Critical Analysis of Different Solar Assisted Heat Pump Configurations in Cold Climate

H. M. Teamah, M. F. Lightstone
McMaster University, Mechanical Engineering Department
Emails: teamahhm@mcmaster.ca , lightsm@mcmaster.ca

Abstract

The integration of a heat pump into a solar domestic hot water system has been reported as a promising approach for enhancing overall system performance. The benefits arise from a reduction in solar collector heat losses and/or an increase in the heat pump coefficient of performance which increases the "free energy" captured by the heat pump. In northern climates, which typically use covered or insulated solar collectors, careful design of the solar assisted heat pump system (SAHP) is required in order for the system to benefit from the "free energy" associated with the heat pump. This paper presents a comparison of predicted annual system performance for conventional solar domestic hot water (SDHW) systems, series SAHP systems, and parallel SAHP systems. The modelling is performed using the TRNSYS simulation environment using weather for Toronto, Canada. The system performance criterion is the free energy ratio (FER). It quantifies the contribution of free resources (solar and air energy if present) to meet the load. The results highlight that the role of the heat pump in a series-arrangement is to lower collector heat losses, rather than to transfer energy from the ambient air. This is a result of the collector insulation which impedes heat exchange with the environment. This is in contrast with standard heat pump design which requires enhanced heat exchange of the evaporator fluid with the environment to capture the free energy. As a result of this paradox, only marginal increases in system performance are obtained with the series SAHP configuration in comparison to a conventional SDHW system. Moreover, simulations show that parallel integration of heat pumps into a solar domestic hot water system results in a substantial increase in the FER in comparison to both conventional SDHW and series-based SAHP systems.

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Introduction

Canada's residential space and water heating constitute 20% of national greenhouse gas emissions (GHG) [1]. Solar thermal systems can efficiently offset these emissions to meet the national targets. A SAHP configuration has been reported as a promising technology [2-5] due to the apparent synergies between the components: solar collectors raise fluid temperature which increases the heat pump coefficient of performance (COP) and the heat pump reduces collector inlet temperature which lowers collector heat losses. The details of the operation of the conventional solar domestic hot water (SDHW) system and different configurations of SAHP systems are given in the following subsections.

Conventional Solar Domestic Hot Water Systems

A typical solar domestic hot water (SDHW) system suitable for northern climates is shown in Figure 1. It consists of a collector, storage tank, a heat exchanger and a pump. For freeze-protection, a glycol-water mixture acts as the heat transfer fluid in the solar collector loop. If sufficient solar irradiation is available, the pump is engaged and cold fluid from the heat exchanger enters the collector and is heated as it flows through the solar collector. The heated fluid enters the heat exchanger and heat is transferred to water that is circulated through the storage tank. Storage is needed to even out the mismatch between the times of solar supply and times of residential demand. When there is a household demand, hot water is extracted from the tank top. The extracted water is replaced with cold water from the mains. The circulating pump is activated based on the temperature difference between the collector surface and the bottom of the tank. This system is referred to as "indirect system" since the solar collector fluid does not flow into the thermal storage tank. In moderate climates in which freeze-protection is not required, a direct configuration is employed such that there is no need for the heat exchanger. The indirect SDHW system is considered in the current paper due to its suitability for the Canadian climate.

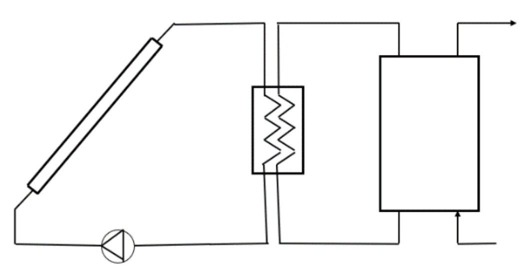


Figure 1: Indirect conventional solar domestic hot water system.

Solar-Assisted Heat Pump systems

Conventional heat pumps transfer heat from a cold temperature reservoir to a hot reservoir and require the use of a compressor as part of the thermodynamic cycle. The performance of the heat pump is dependent on the temperatures of the two reservoirs. Heat pumps typically extract energy for heating from either the ambient air or from the ground via borehole heat exchangers. The performance of an air-source heat pump is mainly determined by the temperature of the outdoor air and deteriorates at cold temperatures. Ground source heat pump offer better performance due to the relatively stable ground temperature, however, have a higher initial capital cost due to the installation of the ground-source heat exchanger. This makes solar-source heat pumps an alluring option to explore in northern climates [2-5].

