UNIVERSITY OF TURKISH AERONAUTICAL ASSOCIATION
INSTITUTE OF SCIENCE AND TECHNOLOGY

STUDY OF COOLING SYSTEM FOR BUILDING BY ABSORPTION SYSTEM CYCLE USING TURBINE EXHAUST GASES

Master Thesis

Khalid Sabah Shaeer SHAEER

Institute of Science and Technology
Mechanical and Aeronautical Engineering Department

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1406080019

IN PARTIAL FULFILLMENT OF THE REQUIREMENT FOR THE DEGREE OF MASTER OF SCIENCE INMECHANICAL AND AERONAUTICAL ENGINEERING

Supervisor: Assist. Prof. Dr. Mohamed Salem ELMNEFI
CO- supervisor: Assist. Prof .Dr .Omar Khaleel Ahmed

Khalid Sabah Shaeer SHAEER, having student number **1406080019** and enrolled in the Master Program at the University of Turkish Aeronautical Association, after meeting all of the required conditions contained in the related regulations, has successfully accomplished, in front of the jury, the presentation of the thesis prepared with the title of: "**STUDY OF COOLING SYSTEM FOR BUILDING BY ABSORPTION SYSTEM CYCLE USING TURBINE EXHAUST GASES**".

Supervisor : **Assist. Prof. Dr. Mohamed Salem ELMNEFI**
University of Turkish Aeronautical Association ……………

Jury Members Asst. **Prof. Dr. Tahsin Çağrı ŞİŞMAN**
University of Turkish Aeronautical Association ……………

: **Assist. Prof. Dr. Mohamed Salem ELMNEFI**
University of Turkish Aeronautical Association ……………

: **Assist. Prof. Dr. Munir ELFARRA**
Yildirim Beyazit University- Ankara ……………

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Khalid SHAEER
07.12.2017

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December.2017 Khalid SHAEER

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RH	Relative humidity [kg moisture/kg dry air]
Q	Heat gain [watt]
U	Over all heat transfer [$w/m^2.°C$]
A	Area [m^2]
CLTD	Cooling load temperature difference [°C]
CLTDc	Correction cooling load temperature difference[°C]
ho	Outside heat transfer coefficient [$w/m^2.°C$]
hi	Inside heat transfer coefficient [$w/m^2.°C$]
K	Thermal conductivity[w/m.°C]
x	Thickness [m]
SHG_{max}	Maximum solar heat gain [w/m^2]
Qs	Sensible heat gain [w]
Q_L	Latinate heat gain [w]
Qeq	Equipment heat gain [w]
Vinf	Volume infiltration [m^3/sec]
$Q_{s,infil}$	Sensible heat of infiltration [w]
Ti	Inside temperature [°C]
T_o	Outside temperature [°C]
$Q_{l,infil}$	Latinate heat of infiltration [w]
TCL	Total cooling load [w]
Wo	Outside specific humidity [kg moisture/kg dry air]
Wi	Inside specific humidity [kg moisture/kg dry air]
SCL	Sensible cooling load [w]
LCL	Latent cooling load [w]
Vs	Supply air [m^3/sec]
Qg	Heat generator [watt]
ṁ	Mass flow rate [kg/esc]

h	Enthalpy [kj/kg.k]
$\dot{m}_{ws}$	Week solution mass flow rate [kg/sec]
$\dot{m}_{ss}$	Strong solution mass flow rate [kg/sec]
Qe	Evaporator heat [kw]
P	Pressure [kpa]
Wp	Work pump [kw]
Q_{exh}	Exhaust heat [MW]
T_1	Air inlet temperature [K]
T	Out let temperature [K]
Cpg	Specific heat of exhaust gases [kj / kg.k]
AC	Air change [dimensionless]
W	Watts
MW	Mega watt
KW	Kilowatt
S	South direction [dimensionless]
SW	South west direction [dimensionless]
W	West direction [dimensionless]
NW	North west direction [dimensionless]
LM	latitude month correction [dimensionless]
K	correction factor [dimensionless]
Jan	January
Feb	February
Apr	April
Aug	August
Sept	September
Oct	October
Nov	November
Dec	December
HORIZ	Horizontal
SC	Shading coefficient [dimensionless]
CLF	Cooling load factor [dimensionless]
N	Number of people [dimensionless]
FC	Special allowance factor [dimensionless]

m^2	Meter square
m^3/sec	Cubic meter per second
L/s.m^2	Later per second for meter square
S.H.F	Sensible heat factor [dimensionless]
λ	Circulation ratio [dimensionless]
ε	Effectiveness of heat exchanger
COP	Coefficient of performance [dimensionless]
KWh	Kilowatt hour
TL	Turkish lira
ID	IRQ dinar
ASHRAE	American society of heating refrigerating and air conditioning engineers

ABSATRACT

STUDY OF COOLING SYSTEM FOR BUILDING BY ABSORPTION SYSTEM CYCLE USING TURBINE EXHAUST GASES

SHAEER, Khalid

M.Sc., Mechanical and Aeronautical Engineering Department

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Human comfort is important nowadays because of the development in life styles and the increase in weather temperatures. Electrical air conditioning systems and machines are not the most suitable solutions for large structures and buildings due to their high electrical power consumption and short lifetime. An absorption air conditioning system is more suitable for operation using various heat sources such as waste heat, liquid fuel and electrical sources. Moreover, they have low economic and operation costs. Large buildings, such as office buildings and commercial complexes, can be provided with absorption air conditioning systems.

This study estimates the cooling loads of an office building located in the city of Salahuddin, Iraq (35°N latitude) using the exhaust gas of gas turbine and the cooling load obtained by using the Cooling Load Temperate Difference (CLTD) method. The building has 29 spaces of two floors in a dry, high temperature climate. The cooling load elements, such as people heat load gain, lighting heat load gain, ventilation and infiltration heat gain, are all fully estimated. The calculations are also performed for cooling heat loads through walls and roofs. In this study, the use of an absorption refrigeration system cycle powered by exhaust waste heat from the electric power generation from a gas turbine could deliver the necessary cooling to decrease cost operations and overall energy consumption. The Microsoft Excel program was used to analyze the system at various generator and condenser temperatures to find a suitable Coefficient of Performance (COP). For the other part of this study, heat exhaust gases and cost savings operations were calculated by comparing between vapor compression chillers and exhaust-driven absorption chillers in Iraq and Turkey. The results showed a total cooling load in the office building of 217.156 KW and an average sensible heat factor (S.H.F) of 0.911. The total supply of air to the spaces was 14.3 m^3/sec. The designed absorption cooling system had varying COP values because we analyzed the effect of the COP in the variation of the generator temperature and condenser temperature. The maximum COP values for Cases 1, 2 and 3 were 0.79, 0.77 and 0.753, respectively. For the waste heat gases, the results showed 155.17 MW as a minimum value and 362.49 MW as a maximum value. Half-yearly cost savings operations of the system were $4846.92 (in Iraq) and $5519.77 (in Turkey).

Keywords: Cooling Load Temperate Difference (CLTD), Absorption system cycle, waste heat gases

ÖZET

TÜRBİN EGZOZ GAZLARI KULLANILARAK ABSORPSİYON SİSTEM DÖNGÜSÜ İLE BİNA İÇİN SOĞUTMA SİSTEMİ ARAŞTIRMASI

SHAEER, Khalid

Fen bilimleri yüksek lisans,Makine mühendisliği

Danışman:Yardımcı Prof.Dr. Mohamed Salem ELMNEFI

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Günümüzde yaşam tarzlarındaki gelişme ve hava sıcaklıklarındaki artış nedeniyle insanların konforlu ve rahat olması önemli hale gelmiştir. Elektrikli klima sistemleri ve makineler yüksek elektrik enerjisi tüketimi ve kısa ömürleri nedeniyle büyük yapılar ve binalar için en uygun çözüm olmamaktadır. Absorbsiyonlu bir havalandırma sistemi, atık ısı, sıvı yakıt ve elektrik kaynakları gibi çeşitli ısı kaynakları kullanılarak çalıştırmak için daha uygundur. Buna ilaveten daha ekonomik olup işletme maliyetleri daha düşüktür. Ofis binaları ve iş hanları gibi büyük binalarda absorpsiyonlu havalandırma sistemleri kullanılabilir.

Bu çalışma, Irak'ın Selahaddin şehrinde (35° K enleminde) bulunan bir ofis binasının soğutma yüklerini, bir gaz türbininin egzoz gazı kullanılarak değerlendirmekte olup soğutma yükü Soğutma Yükü Sıcaklık Farkı (CLTD) metodu kullanılarak elde edilmiştir. Bina içerisinde kuru, yüksek sıcaklık ikliminde olan iki kat ve 29 adet yer bulunmaktadır. İnsan ısı yükü kazanımı, aydınlatma ısı yük kazanımı, havalandırma ve filtreleme ısı kazanımı gibi soğutma yükü unsurları tamamen değerlendirilmiş bulunmaktadır. Hesaplamalar aynı zamanda duvarlar ve çatılardan soğutma ısı yükleri içinde yapılmıştır. Bu çalışmada, bir gaz türbininden gelen elektrik enerjisi üretiminden çıkan egzoz atık ısı ile tahrik edilen absorpsiyonlu soğutma sistemi döngüsünün kullanılmasının maliyet operasyonlarını ve toplam enerji tüketimini azaltmak amacıyla gerekli soğutmayı. sağlayabileceği ortaya konulmuştur. Sistemi çeşitli jeneratör ve kondansatör sıcaklıklarında analiz etmek ve uygun Verim

Katsayısını (COP) bulmak için Microsoft Excel programı kullanılmıştır. Çalışmanın diğer bölümlerinde, Irak ve Türkiye'de buhar sıkıştırmalı soğutucular ile egzoz gücüyle çalışan absorpsiyonlu soğutucular arasında karşılaştırma yapılarak ısı egzoz gazları ve maliyet tasarrufu işlemleri hesaplanmıştır.

Elde edilen sonuçlar, ofis binasındaki toplam soğutma yükünün 217.156 KW olduğunu ve ortalama hissedilebilir ısı faktörünün (SHF) 0.911 olduğunu göstermiştir. Ofis binasındaki yerlere tedarik edilen toplam hava miktarı 14.3 m^3/sn olmuştur. Tasarlanan absorpsiyonlu soğutma sistemi, verim katsayısının etkisinin değişik jeneratör sıcaklığı ve kondansatör sıcaklığında analiz edilmesinden dolayı değişkenlik gösteren COP değerlerine sahip olmuştur. Durum 1, 2 ve 3 için maksimum COP değerleri sırasıyla 0.79, 0.77 ve 0.753 olmuştur. Atık ısı gazları için, elde edilen sonuçlar minimum değerin 155.17 MW ve maksimum değerin 362.49 MW olduğunu göstermiştir. Sistemin yarı yıllık maliyet tasarruf oranları ise (Irak'ta) 4846.92 $ ve (Türkiye'de) 5519.77$ olmuştur.

Anahtar kelimeler: Soğutma Yükü Sıcaklık Farkı (CLTD), Tasarlanan absorpsiyonlu soğutma sistemi, Atık ısı gazları için

CHAPTER ONE

Introduction

Iraq is known by its hot and dry climate, these climatic conditions imposed using air-conditioning equipment as a solution. Many of homes, office buildings and commercial facilities would not be comfortable without a year-round control of the indoor climate. Along with rapid evolution in improving human comfort came the realization of goods that could be produced better, faster and more economically in a properly controlled climate. In fact, many today goods could be not produced if the temperature, humidity and air quality were not controlled within very narrow limits. Many premature cooling systems were invented with varies source of energy process. Today, Energy origin conservation is the top priority of air conditioning designers systems. With regard to energy saving and the pulsations degrees in the factories, it is one of the most important impacts on production cost and profits.

