efficiency scoreboard***

Carleton university's 2019 energy efficiency scoreboard estimated that Newfoundland's electrical energy savings was 0.47% annual incremental savings as a percentage of electricity domestic sales while Ontario's was 1.4% and certain U.S states (with aggressive electricity savings programs) like Vermont had savings of 3% per year. The scoreboard defined electricity savings as having the ability to avoid expensive electricity generation options, increase reliability and reduce risks. For the customer, electricity savings means reduced energy bills, improvement in health and comfort of home environment and increased house durability. For society, the benefits are a reduction in GHG emissions and other negative environmental impacts and a stimulation of the local economy in implementing energy conservation technology [131].

In July 2019, the Canadian government joined the 3% club which is a joint effort between various governments and supporting organizations to work towards achieving 3% annual efficiency improvement. Canada has historically average 1% annual efficiency improvement. Research conducted by the energy scoreboard team suggests energy efficiency spending in Canada increased by 29% from 2016 to 2018 where it was valued at $1.22 billion. Total energy savings were 23.9 PJ in 2018. In 2020, Canada Infrastructure Bank developed a new plan which includes $2 billion of federal capital to be invested in large scale building retrofits throughout the nation [132].

In the 2020 scoreboard, Newfoundland came in the 9th place (second to last followed by Saskatchewan) which is an upward movement of one rank from the last year's scoreboard where NL was dead last. The 2020 scoreboard highlighted that NL faces substantial energy challenges due to cost overruns of Muskrat Falls. A relevant analysis showed that electrification of heat and transportation to be the most valuable mitigation opportunity as it reduces provincial oil expenditure. The province is also preparing to update its building code to increase energy efficiency and commenced rolling out electric vehicle charging network and fuel switching of public buildings from fossil fuels to electricity (supported by the federal low carbon economy fund). The scoreboard suggests NL has an energy poverty problem where more than 38% of the population spend more than 6% of their after-tax income on energy which can be reduced if houses were more energy efficient [133].

iii. *EnerGuide labels*

The Canadian government recommends using EnerGuide labels to shop for energy efficient appliances and heating equipment as they show the product’s energy performance rating relative to the minimum standards set by Canada’s energy efficiency regulations with the most efficient models having the energy star logo/designation [104]. EnerGuide labels show how much energy a product consumes and how it compares to similar models. Installing energy efficient technology raises the home’s energy performance and can increase its resale value while reducing bills and emissions. Canadian and American energy labels follow similar methods to rate energy efficiency but differ due to the models available in each country. EnerGuide label displays: Annual energy consumption (kWh), An arrow that shows the model’s performance relative with the most and least efficient models in the same class, the capacity and type of models in the same class, model number and energy star logo. EnerGuide is compulsory for: clothes dryers, clothes washers, dishwashers, freezers, commercial stoves and ovens, refrigerators, freezers and room ACs. EnerGuide is optional for central ACs, furnaces, water heaters…etc [105].

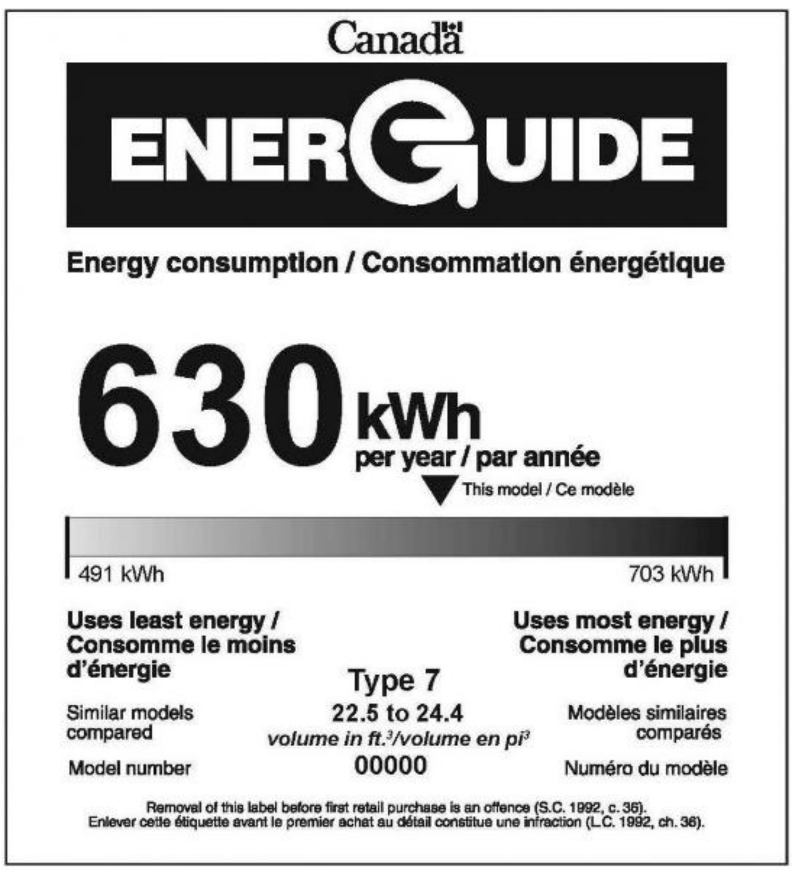


Fig. 53. EnerGuide label [105].

iv. ***Air leakage***

Air leakage (infiltration) is a phenomenon caused by wind, stack effect and mechanical equipment in the building. As wind creates positive pressure on the windward side of the house and negative pressure on the leeward side, air is pulled out of the building (exfiltration) and is replaced by outside air (infiltration). The stack effect is when lighter warm air rises upwards in a building and escapes through the upper levels (exfiltration). This lowers the pressure in lower house levels causing outside air to enter through cracks and gaps (infiltration). This is more prominent in winter since the temperature difference between the indoor and outdoor is higher. Mechanical equipment that exhaust indoor air at a rate higher than that of the air entering the house cause an increase in infiltration. Infiltration is unlike ventilation in the sense that in the former outside air enters the house uncontrollably through cracks and openings while in the latter it is controlled and used to remove moisture and reduce odors. The use of kitchen and bathroom fans is usually sufficient to provide the ventilation needed for simply designed houses. Other more complex measures can include heat recovery and moisture control. It is recommended for both new and old houses to reduce air leakage which is usually higher during colder periods. Higher infiltration can lead to rot or mold in the envelope cavities due to moisture. To air seal a house, an energy audit service can be purchased in which a blower door test is performed where the home is depressurized to reveal the location of most leaks. Sources of leaks can include electrical outlets, door and window frames, vents and fans, gaps around pipes and wires, cable TV and phone lines, baseboards and improperly applied weather stripping around doors. Air sealing can be done by tightening the structure with caulking and sealants. As a result, the tighter house will have lower heating bills, fewer drafts, reduced chance of mold and require smaller heating and cooling equipment [183, 184].

The authors of [185] stated that various studies have concluded that 15% to 50% of heating load in residential and commercial buildings is due to infiltration. A 2020 study in China which examined infiltration in 18 Chinese airports found out that an air change rate of 0.06 - 0.56 1/hr is the most significant contributor to space heating loads as it accounts for 18% to 71% of total heat losses. Improved air tightness and the use of radiant floor heating was found to have the potential of reducing annual heating demand by 84% [186]. A recent study on the contribution of the stack effect on infiltration in high rise buildings in cold climates found that it was responsible for 10.27% of the total winter heating load [187]. A study of the colder region of china showed that for the three zones of public buildings studied air change rate varied from 3.8 to 5.2 1/hr and air infiltration rate from 0.01 to 1.05 1/hr. The study found that the air tightness of buildings built in 2007 were worse than those built in 1990 for the areas in question [188]. European building legislation is establishing increasingly stricter requirements to reduce the energy demand of buildings. Airtightness is an important contributor to air conditioning demand for nearly zero net energy buildings. The current standard in Europe establishes that 0.6 air change rate at 50 pascal as the maximum infiltration rate for all new buildings under the Passivhaus standard regardless of climate zone although the study notes that the effect of infiltration on energy demand is higher for colder climates [189].

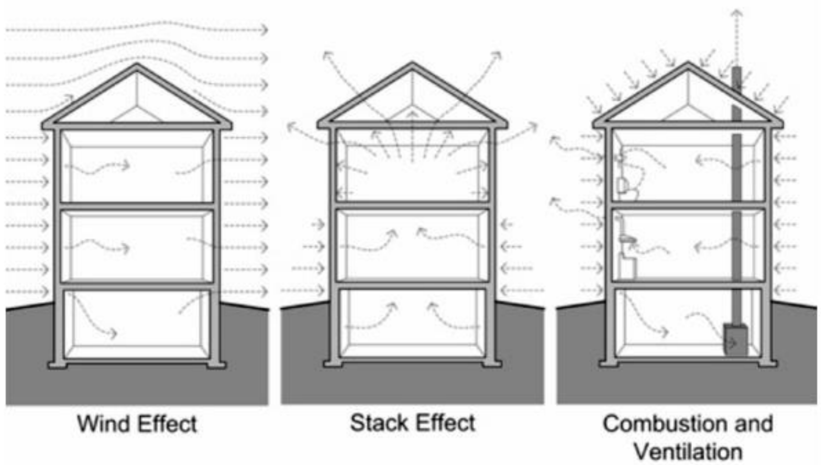


Fig. 54. Sources of air leakage [183].

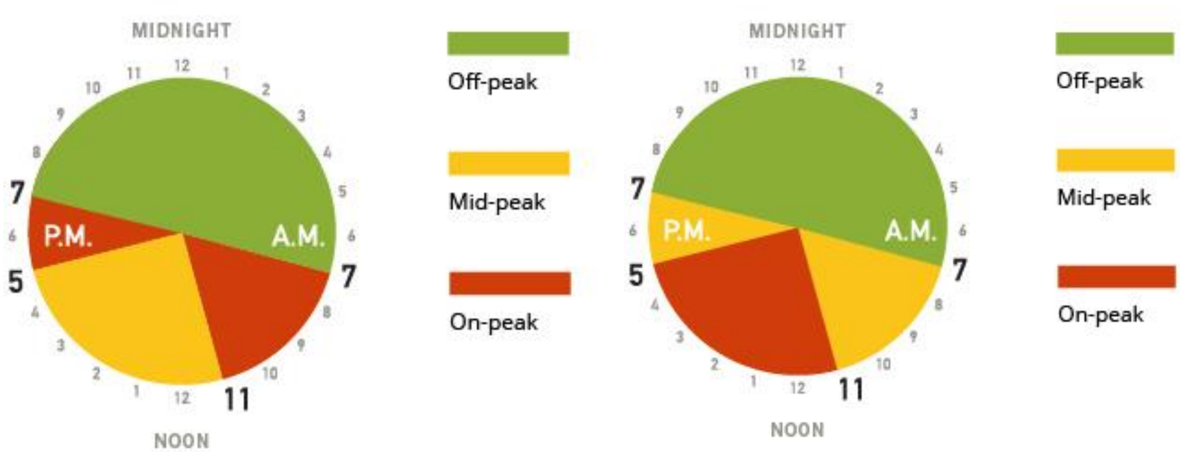
v. ***Demand side management***

According to the EIA, Demand side management (DSM) programs save energy and reduce peak demand which is the most expensive to generate. Demand side management can be divided into energy efficiency and demand response (DR) programs. DR involves the shifting of loads to off peak periods which lowers the chance of blackouts and reduces the use of peak generation (such as combined-cycle gas turbine generators). Around 13 GWh of peak demand savings were achieved in the U.S due to DR in 2018. Energy efficiency on the other hand caused 30 TWh of energy savings and around 6 GWh of peak demand savings. This reduces the need for the addition of extra generation capacity and transmission lines which saves cost and lowers pollution. DR is achieved when smart appliances and smart meters are used thus allowing the utility company to cycle loads to off peak times for lower electricity prices [175,176].

vi. ***Time of use pricing***

Time of use (TOU) pricing refers to a rate plan that charges consumers based on amount consumed and when it is consumed. The prices will vary based on season, time of the day and day of the week. TOU reflects the cost of electricity generation at different times. NL currently has a flat rate of 12.3 cents/kWh while starting November 1st, 2020, Ontario moved back from a flat rate of 12.8 cents/kWh (due to COVID-19) to a TOU pricing scheme. The TOU scheme was set for winter (Nov 1st to April 30th) and will be revised by Ontario's energy board on May 1st, 2021 (for summer rates) and again on Nov 1st, 2021 (for new winter rates). The plan is divided into three categories: off-peak pricing of 10.5 cents/kWh, mid-peak pricing of 15 cents/kWh and on-peak pricing of 21.7 cents/kWh. On-peak is in effect on weekdays from 7 am to 11 am and again from 5 pm to 7 pm, mid-peak from 11 am to 5 pm and off-peak from 7 pm to 7 am while the weekends and national holidays are exempt and priced as off-peak. For the latest available summer season (defined as May 1st to October 31st), On-peak ran from 11 am to 5pm, mid-peak from 7am to 11 am and again from 5 pm to 7pm while off-peak ran between 7 pm and 7 am.

Ontario gives its residence the option to opt out of TOU pricing and instead be billed using a Tiered pricing system. In tiered pricing, a consumption threshold is set where below the threshold the consumer pays a lower price and above it a higher one. The tiered pricing is different between winter and summer seasons. For winter (November 1st, 2020 to April 30th, 2021), below 1000 kWh/month consumption, the rate is 12.6 cents/kWh and above it 14.6 cents/kWh. For the past summer (March 24th, 2020 to October 31st, 2020) the rate was 11.9 cents/kWh for consumptions less than 1000 kWh/month and 13.9 cents/kWh for higher. Because of the similarity in the flat rate, Ontario's TOU pricing can be applied in this project to see what would happen if NL adopts such a policy and to examine the efficacy of smart appliances in reducing the electricity bill however it should be noted that NL is unlikely to adopt TOU since its generation mix will soon be 98% hydroelectric while Ontario's electricity mix, 60% of which came from nuclear in 2018, is more reliant on generation technology that is designed to run all the time (baseload generation) and can not load chase leading to the usage of high cost generation during peak times and lower loads during off-peak times which is a problem for technologies like nuclear that can not be easily scaled back without incurring high costs. Summertime usage is also different in Ontario as the province has high AC ownership (around 80%) due to higher temperatures while Atlantic Canada has the lowest rates in the country (around 30%). An alternate way of looking at this section is to examine what the design house (given its properties and technologies) would pay if it was placed under a different pricing and generation environment. Dr Jim Feehan (Professor at MUN) has also argued back in 2012 that if NL was to implement time of use pricing and energy efficiency measures the province's consumption would decrease to a point where the addition of new generation capacity (Muskrat Falls) would become unnecessary. He argued that an important economic principle is that, for economic efficiency, the price of a commodity should at least be equal to the price of generating an additional unit of it which he proposes means that variable pricing is necessary to deal with seasonal differences in electricity consumption and peaks in demand and to control consumption growth in the province. [12, 177, 178, 181, 182].



a b

Fig. 55. TOU pricing periods; a) Winter TOU periods, b) Summer TOU periods [178].

2) *Electrical appliances*

In this section new appliances will be selected for a retrofit. The models will be selected so as to provide the same/better function as the old/existing devices but at reduced energy consumption/higher efficiency. The models are picked from the 2020 most efficient appliances list from energystar.gov. A second option is chosen which while minimizing cost and increasing efficiency adds the option of demand response compatibility for smart grid solutions. From BEOPT, the highest electrical energy consumer is the miscellaneous category (which includes the plug loads) at a consumption rate of 3889 kWh, followed by lights (2881 kWh) which is followed by large appliances (2348 kWh). So electrical appliances and lighting will be retrofitted. Note: labor costs are assumed as 0 for all electrical appliances as no installers are required. The default appliances in BEOPT are dated back to before 2007 so the effect of newer appliances can be examined in this section. The cooling load here refers to the dehumidifier.

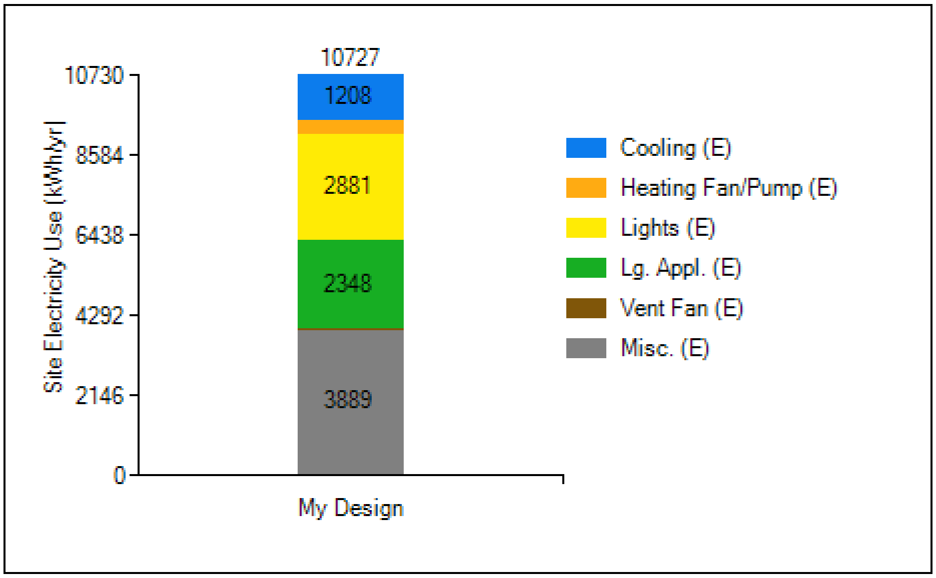


Fig. 56. Electricity consumption from BEOPT.

i. *Clothes washer*

The energy star most efficient designation is awarded to the best (efficiency and price) products in their categories every year. From the list of 2020 best washers, Samsung WF45N53**A* was selected due to price, availability, power consumption and water usage [107]. The device has an estimated yearly energy cost of $7 (for 75 kWh consumption) when used with a natural gas water heater [108]. The base-case top-load washer had a modified energy factor of 2.47 ft^3/kWh-cycle, annual rated consumption of 123 kWh, it was tested in 2007, had a drum volume of 3.68 ft^3, had thermostatic control (also available in replacement model). Occupancy rate and hot water multiplier were kept as 1.0 for both old and new systems. The lifetime of the old system was 14 years and the same was assumed for the new one. By inputting the new washer into BEOPT its electricity use is 36.8 kWh/yr and water volume 1.2 gal/unit/day. The smart washer is Bosch WAW285H1UC (only one available) had an electricity use of 70.3 kWh/yr (higher than base-case) and water volume of 2.8 gal/unit/day.

TABLE XXIX
CLOTHES WASHER PARAMETERS

Metric	Value		
	Old Washer	New Washer	Smart Washer
Load configuration	Top Load	Front Load	Front Load
Electricity use (no heating) (kWh/yr)	46.6	36.8	70.3
Water volume (gal/unit/day)	3.0	1.2	2.8
Volume (ft^3)	3.68	4.5	2.2
Energy Star certified	Yes	Yes	Yes
Connected	No	No	Yes
Integrated Modified Energy Factor (IMEF)	2.47	3.0	2.2
US federal Standard IMEF	--	1.84	1.84
Annual Energy Use (kWh/yr) (rated)	123.0	80	84
Integrated Water Factor (IWF)	--	2.9	3.7
US Federal Standard IWF	--	4.7	4.7
Annual Water Use (gallons/yr)	--	3850	2379
Annual cost with gas DHW ($)	9	7 [108]	7 (assumed)
Price ($)	662	344.5	1699 [115]
Test Date	2007	2016	2020
Lifetime (years)	14	14 (assumed)	14 (assumed)

Definitions:

- Load configuration: Front or top load.
- Volume: Tub capacity measured according to the U.S. DOE test procedure.
- Connection: Whether the model offers advanced controls and functionality allowing a service center to perform remote diagnosis. It also provides smart grid compatibility which allows an energy provider or energy management system to dynamically adjust use to suit hourly energy supply (time of day pricing) so as to reduce electric bills.
- Integrated Modified Energy Factor (IMEF): A performance metric for certified clothes washers. Higher IMEF = higher efficiency. It is the quotient of container capacity divided by total energy consumption per cycle, the energy required to remove moisture from the load and the low power mode consumption.
- U.S Federal Standard IMEF: Minimum IMEF as defined by the U.S DOE.
- Annual Energy Use: Estimated annual energy use under normal conditions including energy used to heat the water. Unlike IMEF it does not address the washer's effectiveness at removing moisture. It is based on annual

usage of 295 loads (6 loads per week). Actual consumption will vary depending on laundry load and settings selected.

- Integrated Water Factor (IWF): Water performance of certified washer. The lower the IWF the more efficient the washer at using water. It is the quotient of total per cycle water consumption divided by the capacity of the washer.
- U.S Federal Standard IWF: Maximum (worst) IWF a certified washer can have.
- Annual Water use: Estimated water use of the washer under normal conditions. Based on a 295 load per year usage.
- Total Electricity use: Total consumption assuming 125 °F water which does not include energy needed to heat the water. (calculated by BEOPT).
- Total Water Volume: Volume of hot and cold water required per day assuming a water heater set point of 125 °F (calculated by BEOPT).

ii. ***Clothes dryer***

According to NRCan, Energy Star certified clothes dryers use 20% less energy than standard models. Some models come with an "extra low heating" setting which dries clothes for a longer period using lower temperatures which saves energy. Another variant is the "heat pump" dryer which consumes less energy by using ambient heat to dry clothes [110]. The cheapest option was Samsung DV22N680*H* Dryer which comes with heat pump technology [112]. The only smart dryer from the 2020 energy star list is Bosch WTW87NH1UC [114]

TABLE XXX
CLOTHES DRYER PARAMETERS

Metric	Old Dryer	New Dryer	Smart Dryer
Type	Electric	Electric	Electric
Heat pump technology	No	Yes	Yes
Drum Capacity (ft^3)	4.0 (assumed)	4.0	4.0
Ventilation	Vent (100 cfm exhaust)	Ventless	Ventless
Connected	No	No	Yes
Combined Energy Factor (lb/kWh)	3.1	5.85	6.8
Estimated Energy Test Cycle (min)	--	60	44
Cost	$760	$899	$1699 [113]
Lifetime (years)	13	13	13
Auto Termination	Timer	Moisture	Moisture

Definitions

- Heat pump technology: This technology heats, dehumidifies and circulates air into the dryer so as to eliminate the need for a vent. A heat pump model solely uses heat pump technology while a hybrid model combines heat pump with traditional electric resistance heating.

