EVALUATION OF AN INDIRECT SOLAR ASSISTED HEAT PUMP WATER HEATER IN THE CANADIAN ENVIRONMENT

by

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Abstract

Solar Domestic Hot Water systems and air-source heat pumps offer the potential for energy savings in residential hot water production, however their performance is limited in cold climates, where the low ambient temperature reduces the collector efficiency or the heat pump coefficient of performance. Combining these systems into a Solar-Assisted Heat Pump can alleviate these limitations by reducing the required collector temperature and by providing an increased heat pump evaporator temperature.

This study is a continuation of the development of an Indirect Solar-Assisted Heat Pump undertaken at the Queen’s University Solar Calorimetry Laboratory. Previously, a numerical study compared its performance to existing technology, and based upon this feasibility analysis, a prototype was constructed for controlled laboratory tests using simulated solar input. In the current study, the prototype was modified to include a novel hybrid collector such that its performance under actual weather conditions throughout the year could be assessed.

On sunny days, the system experienced daily averaged collector efficiencies between 0.47 and 0.88, depending on the flow rate and season. Averaged heat pump coefficients of performance of 2.54 to 3.13 were observed. Overcast days experienced reduced coefficients of performance, between 2.24 and 2.44. However, on overcast days, upwards of 76% of the collected energy gain was from convection with the surroundings.

Based upon these experimental results, a model for the hybrid collector was developed. Annual simulations of the system were conducted to compare the performance of the solar heat pump system when fitted with the hybrid collector relative to cases with more conventional glazed and unglazed collectors commonly used in solar thermal systems. Results were produced for three Canadian cities: Toronto, Vancouver and Winnipeg. The heat pump with the hybrid collector outperformed the other collectors in the Toronto climate, with a free energy ratio of
0.548. Adding a thermally controlled valve to the hybrid collector was proposed to further increase the annual free energy ratio, and was shown to perform best in all three cities, with free energy ratios of 0.558, 0.576 and 0.559 for Toronto, Vancouver and Winnipeg, respectively. It is proposed that additional improvements could be achieved by allowing the collectors to deliver heat directly to the storage tank, by circumventing the heat pump if the conditions were favorable.
Acknowledgements

I would like to thank Dr. Stephen Harrison for his knowledge, experience, and guidance throughout the course of my studies. It was a pleasure to be part of the development of this technology with him.

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Finally, I would like to thank the Ontario Graduate Scholarship program, the Queen’s University Departments of Graduate Studies and Mechanical and Materials Engineering, and the NSERC Canadian Solar Buildings Network for their financial contribution.
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Nomenclature

COP  Coefficient of Performance, (-)
FER  Free Energy Ratio, (-)
SAHP  Solar Assisted Heat Pump
DX-SAHP  Direct Expansion Solar Assisted Heat Pump
ISAHP  Indirect Solar Assisted Heat Pump
SDHW  Solar Domestic Hot Water
A  Area, (m²)
β  Coefficient of Thermal Expansion, (K⁻¹)
CPF  Collector Performance Factor, (-)
C_p  Specific Heat of Constant Pressure, (J/kg-K)
D  Diameter, (m)
E  Energy, (J)
η  Collector Efficiency, (-)
ε  Heat exchanger effectiveness, (-); emissivity, (-)
F  Fin Efficiency, (-)
F'  Collector Efficiency Factor, (-)
F_R  Heat Removal Factor, (-)
G_r  Incident solar radiation on a tilted surface, (W/m²)
g  Acceleration due to gravity, (m/s²)
H  Collector channel length, (m)
h  Enthalpy, (J/kg); convection coefficient, (W/m²-K)
HCW  Wind-induced convection coefficient, (W/m²-K)
k  Specific heat ratio, (-); thermal conductivity, (W/m-K)
L  Length, (m)
\dot{m}  Mass flow rate, (kg/s)
m  mass, (kg)
M  total mass, (kg)
v  kinematic viscosity, (m²/s)
N_cyl  Number of cylinders, (-)
Nu  Nusselt Number, (-)
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<td>Pr</td>
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<tr>
<td>Q</td>
<td>Power, heat transfer rate</td>
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<tr>
<td>q''</td>
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<td>Heat loss coefficient – area product for heat exchangers</td>
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Chapter 1

Introduction

1.1 Background

In Canada, water heating accounts for 17.4% of the residential energy consumption, making it the second most intensive end use of energy in this sector, behind space heating; of this energy, 98% is supplied from conventional purchased sources such as electricity, natural gas and heating oil [1]. With a growing desire to reduce dependence on fossil fuels, research into alternative heating methods has led to the development of various systems as means of reducing or eliminating the consumption of purchased energy for water heating.

One example is the Solar Domestic Hot Water (SDHW) system that uses solar radiation as the heating source, which can be considered ‘free’ energy. These systems typically consist of a solar collector, a storage tank and the plumbing necessary to connect the two. The solar collector converts the solar radiation to thermal energy and transfers it to a working fluid that flows through tubes or channels within the collector. The working fluid then transfers this thermal energy to the storage, either by passing through a heat exchanger, or by entering directly into the tank. An auxiliary heating system (e.g., electrical resistance heaters or natural gas burners) can then heat the water to a desired set-point if the solar output is insufficient.

Many configurations of SDHW systems exist which vary in complexity, and are designed for the climate in which they operate. For example, a tropical location can utilize a direct thermosyphon system, where the tank is located above the collector and water from the tank flows directly through the collector. As the water is heated within the collector, its density decreases creating a pressure differential across the system, inducing a flow; hot water exits the top of the collector, flowing into the top of the storage tank while cold water from the bottom of
the tank flows to the inlet at the bottom of the collector. These systems do not require any pumps to operate, and are therefore inexpensive and reliable.

In northern temperate climates, such as Canada, direct systems are impractical for year-round operation, as cold temperatures experienced in the winter can freeze the water in the system, ceasing its operation and damaging its components. As a result, indirect systems are used, which typically consist of two fluid loops separated by a heat exchanger as shown in Fig. 1-1. The collector loop is charged with a non-toxic anti-freeze mixture, typically 50% propylene glycol / 50% water by volume, that transports heat from the solar collector located outside to the heat exchanger and storage tank located inside, without freezing or harming the system during overnight or inclement weather periods.

![Fig. 1-1: Indirect SDHW system with external natural convection heat exchanger [2]](image)

As the collectors are usually roof-mounted, and the storage tank is typically located on a ground floor or within a basement, a pump is required to circulate the anti-freeze solution through
the collector loop. An external heat exchanger is often used to create a buoyancy-driven natural convection flow to circulate the potable water from the bottom of the storage tank to the top, and create a stratified condition within the tank [3]. Stratification occurs when the flow rate is low enough such that the hot water entering the top of the tank does not mix with the residual colder water. As a result, a stratified storage tank charges from the top downwards, allowing for hot water at the top of the tank to be available to the load while cold water is constantly supplied to the heat exchanger. The cold water supplied to the heat exchanger also ensures that the inlet temperature to the collector is maintained at a low temperature, allowing it to operate efficiently.

Another system which has potential to reduce residential energy consumption is an air-source heat pump water heater, shown in Fig. 1-2. A heat pump consists of four major components: an evaporator heat exchanger, a compressor, a condenser heat exchanger, and a thermostatic expansion valve. The components are plumbed together with a refrigerant as the working fluid. The evaporator is exposed to ambient outdoor air, while the condenser is the heat exchanger connected to the storage tank loop. A temperature-enthalpy diagram of the heat pump cycle is shown in Fig. 1-3, with approximate pressures and temperatures.

![Air Source Heat Pump system schematic](image)

**Fig. 1-2: Air Source Heat Pump system schematic**

3
Refrigerant passes through the evaporator at a low pressure such that its evaporation temperature is lower than the temperature of its surroundings, $T_{\text{evap}}$. As a result, heat is transferred through the evaporator to the refrigerant, which leaves the evaporator as a vapour at Point 1. The refrigerant is then compressed to a higher temperature and pressure at Point 2. At this pressure, the boiling point of the refrigerant is higher than the water temperature on the cold side of the condenser, $T_{\text{cond}}$. As the refrigerant passes through the condenser, it transfers heat to the potable water, exiting as a cooler saturated liquid at Point 3. The refrigerant is then allowed to expand through the valve to Point 4, decreasing in temperature in the process, to continue the cycle. As no work and minimal heat is lost to the environment across the valve, the enthalpies at Points 3 and 4 are approximately the same.

Therefore, the energy that the heat pump delivers to the potable water, $(h_2 - h_3)$, is that which it absorbed through the evaporator, $(h_1 - h_4)$, plus the increase in energy by the compression process, $(h_2 - h_1)$. As Fig. 1-3 shows, the majority of this energy is absorbed through the evaporator, therefore the main benefit of a heat pump is its ability to deliver more energy to the storage tank than required to drive the compressor; the ratio between these two energy values being termed the coefficient of performance, $COP$. As heat pumps transfer more heat than the electricity they consume, they are more energy efficient than electrical heaters, as long as there exists sufficient source heat to transfer.
Both systems have performance limitations in the Canadian climate, primarily during the winter months. Solar collectors are most efficient when the operate at or below the ambient temperature; cold ambient temperatures in the winter cause a decrease in collector efficiency, as there would exist a large thermal gradient between the anti-freeze and ambient temperatures, promoting excess heat loss. Low ambient temperatures also have a negative impact on the air-source heat pump. As less energy would be transferred to the refrigerant through the evaporator, the energy output of the heat pump, as well as the COP, would decrease.

Combining these two systems has the potential to alleviate these limitations. Depending on its configuration, a Solar Assisted Heat Pump (SAHP) would be able to increase the COP by directing heat from the collector loop to the evaporator, increasing its temperature above ambient.
With a higher $T_{\text{evap}}$ in Fig. 1-3, the compression process, $(h_2 - h_1)$, would require less energy to raise the refrigerant above the target $T_{\text{cond}}$.

Additionally, the solar collectors in a SAHP system operate at a higher efficiency than those in a SDHW system with the same load temperature. To supply hot water to a load at 50°C, for example, the fluid circulating through the collectors in a SDHW system must reach a temperature greater than 50°C for heat transfer to occur to the potable water. However, due to the heat pump’s ability to increase the refrigerant temperature through the compression process, an output temperature of 50°C can be maintained for a much lower evaporator temperature. In effect, this would reduce the temperature at which the collector operates, resulting in higher collector efficiencies. This also allows the system to operate for longer periods, as there could exist periods where there is insufficient solar radiation for a SDHW system, yet a sufficient ambient temperature to exceed the required evaporator temperature in a SAHP system.

1.2 Solar Assisted Heat Pumps

Research into SAHP systems has been ongoing since the 1950’s, and many configurations for different applications have been studied, including series, parallel and dual source systems, as explained in Chapter 2. In a series configuration, heat is transferred from the solar collector through the heat pump to the storage, with the collector plumbed in series with an evaporator heat exchanger, or acting as the evaporator itself. The majority of series SAHP systems in the literature are Direct Expansion Solar Assisted Heat Pumps (DX-SAHP), where the solar collector is integral to the heat pump loop, effectively acting as the evaporator. The refrigerant flows through the collector, absorbing thermal energy from the surroundings and the solar irradiance, expanding to a vapour in the process.

Another system type is the Indirect SAHP (ISAHP), which consists of three fluid loops. Essentially, it is a typical SDHW collector loop attached to the source side of the heat pump
evaporator, as shown in Fig. 1-4. While the indirect system has more components and the potential for increased thermal resistance, it has cost and operational benefits compared to the direct expansion system. A DX-SAHP either has long refrigeration lines that run from the compressor and storage tank (usually located indoors) to the collector located outdoors, or alternatively a storage tank located outdoors with the collector/evaporator. The former can lead to expensive installation costs, requiring an onsite refrigeration specialist, while the latter is impractical for regions that experience low ambient temperatures. The heat pump circuit in the ISAHP can be charged with refrigerant and packaged as a unit in the factory, while the collector loop requires lower pressure lines and fittings, simplifying the installation.

Fig. 1-4: Indirect Solar Assisted Heat Pump system schematic
1.3 Solar Collector Types

As with any solar thermal system, the overall performance of the ISAHP depends on the type of solar collector that is used. The three basic types of collectors are unglazed flat plate, glazed flat plate, and vacuum tube, and each has a range of temperatures across which they are the most desirable for a given application: unglazed collectors have the highest efficiency at low operating temperatures, glazed collectors at moderate temperatures, and vacuum tubes at higher temperatures. This is due to the design of each collector type, and the resulting heat transfer resistances between the collector and the surroundings.

Unglazed collectors are the simplest and least expensive of the three collector types. In their basic form, they consist of an absorber plate, typically aluminum, with heat transfer fluid channels attached to the plate or passing through it. The top surface of the absorber is painted or treated with a high absorptivity coating to maximize the fraction of the irradiance that is absorbed by the collector. The rear and sides of the absorber are typically not insulated as they usually operate at temperatures near the ambient air temperature such that thermal losses are insignificant. Glazed collectors are very similar to their unglazed counterparts except the rear and sides of the absorber are usually insulated to limit losses to the surroundings at higher operating temperatures. A transparent cover (or glazing) is mounted above the absorber to further reduce convection and long-wave radiation losses from the absorber. The inclusion of the cover does increase the optical losses, as the cover reflects and absorbs a fraction of the irradiance, typically transmitting around 90% to the absorber. The third collector type, a vacuum tube, varies significantly in its construction compared to the glazed or unglazed flat plate collectors. Each vacuum tube consists of a small absorber plate inserted into a glass cylinder, without contacting the edges. The cylinder is evacuated of air and sealed, producing a vacuum such that any convection or conduction from the absorber to the cylindrical glass wall is inhibited. Multiple vacuum tubes are connected to a single header pipe to produce a collector array. Vacuum tube
collectors have lower heat loss coefficients than flat plate collectors, however, they often have greater optical losses due to a lower aperture to gross area ratio caused by the spaces between adjacent vacuum tubes.

The efficiency, \( \eta \), of a solar collector operating under steady-state conditions is often expressed as a linear function of the inlet temperature, \( T_{in} \), ambient temperature, \( T_a \), and total solar irradiance, \( G_T \), and is given by the Hottel, Whillier and Bliss equation [4], as follows:

\[
\eta = F_R (\tau \alpha) e - F_R U_L \frac{(T_{in} - T_a)}{G_T}
\]  

(1-1)

where \( F_R \) is the collector heat removal factor, \( (\tau \alpha) e \) is the effective transmission-absorption product of the cover and absorber, and \( U_L \) is the overall heat loss coefficient from the absorber to the surroundings. The first term accounts for the energy that is absorbed by the collector, and the second term represents the energy that is lost. The efficiency represents the ratio between useful energy collected to that of solar radiation incident on the total area occupied by the solar collector. Reductions in thermal efficiency are due to optical losses, as well as thermal losses due to convection and radiation heat transfer from the top, sides and back of the collector housing, as shown in Fig. 1-5.

The variability in collector efficiency for sample unglazed, glazed, and vacuum tube collectors is illustrated in Fig. 1-6, where it can be seen that unglazed collectors have the highest efficiency at low temperature levels, due to their lower optical losses. As a result, it has been common practice to use unglazed collectors in SAHP systems to capitalize on the fact that lower collector temperatures caused by the heat pump reduce thermal losses and maintain high efficiency [5] [6] [7]. Additionally, as the unglazed collector has a high \( U_L \), it can, under certain situations, (e.g., when the collector absorber is at lower temperature than the surroundings)
absorb more energy from the surrounding air than the glazed or vacuum tube collectors, due to a decreased thermal resistance.

However, as the COP of the heat pump decreases with the evaporator temperature, it would be detrimental to operate the collector near the ambient temperature during the winter months, as additional electrical energy would be necessary to meet the load. It has been observed that unglazed collectors performed poorly when compared to glazed collectors in annual simulations of a SAHP operating in Canadian cities [8]. On a seasonal basis, a SAHP with a glazed collector has the potential to outperform the unglazed collector in the winter time, with its greater efficiency at larger temperature differences. However, a glazed collector cannot benefit from the below-ambient operation during summer months to the extent that an unglazed collector would. As a result, this study investigates the use of a modified glazed collector with an air channel between the absorber and rear insulation, as shown by the cross-sectional side view in Fig. 1-7, in an attempt to strike a balance between the unglazed and glazed collector performance. The air channel allows the absorber to exchange heat with the ambient air by natural convection like an unglazed collector, while the glazing and insulation limits the heat loss when required. In effect, this becomes a hybrid collector, able to more efficiently absorb energy from solar, as well as, ambient sources.
Fig. 1-5: Generalized efficiency curve showing components that contribute to the lost energy.

![Generalized efficiency curve showing components that contribute to the lost energy](image)

Fig. 1-6: Efficiency curves of three collectors: (a) unglazed, (b) glazed, and (c) vacuum tube.

![Efficiency curves of three collectors](image)
Fig. 1-7: Cross-section of the hybrid collector

- Glazing
- Air gap
- Absorber Plate
- Serpentine Tubes
- Rear Channel
- Insulation
1.4 Problem Definition

A feasibility study of this ISAHP configuration was previously conducted at the Queen’s Solar Calorimetry Laboratory (SCL) with positive results, as discussed in Chapter 2. A numerical model of the system was created in the TRNSYS simulation program by Freeman [9] as discussed in Chapter 3, and an experimental prototype was constructed and tested under controlled inputs by Bridgeman [10] to validate the model. As a continuation of this project, the current study includes an investigation of the performance of the ISAHP under actual weather conditions to highlight performance aspects that would not be apparent under controlled conditions. Additionally, the use of different collector types was investigated to compare their annual performance, and determine the collector type most suitable for the Canadian climate.

1.5 Objective, Approach and Scope

The primary objectives of the study were to modify the prototype developed by Bridgeman to incorporate a flat plate collector located outdoors at the SCL, and to assess the ISAHP’s performance under actual year-round weather conditions. These results would be used to validate and refine an updated TRNSYS model that reflects the installed configuration. This study also included the testing and development of a heat transfer model of a hybrid collector, and to determine, through simulations, if it yields better annual performance than the typical unglazed collector or an unmodified glazed collector. A secondary objective was to determine limitations in the system’s operation, and provide insight for improvement. In order to accomplish these objectives, the flowchart described in Fig. 1-8 was followed.

Initially, a literature review of SAHP systems was conducted, focusing on experimental results for various system configurations. The TRNSYS simulation package and the previously developed system model were reviewed to ensure their proper use. Two identical EnerWorks “Heat Safe”™ glazed collectors were procured and modified; one for conducting controlled tests,
and one to install with the ISAHP system. Performance tests were performed on the ISAHP a range weather conditions that occurred throughout the year, and the thermal characteristics of the collector under controlled input power levels were determined. Concurrently, the collector model was developed based on an existing model, and validated against results from the ISAHP tests. The results from the collector model and experiments were used to configure the TRNSYS model for annual simulations, which were used to assess the performance of different collector types.

The scope of this study included the adaptation the ISAHP prototype to incorporate a real hybrid flat plate collector to replace the electric heaters used in the previous study to mimic the solar collectors. None of the components of the heat pump were replaced or modified, and the existing data acquisition system was used, with the addition of certain sensors for the collector loop.

Day-long storage tank charge tests of the ISAHP system were conducted; therefore before each test the tank was refilled with cold mains water. As a result, the multi–day performance of the heat pump, where the state of charge of the tank varies day-to-day, was not investigated. Additionally, the tests were conducted without water drawn from the tank, such that the end state of the storage tank reflected the energy output of the system. The performance of the hybrid collector was experimentally evaluated and used to validate the collector and TRNSYS models, while the performance of alternate collector types was compared through TRNSYS simulations conducted under identical conditions to ensure consistency in the evaluation.
Fig. 1-8: Project Approach
Chapter 2

Literature Review

2.1 Introduction

Energy use and efficiency has received much attention in recent years with the rising costs of fossil fuels and the threat of climate change. A major factor in residential energy consumption is the generation of hot water, and consequently a number of systems have been developed that heat water with a lower consumption of purchased energy, such as heat pumps and solar hot water systems. Research has shown that the combination of these systems into a solar assisted heat pump can further improve their performance and utilize less energy.

2.2 Solar Assisted Heat Pumps

Many configurations of SAHP systems have been studied in the past 30 years for both space heating and water heating applications. The following sections present a review of the performance of a selection of these studies, with a focus on current results from water heating systems. Additionally, the key findings of the previous Queen’s University SAHP studies are presented as a background to the current work.

2.2.1 External Studies

The idea of a SAHP system has been around since the 1950s, however extensive research into the subject did not begin until the late 1970s. In 1978, Freeman, Mitchell and Audit numerically studied the performance of various SAHP systems for combined space and water heating [11]. The goal of the study was to determine the thermal performance of the systems and give insight into non-intuitive results. Using the TRNSYS simulation package and weather data for Madison, Wisconsin, the authors investigated three SAHP configurations: parallel, series, and dual source; and compared their performance to a base solar heating system and a conventional
heat pump. The base system consisted of a solar collector loop exchanging heat with a storage tank, and a hot water loop that transferred heat from the storage tank to a coil within the air heating duct. A resistance auxiliary heater was located after the hot water coil in the air duct. The parallel system was essentially the base system plus an air-source heat pump with its condenser located ahead of the hot water coil in the duct. In the series configuration, the hot water from the storage tank passed through the evaporator of the heat pump, increasing the refrigerant temperature above ambient. If the temperature of the water was high enough, it would bypass the evaporator and deliver heat directly to the air duct. The dual source configuration was similar to the series system, but the heat pump had an additional air-source evaporator for conditions when the water loop temperature was low. It had three heating modes: direct solar, series SAHP and air-source heat pump operation.

Freeman, Mitchell and Audit based their performance comparison on the fraction of the load that was provided by non-purchased energy, also known as the free energy ratio, FER. This ratio was determined for each system over a range of collector areas while maintaining a 0.75 m³/m² storage volume to collector area ratio. Over all areas, the parallel system had the highest free-energy ratio, while the series system surpassed the conventional heat pump at 12 m² of collector area, approaching the parallel performance at large areas. The dual system performed slightly worse than the parallel system across the entire range. It is interesting to note that the ranking of the COP of each system does not reflect the ranking of the overall performance. The heat pump average COP of each configuration was 2.8, 2.5, 2.1, and 2.0 for the series, dual, conventional and parallel systems. The parallel system heat pump operated less frequently, requiring less electrical energy; yet when it did run, it did so with lower evaporator temperatures, leading to a reduced COP. The series configuration used the highest evaporator temperatures, which gave the highest COP, however as the storage tank was located before the heat pump, its
temperature was not often hot enough to supply direct heating. Therefore the heat pump was running the majority of the time, consuming the most electricity.

A study of similar SAHP configurations for Canadian cities was conducted by Chandrashekar, Le, Sullivan and Hollands in 1981 [8] for single and multiple dwelling buildings. Six different systems were studied that included liquid or rock bed storage for either water or air-based systems. To narrow down the systems to study in depth, simulations were conducted in the simulation program WATSUN for Winnipeg and Vancouver, which were considered extremes in Canadian climates, and energy use was compared to the life cycle cost of each system. Based on the cost assumptions of the authors, none of the systems proved to be more economical than the base air-source heat pump; however all systems provided significant energy savings in Winnipeg, especially the water-based dual systems. In Vancouver, the parallel source system was superior, yet as the Winnipeg climate is more representative of the rest of Canada, in-depth simulations were conducted on the dual source systems. The dual source systems were found to be nearly insensitive to location, with the exception of the air-based system having much greater performance in Vancouver. Energy savings were upwards of 65% and 75% for the air and water based systems, respectively, when compared to resistive heating. It was also found that under rising energy costs, the dual source system would become economical in Vancouver, yet not in Winnipeg. The authors conducted a sensitivity analysis to determine the effect of changing component parameters; notably, a decrease in performance was observed when the systems were operated with unglazed collectors, as opposed to single-cover glazed collectors.

The previous studies have concentrated on space heating with the possibility of cooling, and have suggested that series systems have the lowest performance of the SAHP configurations. However, Morgan argues that series systems designed solely for heating can be competitive, especially if used exclusively for water heating [5]. In a space heating application, the demand for
heat often occurs during periods of low solar input; therefore the series system would be operating with low evaporator temperatures resulting in low system performance. In a water heating application with storage on the condenser side of the heat pump, this demand-offset is not as prevalent. In 1981, Morgan designed and tested a series SAHP for tropical climates with a 140 W hermetic reciprocating compressor and a 1 m² inexpensive, fast-response unglazed collector. The configuration used would later be termed a direct-expansion solar assisted heat pump (DX-SAHP); the collector was plumbed into the heat pump circuit and as such, the refrigerant passed directly through the collector, evaporating as a result of the solar input. The refrigerant would then pass through the compressor, water cooled condenser, and expansion valve to continue the cycle.

The theoretical analysis of the system assumed that the collector/evaporator was operating at steady state, and performance calculations were made using manufacturer’s data for the compressor performance. Weather data was input as an average over five minute intervals while the condenser output was calculated over one minute intervals. The calculated evaporator and condenser temperatures were compared to those found experimentally. Condenser temperatures agreed well over the entire test, while the evaporator temperatures only agreed over the first 80 minutes of the test. The predicted evaporator temperatures varied greatly after the 80 minute mark, suggesting that the steady state assumption of the evaporator was not valid over the time step if large transients occur in the weather data. The experimental system $COP$ varied between 2.0 and 3.25, however the predicted values were not presented as it was stated that the model was unable to accurately predict the $COP$.

A similar DX-SAHP was designed and tested in 1983 by Chaturvedi and Shen under winter conditions in the Eastern United States [6]. The system consisted of \(\frac{1}{2}\) hp (373 W) open-type compressor, a water chilled tube-in-shell condenser, and a 3.39 m² back-insulated, unglazed
A back-up fan and coil air-source evaporator was connected in parallel to the collector, and isolated by valves, for use in the absence of sufficient solar radiation. An algebraic model for the steady state operation was compared to quasi-steady state experimental data obtained under clear skies around solar noon. Across various tests, the ambient temperature ranged from -4 to 22°C, the solar radiation ranged from 700 to 1000 W/m², with condensing temperatures varied between 40 and 50°C. Collector temperatures were maintained between 0-10°C above ambient, within the authors’ desired range. Collector efficiencies between 0.4 and 0.7 were found and the system $COP$ varied between 2.0 and 3.0 for ambient temperatures between 0-22°C, which surpassed an air-source heat pump that had a $COP$ around 1.5 during the same test period. The authors claim the $COP$ of the SAHP could reach 3.0-6.0 with the use of a more efficient, hermetic compressor with a larger capacity.

Chaturvedi and Shen discussed the advantages and disadvantages of a DX-SAHP system. As the collector was plumbed directly into the heat pump loop, there was little thermal resistance through the system, which could lead to superior thermal performance under ideal conditions. However, the use of long refrigeration lines that connect the collector located outside to the rest of the system located inside has an increased potential for leaks and trapping of oil within the evaporator. Additionally, as there would be little thermal resistance and mass, large fluctuations in solar radiation and ambient temperature could result in significant mismatch between the compressor and evaporator, reducing thermal performance, and possibly damaging the system.

In 1994, Morrison studied the commercial systems available in Australia [12], and found that commercial success was limited due to the high installation costs associated with charging the long refrigeration lines on site. The author studied two systems in TRNSYS to provide performance data to help in the commercialization process. The first system was a DX-SAHP with integral wrap-around condenser on the storage tank, which eliminated the need for any
additional pumps in the system, eliminating parasitic loads. The second system was an integral DX-SAHP, where the collector/evaporator was built into the outer surface of the storage tank. As such, the system could be assembled, charged and shipped as a single unit, virtually eliminating installation costs. However, this system required outdoor installation, which limited the climatic regions in which it would be an acceptable product. In each configuration, the collector/evaporator was unglazed, and as a result, the absorber would be in direct contact with the ambient air. Morrison noted that under limited solar radiation, the collector would operate below ambient temperature, providing heat gains by convection. Therefore, Morrison included terms for these gains in his collector modeling. Energy savings, when compared to a base electric system, were found to vary between 65-75% throughout the year for the DX-SAHP, while the monthly average COP varied from 2.5 to 3.3. Energy savings were not reported for the integral system, however it would be expected to be lower than the DX-SAHP, as the COP was approximately 0.25 lower for the same monthly averages. The systems operated more effectively than those studied by Chauvedi and Shen, which can be attributed to the higher ambient temperatures associated with the Australian climate as opposed to the Eastern United States.

Recently, research has continued on SAHP systems in the tropics, notably with Hawlader et al. investigating the DX-SAHP system [7] [13], and Huang et al. investigating the integral system [14] [15] [16]. Hawlader et al. conducted experiments on a DX-SAHP with two 1.5 m² unglazed collectors in series and a variable speed compressor [7]. The second collector could be by-passed depending on the compressor speed, which helped to reduce the mismatch between the collector/evaporator load and the compressor capacity. Throughout the tests, which lasted 4 hours under high levels of irradiance, it was observed that the COP and collector efficiency decreased significantly due to the rising temperatures within the condenser tank, which consisted of a wrap-around integral condenser. The COP was reduced from 8.5 to 4 within this time span, while the
collector efficiency decreased from 0.85 to 0.7, or from 0.75 to 0.65, depending of the compressor speed. This significant decrease can be attributed to the inability of the storage tank to provide a constant, cool temperature to the condenser. Later tests, when this system was used in a crop drying application [13] did not show this decreasing trend, presumably due to the increased thermal load.