1.1 Air Conditioning

The heat energy consumption by buildings structure and industries are answerable for these issues. Elaborated estimation of cooling heat, heating capacity, appropriate sizing of the heat ventilation air- conditioning (HVAC) system and optimum control of the (HVAC) systems are significant to decrease energy consumption. Structure cooling heat load ingredients are solar radiation, ventilation load, infiltration heat load, transmission load and internal heat load. Evaluating each of these loads separately and counting them up gives the guess of total cooling load. The load, thus considered, the establisher total sensible heat load. Normal practice, which imposed on the building stricture type and sure percentage of it, are added to income care of latent heat load. Smearing the laws of solar radiation and heat transfer makes load approximations. For the design, conditions and the building used materials, cooling load temperature difference.

cooling load operators and solar heat gain factors are calculated. Values of solar heat energy estimation applied to estimate the indirect and direct heating solar parts of the building structure. All these results, when added, they are giving the total cooling load of the building. The possible loads comprised in the procedures of load sheet. These consist of solar radiation, transmission load done exposed walls, partition walls, roof, floors and outdoor air load. Estimation of thermal cooling load of building is very vital to find suitable air-conditioning equipment and air handling unit, to achieve comfort operation and good air distribution in the air- conditioned zone. This study is dealing with Cooling load prediction for an office building by the method of **(**CLTD) **[1].**

1.2 Absorption Cooling System

Waste heat from gas turbine is the heat rejected for surrounding as a waste heat. This exhaust waste heating can be unlike the air-conditioning, which uses heat operating air-conditioning system cycle, like an absorption air-conditioning cycle. Electricity bought from the companies for the conventional vapor compression air-condition system can reduced. The usage of heat which worked in air-conditioning system cycle help to decrease problem connected to global environmental, like the so name greenhouse effect from (CO_2) emission of the combustion of firing fuels in a station of power plants. Greatest vapor compression systems cycle ordinarily use chlorofluorocarbon refrigerants (CFCs), Through the limited using of (CFCs), due to reduction of the ozone layer that will make the absorption systems additional outstanding. Absorption systems appear to provide several benefits, vapor compression system cycle yet dominate all market segments. In order to help the using of the absorption systems, further development is needed to increase their performance and reduction the cost **[2].** The fig 1.1 shows the process of absorption air condition system cycle.

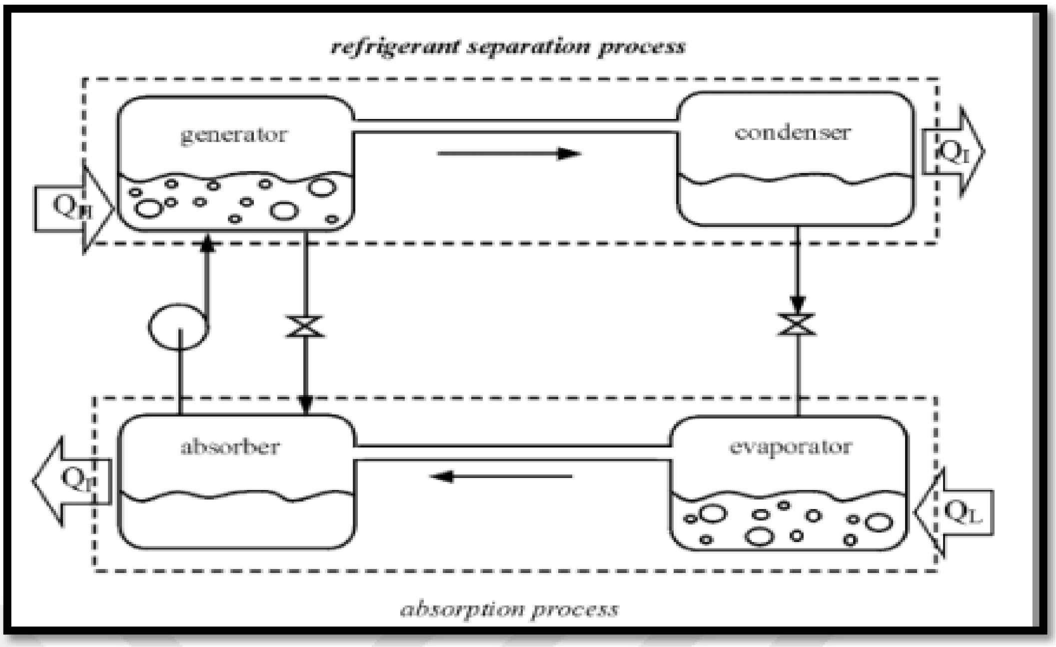


Figure 1.1: Processes of absorption air-conditioning system cycle.

The mixture called Aqua Ammonia was used inside absorption systems years' earlier (LiBr-water) the combination became more popular. The absorption cycle systems have experienced various downs and ups. The absorption system was the predecessor of the vapour compression cycle in (19^{th}) century. Aqua -Ammonia system liked wide applications of domestic refrigerators, big industrial connections in the process and chemical industries. The LiBr-water system was commercialized in the middle of the (20^{th}) as water chillers for big air conditioning of buildings **[2]**.

In the (LiBr-water) system, water is used as a refrigerant inside the system cycle. This type of the systems don't use for applications requiring temperatures below Zero Degree Celsius, And the LiBr life has a salt recrystallization zone when it converts solid and later imposes flow limitations. The vapour absorption system cycle practically LiBr-water system cycle has bounced back when it is using the solar energy and the recent emphasis on co-generation which brands obtainable. Then the waste heat **[2].**

This study contributes in an important knowledge for **study of cooling system for building by absorption system cycle using turbine exhaust gases** that decrease energy ingesting, reduction operational costs and develop environmental benefits in a light commercial buildings and residential. The study performed in Salahuddin city (Iraq).

1.3 Problem Statement

Most of the buildings in Iraq environment consumes high amount in term of energy and consequently higher electricity. Electricity tariff will be rise in the matter of time. Beside the cost saving, this study also carried out the concern on the optimal calculation-cooling load in the commercial building. The optimal design of cooling system will define the efficiency of system operation in the building. Cooling load of building need to define clearly in order to have the optimal design of cooling system .This study will address the optimal design of the exhaust gases absorption system by considering both economic and environment. Lack of the study of waste absorption system to Iraq region leads for this study to carry out. In Iraq, The effectiveness of the waste absorption chiller also has not clearly identified. By given the specification data of finished building and other related information. The aim of this study need to be find. Study of the energy consumption carried out to do the analysis. Result of cooling load of the building will determine the capacity of the waste absorption cooling system and heat waste using to operate the cooling absorption cycle.

1.4 Thesis Objective

The aim of this thesis is to predict the cooling load of the building to find the suitable amount of energy requirements to cool the building and design absorption cooling system to cool this building. This thesis include the utilize use of gas turbine exhaust gases to drive absorption cooling system and study on the effectiveness of gas turbine exhaust gases for absorption the cooling system in Iraq environment. Lead to a cooling system application with reduce energy consumption, decrease capital operation, also benefits the environment. This study will also include economic analysis on building by using the waste heat from the gas turbine exhaust with absorption cooling system and comparing the findings with case study in Turkey .

1.5 Organization of the Thesis

Chapter two: This chapter describes a general literature review of the study and description of old studies that carried out various researches of cooling load of a building, design or analyses the absorption system, waste heat recovery from gas turbine gases and the reign of study.

Chapter three: This chapter is presenting the methodology of thesis to calculate:

- The estimation cooling heat load the office building, the cooling heat load of the office building comes from the external heat load and internal heat load. The external heat load comes from walls, windows and roofs and the internal load comes from occupancy, lighting, equipment and infiltration.
- The analyses and estimate of absorption cooling system cycle estimate appropriate absorption system COP by study three cases and each case has 6 various the generator temperature.
- The analyses the gas turbine waste gases which comes from exhaust gas a turbine that use to fire absorption system and calculate heat quantity at varying electric generation load.
- The make comparative cost operation between compression cooling chiller and absorption cooling chiller to know economizers save per 6 months.

Chapter four: This chapter is showing results and discussions the absorption system cycle and cooling load of the building, estimation of, result the quantity of waste gas turbine gases that use to fire the absorption cooling system, economic cost operation and dissect the results.

Chapter five: This chapter includes Conclusion of the thesis, and the future work.

1.6 Aim the Present Work

In Iraq most the buildings using conventional air conditioning system which uses to drive the system which exhibited malty clear disadvantages like high energy consumption, high quantity electric energy at peak load consumption. Because of high energy cost, the degrease of fossil fuel resources, the utilization the low level renewable energy sources in refrigeration system have become a way to address these problems. In this research work, a new study system in Iraq environment was suggest and it will be studied. The Iraq country has not previous studies on using the waste heat in the past with the air-conditioning of buildings, this research can make leading of entry into this filed. Gas turbine engines produce the most electric energy in Iraq and it produces much quantity of waste gases that can help to reduce the capital cost operation, and it reduces the environmental damage and climate change. This study can consider as a guide for further study in the future in Iraq.

CHAPTER TWO

LITERATURE REVIEW

2.1 Introduction

The first main purpose of air conditioning system is to supply a comfortable environment for people or machines, computer centers, office building, metrology and laboratories inside a closed space. Therefore, the cooling process, reduce the temperature inside the space; heating processes define the meager processes of the air-conditioning. The temperature of human body generally estimated at 37 °C. The body of human feels comfortable when the heat level is transferred to the space at comfort rate. The heat transferring rate quantity depends on the same properties of air space temperature, humidity, and air velocity. In building of office, hospitals, hotels, restaurants and homes air conditioning is no longer auxiliary but also an impart of modern living [3]. The absorption machines might represent an interesting alternative to traditional vapour compression units as they are free from ozone depleting fluids and they can be motivated by thermal energy at low temperature levels, as the heat recovered from an industrial operation or from a thermal engine. It is possible to find several applications of absorption chillers driven by the heat recovery on the exhaust from a gas turbine plant. The gas turbines have established an important role in the industrial production of mechanical energy owing to the very high power-to-weight ratio achievable with simple-cycle configurations and to the high conversion efficiency that can obtained in systems that envisage waste heat recovery from the exhaust gases. The rising popularity of gas turbines in new year's is attributable to the rapid modifications in this technology, which have led to enhancements in the design of both the individual components and the system as a whole. These technical developments, that concern significant progress in materials, construction techniques, blades cooling, control of pollutant emissions, reliability and availability of machines, have improved the performance of modest cycles gas turbines, in terms of electrical efficiency and unit size. They have also led to an important increase in temperature.

