INVESTIGATE THE POTENTIAL USE OF ABSORPTION CHILLER FOR WASTE HEAT RECOVERY OF DATA CENTRE; Case Study of Earth Ranger Centre

by

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Abstract

Data centres are great contributors to global warming due to their high energy consumption. The same amount of energy being used in data centres to run the servers is required for cooling to prevent any damage as the result of overheating of the equipment. Reusing the exhaust heat for cooling purposes reduce energy demand for cooling and heat removal systems. This research proposal has focused on waste heat recovery of data centres located in Woodbridge, Ontario by using absorption chillers. A mathematical model was developed to analyze the performance of the system based on the temperature of the generator and absorber in Simulink/MATLAB. It is concluded that the COP of the system improves with the increase of the generator temperature and the decrease of the absorber temperature. Ultimately, a secondary energy source is required along with the rejected heat from the data centre to support the cooling load.

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1. Introduction

Data centre industry has grown significantly, which has led to an increase in power density of server components and electricity consumption. Data centre energy use for Canada is estimated to be 6,056 kWh/m^2 [1]. Almost half of it is consumed by computing servers, 40% is dedicated for cooling the systems [2], and the rest is consumed by other facilities. It was stated by Ebrahimi et al. [3], that almost all the energy consumed in data centres is converted to waste heat, which in Canada it leads to 2,422.5 kWh/m^2 electricity turning to waste heat throughout a year. Therefore, the same amount of energy is required for cooling the systems to prevent any damage as the result of overheating of the equipment. Capturing the exhaust heat and reusing it for cooling purposes would result in reduction of energy demand for cooling and heat removal systems.

The problem with the data centre energy recovery is the low-quality dissipated heat. Prior to the literature review a detailed analysis of data centres energy consumption and thermal loads is discussed. In the literature review conducted for this research proposal absorption chiller and solar absorption chiller are discussed as the heat recovery strategy for waste heat of data centres. A comparison between heat recovery techniques and their effectiveness for data centre applications has been analyzed by Ebrahimi et al. [4], and absorption refrigeration was introduced as the most efficient technology for heat recovering in data centres.

This paper focuses on the heat recovery for a data centre in an educational facility using absorption chiller in Woodbridge, Ontario. A comprehensive literature review on energy performance of the absorption chiller and solar absorption cooling system has been conducted. The aim of the research is to analyze the potential use of absorption chillers for waste heat recovery of the data centre in Earth Rangers Centre, based on the available dissipated heat. Additionally, the relationships between the COP of the system and the temperature of the generator and absorber have been studied throughout this research. Ultimately, the necessity of adding a secondary energy source has been discussed.

2. Overview of Data Centres

Prior to the literature review, an overview of data centres is brought in accord to their physical organization, thermal loads and temperature limits, and power consumption to provide a better understanding of the functionality of space. Additionally, typical cooling systems used for data centres is discussed.

2.1 Physical Organization

A data centre is a space that locates the servers, switches, and storage facilities, which should be kept in certain temperature and relative humidity to operate efficiently to avoid any damage [4]. The data centre could be the size of a room or a warehouse depending on the scale of the business. Rack is a metal frame to keep the equipment. Therefore, it could be one single rack of equipment or multiple racks and cabinets in a data centre [5]. Typical dimension of a standard rack is 0.5m x 0.7m with the height of 2m [5]. One-unit height of a rack is called 1U, which equals to 44.45 mm and depending on the type of processors, the mounted equipment takes space inside the rack [5]. Other equipment which might be installed in a rack are power distribution units and built-in KVM (Keyboard, Video, Mouse) switches [5]. A series of racks in a data centre is called server farm [6], and they can each contain several servers.

Power supply equipment in data centres is usually provided through a high voltage transformer, which serves as an uninterruptible power supply (UPS) unit [6]. Power distribution units (PDUs) are supplying the power to the ICT (Information and Communication Technology) equipment according to the demand of servers in the racks [6]. However, backup generation is always available in case of power supply failure from utility companies. Furthermore, another important component of data centre is energy storage to bring down the start-up time of systems to almost zero [6]. A server contains microprocessors, additional memory (DIMMS), auxiliary components mass storage, a power supply [4].

2.2 Data Centre Power Consumption, Thermal Loads and Temperature Limits

With the growth of demand and technology, ICT services have been designed to operate faster needing for more electricity. Therefore, electricity consumption and heat production increased significantly, which required more energy for both running and cooling the systems. Murphy et al. [7] stated that data centres can be considered as thermal energy resources due to their extensive dissipated heat. The dissipated energy flux was 430 – 861 W/m^2 in the early generation of data centres and it has risen significantly to almost 6,458 – 10,764 W/m^2 [8]. ASHRAE estimates that power consumption of a single rack, containing extreme density communicating equipment, and a single rack equipped with high density computer servers to be 60 kW and 35kW, respectively. [9]. Furthermore, manufacturers should be able to provide a balance between maintaining the temperature limit and the ability of handling the increasing thermal load for electronic devices for operating at a safe level. Other than microprocessors, storage devices and hard disks also contribute to the power consumption. Dissipated power from a hard disk can reach 12 kW [8], while storage devices consume 20-30% of the total power. A typical data centre characteristic is brought in table 2-1.

Most of the researches related to the electronics thermal management, considered 85°C as the maximum allowable junction temperature for the safe and effective operation of microprocessors [8]. Oró et al [10], specified the allowable dry bulb temperature and relative humidity of the date centre based on their scale and security risks, to be in the range of 5 to 45°C and 8 to 90%, respectively. Table 2-2 demonstrates the ASHRAE air temperature and humidity ranges for data centres [11]. Data centres are categorized according to the three factors of uptime, redundancy and hours of downtime per year. Class 1 data centre are utilized by small businesses and has lower uptime, while operating on a yearly basis with no redundancy [12]. Class 2 data centre has a redundant-capacity component site infrastructure [12]. In other words, it has partial redundancy in power and cooling. Class 3 and 4 have the lowest downtime in compared to class 1 and 2. In addition, they have power outage protection, which prevents any disturbances to the services to end users, while maintaining them [12].

Table 2- 1: Summary of typical data centre thermal loads [4]

Power loads

Components	**Thermal Load**
Processors	60 – 75 W each
DIMM	6 W
Auxiliary power per server	150 – 250 W
Total power per server	300 – 400 W
Rack Capacity	1U servers, up to 42 per rack Blade servers at 10U, up to 64 per rack
Total rack power	13 – 26 kW
Total power per data centre	3.2 – 6.5 MW

Table 2- 2: Thermal guidelines for data centre [11]

	Dry-bulb temperature	Humidity range	Maximum dew point
Class A1 and A4	**Recommended** 18 to 27	5.5 °C DP to 60% RH and 15 °C DP	-
Class A1	**Allowable** 15 to 32 °C	20% to 80%	17°C
Class A2	10 to 35°C	20% to 80%	21 °C
Class A3	5 to 40 °C	8% to 85%	24 °C
Class A4	5 to 45 °C	8% to 90	24 °C

2.3 Management of waste heat sources and streams in data centre cooling system

The cooling system in a data centre is called Computer Room Air Conditioning (CRAC) and controls the temperature and relative humidity of indoor air [4]. The size and capacity of the CRAC system for each data centre is designed by considering it to operate at full capacity. It means all the server farms are running at their highest speed. Furthermore, the designed forced-air cooling system for older data centres cannot be implemented for the newly designed high-power data centres [13]. It is because the newer systems are more advanced and require more energy, which result in higher cooling load. The typical cooling systems used in data centres are air-cooled systems, water-cooled systems, and two-phased cooled system [4].

In an air-cooled data centre, the configuration of the racks has been set to provide cold and hot aisles [4] (Figure 2-1). Cold aisles are the ones that the front of the server racks is facing each other and intake the cooled air supplied to them. On the other hand, the rear side of the server racks are facing each other, and the dissipated air from each server rack is being returned to the CRAC. Water-cooled or liquid-cooled systems provide a closed water or liquid loop going inside of each rack to capture the dissipated heat from servers [14] and bring it to either liquid or air heat exchanger (see Appendix A-1 – A-3). The captured heat through water or liquid cooled systems provide a higher quality waste heat available for heat recovery [4]. Lastly, on-chip two-phased cooled systems uses a liquid or refrigerant as the primary coolant to capture both latent and sensible heat from the main processors and auxiliary electronics [13].

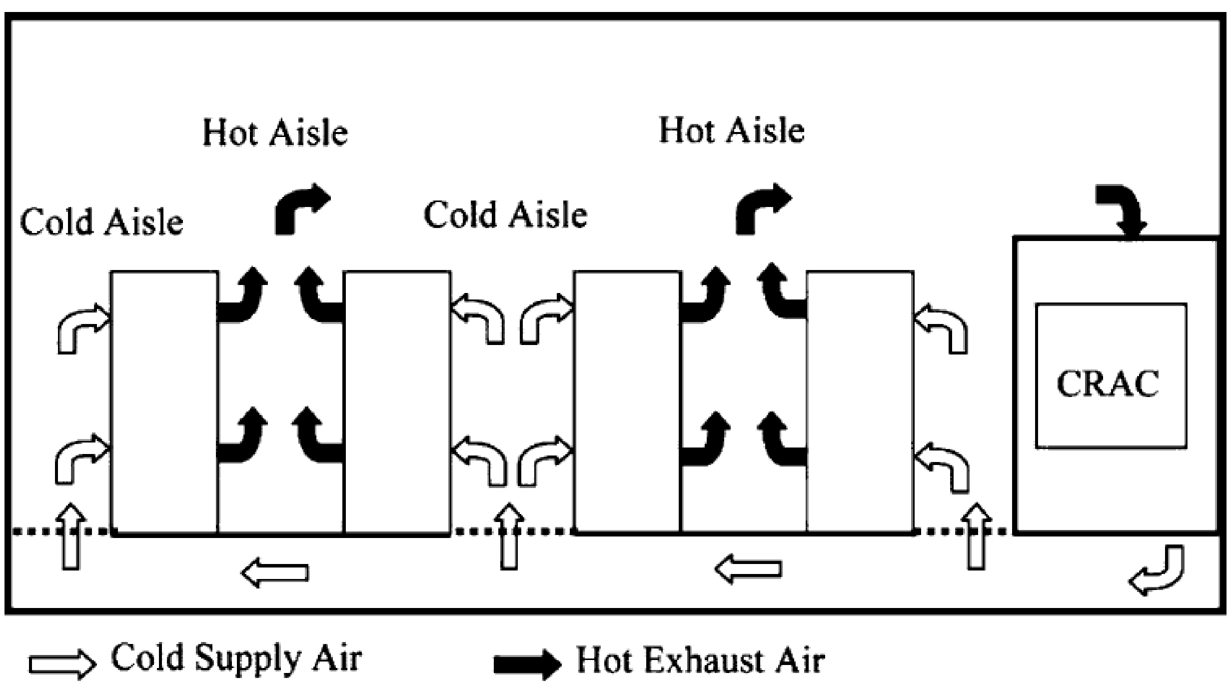


Figure 2- 1: Typical air-cooled system supply and return air flow [15]

3. Literature Review

Data centres are great contributors to global warming due to their high energy consumption. Almost all the cooling energy used for keeping the indoor air temperature of data centres in the safety range would end up as low-grade waste heat [4]. Ebrahimi et al. [4] introduced absorption chillers as one strategy towards heat recovery of the dissipated heat from data centres. However, they have noted that the main barrier to the implementation of waste heat recovery in data centre is the low temperature of the dissipated heat from the equipment. The literature review for this proposal is related to the waste heat recovery in data centres using absorption chillers. In addition, a literature review on energy performance of solar powered absorption chillers is brought.

3.1 Absorption Chillers

The primary energy for most of the HVAC systems is electricity, while the absorption chillers use heat. The primary approach is to analyze the implementation of absorption chillers that use the wasted heat to provide cooling. Haywood et al. [16] have done research on a single effect lithium bromide-water (LiBr- H_20) absorption chiller for heat recovery of a data centre and reducing the cooling load on the CRAC. This system uses Li-Br as the absorbent and water as the refrigerant [16] (see Appendix A-4). The variables effecting the performance of the system are temperature of chilled water inlet, flow rate of the heat medium, and the temperature of the input heat, which identify the resulting cooling capacity [16]. They have analyzed the performance of an existing absorption refrigeration system and then proposed it for use in data centre. From the actual system's data sheet, they observed that "the hotter the medium and the lower the cooling water temperatures, the more cooling capacity will be generated" [16].

The system designed by Haywood et al. [16] considered the dissipated heat only from 70% of the components inside a 42U rack. They have designed the space to contain two rows with two racks. Figure 3-1 illustrates the heat flow from data centre designed for system analysis. It can be seen that the captured heat from data centre ($Q_{HiT,S}$) can be sent to the LiBr absorption chiller, or to the thermal storage. In addition, a portion of heated air ($Q_{Air,C}$) can be removed directly by absorption chillers, which increases the amount of heat available for the system to be cooled. However, $Q_{Air,C}$ has lower quality heat compared to the dissipated heat from the ICT systems due to its lower temperature. The remaining heat inside the room is removed by CRAC or by economizer, which is rejected to the cooling tower. Their challenge was to transfer heat from the ICT systems to the generators of the absorption chiller with minimum heat loss. They have proposed a water-cooled heat sink to be in thermally contact with the ICT equipment, since water has higher heat capacity compared to air [16].