The coupling of a heat pump with a solar thermal system can be done in varying ways. One configuration is the series solar-assisted heat pump system. Figure 2 shows an indirect system. the heat pump components are between the solar collector loop and the thermal storage loop. The refrigerant within the evaporator is exposed to hot fluid from the solar collector which acts to increase the COP of the heat pump. The glycol/water heat transfer fluid in the solar collector loop leaves the evaporator at a cold temperature which acts to reduce the heat losses from the solar thermal collector [5]. Cold climates require an indirect configuration for freeze protection purposes [6]. The indirect SAHP configuration is considered in the current paper.

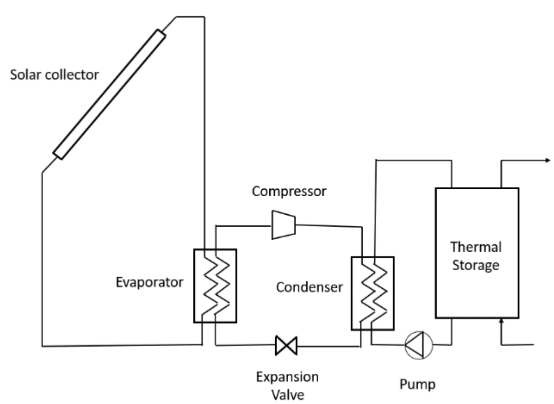


Figure 2: Series configuration: indirect solar assisted heat pump system.

As an alternative to the series arrangement of the heat pump and solar collector, the two components can be placed in parallel (Figure 3). In this arrangement, the heating can be supplied either by a conventional SDHW system or a traditional air source heat pump. This configuration benefits from the free energy of the air when the heat pump is in operation. It also benefits from the abundant solar energy when the SDHW system is in operation. Heat pump operation is, however, impacted by the temperature of the outdoor air. As such, the parallel configuration would appear to not benefit from the increased heat pump coefficient of performance attained with the series configuration.

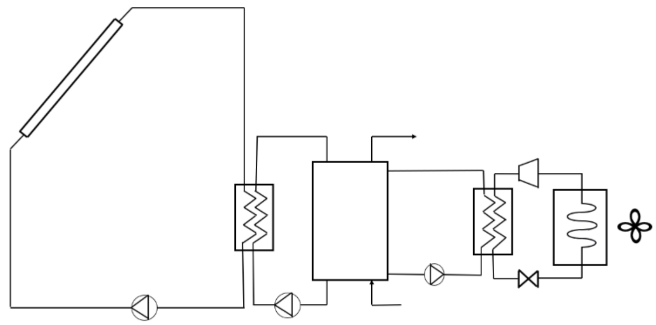


Figure 3: Parallel solar assisted heat pump system.

In addition to the configurations discussed above, SAHP systems are broadly categorized into two-exchanger systems and three exchanger systems. Each category consists of versatile ways to connect the domestic loop to the refrigerant loop. The configuration of the heat pump is the main factor in the initial cost, operating cost and operational reliability [9]. Recent review articles have been directed towards the advances in direct expansion SAHP systems [10]. Direct expansion systems use the solar collector as the evaporator for the heat pump, and are suitable for mild climates. Several aspects of SAHP implementation in domestic applications are discussed. The review presented the system configuration, modelling, and optimization. It reported the escalating trend towards commercializing direct SAHP technology. In addition, it provided future recommendations to enhance system performance and widen its applicability. Several parameters that influence the performance of direct expansion SAHP have been discussed [11,12,13]. As expected, the COP was found to be significantly higher as ambient temperature increases or the irradiation increases [11,12]. Increasing the storage volume was found to enhance system solar fraction as cooler water enters the collector reducing its losses [13].