- Ventilation option: Vented Dryers exhaust evaporated moisture to outside of the house. Ventless dryers use a closed loop system with a condenser to remove moisture from heated air and discharge it down the drain.
- Connection: Whether or not the dryer is connected and can be operated remotely making it smart grid and demand response management compatible.
- Combined Energy Factor (CEF): Energy performance metric which if higher represents higher efficiency, it is in lbs/kWh unit as it is obtained by dividing the weight of the test load by the sum of standby, off mode energy consumption and per cycle electricity consumption.
- Estimated annual energy use: It is based on CEF and an annual usage of 283 cycles per year. Actual consumption will vary depending on usage patterns such as type of cycle selected, load size and frequency of use.
- Estimated Test cycle time: Products must complete the test cycle in less than 80 minutes to obtain ENERGY STAR certification.
- Auto Termination: Determines how the dryer terminates a cycle (time, moisture or temperature). Newer dryer use moisture sensors.

iii. ***Refrigerator***

The old freezer had an annual consumption of 434 kWh/yr, volume of 18 ft3, adjusted volume of 20.9 ft^3, rated annual consumption of 434 kWh/yr, no ice dispenser (ice dispenser adds to energy consumption), occupancy energy multiplier of 1.0, automatic defrost and a lifetime of 17.4 years. The selected fridge (Insignia NS-RTM18SS7) has a 18.0 ft3 (14 ft^3 fridge and 4.1 ft^3 freezer), annual consumption of 362 kWh/yr, no ice dispenser and automatic defrost. It costs $500. The U.S federal standard (maximum permittable energy consumption) is 403 kWh/yr [109]. The smart fridge is LG LRMXC1813* (only connected model on the list) which is available from best buy at $1,999 black Friday price [120,121].

TABLE XXXI
REFRIGERATOR PARAMETERS

Metric	Value		
	Old fridge	New fridge	Smart Fridge
Annual consumption (kWh/yr)	434	362	576
Volume (ft^3)	18.0	18.0	18.3
Adjusted Volume (ft^3)	20.9	20.7	20.9
Rated annual consumption (kWh/yr)	434	362	576
Ice dispenser	No	No	Yes
Defrost	Automatic	Automatic	Automatic
Demand response control (connected)	No	No	Yes
Price ($)	619.15	500	1,999
Lifetime (years)	17.4	17.4 (assumed)	17.4 (assumed)

Definitions

- Defrost Type: Can be automatic, manual or partial.
- Thru the door dispenser: Allows access to water and or ice through an external dispenser (attached to the door). This however increases energy use by 14-20% [109].
- Capacity: Total volume of the fridge and freezer combined. The larger the size the higher the energy consumption.
- Connection: Whether the model offers advanced controls and functionality allowing a service center to perform remote diagnosis. It also provides smart grid compatibility which allows an energy provider or energy management system to dynamically adjust use to suit hourly energy supply (time of day pricing) so as to reduce electric bills.
- Annual Energy Use: Estimated annual use in kWh of the refrigerator under typical conditions. Actual consumption varies depending on frequency of opening/closing, ambient temperature, how full the refrigerator is kept.
- US Federal Standard: Maximum energy consumption allowed under U.S DOE regulations for a refrigerator of similar size and type.
- Adjusted volume: Fresh volume + 1.63 * freezer volume.
- Annual consumption: Actual fridge consumption as calculated by BEOPT.

iv. ***Dishwasher***

The base case dishwasher has 318 kWh rated power, certified by Energy guide in 2007 and consumes 148 kWh/year according to BEOPT. The replacement Dishwasher was selected based on minimization of power, water usage and cost while maintaining energy star most efficient designation. The new dishwasher model is Beko DUT25401** [122] it costs $479 and is 24% more energy efficient than the U.S. standard and 47% more water efficient. The smart dishwasher is Blomberg DWT 81900 **** [123]. It costs $1049, has a rated power of 225 kWh/yr and is 27% better than the U.S. standard in terms of energy consumption and 53% in terms of water consumption. It was assumed that all units have internal heat adjustment ability and unitary occupancy energy and hot water multipliers.

TABLE XXXII
DISHWASHER PARAMETERS

Metric	Value		
	Old Dishwasher	New Dishwasher	Smart Dishwasher
Wi-Fi	No	No	Yes
Number of place settings	8	14	16
Annual rated consumption (kWh/yr)	318	234	225
US Federal Standard (kWh/yr)	307	307	307

% better than US federal standard	-3.5	24	27
Water Use (gallons/cycle)	--	2.66	2.37
US Federal Standard (gallons/cycle)	5	5	5
% better than US Federal Standard	18	47	53
Electricity Usage (kWh/yr)	148	90.8	74.7
Hot Water Volume (gallons/day)	4.1	1.3	1.1
Price ($)	596	479	1049
Lifetime (years)	11	11	11
Energy Guide Annual Operating Cost ($)	24	21	20
Energy Guide Date	2007	2020	2020

Definitions

- Dishwasher type: Standard or compact with standard-sized being defined as a unit that has at least eight place settings and six serving pieces.
- Connected: Ability to remotely operated and controlled for demand side management purposes
- Annual Energy Use: Estimated annual energy use under normal conditions. This includes machine energy, water heating energy and standby losses. It is based on a 215 load per year usage (4 loads/week). Actual usage will vary based on usage pattern.
- US Federal Standard (energy): The mandatory energy standard for residential dishwashers as expressed by the U.S. DOE.
- Water Use: Estimated per cycle water use under normal conditions. It is the number of gallons delivered to the machine to complete a cycle.
- US Federal Standard (water): The mandatory water efficiency standard for residential dishwashers as expressed by the U.S. DOE.
- Electricity Usage: Total electricity consumption of the unit assuming a hot water temperature of 125 °F (calculated by BEOPT). Does not include water heater energy consumption.
- Hot Water Volume: The volume of water required by the unit assuming a hot water temperature of 125 °F in terms of gallons per day (calculated by BEOPT).

v. ***Cooking range***

There are no ENERGY STAR label for residential ovens, ranges or microwaves [116].

3) Heat generators

i. *Boiler*

The existing boiler is 80% efficient (Annual Fuel Utilization Efficiency, AFUE) and uses heating oil with a design temperature of 180 °F. According to the calculation presented in the methodology the maximum deficit for the lower boundary is -27.11 kW i.e. there is no hour of the year where the load required by the boiler is higher than 27.11 kW = 92 kBTU/hr. The chosen boiler is the Energy Kinetics' Resolute EK1RT model which is an oil or natural gas fired boiler which has an AFUE of 91.3% and a capacity of 95 kBTU/hr which represents 14% savings over the federal minimum. It comes with an energy manager which increases efficiency, uses high performance plate heat exchanger, is compatible with the existing baseboard heater infrastructure, produces less noise, is a noncondensing combi boiler, is energy star certified, is a 2018 model and has a 30 year lifetime. Manufacturer claims it can provide up to 40% oil bill reduction over standard boilers [117,118].

The model was inputted to BEOPT where $1000 installation fee was assumed [119] and 16.32 $/kBTUhr was calculated. The cost of boiler was not obtainable and was assumed as $2155.61 (including installation) based on an oil burner of similar size and efficiency [119]

TABLE XXXIII
BOILER PARAMETERS

Metric	Value	
	Old Boiler	New Boiler
Fuel type	Oil	Oil or propane
System type	Hot water forced draft	Hot water forced draft
Rated AFUE (Btu/Btu)	0.8	0.913
Installed AFUE (Btu/Btu)	0.8	0.893
Design Temperature (°F)	180	180
Has Open Flue	True	False
Modulation	False	True
HIR	1.25	1.12
Cost ($)	1933.5	2155.6
Lifetime	24	30

Definitions

- System Type: Choice between "Hot water, condensing", "Hot water, forced draft" and "steam".
- Rated AFUE: Measure of the annual or seasonal efficiency of furnace or boiler. It takes into account energy losses as the unit responds to changing loads which is affected by weather and occupant thermostat setting.
- Installed AFUE: Accounts for performance degradation *relative* to rated AFUE value. For the selected boiler the installed AFUE is 89.3% (2% lower than rated).

- HIR: Heating input ratio to the boiler. How much heating input is needed to produce one unit of heating output from the boiler (inverse of AFUE).
- Design Temperature: Temperature of the outlet water.
- Modulation: Whether the boiler can be fully modulated (reduce its output) or not. Higher efficiency boilers can be modulated.

a) Annual Fuel Utilization Efficiency

The AFUE measure is based upon a heat loss method and involves measurement of excess air and flue gas temperature over operating cycles considered typical of U.S. average conditions. This method considers heating load only, not DHW. In the case of a boiler, the prescribed conditions are supply temperature 140 °F, return temperature 120 °F, burner average on-time (9.68 minutes) and burner average off-time (33.26 minutes). The standard for this measure for boilers is maintained by ASHRAE [123].

According to Energy Kinetics (company that makes EK1RT) AFUE is not an accurate measure of boiler efficiency as typical boilers frequently perform way below their rated AFUE. Idle losses occur when heating a heavy boiler to 180 °F, which can take up to 15 minutes just to heat the boiler itself, if that energy can not be extracted before it dissipates the fuel used is considered wasted. This can lower the AFUE of a boiler by 15% to 40%. The company states that efficient boilers perform closer to their rated AFUE and are characterized by low mass, low water content, high insulation, use of plate heat exchangers, lack of draft regulator and presence of thermal purge control. The thermal purge control predicts end of heat call and pumps the left-over heat in the boiler out to the hot water tank to prevent that energy from being wasted, this results in year-round increased efficiency. This claim relating to the inaccuracy of AFUE (especially for boilers that make hot water) is stated to be supported by a 2007 U.S. DOE study [123][124].

The company claims that a typical condensing cast iron boiler with tankless coil will have an actual AFUE more than 35% lower than its rated AFUE while the Resolute series (which includes the selected boiler) has an actual AFUE only 2% less than its rated. The company states that rated AFUE is more representative of peak efficiency. Based on this they claim that their 91.3% boilers are more efficient than a 95% AFUE condensing boiler [124].

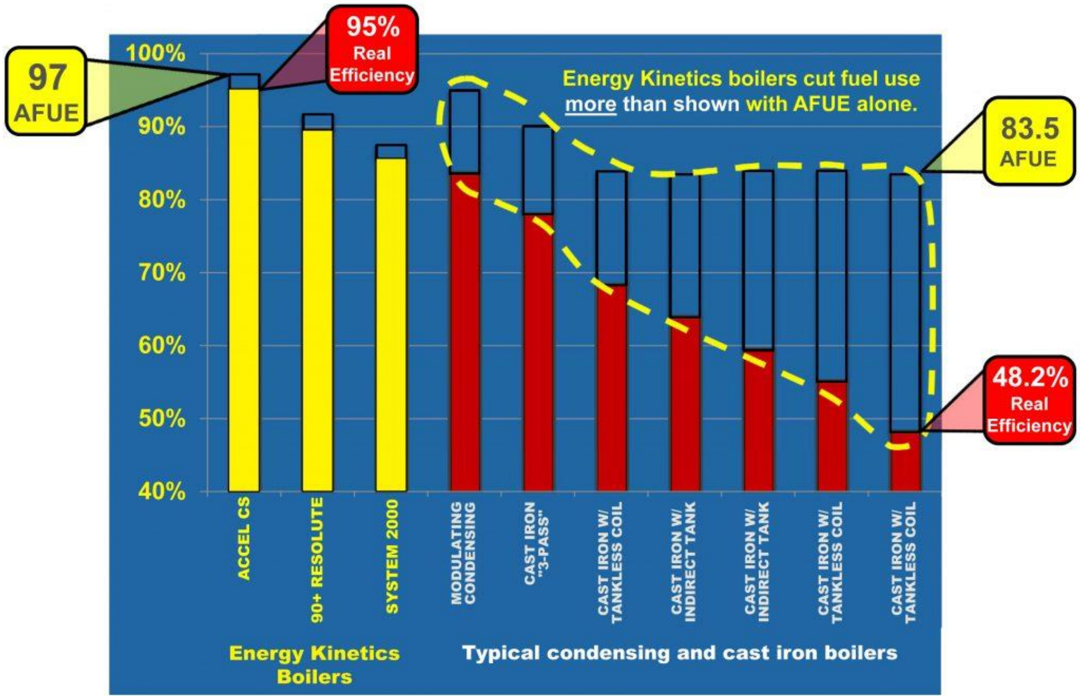


Fig. 57. AFUE comparison by Energy Kinetics [124].

ii. *Water heater*

The existing water heater (WH) is a standard oil fueled water heater with an energy factor of 62%, tank volume of 50 gallons, input rating of 90 kBtu/h and a lifetime of 11 years. The new water heater is A. O. Smith - Heat Pump: HP1080H45DV. It is an electric water heater that uses heat pump technology to achieve an energy factor of 3.45. It has an input rating of 4.5 kW and storage volume of 82 gallons. It comes with a 10-year warranty but electric water heaters according to BEOPT and reference [135] are expected to last for 13 years. The water heater costs 1899 before rebate and is claimed to be able to save up to 72% per year in energy costs and has a payback period of 2-3 years. As the heat pump design allows heat from the surrounding air to be transferred to the hot water it will be placed in the basement (same as old WH) as it is warmer due to it being underground and heated. BEOPT assumed an installation cost of $180 which will be used [136,137].

DOE suggests that an Air Sourced Heat Pump Water Heater (ASHPWH) needs to be installed in locations that remain between 40°F to 90°F and have at least 1000 ft^2 of air space. Placing them in the same room as the boiler can mean they can absorb some of the heat released by the boiler and cool down the room temperature [138]. Since the ASHPWH is a hybrid of electric heating and heat pump heating, it can be used in electric mode in the winter and heat pump mode in the summer as removing heat from ambient air is more desirable in the summer than winter. In fact, an ASHPWH is so efficient at removing heat that it can be used for space cooling depending on the hot water consumption of the house. Independent testing of an A.O. ASHPWH proved its COP (energy factor) is as the manufacturer claims (for the same test conditions) [139]. A 2014 study by the Canadian government has shown that ASHPWH (installed in the basement of an R-2000 house) had no impact on winter heating loads but produced significant reduction in summer cooling loads [140]. For this project, for the above-mentioned reasons and since

BEOPT default HPWH model assumes installed energy factor as 95% of COP, the installed energy factor will be assumed to equal 95% the rated energy factor or 3.3.

TABLE XXXIV
WATER HEATER PARAMETERS

Metric	Value	
	Old Water heater	New Water heater
Type	Tank	Tank
Fuel	Oil	Electric heat pump
Volume (gallon)	50	82
Rated Energy Factor	0.62	3.45
Installed Energy Factor	0.62	3.3
Input (fuel or electric)	90 kBtu/h	4.5 kW
Recovery efficiency (%)	78	233
First Hour Rating (gallons/hr)	--	86
Date certified	2007	2020
Cost ($)	1038	1900

4) Lightening

From BEOPT results, lighting constitutes 2881 kWh of the annual load (27% of the load). It is the second largest contributor to the electric load and therefore it is expected that replacing the existing lighting arrangements which consists of 34% Compact Florescent Lightening (CFL) and 66% incandescent lighting (according to owner of the house) will reduce the electric load. The current arrangement has CFLs at an efficacy of 55 lm/W and incandescent light at an efficacy of 15 lm/W.

LED, invented in 1962 by Nick Holonyak, is a type of solid-state lighting that uses a semiconductor to convert electricity to light. They have a smaller environmental impact than incandescent light and CFL (which contains mercury). It is estimated that by 2030 LEDs will make up 75% of all lighting sales [127]. According to the U.S. DOE, replacing incandescent bulbs with energy efficient ones (CFLs and LEDs) can save the house $75 each year. CFLs and LEDs have 25%-80% less energy consumption than normal incandescent bulbs and last 3-25 times longer (especially Energy Star rated LEDs). LED bulb life is around 25,000 hours (which is over 5 years if operated for 12 hours per day) compared with incandescent bulbs which only work for an average of 1000 hours and so LEDs save the most money despite being initially the most expensive [125]. On a larger scale, it is estimated that widespread use of LEDs in the U.S. can save about 348 TWh of electricity by 2027 which represents almost $30 billion savings in electricity consumption and curtails electricity consumption growth rate as well save millions of tons of carbon emissions [126]. LEDs also emit very little heat whereas incandescent bulbs release 90% of their energy as heat and CFLs 80%. Another advantage of LEDs is that they are directional, so no light reflectors are needed to focus the light which increases efficiency [126].

The chosen new lighting is Philips LED – 9290018397. It has a 5-year warranty and consumes 5.5 W which replaces 50 W incandescent bulbs. It can operate for 25,000 hours and has a brightness of 850 lumens at an efficacy of 154.5 lumens/Watt which is the highest of all 2020 Energy Star rated bulbs [128,129]. The price is $13/bulb [130]. There are 12 bulbs per level so 24 bulbs in toto, the area of both levels combined is 3704 ft^2+ which makes the price density 0.34 $/$ft^2$ (0.26 $/$ft^2$ after rebate) while for the base case it was 0.027 $/$ft^2$ (almost 10 folds increase in price density). The base case option had an annual electricity consumption average of 2880 kWh while for the LED case it is 1319 kWh (calculated by BEOPT)

Through surveying the property owner, it was determined that the house uses 34% CFL and 66% incandescent lighting which might be untrue since Canada has banned the import and manufacture of 40-100 Watt low efficiency incandescent bulbs. However, due to certain exemptions, such as those for decorative lamps, appliances and chandelier bulbs, incandescent bulbs continue to be sold by, for example, Canadian Tire till today [143-145]. For the sake of continuity, the value determined by the survey used in project A will continue to be used in this section.

Comparatively, the U.S. enacted a low efficiency light regulation in 2007 banning most incandescent light bulbs starting 2012 and by 2020 another level of restrictions would kick in requiring all general-purpose bulbs to produce at a minimum 45 lumens/watt. However, in 2019 under the trump administration, some of the planned changes were reversed which was met with critic from energy and environment conscious entities [146].

5) *Building Elements*

i. *Roof tiles*

The current asphalt shingles are dark in colour and have an absorptivity of 0.92 and emissivity of 0.91. Ideally, for winter heat trapping, absorptivity should be as high as possible while emissivity as low as possible. For summertime cooling the reverse. To replace the current system with a more winter ideal solution would lead to lower heating loads but also increase summer time temperatures and since the design house has no cooling mechanism this could negatively affect the comfort of the occupants and so the roof tiles won't be changed [142].

ii. *Windows*

The new windows (CLERAHIGHPROFILE2) are from Clera Windows they have a U factor of 0.16 Btu/h/ft² - °F (the minimum from the list is 0.13) whereas the old windows have a U factor of 0.37 Btu/h/ft² - °F. The U factor is the heat transfer coefficient of the windows. Lowering it is expected to reduce heat losses during winter. The new windows have a solar heat gain (SHG) of 0.65 (out of a maximum of 0.69) while the old windows had an SHG of 0.3. SHG is the ratio of solar heat energy that can pass through the windows, increasing it is expected to reduce winter heating loads during the daytime. The new window has 3 layers of glazing (old one had 2) and an energy rating of 58 out of 58. Higher energy ratings signify a slower heat transfer, higher solar gain and lower air leakage and is therefore the most important parameter in window selection. A full house window replacement from the same company costs around $11,000 which is equivalent to 48 $/$ft^2$ (higher than any of BEOPT's saved models) for a total of 226 ft^2 of window area [166,167]. According to Clera Windows, their vinyl windows can last 30 years on average [173]

iii. *Insulation and infiltration*

Living space and basement wall insulation will require labor while attic insulation projects are do it yourself (DIY) jobs that can take as little as 1 day to finish [168,169]. The current wall insulation is R-10 Extruded Polystyrene (XPS) insulation, attic insulation is R-30 fiberglass, finished basement R-10 XPS and infiltration 7ACH50 (which corresponds to an air exchange rate of 0.42 1/hr). Wall insulation will be upgraded to R-15 XPS. Attic insulation will be upgraded to R-60 cellulose insulation, basement insulation to R-20 XPS and infiltration to 1ACH50 (which corresponds to air exchange rate of 0.06 1/hr) based on the work of [101,102]. Based on approximate measurements the total wall area of each level of the house is 1507 ft^2 whereas the cost of R-20 XPS insulation from BEOPT is $2.5/$ft^2$, R-15 XPS $1.97/$ft^2$ while the cost of R-60 cellulose insulation is 2.69 $/$ft^2$. Average attic size of 1200 ft^2 [170] and the lifespan of any insulation is assumed as 30 years [171]. The built-in cost of 1ACH50 in BEOPT is $0.65 per ft^2 of finished floor (which is 3703 ft^2). According to [172] the cost of installation is $1000 for a 1250 ft^2 house and so it is $1250 for a 1500 ft^2 house. This value is close to the $0.95/ ft^2 installation cost given in [174].

K is thermal conductance. It has a unit of Watts/m^2 Kelvin. For example, for 1 K temperature difference across a wall with K = 1 W/m^2 K, 1 watt of thermal energy will flow across one square meter of the wall. R-Value which is thermal resistance is the inverse of the K-value. in warm countries insulation serves to reduce heat entering the building while in cold countries it reduces heat escaping the building. In S.I. units R has a unit of m^2 K/W while in English units it is hr-ft^2-°F/Btu. R value is directly proportional to the thickness of the wall. If thickness is not given it is referring to a standard thickness of 1 inch [180].

6) Rebates

Based on the design house zip code the following rebates [111,134] were offered and so were subtracted from purchase cost.

TABLE XXXV
OFFERED REBATES

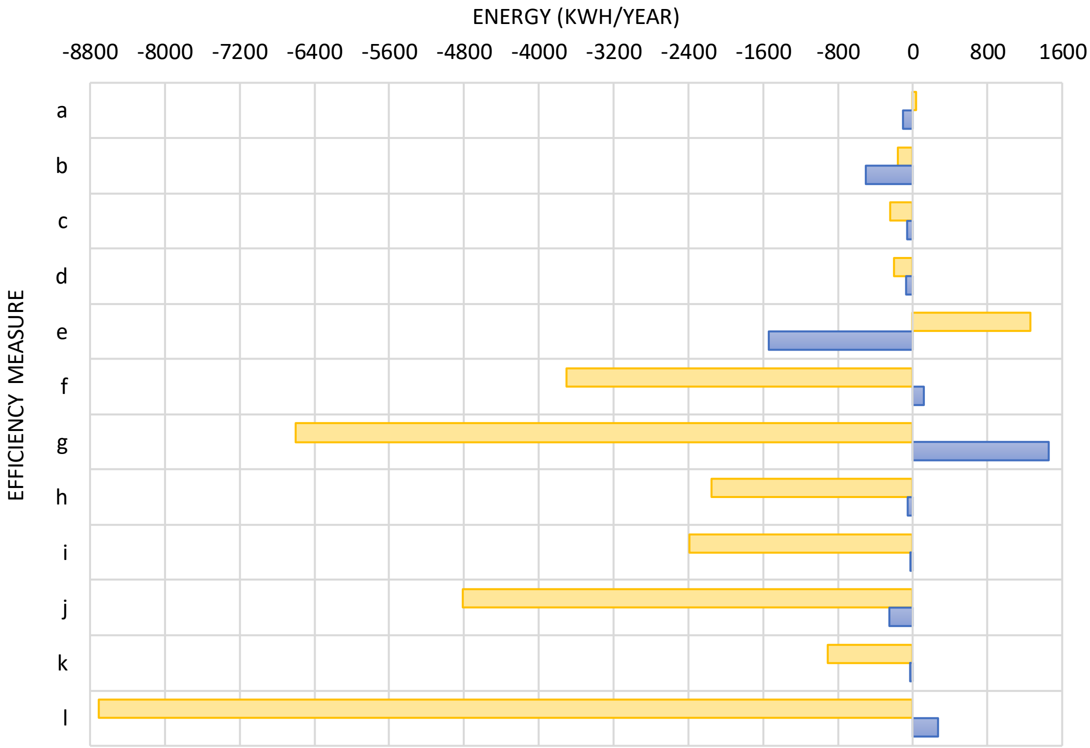
Appliance	Rebate
Clothes Washer	$75
Clothes Dryer	$100
Fridge	$100
Dish washer	$40
Boiler	$100-$400 (assume $250)
Water heater	$500
Cooking range	--
Lighting	$1-$6 (assume $3)

7) *Simulation Results*

For detailed simulation results for each efficiency measure (EM) see appendix D. In the figures shown in this section, information from the results of BEOPT simulations were extracted and processed to obtain the following metrics.