Huang et al. have studied the integral system extensively. An initial feasibility study of the system, with a 1.44 m² collector, a 105 L storage tank, and a 250 W compressor [14] showed COP values of 3.5 could be obtained under warm, sunny conditions, while this decreased to 1.34 under overcast and cooler conditions. Concerned with the unsteady nature of the energy input, and its effects on the durability of the compressor, a 13,000 hour continuous test was performed, which showed no mechanical failure [15]. Average energy consumption for the system was 0.019 kWh/L of water at 57°C, which was significantly less than the 0.06 kWh/L determined for a base electric heater. One drawback of series SAHP systems was that the compressor must always be on to transfer heat from the collectors to the storage. To reduce compressor run time, and consequently extend system life, an adaptation was made to the integral system to incorporate a second operational mode: a heat pipe mode with no electrical consumption [16]. Under sufficient irradiance, the heat pump would deactivate and the compressor and expansion valve would be bypassed. The collector/evaporator was modified from the previous study to act as a heat pipe, in which fluid circulated through the system by buoyancy effects in the evaporation and condensation processes. The daily average COP of the heat pipe enhanced system was increased by 28.7% operating in hybrid mode, when compared to operation in strictly heat pump mode. This is due to the fact that the compressor operates for less time, consuming less energy, while the system delivers the same output. However, it should be noted that lower instantaneous COP values were observed for the hybrid mode, as when the compressor was running it was doing so
at lower average evaporator temperatures, similar to the parallel system studied by Freeman, Mitchell and Audit [11].

Intuitively, a SAHP performs at its best under sunny conditions; however due to the heat pump’s capacity to lower the collector temperature, the collector could operate below ambient temperature on days with insufficient irradiance. In 2006, Xu et al. took advantage of this operational mode by incorporating a specially designed collector into a DX-SAHP to increase the energy gains from the ambient air [17]. Their design was a 2.2 m² serpentine unglazed flat plate collector with a spiral fins along the tube length, which increased the area for heat transfer by convection. This collector was coupled to a 400 W rotary compressor with a 150 L storage tank connected to the condenser. The water was circulated through the condenser by a pump with a constant flow rate of 0.125 L/min. Simulated results for sunny days in the spring and autumn in Shanghai, China, resulted in an average COP of 4.32, while the collector gained 9% of its useful energy from the ambient air. On clear summer days, with warmer ambient air, 21% of the useful energy was gained by convection, and the average COP was 4.69, while for sunny winter days, the COP dropped to 3.83, and 31% of the energy was from the ambient air. Overcast winter days showed a COP of 3.3, with 85% of the energy absorbed by convection from the surrounding air, effectively making the system an air-source heat pump. In all these simulations, it was shown that the collector always operated below ambient temperature.

While DX-SAHP and integral systems have been shown to be viable options in warm regions, the performance of SAHP systems in regions that experience colder winter conditions, such as Canada, is of primary interest. One such system operating in Germany was studied experimentally and numerically by Bertram et al. in 2008 [18]. This system used a 44 m² array of unglazed collectors as the heat source for a 16 kW ISAHP to provide domestic hot water and radiant floor heating. Unglazed collectors were chosen due to their high collector yield (i.e.,
efficiency) at low temperature levels, which can be advantageous in a heat pump system. However, it was noted that during the winter, when heating demand was at a maximum, the unglazed collectors could only provide low output temperatures, due to the collectors’ high heat loss. Therefore the system also incorporated 14 vertical borehole heat exchangers, each with a depth of 17 m, as a secondary heat source independent of the ambient temperature. The boreholes were plumbed in series after the collectors, allowing for regeneration of ground temperatures during the summer months; however significant seasonal storage was not maintained. The system was able to meet the demand of a 300 m$^2$ single-family home, operating with an annual $COP$ of 4, by providing 33.3 MWh annually. Of this, 24 MWh was provided by the collector array, while the boreholes supplied the remainder. While coupling the ISAHP to borehole heat exchangers improved the winter performance of the system, boreholes can significantly increase system costs, and are not ideally suited for retrofit installations. As Chandrashekar et al. found, unglazed collectors can lead to performance limitations in cold weather. If glazed collectors had been used by Bertram, there is a possibility that less investment into the boreholes would be required.

In 2011, a numerical feasibility study of two ISAHP systems was conducted by Sterling [19], based on the climatic conditions of Toronto, Ontario, Canada. Their performances were compared to a base electric heater and a traditional SDHW system. The first system, termed a dual-tank ISAHP, consisted of three fluid loops. The collector loop consisted of an unglazed collector through which a glycol/water solution was pumped directly into a 500 L thermal storage float tank. The heat transfer fluid was circulated back to the collectors from this tank. A heat exchanger associated with this float tank acted as the evaporator for a heat pump circuit, which transferred heat to a condensing heat exchanger associated with a second tank, which acted as the DHW storage. In parallel with the heat pump between the two tanks was a conventional heat
exchanger, though which fluid could be pumped if the temperature in the float tank was above the set point of the DHW tank. If the float tank was at a lower temperature, the heat pump would be used to transfer energy to the DHW tank. The float tank was stratified to ensure sufficient temperatures were available for the heat pump or heat exchanger, while providing cool fluid to the collector to increase its efficiency.

The second system was termed a solar-side ISAHP, and was very similar to a conventional SDHW system; however a heat pump circuit was connected between the inlet and outlet of the collector. Fluid exiting the collector would then pass through the condenser of the heat pump, resulting in a further increase in thermal energy. The fluid would then transfer this heat to the storage tank by passing it through an external heat exchanger. Once the fluid transferred its heat to the storage, it would pass through the evaporator of the heat pump. The fluid would then leave the evaporator at a lower temperature, resulting in higher collector efficiency. Like the first configuration, the system would be able to deliver heat without the need of the heat pump on days with adequate irradiance. Annual simulations of the systems were ranked by solar fraction (i.e., the percentage of the load met by collected solar energy). Values of 0.67, 0.66, and 0.58 were obtained for the dual tank ISAHP, the solar-side ISAHP and the conventional SDHW systems, respectively. While the dual tank system had the highest collected solar energy, it also had the most standby losses from the DHW thermal storage, as it was constantly maintained at an elevated temperature. Reducing the losses from the tank could further increase the solar fraction of the dual tank system above the others. Annual operating costs were calculated based on the electricity usage of each system, with a $0.075/kWh cost, and were $418.13, $181.55, $153.89 and $135.91 for the electric, SDHW, dual tank ISAHP and solar-side ISAHP systems, respectively. However, component costs were not tabulated; therefore the lifecycle cost of each system could not be determined.
2.2.2 Queen’s University Solar Calorimetry Laboratory Studies

The current study is a continuation of two previous Master’s thesis projects conducted at Queen’s University [9] [10]. The first thesis, conducted in 1997, consisted of a feasibility study to determine the performance of the ISAHP system described in Chapter 1 when compared to an air source heat pump and a conventional SDHW system, and included the effects of varying component sizes and collector types. In 2008, under the second thesis, a prototype of this system was constructed and tested under controlled laboratory conditions to validate and improve the numerical model. The updated model was then used to predict the annual performance of the system for five Canadian locations.

2.2.2.1 Feasibility Study

Freeman [9] conducted the feasibility study on the Queen’s ISAHP system by creating a system model in the TRNSYS simulation program [20]. As the standard heat pump component models in TRNSYS (referred to as component “Types”) could not provide sufficient detail for their use in an ISAHP system, Freeman created a steady-state model of a vapour compression cycle in the Engineering Equation Solver Program [21], which included detailed models of the 500 W reciprocating compressor, heat exchangers and thermal expansion valve. A description of these models is presented in Chapter 3. The EES model was used to create a performance map based on inlet temperatures and flow rates that could be read by a TRNSYS type for dynamic simulations of the entire system. Simulations were run using typical meteorological year (TMY) weather data for Toronto, Vancouver, Montreal, Winnipeg and Halifax, Canada, with a 239 L daily hot water load. The performance of the system using unglazed and glazed collectors was compared, and it was found that in most locations the unglazed collectors resulted in a higher free energy ratio, which was due to their ability to absorb more energy from the ambient air than the glazed collectors. Freeman found that the life-cycle cost of the system varied with location. In
Vancouver, the ISAHP showed savings of up to 29% over a SDHW system, and 46% over an electrical system. In Halifax, the savings were upwards of 18% and 39%, over the SDHW and electrical systems, respectively; while in Toronto, Montreal and Winnipeg, the ISAHP had similar life-cycle costs as a SDHW, and offered upwards of a 40% savings over the electrical systems.

2.2.2.2 Prototype Development

While Freeman had attempted to validate his simulations against manufactures’ data for the compressor performance, it was necessary to construct a prototype of the system to further validate and improve the model. Bridgeman [10] constructed and instrumented the heat pump and thermal storage loops, while the collector loop input was simulated by a glycol/water circuit with controllable electric heaters. Constant evaporator inlet temperature tests as well as simulated solar profile tests were performed, and the heat transfer rates, natural convection flow rate and heat pump COP were calculated and compared to the results from Freeman’s model. It was found that the simulations overestimated the experimental results by 12-15%, which was attributed to lower heat exchanger heat transfer coefficients (i.e., $UA$ values) than assumed by Freeman. Bridgeman utilized new empirical relationships for the heat exchangers and refined the model such that the simulations agreed within 5% of the experiments. Annual simulations were repeated, and the changed $UA$ values decreased the predicted annual performance. Using unglazed collectors as the heat source, the free energy ratios for the five Canadian cities were reported, with 56%, 52.7%, 55.6%, 55.8%, and 58.3% for Toronto, Winnipeg, Vancouver, Montreal, and Halifax, respectively. Bridgeman suggested conducting an annual test, in order to compare experimental and simulated results, and identify potential problems in the system that could occur during climatic extremes. Bridgeman predicted that such problems could be mitigated by modifying the collector type, compressor capacity or control scheme.
Chapter 3

Modeling of the Indirect SAHP and Hybrid Collector

3.1 Introduction

As mentioned in Chapter 2, a numerical model of the SAHP system was created in TRNSYS by Freeman [9] and refined by Bridgeman to give better agreement with preliminary experimental results [10]. This model was used in the current study, with modifications made to reflect the installed configuration of the experimental apparatus. Most notably, the collector model was changed to simulate the performance of the hybrid collector, as opposed to the unglazed collector used by Freeman and Bridgeman.

The system was modeled in TRNSYS [20], a transient simulation program. TRNSYS can be used to solve complex thermal systems by connecting the inputs and outputs of individual component models, or Types, much like the physical system. The behavior of the system can then be solved at discrete time steps, simulating the transient response of the system. The TRNSYS package includes an extensive library of component models, as well as the ability to incorporate custom components into the simulation.

3.2 Indirect SAHP Model

Freeman was unable to use the heat pump model from the TRNSYS library when modeling the ISAHP system, as the built-in component was unable to calculate the temperature change on the water side of the condenser, did not include heat transfer characteristics of the evaporator, and was insensitive to condenser temperature and system mass flow rates. Consequently, Freeman created a new component for the heat pump circuit by first developing a steady-state model in EES which simultaneously solved the equations of the compressor, evaporator, condenser and thermostatic expansion valve for various input conditions. A summary
of the EES model is presented in this section; however the reader is encouraged to refer to Freeman’s thesis [9] for an in-depth description. In the following section, temperatures and pressures are labeled according to their position, or state, in the cycle as indicated in Fig. 3-1.

![ISAHP schematic with numbered states](image)

3.2.1 Components

The rate of heat transfer through the evaporator heat exchanger was determined by calculating its effectiveness, which is dependent on the properties of the heat exchanger and the specific heats of the fluids. In heat exchangers where one fluid (e.g., the refrigerant) experiences a phase change, such as evaporation or condensation, that fluid effectively acts with infinite specific heat and the effectiveness, $\epsilon$, can be calculated by the simplified equation:

$$
\epsilon = 1 - \exp\left(\frac{-UA}{(\dot{m}C_p)_{min}}\right)
$$

(3-1)
where $UA$ is the product of the heat transfer coefficient and heat transfer area within the heat exchanger, and $(\dot{m}C_P)_{min}$ is the heat capacitance rate for the glycol solution, also termed $(\dot{m}C_P)_{gly}$. Freeman assumed a $UA$ value of 0.200 kW/°C. The effectiveness is defined as the ratio between the actual heat transfer through the device, $Q_{evap}$, and the maximum possible heat transfer that could occur, $Q_{max}$:

$$\varepsilon = \frac{Q_{evap}}{Q_{max}} \quad (3-2)$$

The maximum heat transfer is that which would occur if the temperature of the glycol entering the collector, $T_5$, was reduced to the temperature of the cold refrigerant entering on the opposite side, $T_4$, and is given as:

$$Q_{max} = (\dot{m}C_p)_{min}(T_5 - T_4) \quad (3-3)$$

Knowing the effectiveness and the maximum heat transfer, Freeman was able to calculate the evaporator heat transfer, and therefore the temperature of the refrigerant, $T_1$, and glycol, $T_6$, exiting the evaporator from an energy balance on the glycol and refrigerant sides of the evaporator:

$$Q_{evap} = (\dot{m}C_p)_{gly}(T_5 - T_6) = \dot{m}_{ref}(h_1 - h_4) \quad (3-4)$$

where $\dot{m}_{ref}$ is the refrigerant mass flow rate, and $h_1$ and $h_4$ are the refrigerant enthalpies at states 1 and 4. The work required by the reciprocating compressor was approximated by assuming the compression process followed a polytropic relationship. Assuming the compressor operation was isentropic with a constant specific heat ratio, $k$, the compressor work, $W_{comp, isen}$, was determined as a function of the inlet and outlet pressures, $P_1$ and $P_2$, given as:

$$W_{comp, isen} = N_{cyl} \left(\frac{k}{k-1}\right)P_1V_{act} \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}\right] \quad (3-5)$$
where \( N_{cy} \) is the number of cylinders, and \( V_{act} \) is the volume of gas drawn into the compressor in each cycle. Knowing the rotational speed of the compressor, the isentropic compressor power could be calculated. As the compression process is not isentropic, the actual compressor power was calculated by determining the electrical and mechanical efficiencies, \( \eta_{elec} \) and \( \eta_{mech} \), and the effects of fluid friction through the valves, \( \eta_{valve} \):

\[
\dot{W}_{comp} = \frac{W_{comp,isen}}{\eta_{elec}\eta_{mech}\eta_{valve}}
\]  

(3-6)

Values for these efficiencies were determined by Freeman [9] and Bridgeman [10] using manufacturers’ and experimental data. The condenser was modeled much the same as the evaporator, using the effectiveness method, and assuming the same \( UA \) value of 0.200 kW/°C. As in the evaporator, the refrigerant passing through the condenser experienced a phase change, and therefore acted with infinite specific heat. The effectiveness was therefore:

\[
\varepsilon = 1 - \exp \left( \frac{-UA}{(\dot{m}C_p)_{min}} \right) = \frac{Q_{cond}}{Q_{max}}
\]  

(3-7)

In this case, the maximum heat transfer was dependent on the temperature difference between the refrigerant entering the condenser at \( T_2 \), and the water entering at \( T_7 \). However, as a phase change was involved, the majority of the heat transfer occurred with the refrigerant at the condensate temperature, \( T_3 \). The average of the refrigerant inlet and outlet temperatures was used to evaluate \( Q_{max} \):

\[
Q_{max} = (\dot{m}C_p)_{water} \left( \frac{T_2 + T_3}{2} - T_7 \right)
\]  

(3-8)

Again, once the maximum possible heat transfer rate and the effectiveness were calculated, the actual heat transfer rate could be calculated by equation 3-9. The temperatures of the refrigerant, \( T_3 \), and water, \( T_8 \), exiting the condenser were then determined by an iterative procedure, knowing:
The final component in the heat pump circuit, the thermostatic expansion valve, was modeled as a constant enthalpy process, as no work occurs in the expansion of the refrigerant and negligible heat transfer occurs to the surroundings. As a result, this assumption leads to:

\[ Q_{\text{cond}} = (\dot{m}C_p)_{\text{water}}(T_B - T_7) = \dot{m}_{\text{ref}}(h_2 - h_3) \quad (3-9) \]

Additionally, the valve controlled the flow through the heat pump to ensure a 5°C superheat through the evaporator.

### 3.2.2 Implementation into TRNSYS

A parametric table was created by Bridgeman from the EES model for a range of \( T_5 \), \( T_7 \), and \( \dot{m}_{\text{water}} \) that would be experienced in typical operation. The table was read by the Type299 heat pump component in TRNSYS, which was supplied these inputs. Type299 then interpolated the appropriate results from the table, outputting evaporator and condenser outlet temperatures on the glycol and water sides, as well as the compressor power, heat pump COP and condenser heat transfer rate.

### 3.2.3 Validation and Refinement

Bridgeman constructed a prototype of the heat pump and storage tank to conduct controlled laboratory tests of the system using constant evaporator inlet temperatures and simulated solar input profiles. Bridgeman found that when experimental results were compared to simulations using Freeman’s model, the simulations over-predicted the results by 12-15%. By comparing the effectiveness of the heat exchangers between the experimental and simulated results, Bridgeman discovered that the \( UA = 0.200 \text{ kW/}^\circ\text{C} \) assumption was invalid. It was shown that for the condenser, the \( UA \) value was a function of \( T_7 \), given as:

\[ h_3 = h_4 \quad (3-10) \]
A simple relation was not found to represent the $UA$ value of the evaporator, due to varying glycol flow rates, evaporator superheat, and inlet temperatures. However, $UA$ values ranged between 0.10 kW/°C and 0.15 kW/°C, therefore a value of 0.12 kW/°C was assumed. Consequently, the simulated results were brought to within 5% of the experimental values [10].

### 3.3 Hybrid Collector Model

The collector used in the experimental set-up was a commercially available EnerWorks residential “Heat Safe”™ glazed collector. What distinguishes this collector from a standard glazed collector is its integral stagnation over-heat prevention. In a basic glazed collector, the absorber sheet is located within an insulated casing, with the absorber in contact with the rear sheet of insulation. In the Heat Safe collector, there exists an air channel between the absorber and the rear insulation that is open to the surroundings at the top and bottom of the collector to cool the absorber under stagnation conditions [22]. The opening at the top of the channel usually contains a thermally actuated valve to limit airflow through the channel under normal operating conditions; however in this study, the valve was removed. When a temperature difference between the absorber and the ambient temperature exists, air flows through the channel by natural convection, transferring heat to or from the absorber. Additionally, as the rear of the absorber and the surface of the insulation within the channel have high emmissivities, significant radiation exchange occurs between these two surfaces, which aids in the heat transfer process. Consequently, the hybrid collector has additional heat transfer mechanisms that a typical glazed flat plate collector does not, which must be considered when modeling its thermal performance.
3.3.1 Hottel, Whillier and Bliss Collector Efficiency

The efficiency of a solar collector is defined as the ratio of the useful energy collected, \( Q_U \), to the available solar irradiance:

\[
\eta = \frac{Q_U}{A_c G_T} = \frac{(mC_p)_{bly}(T_{IN} - T_{OUT})}{A_c G_T} \tag{3-12}
\]

While the efficiency, \( \eta \), can be calculated from data using equation 3-12, the performance of a solar collector can be predicted by the Hottel, Whillier and Bliss method [4], where the efficiency is a function of the collector’s optical properties, as well as the temperature difference between the collector and its surroundings:

\[
\eta = (\tau \alpha)_e - U_L \frac{(T_{PM} - T_A)}{G_T} \tag{3-13}
\]

where \( T_{IN} \) and \( T_{OUT} \) are the collector inlet and outlet temperatures, \( A_c \) is the gross collector area, \((\tau \alpha)_e\) is the effective transmissivity-absorptivity product for the collector cover and absorber sheet, \( U_L \) is the overall heat loss coefficient for the collector, \( T_{PM} \) and \( T_A \) are the mean absorber plate and ambient temperatures, respectively, and \( G_T \) is the incident solar radiation. Typically the absorber temperature is difficult to determine during operation. Therefore the efficiency is often expressed as a function of the inlet fluid temperature. As the fluid passing through the collector increases in temperature along its length, it is then necessary to incorporate a factor that relates the actual energy gain to the gain that would be achieved if the entire absorber was at the inlet temperature, known as the heat removal factor, \( F_R \). The efficiency is then expressed as:

\[
\eta = F_R (\tau \alpha)_e - F_R U_L \frac{(T_{IN} - T_A)}{G_T} \tag{3-14}
\]

As it is related to the temperature distribution in the collector, the heat removal factor is in turn dependent on the mass flow rate, heat loss coefficient and collector efficiency factor, \( F' \):

\[
F_R = \frac{mC_p}{A_c U_L} \left[ 1 - \exp \left( -\frac{A_c U_L F'}{mC_p} \right) \right] \tag{3-15}
\]
The collector efficiency factor represents the ratio between the heat loss factor from the fluid to the heat loss factor from the absorber, each to the ambient air:

\[
F' = \frac{1/U_L}{W \left( \frac{1}{U_LWF} + \frac{1}{\pi Dh_f} \right)}
\]  

(3-16)

where \( W \) is the width of the absorber between the fluid tubes, \( F \) is the fin efficiency along that width, \( D \) is the inner diameter of the tubes, and \( h_f \) is the convective heat transfer coefficient within the tubes.

In a SAHP application, where the heat pump could potentially cool the collector below the ambient temperature, such that the collector gains energy by convection as well as radiation, apparent efficiencies greater than 100% could be obtained. It should be noted that under such circumstances, the use of the term “efficiency” is improper. Therefore in this study, while having the same definition, values greater than 100% will be termed the Collector Performance Factor, \( CPF \), which indicates the relative contributions of radiation and convection to the energy gain.

The heat loss coefficient, \( U_L \), is composed of three parts: heat loss through the top, edges, and back of the collector. Each can be evaluated separately from theoretical analysis or experimental results. Harrison [23] created a thermal model for the performance of a standard glazed collector that calculated the useful energy output by evaluating the mean absorber plate temperature, heat loss coefficient and heat removal factor for given inlet and weather conditions, resulting in the calculation of the collector efficiency. This model, COLSIM, was used as the basis for the hybrid collector model, with adjustments made to the calculation of the back heat loss coefficient.

3.3.2 Top Loss Heat Transfer Coefficient

The thermal losses through the top of the hybrid collector (i.e., from the absorber through the glazing) are the same as those through a standard single-glazed collector. The dominant heat
transfer mechanisms are radiation and convection from the absorber to the cover and from the cover to the surroundings. As glass is used as the cover material, limited long-wave radiation from the absorber is transmitted to the surroundings. A thermal resistance network is shown in Fig. 3-2 for the top loss components, as used by Harrison.

![Thermal network for collector top heat loss](image)

**Fig. 3-2: Thermal network for collector top heat loss [23]**

The terms \( h_{cv,c-a} \) and \( h_{cv,p-c} \) represent the convective heat transfer coefficients from the cover to the surroundings and the absorber to the cover, respectively, while \( h_{r,c-a} \), \( h_{r,p-c} \), and \( h_{r,p-a} \) represent the radiative heat transfer coefficients from the cover to the sky, absorber to the cover, and absorber to the sky. Again, the reader is encouraged to refer to Harrison’s work for an in-depth discussion of the calculation of these values, while a summary of the governing equations used to determine these parameters is presented here.
The radiative heat transfer from the absorber to the cover, \( Q_{\text{rad},p-c} \), was approximated using the infinite parallel planes relationship [24] as the distance between the planes is small compared to their equal areas. The relationship is given as:

\[
Q_{\text{rad},p-c} = A_c \sigma (T_{pm}^4 - T_c^4) \frac{1}{1/\varepsilon_p + 1/\varepsilon_c - 1}
\]

(3-17)

where \( \sigma \) is the Stephan-Boltzman constant, \( T_c \) is the cover temperature, \( \varepsilon_p \) is the absorber plate emissivity, and \( \varepsilon_c \) is the cover emissivity. A radiative heat transfer coefficient can be defined analogously to a convection coefficient:

\[
h_{r,p-c} = \frac{Q_{\text{rad},p-c}}{A_c (T_{pm} - T_c)}
\]

(3-18)

which can be reduced to:

\[
h_{r,p-c} = \frac{\sigma (T_{pm}^2 + T_c^2)(T_{pm} + T_c)}{1/\varepsilon_p + 1/\varepsilon_c - 1}
\]

(3-19)

Besides the thermal radiation, either of two heat transfer mechanisms can occur within the air between the plate and the absorber, depending on their relative temperatures. If the absorber temperature is sufficiently greater than the cover temperature, buoyancy forces within the heated air overcome viscous effects, and a recirculatory convection condition develops. Otherwise, if the temperature difference is small or negative, heat transfer occurs by conduction through the still air. A Nusselt number relationship as a function of Rayleigh number for such an enclosure was developed by Hollands [25] for tilt angles, \( \theta \), between 0° and 75° from horizontal:

\[
Nu_L = 1 + 1.44 \left( 1 - \frac{1708}{Ra_L \cos \theta} \right)^+ \left( 1 - \frac{1708 (\sin 1.8\theta)^{1.6}}{Ra_L \cos \theta} \right) + \left( \frac{Ra_L \cos \theta}{5830} \right)^{1/3} - 1
\]

(3-20)

where the Rayleigh number, \( Ra_L \), was calculated as:
\[ Ra_L = \frac{g \beta (T_{PM} - T_C) L_{PC}^3}{\nu^2} Pr \]  

(3-21)

where \( g \) is the acceleration due to gravity and \( L_{PC} \) is the characteristic length (i.e. the spacing between the absorber and the cover). The expansion coefficient, \( \beta \), the kinematic viscosity, \( \nu \), and the Prandtl number, \( Pr \), were evaluated at a mean temperature between \( T_{PM} \) and \( T_C \).

The ‘+’ superscripts indicate that the bracketed term it accompanies only retains its value if it is positive. Otherwise the term equals zero. As equation 3-20 shows, if the \( Ra_c \cos \theta \) product is negative, or does not surpass the critical value of 1708, then the Nusselt number equals one, representing the pure conduction condition. The convection coefficient can be calculated from the definition of the Nusselt number:

\[ h_{cv,p-c} = \frac{Nu_L k_A}{L_{PC}} \]  

(3-22)

The heat loss coefficient from the absorber to the cover is then:

\[ U_{p-c} = h_{r,p-c} + h_{cv,p-c} \]  

(3-23)

The radiative heat transfer coefficient from the cover to the surroundings was calculated assuming that the surroundings act as a black body at an effective sky temperature, which depends on the ambient temperature and ground temperature. The radiative heat transfer rate was calculated as:

\[ Q_{rad,c-a} = A_c \varepsilon_c \sigma (T_C^4 - T_S^4) \]  

(3-24)

And the radiative coefficient is:

\[ h_{r,c-a} = \frac{\varepsilon_c \sigma (T_C^4 - T_S^4)}{(T_C - T_A)} \]  

(3-25)

The convective heat transfer coefficient from the cover to the surroundings is due to wind blowing over the collector. While a number of correlations exist that relate wind speed to the convection coefficient for flat plate collectors at various angles, Harrison did not find a
conclusive relationship [23], and therefore evaluated the collector performance in terms of the convection coefficient as opposed to wind speed. Values were chosen such that:

\[ 5 \text{ W/m}^2\text{K} \leq h_{cv,c-a} \leq 40 \text{ W/m}^2\text{K} \]  

(3-26)

For this study, a value of 20 W/m\(^2\)K was used, according to the measured values typically used in standard CSA rating tests [26]. The resulting heat loss coefficient from the cover to the surroundings was:

\[ U_{C-A} = h_{r,c-a} + h_{cv,c-a} \]  

(3-27)

The coefficient for the radiation from the absorber plate to the surroundings was calculated much the same as for the radiation from the cover to the surroundings, except with the addition of the transmissivity of the glass to long-wave radiation:

\[ h_{r,p-a} = \frac{\tau_{lw} \epsilon P \sigma (T_{PM}^4 - T_S^4)}{(T_{PM} - T_A)} \]  

(3-28)

The resulting overall top loss heat transfer coefficient was therefore:

\[ U_{top} = \left( \frac{1}{U_{p-c}} + \frac{1}{U_{C-A}} \right)^{-1} + h_{r,p-a} \]  

(3-29)

### 3.3.3 Back Loss Heat Transfer Coefficient

The back loss resistive circuit differs from a standard glazed collector, in that there are four temperature nodes (the absorber plate, \(T_{PM}\), the insulation face within the channel, \(T_{IM}\), the back of the insulation, \(T_B\), and the surroundings, \(T_A\)) while a standard collector can be modeled with two nodes. The dominant heat transfer mechanisms in the rear channel are convection from the absorber to the channel air, radiation exchange between the absorber and the insulation, convection from the insulation to the channel air, and conduction losses through the rear of the insulation. The thermal resistive circuit is shown in Fig. 3-3.
Fig. 3-3: Thermal network for collector back heat loss

The convection from the channel surfaces (i.e., the absorber and the insulation planes) is represented by $h_{cv,p-a}$ and $h_{cv,i-a}$, while $h_{r,p-i}$ represents the radiative heat transfer from the absorber to the insulation. $U_{c,i-b}$ and $h_{cv,b-a}$ represent the heat losses through and off the rear of the insulation.

A number of studies have investigated asymmetrically heated inclined channels [27] [28] [29], and it has been shown that, depending on the channel aspect ratio, the natural convective heat transfer can be represented by an equation of the form:
\[ \text{Nu}_S = A (Ra_S \cos \theta)^b \] (3-30)

where the characteristic length for the Nusselt and Rayleigh numbers was the channel spacing, \( S \).