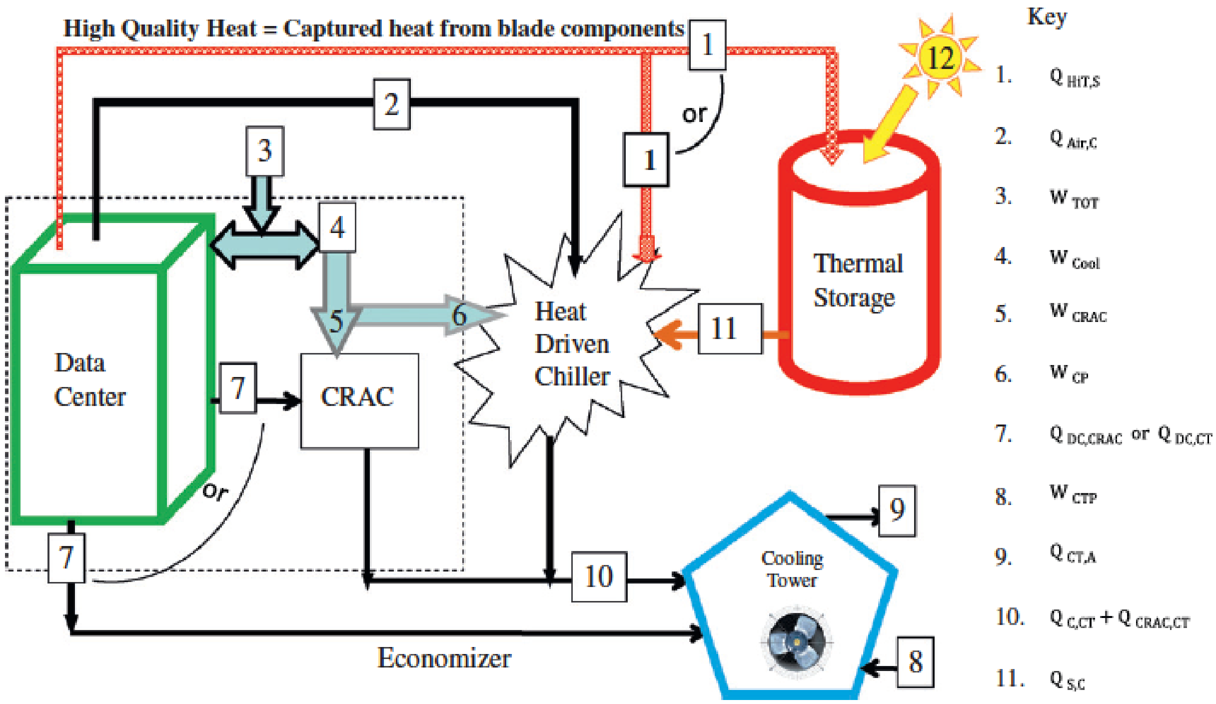


Figure 3- 1:Overall system level diagram showing the work and heat flow paths [16]

A benchmark to analyze data centre efficiency called Power Usage Effectiveness (PUE), has been introduced for the research. It illustrates the relationship between the total amount of power consumed by ICT systems and total facility power [17]. The purpose of using this equation is to "characterize the fraction of the total data centre power consumption devoted to IT equipment" [16]. The ratio between electrical utility power and ICT system power consumption is shown in Eq. (1), where $\dot{W}_{TOT}$ is total facility power and $\dot{W}_{IT}$ is computer rack power. In addition, total facility power is the summation of computer rack power ($\dot{W}_{IT}$), power delivery loss to the racks through PDUs ($\dot{W}_{Loss}$), and power consumed for cooling system ($\dot{W}_{Cool}$). It is noteworthy that energy consumption for lighting was considered to be negligible during normal operation [16]. The goal would be to bring the PUE below one, which means the energy generated from data centre is more than the input electrical energy. In other words, it demonstrates the relation between electric input to the IT equipment and total power consumption. Haywood et al. [16] expected that a PUE < 1 was possible with the aim of external alternative energy source.

$$PUE = \frac{Electrical\ utility\ power}{IT\ equipment\ electric\ power} = \frac{\dot{W}_{TOT}}{\dot{W}_{IT}} \quad (1)$$

The breakdown of the power consumption has been calculated by Eq. (2) and Eq. (3).

$$\dot{W}_{TOT} = \dot{W}_{IT} + \dot{W}_{Loss} + \dot{W}_{Cool}, \quad (2)$$

$$\dot{W}_{Cool} = \dot{W}_{CRAC} + \dot{W}_{AC\_FANS} + \dot{W}_{CP} \quad (3)$$

$\dot{W}_{CRAC}$ is the power consumed to run the vapor compression computer room AC, and $\dot{W}_{AC\_FANS}$ is the power required for operation of supply and condenser fans, $\dot{W}_{CP}$ is the power used by heat driven chiller [16]. By using the COP equations and considering highest heat fraction and capture fraction, the final equation for PUE was simplified to Eq. (4) shown below, where HFF is the highest heat fraction and CF is capture fraction [16]. Capture fraction refers to the maximum amount of heat that can be recovered and highest heat fraction determines the amount of heat dissipated by CPUs [16].

$$PUE = 1 + \frac{\dot{W}_{AC_{FANS}} + \dot{W}_{CP} + \dot{W}_{Loss}}{\dot{W}_{IT}} + \frac{1}{COP_{CRAC}}\left[1 + \frac{\dot{W}_{Loss}}{\dot{W}_{IT}} - HHF \cdot CF - COP_C\left(HFF \cdot CF + \frac{\dot{Q}_{EXT,S}}{\dot{W}_{IT}}\right)\right] \quad (4)$$

The mathematical model was tested for an experimental data centre, which had an area of 15 m^2 running at full capacity [16]. A typical vapor compression unit was selected for the experiment. In addition, COP_C was analyzed for three cases of design point, minimum chiller capacity and maximum chiller capacity [16]. Figure 2-3 illustrates the result of PUE relative to high temperature capture fraction. The external heat source supplying required heat to the generator was modeled and the results showed the value for PUE was below 1.

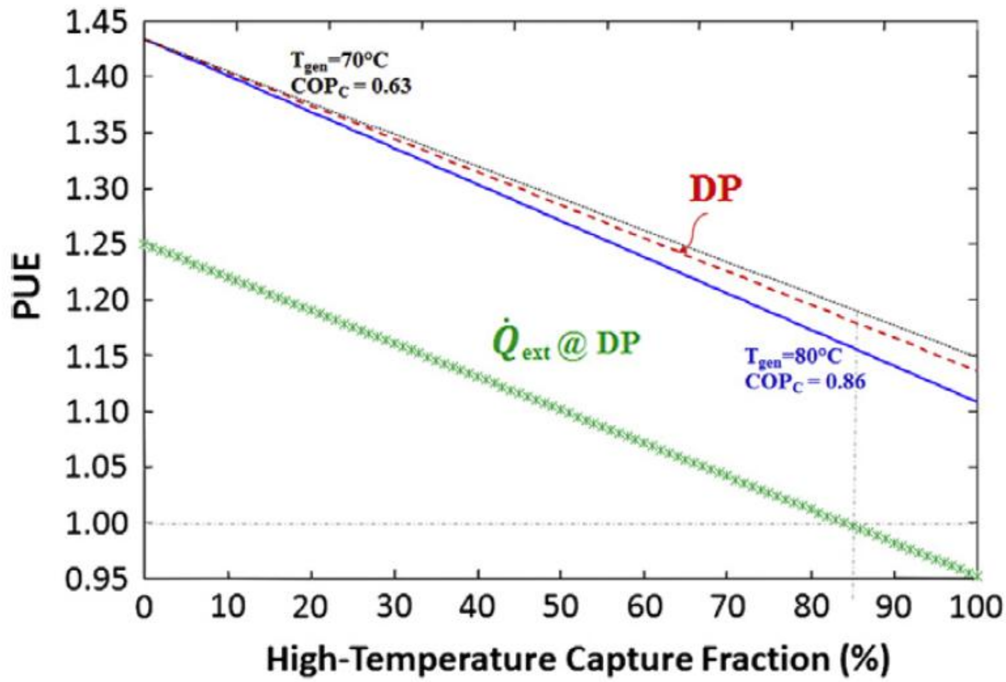


Figure 3- 2: PUE versus high-temperature capture fraction. $\dot{Q}ext$ *is the external heat supplied to the generator at design point condition* [16]

It was concluded that in order to reduce the power consumption of data centres, at least 85% of the dissipated heat should be captured and transported efficiently to the absorption chiller [16]. Furthermore, by adding external heat sources (i.e. solar thermal), it would be possible to reach a power usage effectiveness of less than 1 [16]. Limitation and uncertainties of this paper are related to the energy consumption and heat losses through the refrigeration cycle and capturing heat from the external heat sources. In addition, by considering the external heat sources being used to supply the generator, the overall performance of the system should be analyzed. Given the above, the study focused on a data centre running at full capacity, which would question the performance requirements for other operating conditions.

Another research on using absorption chillers in data centre by Ebrahimi et al. [3] has analyzed a thermodynamic model for a steady state waste heat recovery for a novel configuration of on-chip two-phase cooling system [3]. In addition, the thermodynamic analysis has been done for both lithium bromide-water and water-ammonia solutions in order to compare the COP of the system running with each solution. The energy balance between the dissipated heat and cooling capacity of absorption chillers has been studied. The combination of on-chip two-phase cooling system and absorption chiller was achieved through replacement of condenser in the cooling system, with the generator of the absorption chiller [3]. Variables undertaken for the thermodynamic modeling to calculate the heat transfer rate and COP, were temperature of the absorber, generator, condenser, mass flow rate of the LiBr-water and water-ammonia solutions, and heat transfer coefficient of the solution heat exchanger [3]. It should be mentioned that all the assumptions for modeling the system in the operating condition were listed, such as the negligible temperature variations in the evaporator, condenser, absorbent, and generator.

Schematic diagram of LiBr-water absorption refrigerant is shown in Appendix A-5. The COP of the LiBr-water solution was calculated according to Eq. (5), where $\dot{m}_r$ is the mass flow rate of refrigerant and h_i is the enthalpy for state i based on the diagram:

$$COP = \frac{\dot{Q}_e}{\dot{Q}_g} \tag{5}$$

Where;

$\dot{Q}_e = \dot{m}_r\,(h_{10} - h_9)$ and $\dot{Q}_g = \dot{m}_r h_7 + \dot{m}_6 h_4 + \dot{m}_1 h_3$ (6, 7)

The model was run with a reference data in order to verify the thermodynamic model [3]. There were two LiBr/Water concentration calculated for the model for both week and strong solutions. Therefore, the mass flow rater for each, week and strong solutions was determined by using $\dot{m}_r$. It should be mentioned that the inputs of the model were temperature at the absorber, evaporator, condenser, and generator. The effectiveness of heat exchanger was taken into an account. Additionally, the water-ammonia absorption refrigerant was modeled as shown in Appendix A-6. The main difference between the two solutions is level of toxicity and crystallization. LiBr-water solution has no toxicity but the chance of crystallization, while water-ammonia solution is a toxic material with no chance of crystallization. Therefore, the system required modification for water-ammonia solution. In the water-ammonia absorption refrigeration, water is the absorbent in compared to the LiBr-water solution, where LiBr is the absorbent [3]. The rectifier is necessary since there is water in the ammonia vapor leaving generator, which affects the performance of the whole system [3]. COP of the water-ammonia absorption refrigeration system was calculated by using Eq. (5), where:

$$\dot{Q}_e = \dot{m}_{12}(h_{13} - h_{12}),\ \dot{Q}_g = \dot{m}_8 h_8 + \dot{m}_5 h_5 + \dot{m}_4 h_4 \quad (8, 9)$$

The novel heat recovery system combining the above absorption refrigeration systems and on-chip two-phased cooled system has been analyzed. Figure 3-3 illustrates the conversion of wasted heat to cooling in waste heat recovery system. The heat from the server racks has been delivered to the generator of the absorption chiller [3]. To illustrate the COP of the novel system, the compressor power consumption ($\dot{W}_p$) has been used instead of energy consumption of compressor ($\dot{Q}_p$). Therefore, the COP is defined as Eq. (10):

$$COP = \frac{\dot{Q}_e}{\dot{W}_p} \quad (10)$$

Moreover, the model has been run for four alternative refrigerants such as HFC 245fa, HFC 245ca, HFC 1234ZE, and water [3]. The refrigerants were used for the on-chip two-phased cooled system. Figure 2-7 shows the results of COP for the whole system against temperature of the generator, when four types of refrigerants were used for on-chip two-phased cooled system. It was concluded that the LiBr-Water solution provided better outcome in compared to the water-ammonia solution in the absorption refrigeration cycle [3]. Although, HFC1234ZE provided higher COP for the novel heat recovery system compared to water. Therefore, coolant type would impact the overall efficiency of the equipment. It has been mentioned that the dissipated heat from the servers has been upgraded before delivering it to the generator, which brings the question on what has been done towards upgrading. Furthermore, it should have been mentioned what type of secondary heat source has been implemented for generator.

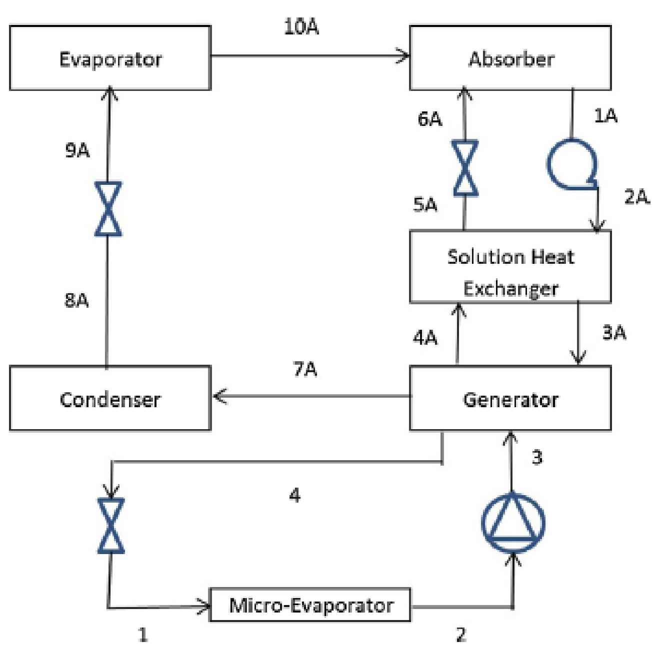


Figure 3- 3: Diagram of novel waste heat recovery system [3]

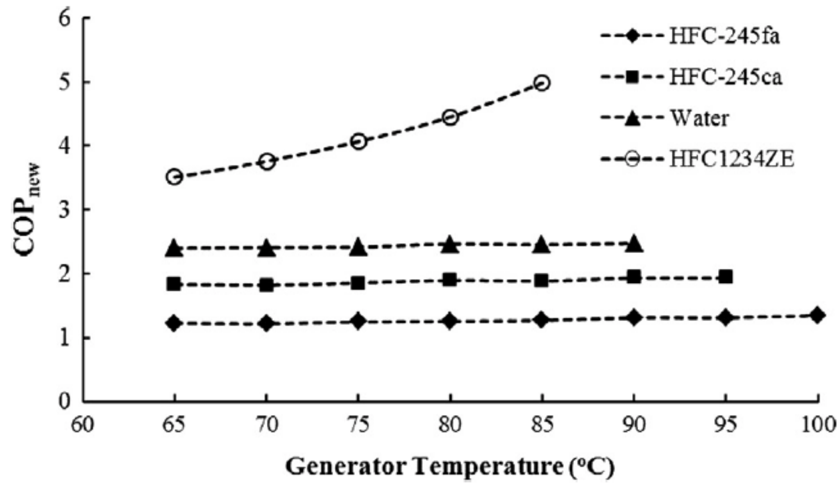


Figure 3- 4: The effect of upgraded waste heat temperature on the COP for the novel heat recovery system [3]

LiBr-water absorption chiller was modeled based on mathematic equations by Osta-Omar and Salem [18]. This research emphasized on the dependability of COP of the absorption chillers on system temperatures. Additionally, it was noted that the increase in temperature of the generator increased the COP of the system [19]. Therefore, temperature of the generator has the most influence on the COP. The increase in the temperature difference between condenser and the absorber would lower the COP of the system and the cooling effect [20]. Also, the study done by Micallef and Micallef [21] indicated that the increase in temperature of the generator or the decrease in the temperature of the absorber would increase the COP of the system and ultimately, the cooling effect. Osta-Omar and Salem modeled the absorption chiller based on Figure 3-5. It was assumed that the refrigerant at the exit of the evaporator is saturated vapour, and solutions at the outlet of adiabatic absorber and the generator are saturated liquid at the unit's temperature [18]. The mathematical model was based on the equilibrium of the pressure between LiBr-Water solution in the absorber and the refrigerant in the evaporator, and the equilibrium of the pressure between the refrigerant in the condenser and the solution in the generator [22].