- Electricity usage change: decrease or increase in yearly electricity usage (kWh/year) as a result of the efficiency measure, negative values imply larger decrease and are better
- Oil usage change: decrease or increase in yearly oil consumption as a result of the efficiency measure, negative values imply larger decrease and are better. Converted from gal/year to kWh/year.
- Utility bill change: decrease in amount spent on energy ($/year) as a result of the efficiency measure, larger negative values imply larger decrease and are better.
- CO_{2e} emissions change: decrease in amount of annual (tonnes/year) and lifetime (tonnes) emissions produced by the house as a result of the efficiency measure, larger negative values imply larger decrease and are better.
- Cost: after rebate cost ($) of purchasing and installing the efficiency measure. Larger positive values imply larger costs and are worse.
- Lifetime: period (years) the efficiency measure is designed to be viable for. Larger values improve the economics of implementing the measure.
- Payback period: number of years required to recover the investment. Calculated using the utility bill savings, EM cost and EM lifetime
- Price of CO_2 abated: monetary value ($) of internalizing the emissions externality calculated by assuming a $50/tonne CO_2 price which is a standard for Canada that will come into effect in 2022 [141]. In NL the tax on light fuel oil increased from 5.37 cents/L to 8.05 cents/L on Nov 7th, 2020 [193].
- Profit: amount of money ($) to be gained after the cost of the EM is paid back. Profit after the inclusion of the value of carbon abated was also calculated.

The potential electricity price increase (to 23 cent/kWh) if the Muskrat Falls price stabilization plan fails [1,2] was also accounted for and its effects on utility bill change, payback period and profit calculated.

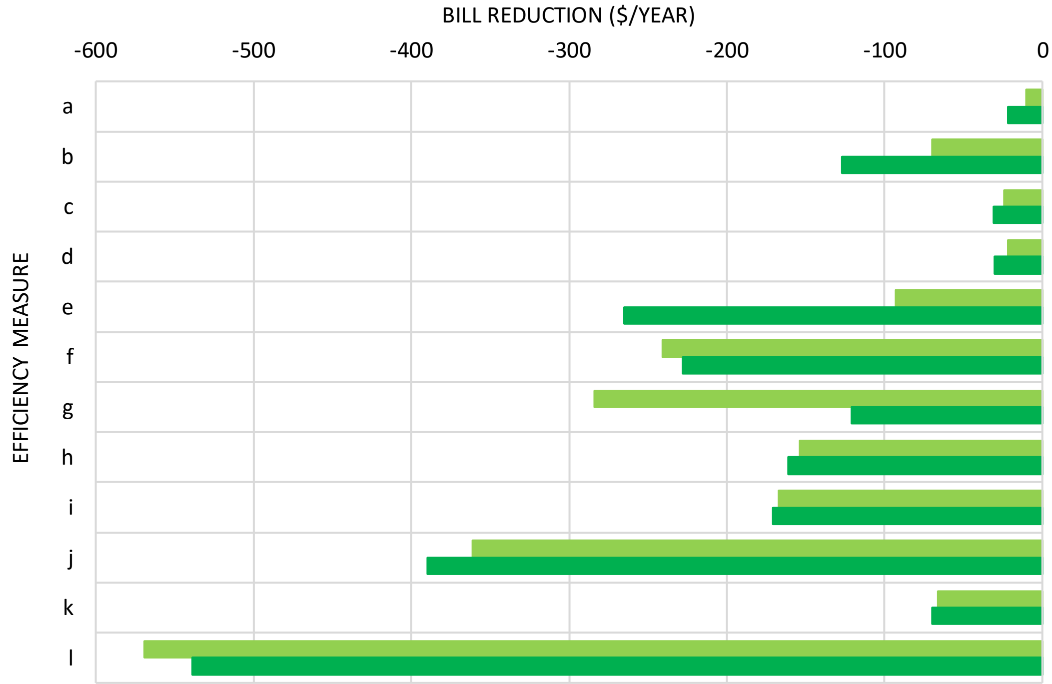


	l	k	j	i	h	g	f	e	d	c	b	a
Oil usage change	-8707	-913	-4814	-2383	-2156	-6605	-3698	1258	-203	-235	-162	41
Electrricity usage change	272	-27	-249	-24	-53	1456	120	-1539	-70	-59	-501	-103

Fig. 58. Change in oil and electricity usage due to EMs application.

New clothes washer is denoted as a, Clothes dryer as b, Dishwasher as c, Refrigerator as d, Lightening as e, Boiler as f, Water heater as g, Attic insulation as h, Basement insulation as i, Windows as j, Living space insulation as k and Air leakage (infiltration) reduction as l.

Here lowering the air leakage resulted in the highest decrease in the thermal load (about 21.5% reduction) while leading to a higher cooling (electric) load which is the dehumidifier load which implies moisture has increased. Replacing the existing oil-based water heater with electric one led to the 2nd highest decrease in the thermal load (16.3% reduction) while leading to the highest increase in electricity consumption (13.6% increase). The switch from incandescent lighting to LEDs led to a 1539 kWh decrease in the electric load (14.3% reduction) while increasing the thermal load by 3%. This is possibly due to the fact that incandescent bulbs emit 90% of their energy as heat and so are a contributor, although an inefficient one, to space heating [126]. Boiler upgrade led to a considerable (9.13%) reduction in the thermal load. It should be noted that full house wall insulation upgrade (basement + living space) resulted in 3296 kWh of reduction to the thermal load while SolarWall installation on the south facing wall alone resulted in 2706 kWh of heating losses recaptured. This could imply that for poorly insulated homes a SolarWall installation might be superior to insulation upgrade.

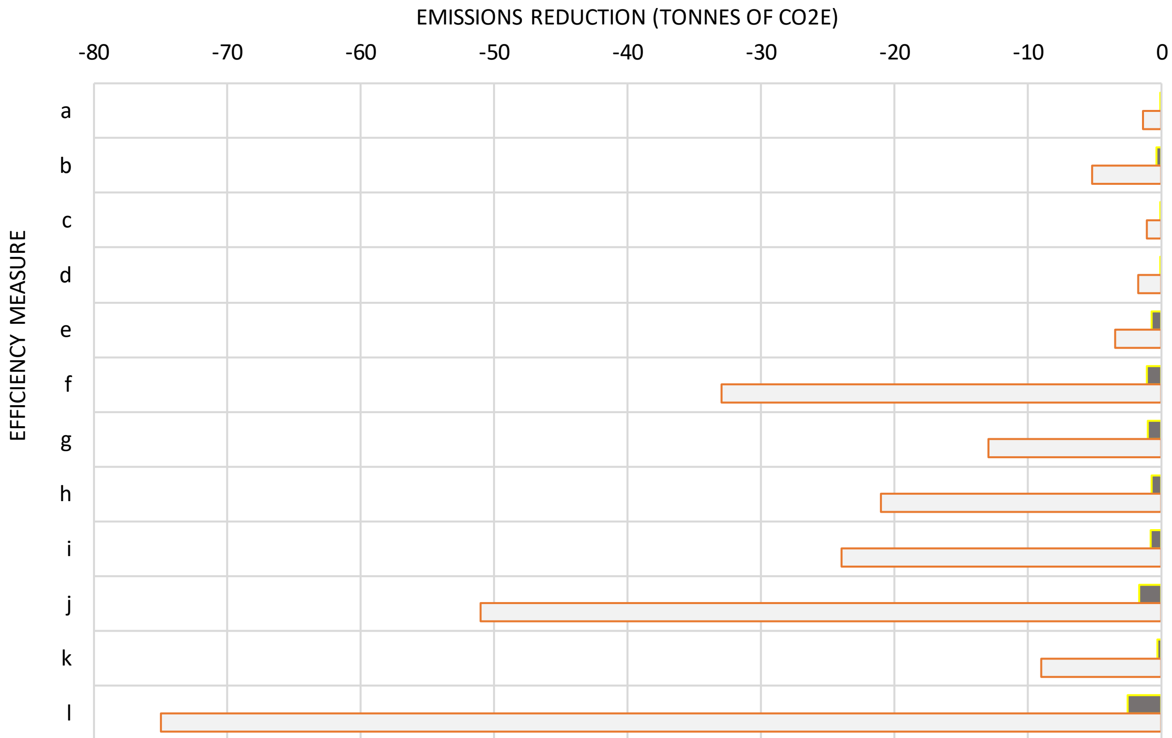


	l	k	j	i	h	g	f	e	d	c	b	a
Utility bill change	-569	-66	-361	-167	-154	-284	-241	-93	-22	-24	-70	-10
Utility bill change (Muskrat)	-539	-70	-390	-171	-161	-121	-228	-265	-30	-31	-127	-22

Fig. 59. Changes in utility bills as a result of EMs application.

New clothes washer is denoted as a, Clothes dryer as b, Dishwasher as c, Refrigerator as d, Lightening as e, Boiler as f, Water heater as g, Attic insulation as h, Basement insulation as i, Windows as j, Living space insulation as k and Air leakage (infiltration) reduction as l.

Figure 59 compares the change in utility bills under the current electricity rate (due to the EMs) to that projected to come into effect if the Muskrat Falls price stabilization plan does not succeed. For most measures the increase in electricity price leads to higher savings due to most measures resulting in lower electricity consumption. Air leakage reduction, water heater and boiler EMs are an exception to the pattern where post price escalation the savings drop somewhat and so profit and payback period also worsen. The measure that stands to offer the most benefit if electricity price increases is the change in lighting as it is the largest contributor to electric load reduction. Among all electrical appliances the clothes dryer offers the most pre and post price escalation savings. It offers over 500 kWh reduction in the electric load and over $70 in savings. Overall air leakage reduction continues to be the dominant source of savings and load reduction followed by window replacement which (from the previous figure) reduce the heating load by 11.9%. Water heater comes in 3rd in terms of pre escalation savings, but its merits drop most significantly post price escalation due to increased electric load.

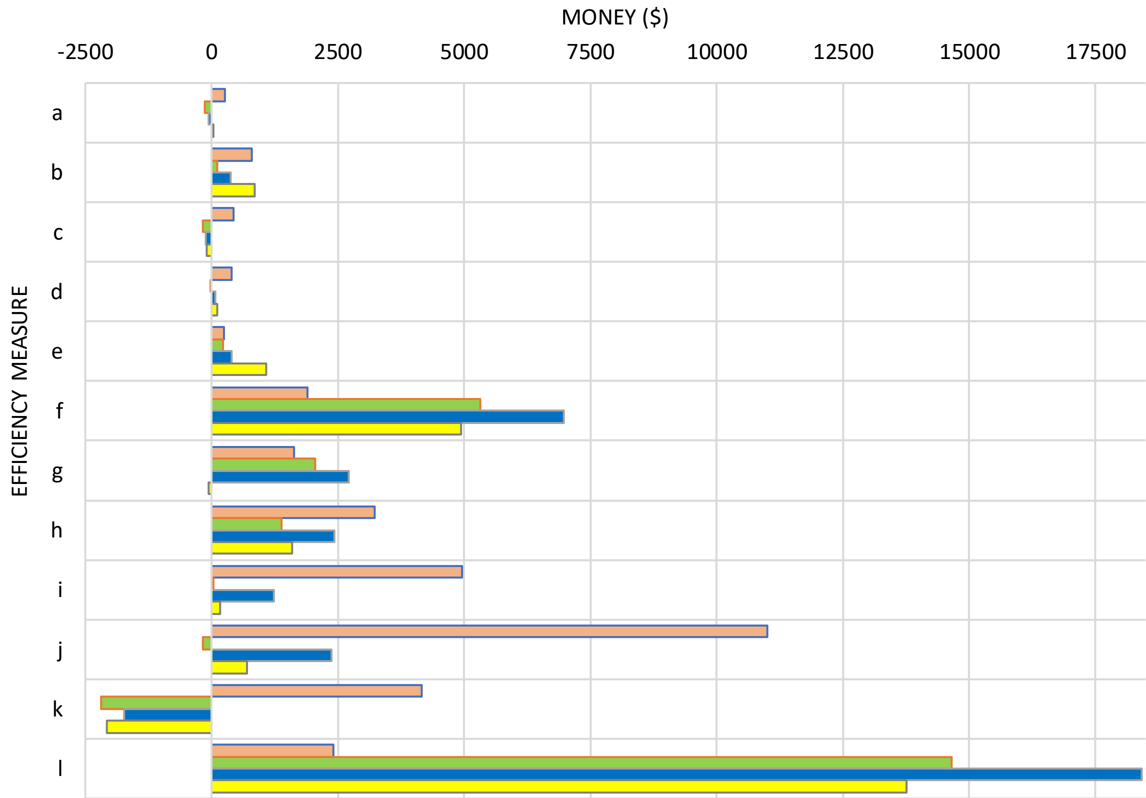


	l	k	j	i	h	g	f	e	d	c	b	a
Emissions per year	-2.5	-0.3	-1.7	-0.8	-0.7	-1	-1.1	-0.7	-0.1	-0.1	-0.4	-0.1
Emissions over lifetime	-75	-9	-51	-24	-21	-13	-33	-3.5	-1.74	-1.1	-5.2	-1.4

Fig. 60. Change in emissions as a result of EMs applications.

New clothes washer is denoted as a, Clothes dryer as b, Dishwasher as c, Refrigerator as d, Lightening as e, Boiler as f, Water heater as g, Attic insulation as h, Basement insulation as i, Windows as j, Living space insulation as k and Air leakage (infiltration) reduction as l.

Figure 60 shows that Air leakage reduction is the dominant contributor to emission reduction by reducing emissions by 13% per year and saving up to 75 metric tonnes of CO_2 over its 30-year lifetime. Followed by windows upgrade which reduces emissions by another 8.5%. Internalizing CO_2 reduction will improve the economics of all EMs. Clothes washer, dishwasher and refrigerator upgrade offered the least abatement of emissions. Clothes dryer was the electric appliance with the highest reduction in emissions. No EM resulted in higher emissions over pre-retrofit levels.

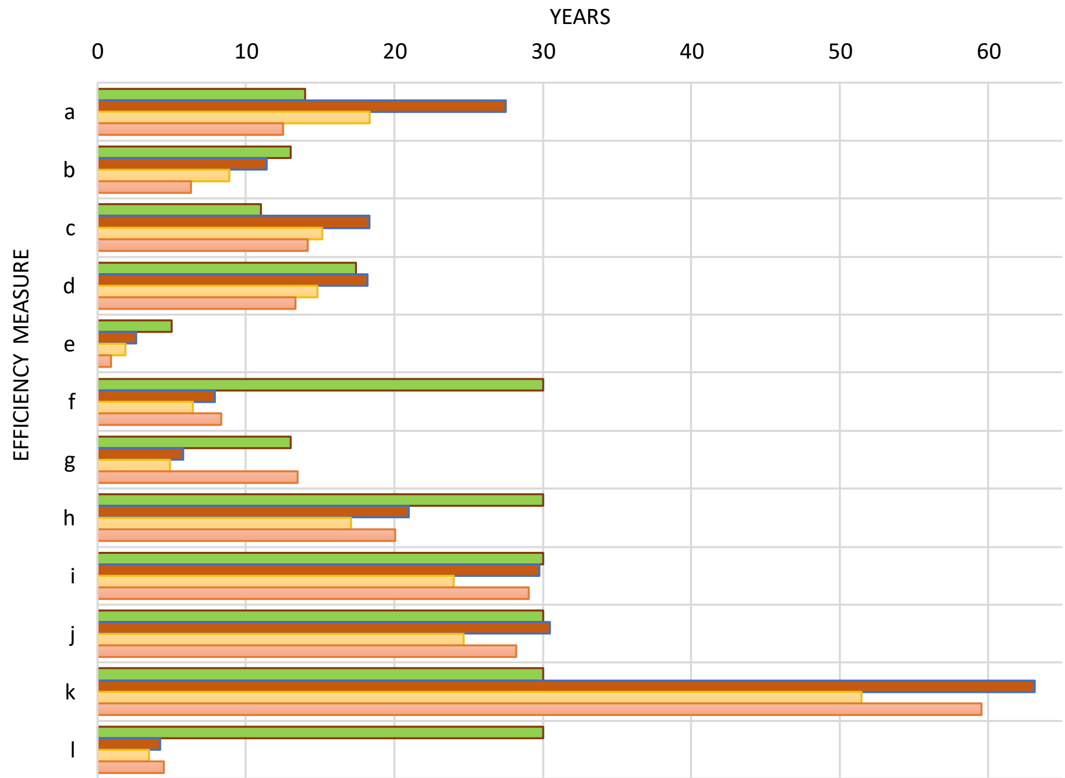


	l	k	j	i	h	g	f	e	d	c	b	a
Cost	2407	4169	11000	4968	3228	1631	1900	240	400	439	799	275
Profit	14,663.1	-2,188.8	-170.0	42.5	1,392.0	2,061.0	5,330.0	225.0	-17.2	-175.0	111.0	-135.0
Profit (Carbon)	18,413	-1,739	2,380	1,243	2,442	2,711	6,980	400	70	-120	371	-65
Profit (Muskrat)	13763.05	-2068.79	700	162.5	1602	-58	4940	1085	122	-98	852	33

Fig. 61. Cost incurred and profit generated as a consequence of EMs application.

New clothes washer is denoted as a, Clothes dryer as b, Dishwasher as c, Refrigerator as d, Lightening as e, Boiler as f, Water heater as g, Attic insulation as h, Basement insulation as i, Windows as j, Living space insulation as k and Air leakage (infiltration) reduction as l.

In figure 61, the cost, profit and profit with CO_2 included are compared for the different EMs. Air leakage reduction is still the largest contributor as it offers the highest pre and post inclusion profits but with a relatively smaller cost this means its payback period is amongst the lowest of all EMs. Living space insulation stands out as the EM to incur the highest negative (worst) pre and post inclusion profits. Meaning that the EM's lifetime (30 years) is shorter than its payback period and so it will never generate profit for the house. Window replacement is the most expensive of all EMs and also generates no profit however since it is the 2nd highest source of emission reduction, its post carbon inclusion profit is positive. A similar trend can be observed for basement insulation. The 2nd most profitable EM is the boiler which generates over $5,300 pre inclusion profit and nearly $7000 in post inclusion profit. Of all electric appliances, again, the clothes dryer is the most profitable. The price escalation due to Muskrat falls results in generally lower profits. The most notable example is that of the electric water heater which becomes unprofitable post escalation despite incurring over $2000 pre-escalation profit. The refrigerator and clothes washer become profitable post escalation despite being non profitable otherwise.



	l	k	j	i	h	g	f	e	d	c	b	a
Lifetime	30	30	30	30	30	13	30	5	17.4	11	13	14
Payback period	4.2	63.2	30.5	29.7	21.0	5.7	7.9	2.6	18.2	18.3	11.4	27.5
Payback period (Carbon)	3.5	51.5	24.7	24.0	17.1	4.9	6.4	1.9	14.8	15.1	8.9	18.3
Payback period (Muskrat)	4.5	59.6	28.2	29.0	20.0	13.5	8.3	0.9	13.3	14.2	6.3	12.5

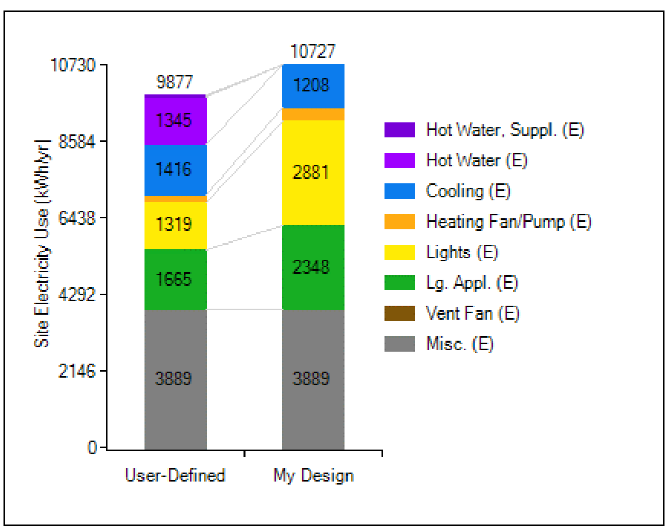
Fig. 62. Payback period of the different EMs.

New clothes washer is denoted as a, Clothes dryer as b, Dishwasher as c, Refrigerator as d, Lightening as e, Boiler as f, Water heater as g, Attic insulation as h, Basement insulation as i, Windows as j, Living space insulation as k and Air leakage (infiltration) reduction as l.

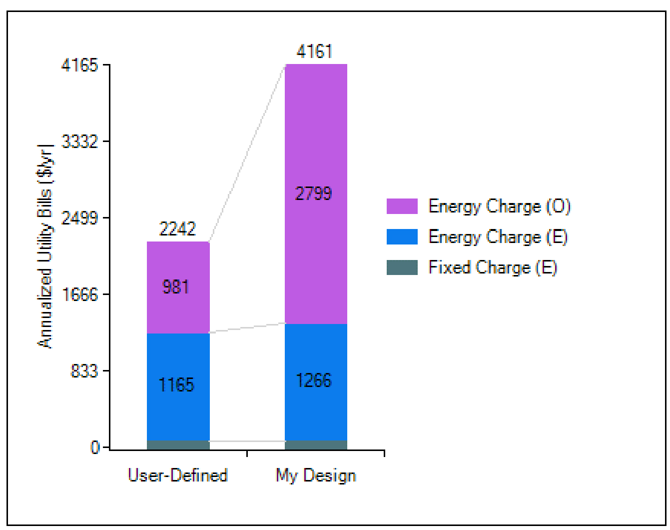
Here the payback period is shown (figure 62) where the smaller the value the better. The payback period for the 0.23 CAD/kWh scenario and the one where the price of carbon is included are also shown. Here EM k (living space insulation) is the most unfeasible where its payback period is more than twice its lifetime. Windows and basement insulation have payback periods of approximately 30 years for a 30-year design lifetime making them profitably neutral. Their economics however improve in the post price escalation scenario and the carbon price inclusion scenarios to marginal profitability. The two EMs with the shortest payback periods are lightening and air leakage reduction. Since the water heater increases electricity consumption its payback period more than doubles with the electricity rate escalation. Clothes dryer is once more the most economically viable electric appliance. It uses heat pump technology to reduce its energy consumption while maintaining the same drum size as the original appliance but at a higher combined energy factor. The dishwasher performed well enough considering that it has 1.75 times the number of place settings as the old dishwasher but at a lower rated electricity and hot water consumption. Oriented Strand Board (OSB) R-15 XPS insulation was also tested for living space insulation but it resulted in worse results than R-15 XPS.