2.2 Building

The practical measurements of the cooling load calculation in Iraq was involved. Testing rooms located in the city of Baquba (Iraq) studied to calculate cooling load. The measurements taken during the summer season in (May, June, July, August and September). The testing room area was 15 m^2 with a height of 3 m roof with a medium weight construction specification. The test room included exterior west and east wall facing, un-conditioning neighboring space with an external roof, exterior west door and east window . The air cooling unit was a window type with a cooling capacity of (7 kW) 2 T.R. cooling load capacity of the room calculated by three calculation methods of cooling load, which were TFM, TETD/TA and CLTD/SCL/CLF methods. There was founded a large difference between experimental and theoretical cooling load with all methods. The variance ranged from 33% to 40%. The malty difference between the theoretical and experimental test was appear because several season [4]**.** The method of heat transfer function was used to calculate cooling load of the building. The hourly cooling load predict due to different types of wall, ceiling and fenestration used in that test. The output data by this method studied and compared with carrier program, they found the results were acceptable. In other cases, the results of cooling load parts studied with ASHRAE solutions, results had some difference noticed for roof and wall. The researcher also calculated the changing of cooler wall on heat load part [5]. The load computation was done in the Student Activity Center, which was having four stories. Each floor of the center was used various objectives, having various floor area and malty number of wall windows and doors. The different floors of the center needed vires cooling load. For the calculating of heat Cooling Load Temperature Difference/Cooling Load Factor (CLTD/CLF) method was utilized which was battered on ASHRAE methods and it's given very good satisfactory result when it is compared with complex cooling load calculation tools [6]**.** Cooling Load calculation for air conditioning cooling system done either by manual estimated or critical calculation depended on an understanding of air conditioning calculation. While hand calculation was hard, calculate depended was liable for incorrect owing a big load computation of computer program automatic was probably going to have a constructive outcome in the dynamic way of air conditioning systems. The researchers developed a modern computer program to calculate and simple run with estimation cooling load in Nigeria. The study revue advanced system that funded cooling loads for air condition system for residential and non-residential structures. It makes the use of method powerful and easy to presenting a private cooling load estimation [7]. The cooling load calculation was done in two levels of a building in Jabalpur city (Saudi). That work presented the CLF/CLTD method that indicated choice and possible by evaporative cooling system type to replace of high power consuming air conditioners. Partially or completely for maintaining thermal comfort in various climatic places without compromising the inside air quantity .that work was applied the method of Jabalpur city characterized by different climate conditions over entire month of the years. The calculation procedure of that work examined the option of space conditioning the interiors of a Hall building in Jabalpur Engineering Collage; Jabalpur uses evaporative cooling in the summer month of April, May, and June. The estimation procedure for cooling loads due to solar radiation transmitted through fenestration was used with a new factor the solar cooling load (SCL) which was more accurate and easier to use [8]**.** The method used to calculate the cooling load of the student room at the institute was worked by the CLTD method. The CLTD method produced by ASHRAE and it is calculated for each wall and roof. Cooling load temperature difference method produced easy to calculate the cooling load by hand and it gives a very good satisfactory outcome compared to the cooling load complex. The student room was needed 27.98 kW for 60 students in the summer class and for the required wind season of 25.8 kW. The total air dehumidifier was 64 m^3 / min for the summer and 41 m^3 / min for the monsoons. There was also seen a variation in the solar air temperature of the to guide different surfaces [9]**.** The compression between CLTD method calculation and hurly analyses program HAP was adopted to calculate cooling loads for the building .the work was developed to find the best way to apply the concept of heat ventilation and air conditioning system in Iraq at Erbil city .The hourly analyses program HAP uses the ASHRAE transfer function method for cooling load calculation. The purpose of the study was to calculate cooling load and compare between a CLTD method and HAP program, the building was consisted tow story with 19 zones to estimate the cooling load. The international standard was used in the study. The work estimated the cooling load for multi climate conditions by using a CLTD method for the building. Cooling load as walls, roofs, lighting, people, infiltration and ventilation heat gain can be easily entered into the Excel program. The results show that there is the same outcome between the two methods. The deference total cooling load between CLTD method and HAP program was 2.1% [10]. The two rooms, particularly Computer Laboratory Room and Excellent Center Room were estimated cooling load by CLTD method at Faculty of University Malaysia Pahang. From that work, the cooling load in Computer Laboratory, Room and in Excellent Center, Room were 20458.6 W and 33541.3 W respectively. The maximum solar radiation was founded at 10 am, which contributed 7.6 % and 8.6 % in room of computer and room of excellent center respectively. The cooling heat load gained from component was taken slightly effect of the total heat gain and also slightly reduced energy consumption of air conditioning system. The rating of cooling capacity delivered by the air conditioning system was above for 25.8 % and 23.7 % of Excellent Center room and Computer Research laboratory room[11]. Thermal analysis of educational construction in the hot and humid areas was shown in Saudi Arabia. The carrier company to estimate cooling load of the building to find a suitable the cooling load produced the HAP program. They used a hand calculation to check results of the program. According to load analysis, appropriate air conditioning systems were selected of the building. The energy analysis was calculated to estimate the effectiveness of system, which shows a good Performance Source result. The performance, was compared with the 4.2 the hourly analyses program result and the outcome was a little difference between two methods. The definition was given of thermal resistance of the materials for wall, ceiling and windows [12]. The spinning workshop of Xi'an locale in the summer for instance, was studding and estimated the cooling load by using the method of cooling load coefficient for each part .The work was used tow method to calculate the cooling load. At that point, they looked into an air treatment technique, which collecting the operation of exhaust distinctly in spinning workshop and computed required cooling limit. They believed that technique was possible and could find exactly to avoiding an original sin that workshop required heat, cooling load limit insufficient by conventional cooling limit calculation [13]**.** The modified method for calculating the value of heating loads and seasonal heating demands for residential buildings was developed mathematically. Study results were compared with ASHRAE standards and the results shows that the data obtained from the modified method were more exact and effective after comparing the results. Moreover, it provided the duration of heating seasons for each building at the same climatic different condition [14]. The cooling load of office working in Beijing, China was studied by the hourly cooling load method. The Hourly Cooling Load Factor Method (HCLFM), that can give speedy and a reasonable gauge of building cooling load for huge scale urban vitality arranging. They presented the suspicions and standards of the suggest method and in addition, examining the suggestion and constraint of the approach. The estimated results demonstrate that the dynamical design of the cooling trend was reasonable [15]. The heat insulation was raised on the load of cooling buildings and cooling system in case Air conditioning was assessed by a central air conditioning system throughout the sample building located In Adana. The study was based on the outcomes for three types of insulation (A, B and C-type buildings) according to the energy efficiency index specified in the thermal insulation list used in Turkey. The cost of the air-conditioning system was calculated using the cooling bin numbers. Life cycle cost analysis Implemented using the current cost method. The results show that both the initial and the operating costs of air conditioning system were significantly reduced for all three-insulation thickness. Though, optimal outcomes were obtained in the light of economic measurements of building using type C [16]. The energy saving period and the repayment period were calculated to build a model structure based on cooling and heating loads. Annual transmission loads were calculated strictly through an analytical method based on complex finite Fourier transform (CFFT). The walls, which there direction to the west and east direction were least favored in the cooling period, while the wall facing the north direction was less preferable in the heating period. It was noted that the direction of the wall had less impact on the optimum insulation thickness. Their study showed that economic parts for insulation cost and building life with energy cost. They also compared the present study with degree days model [17]. The development of a new example of the general weather and the mathematical model was discussed to generate loads of load design using the Monte Carlo simulation of a subtropical climate. The outcomes could useful to determine HVAC, energy consumption in buildings in order to obtain more data representative to predict annual energy consumption. The methods that were formed that would be useful for the efficient design and calculation of precise cooling for air-conditioned buildings, in meeting the demand of people loads and updated outdoor information. The integration of the load time of personal load difference in cooling capacity of the load was ranged varies from 1% to 5%, but a change in the weather year was not significant [18]. The solar heat gain for radiant cooling system was studied. Specific consideration was given to the segment of solar oriented radiation changed over to cooling load without using solar heat absorption because of the heat mass of the room. That the particular segment of the cooling burden was characterized as the Direct Solar Load. The direct Solar Load in cooling load counts was suggest and it was compared with the Heat Balance method and the Radiant Time Series method. The proportion of solar heat picks up gains changed over cooling load at the account of a low heat mass of the roof and it was tested for various types of office rooms with a vast glazed outside surface [19]**.** The new method of cooling field was suggest based of a stochastic estimation Method. Uncertainty ranges were applied to input data and it was disseminated through the building model In order to determine their impact on peak load. The a new method resulted was likened with the conventional solution, and the global compassion examination was taken to identify the most important doubts [20]**.** Load computation in element conditions was a fundamental objective of building vitality reproduction. An unconstrained optimum control process was suggest which used feed-forward to reward the climate conditions and model prognostic software design for set-point tracking. The resulted funded the maximum load capacity depended on the set-back time of the inside temperature and The maxim load was larger, but the heat consuming and the set-back period were same quantity [21]**.** The optimal insulation width giving to the cooling requirements for buildings in a hot climate was determined. An implied narrow difference was investigated a method under constant periodic situations for dissimilar wall directions thru the summer season In Antalya city, Turkey. For this resolution, a computer program established in Mat lab was used. First, thermal the properties for example convection, cooling transmission, time difference, and factor decreases were calculated. It was seen that for the cooling season and the minimum value of insulation was optimal. The thickness of the north facing wall was obtained which had a minimum load cooling while the thickness was higher. The eastern and western walls were obtained to provide maximum cooling load. The outcomes showed the cooling load season at economical orientation was the optimal insulation width of 3.1 cm. The results gotten and compared with the grade of days, and the peak load be contingent on the set-back period of the interior temperature [22]**.**

2.3 Absorption Cooling System

The greatest general cooling and air-conditioning systems are those that work on refrigerant absorption systems. These systems are famous because they are dependable and relatively low-priced and their technologies have a wide scope. However, these systems use high-quality energy (mechanical or electrical) in their operation. Separately from this, the modern development using such as systems by the traditional action of absorption systems is an effect on the ozone layer and the phenomenon of the global warm has forced scientific researchers to search for new systems of cooling systems. Analytical study of the absorption cooling systems conducted. Through the using of the first two laws of the thermodynamics lower and upper bounds for coefficient of performance (COP) cooling absorption system from the upper and lower limits, where they depend on the situation temperature of the system components, also relied on the thermal properties and properties of refrigeration, And their combinations. The aim of that study was to design and study a lithium bromide-absorption (LiP-H2O) absorption refrigerator with 5.25 kW card of capacity. Several stages of the cooling system were used with design evaporator, absorption, heat exchanger solution, generator, condenser and final cup was calculated. The results showed the effect parameters like absorber ,condenser and evaporator to in cares the system COP [23]. The application the firstly and secondly laws of upper and lower thermodynamics derives the limits of coefficient of performance (COP) cooling system absorption. These higher and minor limits, as well as being dependent on conservation temperatures of sequence components, also depended on the characteristics of refrigerants, sorbents, and their mixes. The aim of that study was to compute the result of one vapor bromide water absorption for the refrigerator system with a size of 7kW. Multi stages of the cooling system were shown with evaporator design, absorption, heat exchanger solution, condenser and generator at the end the performance coefficient was designed. The results were shown all parameters condenser ,evaporator absorber and generator mostly influence of the system of Vapor absorption refrigeration COP [24]. The employ of another absorbent used in absorption cooling cycles to change the absorbent now employed in this type of engines, lithium bromide, was considered. The alternative system was consisted of absorbent (LiBr: CHO_2K=2:1 by mass ratio) and the refrigerant (H_2O). To liken the both systems performance, a program to simulate theoretical absorption cycles from empirical data was established. By incomes of the database, the efficiency of absorption cycles was evaluated. In addition, the influence of changes in the operating conditions on the performance of each cycle was analyzed. The study allows an vision into the internal parameters and into the possible behavior for extra simple conditions than those studied [25]**.** Thermodynamic demonstrating and second law studied of a small-size cogeneration system contains about 5 tons of cooling system capacities joined by a thermocouple heat siphon to 28 kW gas turbine natural gas. The planned configuration changes the high temperature source of the coolant absorption, changing the original normal gas fiery system. The computational algorithm calculated to study the global efficiency of combined cooling and electricity plant and refrigerant system performance strictures. The consequences showed the consistency of good performance and planned perfect of the co-generation system. The heat source efficiency of combined cooling system and electricity plant was about 41%, representing a 67% rise for one present of natural gas turbine [26]. The thermodynamic law of vapor absorption systems achieved mono-phase with ammonia-water refrigerant and lithium-bromide water (H_2O-LEPR and NH_3-H_2O). The system contains of triple heat exchangers for example, liquid refrigerant, solution and thermal exchanger solution of chillers. The calculating and designing model developed by a computer program and developed by visual programming language. The designer program allowed to study in the part of the working effect situation, efficient pump solution and the efficiency of heat exchangers on system performance. That study displayed the detailed program package and the thermodynamic study [27]. By waste heat the thermal and tax benefits of single absorption of the lithium bromide water exemplified. The aim of that work designed a default water absorption of the lithium bromide cooling system using the waste heat from any source. Different parts of absorption system were estimated like heat exchanger solution part, evaporator part, condenser and generator. Energy consuming and energy storing were calculated in terms of fuel and energy. The total heat transfer coefficient and the COP for heat exchanger were measured. The crises of energy and global warming had brought the benefits of their restoration to cooling systems driven by heat from the cooling and air conditioning. Refrigerator water absorption of lithium bromide was one of the favorites because of the following specific reasons that could be thermally driven by gas, solar and geothermal energy in addition to waste heat. Which helps to decrease the CO_2 emissions. The Libre-H_2O refrigerator absorption was cooling capabilities ranging from minor residential to big-scale profitable or even industrial refrigeration needs. The outcomes showed a coefficient of performance (COP) varies somewhat (0.65-0.75) and the COP value of the system rise carefully as the generator temperature increased but the system exergy efficiency degreased with the increase in the temperature in generator part. It was also shown that the (COP) value of the system increases with increase the temperature in evaporator part [28]. The numerical simulations for absorption system was estimated to examine the dynamic appearances of the system and performances when it was working to provide shared hot water and air conditioning supply to a hotel positioned near the Yangtze River in China. The outside temperature measured between 29 & 38 °C, the maximum value load air conditioning was 1450 kW, the full capacity of air conditioning was 19,890 kWh and 50 °C of warm water ability of the shower was 20 Ton that needed about 721 kWh of heat on a given day. Under these circumstances, the operation characteristics of system were simulated in the context of full and partial storage strategy. The simulation results expected dynamic characteristics and system performance counting temperature and focus of working liquid. The results also show that the system COP was 3.09 and 3.26 respectively under 2 storage strategies. The efficiency of isentropic water vapor compressor was 0.6% and the Model elation outcomes were very suitable to know the system [29]. The performance of cooling cycle was assessed under mutable bases of power - electricity, conventional type and renewable energy sources. Ammonia water absorption cooling system used and the temperatures measured at every part in the cooling system cycle, for example generators, condensers, evaporators and condensers using the suction system. The outcome experiment was shown when the cooling cycle worked by electric energy; the performance coefficient was different from 0.694 to 1.032 at period the test time. The generator temperature was different from 48.1 Ċ to 101.5 Ċ with a rated efficiency of 57.1% and a rate performance factor of 0.78. When methane was used to drive cooling system cycle, the performance coefficient value was between 0.686 and 0.94 at a generator temperature of 123.3 °C and 127.4 °C and rate efficiency of 40.02% with a 0.735 performance value. When the cooling cycle system was driven by solar thermal power, the performance data was become 0.801 of the generator temperature of 91°C **[30]**.