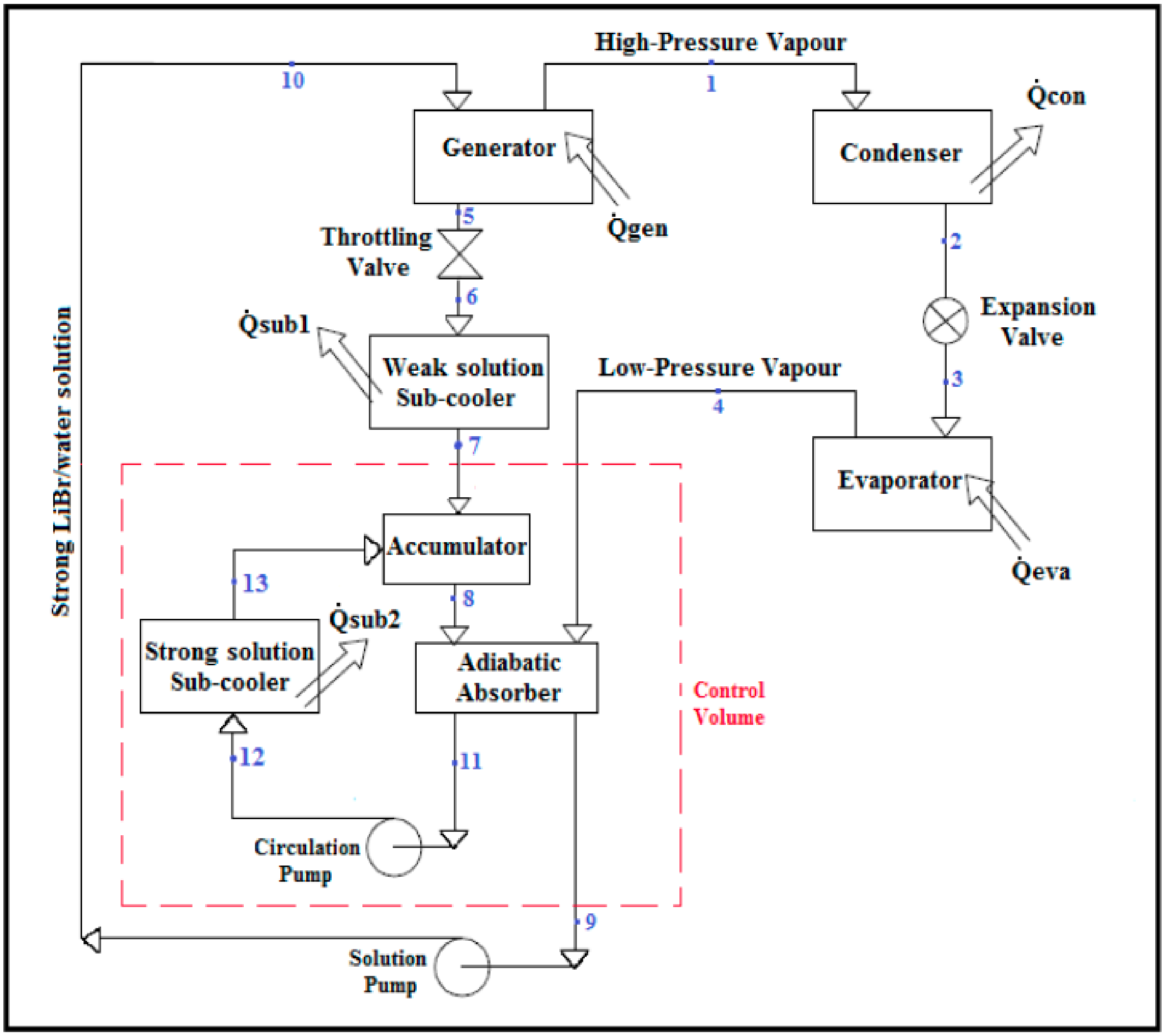


Figure 3- 5: Schematic diagram of a LiBr-Water absorption refrigeration system equipped with an adiabatic absorber **[18]**

The steady state energy equations were used for modeling the system and the model was run with MATLAB based on an algorithm illustrated in Figure 3-6, and invers matrix techniques were used to solve the unknown variables. According to the Osta-Omar and Salem, the COP of the vapor-compression refrigeration cycle can be defined as a ratio of the refrigeration rate to the mechanical work supplied to the compressor [18].

However, “the COP of the absorption chiller is defined as a combination of power cycle and refrigeration cycle” [18]. Therefore, the results indicated that the COP of the system is lower than that of the vapor-compression refrigeration cycle. The research analyzed the COP of the system, the mass concentration of the solution, and the heat transfer in each component according to the specific input design. It was recommended that the weak solution sub-cooler can be replaced by a solution heat exchanger in order to enhance the performance of the system with the same operating process. Lastly, it was recommended that in order to reach higher COP and generator temperature at 80°C, the system can be connected to a solar collector [18].

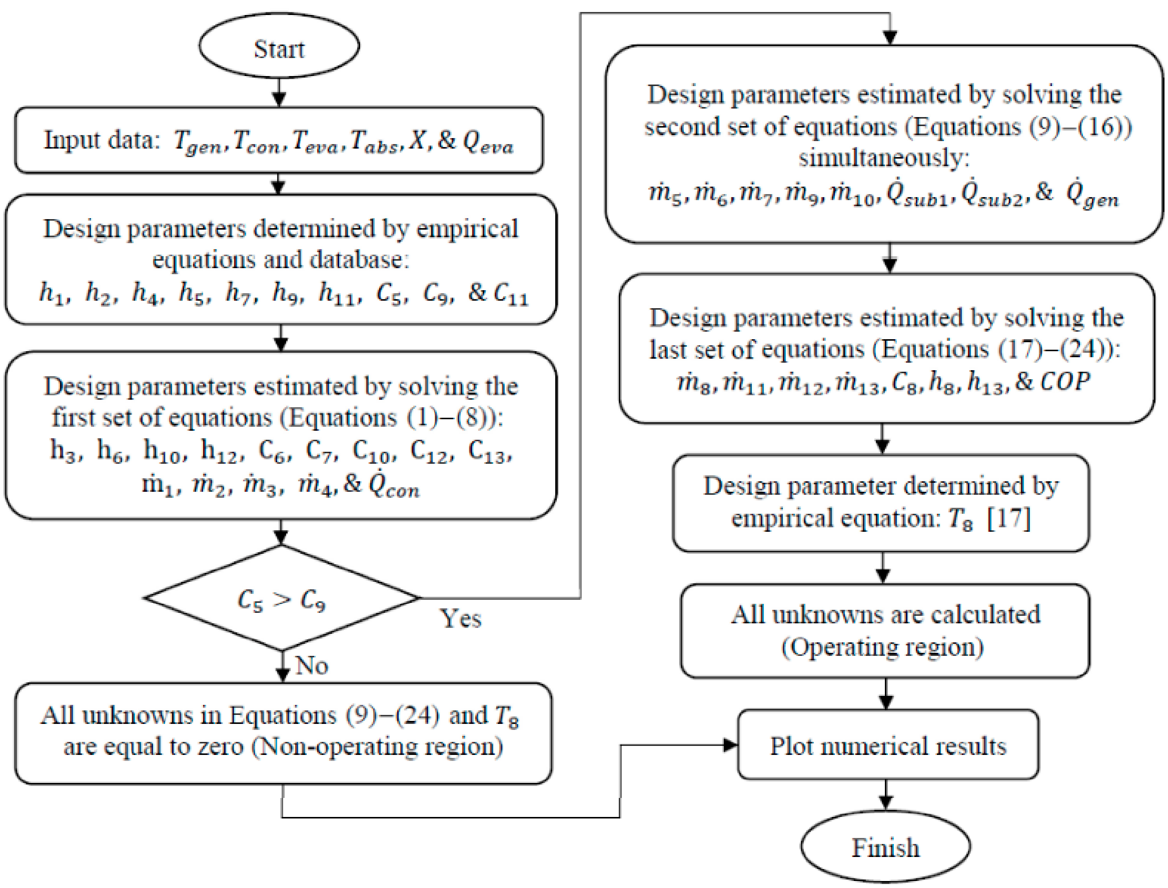


Figure 3- 6: Program flowchart of an LiBr-Water absorption refrigeration system equipped with an adiabatic absorber **[18]**

3.2 Solar Absorption Chillers

Using renewable energy sources such as solar energy as the alternative heat source for absorption chillers would decrease the carbon and GHG emissions. Shirazi et al. [23] have done a parametric study on solar absorption chillers for both cooling and heating applications. They have combined the solar water heater collector and absorption chillers and have done analysis for a variety of configuration to reach the highest outcome. According to the study done by Liu et al. [24] and Ayadi et al. [25], multi-effect chillers do not require as much solar thermal energy as single-effect chillers, while they demand higher driving temperature. Among the solar hot water collectors, flat plate and evacuated tube collectors have higher solar gain per unit area compared to concentrating collectors due to limited direct normal irradiance (DNI) [26]. Shirazi et al. [23] have considered parabolic trough, linear Fresnel concentrating, and evacuated flat plate collectors as part of different configuration for their research. Additionally, three types of LiBr-Water absorption chillers, single-effect, double-effect, and triple-effect, have been modeled for the study. “A parametric study of the proposed configurations is performed to investigate the effect of key design variables such as the storage tank volume and the solar collector area on the plants’ thermal behavior” [23]. The heat loss through the storage tank during winter would impact the thermal performance of the whole system. In addition, the influence of DNI fraction on collector area has been evaluated [23]. Figure 3-5 demonstrates the general layout of a solar heating and cooling absorption chiller.

The model was simulated with TRNSYS for the proposed configurations [23]. Modeling the cooling unit included using the chilled water set-point temperature and the cooling load of the building, while hot water set-point temperature and heating load was used for modeling the heating unit [23]. Recycle fraction of hot water return was evaluated to determine the temperature for the chiller [23]. More importantly, different conditions for storage tank was studied according to the flows and temperature of water entering and exiting the tank [23]. Ultimately, solar area was modeled based on collector flow loop and pump mass flow rate [23]. Location of the simulation was set to Sydney, Australia [23].

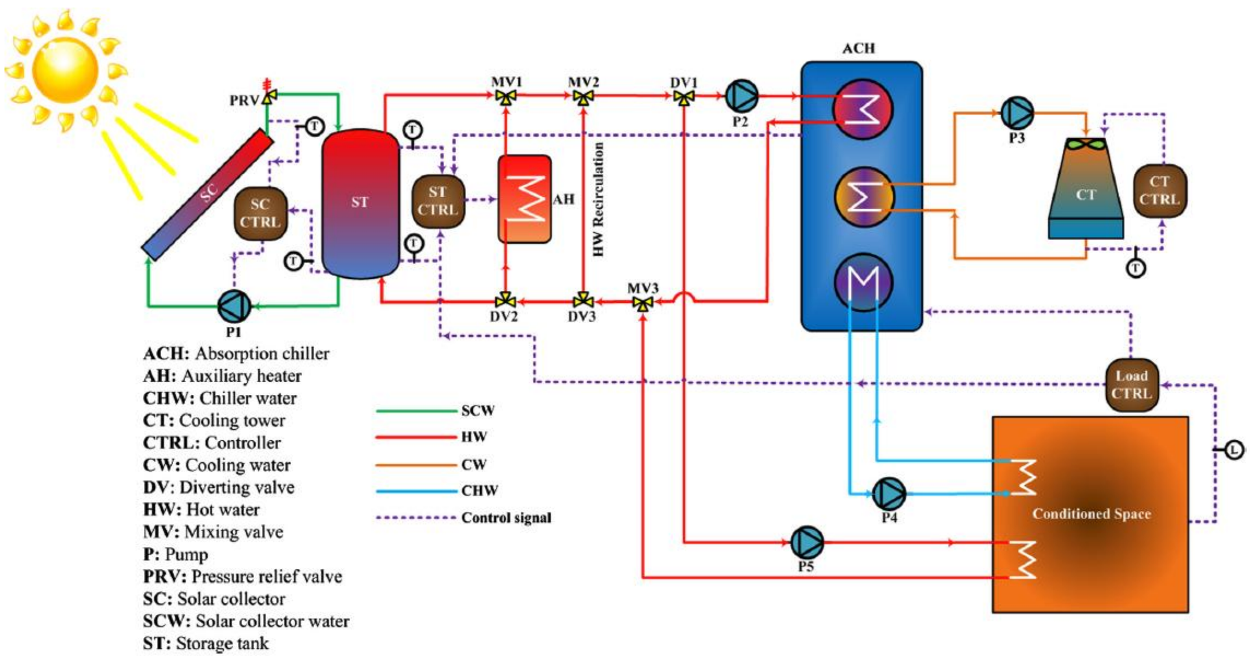


Figure 3- 7: Schematic diagram of a solar heating and cooling absorption chiller plant [23]

COP of the absorption chiller was calculated based on Eq. (11):

$$COP = \frac{\dot{Q}_E}{\dot{Q}_G + \dot{Q}_{aux}} \quad (11)$$

Where;

$$\dot{Q}_E = \dot{m}_{CHW} C_{p,w} \left(T_{CHW,in} - T_{CHW,out}\right), \quad (12)$$

$$\dot{Q}_G = \dot{m}_{HW} C_{p,w} \left(T_{HW,in} - T_{HW,out}\right), \quad (13)$$

$$\dot{Q}_{AC} = \dot{m}_{CW} C_{p,w} \left(T_{CW,out} - T_{CW,in}\right), \quad (14)$$

$$\dot{Q}_{AC} = \dot{Q}_E + \dot{Q}_G + \dot{Q}_{aux}, \quad (15)$$

The model was verified against manufacturer's test reports for each key component [23]. It was noted that the variation between the simulation results and manufacturer's data were less than 2%, which emphasised on the accuracy of the model [23]. Figure 3-6 illustrates performance characteristics of the considered solar collectors.

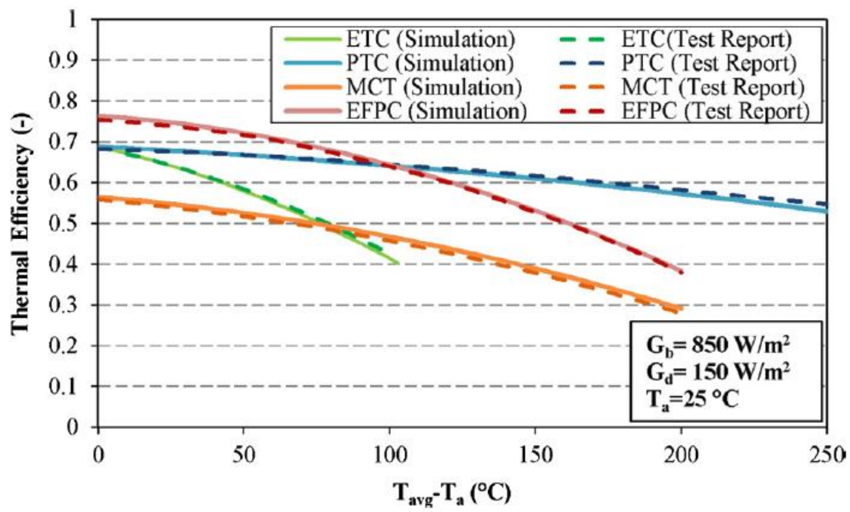


Figure 3- 8: Performance characteristics of the considered solar collectors [23]

The well insulated storage tank was calculated to have the overall U-value of 0.83 W/m^{2}K [27]. The influence of storage tank heat loss was investigated with a range of U-values from 0.8 to 11 W/m^{2}K [23]. Therefore, the plant solar fraction was reduced significantly by increasing the overall U-value of the storage tank [23]. Hence, "a large portion of the solar yield can be wasted through tank heat losses if a poorly insulated tank is used" [23]. Average storage tank volume of collector area was concluded to be 40 L/m^2 for all configurations [23]. Additionally, it was noted that the annual efficiency of the collector was improved by increasing the size of the tank [23]. As for the COP of the chillers, the double-effect chiller in locations with low and medium DNI could reach to 1.3, but for locations with higher DNI, COP could increase to 1.6 [23].