Researchers have investigated the application of indirect SAHP systems in residential applications [14,15,16]. The system performance is found to be sensitive to the initial water temperature in storage tank [14]. An improvement of about 16% in COP was observed when the initial water temperature was increased from 20ºC to 35ºC. However, there is a corresponding increase in energy consumption with increased water temperature. The collector size and storage volume have also a pronounced effect on system performance [15]. It is recommended not to oversize both the collector and the storage relative to the demand. Simplified analytical model can provide an insight into the correct collector area and storage volume based on the application [15]. A 50 liters storage volume for each m^2 of collector area is suitable for sizing residential systems. However, the reported results need further optimization for more accurate prediction. An indirect solar-assisted heat pump system was compared to the conventional solar domestic hot water system in the Canadian climate by Sterling and Collins [16]. They concluded that there is a slight improvement in solar fraction (around 8%) for the series SAHP system.

Parallel SAHP systems, as shown in Figure 3, allow for the heat pump to exchange heat directly with the outdoor air. A comparison between the parallel SAHP system and the conventional system has been conducted in different climates [17,18,19]. An energy saving of 70% can be achieved in the parallel system relative to a conventional system for Athens weather [17,18]. Both series and parallel SAHP configuration were compared for weather conditions in Turkey [19]. The series configuration was found to have higher COPs as the inlet temperature to the collector is lower which enhances heat pump performance. However, more than 26% enhancement in the free energy ratio (FER) was reported for the parallel system relative to the series system. The FER is defined as the ratio of the energy transfer from "free" sources (such as solar irradiation and the ambient air) to the energy required by the load. Comparison between different systems in terms of life cycle impact and materials used was conducted in Switzerland climate [20]. The study highlighted the potential of environmentally friendly performance for the parallel SAHP system relative to the conventional SDHW system as it reduced GHG emissions.

Different operational modes of the parallel SAHP system have been reported [21-27]. The increase in collected solar irradiance has positive effect on the COP. The COP is increased by about 11% for a to 40 m^2 collector case relative to no collector case[21]. The performance of different system configurations was studied for Madison, Wisconsin climate for space heating needs [22]. Collector efficiency was found to be higher for series configuration relative to the parallel configuration (50% and 30% respectively) since collector inlet temperature is reduced in the series configuration. In addition, a higher seasonal COP of the heat pump was found for the series configuration relative to the parallel configuration (2.84, and 2.0 respectively). In Turkey's climate, Kaygusuz and Ayhan [23] studied the same configurations reported by Freeman et al. [22], however, they incorporated phase change material within the tank. They reported a 16% increase in collector efficiency in the series configuration relative to the parallel configuration. In addition, higher COPs were found for the series configuration. In a later study, the performance of covered and uncovered flat plate solar collector in different SAHP configurations were compared [24]. Several operational modes and control strategies to alternate between air source heat pump and solar system were reported for the extremely cold climate of Daqing [25]. A solar fraction of 66% can be obtained in the cold day of December. In addition, collector efficiency can reach 51% using predetermined control strategies to maximize the benefit from both free sources.

The current paper presents a detailed annual analysis to compare the performance of the parallel SAHP system, series SAHP system, and conventional SDHW system for climate conditions of Toronto, Canada. The novelty of this work lies in the control strategy and analysis of system performance. The control strategy in the parallel system herein considered heat pump to replace auxiliary heater operation. The current work highlights clearly the importance of free air energy in enhancing the performance of parallel heat pump configuration. It also provides a thermodynamic background that shows that the indirect configuration loses this benefit. The analysis culminates in a thermodynamic description of the SAHP paradox which occurs when a heat pump is used in a series configuration with insulated solar collectors. The details of system models are given in the next section.

System description

Three systems are compared in the each hour in the morning period (between 6 am and 12 pm).It runs discrete periods varying in length between 6 minutes and 12current paper. The systems are a conventional SDHW system, a series indirect SAHP system, and a parallel SAHP system. Realistic weather data for Toronto, Ontario is applied. The same draw schedule and water delivery temperature are considered in all the systems. The dispersed load profile is used on the load side with a total daily demand of 160 liters (for a typical family of 4)[28]. The demand profile is shown in figure 4. Table 1 includes the common parameters for the three systems.

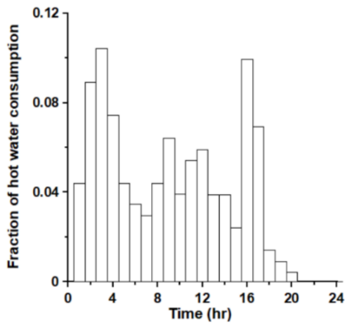


Figure 4: dispersed demand profile.