2.4 Waste Gases of Gas Turbine

The system of waste heat recovery produces energy by taking advantage of the lost thermal energy to the environments of thermal processes, at without any added fuel input. The waste exhaust heat gases has various application that use with our live. For marine vessels, about 50 % of total fuel capacity supplied to the diesel power plant is lost to the surrounding areas. While the total quantity of wasted energy is large, the study measured the potential to recover exhaust heat from a potential of fuel cell gas turbine (FCGT) hybrid to supply thermal energy for building air-conditioning. Initial calculations determined that fuel cell gas turbine FCGT hybrid exhaust may still contain enough thermal energy to provide input to a heating ventilation air conditioning (HVAC) system based on absorption chiller cycle. The amount of steam that could be generated in a heat recovery boiler from the exhaust of a gas turbine system was estimated. A representative urban office building using standard commercial construction was modeled to determine the cooling load characteristics of a typical building in the Northeast, cooling load of one building was 461.36 KW and it was estimated by TRANE program [31]. The usage of gas turbine exhaust gas waste-heat powered, single-effect water lithium bromide (H_2O – LiBr) absorption chillers was thermo-economically evaluated for gas turbine compressor inlet air cooling scheme, with particular applicability to Middle East. The results showed that in extreme ambient conditions representative of summer in the Persian Gulf (i.e., 55 °C, 80% RH), three steam-fired, single-effect (H_2O - LiBr) absorption chillers utilizing 17 MW of gas turbine exhaust gases, could provide 12.3 MW of cooling to cool compressor inlet air to 10 °C. The additional electricity generated through gas turbine compressor inlet air cooling using the waste heat powered absorption refrigeration scheme is of approximately 5264 MWh per year. The study proposed that waste heat absorption cooling cycle is an attractive solution to improve electrical power generation in Middle East through gas turbine inlet air cooling, both in terms of thermodynamic and economic feasibility [32]. To improve on-site power generation capacity and efficiency in process facilities, the thermal coupling of an industrial gas turbine cycle with a bottoming organic Rankin cycle for power plant flue gas waste heat recovery in a process facility was investigated. Using pentafluoropropane (R245fa) as heat carrier in the Rankin cycle, 5.2 MW of additional electric power was generated, enhancing on-site power generation capacity and energy efficiency by approximately 23% and 6%, respectively. The overall energy efficiencies of the waste heat recovery system were estimated at 9 %. Primary energy savings of approximately 1.3 million standard cubic feet per year of natural gas, could be realized with the proposed flue gas waste heat recovery system based on subsidized industrial electricity tariffs in the United Arab Emirates UAE [33].

CHAPTER THREE

METHODOLOGY

3.1 Introduction

This chapter will explain the methodology of estimating the cooling heat load of the building, the size of the absorption system and the gas turbine exhaust gases.

3.2 Study Region

It is substantial to study the climatic condition of any region to know the typical behavior of buildings in multiple seasons and months. The Salahuddin city region in Iraq is dry and has a hot climate during the summer and it is cold and wet during the winter. Moreover, there are large temperature differences between the day and night between the summer and the winter such that maximum temperatures happen in July and minimum temperatures happen in January. Most building construction in Iraq consists of concrete blocks, common bricks and celling concrete, which does not use insulation systems within constructions.

The power station actuality used in this study has four Siemens type V94.2 gas turbine units generating a power of 134 MW each at a nominal power output per unit at a rate of full oil consumption of 7.8 to 9.9 kg/sec [34].

3.3 Calculating the Cooling Load of Building

Before guessing the cooling load of any building there are some basic data that are needed to find the exact cooling load. These data include building orientation, climate data and building construction. The exact cooling loads are derived from exact information of the office building.

3.3.1 Orientation of The building

The orientation of an office building has important influence on energy consumption. The quantity, orientation and proportions of glazed surfaces are three important parameters. The large surface walls and glass surfaces on the southern side of the building contribute to the increase of the heat absorbing capacity of the building structures in the summer season. The two-story building considered in this thesis is situated in Salahuddin city (**Iraq**), which is located at 35°N latitude, 43° longitude and 110 meters above sea level.

3.3.2 Climate Condition

Table 3.1 shows the monthly outdoor temperatures and relative humidity in Salahuddin city in Iraq as having hot weather through the summer and cold weather during the winter. Maximum temperatures occur in July and the lowermost temperatures in January. The lowest and highest temperatures are 1.5°C and 49.5°C, respectively. Table 3.1 presents the high temperature, low temperature and relative humidity (RH) of Salahuddin city in 2016**[35]**.

Table 3.1: Maximum and minimum temperatures with relative humidity.

Month	Jan.	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Max temp.	20.8	27	27.9	39.6	41.9	48.2	49.5	49	44.4	38.2	31	18
Min temp.	6.2	1.5	2.8	6.7	15.8	19.6	24	23.9	15.2	13.6	0.0	2.7
RH%	93	88	71	61	34	24	15.5	22	29	37	43	98

3.3.3 Building Structures.

The building used in this study has one floor with the ground floor with 29 rooms. The external walls of the office building consist of 200-millimeter common bricks, 13-millimeter cement plaster and 13-millimeter gypsum plaster. The interior partitions of the office building consist of 200-millimeter bricks with 13 mm of gypsum plaster on both sides. The roofs of the office building consist of 200-millimeter high density concrete, 100 mm of sand, and 40 mm of cement Shtyger and 10 mm of gypsum plaster. Windows construction is that of an aluminum frame with single planes of 3-millimeter thick glass.

3.3.4 Load Components

The quantity of heat necessary to be extracted from the inside of the office building in order for it to be at the wanted temperature (22.5°C to 26°C) and RH (50%) with the air conditioning equipment is known as the cooling heat load. The heat load of the office building is generated from the outdoor load and internal load [36].

3.3.4.1 External Load and Internal Load

External load is produced from the moved heat energy coming from the outside hot space to the inside of the space. The heat enters the zone from conduction through exterior walls, roofs. Additional sources of cooling heat load are generated from solar radiation out of the windows, doors, and ventilation and filtration systems. The heat load generated from electrical equipment, lights and any occupants are internal sources of heat load. Figure (3.1) illustrates the load components[36].

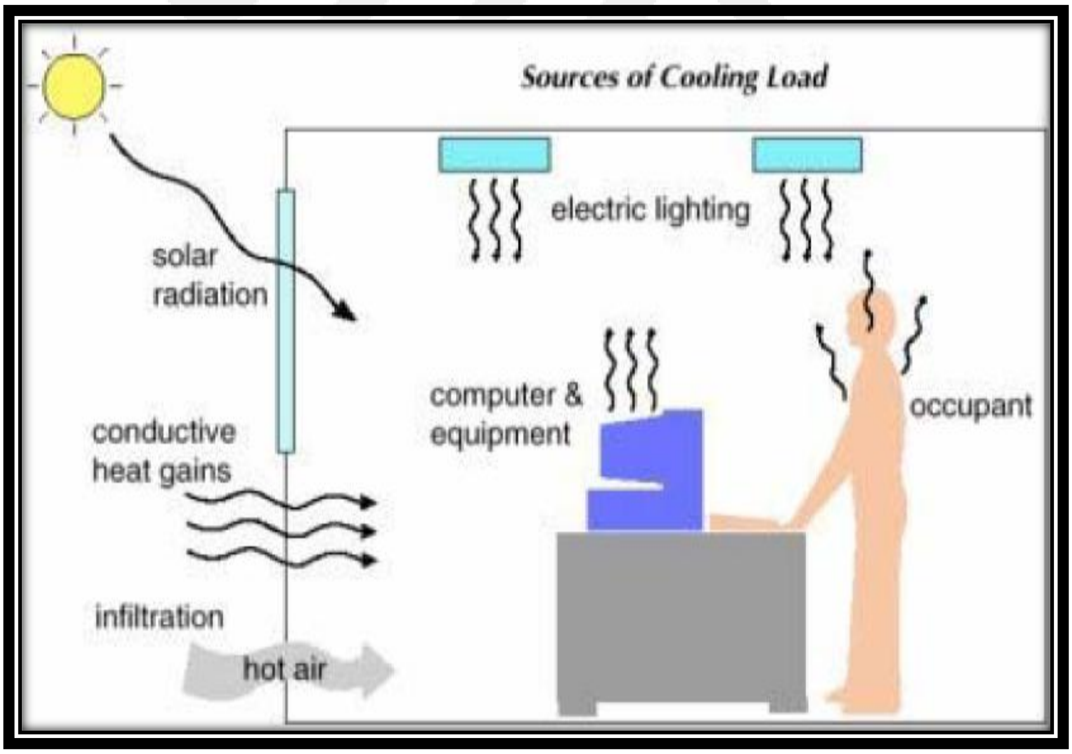


Figure 3.1: load component[36].

3.3.4.1.1 Heat Gain through Walls and Roofs Surface

External walls and roofs are most probably the greatest important parts of an office building envelope. A high quality external wall and roof should be strong, stable, moisture resistant, durable and heat resistant. They should similarly be fire safe; these are the main functional requirements of an external wall and roof .The major part of the sensible cooling load of the building constitutes walls, roofs, floors, doors and windows. The difference in temperature across the opaque surfaces causes heat transfer during wall and roof surfaces. As shown in Figure (3.2), heat load is transferred from the exterior air mainly by convection to the outside surface. Then, the heat load is transferred by plugging through the wall structure to the inner surface.

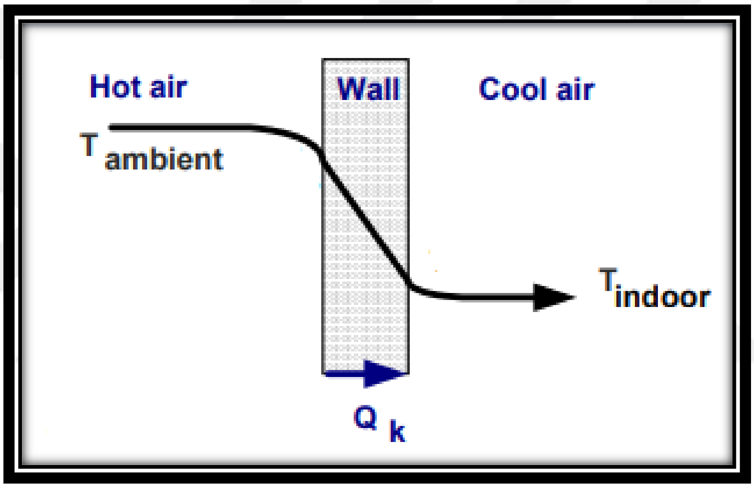


Figure 3.2: Heat transfer through wall surface[36]**.**

The heat transfer from outside to inside is predicted with the equation **[3]**.

$$Q = U \times A \times (CLTD)_c \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3.1)$$

Where:

Q = Heat gain through the walls (W)

U = over all heat transfer coefficient (W/m- Ċ)

CLTDc = cooling load temperature difference correction (°C)

A = wall surface area (m^2)

3.3.4.1.1.1 Overall Heat Transfer Coefficient

The wall, floor and ceiling are made up of layers of dissimilar materials, as shown in Figure 3.3 Then the overall heat transfer coefficient 'U' can be calculated using equation (3.2) **[11]**. The fig 3.3 shows the thickness layers of walls constructions.