The Rayleigh number was evaluated similarly to equation 3-21, with the exception that the temperature difference was calculated as between the average wall temperature and the surroundings temperature.

Azevedo and Sparrow [27] conducted experiments on an inclined channel under three heating modes (i.e.: both walls heated {I}; top wall heated with bottom wall unheated {II}; and top wall unheated with bottom wall heated {III}), to find the values for \( A \) and \( b \) in equation 3-30. For mode II, which is the closest representation of the heated collector channel, the results fit to:

\[ \text{Nu}_S = 0.657 \left( \frac{S}{H} \right) Ra_S \cos \theta^{0.247} \] (3-31)

Similar studies were conducted independently by Manca [28] and Lin [29] which included the effects of radiation exchange between the channel surfaces. These authors each presented their results in terms of modified Rayleigh numbers, based on the heat flux, \( q'' \), from the heated top surface of the channel:

\[ Ra'' = \frac{g \beta q'' S^5}{\nu^2 k L} Pr \] (3-32)

where \( k \) is the thermal conductivity of the air in the channel. Manca determined Nusselt number relations as a function of the modified Rayleigh number based on the convective flux and the combination of convective and radiative fluxes, for channel aspect ratios of \( 10 \leq L/S \leq 32 \) and surface emissivities of 0.8. The correlation for convection only was:

\[ \text{Nu}_S = 0.34 (Ra'' \cos \theta)^{0.25} \] (3-33)

and the correlation for the combined convection and radiation was:
\[ \text{Nu}_S = 0.27 (Ra'' \cos \theta)^{0.26} \]  

(3-34)

The relationship determined by Lin for the average Nusselt number for combined convection and radiation in the channel at a tilt angle of 18°, emissivities of 0.95, and $44 \leq \text{L/S} \leq 220$ was:

\[ \text{Nu}_S = 0.5541 \text{Ra}''^{0.2708} \]  

(3-35)

While these correlations indicate the range in which the Nusselt number falls, neither was chosen to be used in the model, due to the geometry of the channel, with 90° bends at the inlet and outlet. The presence of the serpentine heat transfer tube on the absorber sheet also reduces the cross-sectional area of the channel, increasing the aspect ratio above L/S=82. It was also desirable to have separate relations for convection and radiation. Additionally, when the collector was operating below the ambient temperature, the natural convection occurred in an inclined channel cooled from the top surface, for which information in the literature was limited.

An ideal correlation would represent the Nusselt number in terms of a Rayleigh number based upon $T_{PM}$, such that it would integrate nicely with the COLSIM model. Consequently, experiments were conducted on a modified EnerWorks Heat Safe collector as part of this study to determine appropriate correlations for above and below-ambient operation, the results of which are presented in Chapter 5. These results were used to determine $h_{\text{cv,p}}$.

The coefficient for radiative heat transfer between the absorber and the insulation planes, $h_{r,p-i}$, was determined in a similar fashion to $h_{r,p-c}$ in the top loss calculation:

\[ h_{r,p-i} = \frac{Q_{\text{rad.p-i}}}{A_c(T_{PM} - T_{IM})} \]  

(3-36)

However, where the radiation exchange between the absorber and the cover was approximated by the infinite parallel plates method, temperature profiles obtained through the
experiments presented in Chapter 5 allowed for $Q_{rad,p-i}$ to be calculated using Hottel’s zone method [24], and view-factors determined using the crossed-strings method [30].

The separate determination of the remaining heat transfer coefficients, $h_{cv,i-a}$, $U_{c,i-b}$ and $h_{cv,b-a}$ was unnecessary, as it can be seen in Fig. 3-3 that these coefficients make up the resistances of two parallel paths between $T_{IM}$ and $T_A$. Therefore, these were combined into a single loss coefficient, $U_{i-a}$, representing the heat transfer from the insulation plane to the surroundings:

$$U_{i-a} = h_{cv,i-a} + \left( \frac{1}{U_{c,i-b}} + \frac{1}{h_{cv,b-a}} \right)^{-1}$$ (3-37)

As a result, only the determination of $U_{i-a}$ was necessary. Considering an energy balance at the insulation surface as shown in Fig. 3-4 it may be deduced that the radiative heat transfer from the absorber to the insulation must be dissipated by other heat transfer modes (i.e., convection off the surface of the insulation, and conduction out the rear of the insulation) to maintain a net heat transfer of zero [31].

![Fig. 3-4: Energy balance on the insulation plane](image)

Therefore the heat transfer at the insulation surface can be expressed as:

$$Q_{rad,p-i} = Q_{conv,i-a} + Q_{loss,i-a} = U_{i-a}A_c(T_{IM} - T_A)$$ (3-38)

or, rearranged:
The value for $U_{i-a}$ was also determined from the experimental results presented in Chapter 5. With the individual components determined, the back heat loss coefficient was calculated:

$$U_{back} = h_{ev,p-a} + \left( \frac{1}{h_{r,p-i}} + \frac{1}{U_{i-a}} \right)^{-1} \quad (3-40)$$

To model the collector with a closed channel, a similar analysis was used with the exception that the $Q_{conv,p-a}$ and $Q_{conv,i-a}$ terms would become zero.

### 3.3.4 Edge Loss Heat Transfer Coefficient

The edge of the collector is generally modeled as a one-resistance circuit, where the heat loss is a function of the edge heat loss coefficient, $U_e$, the edge area, $A_e$, and the temperature difference between the absorber and surroundings:

$$Q_{edge} = U_e A_e (T_{PM} - T_A) \quad (3-41)$$

An approximation for the value of $U_e$, where the heat transfer resistance is dominated by conduction through the edge insulation, is:

$$U_e = \frac{k_{ins}}{L_{ins}} \quad (3-42)$$

where $k_{ins}$ is the insulation conductivity and $L_{ins}$ is the insulation thickness. For a $k_{ins}$ of 0.02 W/m-K and a $L_{ins}$ of 0.019 m, the edge loss coefficient equals 1.05 W/m²-K [32]. To make this relationship harmonious with the top and back loss coefficients, which are defined based on $A_C$, the edge loss coefficient was converted to a value based on $A_C$:

$$U_{edge} = \frac{U_e A_e}{A_c} \quad (3-43)$$

The resulting $U_{edge}$ value was then 0.2 W/m²-K, for an edge area of 0.53 m². However, it has been shown by Simko et al. that using this method can lead to underestimations of the overall
heat loss, due to the neglect of edge effects in the top and back heat loss calculations. In their study, Simko et al. calculated an overall heat loss coefficient by the conventional method of 4.3 W/m²·K, and a value of 6.1 W/m²·K when accounting for the extra edge effects, while experiments resulted in a value of 6.2 W/m²·K. From the difference in these results, an effective $U_{edge}$ of 1.8 W/m²·K was chosen to represent the increased edge loss effect.

3.3.5 Overall Heat Loss Coefficient

The overall heat loss coefficient for the hybrid collector is the sum of the top, back and edge heat losses. It is clear that as the different heat loss modes are individually functions of the temperature difference between $T_{PM}$ and $T_A$, so must the overall collector heat loss coefficient be dependent on the temperature difference. Often, the overall heat loss coefficient can be approximated as a linear function of the temperature difference, given as:

$$U_L = A + B(T_{IN} - T_A)$$

and the collector efficiency is therefore modified to become [33]:

$$\eta = a_0 - a_1 \frac{(T_{IN} - T_A)}{G_T} - a_2 \frac{(T_{IN} - T_A)^2}{G_T}$$

(3-45)

The efficiency output from COLSIM, determined over a range of inlet and ambient temperatures and irradiance values, was used to determine a best fit for Eq. 3-45, resulting in a characteristic equation for the collector.

3.3.6 Model Computation

The calculation of the collector efficiency in COLSIM for given inputs of $T_{IN}$, $T_A$, and $G_T$ is not a straight-forward process, as the performance is actually dependent on $T_{PM}$, $T_{IM}$, $T_C$, and the mean fluid temperature, $T_{FM}$. These temperatures are not initially known, therefore an iterative solution was necessary. Initial guesses for $T_{PM}$ and $T_{FM}$ were set to ($T_{IN} + 2$) and $T_{IN}$. $U_L$ and $F_R$
estimates were then calculated, from which the useful energy gain was calculated. A new $T_{FM}$ was calculated based upon [4]:

$$T_{FM,new} = T_{IN} + \frac{Q_U}{A_P} \left( 1 - \frac{F_R}{F'} \right)$$  \hspace{1cm} (3-46)

where $A_P$ is the aperture area of the collector. The new $T_{FM}$ was then used to recalculate the $F_R$, and was iterated until a convergence criterion is met. The converged $T_{FM}$ is then used to calculate a new $T_{PM}$ by:

$$T_{PM,new} = T_{FM} + \frac{Q_U}{h_r W_{per} L_{tube}}$$  \hspace{1cm} (3-47)

where $W_{per}$ and $L_{tube}$ are the wetted perimeter and the length of the heat transfer tube. The entire process was then repeated until both $T_{FM}$ and $T_{PM}$ converged, and the efficiency was calculated.

For each iteration when $U_L$ was calculated, the calculations of $U_{top}$ and $U_{back}$ also required an iterative process to calculate $T_C$ and $T_{IM}$, where the new $T_C$ was calculated from:

$$T_{C,new} = T_{PM} - \frac{U_{top}(T_{PM} - T_A)}{U_{P-C}}$$  \hspace{1cm} (3-48)

and the new $T_{IM}$ was:

$$T_{IM,new} = T_{PM} - \left( \frac{1}{h_{r,p-i}} + \frac{1}{U_{l-a}} \right)^{-1} (T_{PM} - T_A)$$  \hspace{1cm} (3-49)

Flow charts that describe the iterative procedure for the COLSIM program and the UTOP and UBACK sub-procedures are included in Appendix A, while the COLSIM code is included in Appendix B.
3.3.7 Implementation into TRNSYS

Type 1 in the TRNSYS library represents a flat plate solar collector with a second order efficiency, as described by equation 3-45. The required parameters for Type 1 include the coefficients $a_0$, $a_1$, and $a_2$, which can be obtained from manufactures’ ratings for commercially available collectors or the best-fit to the COLSIM output for the hybrid collector. With these parameters, Type 1 calculates the output temperature and the useful energy gain of the collector, based on the inputted weather and operating conditions. The Type is able to adjust the efficiency curve to account for operating at flow rates that differ from the test conditions used to determine the performance parameters, as well as, off-normal incident radiation. Two collector Types were included in the TRNSYS model: one to calculate the performance if the collector was operating above ambient temperature, and the other to calculate the below-ambient performance. A control switch passed on the appropriate outputs, depending on the operational condition.

The weather inputs for the annual simulations are supplied by Type 109, which reads an external weather file of Typical Meteorological Year (TMY) data [34]. Based on this external file, Type 109 can calculate the total and diffuse radiation on a tilted surface, using a user-defined sky radiation model. While Type 109 is able to use a number of radiation models, the Perez model is generally considered the most appropriate [35]. Type 109 can also be used to supply other weather conditions, such as the ambient temperature.

3.4 Final TRNSYS Model

The collector, heat pump, and weather Types alone were insufficient to model the ISAHP performance. The balance of the system must also be considered, which included the hot water storage tank, the interconnecting pipes, and the collector pump. Additionally, as the water flows through the condenser and storage tank by natural convection, a flow rate calculator was required.
to determine this flow rate according to the pressure difference across the condenser. These components are summarized below. An overview of the TRNSYS layout is in Appendix C.

3.4.1 Other Components

3.4.1.1 Storage Tank

Type 60 represents a multiple inlet and outlet liquid storage that experiences thermal stratification [20]. The modeling of the thermal stratification is achieved by separating the tank into N fully mixed equal volume segments along the height of the tank, as shown in Fig. 3-5. The user inputs the tank volume and height, as well as, the desired number of nodes (maximum = 100 nodes). Parameters also include the height of the inlets and outlets, and the model calculates from which node these ports will deliver or draw water. Additionally, the height, power and temperature set-point of auxiliary heaters can be specified, along with an average tank loss coefficient, $U$, and destratification conductivity, $\Delta k$. An energy balance, described schematically in Fig. 3-6, accounts for the flow and heat transfer between nodes, flow in and out of the ports, energy in from the auxiliary heaters, and losses to the environment.
Fig. 3-5: Schematic for the storage tank model and natural convection circuit heights

Fig. 3-6: Energy balance on a storage tank node, [20]
The temperature distribution in the tank was calculated by simultaneously solving N differential equations representing the energy balance of each node, expressed as:

\[
(m_i c_p) \frac{dT_{tank,i}}{dt} = \frac{(k + \Delta k)A_X}{\Delta x_{i+1-i}} (T_{tank,i+1} - T_{tank,i}) + \frac{(k + \Delta k)A_X}{\Delta x_{i-1-i}} (T_{tank,i-1} - T_{tank,i}) - UA(T_{tank,i} - T_{env}) + \dot{m}_{down} c_p(T_{tank,i-1}) - \dot{m}_{up} c_p(T_{tank,i}) - \dot{m}_{down} c_p(T_{tank,i}) + \dot{m}_{up} c_p(T_{tank,i+1}) + \dot{m}_{in,1} c_p(T_{in,1}) - \dot{m}_{out,1} c_p(T_{tank,i}) + \dot{m}_{in,2} c_p(T_{in,2}) - \dot{m}_{out,2} c_p(T_{tank,i}) + Q_{aux} \tag{3-50}
\]

where \( m_i \) is the mass of water in the node, \( k \) is the thermal conductivity of the water, \( A_X \) is the cross-sectional area, \( A \) is the surface area, \( \dot{m}_{down} \) and \( \dot{m}_{up} \) are the mass flow rates in the downwards and upwards directions of the tank, \( \dot{m}_{in,1}, \dot{m}_{in,2}, \dot{m}_{out,1}, \) and \( \dot{m}_{out,2} \) are the mass flow rates through the two inlets and two outlets, and \( Q_{aux} \) is the power supplied by the auxiliary heater. \( T_{tank,i+1}, T_{tank,i}, T_{tank,i-1}, T_{in,1}, T_{in,2} \) and \( T_{env} \) are the temperatures above, in, and below the node, entering the inlets 1 and 2, and surrounding the tank, respectively. While any of the terms in equation 3-50 can equal zero depending on the specific node location and operating conditions, the \( \dot{m}_{down} \) or \( \dot{m}_{up} \) terms can be non-zero at any time step, but not in the same time step.

The volume of the tank was set to 270 L, while the height was set as 1.33 m [36]. Inlet and outlet 1 were located at the top of the tank, while inlet and outlet 2 were located at the bottom. 50 nodes were chosen to accurately represent the stratification in the tank [37], while the destratification conductivity was calculated according the recommendations in the TRNSYS documentation [20] as:
\[ \Delta k = k_{\text{wall}} \frac{A_{X,\text{wall}}}{A_X} \]  

(3-51)

where \( k_{\text{wall}} \) is the thermal conductivity of the wall material, and \( A_{X,\text{wall}} \) is the cross-sectional area of the wall. The resulting value was set as 1.58 W/m-K. The value of \( U \) was calculated as 4.132 kJ/hr-m\(^2\)-K (1.15 W/m\(^2\)-K), from a cool-down test similar to that described in the SRCC TM-1 document [38], shown in Appendix D. The auxiliary heater in the tank was not used, as it would disturb the stratification of the tank above the element. Consequently, a calculator was included which determined the required energy to heat the water to the load set-point for a given flow rate and tank outlet temperature.

3.4.1.2 Pipes

The collector, heat pump, and storage tank Types were connected with Type 31 components, which calculate the energy losses of the fluid travelling through the connecting pipes. Four pipe segments were included between the collector and the heat pump (i.e., the outdoor and indoor sections of the supply and return lines). Bridgeman did not include the indoor sections of the pipes; however they were added to the model, as they allow for the standby temperature of the evaporator to be determined. One pipe length connected the outlet of the condenser to the top of the storage tank to account for any losses in the natural convection loop. As the condenser was connected directly to the outlet at the tank bottom, a pipe segment from the tank to the condenser inlet was unnecessary.

Type 31 calculates the thermal losses by assuming “plug-flow” through the pipe [20]; variable size segments represent the fluid that enters the pipe in each time-step. The mass of an entering segment was calculated as the flow rate multiplied by the time-step. Any segments that were currently in the pipe were pushed out by the incoming flow, conserving the mass flow
through the pipe section. The energy losses from the $j$th element were found from the solution of the differential equation:

$$M_j C_p \frac{dT_j}{dt} = -(UA)_j(T_j - T_{env})$$

(3-52)

And the outlet temperature, $T_O$, was calculated from a mass weighed average of the leaving segments:

$$T_O = \frac{1}{m\Delta t} \left( \sum_{j=1}^{k-1} M_j T_j + aM_k T_k \right)$$

(3-53)

where $a$ is the fraction of the last segment, $k$, that is pushed out of the pipe in the time-step.

### 3.4.1.3 Pump

Type 3b was chosen as a single speed pump for the collector loop. A single speed pump is a good approximation for a positive displacement pump, which tend to have mass flow rates that are insensitive to system pressure differences at a given rotational speed. The parameters for the Type allowed for the flow rate to be set by the user, while the power consumption could be determined from specifications of the specific pump in use.

### 3.4.1.4 Natural Convection Flow Rate Calculator

A custom made type, Type 297, was included to calculate the natural convection flow rate as a function of the driving pressure difference across the condenser, based on the work by Purdy [39] and Lin [40]. The natural convection circuit can be considered as two columns of water of equal height: one through the storage tank, and one through the condenser and the pipe extending to the top of the tank, as shown in Fig. 3-5. Depending on the temperature state within these columns, a hydrostatic pressure difference, $\Delta P_{hydrostatic}$, occurs due to the buoyancy difference between the columns, driving the natural convection flow [39]. The hydrostatic pressure difference was calculated by:
\[ \Delta P_{\text{hydrostatic}} = \rho_{\text{tank}} g H_{\text{tank}} - \rho_{\text{pipe}} g (H_{\text{tank}} - H_{\text{HX}}) - \rho_{\text{HX}} g H_{\text{HX}} \]  
(3-54)

where \( H_{\text{tank}} \) and \( H_{\text{HX}} \) are the heights of the tank and condenser heat exchanger, respectively, while \( \rho_{\text{tank}} \), \( \rho_{\text{HX}} \), and \( \rho_{\text{pipe}} \) are the densities of the water in the tank, heat exchanger and pipe. The densities were calculated from the approximation:

\[ \rho = 1000.3 - 0.00359 T^2 - 0.067 T \]
(3-55)

using the average tank temperature, average condenser temperature, and the condenser exit temperature for the tank, heat exchanger, and pipe calculations, respectively. The study by Lin [40] showed that the flow rate relationship is of the form:

\[ \dot{m}_{\text{water}} = a (\Delta P_{\text{hydrostatic}})^b \]
(3-56)

Bridgeman determined the values for \( a \) and \( b \) to be 5.5442 kg/hr-Pa and 0.4633, from experiments on the ISAHP prototype [10]. However, as the system had since been sitting unused for some time, fouling of the heat exchanger could have occurred, which would decrease the flow rate for a given pressure difference. Consequently, a new fit was determined from experimental results, shown in Fig. 3-7. The relationship shown was found to represent all tests days.

### 3.4.1.5 Mixing Valve

It is possible for the temperature of water entering the tank from the condenser to exceed the desired set-point of 51.7°C. As a result, Type 178 was included on the outlet of the storage tank, which, if the water leaving the storage tank at \( T_{\text{tank,1}} \) exceeds 51.7°C, would mix cold mains water at \( T_{\text{mains}} \) with the water drawn from the tank, according to the energy balance:

\[ \dot{m}_{\text{load}} C_p(55^\circ \text{C}) = \dot{m}_{\text{tank}} C_p(T_{\text{tank,1}}) + \dot{m}_{\text{divert}} C_p(T_{\text{mains}}) \]
(3-57)

and according to the mass balance:
\[ \dot{m}_{\text{load}} = \dot{m}_{\text{tank}} + \dot{m}_{\text{divert}} \]  

(3-58)

where \( \dot{m}_{\text{load}}, \dot{m}_{\text{tank}}, \) and \( \dot{m}_{\text{divert}} \) are the flow rates to the load, from the hot tank, and diverted from the cold mains.

**Fig. 3-7:** Experimental relationship for tank-side natural convection flow rate versus driving pressure difference, October 27th, 2010 data

### 3.4.1.6 Forcing Functions

Type 14 represents a time-dependent forcing function that was used in three situations in the TRNSYS model: first, it was used to specify the mains water temperature at a given point of the year, based on collected data; second, it was used to define the water draw profile, based on the CSA F379-M1982 standard [41], with a total draw of 225 L; third, it was used as a time-based controller to prevent the system from operating outside of the 06:00 – 20:00 hour time frame. The Type operates by outputting a user-defined value depending on the simulation time. If the simulation time exceeds the time range specified in the Type, the function repeats itself, allowing daily patterns repeat every day for a year-long simulation.
3.4.1.7 Controls

The control scheme used by Freeman [9] and Bridgeman [10] was simplified to a more appropriate set-up for the installed system. The time-based controller, described above, and an absolute temperature controller was retained, but the differential temperature controller was removed. The differential controller compared the collector output temperature to the temperature of the glycol leaving the evaporator, and would turn the pump on if the temperature difference more than 5°C. While this set-up worked in previous simulations, it would not work in a practical installation, when the collector temperature may be below the temperature of the room in which the evaporator is located. Depending on the location of sensors, there would occur periods when the heat pump would be operating without the glycol circulating. The performance in such a situation would be decreased due to the lack of heat input, and the evaporator temperature could decrease below the compressor limits.

The remaining controls for the base system were the time-based controller, that limited operation to between 06:00 – 20:00 hrs and prevented harmful compressor cycling during the overnight periods. The absolute temperature controller Type 296, allowed the compressor and glycol pump to run if the system was within limits (i.e., the collector output was above -5°C and the water entering the condenser was below 30°C).

3.5 Performance Indices

The following indices were chosen to indicate how well the system’s components were operating together and individually. These indices were also used to compare the performance of the ISAHP for different configurations.

3.5.1 Collector Efficiency and Collector Performance Factor

The collector efficiency is important in that is indicates how much of the available solar resource is being absorbed, and gives a point of comparison for different collector types operating
under the same environmental conditions. Efficiency values less than the optical efficiency of the collector, \( (\tau_0) \), indicated the collector was losing heat to the environment, while efficiency or \( CPF \) values greater than this signified that the collector was absorbing heat from the surroundings.

3.5.2 Heat Pump \( COP \)

While high collector efficiencies were important, high efficiencies at the cost of reduced collector outlet temperatures could have had a negative effect on the heat pump performance. As mentioned in Chapter 1, the \( COP \) of a heat pump declines with decreasing evaporator temperatures. Achieving a high \( COP \), as well as a high collector efficiency was important to maximize the heat delivered to the load, and to minimize the electrical costs.

3.5.3 Heat Transfer Rates

While certain conditions, such as overcast and warm days, may lead to high collector efficiencies, they may not necessarily result in a high system output. Therefore, knowing the heat transfer rates through the evaporator and condenser, as described by equations 3-4 and 3-9, gave an absolute indication of the power that the system is providing to the hot water.

3.5.4 Free Energy Ratio

The free energy ratio represents the fraction of the load energy requirements that is supplied by non-purchased sources (i.e., the useful gain of the collector from the solar irradiance and ambient air). The total energy delivered to the load is the increase in thermal energy of the water exiting the tank compared to the water entering the tank. Of this increase, a portion was attributed to the electrical work required to run the heat pump, and the auxiliary power supplied by the electrical heaters in the tank. The free energy ratio, \( FER \), was expressed as:
In effect, this indicates the energy savings obtained by the system when compared to a base electrical domestic hot water system. While all these indices are related, and performance is best when all are increased, the \( FER \) is the best index for comparing the overall performance of different configurations over a given time scale.

\[
FER = \frac{\int Q_{tank,\text{out}} \, dt - \int Q_{tank,\text{in}} \, dt - \int Q_{aux} \, dt - \int W \, dt}{\int Q_{tank,\text{out}} \, dt - \int Q_{tank,\text{in}} \, dt}
\]  

(3-59)
Chapter 4

Experimental Description

4.1 Introduction

The ISAHP prototype developed by Bridgeman [10] for controlled laboratory tests was modified to determine its performance under actual environmental conditions. The heat pump and storage tank loops were removed from the auxiliary heaters used to simulate solar inputs, and were connected to a modified EnerWorks “Heat Safe”™ collector which acted as a hybrid heat source, located on a south-facing exterior wall of the Queen’s University Solar Calorimetry Laboratory in Kingston, Ontario.

4.2 Apparatus and Instrumentation

Two separate experimental apparatus were used in the current study. The primary apparatus was the prototype ISAHP connected to the hybrid collector, which was used to determine the system performance. Additional experiments were conducted on a separate, instrumented collector to help determine the heat loss characteristics of the rear cooling channel and to aid in the modeling of the collector performance.

4.2.1 Stagnation Prevention Apparatus

An EnerWorks “Heat Safe”™ collector was acquired and the glazing and absorber sheet were removed. A second absorber sheet was prepared for controlled tests by adhering fifteen thin, flexible electric heating strips (HCW 200 W flexible rubber heaters) to the top surface of the absorber using a thin layer of high temperature silicone adhesive. The heaters were wired in parallel and connected to a Sorensen XFR 150V-8A DC power supply. Between the absorber and the heaters, and array of fifteen 30-gauge T-type thermocouples was positioned according to Fig.
4-1. These thermocouples were sandwiched between two pieces of Kapton® tape to electrically isolate them from the heaters and the absorber sheet.

Fig. 4-1: Collector channel thermocouple layout

The collector was reassembled with the new, instrumented absorber; however the glazing was not reattached. Instead, a layer of fiberglass insulation and a sheet of two-inch isocyanurate rigid insulation (R = 2.16 m²-K/W) were strapped to the top of the absorber to limit heat loss from the top of the collector. The rear of the absorber was painted black (ε=0.95), which is necessary for the stagnation overheat prevention [22].
A similar array of eleven thermocouples was attached to the surface of the rear insulation in the cooling channel. To ensure that the surface temperature of the insulation was correctly measured, each thermocouple was fastened between two small copper tabs, and adhered to the insulation. The thermocouple wire was then fed towards the edge through the insulation to minimize the thermal gradient near the junction. Finally the copper tabs were painted black to match the insulation. Fig. 4-2 (a) and (b) show details of this process. With the thermocouples installed on the insulation, the collector was reassembled and installed on a mounting rack at an inclination of 45°.

Fig. 4-2: Thermocouple attached to insulation with copper tabs (a) and painted black (b), shown by red circles.

An additional thermocouple was included to measure the ambient temperature around the collector. While the heaters were included to provide power to simulate solar input, it was also necessary to cool the collector to simulate operation below ambient temperature. Consequently, a
fluid loop consisting of two VWR chillers and a DC-powered positive displacement pump was connected to the serpentine heat transfer tube on the absorber via insulated flexible vinyl tubing. A chilled mixture of propylene glycol and water could then be pumped through the collector, as the chillers could be set to maintain a desired inlet temperature using their built-in PID control. The fluid temperature rise through the collector was measured by 30-gauge T-type thermocouple wells submersed in the fluid stream at the inlet and outlet of the collector, using custom thermocouple wells as shown in Fig. 4-3. The wells were constructed of a 1/2 in. brass flared elbow, with a 1/8 in. brass tube inserted and soldered into a drilled hole. The flow rate through the collector was determined by diverting the fluid return into a 1 L graduated cylinder, and recording the time necessary to fill it. The thermocouples’ voltage output was read and recorded by a HP 3497A Data Acquisition (D/A) unit, with built in cold junction compensation, connected to a personal computer running Windows XP and LabView ver. 7.1. A schematic of the entire system is given in Fig. 4-4.

![Thermocouple well](image)

**Fig. 4-3: Thermocouple well**
Fig. 4-4: Stagnation Prevention apparatus schematic
4.2.2 Indirect Solar-Assisted Heat Pump

The heat pump loop, as constructed by Bridgeman, was connected to a modified 2.874 m² EnerWorks “Heat Safe”™ collector by connecting the inlet and outlet of the evaporator to the collector loop. From the outlet of the evaporator, the glycol-water mixture passed through a ½ in. flexible hose that connected the heat pump to the wall-mounted plumbing. The fluid then passed through a particulate filter and into the pump. Depending on the flow rate used in the experiments, this was either a Fluid-o-tech MG209 gear pump or a 45 W Iwaki centrifugal pump. The fluid was then pumped to the collector through ¼ in. copper pipes, which matched the serpentine pipe within the collector. The return line from the collector to the evaporator consisted of more 1/4 in. copper pipe until the 1/2 in. flexible hose from the wall back to the evaporator. Two quick-connect ports were located before the pump separated by a ball valve, to facilitate charging of the collector loop. Additional valves were included to aid in charging the system, and isolating sections. An expansion vessel was included to maintain the system pressure in the loop and accommodate any thermal expansion of the fluid.

To measure the useful energy gain by the collector, the thermocouple wells at the inlet and outlet of the evaporator were retained to accommodate $T_5$ and $T_6$, and a Flow Technology Inc. turbine flow meter was added to the outlet of the evaporator to measure the flow rate of the glycol-water mixture. The turbine flow meter outputs a frequency proportional to the flow rate; therefore an Omega FLSC-45 Flow Signal Conditioner was used to convert this frequency to a 4-20 mA signal. This current was passed through a 250 Ω shunt resistor, which resulted in a 1-5 V voltage difference to be read by the data acquisition system. Two additional thermocouples were included to measure the outdoor ambient and the indoor ambient temperatures, while an Eppely PSP pyranometer, mounted in plane with the collector, was included to measure the total normal irradiance. The remainder of the heat pump and instrumentation was left intact. Details of the construction and instrumentation of the heat pump and storage tank loops was discussed by
Bridgeman, and can be found in his thesis [10]. The location and numbering of the instrumentation are detailed in Fig. 4-5, while a summary is presented in Table 4-1. A wiring diagram of the system can be found in Appendix E.