This study undertook the analysis of solar absorption chillers for both heating and cooling purposes. The building type and thermal loads were not specified, which would impact analyzing the results and arrangements for different configurations. Furthermore, the impact of dissipated heat from indoor spaces, which would impact the efficiency and performance of the system, was not determined. Ultimately, the influence of solar collector efficiency on COP of the chillers would be critical.

Another research towards solar absorption chillers has been done by Sioud et al. [28] on a 60kW LiBr-Water absorption chiller and 164 m^2 linear Fresnel collector. Figure 3-7 demonstrates the combination of linear Fresnel collector and single-effect chiller. An ejector has been used in the model, which "uses the vapor produced in the generator to entrain and compress the refrigerant vapor coming from evaporator before flowing in the condenser" [28]. Temperature of chilled water was set to be in the range of 4ºC and 12 ºC, which required higher driving source temperatures in the range of 180 ºC and 210 ºC [28]. Additionally, the optical efficiency of the linear Fresnel collector was between 0.635, under ideal conditions and, 0.663 [28].

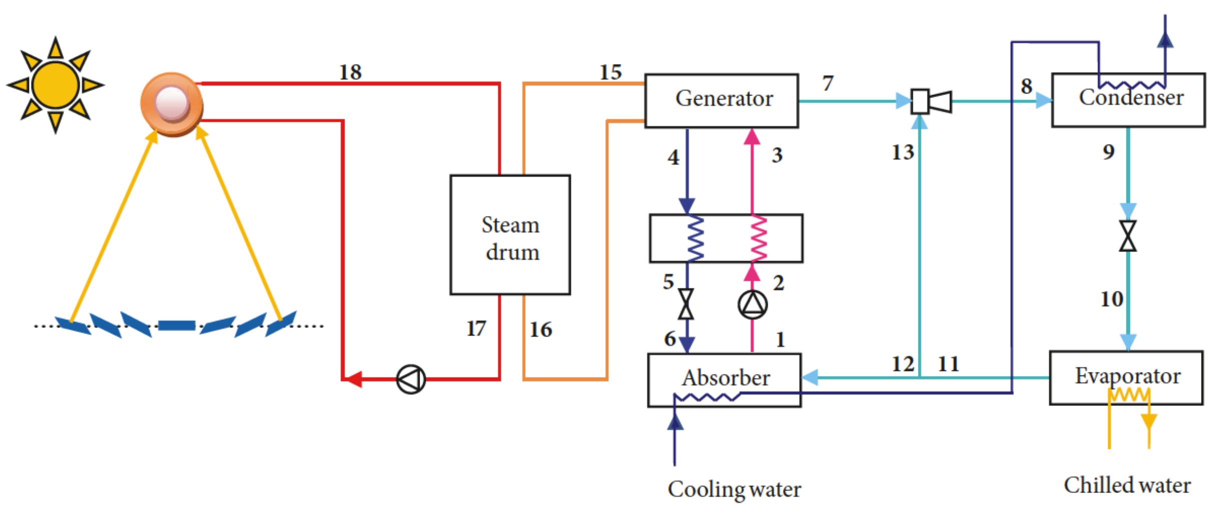
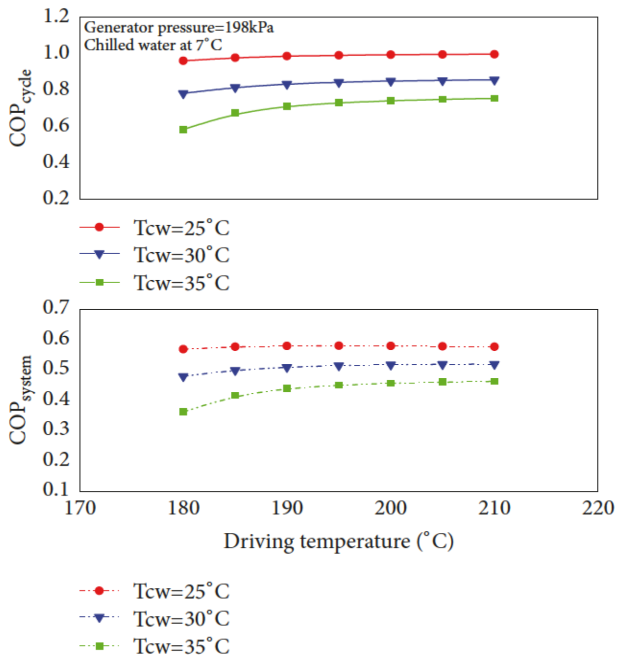


Figure 3- 9: Solar combined ejector single-effect chiller [29]

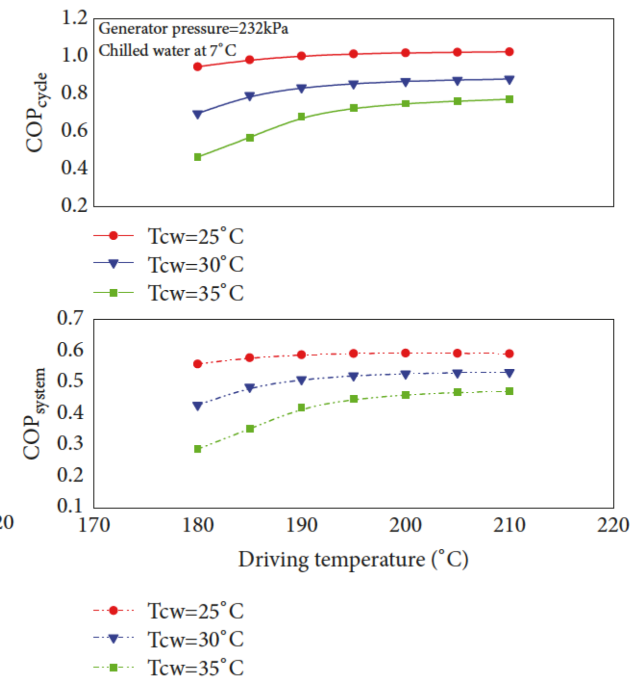
The mathematical model for a steady state process was designed based on mass and energy balances for both absorption chiller and ejector [28]. The results from simulation was validated with the results of a research done by Aphornratana et al. [29] and was noted that the deviations were less than 1% [28]. Moreover, solar combined ejector absorption chiller was modeled to analyze the COP of the overall system [28]. The overall COP of the system (COP_{system}) was achieved by considering the optical efficiency of the solar collectors (η) and COP of the absorption chiller (COP_{cycle}). Therefore, the COP of the entire system is expressed in Eq. (16):

$$COP_{system} = \eta COP_{cycle} \tag{16}$$

The results of the simulation indicated that the COP_{system} was improved with increasing temperature of the generator [28]. Additionally, COP_{system} was increased with reduction of cooling water temperature, while the temperature of the generator was fixed [28]. Ultimately, the maximum COP for both the system and absorption chiller was achieved under high pressure in the generator [28]. Figure 3-8 illustrates the effect of generator temperature on system performance for different pressures in generator and cold-water temperatures for constant chilled water temperature.



(a)



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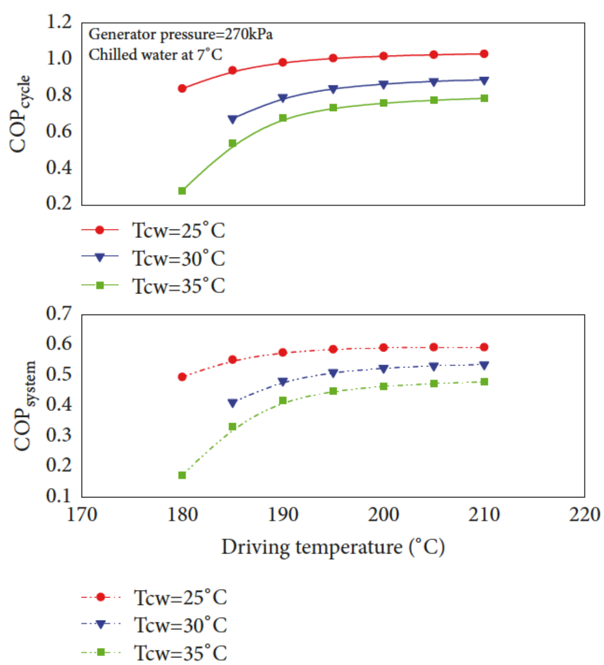


(c)

Figure 3- 10: Effect of generator temperature on system performance for differnt generator pressures and cooling system medium temperatures [28]

The paper analyzed the solar combined ejector absorption chiller in Tunisia, which has the latitude of 33ºN. Analyzing the COP of the system by considering the COP of the absorption chiller and optical efficiency of the collector, provided a significant factor for analyzing the effectiveness of solar absorption chillers. However, the connection between the absorption chiller and the cooling system inside the building was not specified as well as the type of the building and its required cooling load.

4. Case Study

4.1 Earth Rangers Centre (ERC)

The scope of this research is to determine a methodology to analyze the waste heat recovery of a data centre using absorption chiller in Ontario's climatic condition. The selected data centre is located in an institutional building called Earth Rangers Centre for sustainable technology located at 9520 Pine Valley Drive, Woodbridge, Ontario. The building has LEED Platinum certification, although the heat from data centre is not being recovered [30]. The building contains renewable technologies such as solar generation, innovative water and waste water management, green roofs, geothermal heating and cooling, and smart automation and control [31]. The organization aims to educate and inspire people, especially children to protect biodiversity and introduce new environmental activists. "The Centre was designed to embody Earth Rangers' values, showing that we practice what we preach, and inspiring everyone who walks through our doors" [32].

Figure 4- 1: View of the Earth Rangers Centre for sustainable technology **[30]**

4.2 Objective of Study

The literature review conducted for this study determined limitations to current approaches for waste heat recovery on data centres using absorption chillers. This study aims to investigate the potential use of absorption chiller in conjunction with the rest of Earth Rangers Centre's HVAC system only for data centre. In other words, the dissipated heat from existing equipment in the data centre would be the added heat to the generator to supply chilled water to the CRAC. Data collected through previous research on ERC for developing a valid energy model was used for simulating absorption chiller. It should be mentioned that the building has heat recovery system to meet its demands. Although, the wasted heat from data centre is not recovered. Therefore, this study would enable the building to enhance its energy performance by recovering the wasted heat from data centre, while it is considered low-quality heat.

Figure 4- 2: Earth Rangers' Data Centre panoramic view **[31]**

4.3 Earth Ranger Centre; Data Centre

Data centre inside the Earth Ranger Centre is a room located at the centre of the building as shown in figure 4-3. It includes computer servers, hard drives for data storage and back-up purposes, networking infrastructure, security, and cooling system [31]. Additionally, the data centre is involved with email systems, web pages, phone and other modern communication infrastructures, as well as building automation system data processing [31]. The energy consumption of data centre is 12% of total energy consumed in the building, while it is only 16 m^2 [31].

Figure 4- 3: Ground Floor Plan – ERC **[30]**

4.3.1 Computer Room Air Conditioning (CRAC)

The existing cooling system includes a chilled water In-Row Cooling (RC) unit, and the CRAC unit is only placed for back-up. The floor is raised by 2 ft with perforated tiles, which is used while the back-up is operating [31]. The orientation of the data centre is according to the best practice of hot and cold aisles arrangements [31]. Appendix A-7 is the summary of In-Row RC status on Jan 30th, 2019 [31].

4.3.2 Information Technology (IT) System

The relevant description and quantity of every piece are listed in Appendix A-9 [31]. The information given in that table is related to the main rack inside the data centre, which is located to be at the closest distance to the cooling system, and it contains server blades, storage arrays and Networking [31]. Other rack's components are listed in Appendix A-10. Ultimately, the total dissipated heat is broken down in Table 4-1, which refers to the 100% load of servers [31].

Table 4- 1: Racks heat dissipation breakdown **[31]**

Data Centre's Equipment	**No.**	**Heat dissipation (W) – each**	**Heat dissipation (W) - Total**
Pillar Axiom Pilot	8	109	872
Pillar Axiom Slammer	1	592	592
Cisco UCS 5108 Blade Server Chassis PSU	1	2498	2498
Cisco UCS 6248UP PSU	2	732	1464
Cisco S170	1	127	127
Cisco 2911	1	210	210
Miscellaneous	1	2930	2930
		TOTAL	**8693 (W)**

5. Methodology

The research problem entails developing an understanding of waste heat recovery for a data centre using absorption chiller. This can be achieved through the methods used for analyzing the overall COP of the system. According to the reviewed research papers, it can be said that the waste heat recovery from data centres requires field testing, and simulations. The in-situ measurements would help in determining the amount of dissipated heat available in the data centre. Additionally, simulations would help in demonstrating the performance of the system based on the available dissipated heat. Therefore, it would show the energy balance between the thermal energy dissipated from the data centre and cooling capacity of the absorption chiller.

Therefore, the key components of the research methodology are as follows:

- Conducting on-site measurements for simulating the current condition of the data centre with the existing CRAC system, which was done previously by Arbabi [31]
- Simulating the energy performance of the absorption chiller

5.1 System Simulation and Modeling

Simulation proposed for this research includes a mathematical modeling using Simulink/MATLAB to demonstrate the performance of the absorption chillers based on first and second laws of thermodynamics. The input data for the model will be based on previous work done by Arbabi [31]. The variables are divided into independent and dependent variables and are as follows:

A. Independent Variables

1. Indoor air temperature
2. Temperature of the absorber, generator, condenser, and evaporator

3. Mass flow rate of the absorbed refrigerant
4. Efficiency of the CRAC
5. Temperature of the chilled water return from the CRAC
6. Heat transfer coefficient of the solution heat exchanger

B. Dependent Variables

1. Heat transfer rate
2. COP of the absorption chiller

5.2 Mathematical model for simulation

The single effect LiBr-Water absorption chiller was modeled according to Figure 5-1 and following equations. The required input data to the model are temperature of the absorber, generator, condenser, and evaporator. Additionally, the effectiveness (ε) of the solution heat exchanger (SHX) is required for the simulation. The simulation was run with the given dissipated heat from the data centre, and the heat input to the evaporator is compared with cooling load from the simulation results done by Arbabi. Therefore, it would determine whether the dissipated heat is enough to meet the cooling load. Additionally, it would demonstrate whether another source of energy is needed to be added to the system.