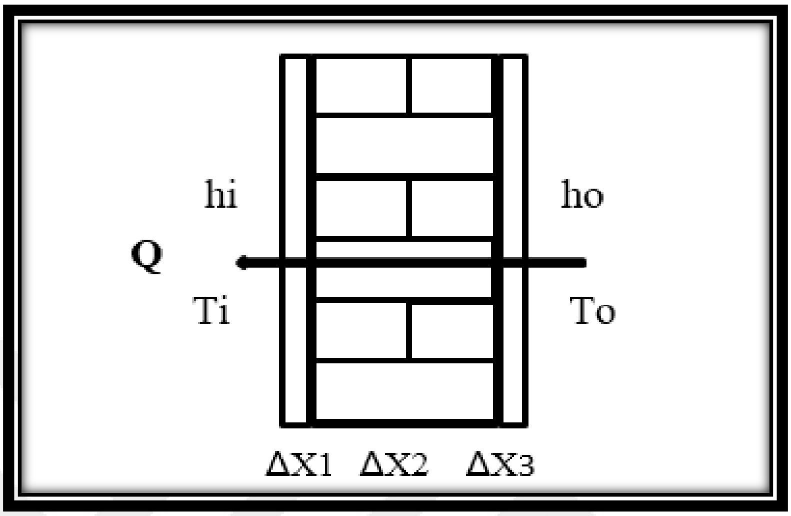


Figure 3.3: Heat transfer through the layers of wall.

$$U = \frac{1}{1/ho + x1/ka + \cdots + 1/hi} \quad \text{.....................................} (3.2)$$

Where:

U= Over all coefficient (w/m^2.c).

ho= Outside heat transfer coefficient w/m^2.c.

x_1= Surface thanks (m).

ka= Thermal conductivity(w/m.c).

hi= Inside heat transfer coefficient w/m^2.c.

The building walls usually consist of various materials such as common bricks, concrete and cement plaster. Heat load transfer through these types of wall is very complex because it involves simultaneous heat load transfer by heat conduction, convection and radiation, as shown in Figure 3.3. The materials in the wall have dissimilar types of thermal properties. The physical thermal properties of the office building materials were measured and presented in ASHRAE and other handbooks **[37].** Table 3.2 shows the thermal properties of usually used building materials.

Table 3.2: Value of thermal conductivity of the stricture material.

Material	K: Thermal conductivity(w/m.c)
Common brick	0.72
Face brick	1.33
concrete	1.73
Cement plaster	0.72
Gypsum plaster	0.81
Cork	0.036
Sand	1.72
Stone	1.8
glass	0.78

Table 3.3 shows for the thermal values for the wall and roof transferring heat through the outside air to the outside surface of the wall and inside the surface to the inside air. Therefore, the air layers coefficient depend on the airspeed outside and inside the wall. The value of heat transfer coefficient depends on the speed of movement of the outer air. The outside heat transfer coefficient (ho) is higher than the inside coefficient (hi) because of the limited air movement in the space.

Table 3.3: Value of surface conductivity of the air film [37].

Air velocity	Surface position	Direction of heat flow	Heat transfer coefficient $w/m^2.c$
Still air	Horizontal	Up	9.4
Still air	Horizontal	Down	6.3
Still air	Vertical	Horizontal	8.5
25 km/h	Any position	Any	35
12 km/h	Any position	Any	23.3

3.3.4.1.1.2 Cooling Load Temperature Difference (CLTD*)*

CLTD is the difference in the effective temperature through the wall or roof. The CLTD represents the impact of radioactive heat like temperature difference. ASHREA has developed the values of CLTD for exterior roofs and walls depending on the kind of solar radiation variation from 40°N latitude on 21 July with 29.4°C outside temperature and 24.5°C indoor temperature and building materials usually used in North America. The cooling load temperature difference value for conduction through walls takes from the table (3.4), and it can correct the value by equation (3.3) **[37]**.

Table 3.4: Equivalent CLTD of B type of the wall constructions **[37]**.

North latitude wall facing	08	09	10	11	12	13	14	15	16	17
N	6	5	5	5	5	5	5	5	6	6
NE	7	7	7	8	8	9	9	10	10	11
E	8	8	9	9	10	12	13	13	14	14
SE	8	8	8	8	9	10	11	12	13	13
S	7	7	6	6	6	6	7	8	9	9
SW	10	9	9	8	8	7	7	8	9	9
W	11	10	9	9	8	8	8	8	8	9
NW	9	8	7	7	7	6	6	7	7	8

The cooling load temperature difference value for conduction through roofs takes from the table (3.5), and it can correct the value by equation (3.3).

Table 3.5: Equivalent CLTD of the roof type 8 with suspended ceiling[37]**.**

Solar time	8	9	10	11	12	13	14	15	16	17
CLTD	10	9	8	8	8	9	11	14	16	19

For various design conditions, the CLTD values presented in Tables 3.4 and 3.5 should be corrected before being used in heat transfer equations because it is out design condition of CLTD. The consequent equation can be used for the correct CLTD value using Eq. (3.3) [11]**.**

$$CLTD_c = (CLTD + LM) \times K + (25.5 - Ti) + (To - 29.4) \quad \text{……..…….. (3.3)}$$

Where:

To = Average outside temperature on design day (°C)

Ti = Inside design temperature (°C)

LM = Latitude month correction given on table (3.4)

K = Correction factor depends on building color and building environment.

K = 1 for dark, light colors and the building in industrial areas and 0.5 for medium colors **[37].**

When the building location out from 40°N latitude on 21 July , 29.4°C outside temperature , 24.5°C inside temperature and building materials doesn't in North America. We should take latitude month correction (LM) on table 3.6 and apply the value on equation 3.3 to calculate ($CLTD_C$) value.

Table 3.6: Latitude month correction (LM) value for 35 °N for walls and roofs **[37].**

Moth	N	NE/NW	E/w	SE/SW	S	HORIZ
Dec	2.9	-5.5	-4.8	0.7	6.2	-10.2
Jan/Nov	-2.7	-5.2	-4.6	0.9	6.4	-9.1
Feb/Oct	-2.4	-4	-2.6	2	6.3	-6.3
March/Sept	-1.8	-2.4	0.0	1.8	4.4	-3.3
Apr/Aug	-1.1	-0.7	0.0	0.4	1.1	0.3
May/July	0.3	0.3	0.0	-0.3	-0.8	0.5
June	0.5	0.9	0.0	-0.7	-1.6	1.1

3.3.4.1.2 Heat through Glass

The convection thermal load is transferred across the glass due to solar radiation. Thermal gains across glass areas form a large part of the cooling load on the cooling device. This can be directly in the form of sunlight or diffuse radiation due to the reflection of other objects outside. The temperature transmitted through a glass window cunts on the wavelength of the radiation, the physical properties the substance and the chemical properties of the glass. Part of the radiation is absorbed and part of it is reflected and the remainder transmitted. Heat is transferred from the glass in two ways: gaining heat transmission and gaining solar heat through the glass. The two equations (Eqs. (3.4) and (3.5)) are used to estimate heat gain through glass **[11].**

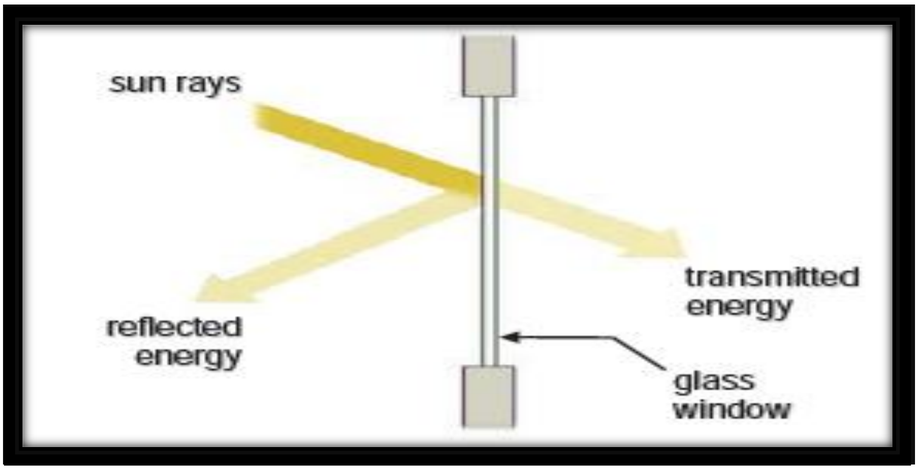


Figure 3.4: Heat gain through glass [36].

Heat gain through glass:

$$Q = UA\,(CLTD)_c \quad \text{……..…………………..…..…..…… (3.4)}$$

$$CLTD_c = CLTD + (25.5 - Ti) + (To - 29.4) \quad \text{………..……... (3.5)}$$

Where:

To = outside design temperature (°C)

Ti = inside design temperature (°C)

The cooling load temperature difference value for conduction through glass takes from the table (3.7), and it can correct the value by equation (3.5).

Table 3.7: CLTD through glass [37].

Solar time	**08**	**09**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**
CLTD	0	1	2	4	5	7	7	8	8	7

Heat gain by solar radiation:

$$Q = A \times SHG_{max} \times SC \times CLF \quad \text{…………………………..…… (3.6)}$$

SHG_{max}: Maximum solar heat gain factor (w/m^2).

SC: Shading coefficient, SC=1.

CLF: Cooling load factor.

Table (3.8) gives the value of sensible heat gain (SHG) factors for (35 degree) north latitude at different months and different orientations.

Table 3.8: Value of SHG_{max} for 35 N latitude **[37]**.

Months	Maximum solar heat gain factors W/m^2					
	N	NE/NW	E/W	SE/SW	S	HORIZ
Jan.	71	80	543	781	790	505
Feb.	83	186	623	764	723	642
Mar.	97	319	707	720	593	762
Apr.	111	456	712	611	410	834
May.	120	532	694	513	278	862
Jun.	146	553	677	465	230	864
Jul.	124	523	680	499	270	850
Aug.	115	438	689	590	397	817
Sep.	100	306	670	707	578	737
Oct.	86	183	596	759	702	629
Nov.	71	80	522	769	778	502
Dec.	65	65	485	764	800	446

The maximum Factor of solar heat gain ($SHGF_{maxs}$), the cooling load factor CLF and the month depend on the latitude of the region and the direction of the window. It is a complete solar heat transfer that directly contains the transfer of energy and indirectly transfers the energy heat gain. The Cooling Load Factor (CLF) for different months and directions are shown in Table 3.9.

Table 3.9: Cooling load factor (CLF) for glass with interior shading coefficient [**37]**

Wall facing	Solar time hr.									
	08	09	10	11	12	13	14	15	16	17
N	0.65	0.73	0.8	0.86	0.89	0.89	0.86	0.82	0.75	0.78
NE	0.74	0.58	0.73	0.29	0.27	0.26	0.24	0.22	0.20	0.16
E	0.80	0.76	0.62	0.41	0.27	0.24	0.22	0.20	0.17	0.14
SE	0.74	0.81	0.79	0.68	0.49	0.33	0.28	0.25	0.22	0.18
S	0.23	0.38	0.58	0.75	0.83	0.80	0.68	0.50	0.35	0.27
SW	0.14	0.16	0.19	0.22	0.38	0.59	0.75	0.83	0.81	0.69
W	0.11	0.13	0.15	0.16	0.17	0.31	0.53	0.72	0.82	0.81
NW	0.14	0.17	0.19	0.20	0.21	0.22	0.30	0.52	0.73	0.82
HORIZ	0.44	0.59	0.72	0.81	0.85	0.85	0.81	0.71	0.58	0.42

3.3.4.1.3 Heat Generation from Occupant

Human bodies in a coolant space forms have a sensible cooling load and a latent cooling load. In an air-conditioned room, a reasonable heat load is attributable to differences in temperature between human bodies and the heat gained from the occupants by the air in the room, which depends on the average number of people or bodies expected to be present in an air-conditioned area. The thermal heat load generated by every person depends on the person's activity inside the space. The rate of heat generation from a human body or occupant is calculated with the following equations **[11]**.

$$Q_{S,\ person} = q_{s,\ person} \times N \times CLF \quad \text{(3.7)}$$

Latent heat gain from occupants:

$$Q_{L,\ person} = q_{l,\ person} \times N \times CLF \quad \text{(3.8)}$$

Where:

$Q_{S,\ person}$ = Sensible heat gain /person (w).

$Q_{L\ person}$ = Latent heat gain/person (w).

N= Total number of people present in conditioned space.

CLF = Cooling load factor.

CLF is the cooling load factor for the human and it value depends on the people spending time in the air-conditioned space when the time intervention since the first entry. The value is equal to 1 if the air cooling system does not worked 24 hours by Day. Table 3.10 shows the rate the heat gain from people at malty activity level based on inside room temperate.

Table 3.10: Rate of heat gain from occupant at activity space **[38]**.