![Wiring Diagram](image_url)

**Fig. 4-5: Instrumentation locations**

<table>
<thead>
<tr>
<th>PART #</th>
<th>DEVICE AND SPECIFICATION</th>
</tr>
</thead>
<tbody>
<tr>
<td>P/T1, P/T4</td>
<td>Pressure/Temperature transducer (0-100 psi, Senstronics LTD)</td>
</tr>
<tr>
<td>P/T2, P/T3</td>
<td>Pressure/Temperature transducer (0-500 psi, Senstronics LTD)</td>
</tr>
<tr>
<td>F – 1</td>
<td>Ultrasonic flow meter (Emerson)</td>
</tr>
<tr>
<td>F – 2</td>
<td>Turbine flow meter (FTI)</td>
</tr>
<tr>
<td>W</td>
<td>Watt meter (ISW8001, Powertek)</td>
</tr>
<tr>
<td>T5-T8, T-Abs, T-Amb, T-Mains</td>
<td>Thermocouple (24 Gauge T-type, Omega)</td>
</tr>
<tr>
<td>T9-T20</td>
<td>Storage Tank Thermocouple Probe (30 Gauge T-type, Omega)</td>
</tr>
<tr>
<td>Qs</td>
<td>Pyranometer, in plane with collector (Eppley Laboratory Inc)</td>
</tr>
</tbody>
</table>
4.3 Experimental Procedure

4.3.1 Stagnation Prevention Apparatus

The experiments conducted on the stagnation prevention apparatus consisted of measuring the steady-state temperatures of the absorber and insulation thermocouples. Depending on the configuration of the collector channel, and the heating mode of the absorber, the heat transfer characteristics of the channel could be determined for above- or below-ambient operation. Experiments with the collector channel open or closed were conducted to determine the heat loss characteristics of the apparatus with and without the increase due to the natural convection flow through the channel. Comparing the results from the open channel tests to the closed channel tests would allow the natural convection effect to be isolated. Four series of experiments were completed as described by Table 4-2. In each series, the heat transfer was induced by varying the heater input power or the fluid inlet temperature within operational limits.

Table 4-2: Experimental Series on the Stagnation Prevention Apparatus

<table>
<thead>
<tr>
<th>Series</th>
<th>Absorber Mode</th>
<th>Channel Configuration</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Heated</td>
<td>Closed</td>
</tr>
<tr>
<td>2</td>
<td>Heated</td>
<td>Open</td>
</tr>
<tr>
<td>3</td>
<td>Cooled</td>
<td>Closed</td>
</tr>
<tr>
<td>4</td>
<td>Cooled</td>
<td>Open</td>
</tr>
</tbody>
</table>

The experimental procedures for Series 1 and 2 are similar, with the exception that the inlet and outlet of the collector channel is blocked with tightly-fitted pieces of isocyanurate insulation and sealed with foil tape for Series 1. The procedures for Series 3 and 4 are also similar, yet with the same exception. A flow rate of 77 kg/hr (1.28 L/min) for the chilled fluid was chosen to be similar to the flow rate recommended by Freeman [9]. Two additional flow
rates were tested at an inlet temperature of 5°C: 38 and 154 kg/hr. The procedure for Series 1 and 2 is as follows:

1. Ensure the channel openings are open or closed, as desired.
2. Turn on the D/A system and record data approximately every 20 seconds.
3. Turn on the power supply and adjust to the desired voltage and current.
4. Wait until the absorber and insulation temperatures have reached steady state, and continue to record for 10 minutes.
5. Power off the system, or repeat at another power level.

The procedure for Series 3 and 4 is slightly different, as it includes the use of the chilled fluid loop to transfer heat from the absorber, as opposed to the electric heaters with supplied the absorber with thermal energy.

1. Ensure the channel openings are open or closed, as desired.
2. Turn on the D/A system and record data approximately every 20 seconds.
3. Turn on the chillers to the desired inlet temperature.
4. Turn on the circulating pump and maintain balance in chiller fluid levels.
5. Wait until the absorber, insulation and fluid temperatures have reached steady state, and continue to record for 10 minutes.
6. Divert the fluid to the graduated cylinder, recording time to fill 1 L.
7. Power off the system, or repeat at another inlet temperature.

Tables outlining the experiments performed are given in Table 4-3 for Series 1 and 2, and Table 4-4 for Series 3 and 4. The results from these experiments, which were used to determine the convection coefficient for the COLSIM model, are presented in Chapter 5.
Table 4-3: Stagnation Prevention Apparatus Test Parameters for Series 1 and 2

<table>
<thead>
<tr>
<th>Collector Heat Loss Tests, 45° Tilt</th>
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### Table 4.4: Stagnation Prevention Apparatus Test Parameters for Series 3 and 4

<table>
<thead>
<tr>
<th>Collector Heat Loss Tests, 45° Tilt</th>
<th>Test Conditions</th>
<th>Glycol Inlet Temperature [°C]</th>
<th>Glycol Flow Rate [kg/hr]</th>
<th>Channel Configuration</th>
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*Below Ambient*
4.3.2 Indirect Solar-Assisted Heat Pump

Experiments were performed on the ISAHP between July 2010 and August 2011, under a variety of natural weather conditions, e.g., solar irradiance levels, ambient temperatures, mains water temperatures, and other climatic variables were allowed to vary. To ensure that appropriate comparisons could be made between different experimental results, care was taken such that the tests were run in a similar manner each day. Consequently, the effects of variable conditions would not be compounded by inconsistencies in the test procedure.

Day-long storage tank charge tests, which used the ISAHP to heat the water within the storage tank from an initially uniform mains temperature to a final fully charged condition, were the most common tests performed. The procedure used for these tests is as follows:

1. One hour before the start of the test, the storage tank was filled with water to a uniform temperature equal to the mains temperature, and left to settle.
2. The D/A system was initialized and data was recorded at 5 second intervals.
3. At 8:00 am, the compressor and collector loop pump were turned on.
4. The system was left on until one of three stopping conditions were met: the temperature of the bottom of the tank reached 30°C; the temperature of collector-loop fluid entering the evaporator decreased below -5°C near sunset; or when the pressure safety switch on the compressor was triggered.

During the winter months, when the sun rose later is the day, and localized shading from a nearby building would block the solar irradiance in the morning, the start of the tests was postponed to 9:00 am. However, the stopping conditions remained the same. Consequently, the tests often lasted until sunset, when the second stopping condition would take effect. These temperatures were chosen based upon practical limits of the system, particularly the operational limits of the compressor.
Additional experiments were conducted to isolate the effect of the open channel in the hybrid collector. In these experiments, a step-change was applied to the operating conditions of the collector, and the resulting change in useful energy gain was observed. The first step-change was to instantaneously shade the collector such that it was only able to absorb energy via convection with the surroundings. To accomplish this, a large reflective sheet was suspended 30 cm above the collector, such that it completely blocked the irradiance, yet did not interfere with the convective heat transfer near the collector surface. The second step-change investigated was to seal the channel openings, such that the increased convective heat transfer from the channel was eliminated. The same tightly-fitting blocks of isocyanurate insulation from the stagnation prevention apparatus were used to stop the natural convection flow through the channel. The step-changes were applied at solar noon to ensure that the collector would be operating under near-steady state conditions, before and after the change. Each of these experiments was conducted in a similar fashion, according to the following procedure:

1. Starting at least two hours before solar noon, the mains water and draw valves on the storage tank were opened, to purge the tank and maintain a constant condenser temperature.
2. Concurrently, the D/A system was initialized and data was recorded at 5 second intervals.
3. The compressor and collector loop pump were turned on, with the pump speed set at 77 kg/hr.
4. At solar noon, the step-change was applied, and the system was allowed to reach a new thermal equilibrium.
5. Once the collector temperatures reached equilibrium, the step-change was reversed, and another hour of data was recorded.
4.4 Uncertainty

Uncertainty is inherent in any physical experiment with measured values. Consequently, a measure of the propagated error in the values derived from these experiments was determined by an error analysis. Bridgeman conducted a thorough analysis for the error associated with the ISAHP measurements, and found that the errors ranged from 8.4 -18.8% for the evaporator heat transfer rate measured from the glycol side, 3.3 - 3.5% measured from the refrigerant side, 2.6 – 2.8% for the heat transfer rate through the condenser, 4.7 – 6.2% for the natural convection flow rate, and 3.2 – 4.2% for the heat pump COP [10]. The replacement of the positive displacement flow meter used by Bridgeman with the FTI turbine flow meter in the collector loop changed the evaporator heat transfer rate error to 8.9% – 13.4%, measured on the glycol side. The error associated with the collector loop efficiency or CPF was calculated based on the glycol side heat transfer rate as 9.1% to 13.4%, while for the refrigerant side it was 3.5% to 4.3%.

The error associated with the experiments conducted on the stagnation prevention apparatus to determine the heat transfer coefficients in the collector channel were determined, as shown in Appendix G. The resulting error values in the Nusselt number correlations ranged from 6.7% at large temperature differences to 30.1% at small temperature differences, for the heated condition. For the cooled condition, the error ranged from 10.3% at large temperature differences to 33.6% at small temperature differences. While small temperature differences led to large relative errors on the Nusselt number, the actual effect of these errors was small, as they corresponded to errors in the heat transfer rates of 14.5 W and 13.9 W. These values are negligible when compared to the irradiance and useful energy heat transfer rates that would occur when operating near ambient temperature, which can be on the order of 1.5 – 2 kW.
4.5 Validation of Data

To ensure that the ISAHP data was valid, a number of performance checks were conducted. Energy balances were conducted across the condenser and evaporators to validate assumptions made in the heat pump analysis, as well as, confirm the collector loop pump was calibrated correctly.

4.5.1 Energy Balances

The energy balance across the condenser can be expressed as:

\[ E_{\text{cond,ref}} - E_{\text{st,tank}} - E_{\text{loss,tank}} = 0 \]  \hspace{1cm} (4-1)

For each storage tank charge test, the total heat lost by the refrigerant as it passed through the condenser, \( E_{\text{cond,ref}} \), was calculated by a numerical integration over each time step of recorded data:

\[ E_{\text{cond,ref}} = \sum_{i=0}^{N} [\dot{m}_{\text{ref,i}}(h_{2,i} - h_{3,i})(t_i - t_{i-1})] \]  \hspace{1cm} (4-2)

where \( t_i \) and \( t_{i-1} \) are the times that data point \( i \) and \( i-1 \) were recorded. The enthalpies \( h_2 \) and \( h_3 \) were determined using the reference tables in EES, knowing the temperature and pressure at each point. The total energy stored by the water, \( E_{\text{st,tank}} \), during the same test was determined by comparing the temperature state at the end to the state at the start of the test for each node, and therefore the entire tank:

\[ E_{\text{st,tank}} = \sum_{i=1}^{10} [m_i (C_{p,i}(\text{end})T_{\text{tank,i}(\text{end})} - C_{p,i}(\text{start})T_{\text{tank,i}(\text{start})})] \]  \hspace{1cm} (4-3)

As the temperature probe for the tank contained 10 evenly spaced thermocouples, the storage tank effectively consisted of 10 nodes. The energy lost by the tank to the environment, \( E_{\text{loss,tank}} \), was determined as the sum of the energy lost during each time step:
\[ E_{\text{loss,tank}} = \sum_{i=0}^{N} \left[ UA(T_{\text{tank,avg}} - T_{\text{env}}) (t_i - t_{i-1}) \right] \quad (4-4) \]

It was observed that \( E_{\text{cond,ref}} \) surpassed \( E_{\text{at,tank}} \) by 1% to 4% over all tests, while the lost energy from the tank during the tests accounted for 1% to 3% of the energy transferred from the refrigerant through the condenser. As a result, the energy balance across the condenser was valid, as the energy transferred from the refrigerant agreed with the energy stored and lost by the tank within the experimental uncertainty of the condenser heat transfer. This balance was achieved by the careful calibration of the ultrasonic refrigerant flow meter and temperature sensors performed by Bridgeman during the initial commissioning of the system [10].

As the condenser energy balance was accurate, the heat pump loop was known to be in working order. Consequently, an energy balance across the evaporator was used to ensure valid results were obtained from the collector loop measurements. Unlike the condenser, where the water-side heat transfer rate was assumed equal to the rate on refrigerant side, the heat transfer rates on either side of the evaporator were determined independently. The heat transfer rate, calculated on the refrigerant side, was determined as:

\[ Q_{\text{evap,ref}} = \dot{m}_{\text{ref}} (h_1 - h_4) \quad (4-5) \]

Once again, the enthalpies were determined from the reference tables in EES, knowing the pressure and temperatures. The heat transfer rate on the glycol-water mixture side of the evaporator was determined by knowing the flow rate and temperatures, with the \( C_p \) evaluated independently at the inlet and outlet temperatures.

\[ Q_{\text{evap,gly}} = \dot{m}_{\text{gly}} \left( C_{p,5} T_5 - C_{p,6} T_6 \right) \quad (4-6) \]

Comparing the evaporator heat transfer rates at each time step throughout the experiments gave an indication of their agreement. The heat transfer rates from the two sides of the evaporator, calculated from data taken on September 22nd, 2010 are shown in Fig. 4-6.
The values agree within 5% for the majority of the day, with the exception of the peaks between 11 am - 12 pm, and 2-3 pm, which were caused by sudden increases in solar irradiance due to cloud movement. With the exception of the highly transient conditions, the agreement indicates that the collector loop flow meter and evaporator inlet and outlet thermocouples were calibrated correctly. During the transient conditions, it was observed that collector loop flow rate remained constant, as would be expected with a positive displacement pump. It was therefore concluded that main source of error was due to the thermocouples on the glycol side of the evaporator.

While the error in the evaporator energy balance in minimal for the September 22nd data, the difference was seen to be more pronounced during other experiments, such as the March 28th, 2011 data presented in Fig. 4-7, where an 8% error was observed. Therefore, due to these results and the lower associated uncertainty, it was determined that the heat transfer rates should be calculated based upon the refrigerant measurements.
Two main assumptions were made in order to analyse the heat pump’s performance. First, it was assumed that the refrigerant exiting the condenser at point 3 was a saturated liquid. While small instances of subcooling could occur, small errors in the temperature measurement without this assumption would cause EES to return the enthalpy values for saturated or superheated vapour. This assumption was justified as the error in the enthalpy associated with small deviations from the saturated liquid value was much less than if the vapour enthalpies were used. In effect, the assumption was validated by the energy balance check on the condenser.

The second assumption was that the flow of the refrigerant through the thermostatic expansion valve was an isentropic process \( (h_4 = h_3) \). This is a common assumption for expansion valves, as no work is done and if properly insulated, the process could be considered adiabatic [42]. This assumption was necessary as the enthalpy could not be determined at state 4 from the temperature and pressure without knowing quality of the mixture. Due to the insulation on the refrigeration lines in the heat pump, and the balanced evaporator heat transfer, this assumption was considered valid for this study.
Chapter 5

Results and Analysis

5.1 Overview

Experimental results from the Stagnation Prevention apparatus and the ISAHP prototype are presented in the subsequent sections. These results were used in part to enable and validate simulations of the system on a daily and annual basis. The validated system model was used to investigate the effect of system parameters to yield the best annual performance for the Canadian climate. In particular, the annual performance using the hybrid collector was compared to the use of commercially available unglazed and glazed collectors.

5.2 Collector Heat Transfer Characteristics

As described in Chapter 4, four series of experiments were conducted on the Stagnation Prevention apparatus to determine the heat transfer characteristics of the open channel. The first series was used to determine the heat loss from the collector when operating above the ambient temperature, and with the collector channel sealed to prevent the natural convection flow. The second series was also heated above ambient, but with the channel open, such that the natural convection heat transfer could be analysed. The third and fourth series had the channel closed and open, respectively, yet with the collector cooled below ambient.

5.2.1 Closed Channel Results

For Series 1, the temperature measurements from the absorber and insulation surfaces inside the channel were recorded at steady state for electrical heater input fluxes of 20, 50, 75, 100, 150 and 200 W/m². The temperatures read by each row of three thermocouples from Fig. 4-1 were averaged to give the temperature of the surface at their position along the length of the
The resulting temperature profiles for the absorber and insulation are presented in Fig. 5-1 and Fig. 5-2, respectively.

**Fig. 5-1: Absorber temperature profiles, plotted by flux [W/m²] (heated, closed)**

**Fig. 5-2: Insulation temperature profiles, plotted by flux [W/m²] (heated, closed)**
Knowing the temperature profile along the channel surfaces, the radiation exchange between the absorber and insulation planes was calculated by Hottel’s zone method [24], with 16 isothermal segments used along the length of the absorber plate, and 14 isothermal segments along the insulation. The number of segments was chosen to correspond to the number of bends in the serpentine tube on the absorber. The temperature profiles were also used to determine the area-weighted mean surface temperatures, $T_{PM}$ and $T_{IM}$, by integrating the profiles with respect to the distance along the collector, and dividing by the total length. Additional heat transfer between the absorber and the insulation occurred by conduction through the still air, and was calculated based upon the mean temperatures:

$$Q_{\text{cond},p-i} = \frac{kA_c}{S} (T_{PM} - T_{IM}) \quad (5-1)$$

By conducting an energy balance on the collector, the remaining power, $Q_{\text{cond},p-a}$, was found to be dissipated from the absorber by conduction through the sides and top of the collector apparatus:

$$Q_{\text{cond},p-a} = Q_\Omega - Q_{\text{rad},p-i} - Q_{\text{cond},p-i} \quad (5-2)$$

where $Q_\Omega$ is the power delivered by the electrical heaters. Fig. 5-3 is a plot of $Q_{\text{cond},p-a}$ as a function of the mean absorber temperature, including the line of best fit to the data. Similarly, the power dissipated through the rear insulation, $Q_{\text{loss},i-a}$, was determined from an energy balance:

$$Q_{\text{loss},i-a} = Q_{\text{rad},p-i} + Q_{\text{cond},p-i} \quad (5-3)$$

$Q_{\text{loss},i-a}$ could then be used to determine the $U_{i-a}$ value for the COLSIM model if the channel was closed:

$$U_{i-a} = \frac{Q_{\text{loss},i-a}}{A_c(T_{IM} - T_A)} \quad (5-3)$$
The $U_{i-a}$ as a function of $T_{IM}$ and $T_A$ is shown in Fig. 5-4. The resulting line of best fit, after translation of the data, was:

$$U_{i-a} = 0.75(T_{IM} - T_A)^{0.25} - 1$$  \hspace{1cm} (5-4)

\[ Q_{\text{cond,p-a}} = 3.1411(T_{PM} - T_A) + 4.1478 \]
\[ R^2 = 0.9992 \]

Fig. 5-3: Heat lost by conduction through the collector top and edges (heated, closed)

\[ U_{i-a} \text{ [W/m}^2\text{K]} \]
\[ T_{IM} - T_A \text{ [°C]} \]

Fig. 5-4: Heat loss coefficient from the insulation through the collector back (heated, closed)
This analysis was repeated for Series 3, where the absorber was cooled below ambient, and the channel was sealed. Inlet temperatures of -10, -5, 5, 10, and 15°C were used to impose a temperature difference below the ambient, which was steady at 22.5 ± 0.5°C. As a result, heat was removed by the glycol-water mixture at a rate of $Q_{\text{fluid}}$, which replaced $Q_{\Omega}$ in the above analysis. The resulting temperature profiles are displayed in Fig. 5-5 and Fig. 5-6 for the absorber and insulation planes. the plots for $Q_{\text{cond,p-a}}$ and $U_{i-a}$ and their respective lines of best fit are shown in Fig. 5-7 and Fig. 5-8.

![Absorber temperature profiles, plotted by inlet temperature (cooled, closed)](image)

Fig. 5-5: Absorber temperature profiles, plotted by inlet temperature (cooled, closed)
Fig. 5-6: Insulation temperature profiles for, plotted by inlet temperature (cooled, closed)

Fig. 5-7: Heat lost by conduction through the collector top and edges (cooled, closed)

\[ Q_{\text{cond,p-a}} = -0.175(T_{PM} - T_A)^2 - 0.564(T_{PM} - T_A) - 14.422 \]

\[ R^2 = 1.000 \]
5.2.2 Open Channel Results

The temperature profiles for the open channel were determined for Series 2 and 4. For Series 2, the heater fluxes were 10, 50, 75, 100, 200, 300, and 400 W/m². Higher power levels were used, such that equivalent temperatures from Series 1 could be obtained. The resulting temperature profiles for the absorber and insulation are shown in Fig. 5-9 and Fig. 5-10.
To determine the convective heat transfer coefficient, it is necessary to isolate the energy that was being lost by convection from the total energy dissipated from the absorber. The energy balance around the absorber plate control volume is:

\[ Q_{\text{conv, \text{p-a}}} = Q_A - Q_{\text{rad, \text{p-i}}} - Q_{\text{cond, \text{p-a}}} \]  

(5-5)

Knowing the temperature profiles, the \( Q_{\text{rad, \text{p-i}}} \) was calculated as in section 5.2.1. The expression for \( Q_{\text{cond, \text{p-a}}} \) as a function of \( T_{PM} \) and \( T_A \), determined from the Series 1 results, was used to determine \( Q_{\text{cond, \text{p-a}}} \) in this case. Subsequently, the value for \( Q_{\text{conv, \text{p-a}}} \) was determined, and is shown in Fig. 5-11. The convective coefficient, \( h_{\text{cv, \text{p-a}}} \), was determined from \( Q_{\text{cond, \text{p-a}}} \):

\[ h_{\text{cv, \text{p-a}}} = \frac{Q_{\text{conv, \text{p-a}}}}{A_C(T_{PM} - T_A)} \]  

(5-6)

For use in the COLSIM model, the Nusselt number was calculated for each data point, and is shown in Fig. 5-12 as a function of the \( Ra \text{g} \cos \theta \) product.
Fig. 5-11: Convective heat transfer from the absorber (heated, open)

Fig. 5-12: Nusselt number versus Rayleigh number (heated, open)
Translation of the data was required to obtain a good fit to a power-law curve similar to Azevedo [27], Lin [29] and Manca [28]. The resulting line of best fit for Series 2 was:

\[ \text{Nu}_S = 0.0808(Ra_S \cos \theta)^{0.365} - 1 \]  

(5-7)

It is impossible for the Nusselt number to be less than 1. Consequently, in the COLSIM model, if the curve fit predicted a value less than 1, it was assumed that convection was minimal, and a value of 1 was used. Another value of interest for the COLSIM model was the heat loss coefficient from the insulation to the surroundings, \( U_{i-a} \). As was done for the closed channel series, \( U_{i-a} \) was calculated from an energy balance on the insulation plane. In this case, the energy balance was as expressed in equation 3-38, and \( U_{i-a} \) was calculated according to equation 3-39. The resulting values are plotted in Fig. 5-13, while translation of the data led to the best fit curve of:

\[ U_{i-a} = 1.915(T_{IM} - T_A)^{0.294} - 2 \]  

(5-8)

![Fig. 5-13: Heat loss coefficient from the insulation (heated, open)](image)
As mentioned in Chapter 3, the radiative exchange between the absorber and insulation in the COLSIM model could be calculated using the infinite parallel plates assumption; however, a more accurate value could be determined using Hottel’s zone method, described in section 5.2.1. The radiative heat transfer calculated from Hottel’s zone method is plotted against the results from the parallel plates method using $T_{PM}$ and $T_{IM}$ in Fig. 5-14. As the linear fit to the data shows, the results from Hottel’s zone method are approximately 1.15 times the infinite parallel plates result. Therefore, it was decided to calculate the radiation exchange in the COLSIM model by the infinite parallel planes method, but multiplied by this factor of 1.15.

The fourth and final series involved the open channel with the collector cooled below ambient. The analysis for this series was similar to the analysis for Series 2, above. Inlet temperatures of -10, -5, 0, 5, 10, and 15°C were used while the ambient temperature remained at $22 \pm 1°C$. The resulting temperature profiles are plotted in Fig. 5-15 and Fig. 5-16.
Fig. 5-15: Absorber temperature profiles, plotted by inlet temperature (cooled, open)

Fig. 5-16: Insulation temperature profiles, plotted by inlet temperature (cooled, open)

The energy balance of equation 5-5 was used to solve for \( Q_{\text{conv,p}} \), with \( Q_0 \) replaced by \( Q_{\text{fluid}}\). The results are plotted in Fig. 5-17, while the Nusselt number is plotted in Fig. 5-18.
Fig. 5-17: Convective heat transfer from the absorber (cooled, open)

Fig. 5-18: Nusselt number versus Rayleigh number (cooled, open)
As the Rayleigh number was negative in this series, the absolute value of $Ra_S \cos \theta$ was required for a curve fit. The Nusselt number relationship, with respect to the Rayleigh number, was therefore:

$$Nu_S = 0.0039 |Ra_S \cos \theta|^{0.6774}$$ (5-9)

$U_{i,a}$ for Series 4 was calculated in the same manner as for Series 2. The values derived from the energy balance on this insulation are plotted in Fig. 5-19.

**Fig. 5-19: Heat loss coefficient from the insulation plane of the channel (cooled, open)**

Finally, the relationship between the radiation calculated by Hottel’s zone method and the infinite parallel plates method is shown in Fig. 5-20. In this case, the linear fit shows that the infinite parallel plates method over-predicts Hottel’s zone method. Therefore, in the COLSIM model, when operating below ambient, the radiation exchange calculated by the infinite parallel plates method was adjusted by a factor of 0.91 to correspond to the Hottel’s zone method results.
To investigate the effect of the glycol-water mixture flow rate on the temperature distribution in Series 4, and what effect this would have on $Q_{\text{conv,p-a}}$ and $Q_{\text{rad,p-i}}$, tests were repeated for an inlet temperature of 5°C, with half and double the flow rate. The resulting absorber temperature distributions are shown in Fig. 5-21, while the results for $Q_{\text{conv,p-a}}$ and $Q_{\text{rad,p-i}}$ are superimposed on the plots for the nominal flow rate of 77 kg/hr, in Fig. 5-22 and Fig. 5-23, respectively. As the results fall along the same curve, it can be seen that the flow rate has little effect on the heat transfer in the channel, as long as the relationships are determined with respect to the mean surface temperatures.

**Fig. 5-20: Hottel’s zone method versus infinite parallel plates, Series 4**

![Graph showing the relationship between Qrad (Hottel's Zone Method) and Qrad (Infinite Parallel Plates) with the equation y = 0.91x and R² = 1.00.](image-url)
Fig. 5-21: Absorber temperature profiles for various flow rates, plotted by inlet temperature (cooled, open)

Fig. 5-22: Convective heat transfer from the absorber for various flow rates (cooled, open)
5.2.3 Summary of Collector Channel Results

The relationships determined by the experiments performed on the Stagnation Prevention Apparatus for use in the COLSIM model are summarized in Table 5-1 below.

### Table 5-1: Summary of Collector Channel Results for COLSIM

<table>
<thead>
<tr>
<th>Condition</th>
<th>Channel Configuration</th>
<th>$U_{i-a}$</th>
<th>$N_u_S$</th>
<th>$Q_{rad,p-i}$ Correction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Above Ambient</td>
<td>Closed</td>
<td>$0.75(T_{IM} - T_A)^{0.25} - 1$</td>
<td>N/A</td>
<td>1.15</td>
</tr>
<tr>
<td>Above Ambient</td>
<td>Open</td>
<td>$1.915(T_{IM} - T_A)^{0.294} - 2$</td>
<td>$0.0808(Ra_S \cos \theta)^{0.365} - 1$</td>
<td>1.15</td>
</tr>
<tr>
<td>Below Ambient</td>
<td>Closed</td>
<td>$0.0095(T_{IM} - T_A) + 0.72$</td>
<td>N/A</td>
<td>0.91</td>
</tr>
<tr>
<td>Below Ambient</td>
<td>Open</td>
<td>$0.005(T_{IM} - T_A) + 1.48$</td>
<td>$0.0039</td>
<td>Ra_S \cos \theta</td>
</tr>
</tbody>
</table>
5.3 ISAHP Experimental Results

Experimental data was collected from the ISAHP apparatus operating on numerous days between July 2010 and July 2011. The majority were day-long storage tank charge tests, where the performance of the system was observed under various weather conditions throughout all seasons. Operational limits were noted, which included the low temperature limit of the collector loop, and the high pressure cut-off of the compressor. The system’s response to step-changes in the collector conditions was also observed.

5.3.1 Derivation of Results

The performance indices of the system were calculated as outlined in Chapter 3. Refrigerant enthalpies were determined using the reference tables in EES, while the glycol-water mixture’s properties were determined from published relationships [43]. As mentioned in Chapter 4, the heat transfer rates determined on the refrigerant side of the evaporator included less error, and were therefore used instead of the glycol side values.