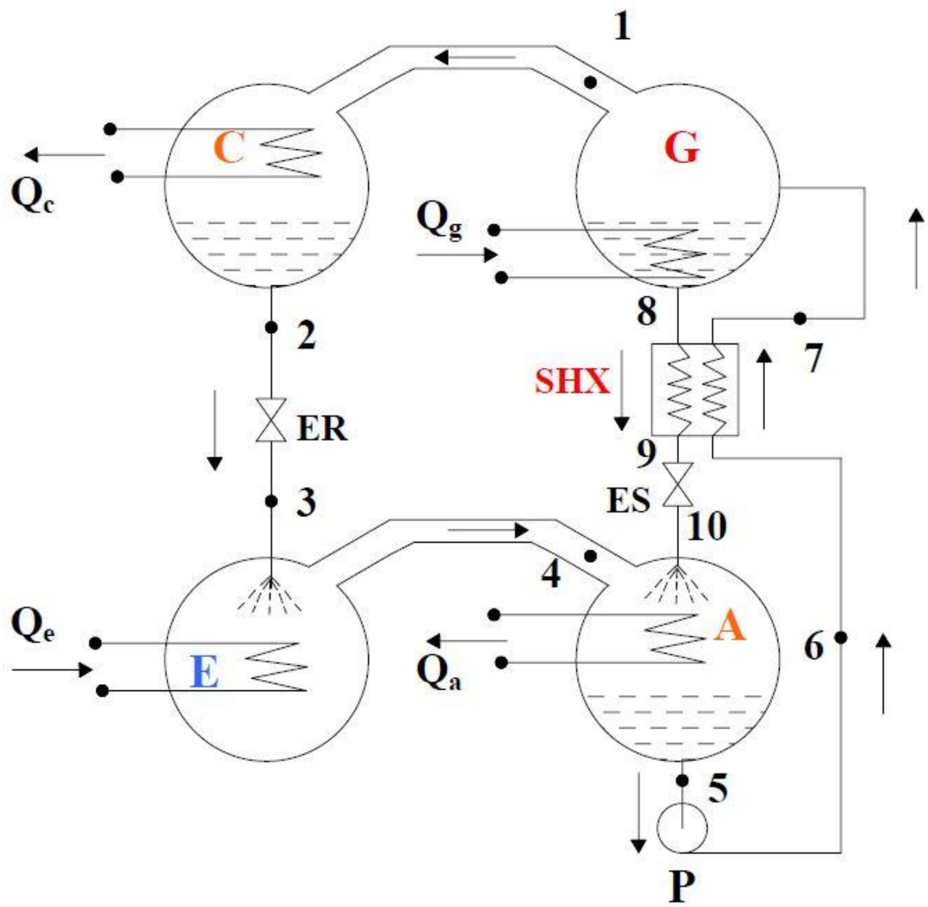


Figure 5- 1: Schematic of a LiBr-Water Absorption Chiller [33]

LiBr Concentration in strong and week solutions can be determined by Eq. (17) and E. (18) [3].

$$X_{strong} = \frac{49.04 + 1.125T_5 - T_4}{134.65 + 0.47T_5} \tag{17}$$

$$X_{weak} = \frac{49.04 + 1.125T_8 - T_2}{134.65 + 0.47T_8} \tag{18}$$

Knowing the solution temperature and concentration, the enthalpy of solution at the outlet of the absorber (State 5) and generator (State 8) can be calculated through the standard formulation developed by Kaita [34].

$$H(\mathrm{T,X}) = (A_0 + A_1X)T + 0.5\,(B_0 + B_1X)T^2 + (D_0 + D_1X + D_2X^2 + D_3X^3) \tag{19}$$

Where H (kJ/kg) is the enthalpy of solution, T (ºC) is the temperature (20 ≤ T ≤ 210 ºC), X (wt %) is the absorption concentration (40 ≤ X ≤65 wt%), A_0 = 3.462023, A_1 = -2.679895*E*-2, B_0 = 1.3499*E*-3, B_1 = -6.55*E*-6, D_0 = 162.81, D_1 = -6.0418, D_2 = 4.5348*E*-3, and D_3 = 1.2053*E*-3.

The enthalpy of liquid water and water vapour at different temperatures and pressures can be obtained from Eq. (20) and (21) respectively, where T_{ref} , the reference temperature, is 0 ºC [33].

$$h_{W,liquid} = 4.19\,(T - T_{ref})kJ/kg \tag{20}$$

$$h_{W,sup} = 2501 + 1.88\,(T - T_{ref})kJ/kg \tag{21}$$

Mass flow rate of the refrigerant ($\dot{m}_r$ (kg/s)) can be defined based on Eq. (22) [3].

$$\dot{m}_r = \frac{\dot{Q}_{eva}}{h_4 - h_2} \tag{22}$$

Where $\dot{Q}_{eva}$(kW) is the heat transfer rate at the evaporator. Therefore, by assuming that the mass flow rate of LiBr in both weak and strong solution streams are the same ($\dot{m}_5X_{strong} = \dot{m}_{10}X_{weak}$), the weak and strong mass flow rate (kg/s) can be found through a mass balance at the absorber [3].

$$\dot{m}_{strong} = \frac{\dot{m}_r\,X_{strong}}{X_{strong} - X_{weak}} \tag{23}$$

$$\dot{m}_{weak} = \frac{\dot{m}_r\,X_{weak}}{X_{strong} - X_{weak}} \tag{24}$$

In order to calculate the temperature values of the strong solution entering the generator (state 7) and weak solution entering the absorber, the energy equation for the heat exchanger is used. Therefore, temperature of the solutions exiting the solution heat exchanger (State 7 and 9) can be obtained based on Eq. (25) and Eq. (26) respectively. It should be mentioned that the enthalpy of weak solution at the outlet of solution valve (State 10) is the same as the enthalpy of weak solution at the inlet of the valve (state 9) [3].

$$T_9 = T_8 - \varepsilon(T_8 - T_6) \tag{25}$$

$$T_7 = T_6 + \varepsilon(T_8 - T_2) \tag{26}$$

Knowing the enthalpy at all state points of the cycle, the heat transfer rate at absorber, generator and condenser can be obtained through the energy balance equations [33]. Heat input to the generator can be calculated through Eq. (29) and ultimately, the COP of the system can be found by Eq. (30).

$$\dot{Q}_{absorber} = \dot{m}_r h_4 + \dot{m}_{10} h_9 - \dot{m}_5 h_5 \tag{27}$$

$$\dot{Q}_{condenser} = \dot{m}_r (h_1 - h_2) \tag{28}$$

$$\dot{Q}_{generator} = \dot{m}_r (h_1 - h_7) + \dot{m}_{strong} (h_8 - h_7) \tag{29}$$

$$COP = \frac{\dot{Q}_{evaporator}}{\dot{Q}_{generator}} \tag{30}$$

6. Implementation of Model in Simulink

The equations brought previously were implemented in Simulink/MATLAB. Figure 6-1 illustrates the implementation of Eq. (17) and (18). It should be mentioned that the simulation is done for steady state. Additionally, the temperature difference inside the evaporator, condenser, absorber and generator are negligible. The temperature of the solution at state 5 (outlet of the absorber) is assumed to be at the absorber temperature. The refrigerant at state 4 (Inlet of the absorber) is assumed to be at evaporator's temperature. The refrigerant at state 1 and weak solution at state 8 are at the generator's temperature. The refrigerant and solution expansion valve are considered to be adiabatic. Therefore, based on the energy balance equation, enthalpy of refrigerant at state 2 and 3 are equal. Figure 6-2 shows the implementation of Eq. (19) for calculating the enthalpy of strong solution. Figure 6-3 entails the model for calculating the enthalpy of liquid water and water vapour for states 2 and 4.

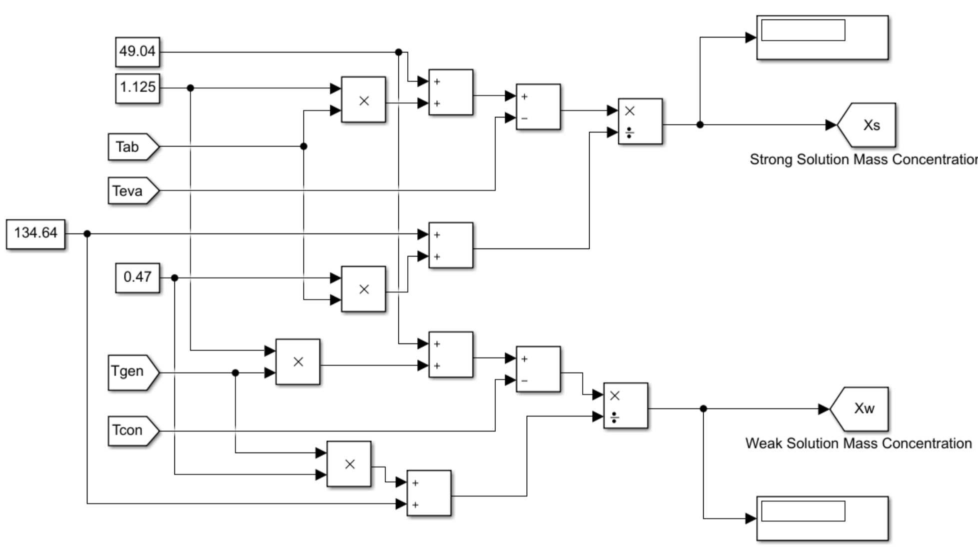


Figure 6- 1: Implementation of equations (17) and (18)

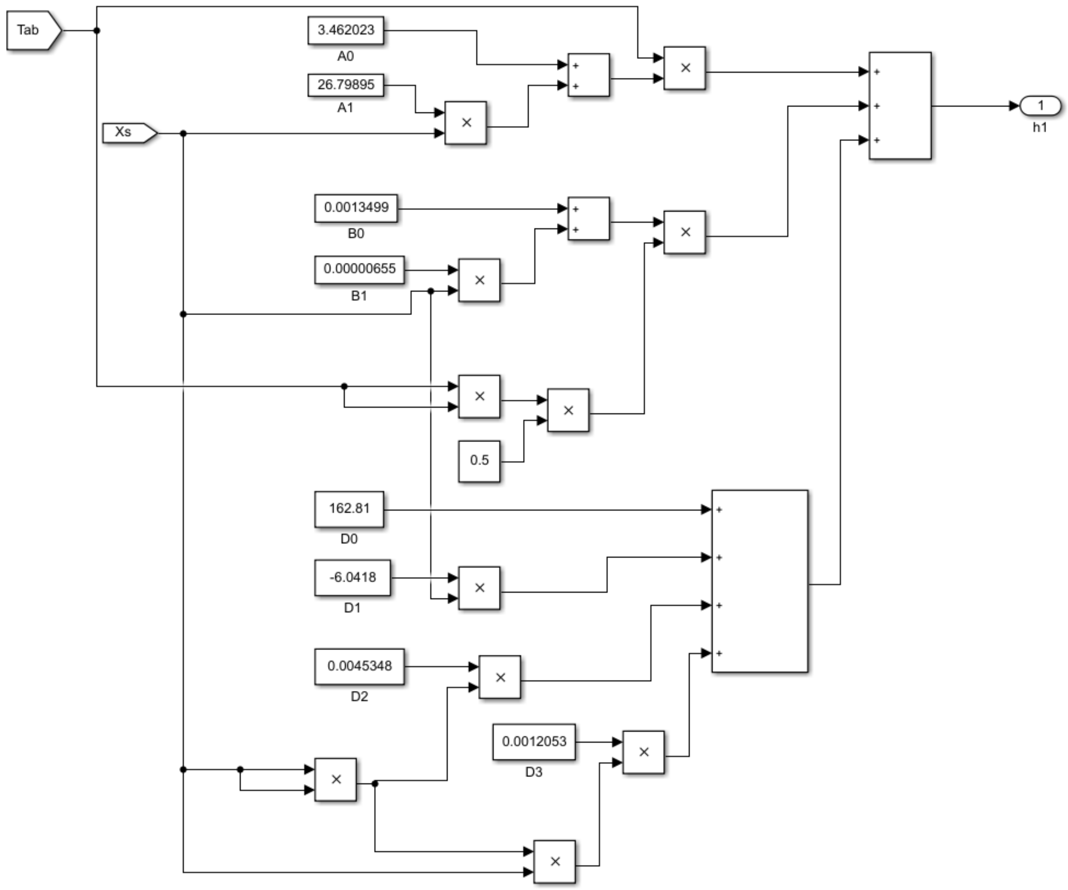
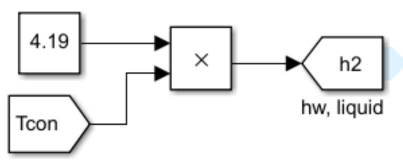


Figure 6- 2: Strong solution enthalpy calculation



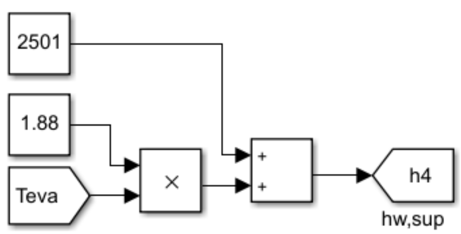


Figure 6- 3: Enthalpy of liquid water and water vapour for states 2 and 4

Figure 6-4 illustrates the calculation of refrigerant mass flow rate, which is used for obtaining mass flow rate of strong and weak solutions shown in Figure 6-5. Figure 6-6 is the implementation of Eq. (25) and (26) for the solution heat exchanger. Heat transfer rate through absorber and condenser are modeled based on Figure 6-7 and 6-8 respectively. Equation of heat input to the generator is demonstrated in Figure 6-9 and ultimately, the COP is shown in Figure 6-10.