Degree of activity	Typical application	Sensible heat (W)	Latent heat (W)
Moderately active	**Office ,hotels, apartments**	75	55
Standing ,light work,wlking	**Departmental store, retail store**	75	55
Walking ,standing	**Drug store, Bank**	75	70
Walking8.4 km/h,light machine work	**Factory.**	110	185
Heavy work	**Factory**	170	255

3.3.4.1.4 Heat Gain from Lighting Equipment

Electric lights generate a sensible cooling heat load equivalent to the amount of electrical power consumed. The heat energy gained from electrical lighting depends on the classification of the light power in watts. Its use of the factors and the allowance factor as an approximate calculation is known as heat gain by Eq. (3.9) [11] .The value of power of lighting is depended on actively inside the space and it takes from table (3.11).

$$Q_{Light} = \text{Light input in watt} \times A \times FC \times CL \quad (3.9)$$

Where:

Light input = Light wattage from the table

A= Area of the space (m^2)

CLF=1, if operation is 24 hours or cooling is not working at night

FC=1.2 for special ballast allowance factor fluorescent fixture **[11].**

Table 3.11: Typical lighting load **[39].**

Type of using	Lighting power W/m^2
Office room	12
Conference /Meeting /Multipurpose	14
Classroom/Lecture /Training	15
Audience Seating area	10
Dining area	10
Laboratory	15
Rest room	10
Electrical/ Mechanical room	16
Workshop	20
Library, reading area	13
Car park	5

3.3.4.1.5 Heat Gain from Office Equipment

The heat load generated from any equipment in a space is determined by the wattage of the equipment. This heat load is calculated with the following equation **[11]**.

$$Q_{equip.} = \text{Total wattage of equipment} \times CLF \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.10)$$

CLF=1.0, if operation is 24 hours or cooling is not working at night or weekends.

Communal office equipment, such as computers, printers, fax machines and photocopiers, are used even when not in use. Desktop computers or other devices in the space also add heat into the air-conditioned area. Table 3.12 shows the power requirements for office equipment **[40]**.

Table 3.12: Heat generate for office equipment (watts) **[39].**

Appliance		Power (W)
computer	**15'' monitor**	110
	17'' monitor	125
	19'' monitor	135
Laser printer	**desktop**	130
	Small office	320
	Large office	550
Fax machine		30
Facsimile machine		25
Image scanner		50
Dot matrix printer		50
Desktop copier		400
Office copier		1100
Coffee brewer		265(Sensible) - 65(Latent)
Refrigerator small size		690

3.3.4.1.6 Loads from Ventilation and Infiltration

Infiltration is air that enters through cracks and interstices around doors, windows, through floors and walls of any type of structure or building. The amount of infiltration depends on the type of construction, the workmanship and the condition of the space. The amount of infiltration is estimated using Eq. (3.11) and the AC value is taken from Table 3.13. The ventilation load is one of the main and important parts of the cooling load on a building space. The amount of air required for load heat ventilation depends on a number of factors, such as application, type of activity inside the space and any combustion sources. The minimum heat load ventilation is calculated using Eq. (3.12), (3.13) and Table 3.14 **[40]**.

$$V_{inf} = \frac{valume\ of\ space \times AC}{60}\ m^3/min \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.11)$$

Where:

AC = Number of air changes value/hr from table (3.13)

Table 3.13: Number of air changes per hours for infiltration load **[37].**

Kind of room or building	Number of air change per hour (A_C)
Room with no window or outside doors	0.5 to 0.75
Room , one wall exposed	1
Room, tow walls exposed	1.5
Room, three walls exposed	2
Room, four walls exposed	2
Entrance halls	2 to 3

Table 3.14: Outdoor air requirements for ventilation per person and per area **[41].**

function	People outdoor air rate m^3/s. person	Area outdoor air rate L/s. m^2
Offices space	0.0025	0.3
Operation theaters	0.0038	0.3
Lobbies	0.0025	0.3
Class rooms/Lecture	0.0038	0.3
Meeting places	0.0025	0.3
Multi-purpose assembly	0.0025	0.3

$$Q_{s,\ infil} = 1.22 \times V_{infil,\ Vent} \times (T_o - T_i) \text{ kw} \quad (3.12)$$

Where:

$V_{inf\ or\ vent}$ = Amount of air infiltration

T_o = Outside temperature (Ċ)

T_i = Inside temperature

$$Q_{l,\ infil,\ Vent} = 3010 \times V_{infil,\ Vent} \times (W_O - W_I) \text{ kw} \quad (3.13)$$

Where:

V_{inf} = Amount of infiltration air

W_o= Specific humidity for outside (kg /kg of dry air).

W_i = Specific humidity for inside (kg/kg of dry air).

3.3.4.2 Total Loads

The total load of the room is the summation of the sensible cooling heat load and the latent cooling heat load. The sensible cooling heat load comes from the summation of the walls load, ceilings load, glass, infiltration load, ventilation load, lights load and the fans, and the latent cooling heat load comes through the infiltration load, the ventilation load of people and appliances [36].

$$TCL = SCL + LCL \quad \text{………………..…...……………..} \quad (3.14)$$

Where:

TCL = Total cooling load (watts).
SCL = Sensible cooling load (watts).
LCL = Latent cooling load (watts).

3.3.4.2.1 Total Sensible Cooling Loads (TSCL)

The sensible cooling load is the heat coming from every type of sensible heat sources of the space, which comes from the summation of the walls load and the ceilings load. The sensible heat load is gained from any occupants, and the sensible heat load is gained from the glass, the infiltration air, the ventilation and the lights [36]

$$TSCL = Q_{walls} + Q_{glass} + Q_{ligting} + Q_{S,occupan} + Q_{S,equipment} + Q_{S,ventilation} + Q_{S,infiltration} \quad \text{…………………………………………..} \quad (3.15)$$

3.3.4.2.2 Total Latent Cooling Loads (TLCL)

The latent cooling heat load is heat from all sources of the latent heat of the space, which comes from the summation of the latent cooling heat load gain due to the infiltration load. The latent heat load is gained from the ventilations, the people and appliances **[36]**

$$TLCL = Q_{L,\,occupant} + Q_{L,\,equipment} + Q_{L,\,ventilation} + Q_{L,\,infiltration} \quad \text{……...……..} \quad (3.16)$$

3.3.4.3 Sensible Heat Factor (SHF)

The Sensible Heat Factor (SHF) is the ratio of total sensible cooling load to the total cooling load of the air conditioning apparatus **[36].**

$$SHF= \frac{SCL}{TCL} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3.17)}$$

Where:

SHF= Sensible heat factor.

SCL= Sensible cooling load.

TCL= Total cooling load.

3.3.4.4 Total Cooling Load in Tons

We can convert the total cooling load from watts to tons with Eq. (3.18**) [36].**

$$\text{Total cooling load in tons} = \frac{total\ cooling\ load\ in\ watts}{3516} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3.18)}$$

3.3.5 Air Supply

Design and study of air conditioning systems includes the selection of appropriate inside and outside design conditions, the calculation of the desired capacity of heating and cooling equipment, the selection of appropriate cooling/heating systems, the selection of supply conditions, the design of the air conveyance and supply systems, and so on. Generally, the inputs are the building specifications and the usage pattern and any other special requirements. Based on the following known information sensible room, latent heat gains, outside and inside design conditions, ventilation (outside) air requirements and either CFM or DB temperature of the supply' one of these is selected and the other is then determined from the sensible heat equation. However, both must be in a range that is considered satisfactory for “good practice.” The feeding air temperature values are usually selected so that the temperature difference between the room and the supply air is between (15 to 30) $^{\circ}$F. Factors like the kind and location of air supply outlets will affect the selected temperature difference [42]**.**

$$Vs = \frac{Qs}{1.22(Tr-Ts)} \quad \text{……………...……...…...…………} \quad (3.19)$$

Where:

Qs= Sensible heat gain (watts)

Vs = Air supply to the space (m^3/sec).

q_s = Sum sensible heat gain of the space (watts)

Tr = Air temperature of room (Ċ).

Ts = Supply air temperature (Ċ).

3.4 Absorption Cooling System Cycle

Steam absorption cooling systems are used with large-scale water-lithium bromide in large size air conditioning systems cycle. In these systems, the water is used as a refrigerant and the bromide is used as the absorbent. Since water is used as a refrigerant in these systems, cooling cannot be providing at temperatures below zero. It is only used with applications requiring cooling at temperatures over zero degree. These systems cycle are used for air conditioning applications and examination of these systems is relatively simple since the steam vapor generated in the generator part is almost pure water [36]. Figure 3.5 shows gas turbine cycle with waste absorption cooling system cycle.

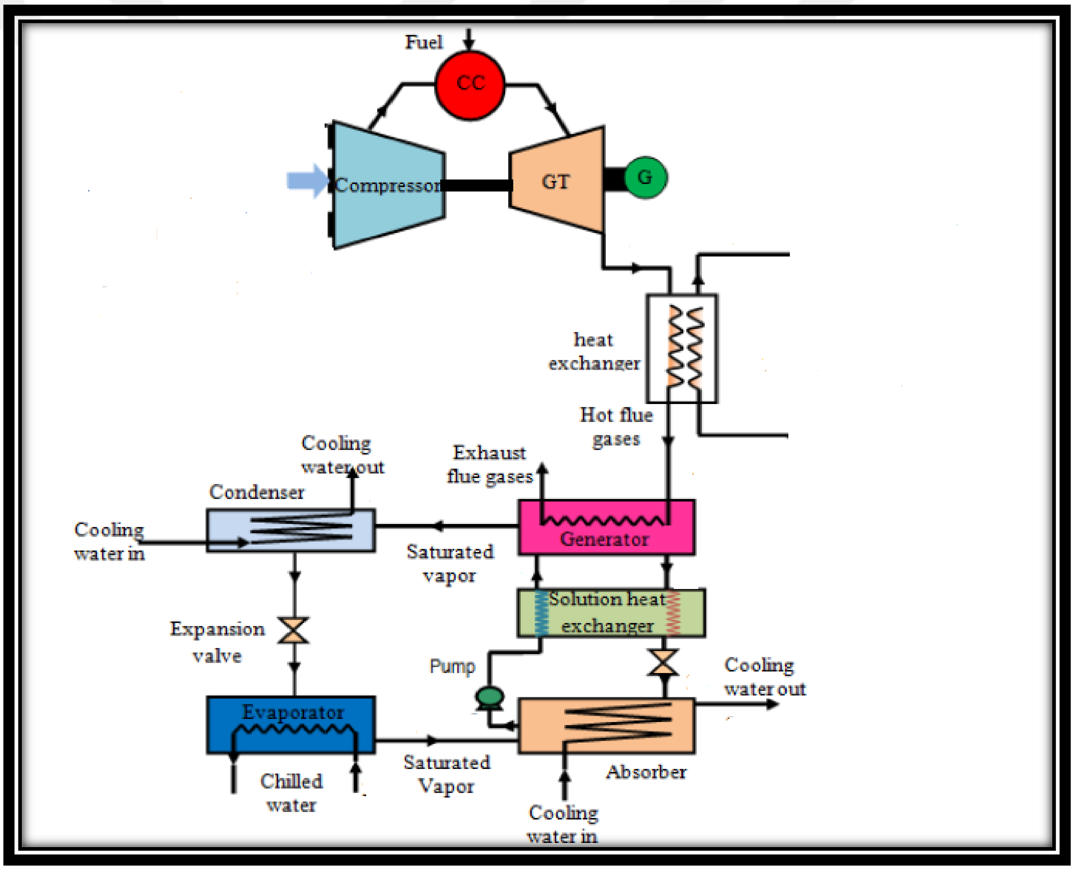


Figure 3.5: Gas turbine cycle with waste gases absorption cooling system cycle.

3.4.1 Components and Analysis of a Vapour Absorption Cooling System

Figure 3.6 shows the schematic of the system representing several state points. Study state flow study of the system is carried out with the flowing assumption **[26]**.

i. Steady state and steady flow.

ii. Changes in potential and kinetic energies through each section are negligible.

iii. No pressure drops due to friction across the system cycle.

iv. Only fresh refrigerant boils inside the generator of the system.

The Figure 3.6 shows the absorption system detail parts.