The calculation of the free energy ratio, described in Section 3.5.4 was not used for the storage tank charge tests, as no water draws were made, and no auxiliary heaters were included. Instead, the daily free energy ratio, \( FER_D \), was calculated in these tests as the ratio of the energy delivered to the storage tank from free sources to the energy required to meet the daily demand of 225 L of water delivered at 51.7°C [41]:

\[
FER_D = \frac{\int Q_{\text{cond}} \, dt - \int \dot{W} \, dt}{m_{\text{load}} C_{p,\text{water}} (T_{\text{set}} - T_{\text{tank, initial}})}
\]  

(5.9)

where \( \dot{W} \) is the electrical power consumed by the compressor and pump. While the compressor power was measured by the D/A, the pump power was estimated from the pump specifications.
5.3.2 Day-long Storage Tank Charge Tests

The results from day-long storage tank charge tests are presented here, with the collector channel open in all instances. Two tests were conducted in July 2010 which highlight the differences in the performance of the system under clear and overcast days, at a nominal collector flow rate of 77 kg/hr. The clear day test was allowed to run until the water leaving the bottom of the tank reached 30°C, while the overcast day was run over the same time span, such that the total energy delivered could be compared directly. The climatic conditions and the resulting performance indices are plotted in Fig. 5-24 and Fig. 5-25 for the clear day, and in Fig. 5-26 and Fig. 5-27 for the overcast day. The temperature profiles within the tanks are shown in Fig. 5-28 and Fig. 5-29.

The solar radiation was significantly lower on the overcast day; however the heat transfer rates did not vary to the same extent, suggesting that the collector was absorbing heat from the ambient air while operating below ambient temperature. On the overcast day, the CPF was consistently above unity. This confirms that the heat gained from the ambient air was the primary source of energy on warm, overcast days. On the clear day, the efficiency fluctuated during the first half of the test due to variations in the solar radiation, and increasing system temperatures, yet settled to approximately 0.4 for the remaining half.

The COP of the system did not differ to the same extent that the collector efficiency or CPF did between the overcast and clear days. However, as the compressor was operating at a consistent power level, and less energy was transferred through the heat pump on the overcast day, a slightly lower COP was observed. The decline in COP towards the end of the test was due to the rising average storage tank. Stratification of the storage minimized this effect by maintaining a cold inlet temperature to the condenser; however the decline in the natural convection flow rate lowered the COP. Over the identical time span, the ISAHP delivered 19.8% percent more energy to the storage tank on the clear day, yet only used 14.1% more energy to
drive the compressor, suggesting the heat pump operates more economically during periods of increased solar radiation.

Fig. 5-24: Heat transfer rates and climatic conditions for July 27th, 2010 (clear)

Fig. 5-25: COP and collector efficiency or CPF for July 27th, 2010 (clear)
Fig. 5-26: Heat transfer rates and climatic conditions for July 23rd, 2010 (overcast)

Fig. 5-27: COP and collector efficiency or CPF for July 23rd, 2010 (overcast)
Fig. 5-28: Stratified tank temperature profile for July 27\textsuperscript{th}, 2010.
Maximum temperature rise of 41.2°C

Fig. 5-29: Stratified tank temperature profile for July 23\textsuperscript{rd}, 2010.
Maximum temperature rise of 33.8°C
The performance on a clear February day at 77 kg/hr was comparable to the clear July day, as seen by the graphs of heat transfer rates and performance indices in Fig. 5-30 and Fig. 5-31. The heat transfer rates through the heat pump reached a similar maximum at midday. This occurred under higher irradiance yet lower temperatures. The higher irradiance can be attributed to a few factors: February 10\textsuperscript{th} is closer to an equinox, where the solar altitude angle at noon is 45°, resulting in lower incidence angles between the sun and collector tilted at 45°; and the ground was snow covered, increasing diffuse reflection. However, the lower ambient temperatures increased the heat loss from the collector, limiting the collected energy. This effect was particularly apparent near noon, where the decrease in heat transfer rates corresponded to a decrease in ambient temperature. Additionally, the heat transfer rates changed more rapidly in the morning and afternoon, as the collector could not gain sufficient energy from the low ambient temperature when the irradiance levels were low.

Even though the low ambient temperature increased the collector heat loss, and consequently lowered the collector temperatures, sufficient energy was still able to be delivered to the storage tank. The tank temperature profile for February 10\textsuperscript{th}, 2011 is shown in Fig. 5-32, and it can be seen that the mains temperature and therefore initial tank temperature was 5°C as opposed to 21°C for July 27\textsuperscript{th}. Similar heat transfer rates were observed for collector loop temperatures between 20°C-30°C on February 10\textsuperscript{th} as with the 40°C-50°C temperatures observed on July 27\textsuperscript{th}. This was possible because low ambient temperatures coincide with low mains water temperature, and in effect, the temperature difference across the heat pump was similar. While the temperature difference across the system was similar, the lower condenser temperatures corresponded to lower condenser pressures in the heat pump circuit, requiring less compressor work. Consequently, the lower condenser temperature resulted in an increased \textit{COP}. 

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Fig. 5-30: Heat transfer rates and climatic conditions for February 10th, 2011 (clear). Morning scatter in data due to shading from a smoke plume from a nearby chimney.

Fig. 5-31: COP and collector efficiency or CPF for February 10th, 2011 (clear). Morning scatter in data due to shading from a smoke plume from a nearby chimney. Frequency observed at the start and end of the test caused by the expansion valve fluctuating.
Fig. 5-32: Stratified tank temperature profile for February 10th, 2011.

Maximum temperature rise of 41.7°C

The performance of the ISAHP on a colder overcast day was also observed on November 17th, 2010. Due to the cold collector loop temperatures, the flow rate was reduced to 55 kg/hr to avoid an operational limit described in Section 5.3.4. The resulting heat transfer rates and performance indices are shown in Fig. 5-33 and Fig. 5-34, while the tank profile is shown in Fig. 5-35. Under these conditions, a marked decrease in performance was observed when compared to the July 23rd, 2010 data, which can be attributed the combination of the lower ambient temperature and the lower flow rate. In both these experiments, the average collector loop fluid temperature was approximately 14°C below ambient. Therefore under the same temperature difference, a lower flow rate would lead to less energy collected. The lower temperatures also led to a slightly reduced COP, as the compressor power did not vary to the same extent as the heat transfer rates.
Fig. 5-33: Heat transfer rates and climatic conditions for November 17th, 2010 (overcast)

Fig. 5-34: COP and collector efficiency or CPF for November 17th, 2010 (overcast)
Maximum temperature rise of 32.0°C

While the above experiments, particularly the clear days of July 27th, 2010 and February 10th, 2011, had $Q_{\text{cond}}$ values peaking just below 1.5 kW, higher heat transfer rates were observed on days with lower irradiance levels or ambient temperatures. For example, consider the results for October 27th, 2010, shown in Fig. 5-36 and Fig. 5-37, with the tank profile shown in Fig. 5-38. The $Q_{\text{cond}}$ reached a maximum of 1.85 kW, a 23% increase over the July 27th maximum, with only a 10% increase in peak irradiance and an 11°C decrease in average ambient temperature. This difference in performance appeared to be attributed to a decrease in evaporator $UA$ values. By manipulating equations 3-1 through 3-4, the evaporator effectiveness and the $UA$ values were calculated for each test. The $UA$ values calculated throughout the tests for July 27th and October 27th, 2010 are compared in Fig. 5-39.
Fig. 5-36: Heat transfer rates and climatic conditions for October 27th, 2010 (clear)

Fig. 5-37: COP and collector efficiency or CPF for October 27th, 2010 (clear)
Fig. 5-38: Stratified tank temperature profile for October 27th, 2010.

Maximum temperature rise of 50.3°C

Fig. 5-39: Comparison of evaporator UA values for July 27th and October 27th, 2010
Bridgeman determined that the evaporator $UA$ value varied with collector loop flow rates, evaporator superheat, and temperatures, yet found it was difficult to develop a single curve fit based upon these values. Bridgeman found during constant temperature tests that the evaporator $UA$ values remained between 0.10 and 0.15 kW/°C \cite{10}, however throughout these day-long tests, $UA$ values ranged from 0.02 to 0.18 kW/°C. Tests which had an average evaporator $UA$ value throughout the day of 0.06 kW/°C and higher tended to show significant increase in performance, as summarized in Table 5.2.

Further increases in performance were observed by increasing the collector loop flow rate. In effect, increasing the flow rate increased the evaporator $UA$ value, because of higher convection coefficients within the heat exchanger. Additionally, higher flow rates lead to higher collector efficiencies, continuing the increase in performance with flow rate. Significant improvements can be seen in Table 5.2 between the July 2011 data at 288 kg/hr to 298 kg/hr and the July 2010 data at 77 kg/hr. However, June 2011 tests at 158 kg/hr achieved comparable performances to the July 2011 tests, which can be attributed to the fact that the collector approached its optical efficiency, and further increases in flow rates had diminishing returns on the efficiency increase. Therefore expending more pump power to increase the flow rate beyond 158 kg/hr was not beneficial. This collector flow rate corresponds with the flow rate of 154 kg/hr that Bridgeman used in his optimum configuration for annual simulations \cite{10}. Throughout these differing flow rates and temperatures, the flow remained laminar with Reynolds numbers between 500 and 1000. Therefore, no effects of transition or turbulence significantly influenced the heat transfer characteristics in the collector loop.
The maximum flow rate obtained with the positive displacement pump was 142 kg/hr. To achieve the higher flow rates, a centrifugal pump was used. Centrifugal pumps, unlike positive displacement pumps, provide flow rates that are sensitive to the pressure drop through the system. Therefore during the tests which used the centrifugal pump, the flow rate increased throughout the day as the glycol-water mixture warmed and became less viscous. The flow rate reported in Table 5.2 are the flow rates measured once the system had warmed up.
Table 5-2: Summary of Average Values from Day-long Storage Tank Charge Tests

<table>
<thead>
<tr>
<th>Date</th>
<th>$T_{amb}$ [°C]</th>
<th>$T_{mains}$ [°C]</th>
<th>Radiation [W/m²]</th>
<th>Flow Rate [kg/hr]</th>
<th>$Q_{cond}$ [kW]</th>
<th>COP</th>
<th>$\eta$ or CPF</th>
<th>FERD</th>
<th>Evap. UA [kW/°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>23-Jul-10</td>
<td>20.8</td>
<td>19.0</td>
<td>94</td>
<td>77</td>
<td>1.05</td>
<td>2.38</td>
<td>3.13</td>
<td>0.54</td>
<td>0.05</td>
</tr>
<tr>
<td>27-Jul-10</td>
<td>27.6</td>
<td>20.8</td>
<td>723</td>
<td>77</td>
<td>1.31</td>
<td>2.54</td>
<td>0.59</td>
<td>0.75</td>
<td>0.04</td>
</tr>
<tr>
<td>02-Sep-10</td>
<td>26.8</td>
<td>22.7</td>
<td>478</td>
<td>77</td>
<td>1.34</td>
<td>2.57</td>
<td>0.91</td>
<td>0.80</td>
<td>0.06</td>
</tr>
<tr>
<td>17-Sep-10</td>
<td>17.3</td>
<td>20.0</td>
<td>835</td>
<td>77</td>
<td>1.57</td>
<td>2.83</td>
<td>0.66</td>
<td>0.84</td>
<td>0.06</td>
</tr>
<tr>
<td>22-Sep-10</td>
<td>20.5</td>
<td>19.4</td>
<td>259</td>
<td>77</td>
<td>1.16</td>
<td>2.44</td>
<td>1.65</td>
<td>0.69</td>
<td>0.09</td>
</tr>
<tr>
<td>20-Oct-10</td>
<td>14.4</td>
<td>15.6</td>
<td>667</td>
<td>77</td>
<td>1.41</td>
<td>2.66</td>
<td>0.75</td>
<td>0.78</td>
<td>0.07</td>
</tr>
<tr>
<td>27-Oct-10</td>
<td>15.9</td>
<td>14.9</td>
<td>788</td>
<td>77</td>
<td>1.56</td>
<td>2.81</td>
<td>0.71</td>
<td>0.79</td>
<td>0.07</td>
</tr>
<tr>
<td>02-Nov-10</td>
<td>8.3</td>
<td>14.0</td>
<td>801</td>
<td>77</td>
<td>1.65</td>
<td>2.72</td>
<td>0.57</td>
<td>0.81</td>
<td>0.10</td>
</tr>
<tr>
<td>17-Nov-10</td>
<td>8.8</td>
<td>12.4</td>
<td>57</td>
<td>55</td>
<td>0.86</td>
<td>2.24</td>
<td>4.28</td>
<td>0.41</td>
<td>0.06</td>
</tr>
<tr>
<td>15-Dec-10</td>
<td>-7.2</td>
<td>11.9</td>
<td>670</td>
<td>77</td>
<td>1.05</td>
<td>2.64</td>
<td>0.54</td>
<td>0.48</td>
<td>0.05</td>
</tr>
<tr>
<td>10-Feb-11</td>
<td>-6.1</td>
<td>5.1</td>
<td>786</td>
<td>77</td>
<td>1.19</td>
<td>2.73</td>
<td>0.49</td>
<td>0.54</td>
<td>0.04</td>
</tr>
<tr>
<td>15-Feb-11*</td>
<td>-7.4</td>
<td>4.8</td>
<td>806</td>
<td>77</td>
<td>1.32</td>
<td>2.89</td>
<td>0.52</td>
<td>0.61</td>
<td>0.04</td>
</tr>
<tr>
<td>03-Mar-11</td>
<td>-6.4</td>
<td>5.0</td>
<td>815</td>
<td>77</td>
<td>1.25</td>
<td>2.81</td>
<td>0.47</td>
<td>0.60</td>
<td>0.05</td>
</tr>
<tr>
<td>18-Mar-11</td>
<td>11.2</td>
<td>5.5</td>
<td>723</td>
<td>142</td>
<td>1.64</td>
<td>2.95</td>
<td>0.66</td>
<td>0.80</td>
<td>0.12</td>
</tr>
<tr>
<td>28-Mar-11*</td>
<td>4.1</td>
<td>5.8</td>
<td>945</td>
<td>142</td>
<td>2.03</td>
<td>3.28</td>
<td>0.61</td>
<td>0.88</td>
<td>0.15</td>
</tr>
<tr>
<td>30-Mar-11</td>
<td>6.6</td>
<td>4.8</td>
<td>864</td>
<td>142</td>
<td>1.70</td>
<td>3.05</td>
<td>0.57</td>
<td>0.79</td>
<td>0.10</td>
</tr>
<tr>
<td>15-Apr-11</td>
<td>5.8</td>
<td>9.8</td>
<td>857</td>
<td>116</td>
<td>1.63</td>
<td>2.97</td>
<td>0.44</td>
<td>0.82</td>
<td>0.08</td>
</tr>
<tr>
<td>31-May-11†</td>
<td>25.3</td>
<td>15.6</td>
<td>719</td>
<td>148</td>
<td>1.78</td>
<td>3.08</td>
<td>0.76</td>
<td>0.84</td>
<td>0.12</td>
</tr>
<tr>
<td>07-Jun-11†</td>
<td>21.9</td>
<td>12.8</td>
<td>735</td>
<td>158</td>
<td>1.85</td>
<td>3.13</td>
<td>0.77</td>
<td>0.88</td>
<td>0.08</td>
</tr>
<tr>
<td>10-Jun-11†</td>
<td>21.3</td>
<td>13.8</td>
<td>638</td>
<td>158</td>
<td>1.74</td>
<td>3.02</td>
<td>0.88</td>
<td>0.92</td>
<td>0.11</td>
</tr>
<tr>
<td>15-Jun-11†</td>
<td>21.3</td>
<td>14.1</td>
<td>731</td>
<td>228</td>
<td>1.84</td>
<td>3.11</td>
<td>0.80</td>
<td>0.90</td>
<td>0.10</td>
</tr>
<tr>
<td>05-Jul-11† †</td>
<td>24.2</td>
<td>17.3</td>
<td>713</td>
<td>298</td>
<td>1.85</td>
<td>3.05</td>
<td>0.87</td>
<td>0.91</td>
<td>0.09</td>
</tr>
<tr>
<td>14-Jul-11† †</td>
<td>26.6</td>
<td>17.7</td>
<td>734</td>
<td>288</td>
<td>1.84</td>
<td>3.10</td>
<td>0.88</td>
<td>0.95</td>
<td>0.07</td>
</tr>
</tbody>
</table>

*Closed Collector Channel
†Centrifugal Pump
††Centrifugal Pump, Lower Pressure Drop
Tests were conducted with the collector channel closed on February 15th and March 28th, 2011, using the fitted blocks of insulation from the Stagnation Prevention apparatus. When compared to the results obtained on February 10th and March 30th, the closed channel tests had better performance due to the reduction of the convective heat loss. While the open channel allowed the collector to absorb heat by convection when below the ambient temperature, it is beneficial to have a sealed, glazed collector when the system is operating above the ambient temperature, particularly during the winter months. Consequently, incorporating a valve into the channel, which would open if the collector was below ambient and close if above, was investigated through TRNSYS simulations in Section 5.4.3.

5.3.3 Step-Change in Collector Conditions

As described in Section 4.3.2, the performance of the system was observed when subjected to specific step-changes under quasi-steady weather conditions. The first step-change was to block the irradiance, such that the collector decreased in temperature and gained energy by convection. The heat transfer rates and climatic conditions are shown in Fig. 5-40, and it can be seen that $Q_{\text{cond}}$ and $Q_{\text{evap}}$ dropped to 48% and 43% of their previous values, respectively. While the collector was shaded, the ambient temperature increased by 1°C, causing a slight upwards drift in the heat transfer rates. The unsteady spikes during the test were caused by the expansion valve oscillating to modulate the refrigerant flow, ensuring the correct evaporator superheat.

The second step-change was to seal the collector channel to observe the change in efficiency. The climatic conditions and the resulting change in heat transfer rates when the channel was sealed are shown in Fig. 5-41. The ambient temperature dropped by 2°C while the channel was closed, yet little effect was seen on the heat transfer rates as the collector was less
sensitive to the ambient temperature when sealed. $Q_{\text{cond}}$ and $Q_{\text{evap}}$ increased by 8% and 9%, respectively.

Modest gains were achieved by sealing the collector channel, yet having the open channel allows for significant useful energy gains in the absence of sufficient solar irradiance. This supports the proposition for a thermally controlled valve, which would maximize the useful energy gain whether the collector is operating above or below ambient temperature.

Fig. 5-40: ISAHP response to shading at solar noon, showing a decline in heat transfer
5.3.4 Operational Limits

During certain day-long tests, two effects were observed that limited the operation of the ISAHP: cavitation in the collector loop, and triggering of the compressor safety shut-off switch. Cavitation in the collector loop occurred when the temperature of the glycol-water mixture was low; the specific temperature at which this occurred depended on the charge pressure of the collector loop and flow rate. The viscosity of the glycol-water mixture is highly sensitive to changes in temperature. When the glycol-water mixture became cold, the pressure loss through the system was increased with the rise in viscosity. As a result, the static pressure at the inlet to the pump decreased sufficiently to cause the pump to cavitate, such that bubbles were observed through the flexible tubing. During preliminary system checks at 77 kg/hr, the charge pressure of the collector loop was 15 PSI, and cavitation was observed when the evaporator inlet temperature decreased below 5°C. The system was recharged to 30 PSI, and care was taken to ensure that expansion tank had a reserve of fluid to accommodate a decrease in system temperature, yet
sufficient remaining capacity for the increase in glycol-water volume under stagnation conditions. The evaporator inlet temperature at which cavitation occurred was then reduced to -8°C. Therefore, a temperature of -5°C was chosen as the lower temperature limit described as a stopping condition in Section 4.3.2 to ensure no cavitation occurred.

The flow rate through the collector loop also affected the temperature at which cavitation occurred, as a lower flow resulted in less pressure drop through the collector loop. Consequently, the November 17th, 2010 test was run at 55 kg/hr to avoid cavitation from the cold operation. At the 142 kg/hr flow rate, cavitation was observed when the evaporator inlet temperature dropped below 0°C, which resulted in a reduction of the operational range of the system. The stop criterion was adjusted accordingly for the higher flow rate. The highest speed of 298 kg/hr was achieved without cavitation by increasing the diameter of the collector loop tubing, reducing the pressure loss.

The second operational limit was the deactivation of the compressor when its safety switch was triggered by excess outlet pressures. While the goal of maintaining the temperature of the water entering the condenser below 30°C was keep the compressor within safety limits, some conditions still occurred below this temperature that caused the compressor to cycle off and on. The compressor was observed to deactivate once the high side pressure, $P_2$, exceeded 1730 kPa (250 PSI), and would reactivate once the pressure fell below 1250 kPa (180 PSI), as shown in Fig. 5-42 for the afternoon of the September 17th, 2010 test.
Fig. 5-42: Compressor outlet temperature and pressure for September 17\textsuperscript{th}, 2010, showing compressor shut-off from excess pressure.

This event was observed on two of the test days, September 2\textsuperscript{nd} and September 17\textsuperscript{th}, 2010. In each case, the temperature of the water at the bottom of the tank was at 24°C when the compressor safety switch was triggered. Due to the low frequency of occurrence, the stop criterion for the condenser inlet temperature was not reduced from 30°C to 24°C. However, in the development of a commercial product based upon this prototype, a control system would be necessary to deactivate the compressor and collector loop pump together before the high pressure limit is reached. It is possible that the inclusion of hot water draws could eliminate this issue by reducing the storage tank temperature, and by association, the condenser pressure.

5.4 Simulation Results

Simulations of the ISAHP performance were completed using an updated version of the TRNSYS model created by Freeman [9], and modified by Bridgeman [10]. To ensure that the collector model and the resulting characteristic equation for the hybrid collector were valid, the
simulated results were compared to the collector data collected from the experiments in Section 5.3. The TRNSYS model was checked against the system performance from the day-long storage tank charge tests to ensure it was configured correctly for subsequent annual simulations. These simulations were used to determine the relative performance of the system using different collector types under identical operating conditions.

5.4.1 Validation of Collector Model

The modified COLSIM model for the hybrid collector was validated by using the irradiance, inlet temperature, ambient temperature, and mass flow rates from the experimental tests as inputs in the parametric table in EES. The resulting modeled efficiency or CPF was compared to the calculated efficiency or CPF from the experimental results, shown in Fig. 5-43 for June 15th, 2011. Over a majority of the day, a slight over-prediction of 1.7% is shown; however it is well within experimental uncertainty. The discrepancy during the start of the day is due to the experimental results being affected by the thermal capacitance of the system, with components initially at equilibrium with the indoor and outdoor ambient temperatures.

Such a strong agreement was not seen on every test day. The efficiency/CPF comparisons for November 2nd and September 22nd, 2010 are shown in Fig. 5-44 and Fig. 5-45, where it can be seen that the simulated values over-predict the above-ambient results by 10% and under-predict the below-ambient results by 14%. It was hypothesized that the discrepancy could be caused by an increased heat transfer coefficient due to wind flowing over the collector.
Fig. 5-43: Simulated and experimental collector performance, June 15th, 2011

Fig. 5-44: Simulated and experimental collector performance, November 2nd, 2010. Morning scatter in data due to shading from a smoke plume from a nearby chimney.
Cooling channels behind photovoltaic panels have been studied extensively by Brinkworth, including the induced flow through the channel due to wind effects [44]. Brinkworth modeled the natural convection flow through a cooling channel from a hydraulic standpoint, where the buoyancy induced pressure difference created a flow opposed by the major and minor hydraulic losses through the channel. Static pressures at the inlet and outlet of the channel were modified by the wind speed, knowing the coefficient of pressure for wind flowing over the panels, which would increase or decrease the channel flow. The determination of the coefficient of pressure as a function of wind speed and direction is complex and beyond the scope of this study. However a series of static pressure measurements at the openings of the collector were recorded and qualitatively compared to the southerly component of the wind speed, in Appendix H. A correlation between the wind speed and the pressure difference was observed, and it was concluded that the wind could indeed induce mixed flow through the channel, increasing the value of \( U_{\text{back}} \). Consequently an extra coefficient, \( HCW \), was added to the COLSIM model to account for the possibility of increased convection by the wind-induced flow. This value was

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**Fig. 5-45: Simulated and experimental collector performance, September 22\textsuperscript{nd}, 2010**

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applied for each test day that experienced the performance discrepancy, in order to determine the magnitude of the possible wind effect. HCW values ranged from 0 to 4 W/m$^2$K; however most tests were between 1 and 2 W/m$^2$K. Fig. 5-46 and Fig. 5-47 show the results from the COLSIM model with these modifications applied, where the discrepancy was reduced below 4%. For the subsequent annual simulations, an HCW value of 4 W/m$^2$K was chosen to represent a worst-case high-wind scenario. The actual performance of the hybrid collector would fall somewhere between the simulated results for HCW of 0 and 4 W/m$^2$K.

Due to the difference in the collector heat loss coefficients for above and below ambient operation in Table 5-1, a single curve fit for equation 3-45 could not be applied to the entire range of operating conditions. However, a curve fit could be applied separately for below and above ambient operation. The resulting coefficients for these curves are given in Table 5-3, evaluated at the same conditions from the SRCC test for the unmodified EnerWorks “HeatSafe™” Collector, Appendix I. It was therefore necessary to include two collector Types in the TRNSYS model to represent the above and below ambient characteristics, and an ‘IF’ statement to control which outputs were passed on to the balance of the system, shown in Appendix C. The switch between the curve fits was seen to occur at a temperature difference-irradiance ratio of -0.02 K-m$^2$/W for all flow rates in the simulation range.

**Table 5-3: Efficiency or CPF Curve Fit Coefficients**

<table>
<thead>
<tr>
<th>HCW</th>
<th>a$_0$</th>
<th>a$_1$</th>
<th>a$_2$</th>
<th>a$_0$</th>
<th>a$_1$</th>
<th>a$_2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.708</td>
<td>4.255</td>
<td>-0.0105</td>
<td>0.689</td>
<td>6.991</td>
<td>0.0152</td>
</tr>
<tr>
<td>2</td>
<td>0.653</td>
<td>6.624</td>
<td>-0.0065</td>
<td>0.635</td>
<td>8.481</td>
<td>0.0161</td>
</tr>
<tr>
<td>4</td>
<td>0.595</td>
<td>8.831</td>
<td>-0.0031</td>
<td>0.585</td>
<td>10.468</td>
<td>0.0123</td>
</tr>
</tbody>
</table>
Fig. 5-46: Simulated and experimental collector performance, November 2\textsuperscript{nd}, 2010.
Collector model adjusted with $HCW = 3$ W/m$^2$K. Morning scatter in data due to shading from a smoke plume from a nearby chimney.

Fig. 5-47: Simulated and experimental collector performance, September 22\textsuperscript{nd}, 2010.
Collector model adjusted with $HCW = 4$ W/m$^2$K
5.4.2 Day-long Storage Tank Charge Simulations

Simulations were run for the day-long tests with the updated TRNSYS model supplied with weather files built from the experimental data. The load draws and auxiliary heater were deactivated, and the resulting simulated heat transfer rates were compared to the experiments, with examples shown in Fig. 5-48 and Fig. 5-49. It can be seen that the heat transfer rates from the simulation compare well to the experimental results, for both an overcast day at 77kg/hr, and a clear day at 116 kg/hr. The $Q_{\text{evap}}$ for the April 15th, 2011 simulation is slightly under-predicted by 8%, which was caused by a lower average $UA$ value than what is used in the heat pump model. The days with the higher flow rates provided by the centrifugal pump could not be compared to simulations, due to their unsteady flow rate; the heat pump Type utilizes a look-up table for a specific collector loop flow rate. However, the overall agreement from the lower flow rates suggests the TRNSYS model was configured appropriately to conduct annual simulations.

![Graph](image)

**Fig. 5-48:** Simulated and experimental ISAHP performance, September 22nd, 2010.

Collector model adjusted with $HCW = 4 \text{ W/m}^2\text{K}$
5.4.3 Annual Simulations

The TRNSYS model was used to simulate the annual performance of the ISAHP prototype using different collectors in three Canadian cities: Toronto, Vancouver and Winnipeg. These cities were chosen based upon their respective climatic differences. While Winnipeg is typically sunny, and experiences very cold winters, Vancouver has a milder climate, with greater instances of cloud cover. Toronto was chosen as it falls in between, and represents the climate of a large portion of the Canadian population located in southern Ontario.

The collectors that were compared were a Heliocol unglazed flat plate, an EnerWorks “HeatSafe™” glazed flat plate, and the hybrid glazed flat plate. The unglazed collector has an area of 2.7 m$^2$, and the EnerWorks and hybrid collector have an area of 2.874 m$^2$. For consistency, an area of 2.874 m$^2$ was used for all simulations. A collector flow rate of 154 kg/hr was used based upon the observations from Section 5.3.2, and consequently, the minimum collector output temperature for the system controller was set at 0°C. In regards to the wind-
induced flow through the hybrid collector channel, a case with an $HCW$ value of 0 W/m²K was simulated along with a value of 4 W/m²K, in order to bracket the performance range. The $FER$ values from the simulations are shown in Fig. 5-50 through Fig. 5-52.