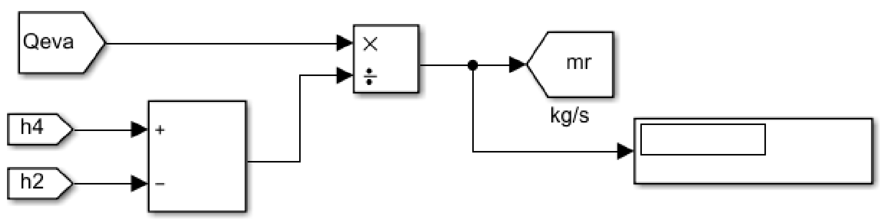


Figure 6- 4: Refrigerant mass flow rate demonstration

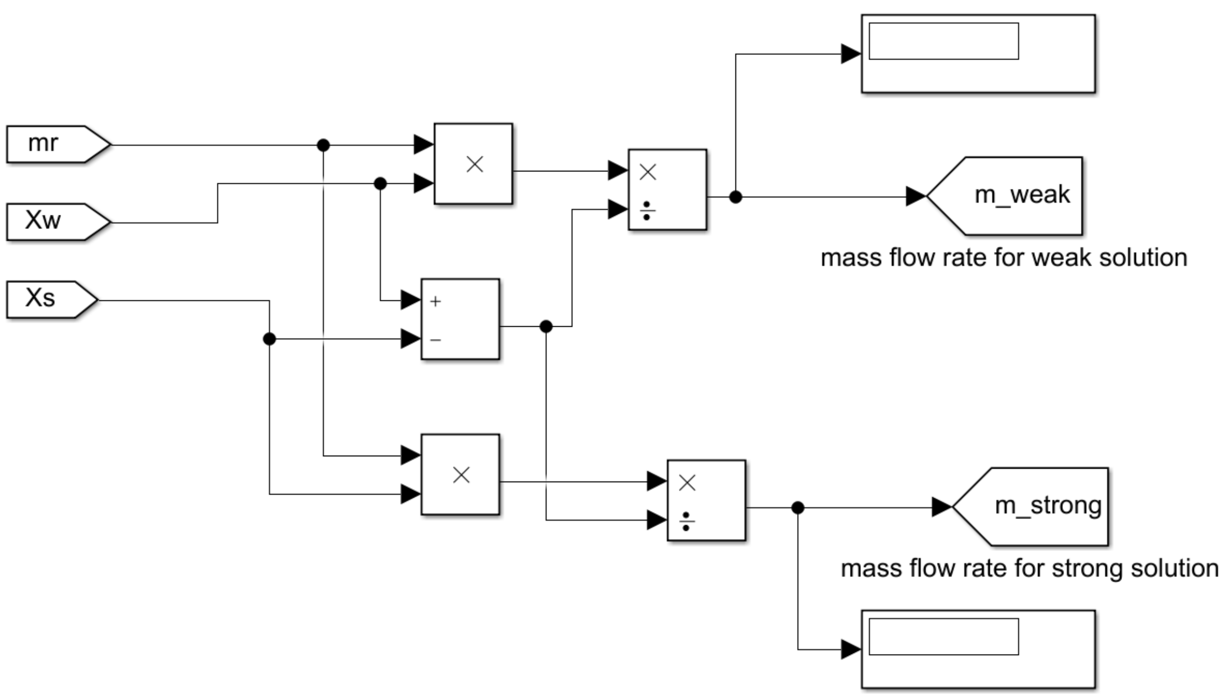


Figure 6- 5: Strong and weak solution mass flow rate

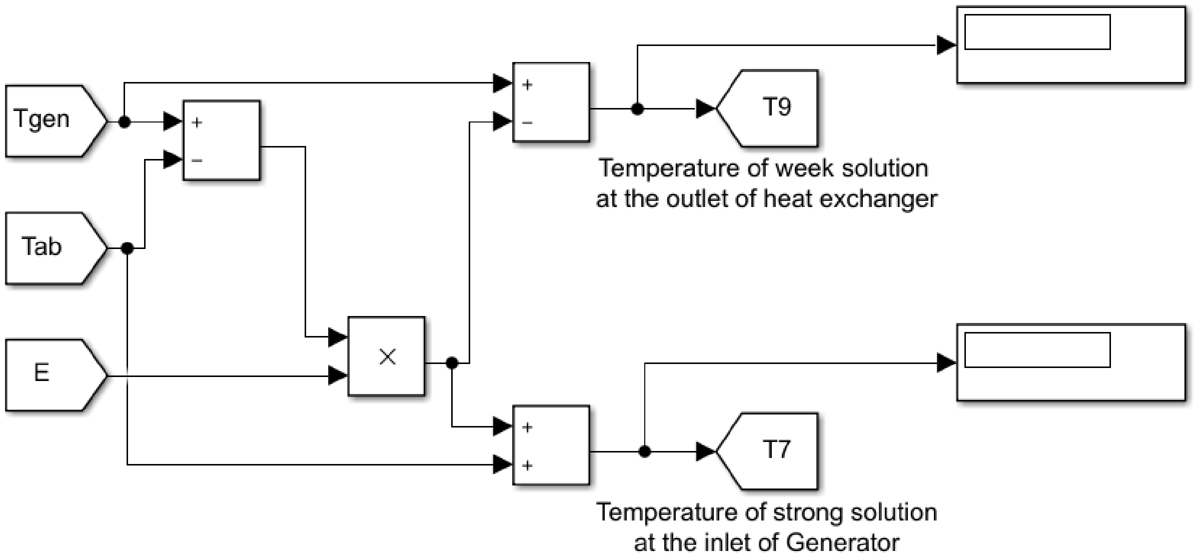


Figure 6- 6: Implementation of Eq. (25) and (26)

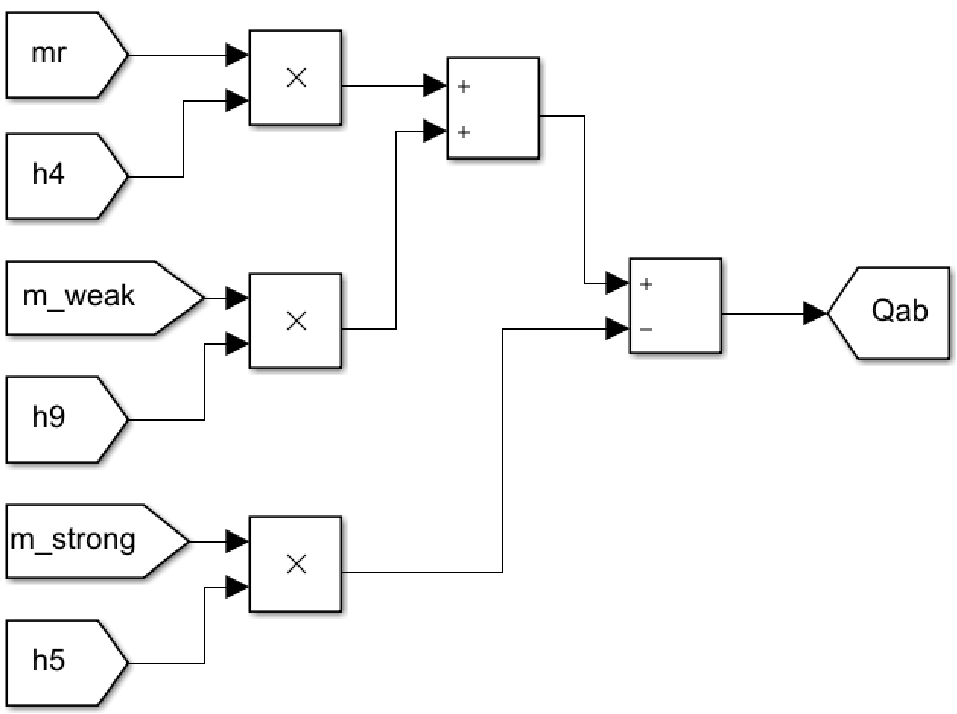


Figure 6- 7: Calculation of Q_absorber (Eq. (27))

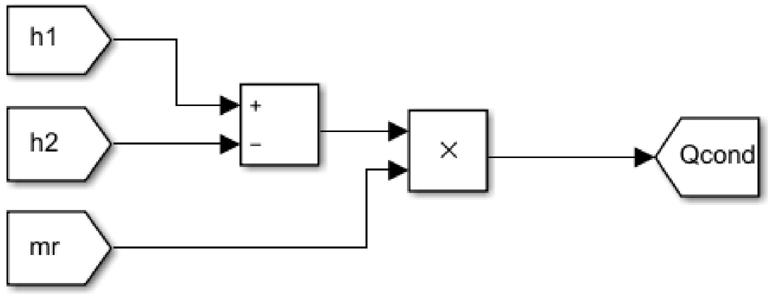


Figure 6- 8: Calculation of Q_condenser (Eq. (28))

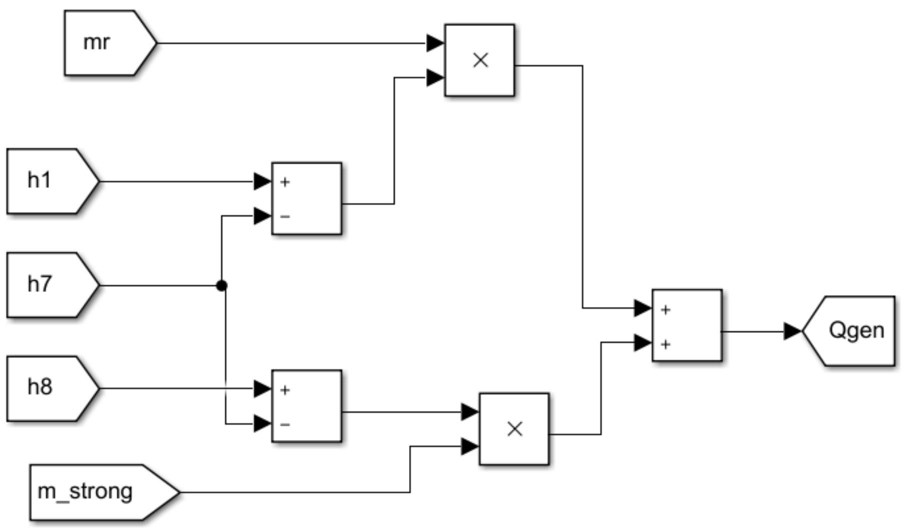


Figure 6- 9: Implementation of heat input to the generator

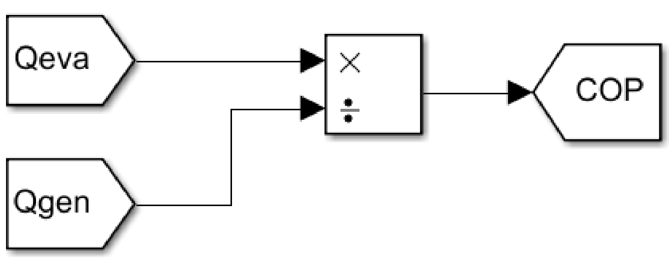


Figure 6- 10: Demonstration of Eq. (30) in the model

7. Simulation Results

The model was run based on the dissipated heat from the data centre, which was estimated to be 8.7 kW. This source of energy was assumed to be the input energy to the generator to analyse the cooling capacity of the evaporator as well as the COP of the system. Mass concentration of the strong and weak solutions is limited ($0.4 \leq X \leq 0.6$), and defined according to the temperature of the absorber and generator. Therefore, the model was run for different absorber and generator temperatures. Table 6-1 demonstrate the range of temperature differences for both absorber and generator, while the mass concentration of the solutions was in the acceptable range.

Table 6- 1:Input temperatures for absorber and generator

Temperature of the Absorber (ºC)

25	30	35	40	45

Temperature of the Generator (ºC)

65	67.5	70	72.5	75	77.5	80	82.5	85	87.5	90	92.5

Additionally, temperature of the evaporator and condenser and effectiveness of the solution heat exchanger were among the inputs. In order to be able to compare the results under the same conditions, these inputs were not changed during the simulation. These variables were given values according to the research done by Rubio-Maya et al. [35] listed in Table 6-2.

Table 6- 2: Input values for condenser and evaporator temperatures and solution heat exchanger effectives

Temperature of the Condenser (ºC)	39.8
Temperature of the Evaporator (ºC)	9
Solution Heat Exchanger (ε)	0.7

LiBr mass concentration in the weak solution rises with the increase in the temperature of the generator and is not dependent on the temperature of the absorber. On the other hand, LiBr mass concentration of the strong solution escalates with the increasing absorber temperature. These variations of the mass concentration of LiBr is shown in Figure 6-1 and 6-2. It can be said that the relation between LiBr-Water mass concentration of the solutions and the associated temperatures of the components are almost linear.

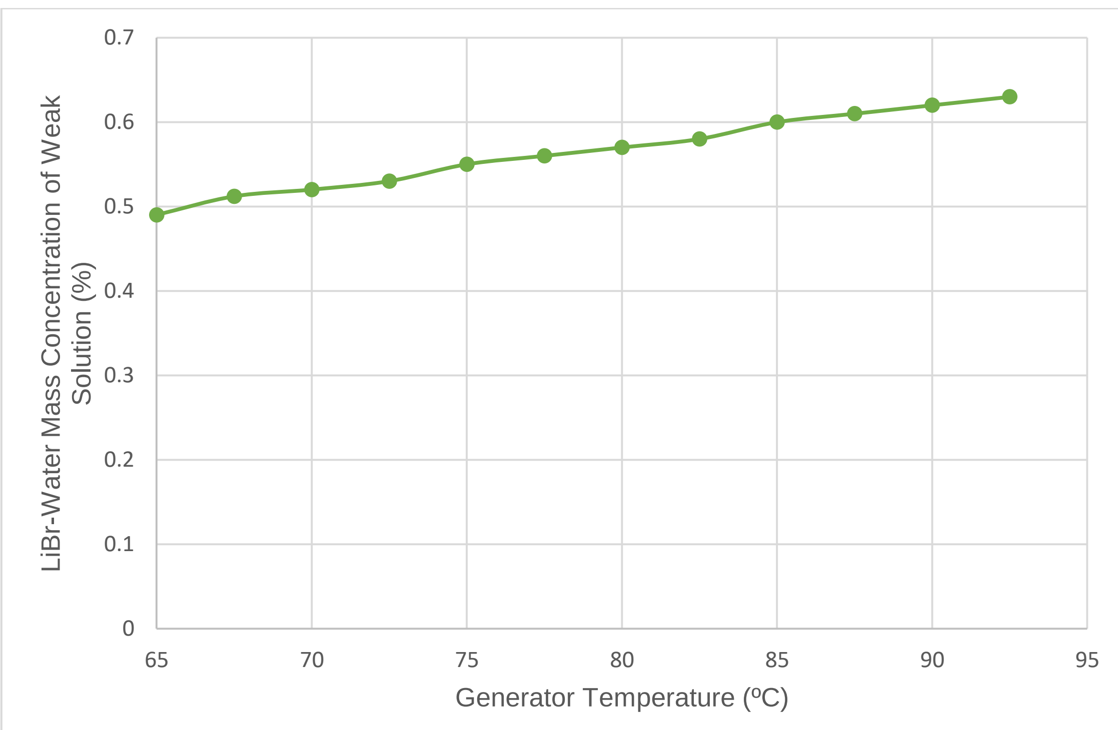


Figure 6- 11: Variation of mass concentration of weak solution with generator temperature

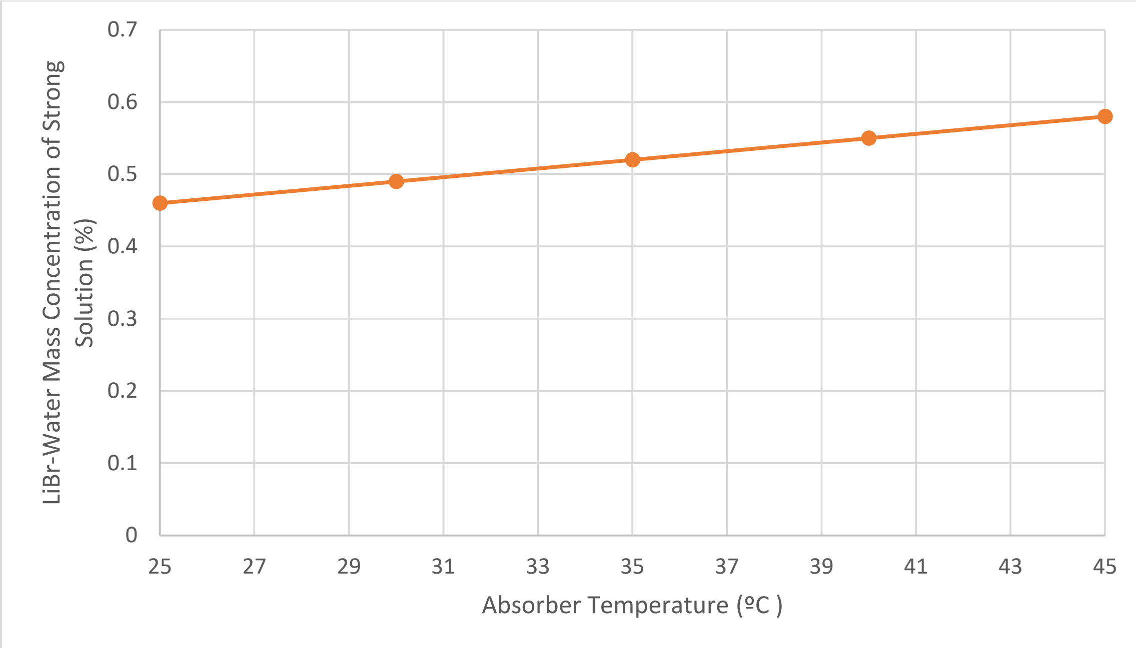


Figure 6- 12: Variation of mass concentration of strong solution with absorber temperature

Figure 6-3 illustrates a rapid decrease in the evaporator heat input for the absorber temperatures of 25ºC**,** 30ºC, 35ºC, 40ºC, and 45ºC, when the generator temperatures are below 67.5ºC**,** 70ºC, 75ºC, 80ºC, and 85ºC respectively. The heat input of the generator was set to be at 8.7 kW during the simulation since this value depends on the data centre heat rejection. It can be said that with the same heat input to the generator, COP of the system is at the highest of 0.79 with the lower absorber temperature of 25ºC and higher generator temperature of 92.5ºC. Figure 6-4 and 6-5 illustrate the influence of generator and absorber temperature on COP of the LiBr-Water absorption refrigeration. As the generator temperature increases (65 ºC – 92.5 ºC), the COP improves rapidly. In contrast, the system COP decreases with the increase in absorber temperature. Figure 6-6 illustrates the variation of COP with temperature of absorber and generator.