8) Final results

Based on the previous simulations clothes washer and dishwasher were unprofitable upgrades however the cost they incurred is relatively small and the devices do offer increased functionality over the existing appliances and therefore will not be eliminated. Living space insulation on the other hand performed worst on the previous metrics and so will be removed from consideration. The final system results are shown in the figure 63. The results are an 850 kWh (7.9%) net decrease in electricity consumption (due to lighting and large appliances) despite the switch to an electric water heater causing a 1345 kWh increase and the dehumidifier consuming 208 kWh more electricity. Oil consumption decreased by nearly 650 gallons which corresponds to a 26.4 MWh decrease which is 52% lower fuel consumption/heating load. This resulted in the house saving $1919 on energy per year and 8.7 tonnes of CO_2 being abated per year. The cost of the retrofit is $27,286, by taking the project lifetime as the average lifetime of all included EMs (20 years) this results in 14.2 year payback period (or 11.6 years if carbon price is included) which corresponds to a profit of over $11,000 which increases to $19,700 once the price of the 174 metric tons of emissions abated is included. The NPV of the system is $55,229 assuming an inflation rate of 2.4% and a discount rate of 3% (BEOPT's default). The ROI is 40.3% at an annual ROI of 1.7%. If the price of electricity escalates due to Muskrat falls, the annual energy savings will increase by an additional $336/year (the overall bill is higher though). This results in the payback period being 2.1 years shorter which leads to a 61% increase in profit. Although in this case the switch to an electric water heater might be less prudent. So, Muskrat Falls electricity rate escalation can be seen as an opportunity/incentive for dated homes in the province to undergo energy efficiency retrofits.



a



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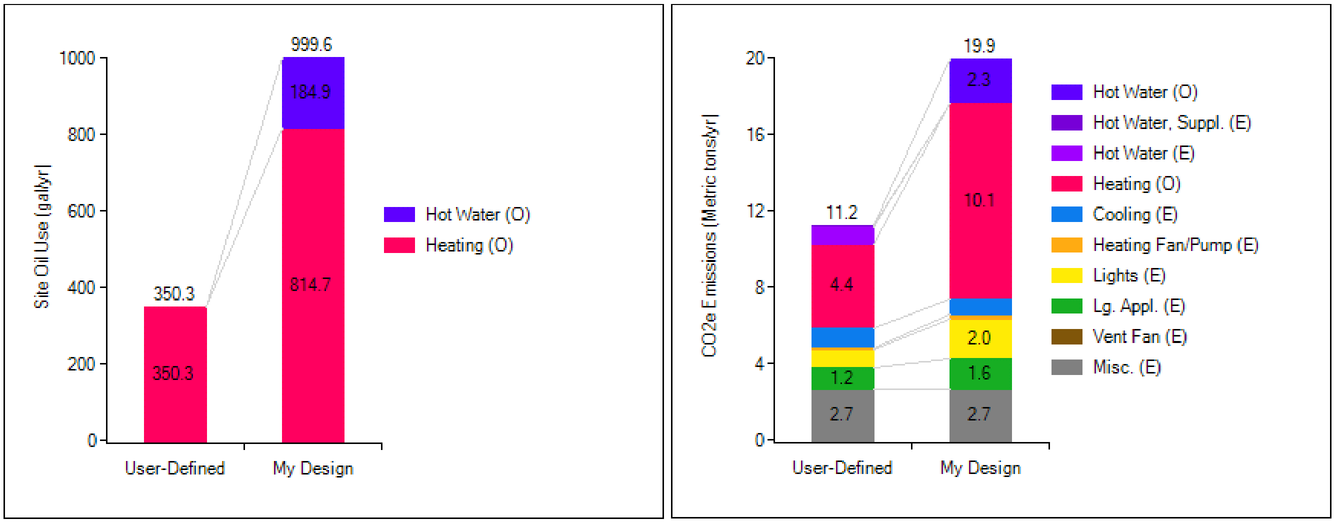


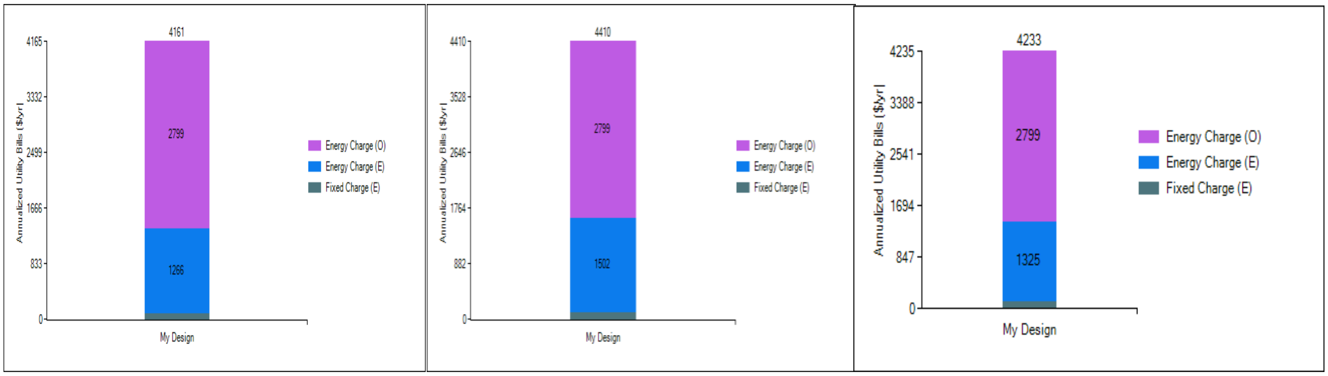
c d

Fig 63. Final system results: a) Electricity use, b) Utility bills, c) Oil use, d) CO2e emissions

9) TOU and Tiered pricing

As can be seen from figure 64, by applying Ontario's TOU pricing regime to the design house, electricity charges increase by $236 while applying tiered pricing the bill increases by only $59. This could imply that a lot of the house's loads are by default scheduled for On-peak times (a comparison between DR schedule and normal schedule can be seen in the appendix) or that the house mostly consumes less than 1000 kWh/month which is likely since the house's annual load is 10,727 kWh/year. Since Ontario residents are offered a choice to opt out of TOU pricing, BEOPT or past electricity bills can be used to determine the savings to be incurred via switching policies.

It was initially intended to test every smart device similar to the manner in which the energy efficiency devices were tested to see the effects of demand response under the presence of TOU pricing. However, it was not possible to create DR schedules for all devices except for the clothes dryer and refrigerator. So, the last two figures in appendix D compare the smart clothes dryer with the old dryer and the energy efficient dryer (without DR). It can be seen that the smart dryer causes 578 kWh of energy savings which is 5.4% of the load versus the old dryer which corresponds to $94 in annual bill savings. The cost of the device is however twice that of the energy efficient dryer which makes its payback period 4 years longer than its lifetime and therefore it is not economically viable even after including the price of CO_2 abated. The energy efficient washer on the other hand consumes 77 kWh more electricity which corresponds to $9 higher energy bills but it only costs $800 making its payback period 9.4 years for 13 years design lifetime which corresponds to $565 in profit over the projects lifetime. In conclusion, given the methods adopted in this work it might be more advisable to buy a cheaper energy efficient appliance than to invest in one with DR functionality. The inconvenience of load shifting should also be considered.



a b c

Fig 64. Utility bills for a) base case, b) TOU pricing, c) Tiered pricing.

VI. Conclusion

This project's motivation was to introduce a solar thermal system to replace the existing heating oil system. From project A the thermal load serving scenario (scenario C) was found unfeasible. However heating oil is very polluting and therefore unsuitable as one of Newfoundland's energy sources especially since the province has committed to reduced emissions. Information related to solar thermal energy technology's classifications, components, distribution, storage, protection, innovations, sizing and economics was extracted from a variety of sources and presented.

Energy efficiency and output of a solar collector was calculated in excel using Hottel-Whiller-Bliss and ten other equations. The calculation utilized hourly inputs such as inlet fluid temperature, ambient temperature, collector characteristics, irradiance at the given location and the design house's space heating and hot water thermal loads obtained from BEOPT. The inlet fluid temperature was varied from an assumed temperature of 15 °C to temperatures of the water main and then to the house's ambient temperature depending on whether the space heating load or DHW load is dominant. This resulted in a lower boundary (space heating is dominant) which was lower than the results of the assumed case and an upper boundary (water heating is dominant) which surpassed the assumed case. Excess energy, unmet load, oil needed, oil gallons offset and payback periods were calculated for all three cases for a number of collectors ranging from 0 to 36. Since the percentage of unmet load and excess energy were high, a sensible heat energy storage was sized and included which resulted in the solar fraction reaching as high as 50% for a 75 L/m^2 tank size. The f-chart was then calculated for the system after changing the size of the implicit storage to include the existing storage tank. The result showed that for small tank volumes the f-chart was squarely in the middle of the upper and lower boundaries but as the tank size increased (due to an increase in collector size) the f-chart converged on the lower boundary. This could be due to the effect of water consumption being less influential on the lowest node of the stratified tank as the size of tank increases. Different collectors were then compared and the results show that an evacuated tube collector can produce higher solar fractions but takes up a much larger area.

The combi system was then simulated in polysun, this resulted in a solar fraction of 50.5% if the tank is placed outside which is very close to the fraction obtained earlier. The cost of the collector array, tank, pumps, piping, valves, controllers and installation were then obtained and used to calculate the economics of the project such as payback period, total savings and LCOE. A comparison between Scenario E and scenario C showed the combi system to be both feasible and economically superior. Installing the system also led to per capita emissions of 0.82 tonnes/year for the house's inhabitants which is less than half the national average.

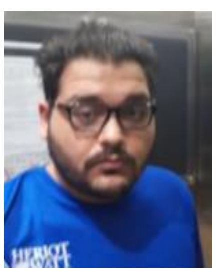
A PVT system was then designed using the existing storage tank. The system was able to provide 106% of the electric load and 12% of the thermal load. A comparison of the system with the combi system and PV system (while maintaining the same area/power) showed that the PVT system's generation is 36% more valuable than the PV system and 44% more valuable than the combi system while taking up the least amount of space. The economics of the PVT system was however worse with a much smaller return on investment when compared with the combi system.

RETScreen was then used to design scenario G which consisted of a solar water heater and storage, a SolarWall air heater and a glazed solar air collector. Inputs to the software included location, climate zone, elevation, facility type and size, electricity consumption, oil consumption, fuel used, fuel and electricity price, set point temperatures, boiler and water heater efficiencies, hot water usage and number of occupants. The solar water heater used AE-21 collector and the solar air heater used Cansolair RA 240 Solar max. The solar water heater generated 3481 kWh of thermal energy at a solar fraction of 48.5% using a 535 L tank. The SolarWall had an area of 24.7 m^2 (windowless portion of the south facing wall) and resulted in 2904 kWh of thermal energy generation and an additional 2706 kWh of the building heat recaptured, the sum of both correspond to 33% of the half load. This was later confirmed as more economic than upgrading the wall insulation from R-10 to R-20. The solar air heater generated 3717 kWh at a seasonal efficiency of 12.7% and a solar fraction of 22.1% which is lower than the total effect of the SolarWall. The payback period and profit were also lowest for the solar air heater whereas the SolarWall generated the most profit and emissions reduction.

The last module of this work used energy efficiency measures to retrofit the design house and reduce its load/bills. Energy efficiency measures were divided into electrical appliances (clothes washer, clothes dryer, fridge and dishwasher), heat generators (boiler and water heater), lighting and building elements (insulation, windows and air leakage reduction). EnerGuide labels and energy star most efficient designation were used to guide component selection. Each energy measure was simulated one at a time and the results used to calculate performance metrics including energy usage change, bill change, emission change, payback period and profit. The metrics were also repeated for the case were emissions were monetized and the case of electricity rate escalation due to Muskrat Falls. The results showed a 21.5% reduction in thermal load and 13% reduction in emissions due to air leakage reduction. Followed by heating load reductions due to the switch to an electric water heater (13.6%) and the upgrade of the boiler (9.13%). The switch to LED lighting produced a 14.3% reduction in the electric load. Windows reduced emissions by 8.5% per year. Air leakage reduction was by far the most profitable under all price scenarios followed by the boiler upgrade. While the dishwasher and living space insulation upgrades were unprofitable. By omitting living space insulation upgrade, the final system results were calculated which showed a 7.9% reduction in the electric load and an outstanding 52% reduction in thermal load corresponding to $1919 in annual savings which increase if electricity price escalates. The project had an ROI of 40.3% and generated $19,700 if the price of the abated emissions is included. The study concludes by applying Ontario TOU and tiered pricing to the system and arriving at the conclusion that tiered pricing is more affordable than TOU and that DR compatible smart devices are not as economical as energy efficient devices without DR due to the higher price point.

Future work

While the work conducted in projects A and B has been both thorough and essential, it can always be developed further. Future expansion of this work can include generalization to a broader level of analysis such as entire neighbourhoods or cities. More advanced forms of energy storage such thermochemical energy storage can be studied. A stratified tank model can be modeled in Mathcad to observe the dynamics of water stratification for the current system. The impact of snow buildup can be studied, and new removal methods devised. A full smart home can be modelled at various levels and its renewable sources designed. An electric vehicle charging station that incorporates renewables can be designed. Peer to peer energy trading or more generally the dynamics of a neighborhood of smart homes with distributed generation can be modelled. Larger buildings in St. John's can be studied and have sustainable systems and efficiency measures tailored to them.



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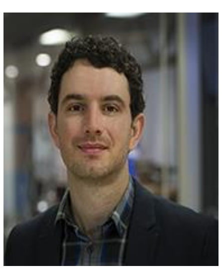

Dr. Kevin Pope has earned a BEng from the University of Ontario Institute of Technology in Energy Systems Engineering (2008). His MASc thesis (2009), in Mechanical Engineering, involved the transient behaviour of wind turbines. His PhD thesis (2012), in Mechanical Engineering, focused on physiochemical characterization, multiphase flow and heat transfer for thermochemical energy conversion and storage. Before joining Memorial University, He investigated sustainable energy systems for large urban areas by examining the effects of different socioeconomic development pathways on resource demands and infrastructure needs.

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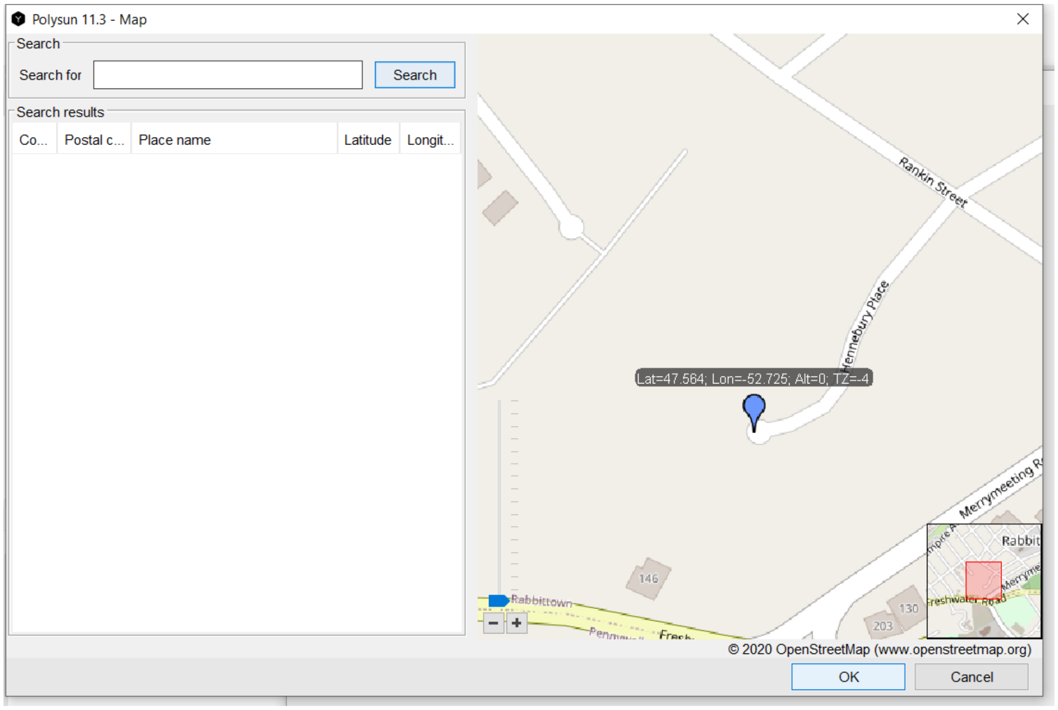
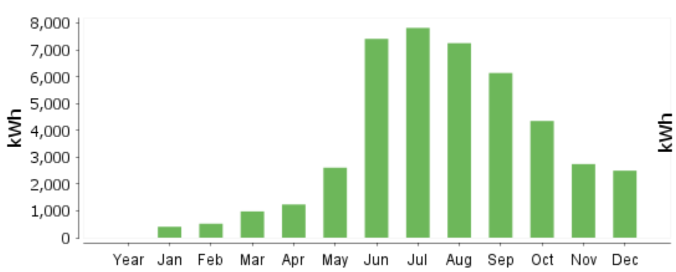
Appendix