![Graph showing FER values for Toronto](image1)

**Fig. 5-50:** Monthly $FER$ values for Toronto

![Graph showing FER values for Vancouver](image2)

**Fig. 5-51:** Monthly $FER$ values for Vancouver
For Toronto, the collector which provided the highest FER was the base hybrid collector, at 0.548, while the unglazed, glazed and increased-loss hybrid collectors had values of 0.536, 0.534, and 0.512 respectively. The unglazed collector outperformed the base hybrid collector in Vancouver by a minimal margin of 0.569 to 0.568, while the increased-loss hybrid achieved a FER of 0.548. The glazed collector had the worst performance in Vancouver, at 0.537. On the contrary, the best collector for the Winnipeg climate was the glazed collector, at 0.549, followed closely by the base hybrid at 0.547. The unglazed and increased-loss hybrid collectors performed worse, at 0.514 and 0.503, respectively. It is important to note that the results for the unglazed collector were based upon a thermal performance test conducted by the SRCC, where the effect of high winds was not included. Such winds would have a large effect on the unglazed collector due to the lack of a insulated casing, reducing its performance. It should therefore be taken into consideration that the presented values for the unglazed collector are a best-case scenario, and the actual performance of the unglazed collector would be penalized similarly to the hybrid collector.
The glazed collector performed the best during cold, sunny conditions, while the hybrid and unglazed collectors performed better in warmer conditions, even when overcast. Compromises occurred for each collector when operating outside their preferred conditions. The goal of this study was to determine which collector had the best average performance through all conditions; however a different collector performed the best for each city. It was proposed that incorporating a thermally controlled valve into the channel of the hybrid collector could increase the annual performance, as the collector could change between the glazed and hybrid configurations automatically, maximizing its performance and utilization. As such, simulations were run for a valved configuration, where the hybrid and glazed collector types had identical inputs. An ‘IF’ statement compared the resulting output temperatures, and the information from the collector with the maximum output was passed on, as shown in Appendix C. The performance of the valved collector, with and without the wind-induced losses, was compared to the best collector for each city, in Fig. 5-53 through Fig. 5-55.

In Toronto, the base valved collector outperformed all others, with an $FER$ of 0.558, while the increased loss version achieved 0.544. The lower performance from the valved collector still outperformed both the glazed and unglazed types. A similar trend was observed in Winnipeg, where both valved versions outperformed all other types, at 0.559 and 0.551 for the base and increased loss versions. In Vancouver, the base valved collector outperformed all others with an $FER$ of 0.576, while the increased loss version had 0.553, outperforming the glazed collector.
Fig. 5-53: Monthly FER values of the valved collector for Toronto

Fig. 5-54: Monthly FER values of the valved collector for Vancouver
Fig. 5-55: Monthly FER values of the valved collector for Winnipeg

A summary of the annual results for the three cities is shown in Fig. 5-56 and Table 5-4 for all collector types.

Fig. 5-56: Annual FER values for the different collector types in each city
Table 5-4: Summary of Simulated Results

<table>
<thead>
<tr>
<th>Index</th>
<th>Collector</th>
<th>Glazed</th>
<th>Unglazed</th>
<th>Hybrid (HCW=0)</th>
<th>Hybrid (HCW=4)</th>
<th>Valved (HCW=0)</th>
<th>Valved (HCW=4)</th>
</tr>
</thead>
<tbody>
<tr>
<td>FER</td>
<td></td>
<td>0.534</td>
<td>0.536</td>
<td>0.548</td>
<td>0.512</td>
<td><strong>0.558</strong></td>
<td>0.544</td>
</tr>
<tr>
<td>COP</td>
<td></td>
<td>2.720</td>
<td>2.776</td>
<td><strong>2.815</strong></td>
<td>2.736</td>
<td>2.763</td>
<td>2.722</td>
</tr>
<tr>
<td>η or CPF</td>
<td></td>
<td>0.758</td>
<td>1.007</td>
<td>0.650</td>
<td>0.776</td>
<td>0.701</td>
<td>0.859</td>
</tr>
<tr>
<td>E Cond [kWh]</td>
<td></td>
<td>3909</td>
<td>3766</td>
<td>3776</td>
<td>3631</td>
<td><strong>3983</strong></td>
<td>3959</td>
</tr>
<tr>
<td>FER</td>
<td></td>
<td>0.537</td>
<td>0.569</td>
<td>0.568</td>
<td>0.548</td>
<td><strong>0.576</strong></td>
<td>0.553</td>
</tr>
<tr>
<td>COP</td>
<td></td>
<td>2.689</td>
<td>2.733</td>
<td><strong>2.811</strong></td>
<td>2.700</td>
<td>2.725</td>
<td>2.696</td>
</tr>
<tr>
<td>η or CPF</td>
<td></td>
<td>0.847</td>
<td>1.261</td>
<td>0.691</td>
<td>0.921</td>
<td>0.779</td>
<td>0.997</td>
</tr>
<tr>
<td>E Cond [kWh]</td>
<td></td>
<td>4239</td>
<td>4369</td>
<td>4140</td>
<td>4226</td>
<td><strong>4421</strong></td>
<td>4267</td>
</tr>
<tr>
<td>FER</td>
<td></td>
<td>0.549</td>
<td>0.514</td>
<td>0.547</td>
<td>0.503</td>
<td><strong>0.559</strong></td>
<td>0.551</td>
</tr>
<tr>
<td>COP</td>
<td></td>
<td>2.802</td>
<td>2.810</td>
<td><strong>2.835</strong></td>
<td>2.783</td>
<td>2.821</td>
<td>2.804</td>
</tr>
<tr>
<td>η or CPF</td>
<td></td>
<td>0.682</td>
<td>0.817</td>
<td>0.598</td>
<td>0.650</td>
<td>0.644</td>
<td>0.743</td>
</tr>
<tr>
<td>E Cond [kWh]</td>
<td></td>
<td>4137</td>
<td>3750</td>
<td>3972</td>
<td>3656</td>
<td><strong>4176</strong></td>
<td>4143</td>
</tr>
</tbody>
</table>

As mentioned in Section 5.3.4, the minimum collector output temperature was restricted by the pressure drop through the collector loop in order to prevent cavitation. Consequently, raising the flow rate from 77 kg/hr to 154 kg/hr resulted in an increase of the minimum collector output temperature to 0°C from -5°C. However, if a redesign of the collector loop was completed to reduce the hydraulic losses, a lower minimum temperature would be attainable. The effect this would have on the FER is shown in Fig. 5-57. It can be seen that decreasing the minimum temperature does not necessarily increase the system performance; while a lower allowable collector output temperature extended the time that the system operated, the lower temperatures led to a decreased COP as shown in Fig. 5-58. For Vancouver, the optimum temperature limit was 0°C. The optimum for Toronto and Winnipeg was -5°C, however switching to -5°C from 0°C only increased the FER by 0.3% and 0.6% respectively.
It is interesting to note that throughout all these simulations, the monthly $FER$ values are relatively constant from March through October. This was not due to an averaging effect, where higher $FER$ values for sunny days are balanced by lower values from overcast days during these months. As shown in Fig. 5-59, the daily values were limited to approximately 0.60. Because the compressor is always consuming power when the system is running, and the $COP$ of the heat pump limits to $FER$, not accounting for auxiliary and pump power, to:

$$FER = \frac{COP - 1}{COP}$$  \hspace{1cm} (5-10)

which, for an average $COP$ of 2.763 from the base valued collector operating in Toronto, was a limit of 0.638. The inclusion of the auxiliary and pump power, $E_{PUMP}$ and $E_{AUX}$, shown in Fig. 5-60, would reduce this limit, bringing the $FER$ in range of what was observed in the annual simulations.
Fig. 5-57: Variation in annual \textit{FER} with minimum collector output temperature, for the valved collector ($HCW = 0 \text{ W/m}^2\text{K}$)

Fig. 5-58: Variation in annual \textit{COP} with minimum collector output temperature, for the valved collector ($HCW = 0 \text{ W/m}^2\text{K}$)
Fig. 5-59: Daily FER values showing the upper limit due to COP effects, for Toronto with the valved collector ($HCW = 0$ W/m$^2$K)

Fig. 5-60: Daily combined pump and auxiliary energy requirements for Toronto with the valved collector ($HCW = 0$ W/m$^2$K)
Chapter 6
Discussion of Results

6.1 Collector Channel Results

Relationships for heat transfer coefficients as a function of temperature associated with the inclined channel within the EnerWorks “HeatSafe™” Heat Safe residential collector were required to model the thermal performance of the collector. Due to the channel geometry, including 90° bends at the openings and the presence of the serpentine heat transfer tube attached to the top surface of the channel, published relationships available in the literature were inappropriate. Consequently, the required relationships were derived from experimental results from tests conducted on the Stagnation Prevention apparatus.

6.1.1 Nusselt Number Relationships

The Nusselt number relationship observed for the above-ambient case, Series 2, follows a power-law curve as expected from previous studies [27] [28] [29]. Compared to the results from Azevedo and Sparrow [27], the experimental results fall within the same range, but do not agree with the curve in Fig. 6-1, as Azevedo and Sparrow’s results were for an isothermal plate condition.

A general relationship for the natural convective heat transfer for a cooled channel was not found in the literature, however it was seen in the experimental results that for a given Rayleigh number, the Nusselt number was higher in the cooled case, Series 4, than for Series 2. The increased heat transfer could be caused by the development of three-dimensional flow at the absorber surface. For natural convection under an inclined surface, the buoyancy force has a component that is normal and parallel to the surface. When heated, the normal component forces the fluid towards the plate, and maintains the development of a boundary layer flowing up the
incline. However, if the surface is cooled, the normal component of the buoyancy force pulls the fluid away from the surface, disrupting the boundary layer. To maintain continuity, warmer fluid circulates to the surface, reducing the boundary layer thickness and increasing heat transfer [45].

![Graph showing comparison of experimental Nusselt number from Series 2 to the relationship published by Azevedo and Sparrow.](image)

**Fig. 6-1: Comparison of experimental Nusselt number from Series 2 to the relationship published by Azevedo and Sparrow**

### 6.1.2 $U_{i-a}$ Relationships

The $U_{i-a}$ relationship, being the heat loss coefficient from the insulation to the surroundings as a function of the temperature difference between the insulation plane and the ambient, also followed a power-law curve for Series 2 (the heated absorber, open channel condition). When compared to the convection coefficients from the absorber surface, which reached just under 3.5 W/m²K at a temperature difference of 60°C, the $U_{i-a}$ values were higher, only needing a temperature difference of 40°C to surpass 3.5 W/m²K. This is because the $U_{i-a}$ values represented not only the convection off the insulation surface within the channel, but also the conduction through the back of the insulation. For fibrous insulation, the effective
conductivity increases with temperature, due to increased radiative exchange between the fibres [46]. Additionally, as the insulation was a heated, upwards facing surface it experienced the same three-dimensional flow effect as the downward facing cooled surface.

For Series 4 (the cooled absorber, open channel condition), the $U_{i-a}$ values were calculated to be nearly constant, remaining between near 1.4 W/m$^2$K and 1.46 W/m$^2$K over an 11°C temperature difference. The $U_{i-a}$ value was shown to decrease with an increasing temperature difference, which can be attributed to opposing factors. First, when cooled, the lower surface of the channel did not experience the increased heat transfer from three-dimensional effects, resulting in a lower convection coefficient than in the heated case, for the same temperature difference. The convection coefficient component of $U_{i-a}$ would increase with the magnitude of the temperature difference, however, lower absolute temperatures reduced the insulation conductivity, reducing the $U_{i-a}$ value.

6.2 ISAHP Prototype Results

The performance of the ISAHP prototype with the hybrid collector was observed while operating under real weather conditions for independent daily tests from July 2010 through July 2011. The results showed that the system was able to fully charge the storage tank for the majority of sunny days throughout the year, while a significant portion of the tank was charged on overcast and colder clear days.

6.2.1 Heat Transfer Rates

The observed evaporator heat transfer rates varied from 0.5 kW to 1.5 kW depending on flow rate, time of day and the weather conditions. The condenser heat transfer rates were observed 0.4 to 0.7 kW higher than the evaporator, due to the addition of the power input from the compressor. On clear days, the heat transfer rates through the heat pump increased with the solar irradiance, and typically remained lower than the available irradiance. However, at certain
points, namely the morning and evening when the irradiance was low, it was observed that the evaporator was absorbing more energy than what was supplied by the irradiance. It was in these cases that it became apparent the system was absorbing a significant portion of energy by convection with the surrounding air.

This effect was also seen on overcast days, where the heat transfer rates through the evaporator were consistently higher than the available irradiance. While the lack of solar input reduced the evaporator heat transfer rates below that for the sunny days, down to 0.75 kW, the performance on these days highlighted one of the main benefits of an ISAHP system: the ISAHP system could deliver free energy to the storage tank outside the operational range of an SDHW system. As opposed to a SDHW system, where the collector output must be above the tank temperatures to achieve heat transfer, the ISAHP could still accomplish heating with lower collector loop temperatures, as a result of the heat pump’s ability to upgrade the temperature of the heat transfer.

6.2.2 Coefficient of Performance

The two main variables which affected the COP of the heat pump were the input temperatures to the evaporator and condenser, on the collector and storage tank sides, respectively. Higher evaporator inlet temperatures were often related to high heat transfer rates, and as the compressor power did not increase by the same magnitude, the ratio between the two, being the COP, also increased.

The storage tank temperature has a large influence on the COP as well. As the tank was heated during the tests, the natural convection flow rate decreased. Even though the temperature of water entering the condenser remained constant through most of the day, the reduced flow rate led to higher outlet temperatures, and therefore higher average condenser temperatures. To be able to transfer heat to the warmer water, the refrigerant had to be compressed to a pressure which
had a saturation temperature higher than the water temperature. Compressing the refrigerant to these higher temperatures and pressures required additional compressor work for the same heat transfer rate, leading to a lower $COP$.

While winter days saw a decrease in evaporator temperatures, which would lead to a decrease in $COP$, the colder mains water caused lower storage tank and condenser temperatures. While the effects of these temperatures on the $COP$ oppose each other, a net increase in $COP$ was often observed, aiding the winter performance of the system.

### 6.2.3 Collector Efficiency and CPF

The collector efficiency and the Collector Performance Factor indicate the collector's performance as the ratio of the useful energy gain, which could be collected from multiple sources (i.e., solar irradiance and ambient convection), to the available solar irradiance. Daily average values of 0.44 to 4.28 were obtained. Values higher than the effective transmissivity-absorptivity product, $(\tau\alpha)_e$ of approximately 0.89, signified that the mean collector temperature was below the ambient temperature, and thermal gains were achieved by convection with the surroundings. For efficiencies below $(\tau\alpha)_e$ the mean collector temperature was greater than the ambient, resulting in thermal losses to the surroundings. In general, higher collector efficiencies were observed for the ISAHP simulations than for simulations of a comparable SDHW system conducted in a previous study, where a SDHW system operating in Montreal experienced an average annual collector efficiency of 0.383 [47]. These higher efficiencies for the ISAHP system were the result of the heat pump effectively cooling the collector loop.
6.2.4 Free Energy Ratio

By the definition in Eq. 5-9, values for the daily free energy ratio for the experimental storage tank charge tests were calculated between 0.41 and 0.95. These values are not comparable to the $FER$ values determined in the simulations due to their slightly different definitions. As no draws occurred during the experiments, the $FER_D$ value was the ratio of free energy delivered to the tank to the energy required to heat 225 L of water from the mains temperature to the set point of 51.7°C. As the tank size was 270 L, and there was fixed limit on the temperature delivered to the tank, more energy could be delivered to the tank during one day than would be required to meet the load. In the simulations, the values were calculated based upon the energy removed from the tank during a draw, and accounted for the mechanical and auxiliary power involved with meeting the load. While the experimental values cannot be directly compared to simulated results, they were useful as a comparison between the storage tank charge tests themselves, indicating which conditions led to better system performance.

6.2.5 Evaporator Performance and Heat Pump Capacity

The evaporator experienced poor performance when the heat pump was operating over capacity. The poor performance manifested itself in what appeared to be decreased evaporator $UA$ values, calculated in section 5.3.2. When the heat pump was beyond capacity, the system was unable to remove sufficient heat from the collector loop, as the compressor could not circulate enough refrigerant. As a result, the evaporator increased in temperature, even beyond that of the condenser. Under such conditions, the refrigerant would enter the evaporator with a high quality, reducing the heat transfer that occurred with a latent phase change of the refrigerant. As a result, the refrigerant side could become the limiting factor for the effectiveness of the now ‘dry’ evaporator, as it would no longer benefit from the increased heat transfer that occurs with the near-infinite heat capacity associated with the phase change.
There is a possibility that these adverse conditions could be avoided entirely. If the collector loop temperatures are greater than the condenser temperature, then there is no need to operate the heat pump, as heat transfer could occur directly between the collector loop and storage tank, by-passing the heat pump. A system with this capability could experience a higher annual \textit{FER}, as the less purchased power would be necessary if the compressor was not always operating.

\textbf{6.2.6 Flow Rate}

As was mentioned in the results, increasing the collector loop flow rate led to increased system performance. Higher flow rates led to higher collector efficiencies because of decreased heat loss with a smaller temperature rise over the collector. Therefore more energy was available to be delivered to the evaporator. The increased flow rate also aided in lowering the evaporator temperature below that of the condenser, reducing the frequency that the heat pump was operating in abnormal conditions.

\textbf{6.3 Simulated Performance}

TRNSYS simulations allowed the variations in the ISAHP system to be evaluated under identical climatic conditions, as opposed to the experimental results, where the changes to the system were evaluated under varying conditions. Therefore the simulations allowed for the effect of the variations to be isolated, yet were subject to assumptions and approximations typically associated with modeling.

\textbf{6.3.1 Monthly Performance}

Each city exhibited different performance trends throughout the year, due to the different climates. In all cities, the \textit{FER} values between April and October were relatively constant, and
minor differences between collector types were observed. It was during the winter months that
the difference in location and collector types became apparent.

In Vancouver, with its mild winter temperatures, only slight decreases for each collector
type were observed. The glazed collector had the worst drop in performance, as it was unable to
take advantage of the mild winter temperatures when the irradiance levels decreased. The
unglazed and base hybrid collectors on the other hand showed little decrease, as they were able to
absorb sufficient energy from the ambient air.

In Winnipeg, the extreme cold in the winter months caused significant decreases in
performance for all collector types. The unglazed and increased-loss hybrid collector saw a
decrease in the monthly $FER$ of 0.30 due to their higher heat loss coefficients, while the glazed
collector, with its increased insulation, had the lowest decrease of 0.11.

A similar trend was observed in Toronto as for Winnipeg, however with less extreme of a
performance drop in the winter. The glazed collector still performed best during the winter
months, but lacked the ability to absorb energy from the warmer summer temperatures. While the
performance of the ISAHP was less sensitive to collector type in warmer climates, such as
Vancouver, it is very important to select an appropriate collector for regions that experience
prolonged periods of sub-zero temperatures.
6.3.2 Collector Improvements

It was seen that for each of the three Canadian cities studied, a different collector provided the best performance due to their respective abilities to maximize energy collection in the different climates. However, it was shown that a collector that could adapt to the different seasonal conditions could yield further improvements in performance. The concept of the valved collector is promising, and should be investigated further.

6.3.3 System Improvement

The experimental results highlighted the fact that as the glycol-water mixture decreased in temperature, its rise in viscosity caused an increased pressure drop through the collector. At certain temperatures, this pressure drop became large enough to cause cavitation. Consequently, a lower collector output temperature limit was imposed on the system, which was dependent on the flow rate. Rebuilding the collector loop to reduce the pressure loss showed that lower temperatures could be obtained at higher flows, before cavitation occurred. In effect, this would extend the operational range of the system. However, the simulations for varying minimum collector output temperature showed that there existed an optimum temperature for each city. Even though the system could operate for longer periods, it would do so under increasingly poorer conditions for the heat pump, decreasing the observed annual $COP$.

While the $FER$ values were fairly consistent throughout most of the year for all three cities, a significant increase in $FER$ was not observed during the summer months. During the summer, an SDHW system can achieve very high monthly $FER$ values, in the 0.9 to 1.0 range, requiring very little auxiliary power [9]. It was shown that the auxiliary power requirements for the ISAHP were practically non-existent between April and October; however the $FER$ remained near 0.6. This was because the compressor was always running when the system was collecting energy, and the performance was limited by the $COP$ of the heat pump. While some of the
electrical energy consumed by the compressor supplants the auxiliary needs, the ISAHP may be consuming more electrical energy than is required to meet the load, during warm days with high irradiance. Under such conditions it may be beneficial to by-pass the heat pump circuit and deliver heat directly to the storage tank, in effect creating an SDHW system. Therefore the compressor would only be used when necessary, increasing summer $FER$ values. This concept has been used in a previous study [16], and resulted in a 28.7% increase in the daily $COP$ for a clear day. It would be beneficial to use the valved collector to maintain the higher collector temperatures necessary in the SDHW mode, while also allowing the collector to have increased gains when operating below the ambient in the ISAHP mode.
Chapter 7

Conclusions and Recommendations for Future Work

7.1 Conclusions

An Indirect Solar Assisted Heat Pump prototype, previously built for controlled laboratory tests with constant temperature and simulated solar inputs, was commissioned with a novel hybrid solar collector to determine its performance under actual operating conditions. The performance of the system was investigated for clear and overcast days between July 2010 and July 2011 in Kingston, Ontario. The tests were also used to highlight operational limits of the system, as well as validate a numerical model for the performance of the hybrid collector. With the model of the hybrid collector complete, annual simulations of the system were performed in TRNSYS to compare the performance of various collectors for Toronto, Vancouver and Winnipeg.

Experimental results varied depending on the climatic conditions and the collector loop flow rate. For clear days with a 77 kg/hr flow rate, daily average COPs ranged from 2.54 to 2.83, while the effective collector efficiency ranged from a minimum of 0.47 in the winter to a maximum of 0.75 in the autumn. On overcast days, the heat pump would cool the collector loop below the ambient temperature, such that significant gains would results from the collector absorbing energy by convection with the surrounding air. CPF values upwards of 4.28 were observed, while the lower collector temperatures resulted in decreased daily average COPs, between 2.24 and 2.44.

Increasing the collector loop flow rate up to 154 kg/hr led to better performance, with higher COPs between 2.95 and 3.13 and daily average efficiencies upwards of 0.88 for clear days. This increase was attributed to two factors: lower temperature increases over the collector which reduced heat losses; and higher convection coefficients in the evaporator. Also, at the
higher flow rates, the lower temperature rise helped the average evaporator temperature remain lower than the average condenser temperature, ensuring the heat pump remained in normal operational conditions. Under certain high irradiance conditions, the output from the collector surpassed the capacity of the heat pump. Consequently, decreased heat transfer rates were observed due to reduced evaporator $UA$ values. By-passing the heat pump loop during these situations would allow for the delivery of energy to the storage tank without the consumption of the additional power required to run the compressor.

Aside from the over-capacity conditions, two other operational limits were observed. First, a lower collector output temperature limit was determined from associated pressure losses through the collector loop. While this lower limit could be decreased by reducing the hydraulic resistance of the collector loop, results showed that the optimum temperature limit varied between 0°C and -5°C, depending on the system location. However, a reduced pressure drop would result in decreased pump power requirements. The second observed limit was the high pressure compressor shut-off, which occurred when the condenser pressure reached 250 PSI. While this occurred with low frequency, and may be eliminated by draws by the load, or by-passing the heat pump, the inclusion of a collector pump shut-off associated with the compressor shut-off would be required to prevent damage to the system from excess temperatures.

Experimental results from an instrumented collector channel were used to determine the heat transfer coefficients from the rear of the absorber in the hybrid collector. Nusselt number relationships were determined for above and below ambient operation for use a modified version of the COLSIM flat plate collector model initially developed by Stephen Harrison. The results from the collector model were compared and validated with experimental data collected from the ISAHP prototype. The resulting collector characteristic equation was used in an updated version of the TRNSYS model developed by Grant Freeman and Andrew Bridgeman.
The system model showed agreement with the experimental results that were not affected by reduced evaporator $UA$ values. Subsequent annual simulations showed that different collectors yielded the highest performance in the three cities investigated, due to their respective climates. The hybrid, unglazed, and glazed collectors performed the best in Toronto, Vancouver, and Winnipeg, respectively, with $FER$s of 0.548, 0.569 and 0.549. The incorporation of a thermally controlled valve into the hybrid collector was proposed, and simulations showed an increase in $FER$ to 0.558, 0.576 and 0.559 for Toronto, Vancouver and Winnipeg, outperforming all other collector types.

### 7.2 Recommendations for Future Work

Improvements in the seasonal performance were shown for using a valved hybrid collector; however the $FER$ of the system was limited by the $COP$ of the heat pump. The utilization of a more efficient compressor should be investigated to raise this limit. The development of a system configuration and control scheme to by-pass the heat pump circuit when it is not required to deliver energy to the storage tank could also lead to further increases in the annual performance of the system.

The effect of load draws should be investigated to assess how the heat pump adapts to the reducing condenser temperatures before conducting a full year continuous operation test of a controlled system. The construction and use of a prototype valved hybrid collector in this full year test could confirm its potential to increase the system performance.

While the simulations showed agreement with the experimental results which experienced evaporator $UA$ values near the 0.12 kW/°C assumed in the heat pump model, further accuracy could be obtained through a comprehensive characterization of the evaporator for varying level of superheat, flow rates, and glycol temperatures. Such an evaluation would be useful in properly sizing the heat pump components to prevent over-capacity operation.
References


Appendix A: COLSIM, TOPL, and BACKL Flow Charts

Flowcharts outlining calculation procedure in the COLSIM program, and the TOPL and BACKL subroutines are shown in Fig. A-1 through Fig. A-3. Information on the additional subroutines mentioned in the flowcharts can be found in the work by Harrison [23].
Fig. A-1: Flowchart overview of the COLSIM program, modified from Harrison [23]
Fig. A-2: Flowchart overview of the TOPL Subroutine, modified from Harrison [23]
Fig. A-3: Flowchart overview of the BACKL Subroutine
Appendix B: COLSIM EES Code and Sample Output

"COLSIM Collector Performance Model
Original Author: Stephen Harrison, 1982
Adapted from FORTRAN for use in EES
Modified By: Bryn Elliott, 2011
Modified to model a hybrid collector with a open channel behind the absorber"

PROCEDURE tsky(TA,EMG,TILT;RS,TS)
"SUBROUTINE TO CALCULATE THE EFFECTIVE SKY TEMPERATURE OF THE
ENVIRONMENT AS SEEN BY A SURFACE INCLINED AT AN ANGLE 'TILT'
TO THE HORIZONTAL.

TA - AMBIENT AIR TEMPERATURE, (DEG. KELVIN)
EMG - TOTAL EMMITTANCE OF THE GROUND TO LONGWAVE RADIATION
TILT - TILT ANGLE OF SURFACE TO HORIZONTAL (DEG.)
SLOPE - TILT ANGLE IN RADIANS
THETA2 - REFRACTION ANGLE (RADIANS)
RA - ATMOSPHERIC RADIATION INCIDENT ON A HORIZ. SURFACE
RAS - ATMOSPHERIC RADIATION INCIDENT ON INCLINED SURFACE
RGS - LONGWAVE RADIATION ON SURFACE DUE TO THE GROUND
RS - TOTAL LONGWAVE RADIATION INCIDENT ON SURFACE
K1 - CONSTANT FOR RAS CALC. FROM COLE
K3 - CONSTANT FOR RAS CALC. FROM COLE
SIGMA - STEPHAN-BOLTZMAN CONSTANT
TS - EFFECTIVE SKY TEMPERATURE, (DEG. K)
TG - TEMPERATURE OF GROUND, ASSUMED = TA"

SIGMA=5.670E-08

"CALCULATE SLOPE ANGLE (RADIANS)"
SLOPE=TILT*pi/180.0

"CALCULATE RA"
RA=-119+1.06*SIGMA*(TA**4)

"CALCULATE RAS"
X=TILT
K1=1.00004-4.46622E-06*X-7.67489E-05*(X**2)+3.42187E-08*(X**3)+1.48096E-09*(X**4)
K3=-9.00303E-05+6.1022E-04*X+1.71856E-04*(X**2)-2.17531E-05*(X**3)+7.38927E-09*(X**4)
RAS=RA*K1+0.09*K3*SIGMA*(TA**4)

"CALCULATE RGS"
TG=TA
RGS=EMG*SIGMA*(TG**4)*((1.0-cos(SLOPE))/2.0)

"CALCULATE RS"
RS=RAS+RGS

"CALCULATE TS"
TS=(RS/SIGMA)**0.25

END

PROCEDURE chtpc(TPM,TC,PATM,LPC,ALFAP,TILT;HPC)
"THIS ROUTINE CALCULATES THE VALUE OF THE CONVECTIVE HEAT LOSS"
COEFFICIENT BETWEEN THE ABSORBER PLATE AND COVER.

SET INITIAL VALUES*
  SIGMA=5.670E-8
  SLOPE=TILT*pi/180.0

"CONVective HEAT Transfer COEFFICIENT (ABSORBER PLATE TO COVER)
AIR LAYER PROPERTIES (ASSUME AT A TEMPERATURE OF TALM)"
  TALM=(TPM+TC)/2.
  DTAL=TPM-TC
  XC=LPC
  XA=100./TALM

"CALCULATE RAYLEIGH NUMBER"
  RA=2737.*((1.+2.*XA)**2)*(XA**4)*DTAL*((100.*XC)**3)*(PATM**2)

"CALCULATE THERMAL CONDUCTIVITY OF AIR AT THE ASSUMED TEMPERATURE"
  KA=0.002528*(TALM**1.5)/(TALM+200.)

