Adaptation of Solar Energy Driven Absorption Chillers for Air Conditioning in Commercial Building

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Master of Science Thesis
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Energy Technology EGI-2016-018MSC EKV1130
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Abstract

The most recent analysis of energy usage in the country revealed that nearly 50% of the power generation is used for air conditioning and mechanical ventilation most of which is used by commercial organizations. The grid generation mix that contains a high percentage of fossil fuel makes such energy usage environment unfriendly. Although absorption refrigeration is an old technique its economical application is limited to applications where cheap or waste heat energy is available despite decades of R&D, due to low COP, high initial cost and larger size. Heat input at Moderately high (over 120°C) temperature and need to release large amount of heat to the environment through liquid or air cooling makes absorption chiller less conducive in cooling. Yet, being a tropical country, Sri Lanka has a better potential in adopting solar driven absorption refrigeration, if the chillers are operated at low temperature heat input that also promotes efficiency in storage that is mandatory due to fluctuation of energy source, subject to economic feasibility.

The project aims designing and modeling of a solar power driven absorption chiller system that is adoptable to a selected medium size commercial organization. The proposed system uses heat energy around 100°C and reusing fraction of energy expelled to the environment by suitably modifying operating parameter and thereby increasing efficiency of the system. Reduction of such heat losses and reducing heat input is achieved with the use of secondary heat exchange (brine) system that optimizes the energy usage. This arrangement will make efficient usage of solar heat storage, even in the considerable absence of solar power. System modeling and simulation of both basic double effect chiller and its modified versions were carried out and compared to evaluate improvement. The simulation of the modified system was used to obtain working parameters of the chiller so that a suitable solar collector, chilled water and heat rejection systems can be designed. Operational conditions of the cooling system are measured by the state sensors that feed inputs to the control system to achieve the optimum efficiency and their technical details are also included in the report.
ACKNOWLEDGEMENT

The thesis project is a partial fulfillment of the three years of the Distant Master Programme in Sustainable Energy Engineering (DSEE) programme, where I benefitted from the guidance and support of many persons who made my academic career challenging and helped achieving the goals.

I greatly appreciate KTH, Sweden for offering me the opportunity to follow a postgraduate course on Sustainable Energy Engineering and for inculcating in me an energy consciousness, which now extends to almost all technical decisions I make as a professional. I am also thankful to ICBT campus for the role they played in making the programme accessible to employed students.

I greatly value the support and guidance of the KTH staff who kept us alive and academically involved right throughout during the academic programme. I have been fortunate to benefit from associations of my colleagues in this study programme whose names cannot be cited individually here due to space limitations; they were a great source of encouragement to me throughout the programme.

My deep gratitude goes to Dr. Primal Fernando, who encouraged us from the inception and provided us with theoretical and experiential guidance and to Ms. Shara Ousman for her coordination support through constant reminders about deadlines and schedules. I am also thankful for to Prof. Andrew Martin of KTH, who encouraged and motivated me to be active during the campus life, during the meeting at ICBT Campus.

The last phase of the programme, thesis research and report writing, would not have been possible unless Open University of Sri Lanka (OUSL) extended their timely support. I consider it a great fortune that Eng. P.D. Sarath Chandra, Senior Lecturer, agreed to be the supervisory of my research work. Without his valuable advice and guidance in this research work and report writing I would not have completed the thesis report. I sincerely thank him for spending his valuable time on my thesis and for his patience. I convey my sincere gratitude to Dr. Sad Jarall of KTH for playing the roles of a supervisor from KTH in this research for spending his time to communicate with me in giving invaluable guidance and also by providing literature that I consider as a kind gesture beyond his role.

I greatly appreciate Mr. Ruchira Abeyweera of OUSL for his role in coordinating our thesis work right from the proposal stage. He made a tremendous contribution providing many supports and directions to make my endeavour a success. I also convey my gratitude to Dr. Nihal Senanayaka and other staff members of Mechanical Engineering Department of OUSL, for participation in evaluation panels, for their advice during presentations and also providing modeling software programme.

Last but not least, I would like to convey my gratitude to my wife, Jani, for providing moral support and encouragement when I was under pressure due to the work of my academic endeavor. I appreciate support received from my daughter, Menusha, who is at the National University of Singapore. She shared her experience with me and provided valuable information as far as this research work is concerned.
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Abbreviations

ACs - Air Conditioners
AHU - Air Handling Unit
ASHRAE - American Society of Heating, Refrigerating and Air-Conditioning Engineers
CFC - Chlorofluorocarbon
COP - Coefficient of Performance
COPe - Coefficient of Performance with electrical input power
COPh - Coefficient of Performance with heat as input power
COPt - Coefficient of Performance with input power of all types
CSP - Concentrated Solar Power
DB - Dry Bulb Temperature
DC - Direct Current
DDC - Direct Digital Control
DNI - Direct Normal Irradiance
EES - Engineering Equation Solver (Modeling Programme)
FCU - Fan Coil Unit
HP - High Pressure
HT - High Temperature
HTF - Heat Transfer Fluid
HVAC - Heating, Ventilation and Air Conditioning
HX - Heat Exchanger
KTH - Kungl Tekniska Högskolan (Royal Institute of Technology)
LCC - Life Cycle Cost
LP - Low Pressure
LT - Low Temperature
MV - Motorized Valve
NREL - National Renewable Energy Laboratory
NV - Non-Return Valve
PLC - Programmable Logic Controller
P-T Chart - Pressure Vs. Temperature Chart
RH - relative Humidity
RV - Relief Valve
SP - Solution Pump
Tr - Tons of Refrigeration (= 12,000 BTU/hr)
TV - Throttle Valve
VSD - Variable Speed Drive
1. Introduction

1.1 Air Conditioning of Commercial Buildings in Sri Lanka

Air Conditioning (AC) systems are the major power consumer of total power consumed by the commercial sector in Sri Lanka. Heavy use of computers, server equipment and other office equipment increases the AC load considerably due to increased sensible heat energy generation. Most of the building designs are also aimed at using mechanical ventilation and air conditioning due to many different types of activities and to have a cleaner environment suitable for modern office environment. In general, there are two types of vapour compression air conditioning systems – individual air conditioners with split or window units and central systems with distributed fan coil units or air diffusers. The former is more commonly used in small and medium commercial buildings due to lower initial cost and easy design, though not energy efficient. Central types are used in large multi-storied complexes, where the use of individual units is impractical.

Whatever the type of air conditioning used in a building, its prime requirement is to achieve indoor comfort, as urban built environment cannot have natural ventilation with the present day congestion of buildings. Nevertheless the use of huge amount of electrical energy for indoor comfort has a negative impact on the natural environment, as a major portion of electricity generation uses non-renewable primary energy sources (currently non-renewable share of generation stands around 63% (1)).

Sri Lanka, being a tropical country with moderate temperature and year-through availability of sunlight with its location very close to the equator (the capital, Colombo has N-S bearing of +6°) has solar energy in abundance. Hence the environmental factors are conducive for the use of solar energy for air conditioning applications. The average climate in Colombo and suburbs, where most of the business organizations are located, has monthly average temperature range of 23 °C to 31 °C throughout the year with a relative humidity variation from 79% to 86%.

![Figure 1: Average monthly DB and RH (%) of Colombo](image)

Most of the modern offices and commercial buildings in Colombo and suburbs are air conditioned due to limitations for natural ventilation and high level of air pollution that affect operations of the commercial
buildings and maintenance of office and IT equipment. This makes the usage of air conditioners in most of the commercial buildings in Colombo and suburbs common, in contrast to the situation 30 years ago, where only a few buildings were air conditioned, as the building design had a high level of natural ventilation. The trend of built environment had changed from natural ventilation to mechanical dependent thereby increasing the electrical energy usage.

1.2 Non-conventional Air Conditioning

Vapour compression type chillers are used in all mechanical ventilation and air conditioning systems in Sri Lanka and is considered as the conventional type of air conditioning. This type of air conditioning is responsible for consuming major portion of electricity in commercial buildings. Modern split air conditioners use inverter technology to control speed of compressors with COP over 3.8, (e.g. York, Model: YMKJYCY024BAMSA-X, 24,000 BTU/Hr. Inverter type split units have a COP of 3.89 at its best operating point), the best achieved efficiency for a commercially available small compression type air conditioners.

Absorption type refrigeration is an old technique that has limited applications in air conditioning systems and refrigeration applications, where abundant waste heat or renewable energy, such as solar, exhaust, biogas, etc. is available. The limitations of absorption chiller applications mainly attribute to the following.

- Low Coefficient of Performance
- Lower ratio of cooling capacity to volume
- High initial cost compared to that of vapour compression type chillers

Absorption chiller systems are already in use and theory of operation has been well documented with many variations to improve COP. However, the technology is not popular compared to compression type chillers due to the reasons given above. The research focuses on an absorption chiller with a closed loop brine system and state sensors to increase COP of the chosen chiller system by effectively managing heat expelled and absorbed in the various stages of the system. This will optimize the heat energy used in the input stage of the system thereby making it more attractive in commercial applications with solar energy.

1.3 Research Objectives

1.3.1 Main Objectives

- The main objective of this thesis project is to design a solar energy operated absorption chiller with improved efficiency for air conditioning in a medium size commercial building, so that it can utilize nearly 100% solar energy fraction with suitable measures to overcome environmental limitations and adaptable in the available space.

1.3.2 Specific Objectives

- Assessment of the highest cooling requirement of a selected, medium, sized commercial building.
- Study on solar heat energy availability in the area of implementation of the selected absorption chiller an environmental conditions that affect performance of an absorption chiller.
iii. Design of a modified version of double effect absorption chiller driven by solar energy that is practically feasible in implementation in view of overall size of the solar aperture and heat storage required
iv. Simulation of basic version and modified version of double effect, series flow LiBr-H2O absorption chillers and comparison of performance.
v. Design of solar energy extraction and storage system that provides 100% solar energy for driving the chiller under worst weather condition using parameters obtained from the simulation, proposing suitable heat rejection system and digital control system with component details.

The following components are included in the system considered.

i. Solar energy collector
ii. Energy storage system
iii. Double–effect LiBr-H2O Absorption chiller system
iv. Cooling tower
v. Ancillary equipment – circulation pumps, control valves system, sensors
vi. Digital control system

1.4 Methodology

To achieve the objectives described in the previous section in an acceptable degree, new developments in the field of absorption chillers and solar energy extraction methods need to be reviewed. Several research papers on the subject will be studied and useful information has been added in this research as building blocks of the design. The following steps will be followed in the research.

i. Selection of a medium sized commercial building and assessment of energy requirement for air conditioning – Building energy simulation.
ii. Theoretical background of the complete system from the information obtained from literature survey and development of basis for calculation to obtain final system parameters after simulation.
iii. Modifications to the conventional configuration to achieve the objectives and simulate to verify its performance and comparison with the original version.
iv. Design of solar collector and storage systems that could achieve 100% solar fraction to run the chiller using operating parameters obtained from the simulation.
v. Setting design parameters for the components of heat rejection and control systems from results of simulation used in the system using commercially available components.
vi. Performance analysis based on the simulation results and discussion on technical feasibility.
vii. Discussion on application of the proposed system beyond simulation, possible pitfalls and limitations.

Descriptions of implementation of the above steps of methodology are given in details in the next sections.
2. Literature Survey

The solar powered absorption chiller that is being designed for an existing commercial building requires a combination of different technologies of which theoretical and experiential information can be grouped into the following areas.

i. Evaluation of energy requirement for chiller capacity of an existing building.
ii. Absorption Chiller system
iii. Solar energy capture and storage to meet the requirements of the absorption chiller.
iv. Control system that manages energy efficiency of the system while providing required effect according to the demand.

There are many methods for solar energy collection and absorption chiller designs are available. They have advantages and disadvantages in implementation in different environments. The required information was extracted from the previously carried out researches on similar system designs, and theoretical information from text books and information provided by KTH, Sweden. The following sections provide information about each of the above areas and other relevant information.

2.1 Evaluation of Energy Requirement

Solar energy collection systems as well as absorption solar chillers are large in size and costly in implementation. Hence it is important to evaluate the correct maximum cooling load of the selected building, as the available space for such system is limited in practical situations.

The potential of solar power extraction is limited by the footprint of a building and hence the capacity of the chiller that is expected to operate only on solar power. Both solar power extraction system and absorption chiller system selected has limitations. A combination of building load, solar thermal system and an absorption chiller for optimum cooling output was investigated and performances by modeling was carried out and published (2). According to this paper the solar aperture area per kW of cooling load varies from 0.5m² to 5 m².

Energy requirement of a selected building for air conditioning can be evaluated in two methods

a) By evaluating the requirement using energy modeling software and
b) By actual energy measurement

The accuracy of maximum cooling load assessment using building modeling depends on the accuracy of weather and building (constructional) data used. Actual measurement of electrical power used on air conditioning of the building would provide an indication that the simulation results are closer to the real requirement. It is not possible to depend solely on the actual measurement, as the measurement provides the loading on the particular day of a month and also it was not possible to measure Ac load alone as load circuits includes other loads as well. Whereas the simulation provides the maximum cooling load according to the weather data that is plugged into the software.
2.2 Non-Conventional Chiller Systems

Chiller systems that do not depend on gas compression or ‘non-conventional’ falls basically into two categories- Absorption chillers that use liquid /vapour in refrigeration process and Adsorption chillers where movement of refrigerant is achieved using solid absorbent material. However, the latter is not widely used compared to the former due to its low efficiency. A detailed study on adsorption refrigeration (3) concludes that it COP is within the range of 0.2 to 0.6 typically and is more appropriate in refrigeration not for air conditioning that is expected to achieve 100% solar energy utilization, thus leaving liquid / vapour based absorption refrigeration as the most suitable solution that will be used in the report.

The working fluid of absorption chillers consists basically of two components – an absorbent (or transport agent) and a refrigerant. Although several working fluids have been tried in absorption chiller systems, H₂O –NH₃ and LiBr-H₂O are the most successful mixtures adopted in practical systems (4)

2.3 Working Fluid of Absorption Chillers

Although H₂O –NH₃ and LiBr-H₂O are the best working fluids used in absorption cooling systems, both of them have pros and cons that limits their application in all types of applications. The advantages of LiBr-H₂O mixture over the other in solar energy applications can be summarized (4)

a) LiBr-H₂O mixture can provide higher COP
b) It can operate at temperatures as low as 70°C – 88°C, that makes it ideal for low temperature solar energy sources (NH₃-H₂O operates within 90°C – 180°C and COP is low at low temperatures)
c) LiBr-H₂O systems are simpler compared to the similar configurations of the other that requires gas separator (rectifier)
d) NH₃ is not an environment-friendly gas and hence restricted in building applications
e) NH₃-H₂O systems operate at higher pressure thus requiring higher pumping power.

The only significant disadvantages of LiBr-H₂O systems are
f) LiBr crystallization at higher concentration and as marked by crystallization boundary line on Pressure – temperature property chart.
g) The refrigerant of the system, water, cannot be used to achieve lower cooling temperatures due to freezing, where NH₃-H₂O can perform at temperatures well below 0°C thus making it suitable for refrigeration applications.

In view of the above features of two solutions, LiBr-H₂O solution becomes the best choice for air conditioning applications.

Another comparison done by Viktoria Martin of KTH (5) and is given in Table-1.
Although H2O/LiBr is not toxic, it is a highly corrosive at higher temperatures, say around 140°C. Hence, in a practical system, precautions need to be taken to prevent corrosion (use of corrosion inhibitors or the use of lower operating temperatures). In general solar driven chillers do not intend to attain such high temperature and the design temperature of the heat source is set to a temperature below the above level, thus making the system less prone to corrosion.

### 2.4 Configurations of Absorption Chillers

With the conclusion that LiBr-H2O is the most favourable option for chillers used in air conditioning applications, the research focuses more on this type of systems, unless specifically stated. There are many configurations of fluid based chillers that are based on two common configurations.

- Single –effect absorption chiller systems
- Double – effect absorption chiller systems

There are many variations and combinations of the two systems, especially of double-effect system that are used under different working conditions to meet specific objectives, some of which are discussed below.

#### 2.4.1 Single Effect Absorption Chillers

The single-effect refers to the transfer of fluids through the four major components of the refrigeration machine – evaporator (E), absorber (A), generator (G) and condenser (C), as shown by its schematic diagram in Figure 1. Single-effect LiBr-H2O absorption chillers use low pressure steam or hot water as the heat source. The circulating chilled water provides heat supply to evaporate water (refrigerant) in the evaporator because the latter is under very low pressure.
This configuration has been used with most of the studies in solar air conditioning as the basic configuration. Heat input $Q_g$ is provided from a solar energy collection system. Heat rejection from the absorber and condenser can be achieved using air or water cooling system. The choice of the system depends on the environmental conditions. Air cooling with higher ambient is unfavorable as it would lead to higher solution temperature in the absorber at a risk of crystallization of LiBr salt that obstructs the solution flow. On the other hand, effectiveness of water cooling system that generally uses a cooling tower is decided by the wet bulb temperature. Simulation carried out by Tierney, M.J (6) shows the effect of wet bulb temperature on COP of the system in which COP varies from 0.83 and 0.78 with the variation of wet bulb temperature within $15^\circ C$ and $21^\circ C$. Generally absorber temperature is set $10^\circ K$ above wet-bulb temperature.

Thermodynamic cycle of single-effect system can be presented on Vapour pressure Vs, Temperature diagram as shown in Figure-2 that depicts components placed on a Duhring Plot to illustrate heat transfer. The aligned black flow lines show constant LiBr concentration as in the Duhring plot.

![Figure 3](image)

**Figure 3 : Single Effect Absorption Refrigeration Cycle**

(Blue lines show steam flow and Black lines show solution flow)
A mathematical modeling carried out by Bajpai, V.K. (7) on a NH3-H2O single-effect system using Enthalpy Vs. Concentration plot and manual calculation shows that COP of the chiller was 0.69 with heat source temperature of 84°C.83 (fed from flat plate solar collector). However, comparable LiBr-H2O system would have slightly higher COP as shown by Tierney, M.J (6).