A. Appendix A- Polysun

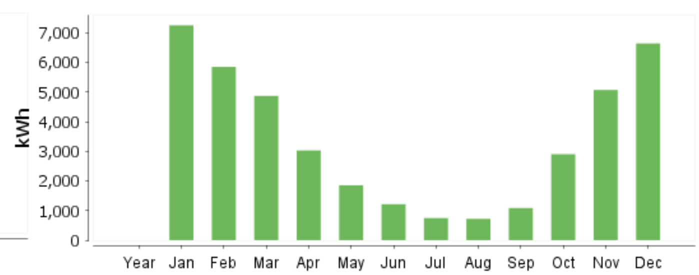


Location

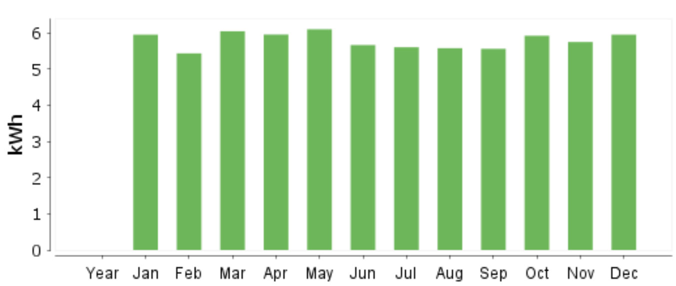
1) *A-1 Combi-system results*



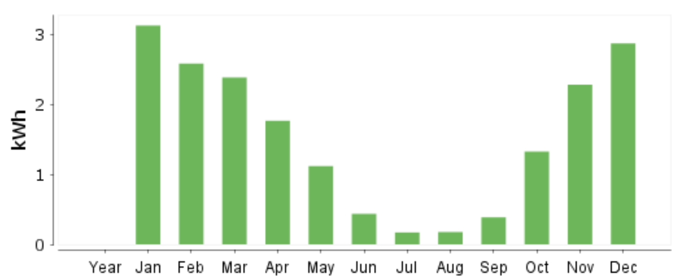
Solar thermal energy to the system



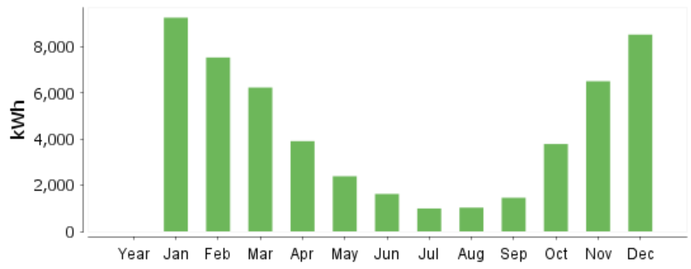
Heat generator energy to the system



Energy Deficit

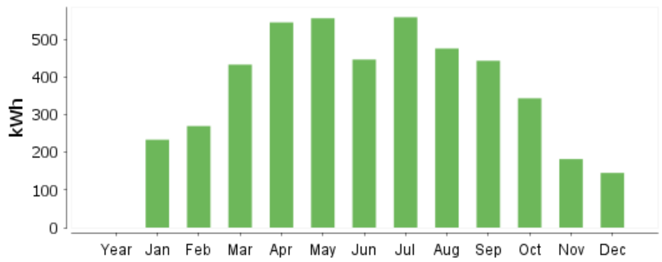


Electricity consumption

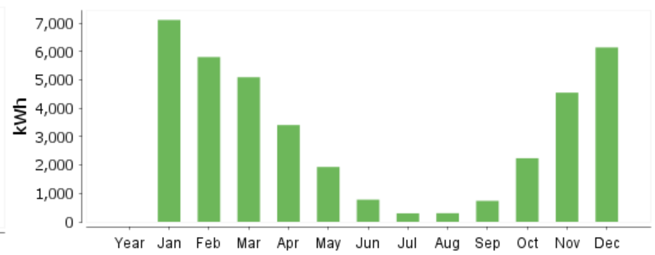


Oil consumption

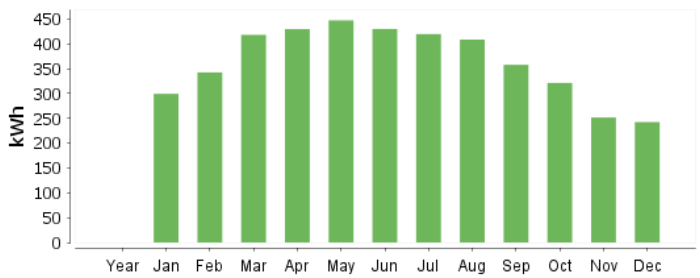
2) A-2 PVT system Results



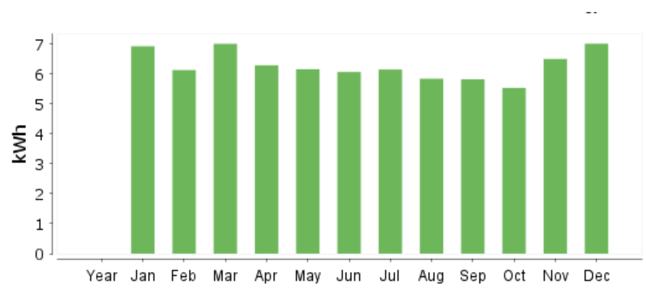
Solar thermal energy to the system



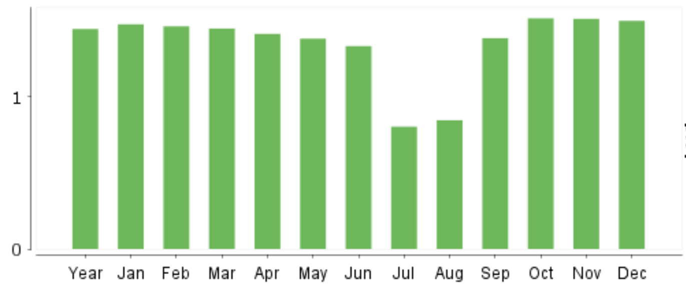
Heat generator energy to the system



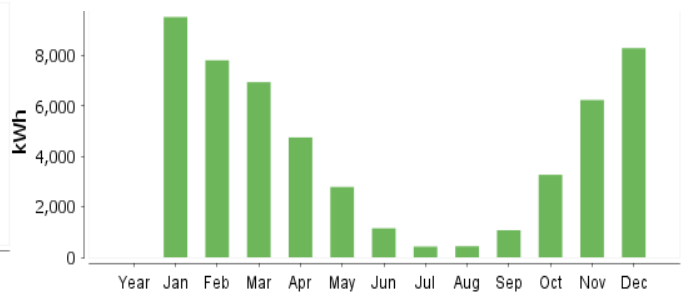
Electricity consumption



Energy Deficit

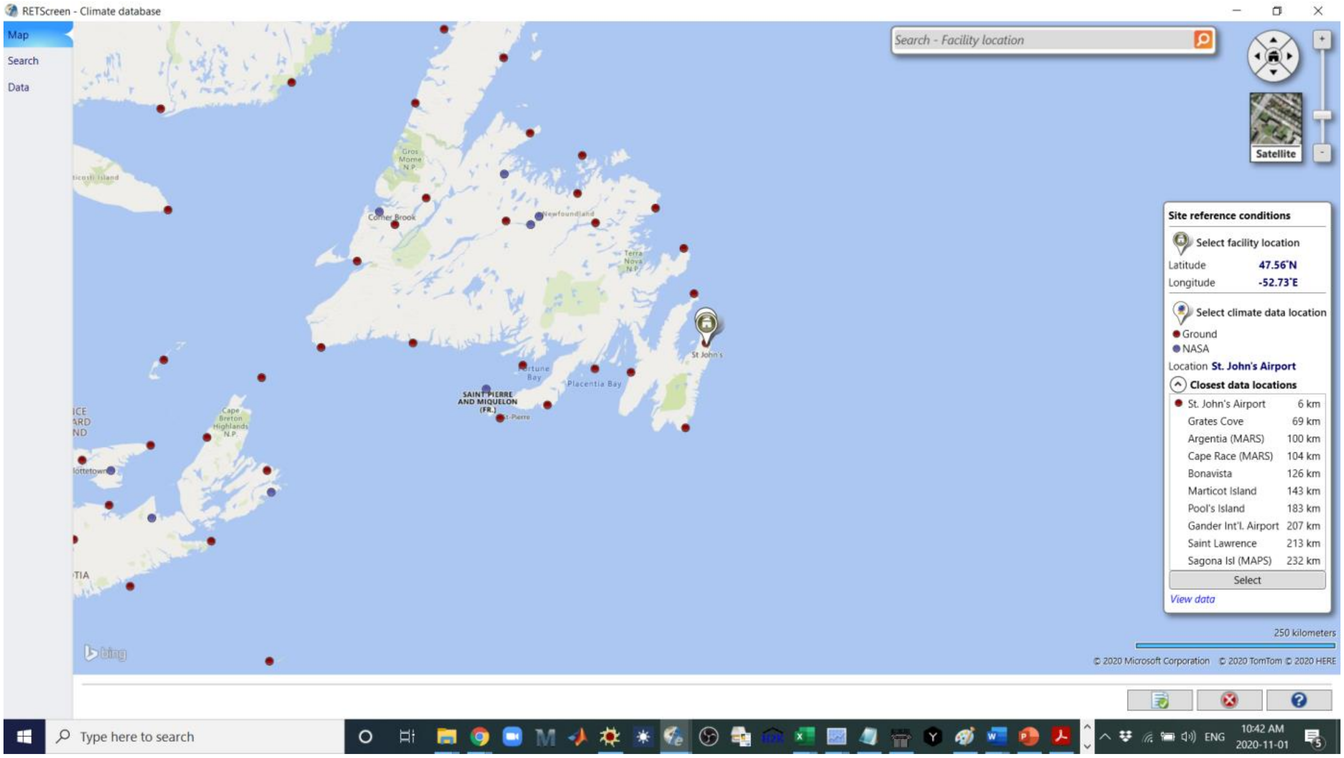


Primary Energy Factor

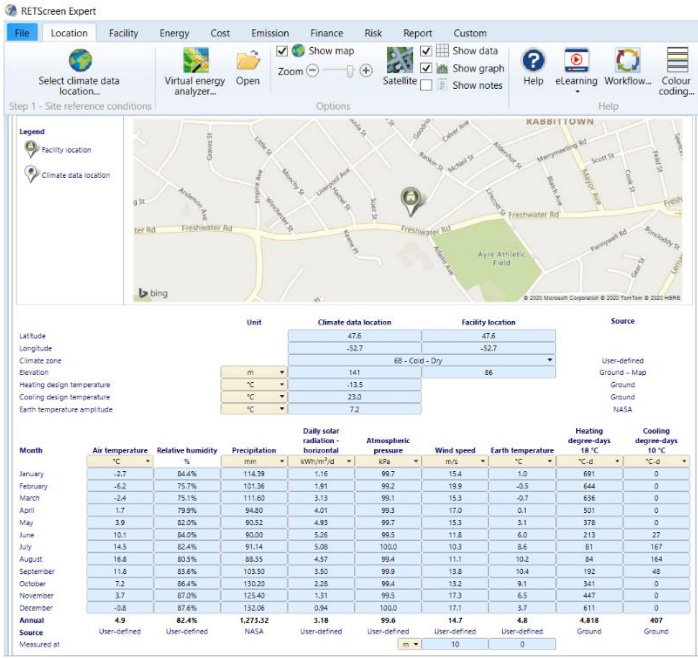


Oil consumption

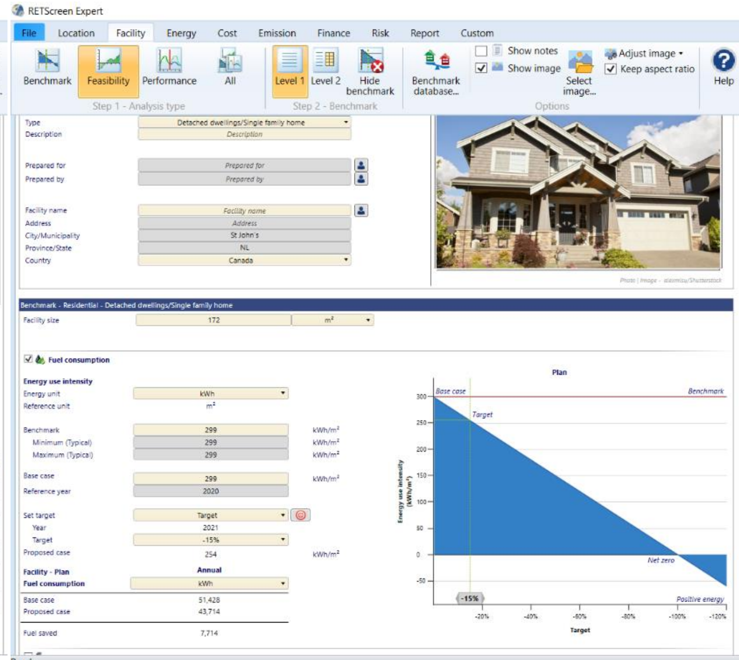
B. Appendix B- RETScreen



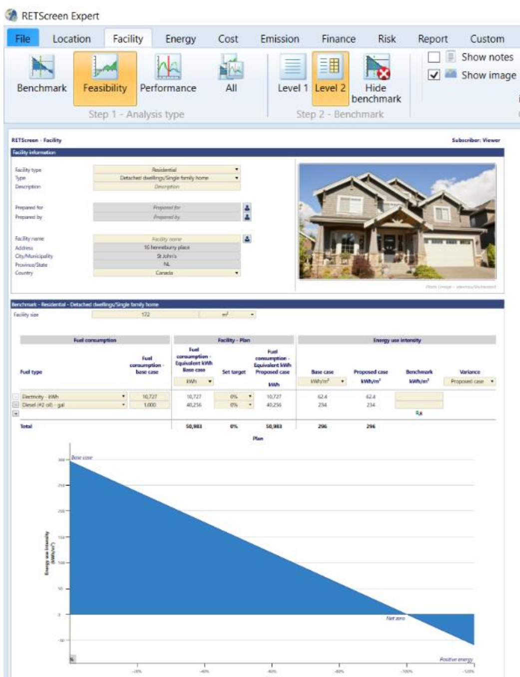
Location selection



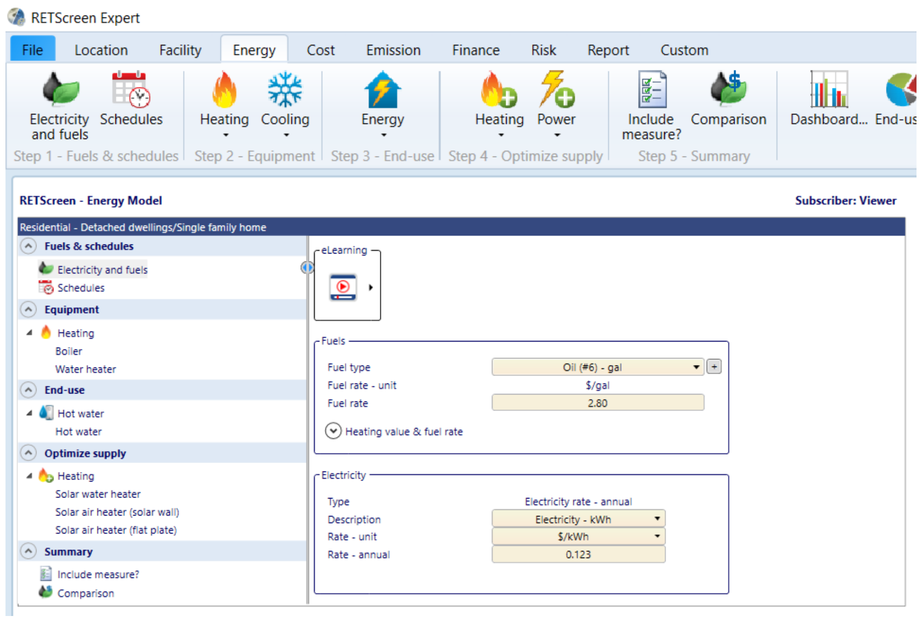
Location window values



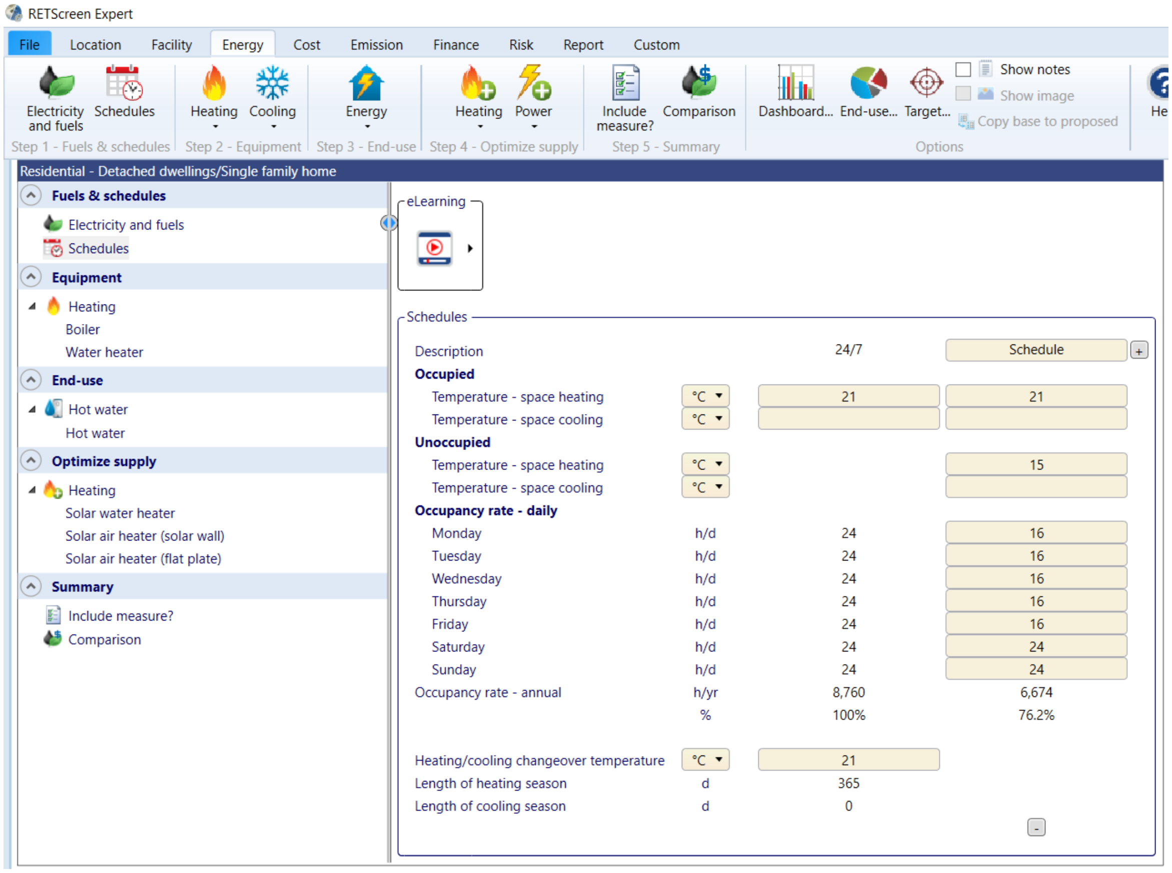
Facility level 1



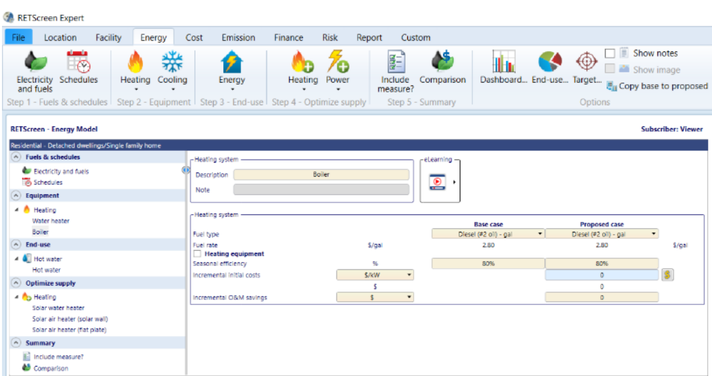
Facility level 2



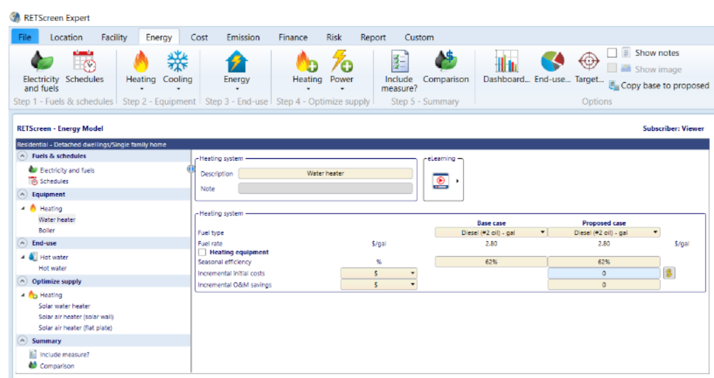
Electricity and fuels



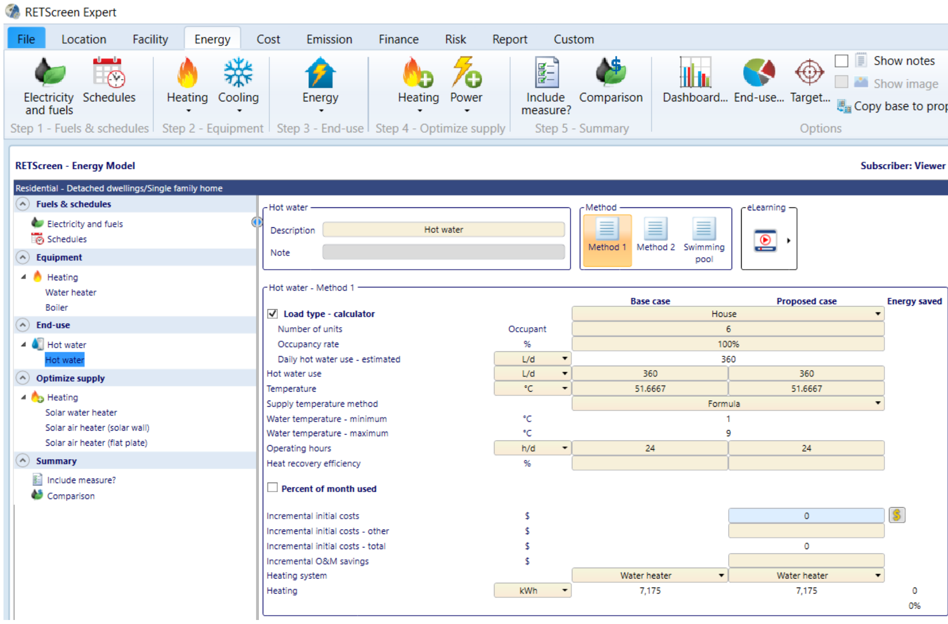
Schedules

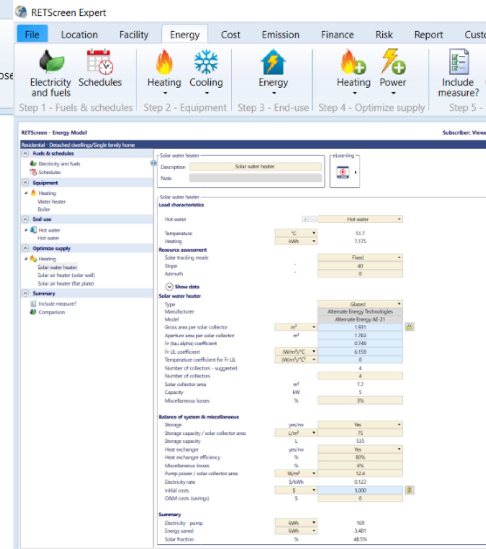


Boiler



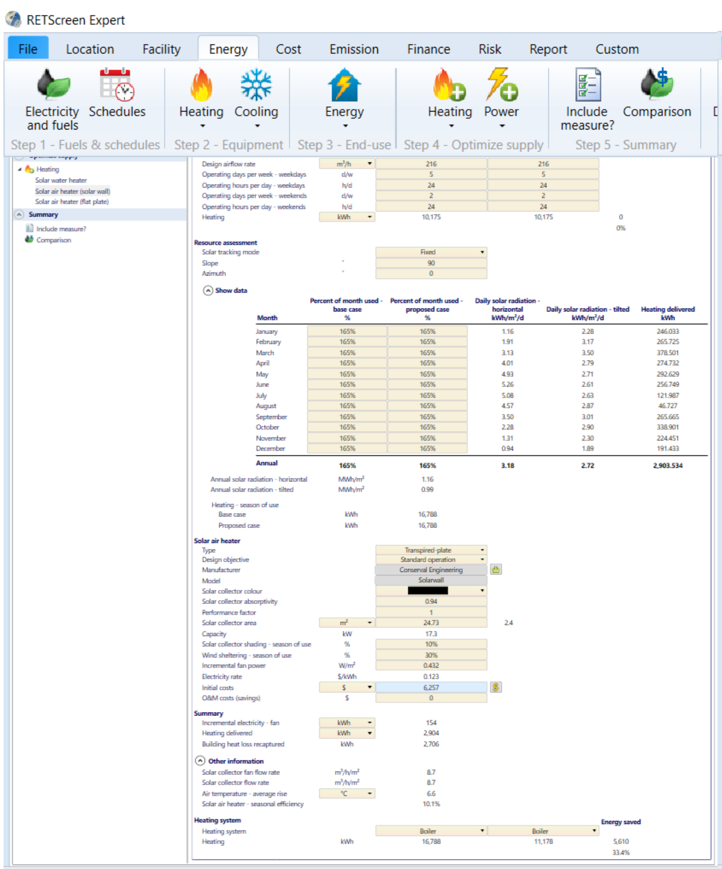
Water heater

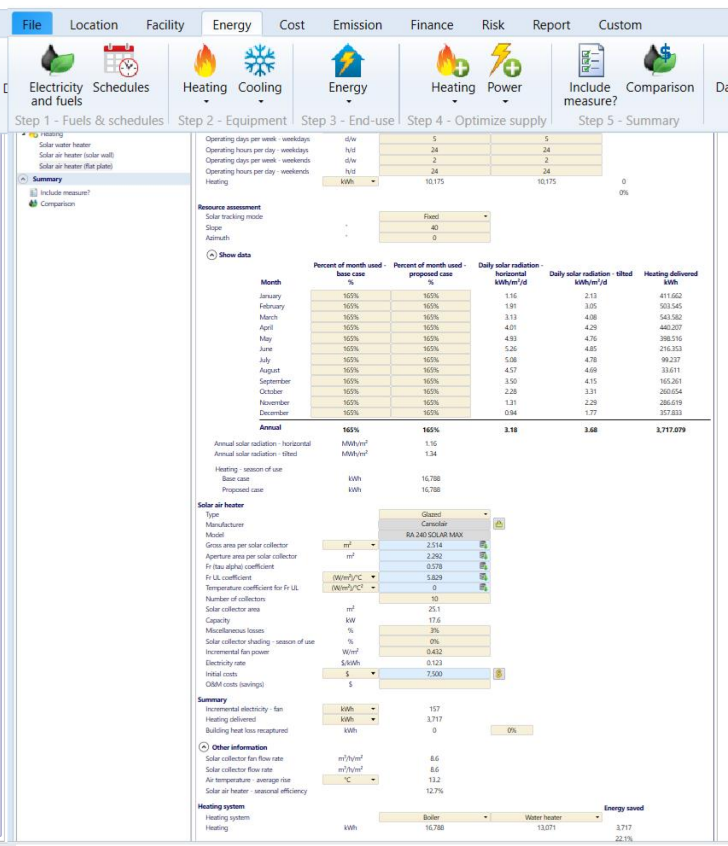




Hot water