"CALCULATE THE NUSSELT NUMBER (BY TERMS)"
  IF(TPM>TC) THEN GOTO 100

"CONDUCTION ONLY CONDITION IF BELOW AMBIENT"
  X1=0.0
  X2=0.0
  X3=0.0

GOTO 110
100: X1=1.-(1708./(RA*cos(SLOPE)))
  IF(X1<0.) THEN X1=0.0
  X2=1.-((((sin(1.8*SLOPE))**1.6)*1708.)/(RA*cos(SLOPE)))
  X3=((((RA*cos(SLOPE))/5830.)**.333333)-1.0)
  IF(X3<0.) THEN X3=0.0
110:NU=1.0+(1.44*X1*X2)+X3

"CONVective HEAT Transfer COEFFICIENT (ABSORBER PLATE TO COVER)"
  HPC=NU*(KA/LPC)

END

PROCEDURE optcal(N,KLC,ANGLE:TRANSC,REFC,TRG)

*SUBROUTINE TO CALCULATE THE OPTICAL PROPERTIES OF THE GLAZING

N - INDEX OF REFRACTION FOR GLAZING
KLC - PRODUCT OF EXTINCTION COEF. & COVER THICKNESS
ANGLE - ANGLE BETWEEN GLAZING SURFACE NORMAL & BEAM RAD. (DEG.)
THETA1 - INCIDENT ANGLE (ANGLE) EXPRESSED IN RADIANS
THETA2 - REfraction ANGLE (RADIANS)
ROWPAR - AIR/GLASS INTERFACE REFLECTIVITY,PARALLEL COMP.
ROWPER - AIR/GLASS INTERFACE REFLECTIVITY, PERPENDICULAR COMP.
TRG - TRANSMITTANCE OF GLAZING DUE TO ABSORPTION ONLY
REFC - REFLECTANCE OF COVER FOR BOTH COMP. OF POLARIZATION
REFPA - REFLECTANCE FOR GLAZING, PARALLEL COMP
REFPE - REFLECTANCE FOR GLAZING,PERPENDICULAR COMP.
TRANPA - TRANS. OF GLAZING DUE TO ABSORP. AND REFLECTION
TRANPE - TRANS. OF GLAZING DUE TO ABSORP. AND REFLECTION
TRANSC - AVERAGE TRANSMITTANCE OF COVER

IMPLICIT REAL(A-H,J-Z)"

"CALCULATE INCIDENT ANGLE (RADIANS)"
IF (ANGLE>90.0) THEN
THETA1:=pi/2.
ELSE
THETA1:=ANGLE*pi/180.0
ENDIF

"CALCULATE REFRACTION ANGLE (SNELL'S LAW)"
THETA2=sin(THETA1)/N
TEMP=sqrt(1.-THETA2*THETA2)
THETA2=arctan(THETA2/TEMP)

"CALCULATE REFLECTIVITY AT AIR-GLAZING INTERFACE FOR PARALLEL"

"AND PERPENDICULAR COMPONENTS (FRESNEL'S FORMULA)"
RPE=sin(THETA1-THETA2)/sin(THETA1+THETA2)
RPA=RPE*(cos(THETA1+THETA2)/cos(THETA1-THETA2))
ROWPAR=RPA*RPA
ROWPER=RPE*RPE

GOTO 50

"FOR THE CASE THETA1=0, THEN THETA2=0 AND ROWPAR=ROWPER GIVEN BY:"

40:    THETA2=THETA1
ROWPAR=((N-1.)/(N+1.))**2
ROWPER=ROWPAR

"CONTINUE"

"CALCULATE TRANSMITTANCE OF GLAZING DUE TO ABSORPTION OF GLAZING"

50:   TRG=exp(-1.*KLC/cos(THETA2))
STRG=TRG*TRG

"REFLECTANCE VALUES (SOL'N OF STOKES EGNS FOR MULTIPLE REFLECTIONS)
FOR THE COVER."
REFPA=ROWPAR*(1.+(STRG*(1.-ROWPAR)**2)/((1.-ROWPAR)**2)*STRG))
REFPE=ROWPER*(1.+(STRG*(1.-ROWPER)**2)/((1.-ROWPER)**2)*STRG))

"AVERAGE VALUES OF REFLECTANCE FOR THE COVER"
REFC=(REFPA+REFPE)/2.

"TRANSMITTANCE OF GLAZING DUE TO ABSORPTION & REFLECTION"  
TRANPA=TRG*(((1.-ROWPAR)**2)/1.-((ROWPAR)**2)*STRG))
TRANPE=TRG*(((1.-ROWPER)**2)/1.-((ROWPER)**2)*STRG))

"AVERAGE TRANSMITTANCE FOR THE COVER"
TRANSC:=(TRANPA+TRANPE)/2.

END

PROCEDURE
topl(TOL,EMC,EMP,EMG,LPC,ALFAP,TRNLW,TILT,TPM,TA,HW:UTOPL,HRCA,HRPC,HPC,UR PS,UPC,UCA,TS,TC)

"THIS ROUTINE CALCULATES THE VALUE OF THE TOP LOSS COEFFICIENT
ESTIMATE MEAN ABSORBER TEMPERATURE AND CALCULATE HEAT TRANSFER RATES,
THEN ITERATE TO REFINE ESTIMATES TO WITHIN SET TOLERANCES.

SET INITIAL VALUES"

PATM=1.0
SIGMA=5.670E-8

90:   TG=(TPM+TA)/2.0
      CALL TSKY(TA,EMG,TILT:RS,TS)

"HEAT TRANSFER COEFFICIENT (COVER TO SURROUNDING ENVIRONMENT)"
\[ HW = 5.7 + (3.8 \times VW) \]
\[ HW = 4.55 + 3.71 \times (VW^{0.63}) \]
\[ DCA = (TC - TA) \]
\[ \text{IF} (DCA < 0.0) \text{ THEN } DCA = 0.0 \]

\[ I = 0 \]

100: \[ HRCA = EMC \times \text{SIGMA} \times ((TC^{4.0} - (TS^{4.0}))/DCA) \]
\[ \text{IF} (HRCA < 0.0) \text{ THEN } HRCA = -1 \times HRCA \]
\[ UCA = HW + HRCA \]

**HEAT TRANSFER COEFFICIENT (ABSORBER PLATE TO COVER)**

**CALCULATE CONVECTIVE HEAT TRANS. COEFF., PLATE TO COVER**

\[ \text{CALL CHTPC}(TPM, TC, PATM, LPC, ALFAP, TILT; HPC) \]
\[ HRPC = \text{SIGMA} \times ((TPM^{2.0} + TC^{2.0}) \times (TPM + TC))/((1./\text{EMP}) + (1./\text{EMC}) - 1.) \]

**OVERALL HEAT TRANSFER COEFFICIENT (ABSORBER PLATE TO COVER)**

\[ UPC = HPC + HRPC \]

**LONG WAVE RADIATION HEAT LOSS (ABSORBER PLATE TO SKY)**

\[ URPS = \text{TRNLW} \times \text{EMP} \times \text{SIGMA} \times ((TPM^{4.0} - (TS^{4.0}))/\text{TPM - TA}) \]
\[ \text{IF} (URPS < 0.0) \text{ THEN } URPS = -1 \times URPS \]

**OVERALL TOP LOSS HEAT TRANSFER COEFFICIENT (PLATE TO ATMOSPHERE)**

\[ UTOPL = (1.0/(1.0/\text{UPC}) + (1.0/\text{UCA})) + URPS \]

******** CHECK IF ASSUMED COVER TEMPERATURE IS WITHIN TOLERANCES *********

**IF NOT, SET ASSUMED COVER TEMPERATURE TO THE CALCULATED VALUE AND ITERATE AGAIN UNTIL VALUE IS WITHIN TOLERANCES.**

\[ TCC = TPM - ((UTOPL \times (TPM - TA))/\text{UPC}) \]
\[ \text{ERR} = \text{abs}(TCC - TC) \]
\[ I = I + 1 \]
\[ \text{IF} (\text{ERR} < \text{TOL}) \text{ THEN GOTO 200} \]
\[ TC = TCC \]
\[ \text{GOTO 100} \]

200: **END**

**PROCEDURE staeff(N,KLC,ALFAP,ANGLE,TILT,FRACD,REFG,UTOPL,UPC:TAEFF)**

**SUBROUTINE TO CALCULATE THE OPTICAL PROPERTIES OF THE GLAZING**

**N** - INDEX OF REFRACTION FOR GLAZING

**KLC** - PRODUCT OF EXTINCTION COEFF. & COVER THICKNESS

**ANGLE** - ANGLE BETWEEN GLAZING SURFACE NORMAL & BEAM RAD. (DEG.)

**TILT** - SOLAR COLLECTOR TILT IN DEGREES

**FRACD** - FRACTION OF DIFFUSE RADIATION ON THE COLLECTOR TILT

**REFG** - GROUND REFLECTANCE TO SOLAR RADIATION

**UTOPL** - HEAT LOSS COEFF., ABSORBER PLATE TO ATMOSPHERE

**UPC** - HEAT LOSS COEFF., ABSORBER PLATE TO COVER

**TRANSC** - AVERAGE TRANSMITTANCE OF COVER

**TAEFF** - EFFECTIVE TRANSMITTANCE ABSORPTANCE PRODUCT

**TRG** - TRANSMITTANCE OF GLAZING DUE TO ABSORPTION ONLY

**REFC** - REFLECTANCE OF COVER FOR BOTH COMP. OF POLARIZATION

**AGESD** - EFFECTIVE INCIDENCE ANGLE FOR SKY DIFFUSE

**AGEGD** - EFFECTIVE INCIDENCE ANGLE FOR GROUND DIFFUSE

**TEFFD** - EFFECTIVE TRANSMITTANCE TO GROUND AND SKY DIFFUSE

\[ SLOPE = \text{TILT} \times \pi/180.0 \]
\[ \text{GSF} = ((1.-\cos(SLOPE))/2.) \]
\[ \text{SSF} = ((1.+\cos(SLOPE))/2.) \]
"TRANSMISSION OF GLAZING TO BEAM RADIATION"
CALL OPTCAL(N,KLC,ANGLE:TRANSC,REFC,TRG)
ABSEFB=(1.-TRG)

"CALCULATE TRANS. OF GLAZING TO SKY AND GROUND DIFFUSE RADIATION SKY SOURCE DIFFUSE"
EFFECTIVE INCIDENCE ANGLE FOR SKY DIFFUSE
AGESD=59.68-.1388*TILT+0.001497*(TILT**2.)

"CALCULATE TRANSMITTANCE TO SKY DIFFUSE"
CALL OPTCAL(N,KLC,AGESD:TRCSD,REFCSD,TRGSD)

"GROUND SOURCE REFLECTED DIFFUSE"
EFFECTIVE INCIDENCE ANGLE FOR GROUND DIFFUSE
AGEGD=90.-.5788*TILT+0.002693*(TILT**2.)

"CALCULATE TRANSMITTANCE TO GROUND SOURCE DIFFUSE"
CALL OPTCAL(N,KLC,AGEGD:TRCGD,REFCGD,TRGGD)

"EFFECTIVE TRANSMITTANCE TO GROUND AND SKY DIFFUSE"
TEFFD=(TRCSD*SSF+REFG*TRCSD*GSF)/(SSF+REFG*GSF)

"EFFECTIVE DIFFUSE TRANS. OF GLAZING DUE TO ABSORPTION ONLY"
TRGEFD=(TRGSD*SSF+REFG*TRGGD*GSF)/(SSF+REFG*GSF)
ABSEFD=1.-TRGEFD

"TRANSMITTANCE X ABSORPTANCE PRODUCT"
Hemispherical Diffuse Radiation Properties of Glazing
HDANGL=60.0
CALL OPTCAL(N,KLC,HDANGL:HDTRAN,HDREFC,HDTRG)
TRNABD=(TEFFD*ALFAP)/(1.-(1.-ALFAP)*HDREFC)
TRNABB=(TRANSC*ALFAP)/(1.-(1.-ALFAP)*HDREFC)

"EFFECTIVE TRANSMITTANCE X ABSORPTANCE PRODUCT"
TAEFFD=TRNABD+ABSEFD*(1.-UTOPL/UPC)
TAEFFB=TRNABB+ABSEFB*(1.-UTOPL/UPC)
TAEFF=((1.-FRACD)*TAEFFB+FRACD*TAEFFD)

END

PROCEDURE backl(TOL,TPM,TA,TILT,PATM,EMPR,EMI,S,HCW,TIM,UBACKL)

*** THIS SUBROUTINE CALCULATES THE BACK HEAT LOSS COEFFICIENT ***
FROM THE ABSORBER SHEET TO THE OPEN AIR CHANNEL, BASED ON EXPERIMENTAL RESULTS"
TPM=TPM+273.15
TA=TA+273.15

SIGMA=5.67E-8 [W/m^2-K^4]
grav=9.81 [m/s^2]
SLOPE=TILT*pi/180.0 [rad]

"GUESS INSULATION PLANE TEMPERATURE"
"GUESS INITIAL INSULATION PLANE TEMPERATURE"
TIM=0.73*(TPM-TA)+TA
I=0

"EVALUATE FILM TEMPERATURE IN CHANNEL: AVERAGE OF AVERAGE CHANNEL WALL TEMPERATURE AND AMBIENT"
100:TFP=((TPM+TIM)/2+TA)/2
"EVALUATE PROPERTIES BASED ON FILM TEMPERATURE"
DENSITY = DENSITY(AIR, T=TFP, P=PATM)
DYNVISC = VISCOSITY(AIR, T=TFP)
KINVISC = DYNVISC / DENSITY
THERMEXP = 1 / TFP
Pr = PRANDTL(AIR, T=TFP)
KA = CONDUCTIVITY(AIR, T=TFP)

"CALCULATE CONVECTION COEFFICIENT FROM ABSORBER TO AIR THROUGH CHANNEL"
RaSP = (grav * COS(SLOPE) * THERMEXP * (TPM - TA) * S^3 / KINVISC^2) * Pr
IF(TPM > TA) THEN GOTO 110

"CALCULATE HCPA IF PLATE IS BELOW AMBIENT"
NUSP = 0.0039 * (ABS(RaSP))^0.6774
GOTO 115

"CALCULATE HCPA IF PLATE IS ABOVE AMBIENT"
110: NUSP = (0.0808 * RaSP^0.365) - 1
115: IF(NUSP < 1.0) THEN NUSP = 1.0

*************************INCLUDED INDUCED WIND HEAT TRANSFER COEFFICIENT*************************
HCPA = NUSP * KA / S + HCW

"DEFINE CORRECTION FACTORS FOR RADIATION - FROM EXPERIMENT"
IF(TPM > TA) THEN GOTO 120

"BELOW AMBIENT CASE"
C = 0.91
GOTO 125

"ABOVE AMBIENT CASE"
120: C = 1.15

"CALCULATE RADIATION COEFFICIENT FROM PLATE TO INSULATION"
125: HRPI = (C * SIGMA * (TPM^2 + TIM^2) * (TPM + TIM)) / (1 / EMI + 1 / EMI - 1)

"CALCULATE LOSS COEFFICIENT FROM ABSORBER TO INSULATION"
UPI = HRPI

"CALCULATE HEAT LOSS COEFFICIENT FROM INSULATION"
IF(TIM > TA) THEN GOTO 140

"CALCULATE UIA FOR BELOW AMBIENT"
UIA = 0.005 * (TIM - TA) + 1.4821 + HCW
GOTO 145

"CALCULATE UIA FOR ABOVE AMBIENT"
140: UIA = (1.9152 * (TIM - TA))^0.2491 - 2 + HCW
IF(UIA < 0) THEN UIA = 0.001

"CALCULATE HEAT LOSS COEFFICIENT FROM PLATE THROUGH INSULATION"
145: UPIA = 1 / (1 / UPI + 1 / UIA)

*******CHECK IF ASSUMED INSULATION TEMPERATURE IS WITHIN TOLERANCES *******
IF NOT, SET ASSUMED TEMPERATURE TO CALCULATED VALUE AND ITERATE AGAIN UNTIL VALUE IS WITHIN TOLERANCES."
TIMN = TPM - UPIA * (TPM - TA) / UPI
ERR = ABS(TIMN - TIM)
i = i + 1
IF(ERR < TOL) THEN GOTO 200
TIM = TIMN
GOTO 100

"CALCULATE OVERALL BACK HEAT LOSS COEFFICIENT"
200: UBACKL = HCPA + UPIA
END
PROCEDURE flprop(TYP,TFM,CP,PR,DENS,ABVISC,KF,KVISC)
****** THIS SUBROUTINE CALCULATES THE HEAT TRANSFER FLUID ******

PROPERTIES BASED ON THE FLUID TEMPERATURE

TYP   - FLUID TYPE, .EQ. 1 > GLYCOL 50/50 MIX, .NE. 1 > WATER
TFM   - MEAN FLUID TEMP IN K
TFMC  - MEAN FLUID TEMP IN DEG. C
CP    - FLUID SPECIFIC HEAT, J/kg C
KF    - FLUID THERMAL CONDUCTIVITY, W/m
DENS  - FLUID DENSITY, kg/cubic m
KVISC - FLUID KINEMATIC VISCOSITY, sq.m/sec
ABVISC- FLUID ABSOLUTE VISCOSITY, kg/m sec
PR    - FLUID PRANDAL NUMBER

TFMC = TFM - 273.0
IF (TYP = 1.) THEN GOTO 100

"THE FOLLOWING EQUATIONS ARE FOR THE PROPERTIES OF WATER"
BASED ON THE FLUID TEMPERATURE IN DEGREE CELCIUS"

CP = .0149689 * TFMC**2 - 1.37808 * TFMC + 4210.02
KF = -1.39368E-6 * TFMC**2 + 1.85377E-3 * TFMC + .571494
DENS = -.0035868 * TFMC**2 -.0670346 * TFMC + 1000.31
VISC = 7.7628E-10 * TFMC**5 + 1.2175E-7 * TFMC**4 - 7.19655E-5 * TFMC**3 + 9.1386E-3 * TFMC**2 - .54136E-4 * TFMC + 17.7615
KVISC = VISC**11E-7
ABVISC = KVISC * DENS
PR = CP * ABVISC / KF
GOTO 101

"THE FOLLOWING EQUATIONS ARE FOR THE PROPERTIES OF UCAR THERMOFLUID 17"
(50/50 % MIXTURE WITH WATER BY VOLUME) BASED ON THE FLUID
TEMPERATURE IN DEGREE CELCIUS"

100:  CP = -.0137147 * TFMC**2 + 5.99361 * TFMC + 3148.08
      KF = -1.28505E-6 * TFMC**2 + 6.3741E-6 * TFMC + .41369
      DENS = -.001863 * TFMC**2 + .465281 * TFMC + 1077.76
      Y = 1.98631E-8 * TFMC**4 + 5.22061E-6 * TFMC**3 + 0.28159E-4 * TFMC**2 + .0484805 * TFMC + 12.4476
      KVISC = 2.0 / (exp(Y))
      ABVISC = KVISC * DENS
      PR = CP * ABVISC / KF
101:  END

PROCEDURE apfr(TYP,TFM,SMFR,UL,AP,CSAT,NT,DH,L,W,WPER,F:CP,HF,RE,NUF,UF,FRME,F,FR,PR)
**THIS ROUTINE CALCULATES THE VALUE OF FR FOR THE CHAMBERLAIN COLLECTOR, THE RELATIONSHIP FOR UO SHOULD BE CHANGED TO SUIT OTHER COLLECTOR TYPES IF THEY ARE TO BE EVALUATED.

TYP   - FLUID TYPE, .EQ. 1 > 50/50% GLYCOL, .NE. 1 > WATER
TFM   - MEAN FLUID TEMPERATURE, K
SMFR  - SPECIFIC MASS FLOW RATE, kg/sec sq.m of collector
UO    - FLUID TO ATMOSPHERE HEAT LOSS COEFF., W/sq.m C
UL    - COLLECTOR HEAT LOSS COEFF., W/sq.m C
AP - ABSORBER PLATE AREA, sq.m
CSAT - CROSS-SECTIONAL AREA OF FLOW TUBES, sq.m
NT - NUMBER OF FLOW TUBES
DH - HYDRAULIC DIAMETER OF FLOW TUBES, m
L - ABSORBER PLATE EFFECTIVE LENGTH, m
W - WIDTH OF FIN AND TUBE, m
WPER - WETTED PERIMETER LENGTH OF FLOW TUBES, m
HF - HEAT TRANSFER COEFF., FLUID TO TUBE WALL, W/sq.m
RE - REYNOLDS NUMBER OF FLUID
NUF - NUSSELT NUMBER FOR FLUID
CP - SPECIFIC HEAT OF FLUID, J/kg. C
FPRME - COLLECTOR EFFICIENCY FACTOR
FR - HEAT REMOVAL FACTOR FOR COLLECTOR
F - FIN EFFICIENCY OF ABSORBER PLATE
PR - PRANDTL NUMBER

*************** DETERMINE COLLECTOR EFFICIENCY FACTOR, F' ***************

CALCULATE HEAT TRANSFER FLUID PROPERTIES*
CALL FLPROP(TYP,TFM,CP,PR,DENS,ABVISC,KF,KVISC)

VAVE=(SMFR*AP)/(DENS*CSAT*NT)

"CALCULATE THE REYNOLDS NUMBER FOR THE HEAT TRANSFER FLUID"
RE=(DENS*VAVE*DH)/ABVISC

"CALCULATE THE NUSSELT NUMBER FOR THE HEAT TRANSFER FLUID"
NUF=3.66+0.668*(RE*PR*DH/L)/(1.+0.4*((RE*PR*DH/L)**.666666))

"CALCULATE HEAT TRANSFER COEFFICIENT (LIQUID TO ABSORBER TUBE)"
HF=NUF*(KF/DH)

"CALCULATE HEAT LOSS COEFFICIENT (FLUID TO AMBIENT AIR)"
UO=1./(W*((1./(UL*W*F))+(1./(WPER*HF))))

"THE COLLECTOR EFFICIENCY FACTOR, F' IS GIVEN;"
FPRME=UO/UL

"THE HEAT REMOVAL FACTOR, FR IS GIVEN BY;"
A1=SMFR*CP/UL
A2=(-1.*FPRME*UL)/(SMFR*CP)
FR=A1*(1.-exp(A2))

**************************** SPECIFY SIMULATION PARAMETERS ***********************
SIGMA=5.670E-8
TOL=0.05

**************************** SPECIFY THE COLLECTOR CHARACTERISTICS ***********************
"DIMENSIONS (IN MKS UNITS)"
AG=2.874
AP=2.86
A=2.874
AB=2.874
A_edge=0.52
DH=0.0078
LI=0.019
LPC=0.012
D=0.0407
L=18.69
W=0.153
S=0.03

"OPTICAL PROPERTIES"
N=1.518
KLC=0.035
TRNLW=0.02

"HEAT TRANSFER PROPERTIES"
EMC=0.88

"ABSORBER PLATE"
CSAT=\pi*(DH/2)^2
NT=1.0
WPER=\pi*DH
EMP=0.05
ALFAP=0.94
F=0.98
EMPR=0.95
EMI=0.95

"COLLECTOR HOUSING"
KI=0.2
UE=9.71
UE=UE*(A_edge/A)

*************** ENVIRONMENTAL AND OPERATING CONDITIONS ***************

"ENVIRONMENTAL CONDITIONS (TEMPERATURE IN DEGREE K)"
TFI=273.0+CI1
TA=273.0+TAC
PATM=1.0 [bar]

"OPERATIONAL CONDITIONS"
SMFR=MFLR/AP

*************** HEAT TRANSFER MODEL AND ANALYSIS ***************

80: TPM=TFI+2.0
90:
CALL topl(TOL,EMC,EMP,EMG,LPC,ALFAP,TRNLW,TILT,TPM,TA,HW:UTOPL,HRCA,HRPC,HPC,UR
PS,UPC,UCA,TS,TC)
UTOPL=(AP/A)*UTOPL

****** DETERMINE THE VALUE OF THE OVERALL HEAT LOSS COEFFICIENT ******

BACK HEAT LOSS COEFFICIENT"
CALL backl(TOL,TPM,TA,TILT,PATM,EMPR,EMI,S,HCW:TIM,UBACKL)
UB=(AB/A)*UBACKL
"UB=(KI/LI)*(AB/A)"
"UB=(1/95)*(AB/A)"

"OVERALL HEAT LOSS COEFFICIENT"
UL=UB+UE+UTOPL

*************** CALCULATE OPTICAL PROPERTIES***************
CALL STAEFF(N,KLC,ALFAP,ANGLE,TILT,FRACD,REFG,UTOPL,UPC:TAEEF)

************************ DETERMINE THE HEAT REMOVAL FACTOR, FR ******************

TFM=TFI

250:
CALL
APFR(TYP,TFM,SMFR,UL,AP,CSAT,NT,DH,L,W,WPER,F:CP,HE,RE,NUF,OU,FPRME,FR,PR)

**************************** CALCULATION OF USEFUL ENERGY COLLECTED ********************

QU=AP*FR*(G*TAEEF-UL*(TFI-TA))

****** CHECK IF ASSUMED PLATE TEMPERATURE IS WITHIN TOLERANCES ******
IF NOT, SET ASSUMED COVER TEMPERATURE TO THE CALCULATED VALUE AND
ITERATE AGAIN UNTIL VALUE IS WITHIN TOLERANCES."

"CALCULATE MEAN FLUID TEMPERATURE"
TFMN=TFI+(((QU/AP)/(UL*FR))*(1.-((FR/FPRME))))
ERR=abs(TFMN-TFM)
IF(ERR<TOL) OR (ERR=TOL) THEN GOTO 350
TFM=TFMN
GOTO 250

"CALCULATE NEW MEAN PLATE TEMPERATURE"
350: TPMN=TFM+((QU)/(HF*WPER*L*NT))
ERR=abs(TPMN-TPM)
IF(ERR<TOL) OR (ERR=TOL) THEN GOTO 400
TPM=TPMN
GOTO 90

**************************** CALCULATE COLLECTION EFFICIENCY ***********************

400: NITA=(QU/(G*AG))
PNITA=NITA*100.
TFE=TFI+((QU)/(SMFR*AP*CP))

"CALCULATE HEAT LOSS COEFFICIENT"
ADELTI=(TFI-TH)/G

**********************************************************************

TPMNTA=TPMN-TH
TFMTA=TFM-TH
TPPTA2=(TPMN+TH)/2.
FRUL1=FR*UL
END
**********************************MAIN PROGRAM**********************************

THIS PROGRAM SIMULATES THE STEADY STATE THERMAL PERFORMANCE OF A FLAT-PLATE SOLAR COLLECTOR AT SPECIFIC TEST CONDITIONS. THE PROGRAM CALLS THE SUBROUTINE COLSM TO PERFORM THE CALCULATIONS.

AG=2.874
INPTS=1

"DO YOU WISH TO CHANGE TEST CONDITIONS, (Y/N)?