### 2.4.2 Double Effect Absorption Chillers

The desire for higher efficiencies in absorption chillers led to the development of double-effect LiBr/H2O systems. The double-effect chiller differs from the single-effect types with the use of two condensers and two generators to allow for more refrigerant boil-off from the absorbent solution. Although double effect systems can be subdivided into series and parallel flow systems, only series flow type is considered due to its wider application in research work referenced in this thesis project. Figure 3 shows a typical configuration of the series-flow double effect absorption chiller.

![Double Effect LiBr-H2O Absorption Chiller Configuration](image)

The higher temperature (HT) generator uses the externally supplied heat energy, which is steam in this thesis project, to boil off the refrigerant from the refrigerant-rich solution. The refrigerant vapor from the high temperature generator is condensed by releasing heat that in turn is used to provide heat to the low temperature (LT) generator, where more refrigerant is vaporized from the solution. Figure 4 shows the double effect absorption cycle on a (not-to-scale) Vapour Pressure-Temperature diagram.
These systems more commonly use gas-fired combustors or high pressure steam as the heat source. Double-effect absorption chillers are used for air-conditioning and process cooling in regions, where the cost of electricity is high relative to natural gas. Double-effect absorption chillers are also used in applications where high pressure steam, such as district heating, is readily available. Although the double-effect machines are more efficient than single-effect machines, they have a higher initial manufacturing cost. There are special materials considerations, because of increased corrosion rates (higher operating temperatures than single-effect machines), larger heat exchanger surface areas, and more complicated control systems. [6]

Absorber temperature has to be set approximately $10^\circ$K higher than the wet bulb temperature. According to simulation trails carried out by Tierney, M.J (6), COP variation of 1.4 to 1.28 was achieved with a wet-bulb temperature variation of $15^\circ$C and $21^\circ$C (against corresponding COP of 0.83 and 0.78 with single-effect systems. Double-Effect LiBr-H2O absorption chiller are commercially available in the range of 1Tr to 10TR (Tons of refrigeration) as modular units. Capacities of multiples of 10TR are generally designed to suit conditions of the customer site (4). However, now chillers over 150TR are commercially available (e.g. Carrier 16JL/JLR series) but with separately connected heat source and cooling systems.
2.4.3 Other Uncommon Configurations of Absorption Chillers

Li, S.F. and Sumathy, K. (4) have described several absorption chiller configurations that are variations and combinations of basic configurations described above and are used with solar energy input as described below briefly.

a) Single / Double Effect convertible system: Functionally this is similar to double-effect system and the only variation is an addition of a heat recovery stage to transfer part of input energy into the solution following the absorber, after giving up most of energy in the HT generator.

b) Two – Stage System: This scheme was specially developed to use with flat plate solar collectors so that overall system cost can be reduced with the use of low temperature solar collectors. The scheme also prevents possibility of crystallization by means of its low temperature input. The system constitutes of High pressure (HP) and low pressure (LP) stages HP stage has LP generator, Condenser, LP generator and HP absorber. LP stage has evaporator and LP absorber. Steam generation for condenser is produced only by HP generator, whereas LP generator produces refrigerant-rich solution for the former. Concentrated LiBr solution is passed to LP stage to produce solution with high H2O content with steam generated in the evaporator and the mixture is transferred to LP generator. However, this scheme has the disadvantages of system complexity over single-effect system, lower COP at nominal generator temperature and higher system heat rejection (cooling).

c) Two – Stage Dual Fluid System: Two stages if this arrangement, uses LiBr-H2O solution in first stage and H2O-NH3 in the second, both systems can be considered single-effect systems. Heat rejection of absorber of the 2nd stage is provided by the evaporator of the 1st stage and cooling load is provided by the evaporator of the 2nd stage. Heat input of 1st Stage and 2nd Stage can be provided by Flat Plate and evacuated tube collectors respectively. This scheme provides lower COP than the above scheme (Item b), but higher than that of H2O-NM3 2-stage system, while requiring considerably higher cooling water circulation.

d) Dual Cycle System: Two independent LiBr-H2O systems with high and low temperature inputs are the main units of this system. Input heat is supplied to the generator of HT stage and heat rejected from this stage is supplied to the generator of LT stage as its driving energy. Whereas heat rejected by the absorber and the condenser of LT stage is absorbed by the evaporator of HT stage. Heat rejection of the condenser of HT stage can be achieved by an air cooler. The evaporator of LT stage absorbs heat from the chiller load. The main advantage of this system is very low requirement of system cooling so that one air cooler can accomplish the requirement. However, its COP is very low and needs higher driving temperature that needs evacuated tube type collectors.

e) Triple-Effect System: By adding a topping stage above high temperature (HT) stage of double-effect configuration, a triple-effect system can be constituted. Heat rejected by the condenser and absorber of the topping cycle is used to drive the generator of HT stage. Refrigerant effects of all three stages share the cooling load. This type of system needs an input temperature as high as 250ºC and is more suitable with direct fired machines, but its COP is around 1.5.
f) Half-Effect Absorption Cooling (LiBr-H2O) System: A comprehensive description about this system is provided by M.H. Madveshi, P.N. Gupta and Nitin Pal (8). The system comprises of two generators, two absorbers and two heat exchangers, to form two groups (low pressure and high pressure) where generator and absorber are connected via heat exchangers. There are only one absorber and an evaporator. Condenser is connected to HP generator to receive refrigerant (steam) and condensed water is passed to the evaporator. The evaporator produces steam, after absorbing heat from the load, and passes it to LP absorber. The steam produced in the LP generator is fed to HP absorber to continue with the cycle. The advantage of this system is its ability perform with temperature as low as 65 - 75°C. However, the achievable COP of the system is below 0.45. The results also show that the dependency of COP on condenser temperature above driving temperature of 65°C is very small. They can be adopted in environments where condensation temperature is 50°C without the risk of crystallization (9).

2.4.4 Design Considerations and system Integration

In view of analyses in the foregoing sections, it is seen that major challenges of solar driven chiller systems are unsteady cooling capacity due to varying heat input and huge amount of heat rejection from absorbers and condenses. The first challenge is met by two methods – heat energy storage and chilled water storage. Though chilled water storage has low energy loss during storage period (4), the chiller should have spare capacity to generate sufficient chilled water to store while meeting the demand.

In order to meet the second challenge, it is required to select cooling system to achieve workable condenser and absorber temperatures. Investigations by S.M Su, X.D. Huang, R. Du (10) establishes that air cooled condenser cannot maintain lower temperatures than water cooled systems. The absorber temperatures should be set around 10°C above the wet-bulb temperature for a cooling tower to be effective ASHARE sets values of 35°C dry-bulb temperature and 25°C wet-bulb temperature as accepted standards (6). The temperature of heat input is also equally important to achieve best COP.

However, most of the above researches are carried out at locations where climatic conditions were with lower wet-bulb temperatures, even if the dry-bulb temperature was high. The wet-bulb temperature at the location of the research of this project closer to Colombo is high. The design has to meet this challenge.

In order to operate the chiller without auxiliary heat source, the solar collector type and area should be sizes so that it can be practically mounted on the top of the building. A storage tank may help to meet the requirement. There are new technologies that could enhance heat absorption, such as nanofluids that would enhance solar energy extraction per unit area (11). But this technology is not matured to be considered for practical usage and adaptation of such new technologies would still be a challenge.

Internal comfort level is an important factor in designing an air conditioning system and an accepted basis is required for internal conditions to set parameters for modeling. Such information would be obtained from the ‘Code of Practice for Energy Efficient Buildings in Sri Lanka – 2008’ published by Sustainable Energy Authority of Sri Lanka, a governmental body (12).
2.5 Solar Collectors and Storage

The contribution of solar collectors and storage would be equally important as that of chiller system in designing a solar driven chiller. Consideration of different technologies and their suitability in adopting in this project is supported by the literature analyzed below.

2.5.1 Solar Collectors and Performance in Chiller Applications

The type of solar collector mainly depends on the operating temperature required at the chiller input, which is generally a generator or desorber in absorption chiller applications. ‘A review for research and new design options of solar absorption cooling system’ by X.Q. Zhai, M. Qu, Yue Li, R.Z. Wang (9). They have discussed about combinations of flat plate and parabolic trough solar collectors (PTSC) with single and double-effect chiller configurations. It proposes double effect absorption chillers for the building that require high cooling loads, considering lower energy input required compared to single-effect versions, provided high direct irradiation is available. It also recommends the use of high temperature solar collectors such as PTSC with solar cooling based on double effect absorption chillers.

A slightly modified version of double-effect absorption chiller to suit the usage solar as primary energy source and a gas or oil burner to backup during the absence of solar energy was developed by Kawasaki Thermal Engineering Co and its description and performance data was presented by Akira Hirai (13). This solar driven hybrid chiller, as the manufacturer named it, can operate at 100% load with hot water input at 75°C chiller output can be reduced to 40% with input water temperature of 72°C and COP at the full load is 1.3 that can be maintained within the input water temperature range of 75°C to 90°C. Although the developer argued that addition of a burner as an advantage over a backup storage system, it could be considered as environment unfriendly and requires fuel management facilities that are not acceptable in all environments.

In view of sizing solar collector, use of higher driving temperature with double-effect absorption chillers using PTSC would reduce the required aperture area according the above analyses. Detailed heat transfer analysis and modeling of PTSC carried out by National Renewable Energy Laboratory (NREL), USA (14) has published detailed codes and analysis data using real parameters of PTSC elements and environmental conditions. This analysis shows the variations of thermal efficiency of a collector with ambient temperature and wind speed. It also shows how losses of various components of the receiver tube contribute to the total loss of the receiver tube (W/m). The results of the analysis show that Collator temperature up to 120°C (approx., 90°C above ambient) and heat transfer fluid (HTC) temperature difference of approx. 24°C the collector efficiency was around 73% at DNI of 800 – 930w/m². The above analysis show that it is not practical to feed HTF directly to the HT generator of the chiller to maintain required input while achieving the highest output from the collector as power output of the collector and the input to the HT generator are not the same. This concludes that a hot water buffer would be required even with a small capacity to serve for a short period of time.

2.5.2 Energy Buffer Used in Chiller Applications

Solar energy is not a steady energy source and its intensity varies continuously even during the 6 – 8 hours of available period. Similarly, the energy requirement by the chiller may also vary according to the load thereby requiring a buffer to store and supply regulated energy input. The driving energy of the system is given by a solar collector and storage system that provides a regulated heat to the chiller to suit the cooling load. However, with higher difference of temperature between that of the storage temperature and ambient, heat loss would be considerable and needs to be accounted. It is suggested that the optimum storage volume
for cooling applications range from 83kg/m² of the collector area (4). It suggests that storage of chilled water for later usage when solar energy is not available would be advantageous in view of heat loss.

An independent assessment carried out by an reputed institution on double-effect solar absorption chiller, which was converted from gas firing to hot water driven and having a capacity of 70kW, at a commercial building operated with a 106.5m² concentrated parabolic collector operated at 90 - 130°C (15). The calculation of design can be carried out using dynamic models to realize conditions at different time slots during a day. However, for energetic analysis of annual system performance could be obtained using steady state modeling with sufficient accuracy (2). Since the thesis focuses on feasibility of achieving long term results the latter option would be adopted.

It was pointed out that heat input required for the HT generator of the chiller and the optimum energy output of the solar collector are not identical and hence some type of control would be required. A hot water buffer is required is required to control heat input to the chiller while tapping off highest energy from the collector.

### 2.6 Heat Exchangers Used in absorption Chillers

Every main component of an absorption chiller requires a heat exchanger for its function and are of different types depending of the heat transfer method – sensible heat transfer or phase change type. The rate of heat exchanged and fluid flow rates are estimated during the analysis of the system model. Low-and high-temperature solution heat exchangers are important elements that have a great influence on the efficiency of the absorption chiller. Plate heat exchangers of welded construction with high efficiency can easily be miniaturized and connected to keep flow velocity at an optimum level even when the flow rate of circulating solution is low (16).

Compact Brazed Heat Exchangers are used to exchange heat between different flows in the system. Typical applications include heat exchange in the circulating stream between the hot LiBr-H₂O stream from the generator and the low-temperature stream from the absorber. This heat exchanger is often referred to as “high-temperature”. The other Brazed Plate Heat Exchangers can be used to cool further the LiBr-H₂O stream entering the absorber, “low-temperature”, and another to preheat further the stream entering the generator (16).

The above is a guideline for selection of heat exchangers. However, the heat exchangers used in different parts of the system have to be selected from the available types and models, as it is impractical to fabricate the most suitable heat exchanger. The best matching and available heat exchanger will be chosen for all applications, after evaluation of their theoretical parameters of the chiller using simulation.

Absorber and evaporator are two critical components that affect the performance of the chiller. It is required to maintain parameters resulted from modeling to achieve the expected results. Figure 6 shows a conventional horizontal falling-film absorber (shown with the evaporator).

**Figure 6 – Conventional Falling film of absorber & Evaporator**
The strong solution is distributed over the outer surfaces of the tubes, and the cooling water flows inside the tubes. The thin film of solution formed on the cooling tubes provides both good heat transfer to the cooling water and ample surface area for absorbing water vapour. (17).

High temperature generator needs to transfer required heat energy to the rich (weak in LiBr concentration) solution to generate estimated quantity of refrigerant, together with low temperature generator that has a much lower heat input. Shell and tube type heat exchanger will be used for this purpose, as it can deliver a high flow rate. In order to maintain high rate of heat transfer (rejection for the system), the largest of all heat exchangers of the system, the condenser must allow for high flow rate of brine. Shell and tube exchangers with multiple shell-passes would meet his requirement.

2.7 Solar Energy Availability and Climatic Conditions

The total solar radiant power per unit area or radiant flux (measured in W/m²) that reaches a receiver surface is the primary parameter in calculations. When integrating the irradiance over a certain time period, it becomes solar irradiation and is measured in Wh/m². When this irradiation is considered over the course of a given day it is referred to as solar insolation, which has units of kWh/m²/day (or x3.6MJ/m²/day). Solar radiation consists primarily of direct beam and diffuse or scattered components. The term “global” solar radiation simply refers to the sum of these two components. The daily variation of the different components depends upon meteorological and environmental factors (e.g. cloud cover, air pollution and humidity) and the relative earth-sun geometry. The Direct Normal Irradiance (DNI) is synonymous with the direct beam radiation and it is measured by tracking the sun throughout the sky (18).

In Concentrated Solar Power (CSP) applications, the DNI is important in determining the available solar energy. It is also for this reason that the collectors are designed to track the sun throughout the day.

Figure 7 – Daily Solar Irradiance on a flat plate positioned horizontal and tracking the sun and direct normal irradiance (DNI). (Source: Edith Molenbroek, ECOFYS, 2008).
Figure 7 shows the daily solar insolation on an optimally tilted surface during the worst month of the year around the world. Regions represented by light and dark red colors are most suitable for CSP implementation. Based on the information presented here it can be seen that desert and equatorial regions appear to provide the best resources for CSP implementation. Colombo, Sri Lanka, located at latitude of 6.49 N, falls well within equatorial region has highest insolation in the range of 5.0 – 5.9. The annual average solar irradiance of Sri Lanka is 700 – 800 W/m² at an average dry bulb temperature of 30°C (19). This information would be useful to estimate the size of the solar collector and capacity of the hot water storage.

A more precise study carried out by National Renewable Energy Laboratory (NREL) of USA shows that the distribution of annual solar resources in Sri Lanka varies from 15-20 MJ/m²/day (5.0 to 5.9 kWh/m²/day) across the country, with the lowest values occurring in the hill country in the south-central region. This information is within the range of more recent study on solar energy study on Sri Lanka and Maldives by NREL that shows Sri Lanka has steady insolation of 4.5 – 6.0 kWh/m²/day (20). The results also showed that the country does not experience sharp seasonal changes in solar resources. These studies do not include the other solar resource components (Direct Normal Irradiance, or DNI, and diffuse radiation) that are required for other types of solar applications, such as concentrating solar power and building day-lighting and missing information is extracted from other resources. However, this document does not provide any information about the irradiance.

Ambient conditions of the environment affect performance of the entire system. DNI at the location, dry and wet bulb temperatures, average wind speed, etc. These factors affect operating parameters of the chiller and the solar collector. Absorption chiller generally expel large amount of heat and it was stated that wet cooling system is the appropriate type. It is necessary to have adequate water supply to meet requirement of the chiller.

2.8 Integration of Components and Control

In order to achieve highest level of energy optimization, it is required to implement precise flow control of heat source fluid, brine, mixing solution, etc. Such control needs precise measurement of temperature and pressure of critical stages to achieve effective energy management by automation of flow control. Such automation systems include speed control of pumps, fans and valve control too. A control is required for an absorption chiller, as for conventional chiller to adjust the system to suit different chilled water requirements (cooling loads) and the process can be described as follows.

A double-effect, absorption chiller adjusts its capacity during part-load operation by modulation of the heat input to the high-temperature generator by adjusting the hot water supply, or other waste heat supply. A sensor is often located at the outlet of the evaporator to monitor the temperature of the chilled water leaving the evaporator, Tco. As the system refrigeration load falls, Tco decreases accordingly. Once a drop of Tco below a predetermined set point is sensed, the heat input to the high-temperature generator is reduced, and less water vapor is boiled off from the solution in the generator. The hot water supply rate can be modulated for a variation between 30% and 100% of the system design refrigeration load. Below 30 percent of the design load, the heat supply (hot water supply) is cycled on and off, and all refrigerant and solution circulation pumps remain on, so that the system refrigeration load is allowed to drop to 10 percent minimum of the design load. The refrigerant circulation pump is cycled at minimum cooling (21).
As a result of the above, the concentration (LiBr) of the solution entering the absorber drops, less water vapor is extracted to the absorber, and, therefore, the rate of evaporation and refrigeration effect in the evaporator are both reduced until they are balanced with the reduction of refrigeration load so that $T_{co}$ is maintained within acceptable limits. Since less water vapor has been extracted to the absorber from the evaporator, both evaporating pressure and evaporating temperature increases. Since less water vapor is boiled off in the generators, the rate of heat transfer at the condensing surface and the amount of water vapor to be condensed to liquid water in the condenser also reduces, thus reducing the condenser heat transfer rate with possible reduction of brine flow rate.