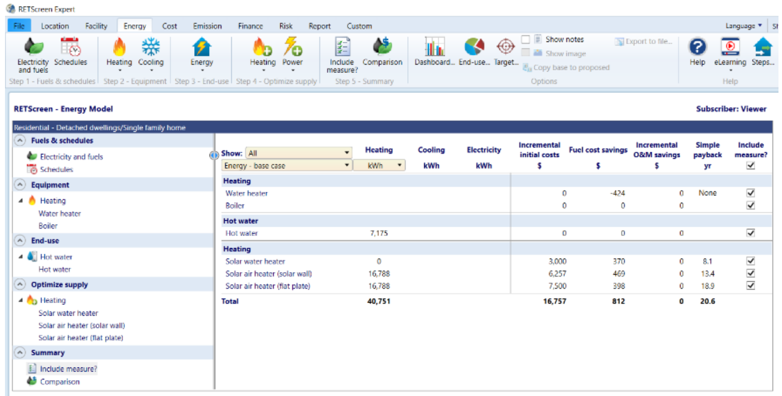
Solar water heater



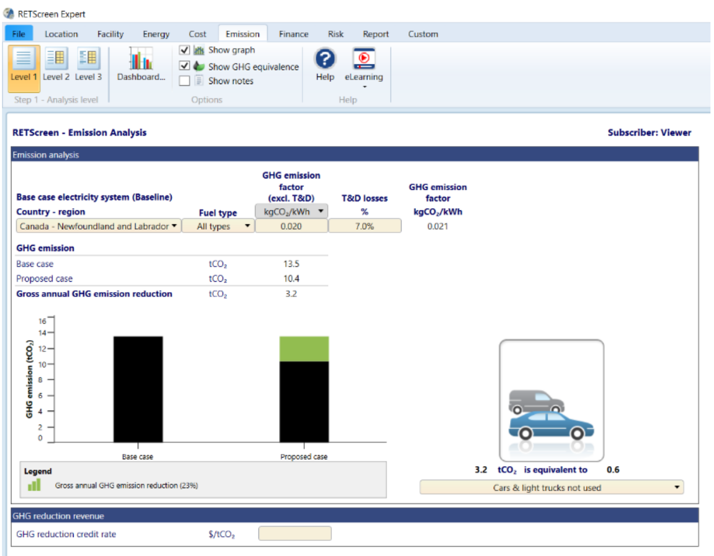


SolarWall

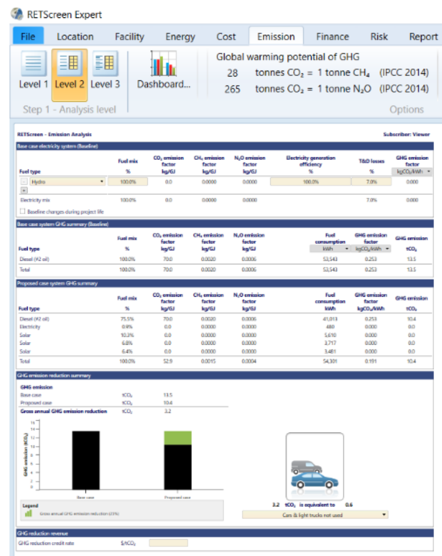
Solar Air heater



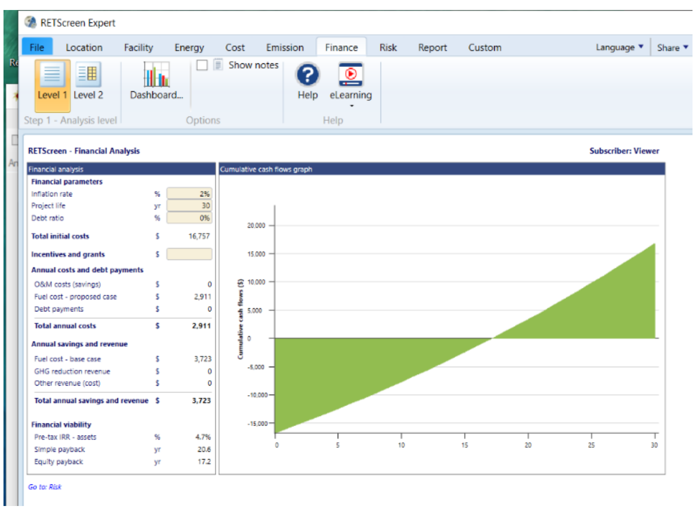
Overall results



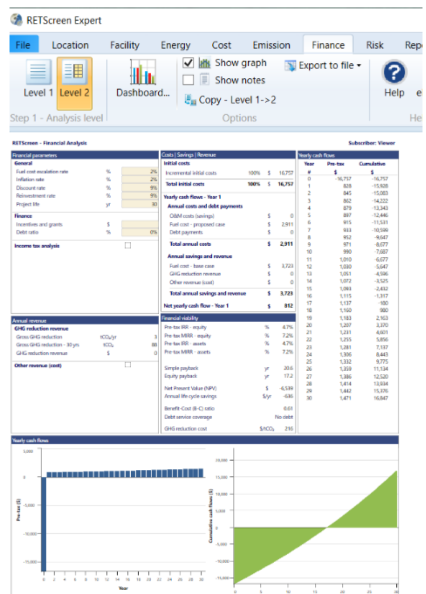
Emissions results 1



Emissions results 2



Financial results 1



Financial results 2

C. Appendix C- LCOE calculator

Simple Levelized Cost of Energy Calculator

Financial

Periods (Years): 30 ?

Discount Rate (%): 6 ?

Renewable Energy System Cost and Performance

Capital Cost ($/kW): 648 ?

Capacity Factor (%): 4.4 ?

Fixed O&M Cost ($/kW-yr): 0 ?

Variable O&M Cost ($/kWh): 0 ?

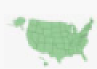
Heat Rate (Btu/kWh): 0 ?

Fuel Cost ($/MMBtu): 0 ?

Today's Utility Electricity Cost

Electricity Price (cents/kWh): 6.9 ?

Cost Escalation Rate (%): 3.0 ?



Results

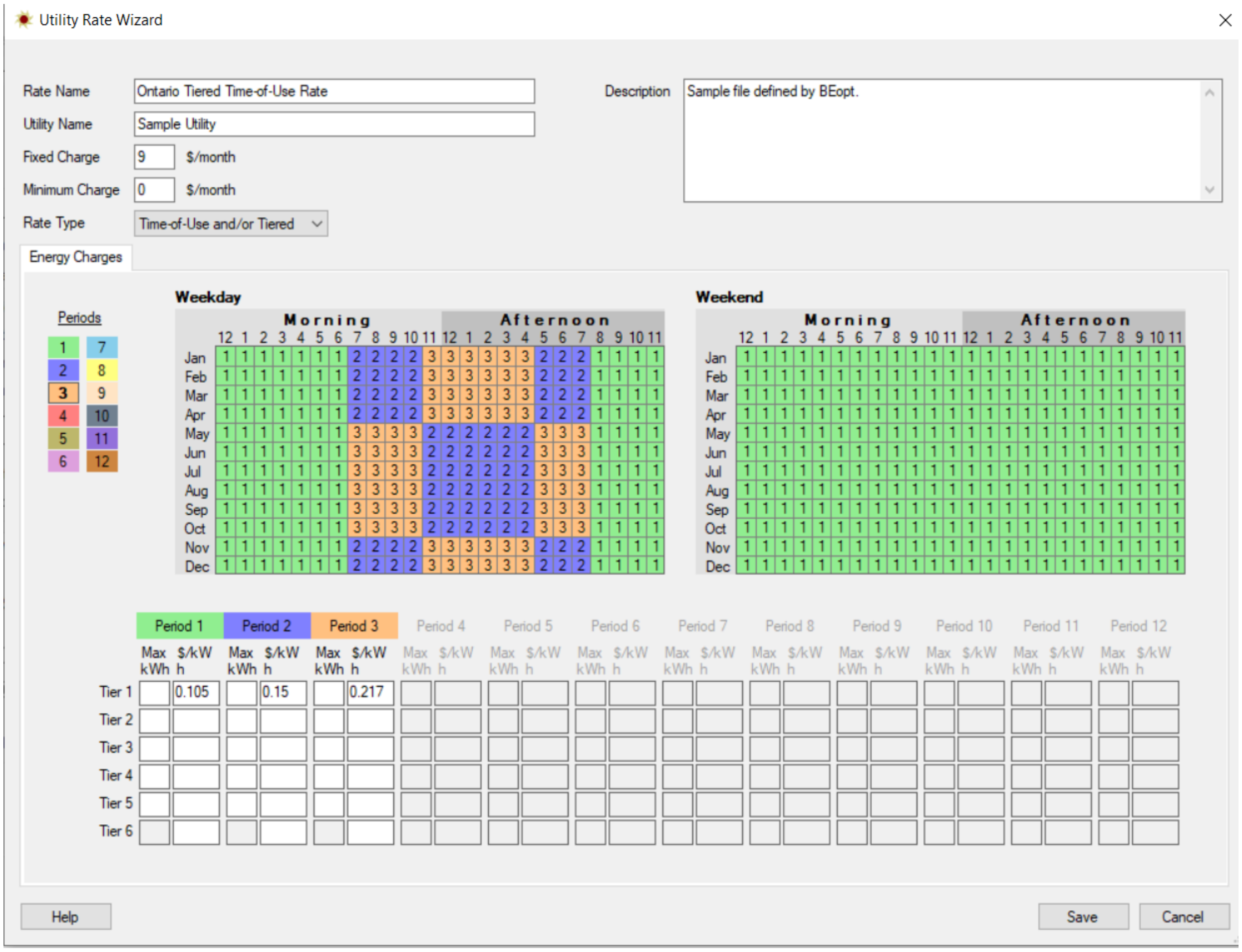
Levelized Cost of Utility Electricity (cents/kWh): 10.0 ?

Simple Levelized Cost of Renewable Energy (cents/kWh): 12.3 ?

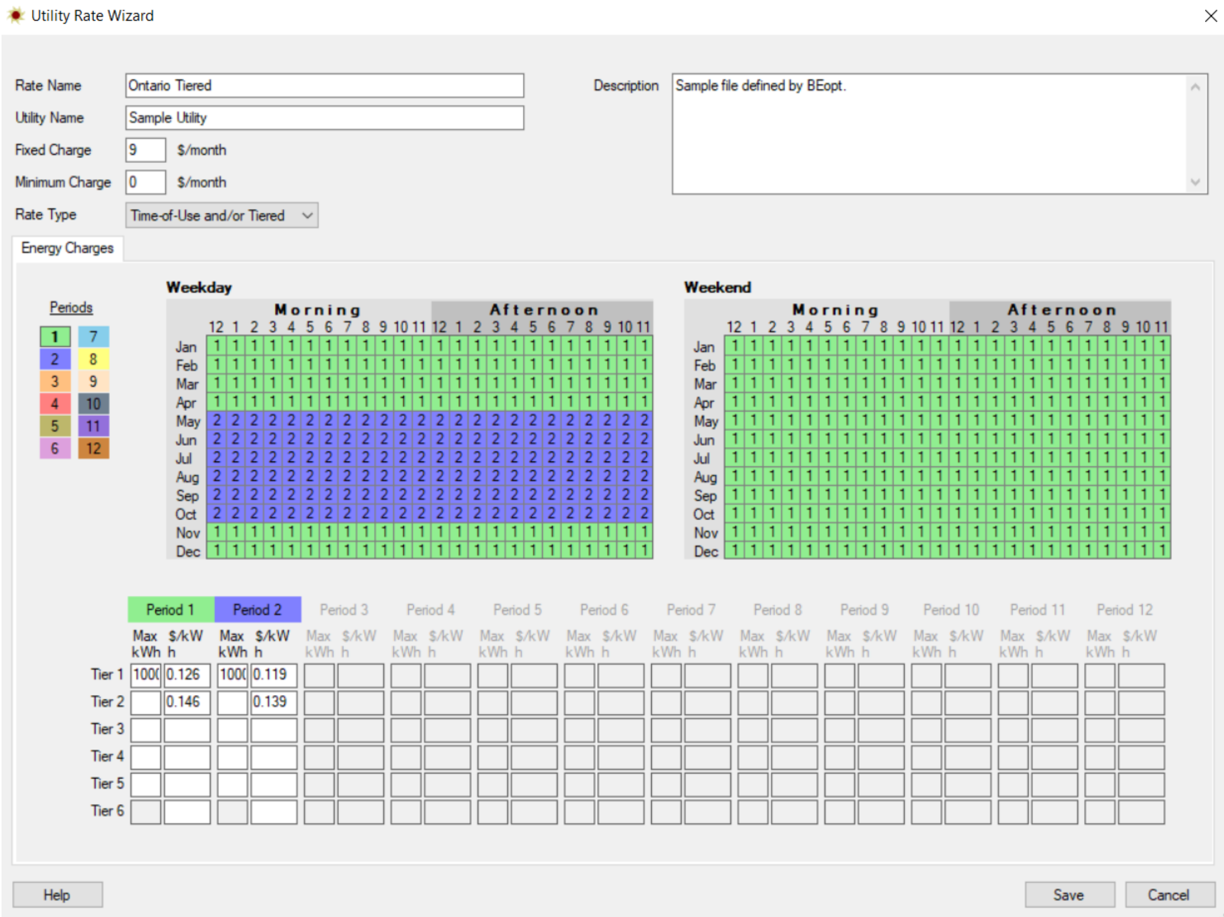
How are these numbers calculated? See documentation

D. **Appendix D- BEOPT Results**

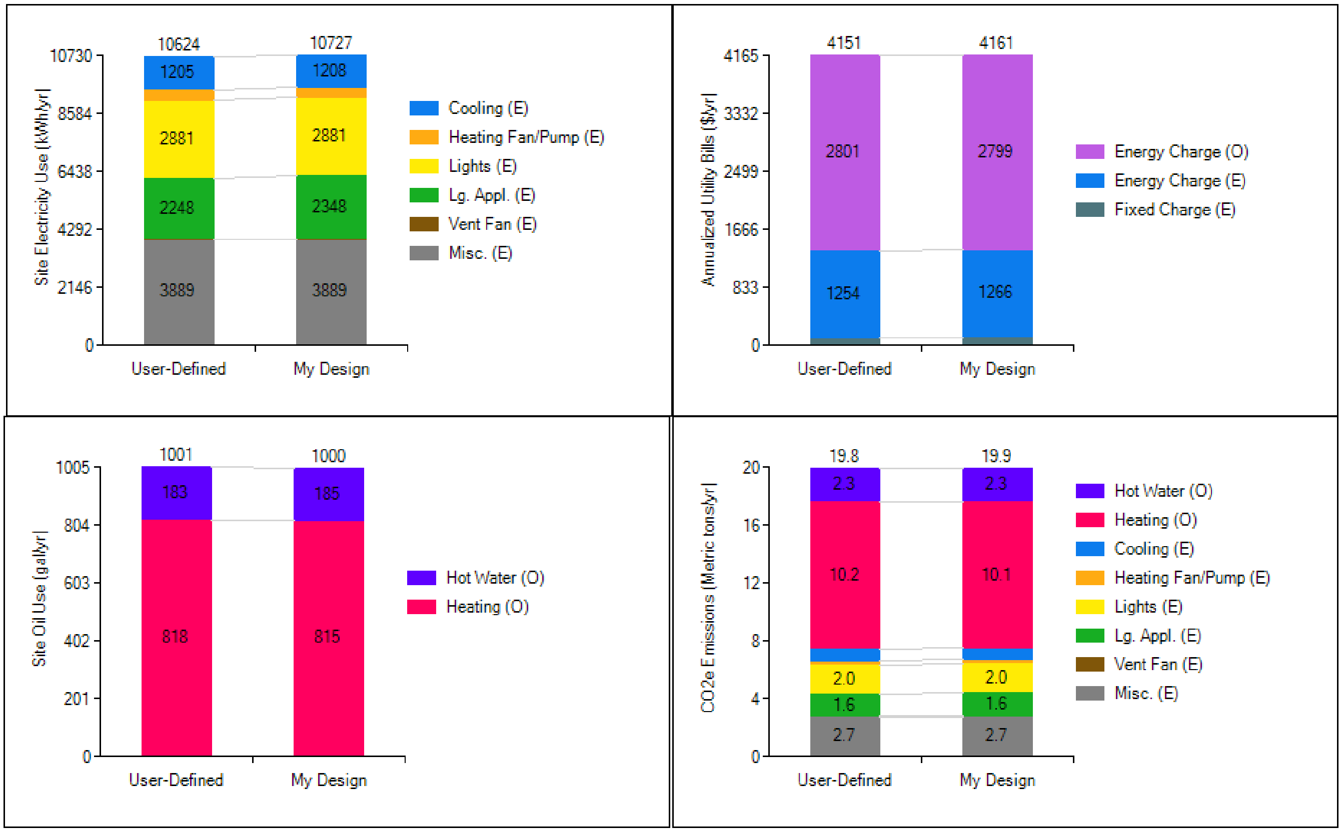
New washer, Smart washer, New dryer, Smart dryer, New fridge, Smart fridge, New dishwasher, Smart dishwasher, New boiler, New water heater.



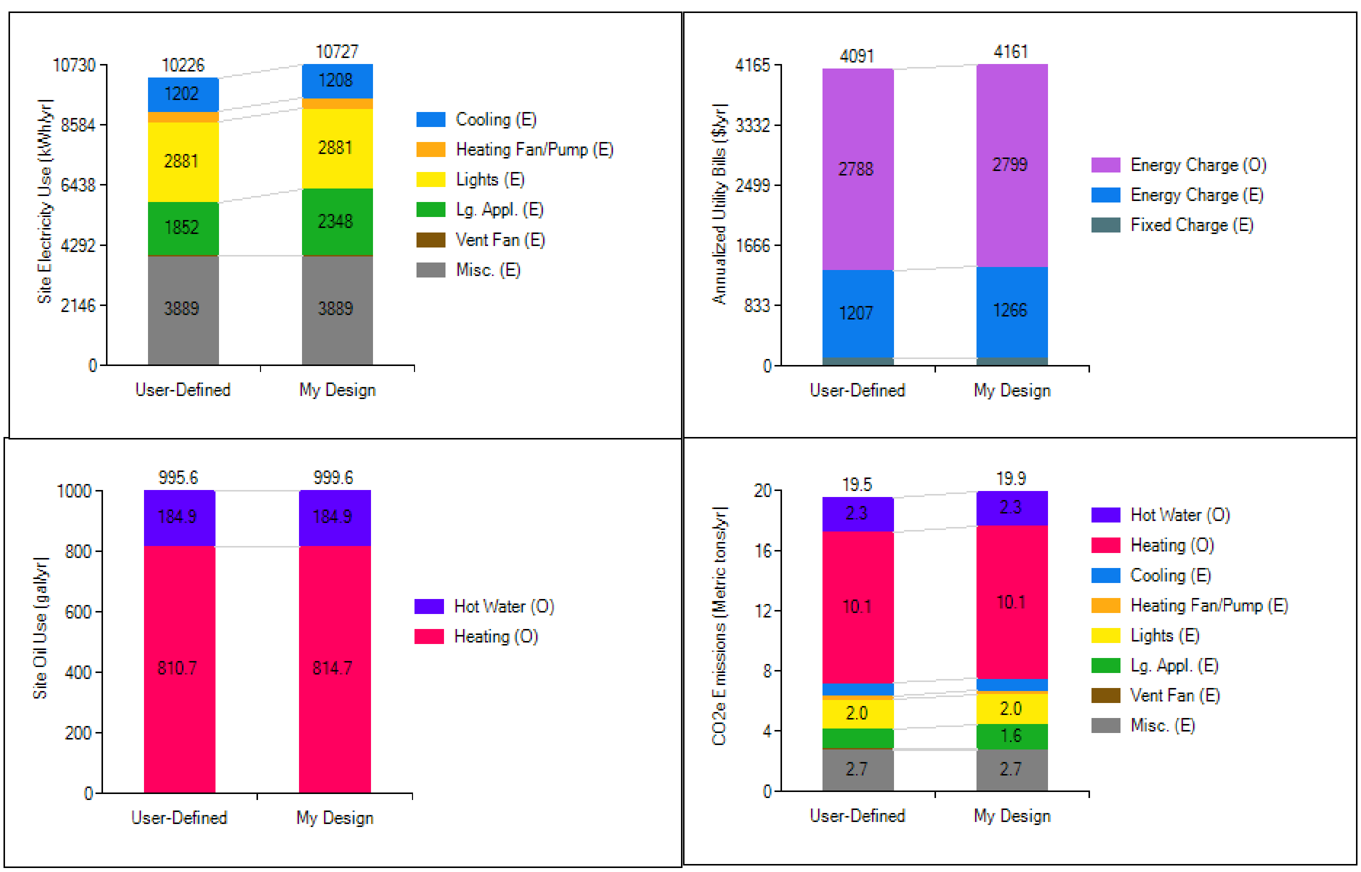
Ontario time of use pricing implemented in BEOPT



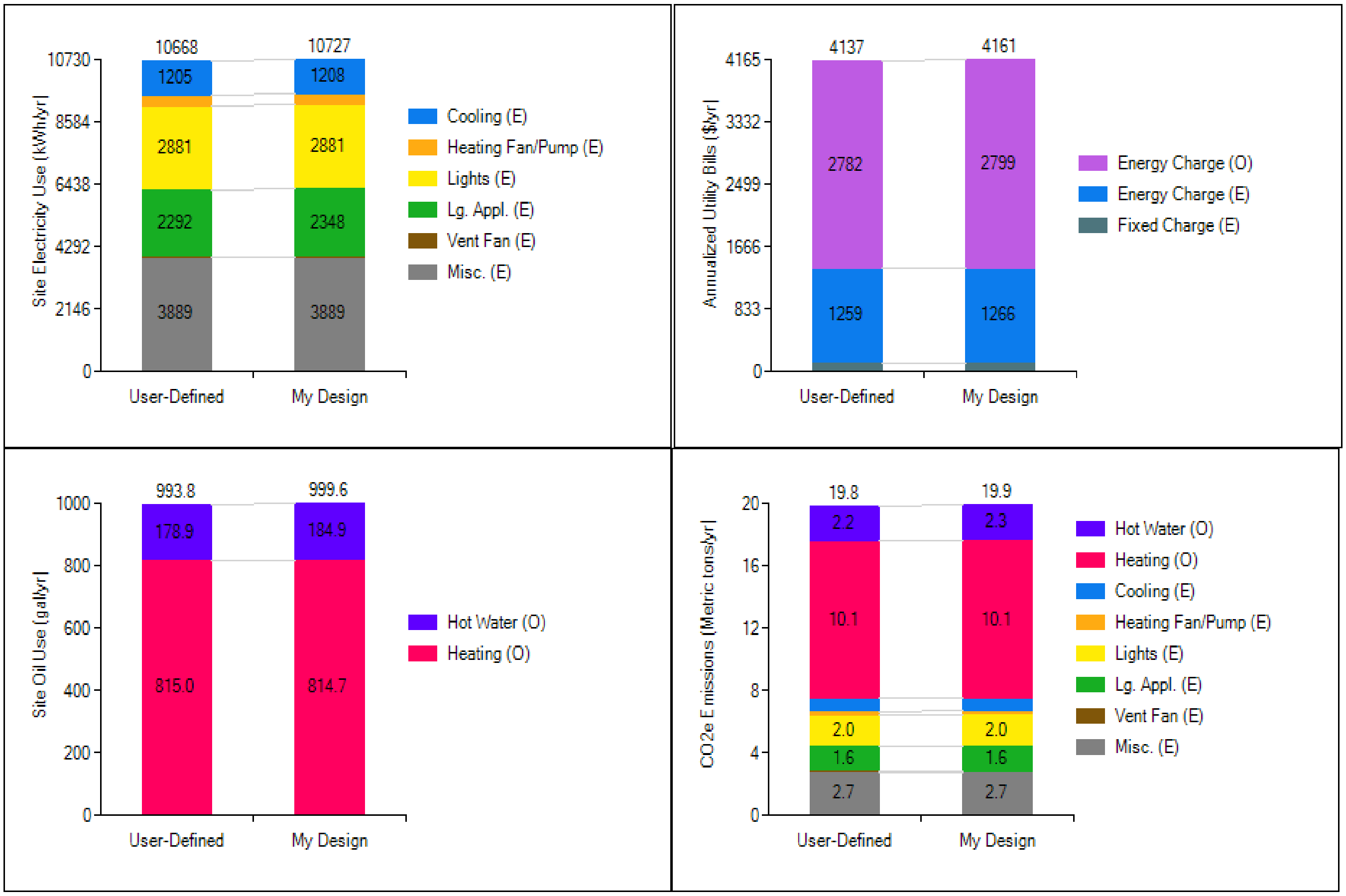
Ontario tiered pricing implemented in BEOPT