Table 1: Common parameters for the simulations

Parameter	Value
Collector area (flat, and evacuated)	6 m^2
Collector slope	45º
Collector loss coefficient for flat plate collector	5 W/m^2K
Collector mass flow rate	0.02 kg/s
Storage tank	200 liters, stratified
Storage tank loss coefficient	0.55 W/m^2K
Tank aspect ratio	2
Daily drawn volume of hot water	160 liters (Edwards et al., 2015).
Discharging flow rate	0.19 kg/s (Edwards et al., 2015).

The flat plate collector in the conventional SDHW system is modelled using the Hottel-Whillier-Bliss equation (standard type 1b in TRNSYS [29]). The net heat gain can be correlated to incident irradiation and losses as follows:

$$Q_u = F_R A_c \left((\tau\alpha)G - U_L(T_{in} - T_{amb})\right) \quad (1)$$

On the fluid side, the useful heat gain is expressed as:

$$Q_u = \dot{m} C_p (T_o - T_{in}) \quad (2)$$

Equating both equation 1, and 2 gives the outlet temperature of the collector in terms of inlet temperature as:

$$T_o = T_{in} + \frac{F_R A_c}{\dot{m} C_p} \left((\tau\alpha)G - U_L(T_{in} - T_{amb})\right) \quad (3)$$

A heat pump is an additional component in the series parallel SAHP systems. The heat pump is of type 927 in TRNSYS. An external file includes the characteristics of the heat pump. The catalogue [29] tabulates the heat pump COP versus inlet fluid temperature from the source side as shown in figure 5. This curve is assumed to be the same during the entire operation

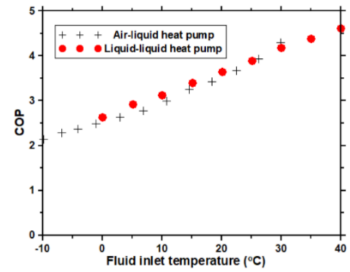


Figure 5: air-liquid and liquid-liquid heat pump coefficient of performance [29]

Details of TRNSYS simulation, component modelling, and several control strategies are presented in the appendix. The present simulation predictions have been verified against the results presented by Sterling and Collins [23]. They presented a comparison between a conventional SDHW system and a dual tank solar assisted heat pump system in a cold climate. The results of the present code were compared to their reported results with good agreement obtained. The details of the verification are shown in the Appendix.

Results and discussion:

Simulations using TRNSYS were performed for three systems: a conventional SDHW system, a series SAHP, and a parallel SAHP. The systems are depicted in Figures 1, 2, and 3, respectively. Since the cold climate of Toronto, Canada is considered, all systems use a glycol-water heat transfer fluid in the collector loop and are thus categorized as indirect systems. A dispersed demand profile is applied. Data for the solar collector and heat pumps are provided in the appendix. The systems effectiveness's were compared based on free energy ratio. It quantifies the contribution of free energy (solar and ambient air) on system performance.

The annual simulations predict a free energy ratio for the conventional SDHW system, the series indirect SAHP, and the parallel SAHP system to be 0.58, 0.62, and 0.88 respectively (Figure 6). This shows a considerable increase in performance for the parallel system relative to the other two.

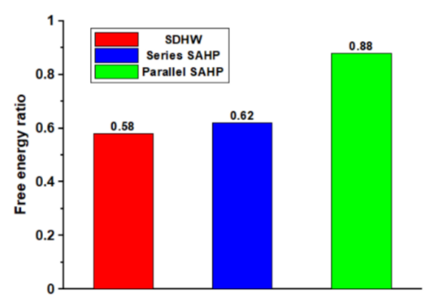


Figure 6: annual free energy

The improvement in the system free energy ratio can be explained by inspecting the normalized energy in different systems. To meet the load in the conventional SDHW system, useful solar energy and auxiliary heat are employed.