If $T_{co}$ rises above a limit, on the other hand, more heat is provided to the generator expelling more refrigerant, the concentration of solution and the refrigeration capacity increase and evaporator temperature, $T_e$ again falls within preset limits. Of course, the increase in solution concentration should not exceed the saturation limit (21).

During part-load operation, the following changes of the operating parameters will occur.

- The mass flow rate of refrigerant is directly proportional to the load ratio (of the full load).
- The evaporating pressure, $P_l$ and evaporation temperature, $T_e$ rise.
- The condensing pressure, $P_m$ and LT condensing temperature $T_c$ will drop.
- The boiled-off temperatures in the high- and low-temperature generators will decrease.
- Heat input to the high-temperature generator is reduced.
- The drop in temperature of cooling water at a lower outdoor wet-bulb temperature lowers the condensing temperature $T_c$ (also $P_m$) and, therefore, the heat input.

In order to achieve most energy efficient system after integration of three main units of the solar driven absorption chiller namely,

- Solar energy supply unit
- Chiller unit
- Cooling brine system

There should be a control system that monitors parameters of the chiller system and regulates heat energy input and cooling system. Such control achieves several objectives in view of energy saving as follows.

a) Maintaining heat storage at the maximum temperature, thereby autonomous functioning of the system is possible, in the absence of long spells of solar energy.

b) To vary heat input to the chiller according to the cooling load

c) To control the secondary cooling (brine) system at the correct operating temperature to achieve best efficiency of the system

d) To switch to a secondary energy source in the event, heat storage is insufficient to run the system, during long periods of the absence of sunlight.

Commercial systems achieve the above requirements in different ways. It is possible to use independent controllers sensing only the relevant parameters (generally temperature or pressure) to control specific operations, as such operations do not depend on several parameters (e.g. cooling water flow (brine) control in condenser senses the condenser temperature only). However, some commercial systems developers have different opinions in this respect.
As an example, SolarNext, a commercial system developer has the following opinion in developing their latest control system for systems. The previous solar cooling demonstrations and pilot projects were using several single controllers (as described above) e.g. for the solar thermal system, for the chiller, for the condenser and for the chilled water or heat distribution, which are together cost intensive and are not always operating optimal together. The alternative was until now an expensive PLC controller which had to be programmed for each single case. Because of that the SolarNext has decided in the year 2007 to develop an own system controller for the whole system, as shown in Figure 6, which has an influence from the automotive sector and is cheap and system oriented \(^{(22)}\).

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Figure 8 – Control system of a solar powered HVAC chiller

The account on control made by SolarNext is not precise in the context of today’s electronic controls with sharply dropping prices. Present day motor control by variable speed drives (VSDs) has more options in control with built-in logic control functions including analog measurements. However, a dedicated system controller would be user friendly so that the user can easily set the required operating conditions. E.g. temperature of outlet chilled water \((T_{co})\). Another important activity of the control system is to monitor state parameters so that they do not reach extreme limits that have adverse effects on the system (e.g. solution concentration that causes crystallization, excessive condenser temperature due to malfunctioning cooling tower, etc.) brief description of safety and interlocking controls are given below \(^{(21)}\)

a) **Low-Temperature Cutout.** If the temperature of the refrigerant in the evaporator falls below a preset value, the system / unit controller shuts down the absorption chiller to protect the evaporator from freezing. As soon as the refrigerant temperature rises above the limit, system / unit controller starts the chiller again.
b) **Chilled Water Flow Switch.** When the mass flow rate of chilled water falls below a limit, a pressure-sensitive or flow-sensitive sensor alerts the system / unit controller, which stops the absorption chiller.

c) **Cooling Water Flow Switch.** When a loss in cooling water supply is detected by the pressure- or flow-sensitive sensor, the system / unit controller shuts down the absorption chiller. The chiller starts again only when the cooling water supply is restored.

d) **High-Pressure Relief Valve.** A high-pressure relief valve or similar device is often installed on the shell of the high-temperature generator to prevent the maximum pressure in the system from exceeding a preset value.

e) **Auxiliary Heat Supply Safety Controls.** The auxiliary heat source such as gas of liquid fuel used for heat supply for the absorption chiller needs controls of high-pressure and low-pressure switches, flame ignition, and monitoring for its burner and generator.

f) **Interlocked Controls.** Absorption chiller should be interlocked with chilled water pumps, cooling-water pumps, and cooling tower fans so that the absorption chiller starts only when these pieces of equipment are in normal operation.

### 2.9 Challenges Addressed in Research

Literature surveyed in the foregoing sections analyzes much research on absorption chillers and solar energy extraction and storage systems. There are several research papers on performance analysis of chillers and solar fraction used for driving chillers under different building environments and climates. Several methods of improving overall efficiency (COPt) are also discussed. However, they lack the following

i. Analysis on whether large area required for solar collector and storage are practically adaptable within the footprint of a building
ii. Use of waste heat in a suitable manner
iii. Minimum usage of electrical power

The research focuses on overcoming the above practical difficulties and improvements in view of feasibility of implementation and minimum use of grid energy. Then proposes methods to overcome such limitations using modifications not discussed in studied research papers. The final objective is to compare performance of the basic and the modified system, how well the resulting solar driven absorption chiller system is adaptable under local environmental conditions and limitations in practical implementation.
3. System Design and Theoretical Analysis

This chapter describes the methods adopted in the thesis research work, based on the theoretical and experiential information discussed in Chapter 2. The methodology used for integrated absorption chiller system is discussed in this chapter.

For the purpose of calculation air condition application of a commercial building was selected. It is required to select a building with appreciable air conditioning load as the starting point of the project. The reason for the requirement of appreciable air conditioning load is for achieving considerable energy saving that can justify the use of an absorption chiller. One of the objectives would be an attractive rate of return on investment (ROI) for the proposed absorption chiller. Selection of a suitable building that falls into the category of medium sized commercial building is the first step of the research, as large buildings generally have centralized air conditioning with high COP than split types. A building with split type air conditioning would be more suitable, as they are less energy inefficient. Having selected the building, the next step is to evaluate capacity of a suitable air conditioning system and the energy requirement for air conditioning. This could be achieved by carrying out power / energy analysis on the building as described below.

3.1 Assessment of Cooling Load

3.1.1 Energy Modeling of Building

Assessment of power and energy requirement is using energy modeling of the building with suitable software, giving input parameters for required internal conditions is the first step of the project. Energy modeling of the building could provide more comprehensive results of maximum cooling load, as it takes the most suitable indoor settings and weather conditions on the hottest day of the year. Indoor conditions of a building have to be set for energy modeling software to calculate energy used for air conditioning (cooling). The required internal temperature is set as described in Section 2.4.5.

Modeling of energy usage of the building for air conditioning requires input parameters that include weather information of the city, where the building is located. The next activity is to obtain structural details of the building that is required for energy modeling of the building using DesignBuilder software that includes weather database of the area where the building is located. Structural details of the building need to be searched with the support of the management.

The recommended indoor conditions as per the local codes are dry bulb temperature of 25°C ± 1.5°C and relative humidity of 55% ± 5%. It also permits higher temperatures within the comfort zone of ASHRAE. The Codes also specifies average outdoor conditions of 31°C dry bulb temperature and 27°C wet bulb temperature.

3.2 Modeling Heat Driven Chiller

It was pointed out under literature survey that there are practical limitations for installation of a solar collectors and thermal storage that are required for a solar drive chiller system in a commercial building. One of the critical limitations is availability of roof area to place the solar collector of required size. In reviewing literature it was concluded that double-effect absorption chiller with PTSC and heat storage is the suitable option. The project aims to improve efficiency of the chiller and reduce the required capacity of cooling system with suitable modification stated in the next section and Engineering Equation Solver (EES) will be used for simulation.
3.2.1 Simulation of Conventional Double-Effect Absorption Chiller

The basic configuration of a series flow double-effect absorption chiller shown in Figure-9 will be used for simulation and its parameters used for simulation are given below.


The following parameters are set with the justification given for initial modeling.

a) Design condenser temperature, \( T_c = 40^\circ C \) to meet the maximum ambient of average \( 30^\circ C \), and setting \( 10^\circ C \) temperature difference of the condenser.

b) Design evaporator temperature, \( T_e = 12^\circ C \), to provide chilled water supply temperature of around \( 14^\circ C \) to allow for heat gains along the chilled water lines and temperature difference in FCUs to provide maximum supply air temperature of \( 20^\circ C \). Expected return water temperature is \( 22^\circ C \).

c) Design temperature of generator heat source, \( T_{gh} = 95^\circ C \). This temperature of the heat source is expected to be provided by a solar heater. This is considered a reasonable temperature with a CSP type solar collector with buffer storage.

Combinations of different values of the above parameters will be used for several trials to achieve the best performance for the chiller. Different parameters will be used in EES for evaluation of performance to achieve the optimum results. The complete simulation results with the best parameter combinations are given in Appendix-3.

3.2.2 Simulation of Modified Double-Effect Absorption Chiller

The configuration of the chiller used for initial modeling is shown in Figure-9 and Table-2. The same initial design parameters that are used for the basic double-effect absorption chiller are used for the modified double-effect configuration for the initial step of modeling. However, these parameters will be changed during performance evaluation to optimize its performance. The following design parameters are used for initial modeling of the system, based on the load and ambient conditions.

a) High temperature generator, \( t_1 = 100^\circ C \)

b) Condenser temperature, \( t_3 = 38^\circ C \)

c) Evaporator temperature, \( t_4 = 12^\circ C \)

The required cooling capacity of the chiller, based on the results of simulation to obtain maximum cooling load and energy measurements carried out (Section 4.1). Modeling of double-effect chiller will allow variation of parameters that are initially set as per theoretical system to achieve better COP\(_h\) and lower heat input to the system. Several trials of the basic system without additional heat exchangers were carried out. Minimum heat source temperature, minimum heat energy input for a given evaporator load with highest COP\(_h\) are the best features aimed at during trials with two internal heat exchangers (as used in basic configurations). The next step is to add an external heat exchanger with external heat input from waste heat energy.

The main requirement of the proposed absorption chiller system is to achieve highest energy efficiency and to reduce load on cooling tower with the state parameters suitable to use with solar energy such as lower temperature of heat source and limitation of solar power availability in a particular application. In order to achieve the above requirement, the design intends to re-use part of the heat rejected by the absorber and condenser. Inlet and outlet temperatures of the heat exchangers used for absorber and condenser in the
above feedback of heat energy, need to be set to enable such heat re-use. However, effectiveness of this modification and setting overall parameters of the system was known only after modeling of the system. The proposed basic configuration of double-effect absorption chiller with its parameters is shown in Figure 9 and Table -2. Refrigeration cycle with components on non-scaled P-T chart is also shown in Figure-16, Section 4.4.1.

Figure 9 – Conceptual configuration of Double-Effect Absorption Chiller

Table- 2 - Description of units and parameters used in Double-Effect Absorption Chiller

| Qa | Rate of heat expelled from absorber | Qg1 | Rate of heat supplied to HT generator |
| Qc | Rate of heat expelled from LT Condenser | Qg2 | Rate of heat supplied through LT generator |
| Qe | Rate of heat absorbed by evaporator | Hx-1, & 2 : Heat Exchangers |
| Qct | Rate of heat expelled to environment (liquid./air) | TV-1 to 4 : Throttle Valves |

The next step of the methodology is to decide the size and type of solar heater and hot water storage to supply heat energy required for operation of the chiller.
3.3 Solar Heater Development

As described in Section 2.5 Concentrated Solar Power (CSP) type solar collector is favourable for solar driven absorption chiller application. The next step is to size the CSP solar collector and the hot water (heat energy) storage. Average solar energy collection information was obtained from research data of NERL as described under Section 2.7.

3.3.1 Operational Details of the Chiller

According to the meteorological information, sunrise and sunset in January is around 06.25 hrs. and 18.15 hrs. respectively, whereas the respective times of mid-year (June) they are 05.55 hrs. and 18.25 hrs. respectively. Hence it can be safely account that solar heating process starts around 07.30 hrs. and ceases at 16.30 hrs during which useful energy is available for heat energy storage and usage in the absorption chiller system. Moreover, for calculation purposes, the average insolation can be considered as 5.5 kWh/m²/day, as described in Section 2.7.

Medium scale commercial building generally starts at 08.30 hrs. and closes at 17.30 hrs., the latest, during week days and many organizations work on Saturdays for approximately 4 to 5 hours. It is shown in Table-3 below based on the simulation results that approximately 5 hours of the working period would require 100% chiller capacity and the rest of the working period (08.30 – 10.30 and 15.30 – 17.30) would need about 60% capacity.

Table – 3: Energy Usage at Different Times of Day

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<thead>
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</tr>
</thead>
<tbody>
<tr>
<td>Power (kW)</td>
<td>17.5</td>
<td>19.0</td>
<td>25.0</td>
<td>27.0</td>
<td>25.0</td>
<td>22.0</td>
<td>16.0</td>
<td>25.0</td>
<td>16.5</td>
<td>13.0</td>
</tr>
<tr>
<td>% of Peak</td>
<td>64.8</td>
<td>70.4</td>
<td>92.6</td>
<td>100.0</td>
<td>92.6</td>
<td>81.5</td>
<td>59.3</td>
<td>92.6</td>
<td>61.1</td>
<td>48.1</td>
</tr>
</tbody>
</table>

Average for 4 hours of low power = 63.89%
Average for 5 hours of high power = 91.85%

According to the above information for solar collector and average insolation,

\[
Q_{gh} \times (5 + 4 \times 0.6) \text{ kWh.} = 7.4Q_{gh} \text{ (kWh)}
\]

Where, \(Q_{gh}\) is the input heat energy rate required for the HT generator at its 100% capacity,

The available daily solar energy input is

\[
I_{av} \times A_a \text{ (kWh)}
\]

Where, \(I_{av}\) is average insolation in kWh/m²/ day and

\(A_a\) is the total aperture area of the CSP collector in m²
With the average value of $I_{av} = 5.5 \text{ kWh/m}^2/\text{day}$, as described in section 2.7.

Required CSP aperture area is given by (from equations 3.6a and 3.6b,

$$A_a = 1.345 \times Q_g \text{ m}^2$$  \hspace{1cm} (3.6c)

### 3.3.2 Parabolic Trough Solar Collector

The above information with energy collection data given in Section 2.7 would make it possible to evaluate the capacity of CSP type solar collector that meets source temperature of the chiller. The proposed solar collector will be of trough type parabolic collectors with absorber of stainless steel with vacuum type glass insulator that prevents losses to a great extent of which details are shown in Figure 10. This type of solar collectors is available in the market now at reasonable costs.

![Parabolic Trough Solar Collector Diagram](image)

**Figure 10 – CSP Collector absorber construction** (23)

The choice of reflector material and construction of the CSP collector are crucial factors in view of durability, cost effectiveness and maintenance. Anodized aluminum sheets would meet most of the above requirements, with the additional advantage of lightweight. It can achieve 85% to 90% reflectivity (24). The collector supporting structure is made of mild steel U- channels and angle iron with a coating of high quality anti-corrosive paints for durability.

In order to collect the maximum possible radiation of the sun, a CSP collector can be rotated about different axes. There are several ways a collector system can be placed and change the facing direction to achieve maximum energy.

a) Rotation about North – South axis
b) Rotation about East – West axis
c) Rotation about an axis inclined and parallel to the earth’s axis (polar axis). (24)

The last method can be considered the most suitable for equatorial countries such as Sri Lanka, as it requires one rotation per day at the rate of 15° per hour.
The entire collector system that may consists of several individual troughs that are placed in parallel on the roof top and all of them can be guided (rotated) with a single drive shaft that is driven by a programmed stepper motor and coupled to axis of each trough with a worm & wheel coupling. It is possible to direct the trough in the minimum energy collection direction to avoid energy collection when the storage temperature tends to rise above a threshold value.

### 3.3.3 Solar Energy Storage

Three primary requirements that have to be met to utilize solar energy were stated early in this section and the next step is to determine a method of solar energy storage, a transport medium and a suitable mechanism. The primary factor for estimation of the storage is to decide the heating medium or brine. Water is considered the best medium due to its high thermal capacity, free availability, environment friendliness and lowest cost, but its boiling point could be a drawback, where operating temperature is above 100°C as a sensible heat storage system. Sensible heat storage is preferred in this application as it can be managed more conveniently in comparison to steam supply and as the operating temperature is slightly above the boiling temperature at atmospheric pressure. A high pressure water circulation system or aqueous solutions (with boiling point above the operating temperature) would be two common solutions for this problem. The former is preferred, as handling of aqueous solutions of large volume would encounter with other complications and higher cost would affect LCC.

The storage should be large enough to maintain sufficient hot water at a temperature usable by the chiller (HT generator). Hence, the correct estimation of the solar collector and storage system is a key activity for successful operation of the system. Such estimation needs the following data and assumptions.

a) Operating temperature of the high temperature generator  
b) Number of hours of daily usage of the chiller at different capacities.  
c) Average daily heat energy availability or insolation (allowing for absence of sunlight)  
d) Type of solar collector

![Figure 11 – Basic Schematic Diagram of a Solar Collector & Storage](image)

The thermal storage is expected to maintain a slightly higher temperature than the operating source temperature of the chiller and the storage outlet water is mixed with water at ambient temperature or return.
water from HT generator to achieve the exact temperature. Relationship for the energy collection by the solar collector during the useful period of 07.30 hrs. to 16.30 hrs. can be given as

\[
\text{Total energy collection during day, } E = \int_{t=07.30}^{16.30} I_s \, dt
\]

- \(I_s\) - Average solar irradiation (w/m²)
- \(E_s\) - Total energy collected per day with undisturbed irradiation in kWh/m²

\[
E_s \cdot A_s = C_p \cdot \rho \cdot \Delta T \cdot \int \dot{m}
\]

Since insolation considered (from literature) considered a constant value for all practical purposes

\[
I_{av} = E_s \cdot A_s
\]

Where,
- \(\Delta T = T_c - T_L\) is maintained a constant by varying \(\dot{m}\)
- \(A_s\) = Total area of solar collector
- \(\dot{m}\) = Water flow rate through the solar collector
- \(C_p\) = Specific heat of water at operating temperature range
- \(\rho\) = Average density of water within the operating temperature range
- \(T_c\) = Temperature of collector outlet water
- \(T_L\) = Temperature of stored water (or water supplied to load)

It is required that \(T_c > T_L\) to have useful output from the solar collector.