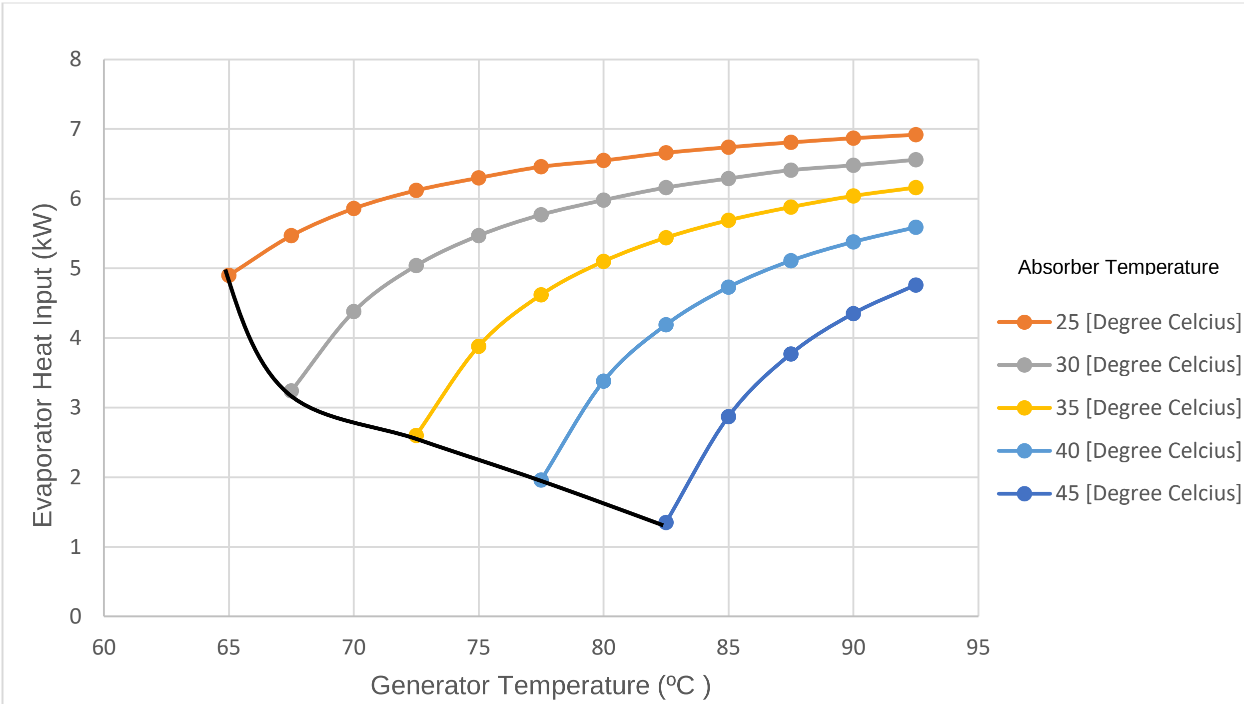


Figure 6- 13: Graph showing the variation of the heat input in the evaporator, with absorber temperature and generator temperature ($\boldsymbol{T_{con} = 39.8\,°C\,, T_{eva} = \ 9\,°C}$)

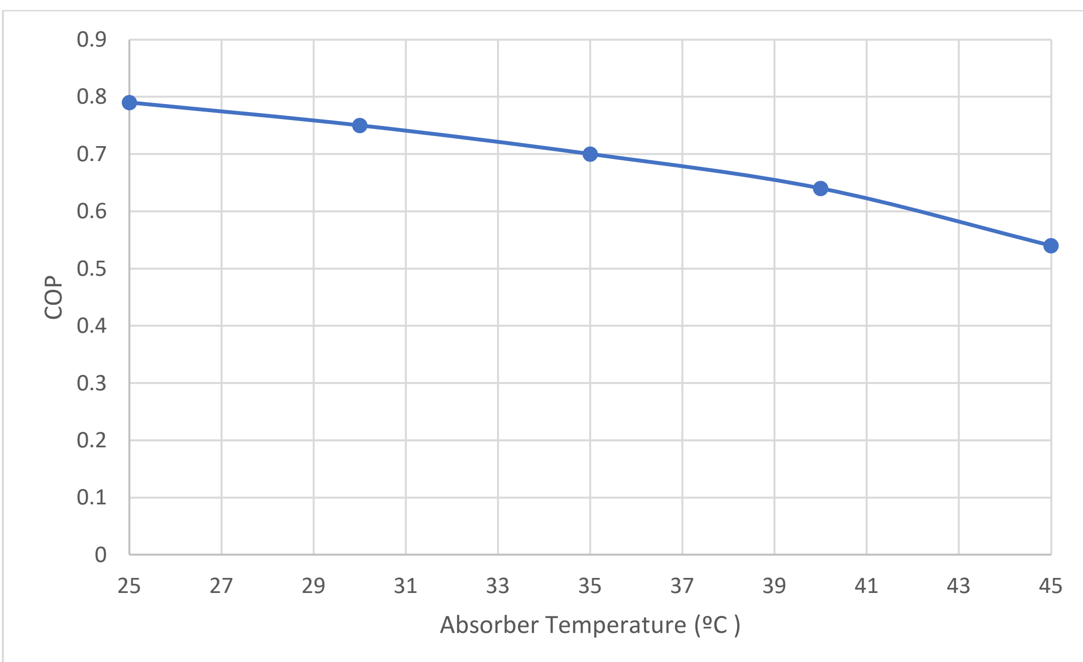


Figure 6- 14: The influence of absorber temperature on the absorption refrigeration system performance

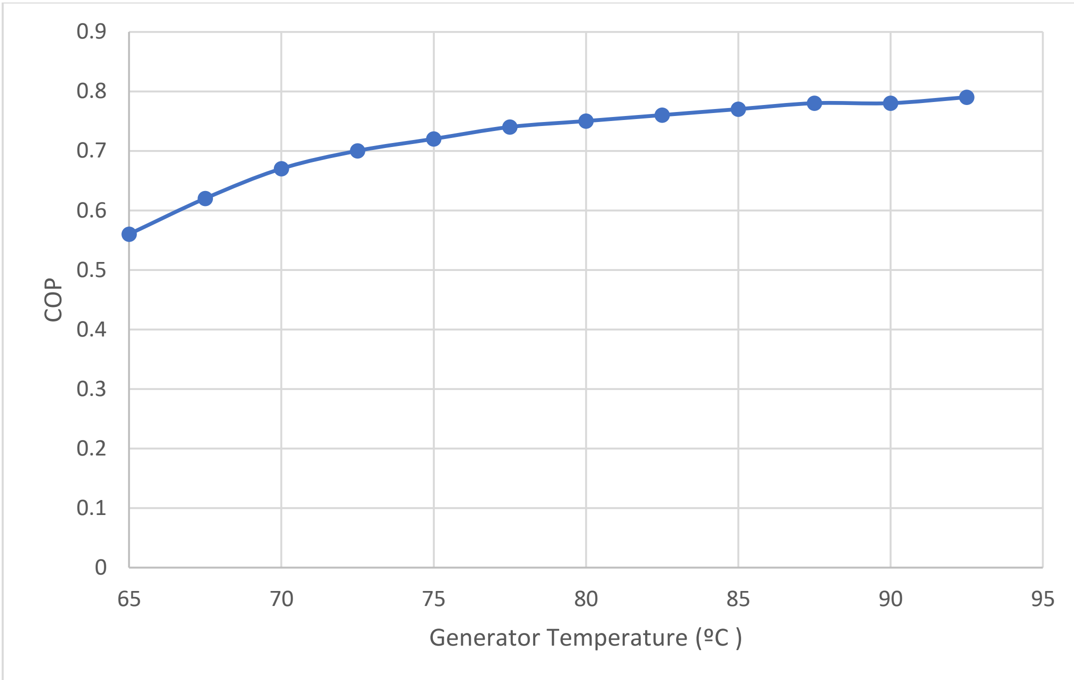


Figure 6- 15: Influence of generator temperature on the absorption refrigeration system performance