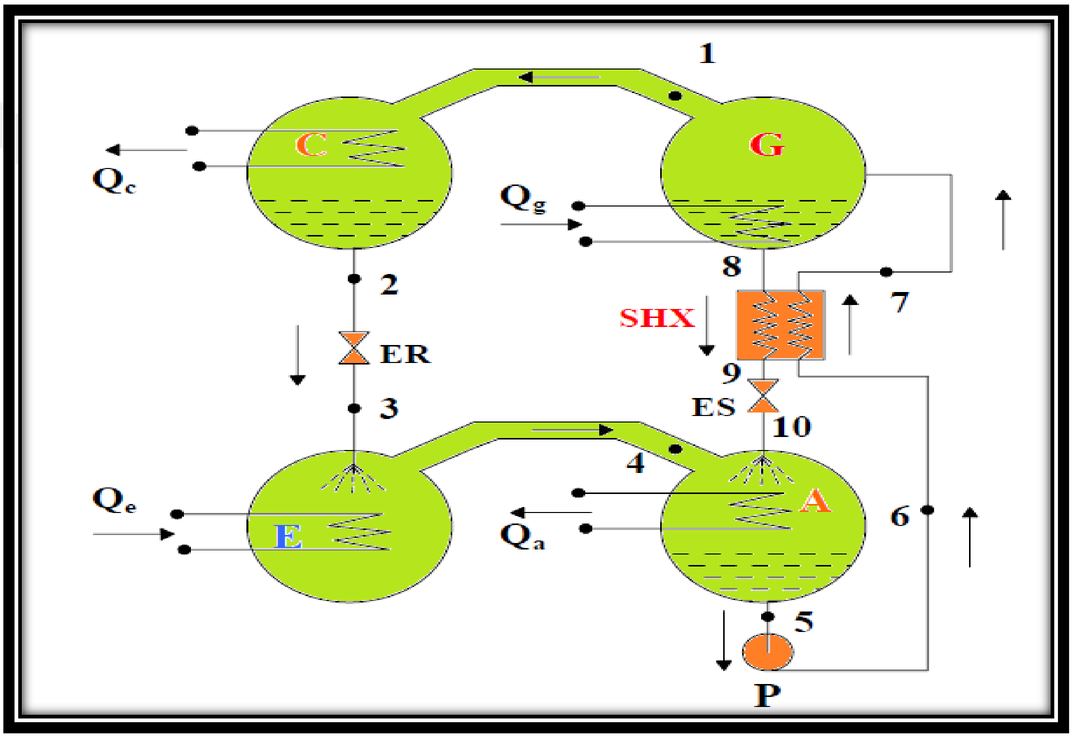


Figure 3.6: Absorption system cycle **[36].**

3.4.1.1 Generator:

The objective of the generator is to transport the refrigerant vapor of the system to parts of the absorption system. It completes this process by detach the refrigerant from the solution. In the generator part, the solution falls vertically above horizontal tubes and a high temperature energy normally of steam or hot water flowing through the tubes. The refrigerant solution absorbs a quantity of heat from the hot steam or water producing the refrigerant to boil and then detach from the absorbent solution part. The refrigerant in the system cycle is boiled afar and the absorbent solution converts additional concentrated. The concentrated refrigerant absorbent solution revenues to the absorber section and the refrigerant vapor travels to the condenser. Heat is added to the generator. The heat added to the generator is calculated using Eq. (3.20).The lithium bromide circulation ratio is calculated using Eq.(3.21) [36][23].

$$Q_{gen} = m\,[(h_1 + h_7) - \lambda\,(h_8 - h_7) \quad \text{(3.20)}$$

$$\lambda = \frac{weak\ sulation\ \%}{(strong\ solutiob\% - weak\ solution\%)} \quad \text{(3.21)}$$

Where:

Q_{gen} = Heat add to generator (watt)

m = Mass of the water refrigerant (kg/sec)

h = Enthalpy of the refrigerant water or liber bromide (kJ /kg)

λ= Circulation ratio

m_{ws} = Mass flow rate of strong solution (rich in LiBr), kg/s

m_{ss} = Mass flow rate of weak solution (weak in LiBr), kg/s

3.4.1.2 Condenser

The objective of the condenser is to condense cooling fumes. Inside the condenser section, the cooling water flows inside the tubes and the hot cooling vapor flows through the around tubs. Heat is moved from the cooling vapor to the water and the refrigerant vapor condenses on the surface of the tube. The liquid coolant collects in the condenser at the bottom of the condenser before traveling to the expansion device. The water cooling system is connected to the cooling tower. The heat rejected through the condenser is calculated with Equation (3.22) **[36][23]**.

$$Q_{con} = m\,(h_1 - h_2) \quad \text{(3.22)}$$

The enthalpy of the superheated vapor in the point (1) with simple equation.

$$h_1 = 2501 + 1.88(T_{gen}.) \quad \text{(3.23)}$$

$$m_5 = m_9 = m_3 = m_4$$

Where:

Q_{con} = Heat rejected from condenser (watt)

m= Mass of refrigerant water (kg/sec)

T_{gen} = Generator temperature (°C)

3.4.1.3 Expansion Valve:

The condensed, cooling fluid flows through the expansion valve in the evaporator. The expansion device is used to keep the pressure difference between the high pressure side (condenser) and low pressure liquid flows across the expansion valve. This causes low pressure which decreases the cooling pressure to that of the evaporator. This pressure limit causes a small part of the liquid refrigerant to boil off thereby cooling the residual coolant to the coveted evaporator temperature. The cooling mixture of the liquid and steam refrigerant then flows into the evaporator. Where: $h_2 = h_3$ (throttling) **[36][23].**

3.4.1.4 Evaporator:

The major point of the evaporator is to cool the circulating water. The evaporator contains a variation of pipes that carry the water to be cooled. In the low pressure portion of the evaporator, the liquid cooling refrigerant ingests heat from the circulating water and evaporates. Refrigerated vapors formed and tend to increase pressure value of the vapors. This in turn increases the boiling temperature and it will not provide the desired cooling result. Therefore, it is required to remove the cooling heat load from the evaporator in low pressure absorption. Physically, the evaporator part and the incoming absorption are within the same casing. This allows the cooling heat load generated in the evaporator part to constantly migrate to the absorption. The heat rejected across the evaporator part is estimated with Eq. (3.24) **[36][23]**.

$$Q_{eva} = m\,(h_4 - h_3) \quad (3.24)$$

Where:

Q_{eva} = Heat rejected or add from the evaporator (watt).

m = Mass of the refrigerant (kg/sec).

3.4.1.5 Absorber:

The refrigerant inside the absorber is absorbed by the solution. The refrigerant vapor is absorbed and it alters from a vapor state to a liquid state, freeing the heat it absorbed in the evaporator part. The heat is freed from the intensification of the refrigerant vapor by its absorption in the solution, which is take away by the cooling water circulation in the absorber tube package. The weak absorbent refrigerant solution is driven to the generator part and the heat is used to operate off the refrigerant. The high value of temperature refrigerant vapors created inside the generator travel to the condenser. The heat rejected value through the absorber is calculated with Eq. (3.25) **[36][23]**.

$$Q_{abs} = m\,[(h_4 + h_5) - \lambda\,(h_{10} - h5)] \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3.25)$$

Where:

Q_{abs} = Heat rejected from absorber (watt).

m = Mass of the water or lithium bromide (kg/sec).

3.4.1.6 Solution Bump

The solution pump represents some of the absorption system parts. It is used to pump the water-lithium bromide refrigerant from the absorber to the generator. We calculate the pump work with Eq. (3.26) [36].

$$W_p = m_{ws} \times V \times (P_6 - P_5) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3.26)$$

Where:

m_{ws} = Mass of the weak solution (kg/sec).

V = Specific volume of the refrigerant (m^3/kg).

P = Pressure across the side the motor pump (kpa).

3.4.1.7 Coefficient of Performance (COP):

In the absorption system, the total refrigerating effect inside the system is dependent upon the quantity of heat load absorbed by the liquid refrigerant inside the evaporator section. The total heat energy in the system is the total of the work performed by the work pump and the heat energy supplied in the generator part. Consequently, the Coefficient of Performance (COP) of the system as theoretically known is given by COP = heat absorbed in the evaporator / (work of the pump + heat supplied in the Generator) **[36][23].**

$$\text{COP} = \frac{Qeva}{Qgen + Wp} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3.27)$$

Where:

Q_{eva} = Heat evaporator (W).

Qgen = Heat generator (W).

Wp= Work done by pump (W).

3.5 Gas Turbine

3.5.1 Gas Turbine Specifications

The gas turbine used in this study is a Siemens gas turbine V94.2 Table 3.15 shows the specifications of the lighting fuel oil of the gas turbine.

Table 3.15: Gas turbine specification [34].

Electric output at 27Ċ	134.7 MW
Exhaust gas temperature	490 Ċ
Exhaust gas flow rate	480 kg/sec
Thermal efficiency	33 %
Full consumption	9.7 kg/sec
Low heat value	42000 kj /kg

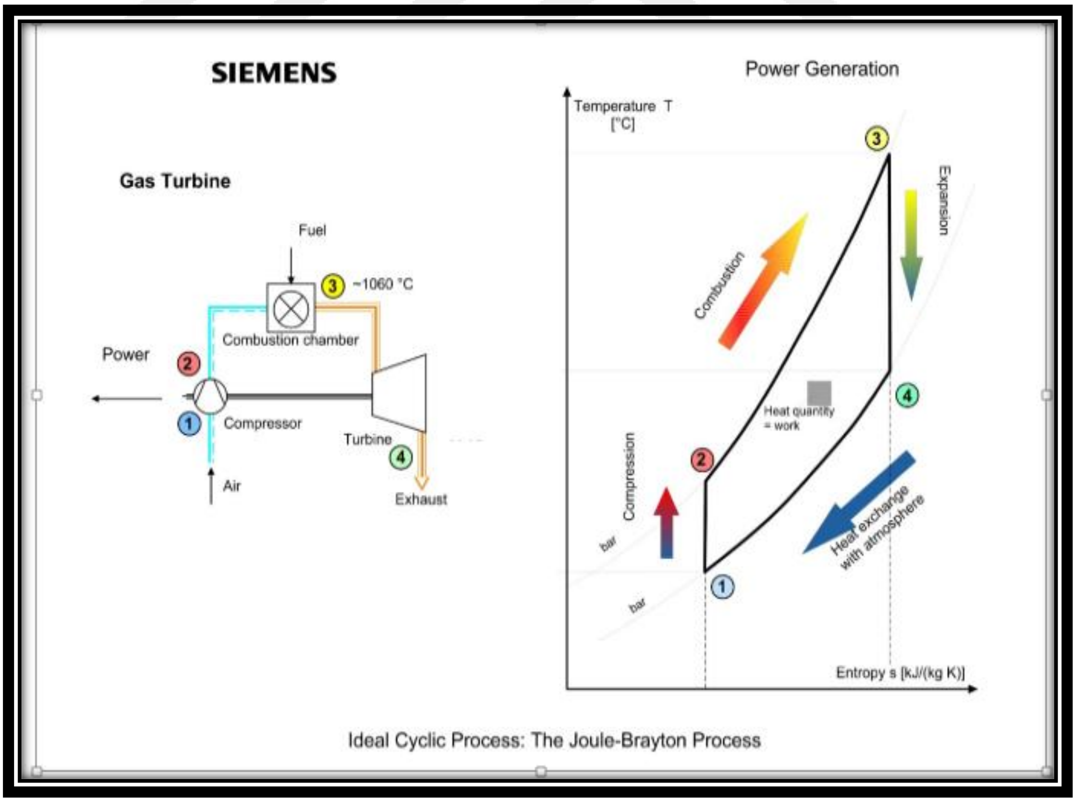


Figure 3.7: Gas turbine cycle

3.5.2 Waste Heat

Waste heat is heat which is produced in an operation of fuel combustion or chemical response, and then deliver into the atmosphere even however it can still be reused for some useful and economic purpose. The essential quality of heat is not its amount but rather its "value." The plan of how to improve this heat count on in portion on the temperature of the waste exhaust heat gases and the economics involved. A huge quantity of hot gases is produced from boilers, gas turbines, ovens and furnaces. If several of this exhaust waste heat can be recovered, a considerable amount of primary fuel can be saved. Table 3.16 shows the value of electric load generation, mass flow of waste heat gases and outlet temperature of the waste gases. The amount of waste heat from the V94.2 gas turbine is 480 kg/sec and 490°C on 134.7 MW[34]. The quantity of heat can be calculated using Eq. (3.28).

$$Q_{Exu} = \dot{m} \times cp_g \times (T - T_1) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3.28)$$

$$Cp_g = [1.8083 - 2.3127\times10^{-3}\, T + 4.045\times10^{-6}\, T^2 - 1.7363\times10^{-9}\, T^{-9}] \text{ [43]} \ldots\ldots\ldots \quad (3.29)$$

Where:

$\dot{m}$ = Mass of gases exit from exhaust gas turbine (kg/sec)

Cp_g = Specific heat of exhaust gases (kj / kg.k),

T_1 = Air temperature enter the compressor (k)

T = Gases temperature exiting from exhaust turbine (k)

Q_{exh} = Heat add (kW).