Since \(T_c\) is not a steady temperature, \(\dot{m}\) is controlled to maintain steady temperature at the storage. It is possible to allow the temperature of the storage to go higher within safe limits (high pressure relief valve is provided), as the temperature of water into the chiller can be mixed with cold water in case mixing with return water cannot reduce the temperature of the inlet water to the chiller within a required level.

Another important aspect of the thermal (hot water) storage tank is its thermal insulation. Overall conductance of heat, \(U\) (\(U= \) overall heat transfer coefficient of the tank shell and \(A = \) heat emitting surface area), of the tank can be decreased to minimize the heat loss, but at a considerable cost that adds to the total system cost.

\[
\text{Heat loss from the storage can be expressed as } Q_{s1} = (UA) \cdot (T_s - T_o),
\]

Where, \(T_s\) is the average temperature of hot water storage and \(T_o\) is the ambient temperature.

### 3.3.4 Energy Storage Sizing

Then, the daily heat energy storage required is \(Q_{gh} \cdot \tau_a\) (kWh) \(\quad (3.7a)\)

Assume that the storage temperature is \(T_s\) (°C), heat source temperature at the HT generator is \(T_{gh}\) (°C) and volume of the storage is \(V_s\) (m³).

The energy storage required during solar power absence is \(C_p \cdot \rho \cdot V_s \cdot (T_s - T_{gh})\) (kJ) \(\quad (3.7b)\)

Where,
- \(C_p\) = specific heat of water (kJ/kg/°C) and
- \(\rho\) = density of water (kg/m³), at storage and generator supply temperatures (Section 4.5.1) Equations 3.7a and 3.7b can be equated with kWh to kJ conversion as follows

\[
Q_{gh} \cdot \tau_a \cdot 3600 = C_p \cdot \rho \cdot V_s \cdot (T_s - T_{gh})\]

\(\quad (3.7c)\)
With the approximate values of \( C_p \) and \( \rho \), Eq. 3.7c can be written as

\[
Q_{gh} \tau_a \cdot 3600 = 4.25 \times 938 \times V_s \times (T_s - T_{gs})
\]

Hence,

\[
V_s = 0.903 \frac{Q_{gh} \tau_a}{(T_s - T_{gs})}
\] (3.7d)

The above equation provides capacity of the storage tank.

It is also important to calculate energy loss and insulation requirement of the storage system. Insulation of the storage is associated with a cost that is added to the overall cost of the project, which can be evaluated with the energy accumulated and that is lost during the non-operational period (non-working).

If the heating cycle of one week is considered,

Total energy extracted during a week is (from. Eq. 3.6b) = \( 7 \times 5.5 \times A_a \) kWh \( (3.7e) \)

Energy used by chiller during 5 ½ day week is (from. Eq. 3.6a) = \( Q_{gh} \times 7.4 \times 5.5 \) kWh \( (3.7f) \)

The excess energy in the system would be = \( 38.5 \times A_a - 40.7 \times Q_{gh} \) kWh \( (3.7g) \)

Equation 3.7g allows estimating allowable maximum heat loss from the storage system, if the total surface area of the storage tank (including area of connecting pipes of the system) and the heat transfer coefficient of the surface insulation, \( U_s \), is known. However, the energy storage could be enhanced by reducing the losses, but at a justifiable insulation cost.

Availability of sun radiation for heating was also discussed above and the absence of sun’s radiation starts around 16.30 hrs. stops around 7.30 hrs. next day. Hence the heat loss continues for approximately 15 hrs., even if the chiller is not used. The allowable maximum rate of heat loss would be calculated using the storage temperature \( T_s \) and the rated temperature required to drive the HT generator for the maximum load. This heat loss will evaluate the required \( (UA) \)s, with the known surface area of the tank, \( Q_s \) is given by

\[
Q_s = A_s \times U_s \times (T_s - T_o)
\] (3.7h)

Where,

\( T_s \) = maximum temperature maintained in the storage tank

\( T_o \) = ambient temperature

\( U_s \) = Overall heat transfer coefficient of the tank with surface insulation

\( A_s \) = surface area of the tank

The insulation of the storage tank is such that value obtained from Eq. \( (3.7h) \) should be less than that is given by Eq. \( (3.7g) \). Higher the excess value makes more reliable operation to allow for the period of the absence of sunlight and limitation of the usage of other energy sources.

### 3.3.5 Integrated Solar Driven Absorption Chiller

Smooth operation of the chiller system depends on the precise control of heating water and coolant flow rate control to suit the cooling demand. Such control can be achieved by pump speed control and motorized valves operation to achieve the required input energy optimization. Figure 12 shows the control diagram of integrated system proposed in the project.

The solar heater and hot water storage comprises of two hot water pumps and two motorized control valves and pressure relief valve.
a) Hot Water Control System
The Hot water collection, storage and supply are done by two pumps that are driven by two Variable Speed Drives (VSDs). These VSDs can have analog inputs - either DC voltage or currents (through 4-20mA transducers) that are provided by temperature transducers. The control of solar collector system was kept independent so that it has the option of running even when the chiller is switched off. However, central control system will have input from these temperature sensors for monitoring purpose and to activate bypass function (using motorized Valve MV-1) when the chiller is not in use.

b) Chiller and Brine Control System
Since these controls are directly related and needs coordination they are controlled using a high-end programmable Logic Controller (PLC) that have capability of sensing analog temperature signals and controlling motorized valves solution pumps and brine pump in a closed-loop control system to enable precise control.

The PLC can be programmed to control the entire chiller system and hot water circulating system according to the operational parameters of the system. However, it may be required to fine tune the system to suit actual conditions. The controller monitors the evaporator temperature to control the heating water supply to the HT generator. When the chiller load is low, the evaporator temperature tend to lower and the PLC gives a signal to the hot water outlet pump (storage to HT generator) to reduce the water flow and vice versa.

Chiller has state monitoring system with five temperature sensors that provide control signals to the PLC, analyzes the parameters of the system. Its control is such that three pumps (two systems pumps and the brine pump), with bypass lines and motorized valves in brine system maintains optimum operations of the brine (secondary cooling) system under different load conditions. The Brine Pump and two system pumps (SP-1 and SP-2) are controlled by a VSD for better harmonization with the system states, whereas recirculation pumps are directly driven only with on off controls.

The cooling tower motor is also controlled by the main system or PLC in order to reduce fan and recirculation pump speeds to suit the cooling load. Figure 12 shows a scheme for the combined system with proposed sensing line (continuous) and control lines (broken) of the system. Technical details of the components of the control system are given in Section 5.4.4.
Figure 12 – Integrated system with control system
4. Modeling and Review of Results

The actual results of the research work according to the methodology described in Chapter 3 and information used in Chapter 2 are described in this chapter.

4.1 Building Energy Modeling and Measurement Results

The selected commercial building for thesis work is located at approximate bearings of 60°.49'N, 79°.55'E and a site map of the building is shown in Figure-13. The site consists of four building blocks of which two large blocks, Building-A and Building-C, only has air conditioning and will be considered in the project.

Both Building-A and -C are two storied and only office areas are air conditioned with split type units. Split type air conditioners are used in the areas that need air conditioning and the capacities (in Tons of refrigeration, Tr) and numbers of split AC units installed are as follows.

Summary of floor details of air conditioned and other areas are given in Table-4.

<table>
<thead>
<tr>
<th>Block</th>
<th>Floor</th>
<th>Total Area (m²)</th>
<th>AC Area (m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Building-A</td>
<td>Ground Floor</td>
<td>4549</td>
<td>446</td>
</tr>
<tr>
<td></td>
<td>Upper Floor</td>
<td>4413</td>
<td>1446</td>
</tr>
<tr>
<td>Building-C</td>
<td>Ground Floor</td>
<td>1760</td>
<td>445</td>
</tr>
<tr>
<td></td>
<td>Upper Floor</td>
<td>1360</td>
<td>1090</td>
</tr>
</tbody>
</table>

**Description of Split Units Used**

a) 1.0 Tr = 7 Nos.
b) 1.5 Tr = 2 Nos.
c) 2.0 Tr = 2 Nos.
d) 3.0 Tr = 3 Nos.
Total = 23 Tr. (80.5 kW)
4.1.1 Energy Modeling of the Buildings

In order to verify the measured air conditioning load and to assess maximum load for air condition, considering the average weather condition during a whole year in the area, energy modeling of the building would be useful. Weather information of Colombo suburb, Ratmalana, which very close to the location of the building, is available from a recommended plug-in used with energy modeling software. The information of building components required for energy analysis can be provided from the available structural details and the main components are as follows.
i. Structure – Reinforced Concrete
ii. Walls – Hollow Cement Blocks
iii. Roof – Asbestos with steel truss with ceiling
iv. Windows panes - Non-glazed single layer 3mm plain glass
v. Window frames – Aluminum
vi. Doors – Anodized aluminum frame and non-glazed single layer 3mm 50% plain and 50% shaded

The information relevant to ventilation under ‘HVAC settings’ option of the analysis programme was set to split type ACs without mechanical ventilation (as split type ACs are presently used without forced ventilation) and 0.5 ACH is considered for air leakages. Other information required by the software is taken from the default values, unless there is a deviation from the actual status.

The above simulation results provide the maximum energy requirement for cooling of the building under consideration on the hottest day of the year as per the weather data. This would determine the size of the chiller capacity for the building. Similar simulation will be carried out for other parts of the buildings and tabulated to find the total energy requirement for cooling of the building complex. Energy information of two buildings is separately tabulated to size fan coil units for individual areas in order to calculate price of the chilled water cooling system.

The cooling data obtained from simulation results of Building-A, Ground Floor are shown below in Figure-14. Similar graphs for ground floor of Building-C are given in Figure-1A and -1B in Appendix-1. The summary of analysis (areas of each floor and results of energy modeling) was obtained from the analysis results given in Appendix-1. Summary of air conditioning load of two floors of each building is given in Table-5 below.

<table>
<thead>
<tr>
<th>Block</th>
<th>Floor</th>
<th>AC Area (ft²)</th>
<th>Cooling Load (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Building-A</td>
<td>Ground Floor</td>
<td>447</td>
<td>7.5</td>
</tr>
<tr>
<td></td>
<td>Upper Floor</td>
<td>1716</td>
<td>26.3</td>
</tr>
<tr>
<td>Building-C</td>
<td>Ground Floor</td>
<td>1039</td>
<td>15.7</td>
</tr>
<tr>
<td></td>
<td>Upper Floor</td>
<td>828</td>
<td>16.9</td>
</tr>
</tbody>
</table>

**Total Design Cooling Load** 66.4 kW

The information in Table-5 provides that the total chiller load required by the complex is 66.4kW according to energy analysis. The cooling load evaluated is 47.5kW according to measurement results of 17kW of electrical power input at an average COP of 2.5. Since simulation of cooling loads was carried out day the of the year, it can be considered that the results are considerably in agreement. Allowing a margin for any measurement errors or assumptions used in simulation, a total cooling load of 70 kW will be used for modeling. Figure-14 shows graphs from the simulation program DesignBuilder.
The split type AC units used in the building are Chinese made and are of less known brands. Hence it can be considered that they fall into the category of low COP. Room air conditioner benchmarking report (25) reveals that low efficient split type AC manufactured during 2002 to 2009 period varies within 2.28 and 2.54. The ACs presently in operation were installed during this period, as the building was constructed early part of this period. Hence it is assumed that the best COP for standard split type AC units is 2.5, base on which the above measurement indicates that the cooling requirement on this day is 42.5 kW. In reality, with old AC units this value of COP would be less.

The average monthly consumption was obtained from the electricity bills and the percentage of energy used for air conditioning was calculated using information obtained from simulation results (Figure-14, Figure-1A and 1B in Appendix-1) with Table-5 and are summarized in Table-6.

**Table 6: Energy Usage Summary**

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average consumption per hour (AC load)</td>
<td>15.5 kWh</td>
</tr>
<tr>
<td>Average consumption per day (AC load)</td>
<td>124.4 kWh</td>
</tr>
<tr>
<td>Average consumption per month (AC load)</td>
<td>2,736.9 kWh</td>
</tr>
<tr>
<td>Monthly total consumption as per bill (kWh)</td>
<td>4650 kWh</td>
</tr>
<tr>
<td>Energy usage for air conditioning</td>
<td>58.86 %</td>
</tr>
<tr>
<td>Maximum electrical power drawn by the AC load</td>
<td>17 kW</td>
</tr>
</tbody>
</table>
4.2 Chiller System Modeling

4.2.1 Basic Double-Effect LiBr-H2O Absorption Chiller

The use of Engineering Equation Solver (EES) for modeling of a double-effect LiBr-H2O absorption chiller was carried as a trial to compare improvements made to the modified chiller. Details of simulation with assumed parameters are given in below and in Appendix-1. The requirement of the building cooling load, which is 70kW as discussed in the previous section. The following assumptions are made in modeling formulation of the system with different parameters to achieve the most energy efficient system.

a) Energy used by the system solution pumps is ignored
b) All units under consideration are in steady state condition
c) Condition of outlet fluid from any unit has the steady state properties of that unit
d) Heat losses in the interconnecting pipes are negligible
e) Pressure drops are negligible.

The initial modeling of double-effect chiller was done without any external heat exchangers and the heat exchangers were introduced in the next cycle. The following parameters were set according to the environmental conditions, requirements of the system. The information in Table-5 provides the total chiller load required by the complex. The following input parameters to suit the real environmental conditions and cooling demand are used for the initial trials.

i. Maximum ambient temperature = 31°C
ii. Maximum relative humidity = 76%
iii. Cooling load = 70kW
iv. Evaporator temperature = 12°C
v. Heat source temperature = 105°C
vi. Refrigerant (water) out from the LT generator is at saturated state

4.2.2 Simulation of Basic System at Full Load (70kW)

Heat exchangers (HX) with external coolant loop were not taken into account in the main programme to make it simpler and easier for debugging. However, two solution heat exchangers (internal), HX-1 and HX-2 in the system are included in the programme, at the points where they interact with the operation of the system. The steady state internal temperatures and heat transfer rates with brine system of each main component of the system were considered for the main programme in modeling. These temperatures, heat transfer rates and other parameters such as flow rates of each component obtained from the main program were taken into account to determine complete design parameters of heat exchangers using a separate EES program. The following results at full load after many modeling trials with different parameter combinations were obtained. EES programme codes for this set of trails are given in Appendix-2 and details of the systems and parameters are given in Table-2A and 2B.

Table-7: Simulation Results of Basic System at Full Load

Variables in Main

| COP=1.231 | E_2=0.0007364 [kW] | P_l=1.599 [kPa] |
| E_1=0.008598 [kW] | P_h=48.67 [kPa] | P_m=5.323 [kPa] |
\[Q_a=81.41 \text{ [kW]} \quad Q_e=70 \text{ [kW]} \quad Q_{gl}=28.07 \text{ [kW]}\]

\[Q_c=45.44 \text{ [kW]} \quad Q_{gh}=56.85 \text{ [kW]}\]

[E1, E2 – Power of Solution Pumps; Pl, Pm & Ph – Absolute Pressure of Low, Medium and High pressure stages; Qa, Qc, Qe, Qgh and Qgl are rate of heat transferred in or expelled out to / from absorber, condenser, evaporator, HT generator and LT generator]

**Local variables TGen1 (Subprogram)**

\[h_e=144.3 \text{ [kJ/kg]} \quad m_{st}=0.01734 \text{ [kg/s]} \quad T_{in}=52.48 \text{ [C]}\]

\[h_{in}=108.1 \text{ [kJ/kg]} \quad P=5.323 \text{ [kPa]} \quad T_{out}=68.27 \text{ [C]}\]

\[h_{out}=143.8 \text{ [kJ/kg]} \quad Q_{vap}=18.37 \text{ [kW]} \quad \text{Units}=2\]

\[h_{st}=2627 \text{ [kJ/kg]} \quad Q_{gl}=28.07 \text{ [kW]} \quad x_{in}=46.17\]

\[m_{in}=0.268 \text{ [kg/s]} \quad Q_{sen}=9.703 \text{ [kW]} \quad x_{out}=49.37\]

\[m_{out}=0.2506 \text{ [kg/s]} \quad T_e=68.27 \text{ [C]}\]

The above parameters are shown in schematic diagram of Figure-15
Figure 15: Configuration of Basic LIBr-H2O Absorption Chiller
4.2.3 Simulation of Modified System at Full (70kW)

The above configuration of the double-effect absorption chiller is modified to add part of the rejected heat into the system by altering system temperatures. Heat input by the third heat exchanger (external), HX-3 that inputs part of waste heat into the system is obtained from modeling.

The following conditions were set in addition to the common input parameters given above.