For the series indirect SAHP system, useful solar energy, and total electrical load (auxiliary heat, compressor work). For the parallel SAHP system, useful solar energy, air energy, and total electrical load are employed. The overall energy balance must satisfy the following equation:

$$q_{solar} + q_{air} + q_{elec} = Q_L \tag{4}$$

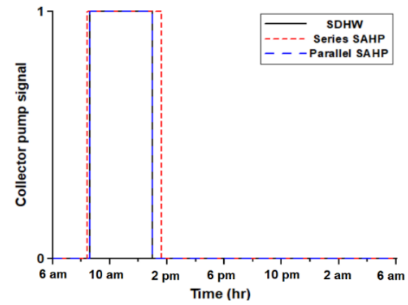
$$q_{elec} = W_{hp} + q_{aux} \tag{5}$$

By dividing the terms of equation (4) by Q_L, we get

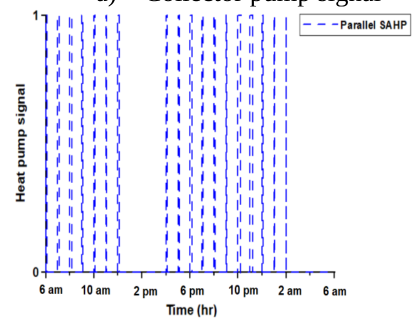
$$\frac{q_{solar}}{Q_L} + \frac{q_{air}}{Q_L} + \frac{q_{elec}}{Q_L} = 1 \tag{6}$$

where $\frac{q_{solar}}{Q_L}$ is the normalized solar energy, $\frac{q_{air}}{Q_L}$ is the normalized air energy, and $\frac{q_{elec}}{Q_L}$ is the normalized electric energy.

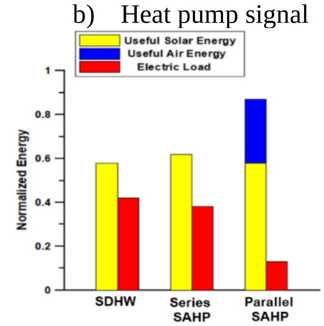
Figure 7 shows the detailed performance of the three systems in a selected spring day. The signal for the collector pump and the heat pump are shown in Figures 7a and 7b. The collector pump is switched on for approximately 4.4 hrs, 5.1 hrs, and 4.4 hrs for the conventional SDHW system, the series SAHP system, and the parallel SAHP system respectively (Figure 7a). The longer collector pump run time for the series SAHP system relative to the conventional SDHW system is attributed to the lower collector losses achieved with the series heat pump integration [32]. According to the control strategy of the parallel SAHP system, the heat pump is switched on when the temperature directed to the load is lower than the setpoint temperature. In that sense, it operates in a similar way to the auxiliary heater. It switches on during the start of each hour in the morning period (between 6 am and 12 pm).It runs discrete periods varying in length between 6 minutes and 12 minutes dependant on the demand profile (Figure 4).In addition, it is switched on during the late afternoon and night hours (4 pm to 2 am). Normalized energy for the three systems is depicted in Figure 7c. The figure shows that for the conventional SDHW system, 58% of the load is met by free solar energy and the rest is met through auxiliary heating. For the series indirect SAHP system, there is a slight increase in solar energy collected (from 58% of load to 62% of the load). This is due to the colder collector inlet fluid for the series SAHP which increases the pump run time. When the heat pump is in a parallel arrangement with the solar collector, there is a significant increase in system performance as a result of heat transfer from the ambient air to the load. In this case, 58% of the load is satisfied by solar contribution. In addition, 29% are satisfied by air contribution and the rest is supplied by electric load. This is reflected in the free energy ratios (figure 7d). Similar behaviour is shown for the rest of the seasons.



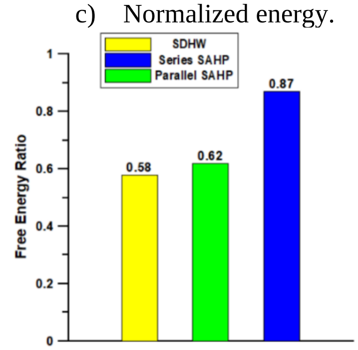
a) Collector pump signal



b) Heat pump signal



c) Normalized energy.



d) Free energy ratio

Figure 7: Detailed analysis of a typical spring day.

The SAHP Paradigm

From the previous analysis, it appears that the relative benefit of incorporating a heat pump into a solar thermal system is strongly dependent on the configuration. In northern climates where collector heat losses are significant, solar collectors are designed to minimize heat losses from the collector while maximizing solar irradiation collection. Heat pump operation, however, is based on enhancing heat exchange with the environment in order to capture "free energy". As such, in a series configuration, an insulated solar collector would appear to not benefit from the full capability of a heat pump for collecting energy from the ambient air; instead, the benefit is in the reduction in heat loss to the environment. The following thermodynamic background explains the reason behind this.