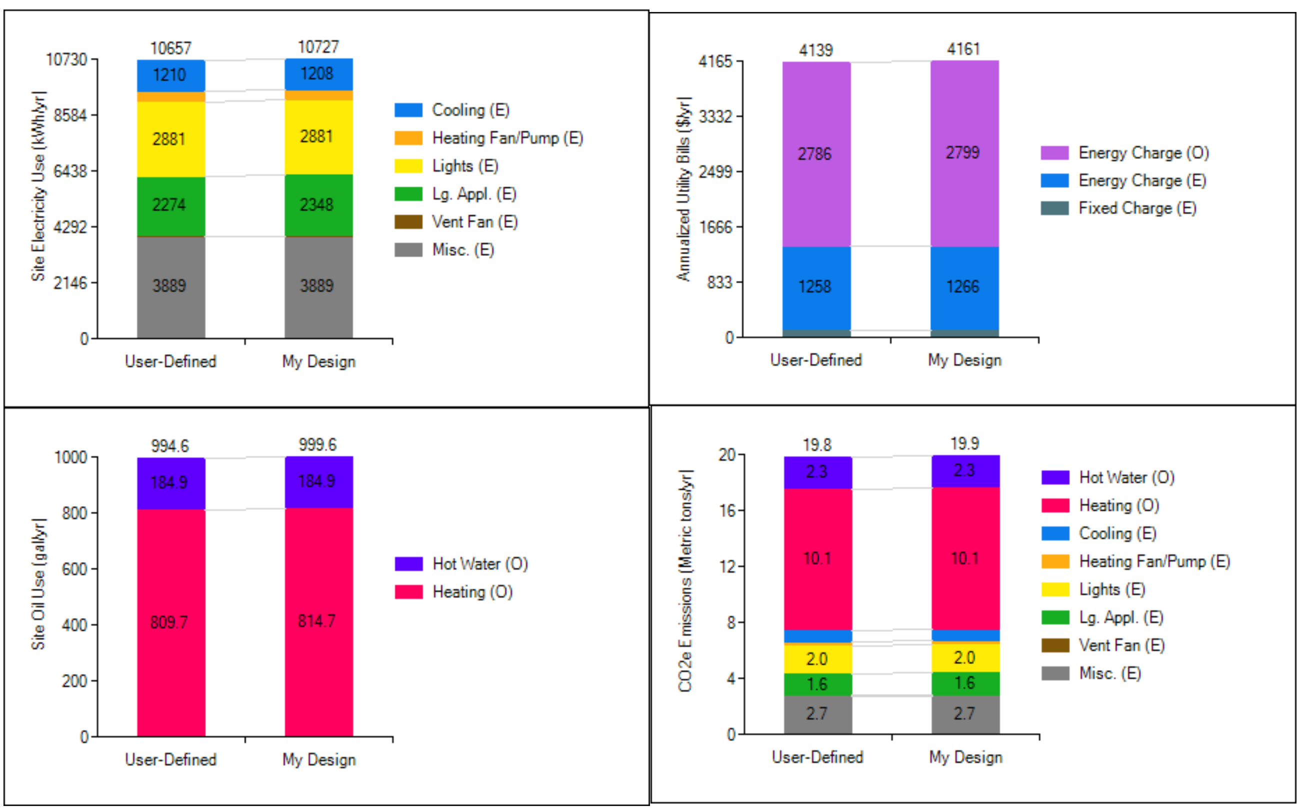
New clothes washer results



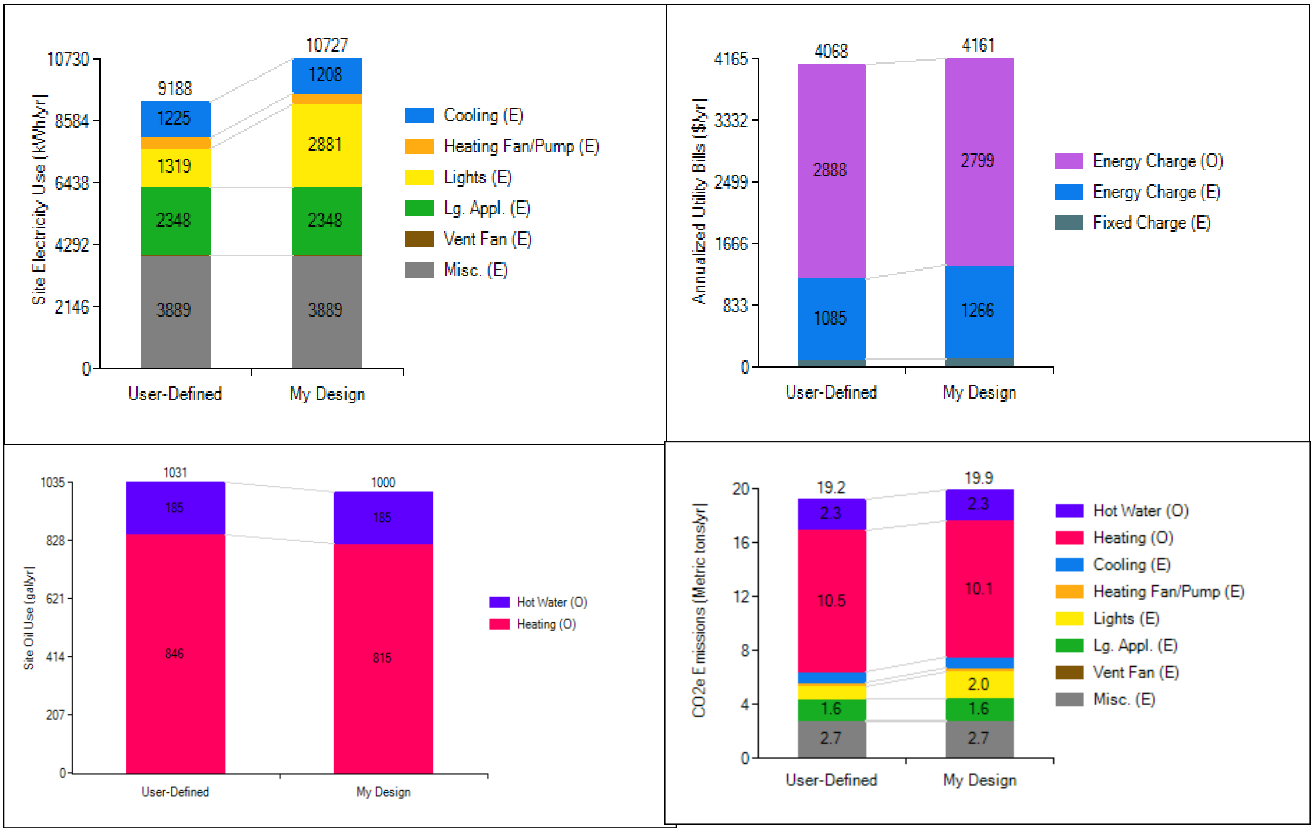
New clothes dryer Results



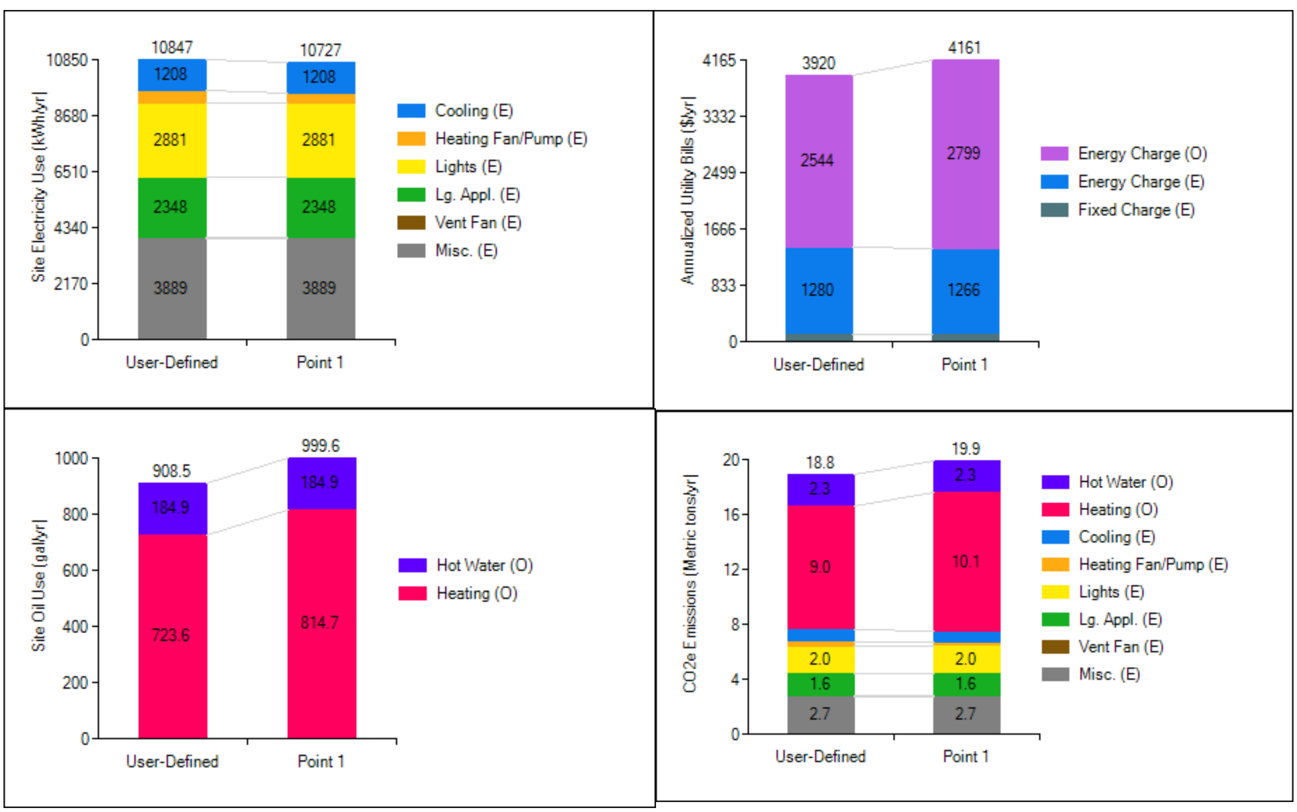
New dishwasher results



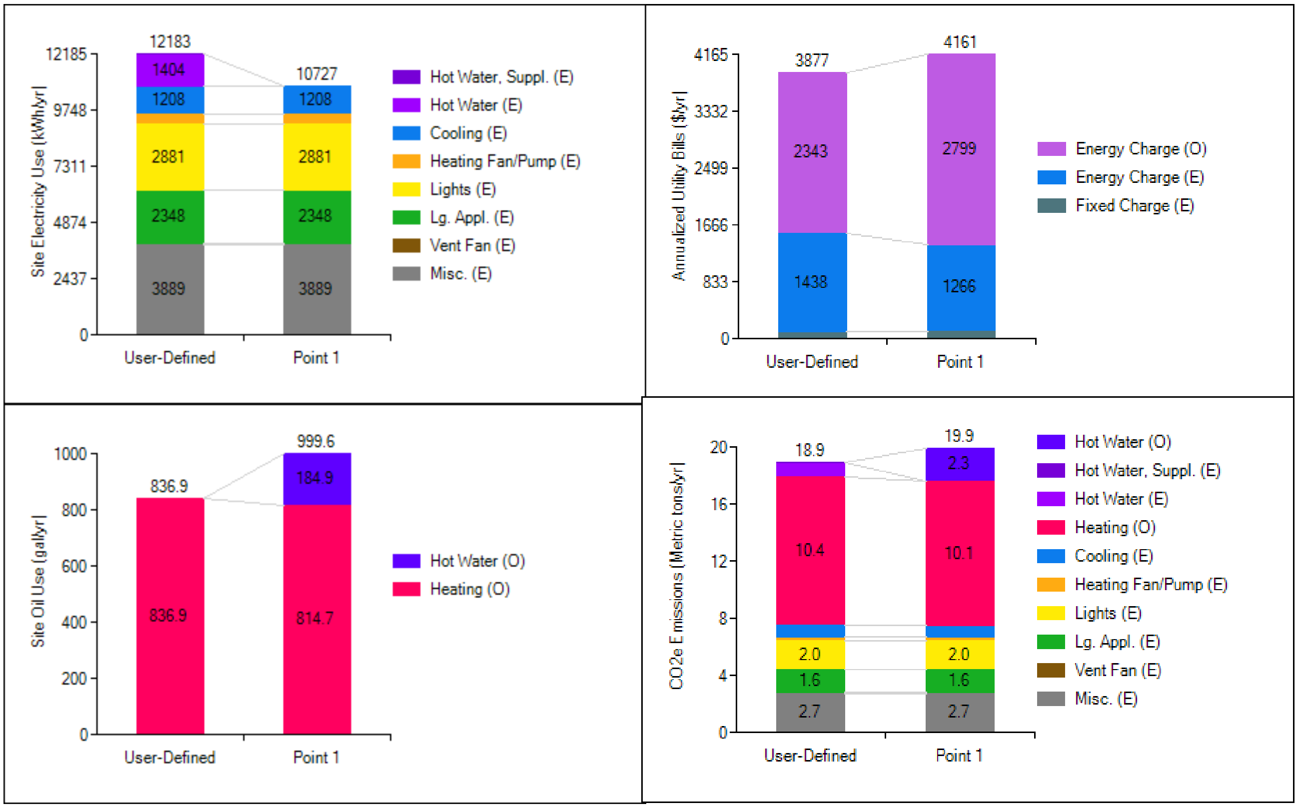
New refrigerator results



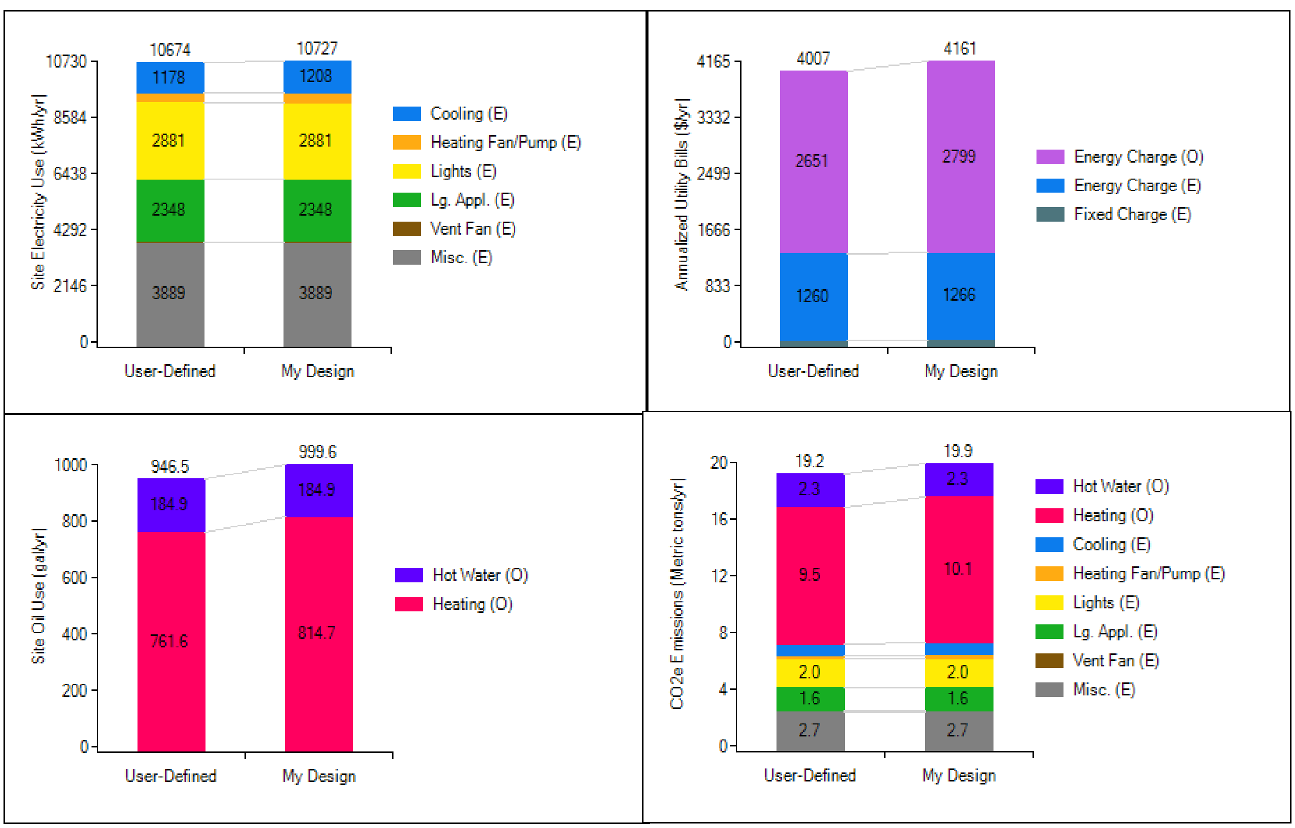
New lighting results



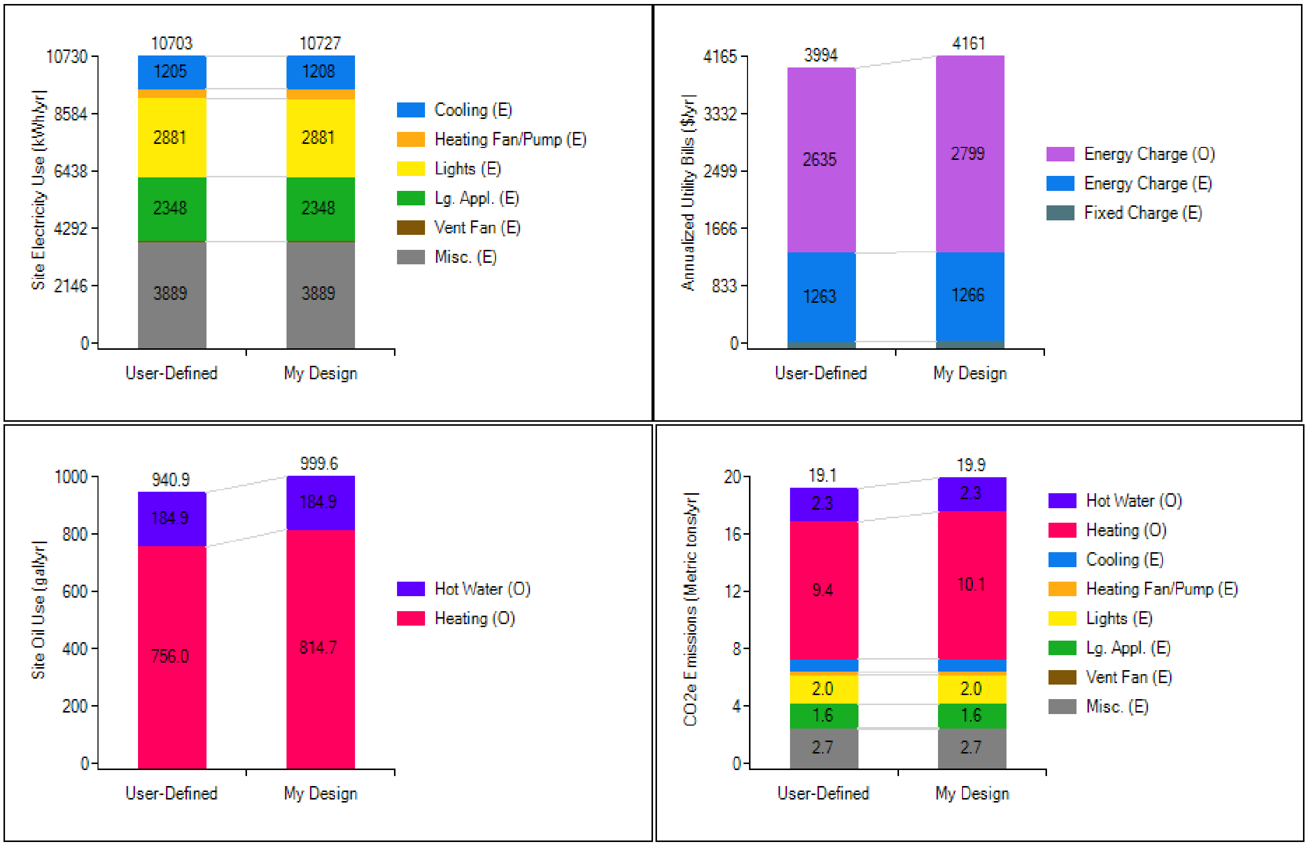
New boiler results



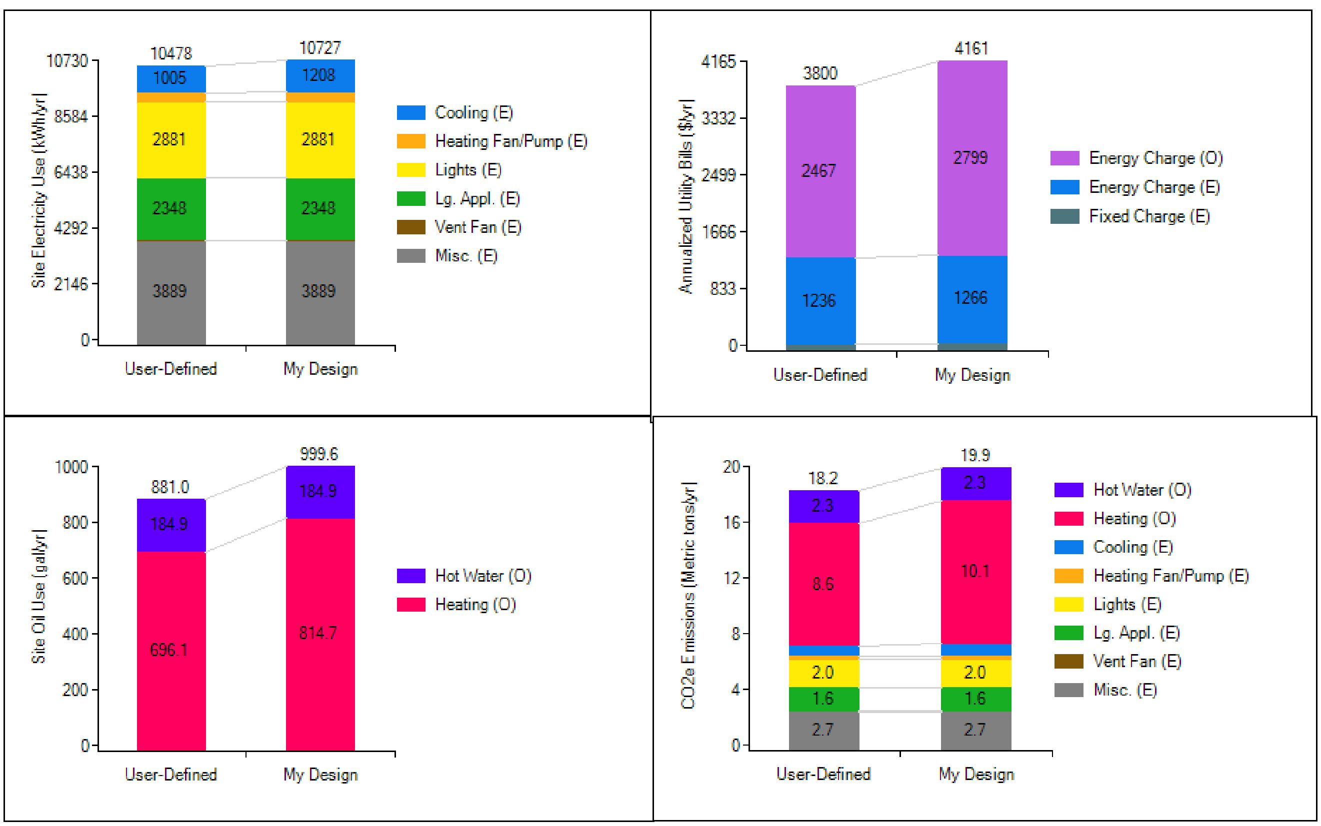
New water heater results



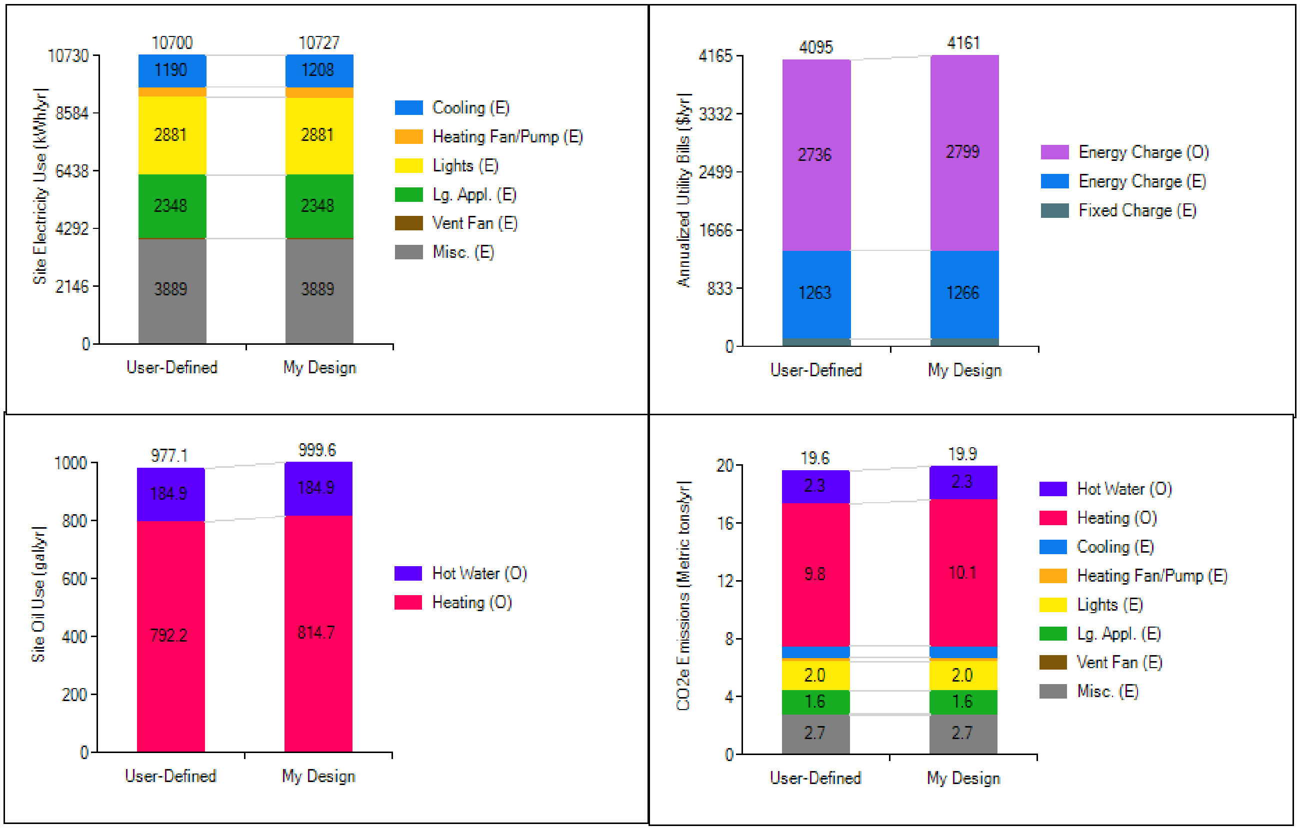
New attic insulation results