Temperature of refrigerant leaving HT generator = 95ºC
(The above determines the temperature in and out solutions of HT Generator and hence that of heat source temperature)

LT Condenser temperature, t[4] = 38ºC

Temperature at the outlet of HX-3 = 32ºC

Temperature of the evaporator = 14ºC
(The above evaporator temperature gives the best performance in modeling trail. Since the expected indoor temperature is 24ºC according Sustainable Energy Authority (12) this would not be a drawback)

<table>
<thead>
<tr>
<th>Variables in Main</th>
<th>Q_a=84.03 [kW]</th>
<th>Q_c=44.94 [kW]</th>
<th>Q_e=70 [kW]</th>
</tr>
</thead>
<tbody>
<tr>
<td>COP=1.303</td>
<td>Pm=6.63 [kPa]</td>
<td>Q_{gh}=53.71 [kW]</td>
<td>Q_{hx3}=5.261 [kW]</td>
</tr>
<tr>
<td>E_1=0.01277 [kW]</td>
<td>Q_{gl}=28.44 [kW]</td>
<td>Q_e=70 [kW]</td>
<td>E_2=0.001518 [kW]</td>
</tr>
<tr>
<td>Ph=48.67 [kPa]</td>
<td>Q_e=44.94 [kW]</td>
<td>Q_{hx3}=5.261 [kW]</td>
<td></td>
</tr>
<tr>
<td>Pl=1.599 [kPa]</td>
<td>Q_e=70 [kW]</td>
<td>Q_{hx3}=5.261 [kW]</td>
<td></td>
</tr>
</tbody>
</table>

[E1, E2 – Power of Solution Pumps;  Pl, Pm & Ph – Absolute Pressure of Low, Medium and High pressure stages;  Qa, Qc, Qe, Qgh, Qgl and Qhx3 are rate of heat transferred in or expelled out to / from absorber, condenser, evaporator, HT generator, LT generator and Heat exchanger HX-3]
Figure 16: Configuration of Modified LiBr-H2O Absorption Chiller
Local variables in LTGen (Subroutine)

- \( h_{\text{e}} = 133.7 \) [kJ/kg]
- \( h_{\text{in}} = 115.7 \) [kJ/kg]
- \( h_{\text{out}} = 132.7 \) [kJ/kg]
- \( h_{\text{st}} = 2618 \) [kJ/kg]
- \( m_{\text{in}} = 0.4178 \) [kg/s]
- \( m_{\text{out}} = 0.4004 \) [kg/s]
- \( m_{\text{st}} = 0.01739 \) [kg/s]
- \( P = 6.63 \) [kPa]
- \( Q_{\text{vap}} = 20.95 \) [kW]
- \( Q_{\text{gl}} = 28.44 \) [kW]
- \( Q_{\text{sen}} = 7.49 \) [kW]
- \( T_{\text{e}} = 63.38 \) [C]
- \( T_{\text{in}} = 55.64 \) [C]
- \( T_{\text{out}} = 63.38 \) [C]
- Units = 2
- \( x_{\text{in}} = 45.22 \)
- \( x_{\text{out}} = 47.19 \)

The above subscripts have the following designations:
- \( e \) – Properties of equilibrium state in LT generator;
- \( \text{in} \) – Properties at inlet;
- \( \text{out} \) – Properties at outlet;
- \( \text{st} \) – Properties of steam generated in LT generator;
- \( Q_{\text{gl}} \) - Rate of heat supplied to LT generator;
- \( Q_{\text{sen}} \) – Rate of sensible heat consumed;
- \( Q_{\text{vap}} \) – Rate of vaporization heat consumed.

Complete solution that includes other parameters of the system and EES program listing is given in Appendix-3.

### 4.2.4 Comparison of Results of Basic and Modified Systems at Full Load

The results obtained from simulation of two systems are shown graphically in the following charts. Power transfer in different units of the system (generator, absorber, condenser and evaporator), pressure in three different levels of the system and system pump power with COP are shown in three charts as one chart would not show distinction between them due to wide range of values.

![Figure 17: Comparison of Heat Transfer in Different Units](image)

As seen from the chart, the difference of total power in and out of the four heat exchanging units is met by the power feedback heat exchanger, HX-3. However, total heat rejection of the modified system has slightly increased (2.12kW) above that of the basic system.
Figure 18: Comparison of Pressure Levels in two systems

There is no significant change in pressure levels of two systems as shown in Figure-18. This is due to selection of saturation temperatures are almost equal. Hence, there is no requirement for changing the design and constructional details of the modified system.

Figure 19: Comparison of COP and System Pump Ratings

Power ratings of solution pumps had increased in the modified system compared to that of the basic system. However, since the values of these electric pumps are very small and they also have speed control to match the cooling load thereby optimizing usage of electrical energy.
The above comparison shows that the modified system has slight improvement in efficiency with corresponding decrease in input power. However, overall heat rejection rate (that of LT condenser and the absorber has slightly increased there by requiring a slightly large cooling tower. It is also observed during different trials of simulation that in order to feed more power it is required to increase the temperature of the condenser that in turn affects the efficiency.

Table – 8: Simulation Results of Modified System at Half-Load Half load (35kW)

<table>
<thead>
<tr>
<th>Variables in Main</th>
<th>Q_e=35 [kW]</th>
<th>Q_gh=26.77 [kW]</th>
</tr>
</thead>
<tbody>
<tr>
<td>COP=1.308</td>
<td>Pl=1.599 [kPa]</td>
<td></td>
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<tr>
<td>E_1=0.008917 [kW]</td>
<td>Pm=5.945 [kPa]</td>
<td></td>
</tr>
<tr>
<td>E_2=0.009015 [kW]</td>
<td>Q_a=45.72 [kW]</td>
<td></td>
</tr>
<tr>
<td>Ph=48.67 [kPa]</td>
<td>Q_c=18.53 [kW]</td>
<td></td>
</tr>
<tr>
<td>Q_e=35 [kW]</td>
<td>Q_hx3=2.483 [kW]</td>
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</tr>
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</table>

<table>
<thead>
<tr>
<th>Local variables in LTGen1 (Subroutine)</th>
<th></th>
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</thead>
<tbody>
<tr>
<td>h_e=138.9 [kJ/kg]</td>
<td>m_st=0.006903 [kg/s]</td>
</tr>
<tr>
<td>h_in=113 [kJ/kg]</td>
<td>P=5.945 [kPa]</td>
</tr>
<tr>
<td>h_out=138.4 [kJ/kg]</td>
<td>Qvap=10.72 [kW]</td>
</tr>
<tr>
<td>h_st=2623 [kJ/kg]</td>
<td>Q_gl=18.28 [kW]</td>
</tr>
<tr>
<td>m_in=0.2922 [kg/s]</td>
<td>Q_sen=7.559 [kW]</td>
</tr>
<tr>
<td>m_out=0.2853 [kg/s]</td>
<td>T_e=65.9 [°C]</td>
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</tbody>
</table>

4.2.5 Observations on Half Load Trial

a) The same heat source with temperature of 95°C, but the rate of heat input to HT generator is reduced nearly to half of that used for the full-load. Solution flow rate was reduced to nearly 70% of the rate used with full-load.

c) COP remains nearly unchanged.

d) Heat transfer rates of the absorber, LT condenser, LT generator and HX-3 are reduced by percentages in the range of 41% to 64%.

e) Refrigerant temperature is 14°C, which is higher than standard chillers (in the range of 5°C to 8°C) due to higher condenser temperature that is required to utilize part of the condenser heat (Figure-16). However, this is not a drawback to achieve the required minimum indoor temperature of 23.5°C (Section 2.1.2)

The complete solution with parameters is given in Section 3B of Appendix-3. Program listing is identical to that of the full-load program and only parameter values of Qe, m[7] and t[4] were changed trial and error to achieve the stable results.
4.3 Setting Ratings for Chiller Components

Having established the operating parameters after many trials with various combinations of input parameters, it was possible to set ratings for the system components of which properties have already been discussed. As described earlier, every component of the chiller contains a heat exchanger, except circulation and mixing pumps. Hence the performance of an absorption chiller almost entirely depends on effectiveness of heat exchangers.

4.3.1 Heat Exchangers

Commercially available absorption chiller systems of large capacities have purpose-built heat exchangers all of them are included in a common module, but this type of fabrication would be labour intensive and costly for the fabrication of a single unit. Hence the trial system in the project uses commercially available heat exchangers with suitable interconnections, except for the absorber and evaporator that intend to transfer the highest amount of heat. The absorber design is different from other heat exchangers, as it needs to perform three important functions that decide overall performance of the system as follows

- Large amount heat rejection from the absorber.
- Absorption of steam (refrigerant) received from the evaporator.
- Efficient mixing of refrigerant and weak mixture flows from the L.T. generator

In order for efficient transfer of steam from the evaporator to the absorber it would be preferable to contain both evaporator and absorber in one shell with an insulated partition between them and an opening for vapour transfer.

Leakages in components and connecting pipes of an absorption chiller would not be so difficult to overcome, compared to that of conventional type systems, as the pressure difference between connected components and between each of them and ambient would not be more than 1.0 bar. The unit that has the highest pressure in the proposed system is hot water storage unit, but it is outside the main system and needs to maintain about 1.3 bars as described below. Although the maximum temperature (at the exit of HT generator) is around 101°C, it is proposed to have the storage temperature at 110°C to allow for losses and temperature variations in the system. The correct temperature can be fed in with the use of mixing with return (cooled off) water (brine).

The details of pumps and control valves in each part of the chiller are described in Section 4.3.2. The parameters of each heat exchanger have been obtained from separate EES programme (Appendix-4) with the required input parameters obtained from the main modeling programme (Appendix-3). However, the design parameters of two heat exchangers HX-1 and HX-2 and that of the system solution pump are obtained from the main programme where their analysis is included. The results and the EES programme equations of the other heat exchangers are given in Appendix-4.

a) Heat Exchanger for High Temperature Generator

Heat input for the chiller is provided by this heat exchanger with necessary temperature and flow control for the inlet hot water. The specifications required according to the system analysis are given below.

i. **Input heat energy transfer rate**, \( Q_{gh} = 53.7 \text{ kW} \)

ii. **Source (hot water in) temperature**, \( t_{hi} = 110^\circ \text{C} \)
iii. Outlet (hot water) temperature, \( t_{ho} = 102^\circ\text{C} \)

iv. Flow rate in heat source line, \( m_{hw} = 1.59 \text{ kg/sec} \)

v. Pressure in heat source line (in tubes), \( P_{gh} = 129.7 \text{kPa} \) (saturated water)

vi. Pressure in the shell (solution), \( P_s = 48.7 \text{kPa} \)

vii. Mean Temp. Difference LMTDg = 3.6\(^\circ\text{C}\)

viii. UA value, \( U_{ag} = 15.32 \text{kW/}^\circ\text{K} \)

ix. Correction Factor, \( F_g = 0.97 \) Description: Shell and Tube type heat exchanger made of copper tubes and steel shell with single-pass tube system and a vent for steam to the HT condenser is suggested for this application. Hot water inlet is connected to the tube bundle and the solution is circulated in the shell. The vent is provided to eject steam (similar to an ejector outlet in process control heat exchangers) to the next stage (LT generator / HT condenser). Solution enters from the top of the shell and hot water also enters from the upper inlet of the tube bundle so that LiBr saturation increases when solution flows to the bottom of the shell, where the outlet directs solution to the LT generator through HX-1.

b) Low Temperature Generator & High Temperature Condenser

This unit is a combination of two functional units – HT Condenser and LT Generator (Figure-16). There is no external heat transfer to this unit and is effective in boosting efficiency of the system. The results of the main program (modeling) show that the temperature of outlet solution in LT generator and refrigerant out of HT Condenser are same.

i. Input heat transfer rate, \( Q_{gl} = 28.44 \text{kW} \)

ii. Inlet refrigerant temperature, \( t_1 = 95^\circ\text{C} \) (steam into HT Condenser)

iii. Outlet refrigerant temperature, \( t_2 = 80.7^\circ\text{C} \)

iv. Inlet solution temperature, \( t_{lgi} = 55.6^\circ\text{C} \)

v. Outlet solution temperature, \( t_{lgo} = 63.4^\circ\text{C} \)

vi. Pressure in the shell (solution), \( P_s = 48.7 \text{kPa} \)

vii. Refrigerant inflow rate, \( m_{[19]} = 0.0174 \text{ kg/sec} \) (superheated steam)

viii. Solution inflow rate, \( m_{61} = 0.418 \text{ kg/sec} \)

ix. Mean Temp. Difference LMTDgl = 26.8\(^\circ\text{C}\)

x. UA value, \( U_{agl} = 1.373 \text{kW/}^\circ\text{K} \)

xi. Correction Factor, \( F_{gl} = 0.77 \)

xii. Description: Shell and tube type is used here too, to handle large rate of heat transfer with refrigerant (steam) in the tubes. Construction of this heat exchanger is also similar to that of HT Generator HX, but of smaller size, as its heat transfer rate is less (28.44 kW).

c) Low Temperature Condenser

4.2 Heat rejection rate, \( Q_c = 44.94 \text{kW} \)

  ii. Brine water inlet temperature, \( T_{ci} = 31^\circ\text{C} \) (Program returns a value of 30.96\(^\circ\text{C}\))

4.3 Brine outlet temperature, \( T_{ci,o} = 36^\circ\text{C} \)

4.4 Condenser inlet refrigerant temps., \( t_{[3]} \) & \( t_{[19]} = 38^\circ\text{C} \) & 63.4\(^\circ\text{C}\)

  v. Condenser outlet refrigerant temp., \( t_{[4]} = 38^\circ\text{C} \)

  vi. Condition of outlet refrigerant = Saturated liquid (water)

  vii. Pressure in tubes (refrigerant), \( P_m = 6.6\text{kPa} \)

  viii. Refrigerant outflow rate, \( m_{[4]} = 0.0296 \text{ kg/sec} \) (saturated water)

  ix. Approx. brine flow rate, \( = 1.783 \text{ kg/s} \)

  x. Mean Temp. Difference LMTDc = 4.33\(^\circ\text{C}\)
xi. UA value, Uac = 10.38 kW/°K
xii. Correction Factor, Fe = 1
xiii. Description: Shell & Tube type heat exchanger would be suitable for this application too. Refrigerant flows in the tubes with combining T-connection for two refrigerant flows (from LT generator and from HT condenser) at the inlet to the condenser. Copper tubes and steel shell can be used in this low temperature and pressure HX.

d) Evaporator

i. Heat absorption rate, Qe = 70 kW

4.5 Evaporation temperature, = 14°C

iii. Inlet refrigerant temperature, t[6] = 14°C (after throttling)
iv. Condition of inlet refrigerant, q[5] = 0.0407 (95.9% liquid)
v. Outlet temperature of refrigerant = 14°C
vi. Pressure of refrigerant (in shell), P = 1.61kPa

4.6 Condition of outlet refrigerant = Saturated vapour

viii. Refrigerant flow rate = 0.0296 kg/sec (saturated steam)
ix. Chilled water out temperature = 15°C
x. Chilled water in maximum temperature = 24°C (set by the load, considered)
xi. Chiller water (brine) flow rate = 1.859 kg/s
xii. Mean Temp. Difference, LMTDe = 3.91°C
xiii. UA value, Uae = 17.91 kW/°K
xiv. Correction Factor, Fe = 1.0

Description: This HX occupies the same shell with the absorber for efficient mixing of steam (refrigerant) with weak solution as described in next item. In order to cater for high rate of heat transfer, 2-pass shell and tube made of copper tubes due to the use of LiBr-H2O solution and steel shell HX would be suitable. A motorized refrigerant circulation pump with a VSD to adjust flow rate thus achieving energy saving is suggested.

c) Absorber

4.7 Heat rejection rate = 84.03 kW

ii. Inlet refrigerant temperature, t[6] = 14°C (from Absorber evaporator)
iii. Condition of inlet refrigerant = Saturated vapour (steam)
iv. Temp. of LiBr solution entering, t[18] = 30.4°C
vi. Heat sink (brine inlet) temperature, Tbi = 25.5°C (Outlet temperature of evaporative cooler (cooling tower))

v. Brine outlet temp., Tabo = 29°C (inlet brine temp. of condenser)
vi. Brine flow rate, mb = 5.716 kg/sec
vii. Mean Temp. Difference, LMTDa = 1.35°C
viii. UA value, Uaa = 63.41 kW/°K
ix. Correction Factor, Fe = 0.982

Description: Special arrangement is required for mixing of refrigerant and solution at the controlled temperature. It is proposed to contain absorber and evaporator in a single shell, as proposed in Section 2.3, in order to achieve compactness and losses. Tube sets for Shell & Tube type HXs are available in the market. Two tube sets to match parameters of Sections 4.5.2 d) and e) will be used for this purpose.
A possible arrangement of Absorber and Evaporator combination in one container is shown in Figure-17. Two circulation pumps make the evaporation process in the evaporator and cooling and mixing processes in the absorber more efficient. The spray system connected to the circulation pump in the absorber creates a slight pressure drop at the entry to the absorber, thus making steam flow from evaporator towards the absorber effective.

**Figure 20: Absorber and Evaporator arrangement**

f) HX-1 (between HT generator & LT Generator / HX-2)

i. Heat transfer rate, \( Q_{x2} = 25.93 \) kW

ii. HT side inlet temperature, \( t[13] = 101.3^\circ C \)

iii. HT side outlet temperature, \( t[14] = 76.3^\circ C \)

iv. HT side flow rate, \( m[13] = 0.418 \) kg/s

v. LT side inlet temperature, \( T[11] = 53.0^\circ C \)

vi. LT side outlet temperature, \( t[12] = 99.6^\circ C \)

vii. LT side flow rate = 0.43 kg/s

viii. UA value, \( UA1 = 3.725 \) kW/°K

ix. Description: Small Shell & Tube heat exchanger would be adequate for this purpose due to small flow rate and moderate heat transfer rate.
g) HX-2 (between Absorber / HX-3 & HX-1 / LT Generator)
4.8 Heat transfer rate = 21.63 kW
   ii. HT side inlet temperature, t[16] = 63.4°C
4.9 HT side outlet temperature, t[17] = 28.8°C
   iv. HT side flow rate = 0.040 kg/s
4.10 LT side inlet temperature = 32.0°C
   vi. LT side outlet temperature = 53.0°C
   vii. LT side flow rate = 0.043 kg/s
   viii. UA value, UA2 = 2.91 kW/°K
   x. Description: Small Shell & Tube heat exchanger would be adequate for this purpose due to small flow rate and moderate heat transfer rate. As in the case of HX-2

h) HX-3 (between Absorber & SP-2 / HX-2)
4.11 Heat transfer rate, Qx3 = 5.26 kW
   ii. HT side (brine) inlet temperature, Tx3bi = 36°C
4.12 HT side (brine) outlet temperature, Tx3bo = 34°C
   iv. HT side flow rate, mbx3 = 0.626 kg/s
4.13 LT side inlet (from Absorber) temperature, t[7] = 26.8 °C
   vi. LT side outlet temperature, t[8] = 32°C
   vii. LT side flow rate, m[7] = 0.43 kg/s
4.14 UA value, Uax3 = 1.124 kW/°K
   xi. Description: Miniature Shell & Tube heat exchanger would be adequate for this purpose due to small flow rate and heat transfer rate.