Thermodynamic background

Heat pumps extract energy from low temperature reservoirs, such as the ambient air or the ground. A simple energy balance on a conventional air-source heat pump (Figure 8) can be expressed as:

$$\dot{Q}_{\text{in}} + \dot{W}_c = \dot{Q}_H \tag{7}$$

where $\dot{Q}_{\text{in}}$ is the rate at which energy is extracted from the ambient air. $\dot{W}_c$ is the rate at which work is done by the compressor, and $\dot{Q}_H$ is the rate of heat transfer to the heated space.

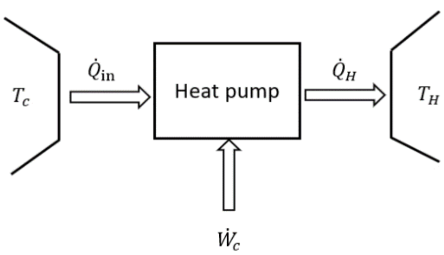


Figure 8: Energy balance for conventional air source heat pump.

The heat transfer from the outdoor air can be determined by considering an energy balance on evaporator coils as shown in figure 9. As the outdoor air passes over the evaporator coils, energy is extracted from the ambient and the temperature is reduced from T_∞ to $(T_\infty - \Delta T)$. This system is referred to as an "open system". For steady conditions, an energy balance yields:

$$\dot{m}_{air}\mathrm{C_p}\mathrm{T}_\infty - \dot{m}_{air}C_p(T_\infty - \Delta T) = \dot{m}_{evap}h_{evap,\,in} - \dot{m}_{evap}h_{evap,\,out} \tag{8}$$

where $\dot{m}_{evap}$ is the mass flow rate of refrigerant in the evaporator loop, $h_{evap,\,in}$ is the enthalpy of refrigerant entering the evaporator, and $h_{evap,\,out}$ is the enthalpy of refrigerant exiting the evaporator.

Therefore:

$$\dot{Q}_{in} = \dot{m}_{evap}(h_{evap,\,in} - h_{evap,\,out}) = \dot{m}_{air}C_p\Delta T \tag{9}$$

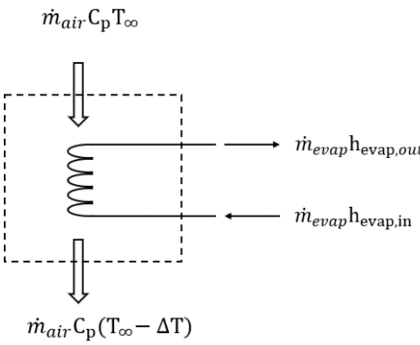


Figure 9: Energy balance on the evaporator.

Returning to the heat pump energy balance:

$$\dot{Q}_H = \dot{Q}_{\text{in}} + \dot{W}_c = \dot{m}_{air}C_p\Delta T + \dot{W}_c \tag{10}$$

The term "$\dot{m}_{air}C_p\Delta T$" is essentially the "free energy" provided by the environment to the load. The magnitude of this term is dependent on the COP of the heat pump and reduces as the outdoor temperature drops. The COP can be defined as follows:

$$COP = \frac{\dot{Q}_H}{\dot{W}_c} = \frac{\dot{Q}_{\text{in}} + \dot{W}_c}{\dot{W}_c} \tag{11}$$

Substituting equation (11) into equation (10) and rearranging yields:

$$\dot{m}_{air}C_p\Delta T = \text{free energy heat transfer rate} = (COP - 1)\dot{W}_c \tag{12}$$

To enhance the rate at which "free energy" is extracted from the environment, there is a motivation to increase the heat pump COP. The COP can be increased by somehow increasing the outdoor air temperature. Solar collectors provide an opportunity to achieve this by preheating outdoor air upstream of the evaporator coils. This is shown schematically in figure 10. With a solar collector, the inlet temperature to the evaporator is given by:

$$\mathrm{T_{in}} = \mathrm{T}_\infty + \left(\frac{\dot{Q}_{sun} - \dot{Q}_{loss}}{\dot{m}_{air}\mathrm{C_p}}\right) \tag{13}$$