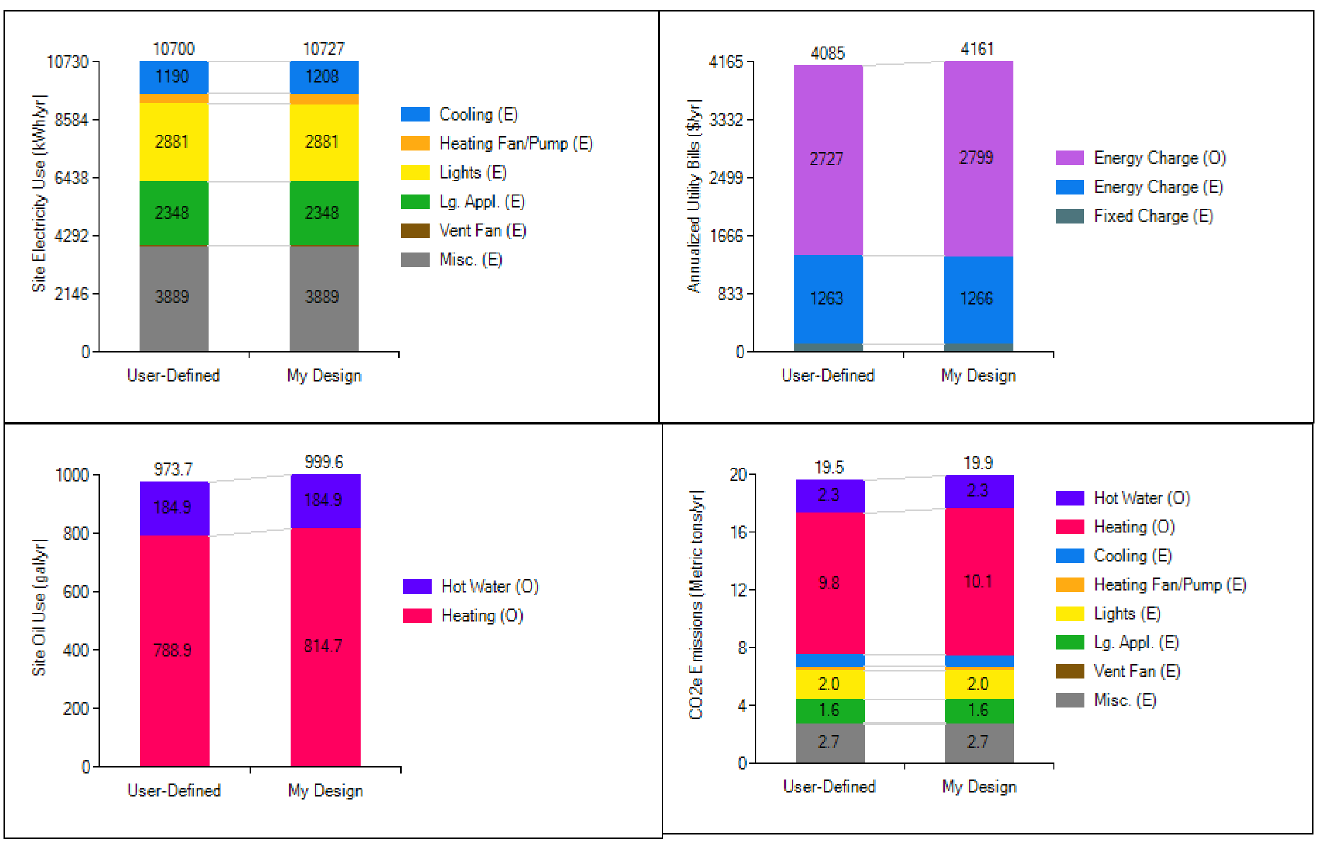
New finished basement insulation



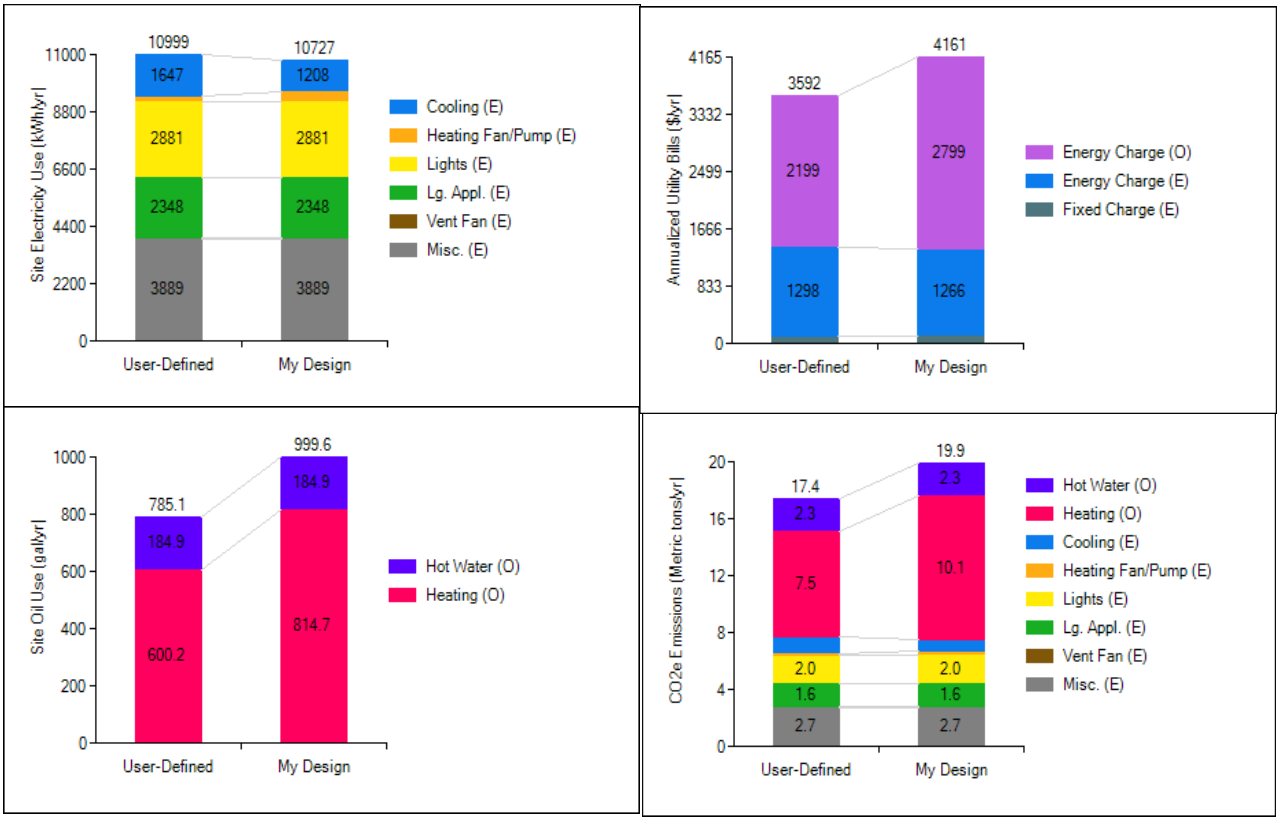
New windows results



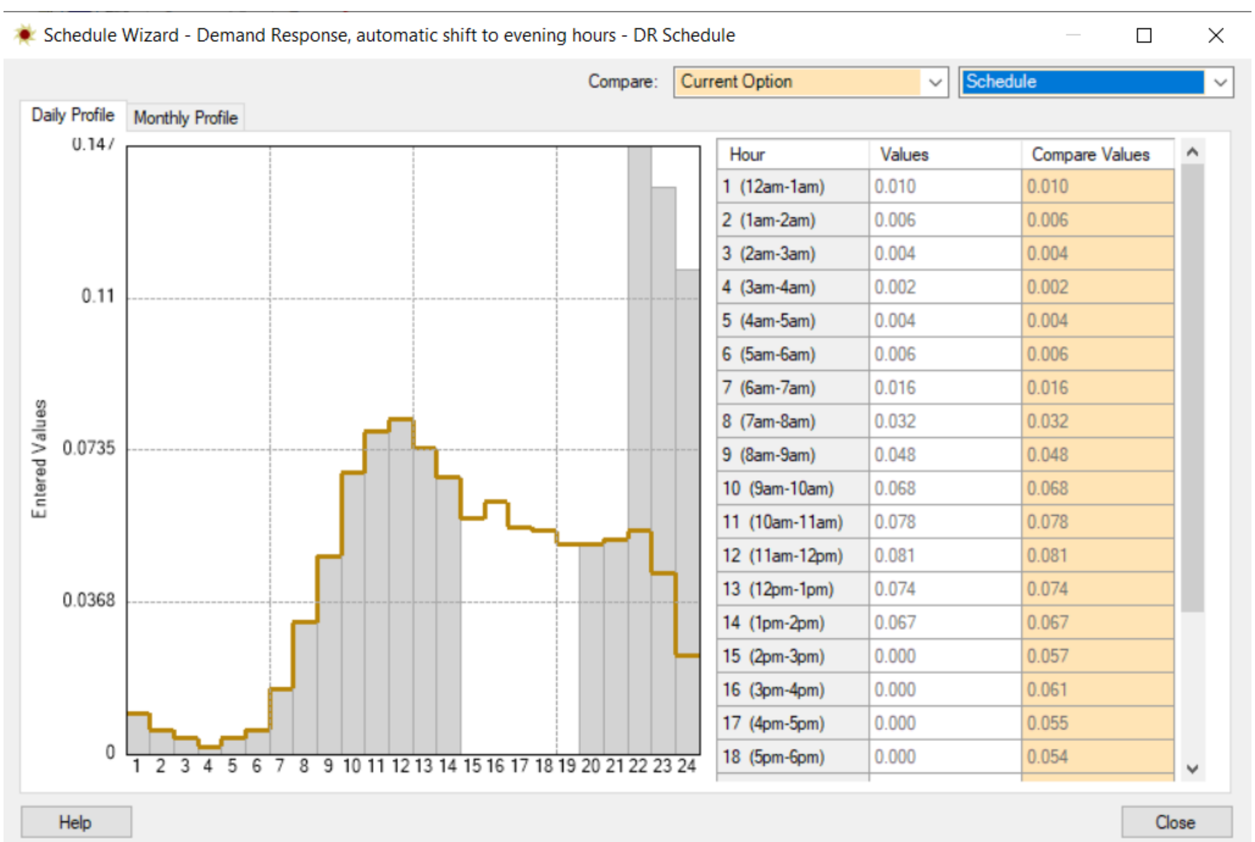
New living space insulation (R-15 XPS) results



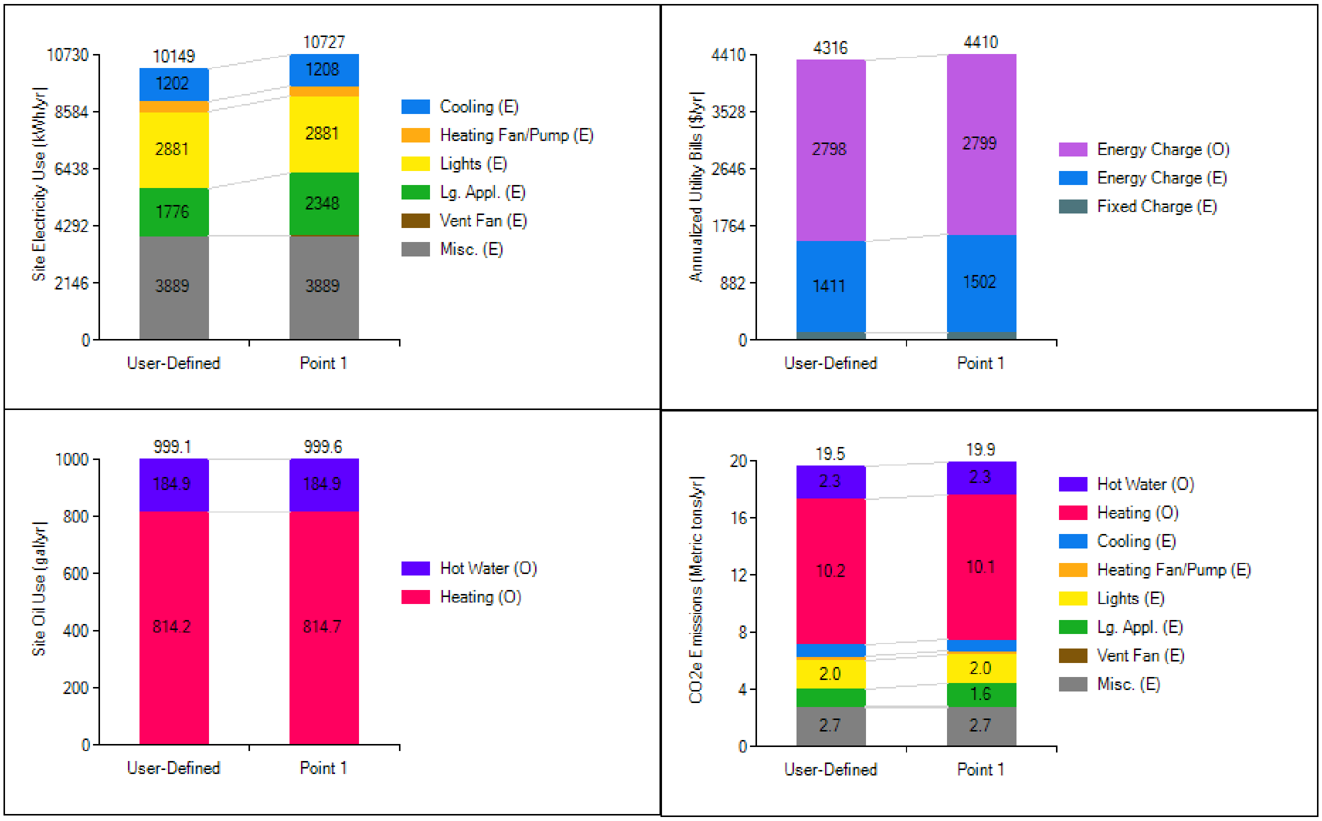
New living space insulation (OSB R-15 XPS) results



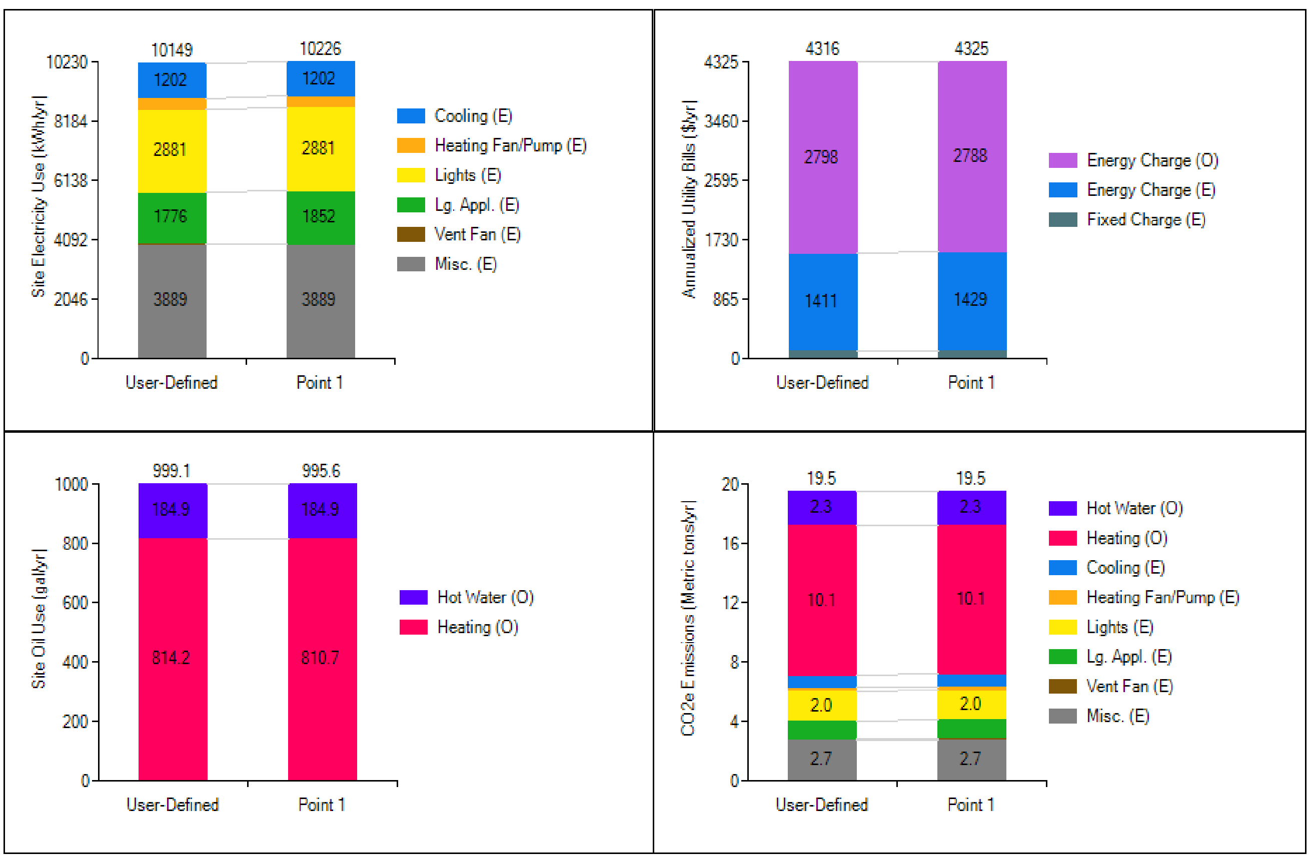
Air leakage reduction



DR schedule (bar chart) versus standard schedule (curve)



Smart dryer versus old dryer



Smart dryer versus energy efficient dryer

E. Appendix E- Plagiarism Check