4.3.2 Pumps and Control Valves
The system needs six pumps for the operation of the chiller alone, excluding chilled water circulation, as follows.
   a) Main system pumps (SP-1 and SP-2)
   b) Heat source pump
   c) Solar heater circulation pump
   d) Absorber recirculation pump
   e) Evaporator recirculation pump
A brief description of operation of each pump and specifications of them are given below to enable to select suitable pumps. (26)

Pump power will be calculated according to the formula

\[ P_p = \left( \frac{V_{\text{dot}} \times \Delta p}{\eta} \right) \times (1+s) \]

Where, \( P_p \) = Power rating of the motor of the required pump (W) (Main system pump ratings are given by the modeling program)

\( V_{\text{dot}} \) = Volumetric flow rate (m³/s)
\( \Delta p \) = Total pressure head requirement (head + frictional loss)
\( \eta \) = Total pump efficiency (0.55 - 0.85)
s = The factor that determines the required motor power (allowance for overloading),
depending on the pump power (consider 0.5 for fractional horsepower and 0.6 for 1 – 2
kW pumps)

a) System Main Pumps
There are two pumps SP-1 and SP-2, of which ratings are 12.8W and 1.5W respectively according to the
modeling. Assuming an efficiency of 75% for both pumps, which is the normal value for small pumps, the
actual power ratings of the pumps would be 17.1W and 2W.

Power rating of SP-1, as described above = 17.1 x (1.5) = 25.6W
Power rating of SP-2 = 2 x 1.5 = 3W
Small solution pumps used for chemical processes can be adopted with variable speed control to
regulate the speed for SP-1, as the motor runs throughout the operation. SP-2 has a very small capacity
and a small DC powered pump would be suitable for this application.

b) Heat Source Pump
This pump carries out an important role of energy saving, by regulating hot water supply to match the
exact heat demand. Hence it is controlled by a variable speed drive to supply correct amount of water
in coordination with the controlled valves MV-1 and MV-2. PLC maintains the input temperature at
the design values of 110°C by measuring temperature of tank outlet water (Tₜₒ) and that of HT
Generator (T[12] in modeling). Motor is set to run at the design flow rate (at full load) but reduces the
rate if the evaporator temperature drops below the set value. MV-1 and MV-2 is controlled to minimize
the use of hot water supply by mixing with returned water (at nearly 100°C), when the temperature is
above the set value.

Full-load design parameters of the pumps are as follows.

Design flow rate = 0.00222 m³/s = 2.22 l/s
Since the chiller and the storage tank have no considerable height difference, the pressure difference is
the frictional head only. The considered heat exchanger is a double-pass, shell & tube type with copper
tubes of 48mm dia. (considering a worst case of single-pass tube) that has a frictional loss of 0.09m/m
(from charts for copper tubes) or 882 Pa/m. The pipe line between water storage and HT generator
are assumed to have a maximum of 10m distance including HX connecting pipe length. Hence the
pressure drop would be 8820Pa plus losses at bends and fittings (say 1000 x dia. Maximum and 48mm
dia.).

The power rating of the pump, at 60% efficiency = [0.00222 x (8820 + 4.8x882)/0.6] x 1.5 = 72.4 W
We would select a 100W, single-phase fractional horsepower with a single-phase VSD.
This power rating is the peak value and the actual wattage depends on the required flow rate that
depends on the chiller load.

c) Solar heater circulation pump
The flow rate of this pump depends on the energy absorbed by each collector (area of 72.2/3 m² per
collector, according to Section 4.5.3) that in turn depends on the average insolation and period of heat
absorbed. Temperature rise across the absorber tube varies according to the heat intensity (radiation)
and the flow rate. Hence for calculation purpose, we assume that the solar heater circulation pump also
delivers the same maximum flow rate as the heat source pump. However, the total pipe length is more than the collector length due to spacing of collectors and distance to the storage. Collectors have a length of 12m and the maximum pipe length could be considered as 20m so that the total effective pipe length of three parallel collectors and a height difference of 3m between the solar collectors and HT Generator to allow chiller at a lower level in case of space problem. Copper pipes with diameter of 48mm will be used as in the previous case.

Head loss due to static head and pipe friction

\[ 882 \times (23 + 12/3) + 3000 \times 9.8 = 52.92 \text{kPa} \]

(1mm of water = 9.8 Pa)

Considering the same amount of drop in obstacles as in the previous case, we have 4.23 (=4.8 x 882) kPa loss.

Hence the total head required = 52.92+4.23 = 52.92kPa

The required temperature difference in HT Generator (Section 4.5.1, Item a) is 8°C and the heat energy rate is 53.7kW. The solar collector will be designed to achieve the same temperature difference and flow rate through the solar collectors, at 90% efficiency. Then the flow rate through the collectors (total of 3 parallel collectors) would be calculated using the formula,

\[ \dot{m} = \frac{Q_{s}}{(\Delta t \times C_{p})} = \frac{53700}{(8 \times 4250)} = 1.58 \text{ kg/s} \]

Hence water flow rate in solar collector line = 1.58 / 938 = 0.001684

Power rating, as for Heat Source Pump = [0.001684 x 52920 /0.6] x 1.5

(Efficiency = 60% and s =0.5)

\[ = 223 \text{W} \]

A standard pump with motor rating 300W, 1-single phase (230V) would be selected for this application.

d) Absorber Circulation Pump

This is a smaller pump similar to solution pumps used for chemical mixing (e.g. swimming pool). The minimum flow rate is set equal to incoming solution flow to the absorber (m[18]) for better mixing. The maximum flow rate (m[18] from modeling) is 0.43 kg/s or 0.302 l/s (density of the solution is 1425 kg/m³). Since the recirculation pipes are short (max. 1 m), the pressure difference would be 1m head + frictional loss that is well below 20W. Hence, a pump of 30W rating with adjustable flow rate is used.

e) Evaporator circulation pump

The function of this pump is also same as in the above case with the only difference that liquid is water, instead of LiBr-H₂O solution. Hence, adjustable pump used for the Item d) can be used in this case too.

f) Motorized Valves

There are four motorized through valves (MV-1 to MV-4) and one motorized mixing valve is shown in Diagram-10. The required specifications of them are as follows
The above specifications are the maximum ratings above the calculated or anticipated values.

They will be controlled by two PLC (or DDC) to achieve the described functions. These motorized valves are preferably low voltage operated (24V DC) that will be fed through the PLC to make the system less complicated.

g) Manual Throttle Valves & Pressure Relief Valves

In order to tune the system to obtain required operational parameters four manual throttle valves (TV-1 to TV-4) and three non-return valves (NV-1 to NV-3) are provided. There is one pressure relief valve, RV-1 on the hot water (storage) tank to release excess pressure from the tank, under any extreme condition for safety of the system and one 3-way mixing valve to feed make up water into the solar heating system.

h) Cooling Towers (Evaporative Condensers)

Combined brine system has to expel heat to the environment through a commercially available cooling tower. The capacity would be approximately the total heat rejection from the absorber and condenser, as that is absorbed by HX-3 is negligible and a rating of 130 kW with inlet temperature of 31°C (precisely 30.96°C) and outlet temperature of 25.5°C would be suitable, as required by the LT condenser and the absorber. It is placed in the same level as the chiller. Manufacturer’s specifications suggest (KS Model KST N-10 with over capacity for additional temperature drop) fan motor is 187.5 kW (0.25 hp) and circulation pump with design pump head of 1.7m would be rated at 100W. Hence the total motor power would be less than 300W (installed rating).

i) Chilled Water Circulation System

The head of water circulation is approximately 5m as the building has two floors only and the maximum length of the chilled water line is approximately 36m (72m pipe length). The chilled water flow rate at 9°C temperature difference in evaporator is approximately 1.86 l/s with 48mm dia. Pipe and the flow speed is 1.03 m/s. Basic analysis for the chilled water circulation system suggest approximately 700W motor using the method described in the beginning of the section. Hence a 750W motors is proposed for the chilled water circulation system.

J) Total Motor Load

A total of seven motors are used in the solar powered absorption chiller system a discussed above, The selected pumps are single phase fractional horsepower induction motors due to their ruggedness for continuous operation. The rating of each motor was set to the next standard rating available above the calculated motor power. A summary of the calculated and commercially available ratings for each application is given below.
### Table-10: Actual Pump Power and Pump Motor rating

<table>
<thead>
<tr>
<th>Function of the Motor</th>
<th>Calculated Power Consumption (W)</th>
<th>Rating of the Proposed Motor (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>System Main Pumps (SP-1 &amp; -2)</td>
<td>28.6 (25.6 + 3)</td>
<td>30 &amp; 10</td>
</tr>
<tr>
<td>Heat Source Pump</td>
<td>72.4</td>
<td>100</td>
</tr>
<tr>
<td>Solar Heater Circulation Pump</td>
<td>223</td>
<td>300</td>
</tr>
<tr>
<td>Evaporator Circulation Pump</td>
<td>20</td>
<td>30</td>
</tr>
<tr>
<td>Absorber Circulation Pump</td>
<td>20</td>
<td>30</td>
</tr>
<tr>
<td>Cooling Tower Fan motor</td>
<td>187.5</td>
<td>187.5</td>
</tr>
<tr>
<td>Cooling Tower Circulation Pump</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>Chiller Water circulation Pump</td>
<td>700</td>
<td>750</td>
</tr>
<tr>
<td><strong>TOTAL CONSUMPTION</strong></td>
<td><strong>1,351.5</strong></td>
<td></td>
</tr>
</tbody>
</table>

#### 4.3.3 Solar Heater and Energy Storage System

The necessary theory and formulae of calculation for dimensioning of the solar collector and storage system was discussed in Sections 3.3 and 3.4. In order to size these components of the system, parameters from the final modeling have been used. The results of the final modeling trial on full-load stated above will be used for sizing the solar energy collector and storage system.

Equation 3.6c provides the method of evaluating the size of required CSP aperture area, as given below.

\[
A_a = 1.345 \times \frac{Q_{gh}}{m^2}
\]

The value of the heat input for HT generator, \( Q_{gh} = 53.7 \, \text{kW} \)

Thus aperture of the CSP, \( A_a = 1.345 \times 53.7 \, m^2 \)

\[
= 72.2 \, m^2
\]

It may not be possible to fabricate a single unit of the above size that could be installed on a roof. Probably three units of 24 m² would be more practical solution. The size of each would be 2 m x 12 m and are oriented along N-S line parallel to the earth axis. Building-A of the office has a roof area with dimensions 160’ x 43’ (48 m x 13 m) that is adaptable for installation of solar collectors that needs maximum area of 12 m x 8 m space including inter-panel clearance of 1 m.

Having decided the size of solar energy collector, the next step is to determine the size of the solar energy tank for hot water storage that stores hot water above the minimum desired temperature. The methodology, assumptions and limitations of the storage were discussed in Section 3.7. Equation 3.7d provides the method of determining the volume of the solar storage by substituting the values of the following parameters.

Input power to the HT generator, \( Q_{gh} = 53.7 \, \text{kW} \)

Heat source (temperature of inlet to heat exchanger), \( T_{in} = 110 \, ^\circ \text{C} \)

(Reference to Section 4.3.1 Item a)
Temperature of the HT generator, more precisely the temperature of return water, $T_{ho} = 102^\circ$C.

The period allowed daily for absence of solar energy, $\tau_a = 2$ hrs.

From Equation 3.4.d,

Volume of the storage $V_s = 0.903 \times 53.7 \times 2 / (110 - 102)$

$= 12.12 \text{ m}^3$

The size of the storage would be approximately one vessel of 2.0m in diameter and 3.858m in length that can be easily accommodated on a roof slab. Alternatively, two interconnected vessels of 1.0m in diameter and 3.858 in length can be used if load of a single unit is a problem for the structure. Additional heating during weekends would enhance the energy storage capability and is not accounted in the above analysis, but can be separately evaluated using Equation 3.7g, keeping allowances for heat loss. Moreover, heat losses through the insulated surface can be further reduced by suppressing natural air draft over the tank surface.

Recalling Equation 3.7g, the excess energy accumulated during non-working days

$38.5 \times Aa - 40.7 \times Qg1 \text{ kWh}$

Hence, the actual excess energy available as per the above equation $= 38.5 \times 72.2 - 40.7 \times 53.7$

$= 594.1 \text{ kWh}$

However, the above calculations are based on the assumptions of solar power absence and chiller operational period and loads. The accumulation of excess energy would increase the storage temperature above the favourable temperature of the system, which is set below 110°C to prevent harmful corrosive effects. Hence it would be required to purge hot water to the discharge tank, if the temperature goes above the limit and deflect the solar collectors to cut off heating process.

### 4.3.4 System Integration and Control

The absorption chiller, solar collector, heat energy storage, cooling tower and control system need to be combined to have the expected performance of the chiller. A brief technical description of each component has been discussed above and integration of them has to achieve the desired results. The energy input, heat expelled, chilled water circulation, etc. have to be coordinated with the load and ambient conditions. This will be achieved by a comprehensive Direct Digital Control (DDC) system that uses programmable logic controller (PLC) and indecently controllable VSD. The final configuration of the developed chiller system is shown in Figure-12.

The modeling process takes the integrated system into account in analyzing performance of the system, except the solar energy control and secondary coolant (brine) control. However, the modeling provides the required input information for the control system. The separate EES programme that is used for evaluation HX parameters provides key information including that of input energy and cooling tower, as described in Section 4.5.1 and in Appedix-4 (results of EES programme with programme equations).

Basic objective of overall system control using DDC is to monitor the system constantly and adjust the parameters to maintain the required load and temperature levels expected from the chiller (Figure-9, Section 3.2). In addition, they carry out safety functions so that system does not enter into certain extreme condition that could be detrimental. Measured parameters and controlled parameters of the system are interrelated to achieve the expected results. Recirculation pumps have small power rating and runs throughout operation
to make the absorber and evaporator functions efficient. A basic description of how the controllers monitor the system (Figure-10) and control is shown in Table-9.

### Table -I1: Control System Components, Sensed / Control Parameters

<table>
<thead>
<tr>
<th>Controlled Unit Description</th>
<th>Measured Parameter</th>
<th>Controlled Parameter</th>
<th>Operational Activity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hot Water feed pump &amp; MV-1</td>
<td>T1 &amp; Tgi (with transducer)</td>
<td>Input source temperature</td>
<td>Economizing usage of hot water supply by mixing with return water</td>
</tr>
<tr>
<td>Solar Collector Pumps</td>
<td>Tsi &amp; Tso (with transducers)</td>
<td>Temperature of water through solar collection system</td>
<td>Optimum usage of solar energy by setting correct level of collector fluid temperature</td>
</tr>
<tr>
<td>MX-1</td>
<td>Tank Level Sensor</td>
<td>Supplies make up water</td>
<td>A simple solenoid valve</td>
</tr>
<tr>
<td>PLC/MV-2, MV-3, Brine Pump</td>
<td>T5</td>
<td>Absorber temperature and brine flow rate</td>
<td>Maintains correct temperature of absorber</td>
</tr>
<tr>
<td>PLC/MV-2, MV-3, Brine Pump</td>
<td>T2</td>
<td>Condenser temperature and hence system efficiency</td>
<td>To maintain proper operation of condenser.</td>
</tr>
<tr>
<td>PLC / System Pumps, Hot Water Pump</td>
<td>T4</td>
<td>Evaporator temperature</td>
<td>Stabilizing chiller temperature at constant chilled water inlet, Tci.</td>
</tr>
<tr>
<td>PLC / Chilled Water Pump</td>
<td>T3 &amp; Tei</td>
<td>Temp. of inlet Chilled water</td>
<td>Changes energy input to the HT generator to match cooling load</td>
</tr>
<tr>
<td>PLC-2 / Aux. Heat source</td>
<td>Tg</td>
<td>Switches aux. heat source</td>
<td>When storage temp. falls below usable state.</td>
</tr>
<tr>
<td>RV-1</td>
<td>Pressure relief and shuts down the system</td>
<td>Reduces pressure in storage when reaches the threshold and diverts solar reflectors.</td>
<td>Rare situation of not using solar power when the storage is full. Tracking system prevents solar energy collection.</td>
</tr>
</tbody>
</table>
The manual valves TV-1 to TV-4 are provided for tuning of the system for obtaining expected design conditions at the start up. Set values for the automatic controls have to be decided during the tuning process so that the system runs at the pre-determined conditions. Thereafter, the DDC will take control of the system according to the algorithm of the control system and with the set input parameters.

4.3.5 Proposed Brands and Models for Components of Control System

Integration, tuning and maintenance of the control system that is vital for proper operation of the system needs quality components. The following table includes the proposed brands and models of control components.

Table 12: Proposed Makes and Models for Control System

<table>
<thead>
<tr>
<th>Component</th>
<th>Make</th>
<th>Model</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>PLC and HMI</td>
<td>Mitsubishi</td>
<td>FX 1S and FX1N-5DM</td>
<td>This PLC can accommodate up to 30 I/O signals with combination of analog &amp; digital signals.</td>
</tr>
<tr>
<td>AC drive for Chilled water circulation Pump</td>
<td>ABB</td>
<td>ACS 350</td>
<td>This drive can be independently of through PLC can be controlled. It has advanced programming ability with PDI control.</td>
</tr>
<tr>
<td>Motorized Valves Diverting type (MV-1, MV-2 &amp; MV-3)</td>
<td>Honeywell</td>
<td>Valve -V5013C</td>
<td>Includes only the valve with stem. Motor units has to be separately supplied</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Motor - M9174C1041</td>
<td></td>
</tr>
<tr>
<td>Motorized Valves Mixing type (MX-1)</td>
<td>Honeywell</td>
<td>Valve -V5013D</td>
<td>Same as above</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Motor - M9174C1041</td>
<td></td>
</tr>
<tr>
<td>Temperature sensors (T1 – T4, Tci, Tgi, Tsi &amp; Tso)</td>
<td>Honeywell</td>
<td>C7031</td>
<td>This sensor that uses NTC sensors with transducer is recommended to be used with DDC system of MVAC systems.</td>
</tr>
<tr>
<td>VSD for small motors (Sp-1/ Sp-2 and solution circulation pumps)</td>
<td>Ironhorse</td>
<td>GSD1</td>
<td>This DC drive can drive motors with rating from 15W to 750W</td>
</tr>
<tr>
<td>VSD for HWP, Solar Trough circulation, Cooling Tower circulation pumps and Cooling Tower fan</td>
<td>Ironhorse</td>
<td>GSD6</td>
<td>Single phase 230V input power and Dc output voltage.</td>
</tr>
</tbody>
</table>
4.3.6 Control System Errors, Methods of Correction and Maintenance

Many components in the control system could cause errors during operation under different operating environments. By choosing high quality components such errors could be minimized. Since the sensors used in the system are only temperature sensors of NTC type and with transducers. In general, errors in the control systems are caused by the sensors that are sensitive to dust and dirt. Hence possibility for errors is minimum in this system compared to that includes optical or proximity sensors.