From an energy balance, the rate of heat transfer to the heated space is given by

$$\dot{Q}_H = \dot{W}_c + \dot{m}_{air}\mathrm{C_p}\Delta\mathrm{T} + (\dot{Q}_{sun} - \dot{Q}_{loss}) \tag{14}$$

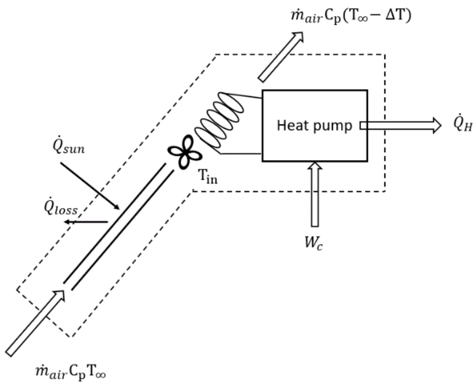


Figure 10: open loop solar-assisted heat pump

In contrast to the "open" system shown in figure 10, solar assisted heat pumps are typically configured as a "closed" system as previously shown in figure 2. The fluid that passes over the evaporator coils is contained in a closed loop. The connection to the environment (normally the cold body) is through heat exchange with the flat plate solar collector.

By considering an energy balance on the closed loop SAHP system, we obtain:

$$\dot{Q}_H = \dot{W}_c + (\dot{Q}_{sun} - \dot{Q}_{loss}) \tag{15}$$

Comparing equation (15) to equation (14), it is seen that the "open" system contains an additional term: $\dot{m}_{air}C_p\Delta T$. This term does not appear in the closed loop system because the fluid that passes over the evaporator is not ejected to the environment. This cold fluid must be reheated as it passes through the solar collector. Moreover, the term $\dot{m}_{air}C_p\Delta T$ represents the "free energy" which is obtained using the heat pump. In a closed system where the heat losses from the solar collector are intended to be reduced, the true benefit of a heat pump (i.e. the capture of "free energy") is lost. In a closed SAHP context, the role of the heat pump is then to lower collector temperatures and thus reduce collector heat loss. As a result of this paradigm which applies to a series arrangement, the parallel arrangement provided significantly improved system performance.

Conclusions

The paper presented a comparison between three different configurations. The configurations included a conventional SDHW system, a series indirect SAHP system, and a parallel SAHP system. The analysis was performed on a high-resolution annual basis for Toronto, Canada climate. It is concluded that for the case study presented, there is a 4% improvement in the free energy ratio in the indirect series SAHP system relative to the conventional system. There is an additional 24% improvement in the free energy ratio of the parallel SAHP system relative to the series indirect SAHP system (i.e. 28% improvement in free energy ratio relative to the conventional system). This is attributed to the contribution of two sources of free energy (solar and air) in the parallel SAHP system. The paper shows that there is a great potential for the parallel SAHP configuration in cold climates. It also highlights the superiority of parallel coupling relative to the serial one to maximize the free energy ratio.

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Appendix:

<u>Details of TRNSYS simulation and component modelling</u>

The present study compares the performance of flat plate conventional SDHW system, series SAHP, and parallel SAHP systems. The schematics of the three systems in the TRNSYS environment are shown in figures A.1, A.2, and A.3. The common components in the three systems are:

<u>The water storage tank:</u> the water tank is a 200 L stratified tank. In TRNSYS (type 4a), the tank is divided into 60 equal-sized segments to account for stratification within the tank. The stratified model ensures that water layers are arranged in a descending order of temperature from top to bottom.

<u>The auxiliary heater</u> If water extracted from the tank top is colder than the setpoint, the auxiliary heater is switched on (type 6).

<u>The flow mixer</u> is used in case of extracting water of temperature higher than the set point. Cold water is mixed with the extracted water using the flow mixer (type 11b) to lower its temperature to set point.

<u>Water pumps, controller, and heat exchanger:</u> A fixed displacement pump (type 3b) circulates fluid in the collector loop. The pump is switched on based on the temperature difference between the collector surface and storage tank bottom. This temperature difference is sensed by a differential controller (type 2b). The pump is switched on when the temperature difference reaches 7 ºC and is switched off when it drops below 2 ºC. There is another pump of the same TRNSYS type that circulates the domestic hot water between the heat exchanger and the storage tank. Due to the climate considered, a glycol solution is circulated as an antifreeze in the collector loop. The antifreeze composition is a 50%:50% glycol-water mixture. A heat exchanger (type 5b) is used to transfer the heat absorbed in the glycol loop to the domestic hot water loop. The overall heat transfer coefficient of the heat exchanger is 3000 kJ/hr K. The overall effectiveness is 0.9.