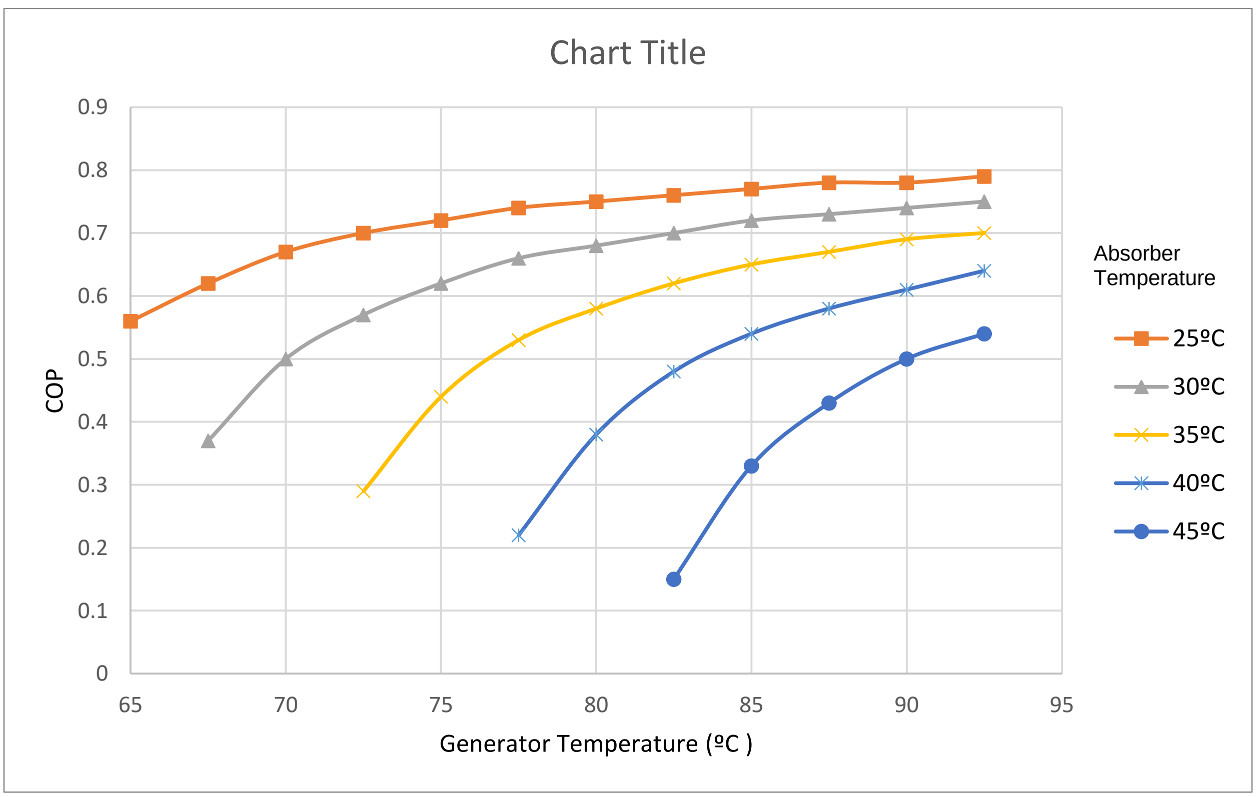


Figure 6- 16: Variation of COP with temperature of absorber and generator

8. Conclusion

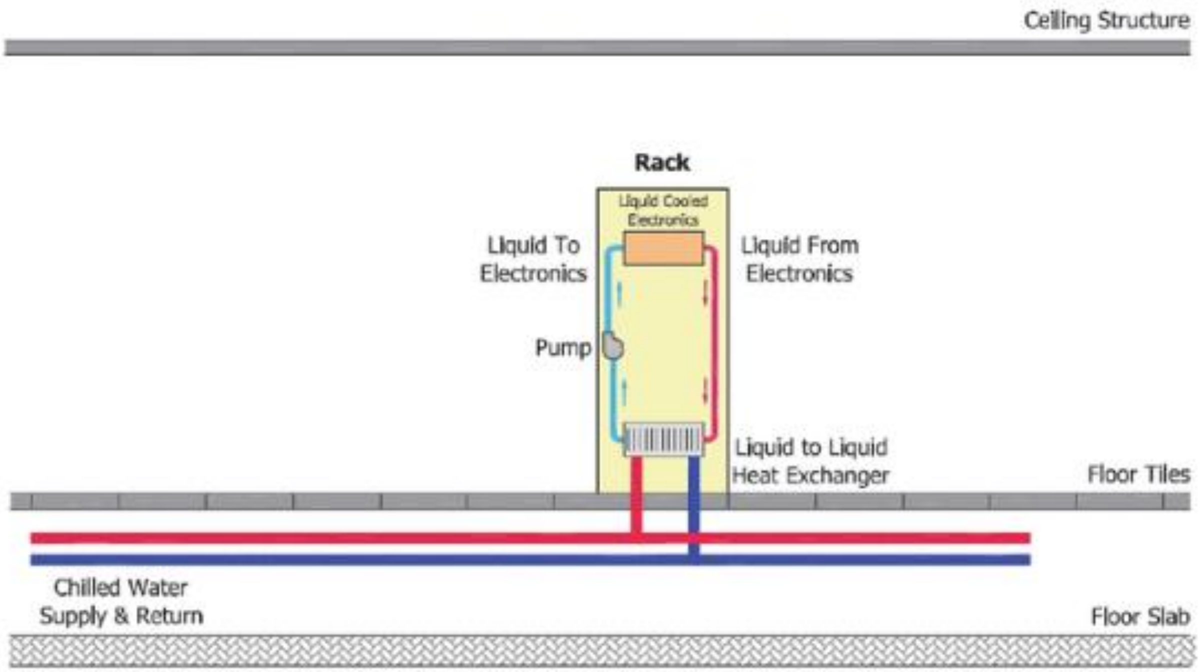
Data centres are one of the biggest contributors of energy consumption in the world, which 40% of this energy is used for cooling purposes. Therefore, reduction in cooling demand would make a significant impact on the energy savings in all industries. This research project aimed to focus on potential use of absorption chiller for heat recovery of a small-scale data centre in Woodbridge, Ontario. Therefore, an absorption chiller was modeled using Simulink/MATLAB to determine the coefficient of performance of the system and analyze the ability of the system in producing enough chilled water. According to the research done by Arbabi, the dissipated heat from the data centre is 8.7 kW based on the energy model of the data centre and data collection she made during her research. This energy was used as the heat input to the generator. The mathematical model for a single stage LiBr-Water absorption refrigeration system has been developed. It has been shown that the generator and absorber temperatures have significant influence on the COP of the system. The result shows that an increase in the generator temperature or decrease in the absorber temperature increases the COP. Due to the fact that the LiBr-Water absorption chiller is influenced by many parameters, and that it is difficult to take every factor into account, the condenser and evaporator temperatures, and the generator heat input were kept constant throughout this analysis. Since the energy input of the system is low quality and is not enough to meet the cooling load (10.6 kW [31]) inside the data centre, it is suggested to use a secondary energy source to run the system more efficiently. The highest cooling capacity that the system could reach is when the absorber and generator temperatures are at 25 ^{o}C and 92.5 ^{o}C respectively. Therefore, the highest possible COP of the system is 0.79 if all the dissipated heat from the data centre is captured and transferred to the system. However, the generator temperature is generally low in data centres and the COP of the system would be lower than 0.79. Therefore, the COP of 0.79 and higher would be ideal for such data centres, while the COP in reality would be lower than 0.79. The selected data centre is located in a building with LEED certification and it has solar panels for energy production. The absorption chiller can be connected to a solar hot water heater in order to achieve the required heating energy to provide enough cooling for the data centre. This can be the focus of future studies.

7.0 Future Work

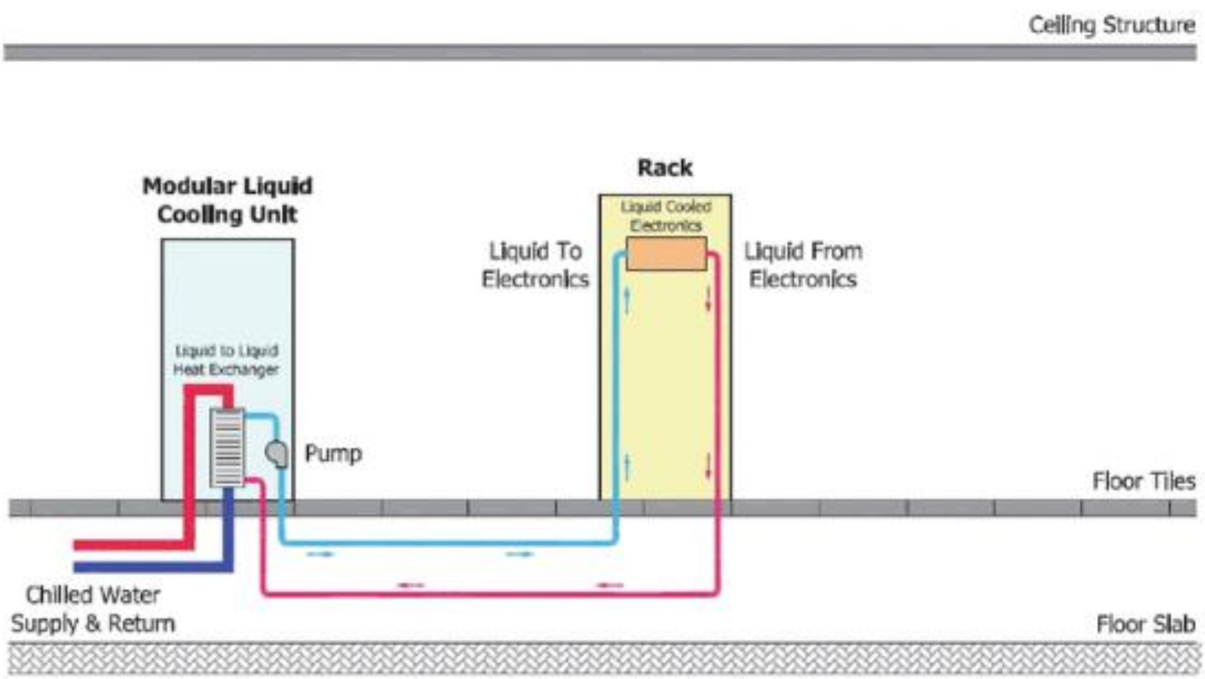
The results of this research lay the foundation for future related research projects. The research at this stage can be the starting point for the following topics:

- Investigating the potential use of solar absorption chillers for waste heat recovery in the Earth Rangers Centre
- Investigating the potential use of absorption chillers for waste heat recovery of large-scale data centres
- Identifying renewable energy sources as the secondary energy input to the generator in data centres

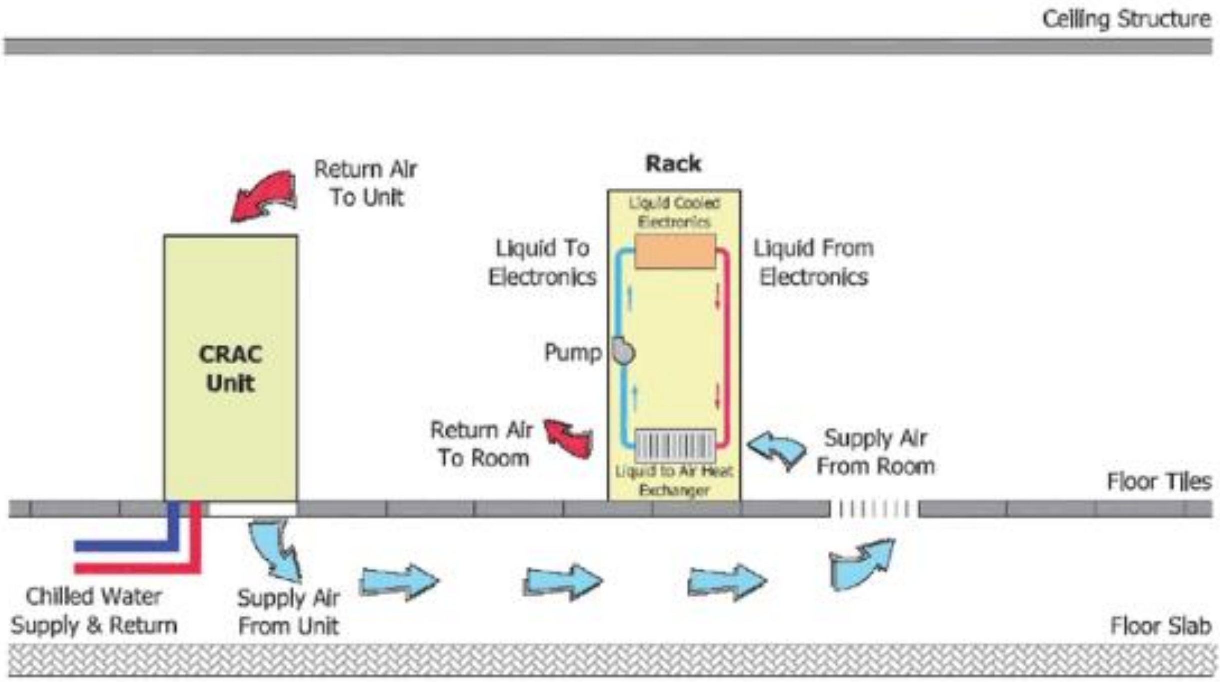
Appendix A



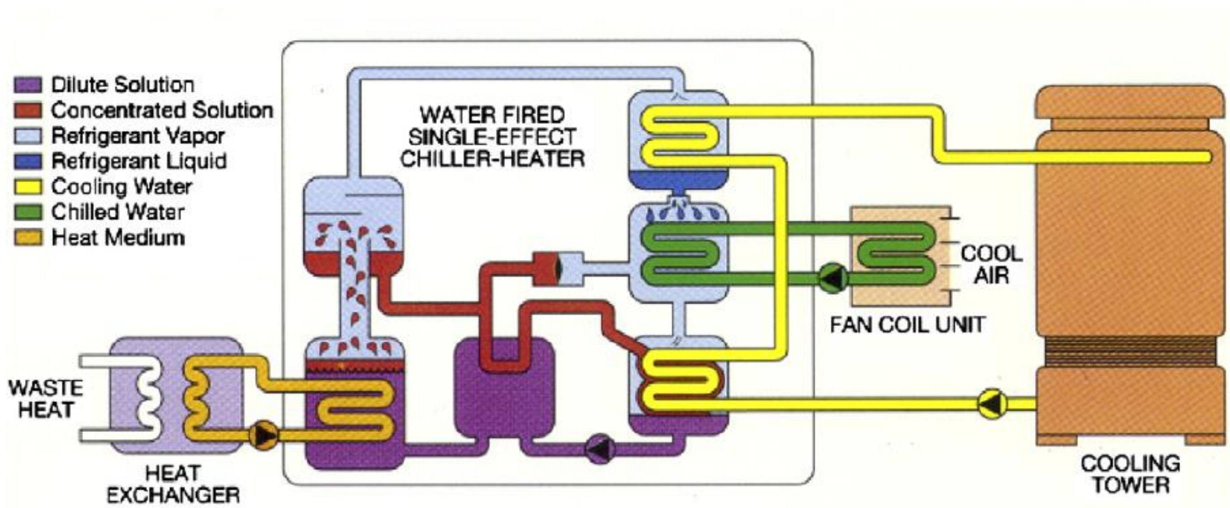
Appendix A- 1: Internal liquid-cooling loop exchanging heat with liquid-cooling loop external to racks [14]



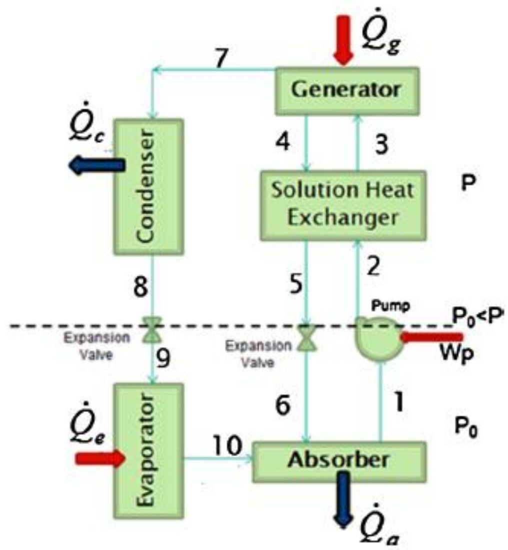
Appendix A- 2: Internal liquid-cooling loop extended to liquid-cooled external modular cooling unit [14]



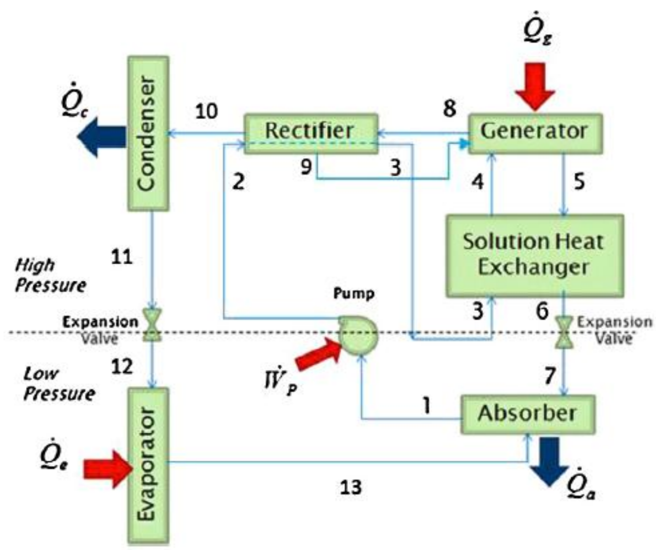
Appendix A- 3: Internal liquid-cooling loop restricted within rack [14]



Appendix A- 4: Waste heat energized cooling and heating (cooling operation) [16]



Appendix A- 5: Schematic view of a LiBr-Water AR system [3]



Appendix A- 6: Schematic view of a Water-Ammonia AR system [3]

Appendix A- 7: In-Row RC status on Jan 30th, 2019 **[31]**

Cool Set point	18 ºC
Rack Inlet Temperature	23.9 ºC
Supply Air Temperature	17.7 ºC
Return Air Temperature	23.2 ºC
Airflow	2265 CFM
Fan speed	72%
Cool Output	7.6 kW
Cool Demand	10.6 kW
Unit Energy	9041 kWh
Unit Power	379 W
Chilled Water Flow	40 L/m
Entering Chilled Water Temperature	12.6 ºC
Leaving Chilled Water Temperature	15.4 ºC
Filter Differential Pressure	12.45 Pa

Appendix A- 8:In-Row RC - ARC100 - 50/60 Hz - 100 - 120v **[31]**

Appendix A- 9:Earth Rangers' rack No. 1 components **[31]**

DC's main Rack		Quantity	Description
Storage	Pillar Axiom Pilot	8	An Application-Aware storage™ system. This feature enables dynamic adjustment in performance of the storage system relative to the requirement of applications. Storage utilization is expected to enlarge due to performance prediction ability. An external interface to the host storage network is provided by the slammer. Pilar axiom system supports one to four slammers.
	Pillar Axiom Slammer	1	
Server	Cisco UCS 5108 Blade Server Chassis PSU	1	6 Rack Unit (6U) Eight half-width slots, four full-width slots Operating Temperature 10°C to 35°C (50°F to 95°F) Max. output power per power supply – 2500W (up to 4 supplies) Max. heat output – 8530 BTU Efficiency - 88% at 10-20% & 100% Load, 92% at 50% Load
Network	Cisco UCS 6248UP PSU	2	one-rack-unit (1RU) 10-Gigabit Ethernet 48-port Fabric Interconnect (consumes less power per port than traditional systems) - Cisco Unified Computing System (UCS), provides network connectivity and management capabilities for the system. - Provided management and communication backbone for the servers. Minimum Output Power – 750W Heat dissipation 2497 BTU/hr (732 W) Efficiency 88 to 92% (50 to 100% load)

Appendix A- 10: Earth Rangers' other racks' active components

DC's Rack (Other)	Quantity	Description
Cisco S170 Web Security Appliance	1	Organizer and traffic control Rack height: 1RU Temperature allowance: 104 F (40 C)
Cisco 2911 Integrated Services Router (ISR)	1	Rack height: 2RU Delivering data, voice, video, and application service with a high security level
Miscellaneous	1	Including other servers, switches, power supplies, etc.

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