Since the pumps and control valves are operated in closed loop control system with digital inputs, it is possible to feed numerical values for input directly and the output response. This would provide a convenient method of verifying error of any controlled components, thereby identifying faulty component in a control loop.

The system consists of many motors. Most of them are of small capacities (less than 100W) and are operated with both AC and DC power. Regular maintenance of motors that include lubrication, replacement of commutator brushes is important for trouble free operation.
5. Conclusions

The basic configuration of a double-effect LiBr-H2O chiller was modified in the project to achieve 100% solar energy for its operation. The added features had improved COP of the modified system over the basic version, thus reducing heat input and size of the solar collection system. However, the efficiency improvements are not very significant according to the simulation results. The simulation results given in the previous chapters and referenced annexes show improvements achieved in the modified version. The sizes of the solar collector and hot water storage are well adaptable within the footprint of the building thus indicating feasibility of solar powered absorption chiller in this type application.

5.1 Beyond this Research

The modeling based on the environmental parameters and indoor conditions could give results with a limited accuracy due to assumptions that certain losses are excluded. Hence, the real results could vary from the results obtained. However, system components used in the design are from the commercially available sources. It would be required to recalculate the system parameters using the actual physical parameters of the components to ascertain how well the simulation results fit in the actual scenario using a model of reduced scale. Such model would pave way to a prototype of the considered or higher capacity chiller that can be used in similar buildings.
Bibliography


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Appendix – 1

Energy simulation of Upper Floors with DesignBuilder

Figure – 1A : Energy Modeling Results of Upper Floor of Building-A
Figure 1B: Energy Modeling Results of Upper Floor of Building-C
APPENDIX – 2
EES Programme Listing & Solutions
for Standard Double-Effect Li-Br Absorption Chiller

Table-2A: Simulation Results of Basic Double Effect Chiller with EES at Full Load

Variables in Main

| COP=1.23 | E_1=0.008599 [kW] | E_2=0.0007752 [kW] |
| Ph=48.67 [kPa] | P_l=1.403 [kPa] | P_m=5.323 [kPa] |
| Q_a=81.41 [kW] | Q_c=45.48 [kW] | Q_e=70 [kW] |
| Q_gh=56.89 [kW] | Q_gl=28.15 [kW] |

Local variables in LTGen1 (Subprogram)

| h_e=144.3 [kJ/kg] | h_in=108.1 [kJ/kg] | h_out=143.8 [kJ/kg] |
| h_st=2627 [kJ/kg] | m_in=0.2679 [kg/s] | m_out=0.2506 [kg/s] |
| m_st=0.01735 [kg/s] | P=5.323 [kPa] | Q_vap=18.45 [kW] |
| Q_gl=28.15 [kW] | Q_sen=9.702 [kW] | T_e=68.27 [C] |
| T_in=52.48 [C] | T_out=68.27 [C] | Units=2 |
| x_in=46.17 | x_out=49.37 |

Table of Parameters Used in Program

<table>
<thead>
<tr>
<th>[i]</th>
<th>t[i]</th>
<th>h[i]</th>
<th>m[i]</th>
<th>q[i]</th>
<th>rho[i]</th>
<th>x[i]</th>
</tr>
</thead>
<tbody>
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<td></td>
<td>[C]</td>
<td>[kJ/kg]</td>
<td>[kg/s]</td>
<td>[kJ/kg]</td>
<td>[kg/m^3]</td>
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<tr>
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<td>48.17</td>
<td>102.9</td>
<td>0.28</td>
<td></td>
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</tr>
<tr>
<td>12</td>
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<td></td>
<td>0.4259</td>
<td></td>
</tr>
<tr>
<td>13</td>
<td>101.6</td>
<td>228.6</td>
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<td></td>
<td>0.4451</td>
<td></td>
</tr>
</tbody>
</table>
PROGRAM LISTING FOR STANDARD DOUBLE EFFECT CHILLER

{LiBr Absorption Chiller Modeling - Series Flow Double Effect Chiller}

{Suprogram to evaluate properties of LT Generator}

SUBPROGRAM LTGen1 (m_in, m_out, P, Q_gl, T_in, x_in, h_out, h_st, T_out, T_e, x_out)
Units=2
(Mass balance)

\[ m_{st} = m_{in} - m_{out} \]

{Equilibrium temperature at P, x_in}

\[ T_e = T_{LIBR}(Units, P, X_{in}) \]

\[ h_e = H_{LIBR}(T_e, x_{in}, Units) \]  \{Enthalpy at T_st, x_in\}

{Properties at inlet - subcooled state}

\[ h_{in} = H_{LIBR}(T_{in}, X_{in}, Units) \]

{Enthalpy at outlet - saturated state}

\[ h_{out} = H_{LIBR}(T_{out}, X_{out}, Units) \]

{Properties of vapor (refrigerant)}

\[ h_{st} = \text{enthalpy}(\text{WATER}, T=T_e, P=P) \]

{LiBr mass balance}

\[ m_{in} \times x_{in} = m_{out} \times x_{out} \]

{Solution & vapour heat transfer}

\[ Q_{sen} = m_{in} \times (h_e - h_{in}) \]  \{Sensible heating of inlet stream\}

\[ Q_{vap} = Q_{gl} - Q_{sen} \]  \{Latent heat\}

END

{Parameter settings to suit controlled conditions}

\[ q[2] = 0 \]  \{Vapour quality\}

\[ q[4] = 0 \]

\[ q[6] = 1 \]

{Temp. settings}

\[ t[6] = 10[\text{C}] \]

\[ t[1] = 95[\text{C}] \]

\[ t[4] = 34[\text{C}] \]

\[ t[8] = 38[\text{C}] \]

"System parameters"

\[ Q_e = 70[\text{kW}] \]

\[ m[8] = 0.28[\text{kg/s}] \]  \"0.26[\text{kg/s}]"  \"Steam requirment is 70/2500=.028[\text{kg/s}]\, steam; sat. solution ratio =6.5"

\[ \text{COP} = \frac{Q_e}{Q_{gh}} \]

\[ h[1] = \text{Enthalpy}(\text{Steam}, T=t[1], P=P) \]
\[ h[13] = h_\text{LiBrH}_2\text{O}(t[13], x[13]) \]
\[ x[13] = x_\text{LiBrH}_2\text{O}(t[13], P) \]
\[ t[12] = T_\text{LiBrH}_2\text{O}(P, x[12]) \]


\[ h[2] = \text{Enthalpy}(\text{Steam}, T=t[2], x=q[2]) \]
\[ t[2] = \text{Temperature}(\text{Steam}, P=P, x=q[2]) \]
\[ P = \text{Pressure}(\text{Steam}, T=t[2], x=q[2]) \]
\[ m[1] = m[2] \]


\[ "h[14] = h_\text{LiBrH}_2\text{O}(t[14], x[14])" \]

\[ m[13] = m[14] \]

\[ \text{CALL Q}_\text{LiBrH}_2\text{O}(h[14], Pm, x[14]: q[15], t[15], x[15]) \]
\[ m[14] = m[15] \]


\[ m[15] \cdot x[15] = m[16] \cdot x[16] \]

\[ \text{Call LTGen1}(m[15], m[16], Pm, Q_{\text{gl}}, T[15], x[15]\cdot 100; h[16], h[19], T[16], T[19], x[16]\cdot 100) \]

\[ \text{TV-2} \]

\[ \text{L.T. Condenser} \]

\[ h[4] = \text{Enthalpy}(\text{Steam}, T=t[4], x=q[4]) \]
\[ t[3] = \text{Temperature}(\text{Steam}, P=P, h=h[3]) \]

\[ h[10] = h_\text{LiBrH}_2\text{O}(t[10], x[10]) \]
\[ x[10] = x_\text{LiBrH}_2\text{O}(t[10], Pm) \]
\[ m[9] = m[10] \]
\[ m[16] = m[17] \]
\[ x[9] = x[10] \]
\[ x[16] = x[17] \]

\[ \text{CALL Q}_\text{LiBrH}_2\text{O}(h[17], Pl, x[17]: q[18], t[18], x[18]) \]
\[ h[17] = h_\text{LiBrH}_2\text{O}(t[17], x[17]) \]
\[ m[17] = m[18] \]

\[ h[8] = h_\text{LiBrH}_2\text{O}(t[8], x[8]) \]

\[ \text{Absorber} \]


\[ h[6] = \text{Enthalpy(Steam,} T= t[6], x= q[6]) \]

\[ Pl = \text{Pressure(Steam,} T= t[6], x= q[6]) \]

\[ Pm = \text{Pressure(Steam,} T= t[4], x= q[4]) \]  \hspace{1cm} \text{(TV-4)}


\[ t[5] = \text{Temperature(Steam,} P= Pl, h= h[5]) \]

\[ m[8] = m[9] \]
\[ x[8] = x[9] \]
\[ h[9] = h_{\text{LiBrH2O}}(t[9], x[9]) \]
\[ h[11] = h_{\text{LiBrH2O}}(t[11], x[11]) \]

\[ Q_e + Q_{gh} = Q_a + Q_c \]

\[ E_2 = \frac{(m[8]/\rho[8])}{\rho[8]} \cdot (Pm - Pl) \]  \hspace{1cm} \text{(Pump-1)}

\[ \rho[8] = \rho_{\text{LiBrH2O}}(T[8], x[8]) \]
\[ h[9] = h[8] + \frac{E_2}{m[8]} \]

\[ E_1 = \frac{(m[10]/\rho[10])}{\rho[10]} \cdot (Ph - Pm) \]  \hspace{1cm} \text{(Pump-1)}

\[ \rho[10] = \rho_{\text{LiBrH2O}}(T[10], x[10]) \]
\[ h[11] = h[10] + \frac{E_1}{m[10]} \]
APPENDIX – 3

Table-3A: Simulation Results of Modified Double Effect Chiller with EES at Full Load

Variables in Main

<table>
<thead>
<tr>
<th>Variable</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>COP</td>
<td>1.303</td>
</tr>
<tr>
<td>E_1</td>
<td>0.01277 [kW]</td>
</tr>
<tr>
<td>Ph</td>
<td>48.67 [kPa]</td>
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<tr>
<td>E_2</td>
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</table>

Local variables in LTGen1 (Subroutine)

<table>
<thead>
<tr>
<th>Variable</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
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<td>115.7 [kJ/kg]</td>
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<td>h_out</td>
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<tr>
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Table of Parameters Used in Program

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<th>q[i]</th>
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<td>0.4306</td>
<td>0.43</td>
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</tr>
<tr>
<td>13</td>
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<td>228.4</td>
<td>0.4431</td>
<td>0.4178</td>
<td></td>
</tr>
<tr>
<td>14</td>
<td>76.26</td>
<td>168.1</td>
<td>0.4431</td>
<td>0.4178</td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>55.64</td>
<td>168.1</td>
<td>0.4522</td>
<td>0.02013</td>
<td>0.4178</td>
</tr>
<tr>
<td>16</td>
<td>63.38</td>
<td>132.7</td>
<td>0.4719</td>
<td>0.4004</td>
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</tr>
<tr>
<td>17</td>
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<td>78.67</td>
<td>0.4588</td>
<td>0.4004</td>
<td></td>
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<tr>
<td>18</td>
<td>30.43</td>
<td>78.64</td>
<td>0.4624</td>
<td>0.007675</td>
<td>0.4004</td>
</tr>
</tbody>
</table>
PROGRAM LISTING FOR MODIFIED DOUBLE EFFECT CHILLER WITH FULL LOAD

{LiBr Absorption Chiller Modeling - Series Flow Double Effect Chiller}
{Modeling of chiller with an additional HX to improve COP at low source temperature}

{Subprogram to evaluate properties of LT Generator}

SUBPROGRAM LTGen1 (m_in, m_out, P, Q_gl, T_in, x_in: h_out, h_st, T_out, T_e, x_out) Units=2

{Mass balance}
\[ m_{st} = m_{in} - m_{out} \]

{Equilibrium temperature at P, x_in}
\[ T_e = T_{LIBR}(\text{Units}, P, X_{in}) \]

{Enthalpy at inlet - subcooled state}
\[ h_{in} = H_{LIBR}(T_{in}, X_{in}, \text{Units}) \]

{Properties at outlet - saturated state}
\[ T_{out} = T_{LIBR}(\text{Units}, P, X_{out}) \]
\[ h_{out} = H_{LIBR}(T_{out}, X_{out}, \text{Units}) \]

{Properties of vapor (refrigerant)}
\[ h_{st} = \text{enthalpy(WATER,} T=T_e, P=P) \]

{LiBr mass balance}
\[ m_{in} * x_{in} = m_{out} * x_{out} \]

{Solution & vapour heat transfer}
\[ Q_{sen} = m_{in} * (h_{e} - h_{in}) \]
\[ Q_{vap} = Q_{gl} - Q_{sen} \]

END

{Parameter settings to suit controlled conditions}
\[ q[2] = 0 \] \quad \text{(Vapour quality)}
\[ q[4] = 0 \]
\[ q[6] = 1 \]

{Temp. settings}
\[ t[6] = 14[\text{C}] \]
\[ t[1] = 95[\text{C}] \]
\[ t[4] = 38[\text{C}] \]
\[ t[8] = 32[\text{C}] \]

{System parameters}
\[ Q_e = 70[\text{kW}] \]
\[ m[7] = 0.43[\text{kg/s}] \]

\[ \text{COP} = \frac{Q_{e}}{Q_{gh}} \]


\[ h[1] = \text{Enthalpy(Steam,} T=t[1], P=Ph) \]
\[ h[13] = h_{\text{LiBrH}_2\text{O}(t[13], x[13])} \]
\[ x[13] = x_{\text{LiBrH}_2\text{O}(t[13], Ph)} \]
\[ t[12] = T_{\text{LiBrH}_2\text{O}(Ph, x[12])} \]


\[ h[2] = \text{Enthalpy(Steam,} T=t[2], x=q[2]) \]
\[ t[2] = \text{Temperature(Steam,} P=Ph, x=q[2]) \]
\[ Ph = \text{Pressure(Steam,} T=t[2], x=q[2]) \]
\( m[1] = m[2] \)


\( \text{HX-1} \)  
\( \text{Added SP-1 between HX-1 & HX-2} \)

\( h[14] = h_{\text{LiBrH}_2\text{O}}(t[14], x[14]) \)
\( m[13] = m[14] \)
\( x[10] = x[12] \)
\( x[13] = x[14] \)

\( \text{CALL Q}_{ \text{LiBrH}_2\text{O}}(h[14], P_m, x[14]; q[15], t[15], x[15]) \)  
\( \text{TV-2} \)

\( h[14] = h[15] \)
\( m[14] = m[15] \)


\( \text{LT Generator} \)

\( m[15] = m[16] + m[19] \)
\( m[15] \cdot x[15] = m[16] \cdot x[16] \)
\( \text{Call LTGen1} (m[15], m[16], P_m, Q_{gl}, T[15], x[15] \cdot 100: h[16], h[19], T[16], T[19], x[16] \cdot 100) \)

\( m[2] = m[3] \)
\( q[3] = \text{Quality(Water,} T=T[3], h=h[3]) \)


\( \text{L.T. Condenser} \)

\( h[4] = \text{Enthalpy(Steam,} T= t[4], x=q[4]) \)
\( t[3] = \text{Temperature(Steam,} P=P_m, h=h[3]) \)
\( q[19] = \text{Quality(Water,} T=T[19], h=h[19]) \)


\( \text{HX-2} \)

\( h[10] = h_{\text{LiBrH}_2\text{O}}(t[10], x[10]) \)
\( x[10] = x_{\text{LiBrH}_2\text{O}}(t[10], P_m) \)
\( m[9] = m[10] \)
\( m[16] = m[17] \)
\( x[9] = x[10] \)

\( h[7] \cdot m[7] + Q_{hx3} = m[8] \cdot h[8] \)

\( \text{HX-3} \)

\( h[8] = h_{\text{LiBrH}_2\text{O}}(t[8], x[8]) \)
\( x[7] = x[8] \)
\( m[7] = m[8] \)

\( \text{CALL Q}_{\text{LiBrH}_2\text{O}}(h[17], P_l, x[17]; q[18], t[18], x[18]) \)  
\( \text{TV-1} \)

\( h[17] = h_{\text{LiBrH}_2\text{O}}(t[17], x[17]) \)
\( m[17] = m[18] \)


\( \text{Absorber} \)

\( h[7] = h_{\text{LiBrH}_2\text{O}}(t[7], x[7]) \)
\( t[7] = T_{\text{LiBrH}_2\text{O}}(P_l, x[7]) \)
\( m[7] \cdot x[7] = m[18] \cdot x[18] \)


\( \text{Evaporator} \)

\( h[6] = \text{Enthalpy(Steam,} T=t[6], x=q[6]) \)
\( P_l = \text{Pressure(Steam,} T=t[6], x=q[6]) \)

\( P_m = \text{Pressure(Steam,} T=t[4], x=q[4]) \)  
\( \text{TV-4} \)