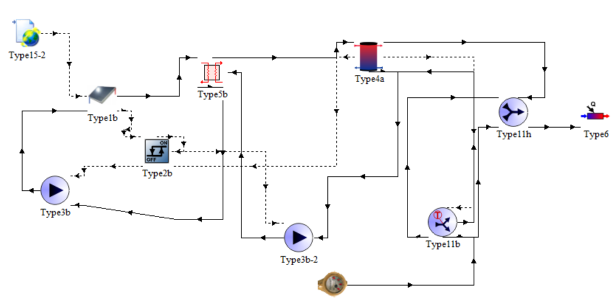

Figure A.1: Schematic of conventional indirect solar domestic hot water system using flat plate collector in TRNSYS environment

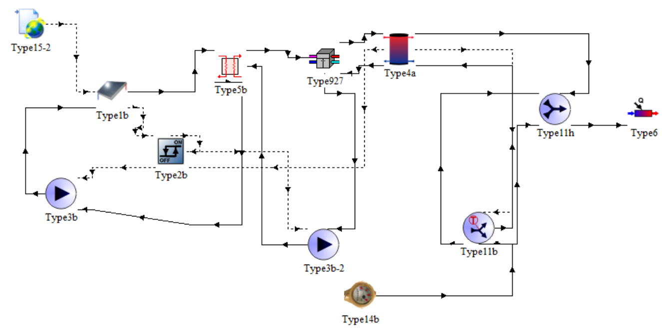

Figure A.2: Schematic of indirect solar assisted heat pump system in TRNSYS environment.

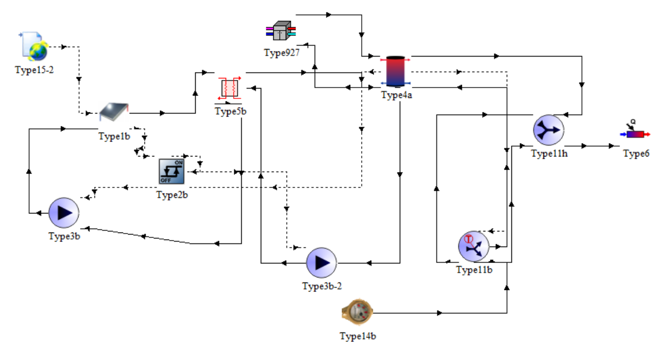

Figure A.3: Schematic of parallel solar assisted heat pump system in TRNSYS environment.

The present simulation predictions have been verified against the results presented by Sterling and Collins [23]. They presented a comparison between a conventional SDHW system and a dual tank solar assisted heat pump system in Ottawa climate. The schematic of their dual tank system is shown in figure A.4. The results of the present code were compared to their reported results.

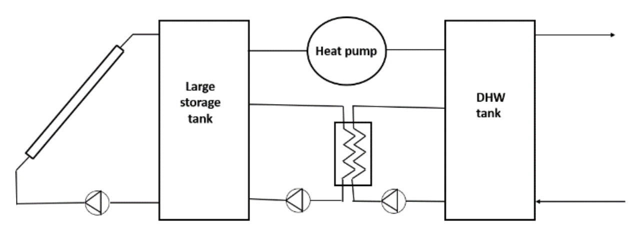

Figure A.4: schematic of dual tank solar assisted heat pump system.

Annual simulations have been performed in TRNSYS [29]. Individual values of input and output energy to both systems are calculated. The systems were simulated using our developed model. The comparison of results is shown in Table A.1. There is about 3% between the solar fraction predicted by present code and their results. Additional verification for the SDHW system performance prediction in TRNSYS was previously presented by Teamah et al. [31].

Table A.1. Comparison between present simulation results and Sterling and Collins results

	Sterling and Collins SDHW	Present results SDHW	Sterling and Collins SAHP	Present results SAHP
Solar fraction	**0.58**	**0.60**	**0.67**	**0.66**