TYP= 1. [TYP=1 FOR GLYCOL, TYP=0 FOR WATER]
TILT= 45 "COLLECTOR TILT, (deg.)"
ANGLE=0 "RAD. INCIDENT ANGLE, (deg.)"
G= 910 "SOLAR INSOL. LEVEL,(W/sq.M)"
FRACD=0.01 "FRACTION OF DIFFUSE RAD."
MFLR=0.058 "FLUID FLOW RATE,(Kg./sec)"
TAC=20 "AMBIENT AIR TEMP.,(deg. C)"
CI1= X "INLET FLUID TEMP.,(deg. C)"
HW=20 "WIND H.T.COEFF.(W/sq.M C)"
EMG=.9 "EMITANCE OF GROUND"
REFG=.1 "GROUND REFLECTANCE"
HCW = 0 "WIND INDUCED H.T. COEFF. INSIDE CHANNEL"

CALL colsm(TYP,TILT,ANGLE,G,FRACD,MFLR,TAC,CI1,HW,EMG,REFG,HCW:PNITA,UTOPL,UBAC
KL,UL)

"CALCULATE DT/G"
FX=(CI1-TAC)/G
"TSC=TS-273.0"

"*********************************END OF PROGRAM**********************************"
Table B-1: Sample output for the hybrid collector, at conditions used in the SRCC test for the Enerworks Heat Safe collector

<table>
<thead>
<tr>
<th>$T_{IN}$</th>
<th>$T_A$</th>
<th>$G$</th>
<th>$(T_{IN}-T_A)/G$</th>
<th>$\eta$ or CPF (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-70</td>
<td>20</td>
<td>910</td>
<td>-0.099</td>
<td>109.6</td>
</tr>
<tr>
<td>-60</td>
<td>20</td>
<td>910</td>
<td>-0.088</td>
<td>104.2</td>
</tr>
<tr>
<td>-50</td>
<td>20</td>
<td>910</td>
<td>-0.077</td>
<td>98.9</td>
</tr>
<tr>
<td>-40</td>
<td>20</td>
<td>910</td>
<td>-0.066</td>
<td>93.7</td>
</tr>
<tr>
<td>-30</td>
<td>20</td>
<td>910</td>
<td>-0.055</td>
<td>88.6</td>
</tr>
<tr>
<td>-20</td>
<td>20</td>
<td>910</td>
<td>-0.044</td>
<td>83.7</td>
</tr>
<tr>
<td>-10</td>
<td>20</td>
<td>910</td>
<td>-0.033</td>
<td>79.4</td>
</tr>
<tr>
<td>0</td>
<td>20</td>
<td>910</td>
<td>-0.022</td>
<td>74.4</td>
</tr>
<tr>
<td>10</td>
<td>20</td>
<td>910</td>
<td>-0.011</td>
<td>68.5</td>
</tr>
<tr>
<td>20</td>
<td>20</td>
<td>910</td>
<td>0</td>
<td>61.8</td>
</tr>
<tr>
<td>30</td>
<td>20</td>
<td>910</td>
<td>0.011</td>
<td>54.5</td>
</tr>
<tr>
<td>40</td>
<td>20</td>
<td>910</td>
<td>0.022</td>
<td>46.9</td>
</tr>
<tr>
<td>50</td>
<td>20</td>
<td>910</td>
<td>0.033</td>
<td>39.1</td>
</tr>
<tr>
<td>60</td>
<td>20</td>
<td>910</td>
<td>0.044</td>
<td>31.0</td>
</tr>
<tr>
<td>70</td>
<td>20</td>
<td>910</td>
<td>0.055</td>
<td>22.7</td>
</tr>
<tr>
<td>80</td>
<td>20</td>
<td>910</td>
<td>0.066</td>
<td>14.3</td>
</tr>
<tr>
<td>90</td>
<td>20</td>
<td>910</td>
<td>0.077</td>
<td>5.7</td>
</tr>
<tr>
<td>100</td>
<td>20</td>
<td>910</td>
<td>0.088</td>
<td>-3.1</td>
</tr>
<tr>
<td>110</td>
<td>20</td>
<td>910</td>
<td>0.099</td>
<td>-11.9</td>
</tr>
<tr>
<td>120</td>
<td>20</td>
<td>910</td>
<td>0.11</td>
<td>-20.8</td>
</tr>
</tbody>
</table>

Fig. B-1: Efficiency or CPF curve for hybrid collector, showing inflection point at the transition between above and below ambient operation
Appendix C: TRNSYS Configuration

A screenshot from the TRNSYS simulation studio is shown in Fig. C-1. The connections between the system components are included. Dark blue lines represent the flow of cold water, while red lines represent hot water flow. Light blue lines are the flow of cold glycol-water mix, while the warm glycol-water mix is represented by orange. Dashed green lines are control signals, while miscellaneous information is passed along the black lines. Two collector types are used to represent the performance of the collector above and below the ambient temperature. The configuration for the system with the valved collector, shown in Fig. C-2 has four collector types to represent the hybrid collector and the glazed collector, for above and below ambient operation. The valve control calculator acts as an ‘IF’ statement, comparing the output of the two collector types, and forwarding the data from the collector with the highest output temperature.
Fig. C-1: TRNSYS Simulation Studio Configuration for Hybrid Collector in Toronto
Fig. C-2: TRNSYS Simulation Studio Configuration for Valved Collector in Toronto
Appendix D: Cool-down Test to Determine Tank Heat Loss Coefficient

To determine an updated heat loss coefficient of the storage tank, a cool-down test was performed according to the SRCC TM-1 document [38]. From the October 27th, 2010 test the storage tank was in the fully charged condition. Tank temperature data was recorded for the following 108 hours, and was averaged over 12 hour intervals. At each interval, the energy lost from the each node was calculated according to:

\[
E_{\text{loss},i,t} = (m_i C_p)(T_{\text{tank},i,t} - T_{\text{tank},i,t-12hr})
\]  

(D-1)

And the average heat transfer rate over each time step was calculated as:

\[
Q_{\text{loss},i,t} = \frac{E_{\text{loss},i,t}}{43200 \text{ s}}
\]

(D-2)

Subsequently, the loss coefficient-area product, \(UA\), was determined for each node at each time step, where \(A_{S,i}\) is the surface area of each node of water in contact with the walls of the tank:

\[
(UA)_{i,t} = \frac{Q_{\text{loss},i,t}}{A_{S,i}(T_{\text{tank},i,t} - T_{\text{amb}})}
\]

(D-3)

At each time interval, an area-weighted average of the \(UA\) values was made. The average of these values was taken as the loss coefficient for the tank, shown in Table D-1. The final value was determined to be 1.148 W/m\(^2\)K (4.132 kJ/hr-m\(^2\)K).
Table D.1: Nodal and Average Heat Loss Coefficients for the Storage Tank

<table>
<thead>
<tr>
<th></th>
<th>UA10</th>
<th>UA9</th>
<th>UA8</th>
<th>UA7</th>
<th>UA6</th>
<th>UA5</th>
<th>UA4</th>
<th>UA3</th>
<th>UA2</th>
<th>UA1</th>
<th>Node Average</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hour 0-12</td>
<td>-1.981</td>
<td>1.136</td>
<td>2.469</td>
<td>2.083</td>
<td>1.673</td>
<td>1.444</td>
<td>1.330</td>
<td>1.360</td>
<td>2.093</td>
<td>1.380</td>
<td>1.042 W/m²-K</td>
</tr>
<tr>
<td>Hour 12-24</td>
<td>-0.006</td>
<td>0.420</td>
<td>0.936</td>
<td>1.855</td>
<td>2.070</td>
<td>1.883</td>
<td>1.693</td>
<td>1.628</td>
<td>1.687</td>
<td>0.837</td>
<td>1.158 W/m²-K</td>
</tr>
<tr>
<td>Hour 24-36</td>
<td>0.381</td>
<td>0.981</td>
<td>0.892</td>
<td>1.154</td>
<td>1.550</td>
<td>1.784</td>
<td>1.812</td>
<td>1.719</td>
<td>1.631</td>
<td>0.835</td>
<td>1.167 W/m²-K</td>
</tr>
<tr>
<td>Hour 36-48</td>
<td>0.546</td>
<td>1.171</td>
<td>1.086</td>
<td>1.097</td>
<td>1.265</td>
<td>1.506</td>
<td>1.715</td>
<td>1.737</td>
<td>1.683</td>
<td>0.798</td>
<td>1.166 W/m²-K</td>
</tr>
<tr>
<td>Hour 48-60</td>
<td>0.596</td>
<td>1.244</td>
<td>1.178</td>
<td>1.153</td>
<td>1.176</td>
<td>1.310</td>
<td>1.500</td>
<td>1.625</td>
<td>1.659</td>
<td>0.974</td>
<td>1.168 W/m²-K</td>
</tr>
<tr>
<td>Hour 60-72</td>
<td>0.690</td>
<td>1.373</td>
<td>1.326</td>
<td>1.291</td>
<td>1.268</td>
<td>1.314</td>
<td>1.454</td>
<td>1.558</td>
<td>1.625</td>
<td>0.899</td>
<td>1.202 W/m²-K</td>
</tr>
<tr>
<td>Hour 72-84</td>
<td>0.751</td>
<td>1.483</td>
<td>1.420</td>
<td>1.369</td>
<td>1.325</td>
<td>1.309</td>
<td>1.380</td>
<td>1.446</td>
<td>1.537</td>
<td>0.811</td>
<td>1.202 W/m²-K</td>
</tr>
<tr>
<td>Hour 84-96</td>
<td>0.678</td>
<td>1.395</td>
<td>1.358</td>
<td>1.329</td>
<td>1.267</td>
<td>1.261</td>
<td>1.287</td>
<td>1.334</td>
<td>1.385</td>
<td>0.747</td>
<td>1.125 W/m²-K</td>
</tr>
<tr>
<td>Hour 96-108</td>
<td>0.725</td>
<td>1.366</td>
<td>1.334</td>
<td>1.299</td>
<td>1.263</td>
<td>1.230</td>
<td>1.273</td>
<td>1.292</td>
<td>1.332</td>
<td>0.664</td>
<td>1.100 W/m²-K</td>
</tr>
</tbody>
</table>

Average: 1.148 W/m²-K

4.132 kJ/hr·m²-K
Appendix E: Wiring Diagram

An updated wiring diagram for the ISAHP prototype, showing instrumentation and connections to the data acquisition system is shown in Fig. E-1.
Fig. E-1: ISAHP prototype wiring diagram
Appendix F: Flow Meter Calibration

The turbine flow meter used to measure the collector loop flow rate outputs a signal as a 0-1200 Hz square wave with 5 V pulses. A flow signal conditioner was used to convert this frequency to a 4-20 mA signal, which when passed through a 250 Ω resistor, caused a 0-5 V signal. A resulting linear relationship between the measured voltage and the frequency is then:

\[ f = 300V - 300 \quad (F-1) \]

A relationship between the flow rate and the resulting signal frequency was determined by measuring the frequency output from the flow meter for a range of flow rates, shown in Fig. F-1. The frequency was measured by a handheld digital multimeter, while the flow rate was determined by measuring the amount of time required to fill a 2L container.

![Graph showing the relationship between flow rate and FTI flow meter output frequency](image)

\[ AV = 0.0051f + 0.0118 \]

\[ R^2 = 1.0000 \]

**Fig. F-1: Relationship between flow rate and FTI flow meter output frequency**

The resulting relationship between the measured voltage and the flow rate is therefore:

\[ AV = 1.53V - 1.5182 \quad (F-2) \]
Appendix G: Propagation of Errors

The error associated with the measurements and the subsequent calculation of values based on these measurements are discussed in this appendix, based upon the root mean squared (RMS) method [48]. Sample calculations for the error associated with the heat transfer rate through the evaporator, calculated on the collector side, are detailed below. Additionally, the propagation of error from the temperature measurements through to the relationships for the various heat loss coefficients derived for the collector back loss are outlined for a sample flux of 200 W/m². A summary of the errors for all power levels tested in also included.

**Collector-side Evaporator Heat Transfer Rate**

As discussed in Appendix F, the calibration of the collector loop flow meter involved comparing the output frequency of the flow meter to the flow rate measured by the time required to fill a 2 L container. The mass flow rate is:

\[
\dot{m} = \frac{V \rho}{t} \tag{G-1}
\]

Assuming a 5% error in volume and density, as well as a 0.1 s error in the time measurement, the error in mass flow rate can be calculated for the 2285 Hz calibration point:

\[
\omega_m = \pm \sqrt{\left(\frac{\partial \dot{m}}{\partial V} \omega_V \right)^2 + \left(\frac{\partial \dot{m}}{\partial \rho} \omega_\rho \right)^2 + \left(\frac{\partial \dot{m}}{\partial t} \omega_t \right)^2}
\]

\[
= \pm \sqrt{\left(\frac{1040}{10.38} \cdot 0.001 \right)^2 + \left(\frac{0.002}{10.38} \cdot 52 \right)^2 + \left(\frac{0.002 \cdot 1040}{10.38^2} \cdot 0.1 \right)^2}
\]

The resulting error was ±0.0143 kg/s, which accounted for 7.1% of the measured flow rate of 0.0200 kg/s. The other values through the calibration curve had errors in the 7.07-7.14% range, therefore 7% was chosen to represent an average error for the mass flow rate measured by
the turbine flow meter. This error contributed to the uncertainty in the evaporator heat transfer rate, calculated on the collector side, where:

\[ \omega_{\text{evap}} = \pm \sqrt{\left( \frac{\partial Q_{\text{evap}}}{\partial \dot{m}} \omega_{\dot{m}} \right)^2 + \left( \frac{\partial Q_{\text{evap}}}{\partial C_p} \omega_{C_p} \right)^2 + 2 \left( \frac{\partial Q_{\text{evap}}}{\partial T} \omega_T \right)^2} \]  

(G-3)

where the error in the specific heat of the glycol-water mixture was approximated as a conservative estimate of 5% to account for a discrepancy in the mixture ratio, and error is the temperature used to calculate the specific heat from the commercial datasheet. The error in temperature was chosen as 0.5°C. As an example calculation was applied to each data point in the November 2\textsuperscript{nd}, 2010 data, and the resulting error ranged between 8.9% and 13.4%.

The error in the collector loop efficiency was calculated using both the glycol side and refrigerant side heat transfer rates through the evaporator, and the error in the pyranometer reading, according to:

\[ \omega_\eta = \pm \sqrt{\left( \frac{\partial \eta}{\partial Q_{\text{evap}}} \omega_{Q_{\text{evap}}} \right)^2 + \left( \frac{\partial \eta}{\partial A_c} \omega_{A_c} \right)^2 + \left( \frac{\partial \eta}{\partial g_T} \omega_{g_T} \right)^2} \]  

(G-4)

The Eppley PSP pyranometer has a 1% error due to temperature dependence over -20°C to 40°C, a 0.5% error in linearity, and a 1% error associated with the incidence angle [49]. Therefore a total error of 2.5% was assumed. From the November 2\textsuperscript{nd}, 2010 data, the error based on the glycol side was 9.1% to 13.4%, while for the refrigerant side it was 3.5% to 4.3%.
Back Heat Loss Coefficients

Error in the output of the power supply was determined based upon the values supplied in the documentation, where a maximum error in voltage is 1.6V, and the maximum error in current is 0.09 A. At 200 W/m², the error can be calculated as:

\[
\omega_{\text{Q}_{\text{rad},i}} = \pm \sqrt{\left(\frac{\partial \text{Q}_{\text{rad},i}}{\partial V} \omega_V\right)^2 + \left(\frac{\partial \text{Q}_{\text{rad},i}}{\partial I} \omega_I\right)^2} = \sqrt{(5.54 + 1.6)^2 + (103.2 + 0.09)^2} \]

\[= \pm 12.8 W\]

For the closed channel condition, the error associated with the heat transfer through the still air in the channel was:

\[
\omega_{\text{Q}_{\text{air}}} = \pm \sqrt{\left(\frac{\partial \text{Q}_{\text{air}}}{\partial k_{\text{air}}} \omega_{k_{\text{air}}}\right)^2 + \left(\frac{\partial \text{Q}_{\text{air}}}{\partial A} \omega_{A}\right)^2 + \left(\frac{\partial \text{Q}_{\text{air}}}{\partial S} \omega_{S}\right)^2 + 2 \left(\frac{\partial \text{Q}_{\text{air}}}{\partial T} \omega_{T}\right)^2} \]

The error in the conductivity was determined from a linear relation for a given temperature. With the error in temperature of 0.5°C and the standard error from the fit, an error for the conductivity was approximated as 0.5%. Error in the area and channel spacing was based upon 0.005 m and 0.001 m error for each length measurement. The resulting \(\text{Q}_{\text{air}}\) for the 200 W/m² case was 21.0 ± 0.1 W.

The RMS method could not simply be used in the determination of the error propagation through the calculation of \(\text{Q}_{\text{rad},p-i}\) due to the matrix inversion required by Hottel’s zone method. Therefore the error was determined by a sensitivity analysis on the MatLab program written to calculate the radiative flux. The inputs, being the temperature profiles, area, and channel spacing were varied individually from the base case, and the resulting change in the output was recorded. For the 200 W/m² closed channel case, applying the 0.5°C error to the temperature profiles resulted in errors of 15.8 W and 14.7 W, for the absorber and insulation temperatures, respectively. The error is area and channel spacing caused a 1.6 W and 2.6 W error in \(\text{Q}_{\text{rad},p-i}\).
respectively. The RMS value of these errors was calculated as 21.8 W, which accounts for an 8% error on \( Q_{\text{rad,}p-i} \). The error in \( Q_{\text{cond,}p-a} \) was then calculated as the RMS value of the \( Q_{\Omega}, Q_{\text{air}}, \) and \( Q_{\text{rad,}p-i} \) as:

\[
\omega_{Q_{\text{cond,}p-a}} = \pm \sqrt{\left( \omega_{Q_{\text{air}}} \right)^2 + \left( \omega_{Q_{\Omega}} \right)^2 + \left( \omega_{Q_{\text{rad,}p-i}} \right)^2} = \pm \sqrt{(0.1)^2 + (12.8)^2 + (21.8)^2} = \pm 25.1 \text{ W}
\]  

(MG-7)

Much of this process was repeated for the open channel case, however with the exclusion of \( Q_{\text{air}} \). Additionally, as \( Q_{\text{cond,}p-a} \) was evaluated from the relation developed from the closed channel case, the associated error was determined from a similar relationship. The upper and lower limits of \( Q_{\text{cond,}p-a} \) were plotted as functions of \( (T_{PM}-T_A) \), shown in Fig. G-1.

![Fig. G-1: Relationships for the error in \( Q_{\text{cond,}p-a} \)](image)

The resulting error in \( Q_{\text{conv,}p-a} \) for the 200 W/m\(^2\) open channel case was calculated as the RMS value of the errors in \( Q_{\text{cond,}p-a}, Q_{\Omega}, \) and \( Q_{\text{rad,}p-i} \):

\[
\omega_{Q_{\text{conv,}p-a}} = \pm \sqrt{\left( \omega_{Q_{\text{cond,}p-a}} \right)^2 + \left( \omega_{Q_{\Omega}} \right)^2 + \left( \omega_{Q_{\text{rad,}p-i}} \right)^2} = \pm \sqrt{(15.3)^2 + (12.8)^2 + (13.6)^2} = \pm 24.1 \text{ W}
\]  

(MG-8)
As a percentage $Q_{\text{conv,p-ar}}$, this error accounts for 8.9%. As the calculation of the convective heat transfer coefficient was based upon $Q_{\text{conv,p-ar}}$, the associated error was also similarly dependent:

\[
\omega_h = \pm \sqrt{\left(\frac{\partial h}{\partial Q_{\text{conv}}} \omega_q\right)^2 + \left(\frac{\partial h}{\partial A} \omega_A\right)^2 + 2 \left(\frac{\partial h}{\partial T} \omega_T\right)^2}
\]

\[
= \pm \sqrt{\left(\frac{24.0}{2.874 + 32.8}\right)^2 + \left(\frac{268.4}{2.874^2 + 32.8^2} \times 0.014\right)^2 + 2 \left(\frac{268.4}{2.874 \times 32.8^2} \times 0.5\right)^2}
\]

\[
= \pm 0.26 \frac{W}{m^2 K}
\]

This accounted for an error of 9.1% on a value of 2.86 W/m$^2$K. The error associated with the Nusselt number was calculated based upon the error in the convection coefficient and the channel spacing. The error in the conductivity of the air was ignored, as its magnitude was much less than the others.

\[
\omega_{\text{Nu}_S} = \pm \sqrt{\left(\frac{\partial \text{Nu}_S}{\partial h} \omega_h\right)^2 + \left(\frac{\partial \text{Nu}_S}{\partial S} \omega_S\right)^2}
\]

\[
= \sqrt{\left(\frac{0.03}{0.0263} \times 0.26\right)^2 + \left(\frac{2.874}{0.0263} \times 0.001\right)^2} = \pm 0.32
\]

Additionally, the error in the heat transfer coefficient from the insulation to the surroundings was determined based on the error in $Q_{\text{rad,p-I}}$, the area, and the insulation and ambient temperatures:
\[ \omega_{\text{U}-\alpha} = \pm \sqrt{\left( \frac{\partial U_{\text{I}-\alpha}}{\partial Q_{\text{rad},p-i}} \omega_{\text{rad},p-i} \right)^2 + \left( \frac{\partial U_{\text{I}-\alpha}}{\partial A} \omega_A \right)^2 + 2 \left( \frac{\partial U_{\text{I}-\alpha}}{\partial T} \omega_T \right)^2} \]

\[ = \pm \sqrt{\left( \frac{13.6}{2.874 + 24.3} \right)^2 + \left( \frac{196.2}{2.874^2 + 24.3^2} 0.014 \right)^2 + 2 \left( \frac{196.2}{2.874 \times 24.3} 0.014 \right)^2} \]

\[ = \pm 0.21 \frac{W}{m^2 K} \]  

This process was repeated for each power level and inlet temperature for Series 2 and 4. The resulting error values are summarized in Tables G-1 and G-2. The error for \( Q_{\text{fluid}} \) was calculated identically to that for \( Q_{\text{evap}} \) in equation G-3. It can be seen in the tables that for the tests with low input power (i.e. a flux of 10 W/m\(^2\) and an inlet temperature of 15°C) that the percentage error is very large. This was due to the small temperature differences associated with these cases, and that the error in temperature was of similar magnitude. While the error may be large, the resulting rates of heat loss are small compared to the irradiance and useful energy transfer rates when operating near ambient temperature. Therefore this error would have little effect on the collector efficiency.
Table G-1: Calculated values and their respective error values for Series 2

<table>
<thead>
<tr>
<th>Flux</th>
<th>$Q\Omega$ [W] +/-</th>
<th>$Q_{rad}$ [W] +/-</th>
<th>$Q_{cond}$ [W] +/-</th>
<th>$Q_{conv,p-a}$ [W] +/-</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>28.64 ± 2.87</td>
<td>5.06 ± 10.42</td>
<td>15.20 ± 9.70</td>
<td>8.38 ± 14.52</td>
</tr>
<tr>
<td>50</td>
<td>142.93 ± 6.42</td>
<td>45.28 ± 11.21</td>
<td>40.12 ± 11.15</td>
<td>57.53 ± 17.07</td>
</tr>
<tr>
<td>75</td>
<td>214.25 ± 7.86</td>
<td>69.99 ± 11.68</td>
<td>53.52 ± 11.93</td>
<td>90.74 ± 18.45</td>
</tr>
<tr>
<td>99</td>
<td>283.42 ± 9.08</td>
<td>94.48 ± 12.10</td>
<td>65.17 ± 12.61</td>
<td>123.76 ± 19.69</td>
</tr>
<tr>
<td>200</td>
<td>571.73 ± 12.84</td>
<td>196.24 ± 13.55</td>
<td>107.05 ± 15.05</td>
<td>268.43 ± 23.98</td>
</tr>
<tr>
<td>300</td>
<td>858.26 ± 15.73</td>
<td>306.07 ± 15.01</td>
<td>145.03 ± 17.26</td>
<td>407.16 ± 27.76</td>
</tr>
<tr>
<td>400</td>
<td>1143.86 ± 18.16</td>
<td>420.41 ± 16.44</td>
<td>180.23 ± 19.31</td>
<td>543.21 ± 31.19</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Flux</th>
<th>$h_{conv}$ [W/m²K] +/-</th>
<th>$\nuS$ +/-</th>
<th>UIA [W/m²K] +/-</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>0.83 ± 1.45</td>
<td>0.99 ± 1.72</td>
<td>0.51 ± 1.06</td>
</tr>
<tr>
<td>50</td>
<td>1.76 ± 0.53</td>
<td>2.06 ± 0.63</td>
<td>2.08 ± 0.55</td>
</tr>
<tr>
<td>75</td>
<td>2.02 ± 0.42</td>
<td>2.35 ± 0.50</td>
<td>1.97 ± 0.35</td>
</tr>
<tr>
<td>99</td>
<td>2.23 ± 0.36</td>
<td>2.58 ± 0.43</td>
<td>2.19 ± 0.30</td>
</tr>
<tr>
<td>200</td>
<td>2.86 ± 0.26</td>
<td>3.26 ± 0.32</td>
<td>2.82 ± 0.21</td>
</tr>
<tr>
<td>300</td>
<td>3.17 ± 0.22</td>
<td>3.56 ± 0.28</td>
<td>3.27 ± 0.18</td>
</tr>
<tr>
<td>400</td>
<td>3.39 ± 0.20</td>
<td>3.75 ± 0.25</td>
<td>3.62 ± 0.16</td>
</tr>
</tbody>
</table>

Table G-2: Calculated values and their respective error values for Series 4

<table>
<thead>
<tr>
<th>Inlet Temp [°C]</th>
<th>$Q_{fluid}$ [W] +/-</th>
<th>$Q_{rad}$ [W] +/-</th>
<th>$Q_{cond}$ [W] +/-</th>
<th>$Q_{conv,p-a}$ [W] +/-</th>
</tr>
</thead>
<tbody>
<tr>
<td>-10</td>
<td>-444.41 ± 23.45</td>
<td>-63.99 ± 12.38</td>
<td>-84.65 ± 13.38</td>
<td>-295.77 ± 29.70</td>
</tr>
<tr>
<td>-5</td>
<td>-360.29 ± 18.99</td>
<td>-55.92 ± 12.21</td>
<td>-65.21 ± 11.09</td>
<td>-239.17 ± 25.15</td>
</tr>
<tr>
<td>0</td>
<td>-265.03 ± 13.93</td>
<td>-47.91 ± 12.02</td>
<td>-49.16 ± 9.14</td>
<td>-167.95 ± 20.54</td>
</tr>
<tr>
<td>5</td>
<td>-209.74 ± 10.92</td>
<td>-41.72 ± 11.86</td>
<td>-38.68 ± 7.83</td>
<td>-129.33 ± 17.92</td>
</tr>
<tr>
<td>15</td>
<td>-90.23 ± 4.75</td>
<td>-21.34 ± 11.09</td>
<td>-18.62 ± 5.09</td>
<td>-50.27 ± 13.09</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Inlet Temp [°C]</th>
<th>$h_{conv}$ [W/m²K] +/-</th>
<th>$\nuS$ +/-</th>
<th>UIA [W/m²K] +/-</th>
</tr>
</thead>
<tbody>
<tr>
<td>-10</td>
<td>4.76 ± 0.49</td>
<td>5.81 ± 0.60</td>
<td>1.39 ± 0.27</td>
</tr>
<tr>
<td>-5</td>
<td>4.47 ± 0.49</td>
<td>5.45 ± 0.59</td>
<td>1.41 ± 0.31</td>
</tr>
<tr>
<td>0</td>
<td>3.72 ± 0.47</td>
<td>4.51 ± 0.57</td>
<td>1.43 ± 0.36</td>
</tr>
<tr>
<td>5</td>
<td>3.35 ± 0.48</td>
<td>4.04 ± 0.58</td>
<td>1.45 ± 0.42</td>
</tr>
<tr>
<td>10</td>
<td>2.71 ± 0.54</td>
<td>3.25 ± 0.65</td>
<td>1.43 ± 0.55</td>
</tr>
<tr>
<td>15</td>
<td>2.60 ± 0.70</td>
<td>2.50 ± 0.84</td>
<td>1.46 ± 0.77</td>
</tr>
</tbody>
</table>
Appendix H: Wind-Induced Pressure Difference

The natural convection flow through the collector channel is caused by a pressure difference between the column of heated (or cooled) air within the channel, and the column of equal height outside the collector at ambient temperature. This pressure difference causes a flow through the collector, which is opposed to the hydraulic losses associated the channel. The static pressure in the top of the collector channel, $P_{top,\, chan}$, is:

$$P_{top,\, chan} = P_{bottom} - \rho_{avg}gL \cos \theta \quad (H-1)$$

where $P_{bottom}$ is the static pressure outside the bottom of the collector, $g$ is the acceleration due to gravity, $L$ is the length of the channel, and $\theta$ is the tilt angle. The average density of the air within the channel, $\rho_{avg}$, is dependent on the temperature of the air, $T_{avg}$. The pressure outside the top of the channel, $P_{top,\, ext}$, can be expressed as:

$$P_{top,\, ext} = P_{bottom} - \rho_{ext}gL \cos \theta \quad (H-2)$$

where $\rho_{ext}$ is the density of the air outside the collector channel, and is also dependent on the air temperature, $T_{ext}$. The pressure difference between the outside and inside of the collector is:

$$P_{top,\, ext} - P_{top,\, chan} = \rho_{avg}gL \cos \theta - \rho_{ext}gL \cos \theta \quad (H-3)$$

Assuming an exterior temperature of 20°C, and an average channel temperature of 55°C, the pressure difference is:

$$P_{top,\, ext} - P_{top,\, chan} = \left(9.81 \frac{m}{s^2}\right)2.45 \, m \, \cos 45^\circ \left(1.076 \frac{kg}{m^3} - 1.204 \frac{kg}{m^3}\right)$$

$$= -2.18 \, Pa \quad (H-4)$$

It is this pressure difference that acts as the driving force for the natural convection flow through the channel, exhausting through the top opening.
To investigate the effect that wind has on the flow through the channel on the hybrid collector, a series of pressure measurements were recorded on both the stagnation prevention apparatus and the collector attached to the ISAHP prototype. An AIR Inc. digital micromanometer was used to measure the static pressure difference between the lower and upper openings of the collector channel. On the stagnation prevention apparatus, the voltage output from the micromanometer was read by the D/A system along with the absorber plate temperatures to the see pressure difference in a wind-free condition. The results for a 400 W/m² test are show in Fig. H-1. It can be seen that the magnitude of the pressure difference increased with the absorber temperature. The fact that the pressure difference is negative signifies that the static pressure at the top of the channel was lower than at the bottom, which would cause an upwards flow, as would be expected from heated air.

![Fig. H-1: Static pressure difference between top and bottom openings of collector channel in still surrounding air, compared to the maximum absorber temperature.](image)

The pressure measurements from the ISAHP collector were compared to the wind speed, $V_w$, and direction, $\theta_w$, measured by a Lufft Ventus ultrasonic anemometer connected to the same
D/A system as the micromanometer. The wind direction was measured as degrees from North. Therefore southerly component of the wind was calculated as:

\[ V_{W,\text{South}} = V_{W} \cos(\theta_{W} - 180^\circ) \]  

(H-5)

The pressure difference was plotted along with the \( V_{W,S} \) over a 60 hour time span, shown in Fig. H-2. It can be seen that spikes in the pressure data correspond with spikes in the wind speed, particularly when the wind speed is above 0 m/s (i.e. blowing from a direction south of East and West). Additionally, when the wind is from the south, a wider range of pressure differences are measured, ranging between -5 and 5 Pa. Compared to the 1 Pa difference observed for the still air condition, it appears that the wind speed does affect the pressure drop across the channel, and therefore could increase or decrease the flow. It is difficult to determine a cohesive relationship between the wind speed and the change in pressure difference, due to a number of location-specific factors including collector orientation, mounting (i.e. roof, wall or rack mounted), local obstructions and recirculation patterns.
Fig. H-2: Static pressure difference between top and bottom openings of collector channel (blue) along with the Southerly component of the wind speed (red). Moving averages of the data are shown by the black lines.
Appendix I: SRCC Collector Test Results

The certification ratings and specifications from the Enerworks Heat Safe glazed collector and Heliocol unglazed collector are given in Fig. I-1 and Fig. I-2.
Fig. I-1: Enerworks Heat Safe collector SRCC Certification [50]
Fig. I-2: Heliocol collector SRCC Certification [50]

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