\( t[5] = \text{Temperature(Steam, } P=\text{Pl, h=h[5])} \)

\( m[8] = m[9] \)
\( x[8] = x[9] \)
\( h[9] = h_{\text{LiBrH2O}(t[9], x[9])} \)
\( h[11] = h_{\text{LiBrH2O}(t[11], x[11])} \)

\( Q_e + Q_{gh} + Q_{hx3} = Q_a + Q_c \)

\( E_{-2} = \left( \frac{m[8]}{\rho[8]} \right) \left( P_{m} - P_{l} \right) \)
\( \rho[8] = \rho_{\text{LiBrH2O}(T[8], x[8])} \)

\( E_{-1} = \left( \frac{m[10]}{\rho[10]} \right) \left( P_{h} - P_{m} \right) \)
\( \rho[10] = \rho_{\text{LiBrH2O}(T[10], x[10])} \)

Table-3B: Solutions & Equations for Modeling of Double Effect Chiller with EES for Final Trial with Half Load

<table>
<thead>
<tr>
<th>Variables in Main</th>
<th></th>
<th>E_1=0.008917 [kW]</th>
<th>E_2=0.0009015 [kW]</th>
</tr>
</thead>
<tbody>
<tr>
<td>COP=1.308</td>
<td>Ph=48.67 [kPa]</td>
<td>Pl=1.599 [kPa]</td>
<td>Pm=5.945 [kPa]</td>
</tr>
<tr>
<td>Q_a=45.72 [kW]</td>
<td>Q_c=18.53 [kW]</td>
<td>Q_e=35 [kW]</td>
<td></td>
</tr>
<tr>
<td>Q_gh=26.77 [kW]</td>
<td>Q_gl=18.28 [kW]</td>
<td>Q_hx3=2.483 [kW]</td>
<td></td>
</tr>
</tbody>
</table>

Local variables in Subroutine ‘LTGen1’

<table>
<thead>
<tr>
<th>h_e=138.9 [kJ/kg]</th>
<th>h_in=113 [kJ/kg]</th>
<th>h_out=138.4 [kJ/kg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>h_st=2623 [kJ/kg]</td>
<td>m_in=0.2922 [kg/s]</td>
<td>m_out=0.2853 [kg/s]</td>
</tr>
<tr>
<td>m_st=0.006903 [kg/s]</td>
<td>P=5.945 [kPa]</td>
<td>Qvap=10.72 [kW]</td>
</tr>
<tr>
<td>Q_gl=18.28 [kW]</td>
<td>Q_sen=7.559 [kW]</td>
<td>T_e=65.9 [C]</td>
</tr>
<tr>
<td>T_in=54.62 [C]</td>
<td>T_out=65.9 [C]</td>
<td>Units=2</td>
</tr>
<tr>
<td>x_in=46.12</td>
<td>x_out=47.24</td>
<td></td>
</tr>
</tbody>
</table>
### Table of Parameters Used in Program

<table>
<thead>
<tr>
<th>[i]</th>
<th>t[i] [°C]</th>
<th>h[i] [kJ/kg]</th>
<th>x[i]</th>
<th>q[i]</th>
<th>m[i] [kg/s]</th>
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</thead>
<tbody>
<tr>
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<tr>
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<tr>
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<td>1</td>
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</tr>
<tr>
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</tr>
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<td>32</td>
<td>63.19</td>
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<tr>
<td>10</td>
<td>52.6</td>
<td>111.5</td>
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<td>111.5</td>
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<td>101.7</td>
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<tr>
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<td>134.5</td>
<td>0.4576</td>
<td></td>
<td>0.2922</td>
</tr>
<tr>
<td>15</td>
<td>54.62</td>
<td>134.5</td>
<td>0.4612</td>
<td>0.007818</td>
<td>0.2922</td>
</tr>
<tr>
<td>16</td>
<td>65.9</td>
<td>138.4</td>
<td>0.4724</td>
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<td>0.2853</td>
</tr>
<tr>
<td>17</td>
<td>42.74</td>
<td>87.57</td>
<td>0.4638</td>
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<td>0.2853</td>
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<tr>
<td>18</td>
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<td>87.54</td>
<td>0.4687</td>
<td>0.01043</td>
<td>0.2853</td>
</tr>
<tr>
<td>19</td>
<td>65.9</td>
<td>2623</td>
<td></td>
<td>100</td>
<td>0.006903</td>
</tr>
</tbody>
</table>

Program is same as for the full-load trail and only the mass flow rate, condenser temperature and evaporator load (35kW) were changed.
Figure 3A: Double Effect Refrigeration Cycle with Waste-heat feeding Arrangement
**APPENDIX -4**

**CALCULATION OF HEAT EXCHANGERS FOR MAIN UNITS OF THE SYSTEM**

Table-4A: Sizing of Heat Exchanger using Simulation Results at Full Load

<table>
<thead>
<tr>
<th>Variables in Main</th>
<th>$C_p=4.2$ [kJ/kg·K]</th>
<th>$C_{pe}=4.183$ [kJ/kg·K]</th>
<th>$C_{p_h}=4.226$ [kJ/kg·K]</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{p-sh}=2.452$</td>
<td>$C_{p-x1l}=2.408$</td>
<td>$F_a=0.9821$</td>
<td>$F_e=1$</td>
</tr>
<tr>
<td>$F_e=1$</td>
<td>$F_{gh}=0.9753$</td>
<td>$F_{gl}=0.7717$</td>
<td>$F_{x1}=0.8437$</td>
</tr>
<tr>
<td>$F_x2=0.8764$</td>
<td>$F_x3=0.9915$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$h_{bc}=121.5$</td>
<td>$h_{cti}=129.7$</td>
<td>$h_{cto}=106.8$</td>
<td></td>
</tr>
<tr>
<td>$LMTD_e=4.328$</td>
<td>$LMTD_g=3.909$</td>
<td>$LMTD_{gh}=3.595$</td>
<td></td>
</tr>
<tr>
<td>$m_{bc}=1.783$ kg/s</td>
<td>$m_{be}=1.859$ kg/s</td>
<td>$m_{bx3}=0.6262$ kg/s</td>
<td></td>
</tr>
<tr>
<td>$m_{gh}=1.589$ kg/s</td>
<td>$m_{gh}=1.589$ kg/s</td>
<td>$m_{gh}=1.589$ kg/s</td>
<td></td>
</tr>
<tr>
<td>$Pa=0.7347$</td>
<td>$P_{gh}=0.7692$</td>
<td>$P_{gl}=0.3629$</td>
<td></td>
</tr>
<tr>
<td>$P_{x1}=0.5176$</td>
<td>$P_{x2}=0.6688$</td>
<td>$P_{x3}=0.2174$</td>
<td>$Pr_{gh}=125$</td>
</tr>
<tr>
<td>$Q_{ci}=30.96$</td>
<td>$Q_{ct}=130.7$</td>
<td>$Q_{ct}=70$</td>
<td>$Q_{gh}=53.7$</td>
</tr>
<tr>
<td>$Q_{gl}=28.44$ [kW]</td>
<td>$Q_{x1}=25.93$ [kW]</td>
<td>$Q_{x2}=21.63$ [kW]</td>
<td>$Q_{x3}=5.26$ [kW]</td>
</tr>
<tr>
<td>$Ra=1.029$</td>
<td>$R_{gh}=0.2125$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$R_{x1}=0.5365$</td>
<td>$R_{x2}=0.8537$</td>
<td></td>
<td>$rho_{e}=998.3$ [kg/m^3]</td>
</tr>
<tr>
<td>$ta_{bi}=25.5$ [C]</td>
<td>$ta_{bo}=29$ [C]</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$tx_{ci}=30.96$</td>
<td>$tx_{bi}=24$ [C]</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$T_{\chi 1 m}=76.3$</td>
<td>$UA_{a}=63.41$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$UA_{c}=10.38$</td>
<td>$UA_{e}=17.91$</td>
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<td></td>
</tr>
</tbody>
</table>

**Program codes for Determining Heat Exchanger parameters that returned the above solutions**

"HT generator"

$t_{hi}=110$[C]; $t_{ho}=102$[C]; $t[12]=99.6$[C]; $t[13]=101.3$[C]; $Q_{gh}=53.7$[kW]; $x[12]=0.43$; $x[13]=0.418$

$LMTD_{gh} = ((t_{hi} - t[12]) - (t_{ho} - t[13])) / (\ln((t_{hi} - t[12])/(t_{ho} - t[13])))$

$P_{gh} = (t_{ho} - t_{hi}) / (t[12] - t_{hi})$

$R_{gh} = (t[12] - t[13]) / (t_{ho} - t_{hi})$

$F_{gh}=LMTD\_CF(\text{'shell\&tube_2'}, P_{gh}, R_{gh})$

$UA_{gh} = Q_{gh} / (LMTD_{gh} * F_{gh})$
Cp_h = Cp(Water,T=106, x=0);       rho_gh = Density(Water,T=106, x=0)  \{Average of t_hi & t_ho\}
Cp_sh = Cp_LiBrH2O(T_hm, x_gh) \{Ts_m - Mean temp. of solution (t[12] & t[13])
, x_gh - Mean concentration of x[12] & x[13]\}
T_hm = (t[12] + t[13])/2;       x_gh = (x[12] + x[13]) / 2
Qgh = m_gh * Cp_h * (t_hi - t_ho)
V_gh = m_gh / rho_gh
Pr_gh = Pressure(Water,T=106, x=0) \{Pressure in HX\}

"LT Generator & HT Condenser"

\[t[1] = 95^\circ C; \quad t[2] = 80.7^\circ C; \quad t[15] = 55.6^\circ C; \quad m[1] = 0.0121[kg/s];\]
\[Cp_gl = 1.96[kJ/kg·K]; \quad m[15] = 0.0418[kg/s];\]
\[t[16] = 63.4^\circ C; \quad Qgl = 28.44[kW]\]
LMTDgl = \left( (t[1] - t[15]) - ((t[2] - t[16])) \right) / \ln\left( (t[1] - t[15])/((t[2] - t[16]))\right)

Pgl = (t[1] - t[2]) / (t[1] - t[15]),
Rgl = (t[1] - t[2]) / (t[16] - t[15])
Fgl = LMTD_CF('shell&tube_1', Pgl, Rgl)
UAgl = Qgl / (LMTDgl * Fgl)

"Condenser (HX) Sizing - used kW for power"

\[Tc_{bi} = 30^\circ C; \quad Tc_{bo} = 36^\circ C; \quad t[3] = 38^\circ C; \quad t[4] = 38^\circ C; \quad t[5] = 14^\circ C; \quad t[6] = 14^\circ C; \quad Tc_{bo} = 36^\circ C; \quad m[15] = 0.0418[kg/s];\]
\[h[19] = 2623[kJ/kg];\]
LMTDc = \left( (t[3] - tc_{bi}) - (T[4] - tc_{bo}) \right) / \ln\left( (t[3] - tc_{bi})/(T[4] - tc_{bo})\right)

Fc = 1
Qc = 44.94[kW]
UAc = Qc / (Fc*LMTDc)

"Evaporator Sizing - used W for power"

"Using LMTD Method"

\[te_{bi} = 24^\circ C; \quad te_{bo} = 15^\circ C; \quad T[5] = 14^\circ C; \quad T[6] = 14^\circ C; \quad Qe = 70[kW];\]
LMTDe = \left( (te_{bi} - T[5]) - (te_{bo} - T[6]) \right) / \ln\left( (te_{bi} - T[5])/(te_{bo} - T[6])\right)

"Fe=LMTD_CF('shell&tube_1', Pe, Re)
\[Pe = (t[6] - t[5]) / (Te_{bi} - t[5])\]
\[Re = (Te_{bi} - Te_{bo}) / (t[6] - t[5])\]
Fe = 1
Qe = Fe * LMTDe * UAe
Cpe = Cp(Water,T=19.5, x=0); \{Average of te_{bi} & te_{bo}\}
\[ \rho_e = \text{Density(Water, T=19.5, x=0)} \]

\[ Q_e = m_{be} \cdot C_{pe} \cdot (t_{eb} - t_{eo}) \]

\[ V_e = \frac{m_{be}}{\rho_e} \]

**Absorber**

\[ t_1[18] = 30.4[^\circ C]; \quad t_1[7] = 26.8[^\circ C]; \quad t_a_{bi} = 25.5[^\circ C]; \quad t_a_{bo} = 29[^\circ C]; \]

\[ \text{LMTDA} = \frac{(t_1[18] - t_a_{bo}) - (t_1[7] - t_a_{bi})}{\ln((t_1[18] - t_a_{bo}) / (t_1[7] - t_a_{bi}))} \]

\[ Q_a = 84.03[kW] \]

\[ F_a = \text{LMTD\_CF('shell\&tube_9', Pa, Ra)} \]

\[ P_a = \frac{(t_1[18] - t_1[7])}{(t_1[18] - t_a_{bi})} \]

\[ R_a = \frac{(t_1[18] - t_1[7])}{(t_a_{bo} - t_a_{bi})} \]

\[ Q_a = U_{Aa} \cdot F_a \cdot \text{LMTDA} \]

**HX-1**

\[ Q_x[1] = m_{[13]} \cdot (h_{[13]} - h_{[14]}) \]

\[ m_{[13]} = 0.43; \quad h_{[13]} = 228.4; \quad h_{[14]} = 168.1 \]

\[ t_1[11] = 53[^\circ C]; \quad t_1[12] = 99.6[^\circ C]; \quad t_1[13] = 101.3[^\circ C]; \quad t_1[14] = 76.3[^\circ C]; \]

\[ \text{LMTD}_{x_1} = \frac{(t_1[13] - t_1[12]) - (t_1[14] - t_1[11])}{\ln((t_1[13] - t_1[12]) / (t_1[14] - t_1[11]))} \]

\[ F_{x1} = \text{LMTD\_CF('parallelflow', P_{x1}, R_{x1})} \]

\[ P_{x1} = \frac{(t_1[13] - t_1[14])}{(t_1[13] - t_1[11])} \]

\[ R_{x1} = \frac{(t_1[13] - t_1[14])}{(t_1[12] - t_1[11])} \]

\[ Q_x[1] = F_{x1} \cdot U_{A_x[1]} \cdot \text{LMTD}_{x_1} \]

\[ C_{p_x[1]} = C_{p\_LiBrH2O(T_{x[1]}m, x_1[11])} \]

\[ T_{x[1]}m = \frac{(t_1[11] + t_1[12])}{2}; \quad x_{1[11]} = 0.431 \]

**HX-2**

\[ Q_x[2] = m_{[10]} \cdot (h_{[10]} - h_{[9]}) \]

\[ m_{[10]} = 0.43; \quad h_{[9]} = 63.79; \quad h_{[10]} = 114.1 \]

\[ t_1[9] = 32[^\circ C]; \quad t_1[10] = 53[^\circ C]; \quad t_1[16] = 63.4[^\circ C]; \quad t_1[17] = 38.8[^\circ C]; \]

\[ \text{LMTD}_{x_2} = \frac{(t_1[16] - t_1[10]) - (t_1[17] - t_1[9])}{\ln((t_1[16] - t_1[10]) / (t_1[17] - t_1[9]))} \]

\[ F_{x2} = \text{LMTD\_CF('shell\&tube_2', P_{x2}, R_{x2})} \]

\[ P_{x2} = \frac{(t_1[10] - t_9)}{(t_1[16] - t_9)} \]

\[ R_{x2} = \frac{(t_1[10] - t_9)}{(t_1[16] - t_1[17])} \]

\[ Q_x[2] = F_{x2} \cdot U_{A_x[2]} \cdot \text{LMTD}_{x_2} \]

**HX-3**

\[ t_1[7] = 25.1[^\circ C]; \quad t_1[8] = 32[^\circ C]; \quad t_{x3}\_{bi} = 36[^\circ C]; \quad t_{x3}\_{bo} = 34[^\circ C]; \quad Q_x[3] = 5.26[kW] \]

\[ \text{LMTD}_{x3} = \frac{(t_{x3}\_{bi} - t_1[7]) - (t_{x3}\_{bo} - t_1[8])}{\ln((t_{x3}\_{bi} - t_1[7]) / (t_{x3}\_{bo} - t_1[8]))} \]

\[ F_{x3} = \text{LMTD\_CF('parallelflow', P_{x3}, R_{x3})} \]

\[ P_{x3} = (t_{x3}\_{bi} - t_{x3}\_{bo}) / (t_{x3}\_{bi} - t_1[7]) \]

\[ R_{x3} = (t_{x3}\_{bi} - t_{x3}\_{bo}) / (t_1[8] - t_1[7]) \]

\[ Q_x[3] = U_{A_x[3]} \cdot F_{x3} \cdot \text{LMTD}_{x3} \]
"Cooling Tower inlet water temperature calculation considering mixing from 3 inlets"

Cp = 4.2[kJ/kg·K]

(Absorber)

\[ h_{bac} = \text{Enthalpy(Water,} T=ta_{bo}, x=0) \]
\[ Q_a = m_{ba} * Cp * (ta_{bo} - ta_{bi}) \]

(LT Condenser)

\[ h_{bc} = \text{Enthalpy(Water,} T=tc_{bo}, x=0) \]
\[ Q_c = m_{bc} * Cp * (tc_{bo} - tc_{bi}) \]

(HX3)

\[ h_{bx3} = \text{Enthalpy(Water,} T=tx3_{bo}, x=0) \]
\[ Q_{x3} = m_{bx3} * Cp * (tx3_{bi} - tx3_{bo}) \]

(Let inlet brine temperature of Cooling Tower be \( t_{cti} \) & s. enthalpy be \( h_{cti} \) - outlet state of mixing chamber)

\[ h_{bac} * (m_{ba} - m_{bc}) + h_{bc} * (m_{bc} - m_{bx3}) + h_{bx3} * m_{bx3} = h_{cti} * m_{ba} \]
\[ t_{ci} = \text{Temperature(Water,} h=h_{cti}, x=0) \]

(Cooling tower capacity)

\[ h_{cto} = \text{Enthalpy(Water,} T=ta_{bi}, x=0) \]
\[ Q_{ct} = m_{ba} * (h_{cti} - h_{cto}) \]