Table 3.16: Electric load generation withe waste mass flow rate and outlet temperature

No	Load (mw)	$\dot{m}$ (kg/sec)	Tout (Ċ)	No	Load mw	$\dot{m}$ (kg/sec)	Tout (Ċ)
1	20	332.64	336	7	80	359.48	490
2	30	333.1	343	8	90	374.37	490
3	40	333.69	377	9	100	402.02	490
4	50	334.15	411.6	10	110	421.6	490
5	60	335.35	460	11	120	437.37	490
6	70	335.35	490	12	134.7	480.26	490

3.6 Comparison Cost between Vapor Compression Chiller System and Vapor Absorption Chillier System.

Waste gas processes are generally characterized by high investment and low operating costs. Thus, the main economic problem is one of comparing an initial known investment with the estimated future operations savings. The cost of each energy delivery process includes multi-costs; however, the operating expenses are the main subject of the thesis. The Ministry of Electricity in Iraq is the main electricity energy utility provider which has to generate electricity constantly to meet demand at peak periods operate at low efficiency during off peak periods. Despite the waste gases, the energy cost is zero and the accessories pump still consumes minimum electricity. In this study is calculating the operation costs, we are compared in Turkey. Table 3.17 shows the data of the vapor compression chiller and exhaust gases absorption chiller, the reference of data is taken from both catalogs.

Table 3.17: Data of the vapor compression chiller and exhaust driven absorption chiller.

Description	Vapor Compression Chiller [44]	Exhaust Absorption Chiller [45].
Chiller Capacity 62 TR	86.46 (kw)	2.2
Fans Chiller	4.1 (kw)	Nil
chilled water pump	3.7 kw	3.7 kw
Cooling water pump	2.79 (kw)	2.79 kw
Cooling tower pump	Nil	5.126 kw
Total Power	97.05 (kw)	13.82 (kw)

Vapor compression chiller cost operation:

1 KWh power cost (industry) = 75 I.D (Iraq)[46]

1 KWh power cost (industry) = 0.252 T.L (Turkey)[47]

= price × Total power ×7 hr. /day×22 day/month×6 month..... (3.30)

CHAPTER FOUR

RESULATS AND DISCUTION

4.1 Introduction

This chapter shows the results and discusses of the cooling load analysis of the building and the simulation analysis of the waste exhaust absorption of the cooling system. An analysis of the exhaust waste heat of the gas turbine will be presented in this chapter and the simulation results will be further explained for the cooling load of the building, the waste exhaust absorption cooling system and the waste exhaust gas turbine. Comparison of the economic costs between the compression cooling chiller and absorbent chiller will be performed**.**

4.2 Thermal Load Calculation for the Building

4.2.1 Heat Transfer Analysis

In any building or space, heat is transmitted through external walls, roof tops, the floor of the ground floor, windows and doors. Heat transfer in space takes place by conduction of heat, convection and radiation. The cooling load of the building depends upon the thermal characteristics of the materials, climate of the region in which the building is located and the type of building. For cooling load calculations, there are many methods to estimate the heat and cooling load. A number of these methods need long and complex data input to estimate cooling loads. Most engineers use simple methods. They select a more compact and normal method for calculating the cooling and heat load of a building. A more basic version for calculating a cooling load using the transfer function method is to use the one-step procedure, which was first presented in the **ASHRAE** Handbook of Fundamentals. This method is called the Cooling Load Temperature Differences (CLTD) method, in this method, a hand calculation is used to find the cooling load of a space. Using an Excel spreadsheet, calculations are performed for the building using all the equations, calculation information and procedures for this part referenced in Chapter 3. The building for the project has a total of 29 spaces of ground and first floors. Each one is a separate calculation. Every equation that was used was required for the heat transfer through the building. For the inside space, the heat loads were used to get the cooling heat load of the space of the office building. Therefore, all the formula equations were put into an Excel spreadsheet to acquire the results of the heat cooling load of the space.

4.2.2 Design Condition

Cooling can keep buildings comfortable during the summer and winter depending on the required indoor and outdoor conditions and the design conditions. These conditions are called the indoor design condition and outdoor design condition. The desired comfort requires an indoor temperature ring from 22.5°C to 26°C and a relative humidity of 50%. The estimated cooling load of the building is best at 24°C dry bulb temperature (DBT) and 50% relative humidity (RH) for inside design conditions. The temperature and relative humidity for outdoor conditions are taken from the Salahaldeen city meteorological office. The outdoor design conditions of Salahaldeen city are 49.5°C DBT and 15.5% RH for July [48].

4.2.3 Overall heat Transfer Coefficient of the Building

Commonly the walls of buildings, roofs and glass may consist of multiple layers and non-similar physical properties. The walls may consist of common cement plaster, common bricks and gypsum plaster and the roofs may consist of shtyger, sand, and gypsum plaster. The heat load transfer from one end to the other end of these layers of the wall is complex as it covers heat transfer by convection, conduction and radiation, as shown in Figure 3.2 in Chapter Three. Different types of walls can have many thermo-physical properties. Wall material properties have been tested and offered in ASHRAE standard and other reference books on refrigeration and air conditioning.

4.2.3.1 Overall Heat Transfer Coefficient Calculation for the Walls (U_o)

The external walls are the most important part of building structures. Most heat loss occurs through external walls. Knowing external wall materials is an important element in knowing a building's energy efficiency and keeping it in a suitable condition. The external walls of an office building have three different layers in their construction: cement blaster, the common bricks and gypsum plaster. The outside coefficient, inside coefficient, thermal conductivity (K) and thermal conductance (h) are taken from Tables (3.1) and (3.2). The details of the wall materials are shown in Table (4.1).

Table 4.1: Detail of walls material

Component	Thickness(mm)	k (W/m.k)	h (W/m^2.k)
Outside coefficient	–	–	23.3
Cement blaster	15	0.72	–
Common brick	200	0.77	–
Gypsum plaster	15	0.81	–
Inside coefficient	–	–	8.5

4.2.3.2 Overall Heat Transfer Coefficient of the Celling

The roof is another important component which in large part, can be attributable for the heat losses of a building. The roofs of the building have four different construction layers, including high density cement, sand, gypsum plaster and shtyger cement with an outside coefficient and an inside coefficient. The thermal heat conductivity and thermal heat conductance are taken from Tables 3.1 and 3.2. Details of the roof materials is shown in Table 4.2.

Table 4.2: Detail of roofs material.

Component	Thickness(mm)	k (W/m.k)	h (W/m^2.k)
Outside coefficient	—	—	9.4
Cement styger	40	1.73	—
sand	100	1.72	—
Cement high density	200	1.73	—
Gypsum plaster	10	0.81	—
Inside coefficient	—	—	6.3

4.2.3.3 Overall Heat Transfer Coefficient of the Partitions

The inside walls of the building represent the inside of the structure of the building and it has no solar radiation through it. The partition walls consist of common brick and gypsum plaster on both sides of the walls. The thermal heat conductance is taken from Tables 3.2 and 3.3. The details of partition material are shown in Table 4.3.

Table 4.3: Detail of partition material.

Component	Thickness(mm)	k (W/m.k)	h (W/m^2.k)
Inside coefficient	—	—	8.5
Gypsum plaster	13	0.81	—
common brick	200	0.77	—
Gypsum plaster	13	0.81	—
Inside coefficient	—	—	8.1

4.2.3.4 Overall Heat Transfer Coefficient of the Glass

Effective windows with glazing technologies can make a space more comfortable. In buildings with high U-value windows, during the heating season, the outside temperature drives down the interior glass surface temperatures to below the space air temperature, which causes discomfort due to radiant heat loss and drafts. Windows with low U-value are a good solution. The windows of the building each have one sheet of 3-millimeter thick glass. Thermal heat conductivity and thermal heat conductance are taken from Tables 3.2 and 3.3. The overall heat transfer coefficient is calculated using Eq. (3.2). Details of the glass material is shown in Table 4.4.

Table 4.4: Detail of glass material**.**

Component	Thickness(mm)	k (W/m.k)	h (W/m^2.k)
Outside coefficient	–	–	23.3
Glass thickness	3	0.81	–
Inside coefficient	–	–	8.5

4.2.3.5 Overall Heat Transfer Co-efficient (U)

The U value is an amount of heat loss in a building structure element such as a wall or a roof. It can also be an indication of the overall heat transfer coefficient and it measures how fine parts of a building structure transfer heat. This means that the higher the U value, the poorer the thermal performance of the building. In Figure 4.1 below, it can be seen that most heat will come from the glass since the (U) value of glass is greatest in comparison to the (U) values of other wall materials. The minimum value for (U) of this building construction material is at the partition. The external wall of this office building has a slightly lower U value. The heat loads coming from the wall are lower compared to the heat coming from the glass area. In this office building, most of the heat loads will come through the glass area facing the sun.

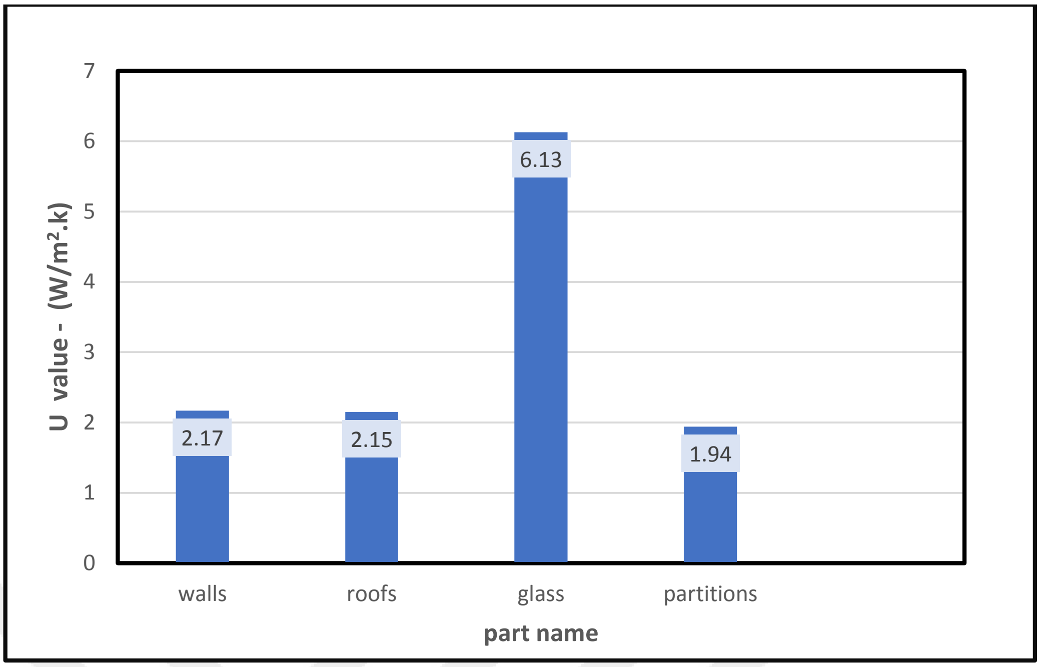


Figure 4.1: U value of the structures office building**.**

4.2.4 Calculation of Correction Cooling Load Temperature

The accuracy of the CLTD value is such that it is not an exact value when the building is not at 40°N latitude. We will use Eq. (3.1) to calculate the $CLTD_c$ of the walls, roofs and windows. It is taken at 3 p.m. because the maximum cooling load occurs at this time of the day and the LMT values are taken in July at 35°N latitude.

Table 4.5: Corrected values for CLTD of a building.

directions	**CLTD(°C)**	**K**	**LM**	**T_o (Ċ)**	**T_i (Ċ)**	**$CLTD_c$ (Ċ)**
N	5	1	0.3	41	24	18.4
E	13	1	0.0	41	24	26.1
S	8	1	- 0.8	41	23	20.3
W	8	1	0.0	41	24	21.1
HORI.	14	1	0.5	41	24	27.6
Glass	8	**—**	–	41	24	21.1

4.2.5. Cooling Load Calculation of the Building

4.2.5.1 Cooling Load Calculation of Walls of the Building.

Figures 4.2 and 4.3 show the heat gain due to conduction and convection through the walls of the building. This situation shows the heat gain through external walls at different orientations, such as from the west, north, east and south. The different values of the heat gain through the walls depend on the area and orientation of the walls. The maximum value of the heat gain through the walls occurs in the lecture room and meeting room as they have walls that are large in area. Moreover, they have the second highest temperature in those directions. Rooms 3 and 4 for both floors have the second highest wall heat gain as they have three exposed external walls. The other walls of the rooms for both floors have different wall heat gain values because they have different orientations and areas. The results of the values agree with the results of the values from research **[1][11]**.

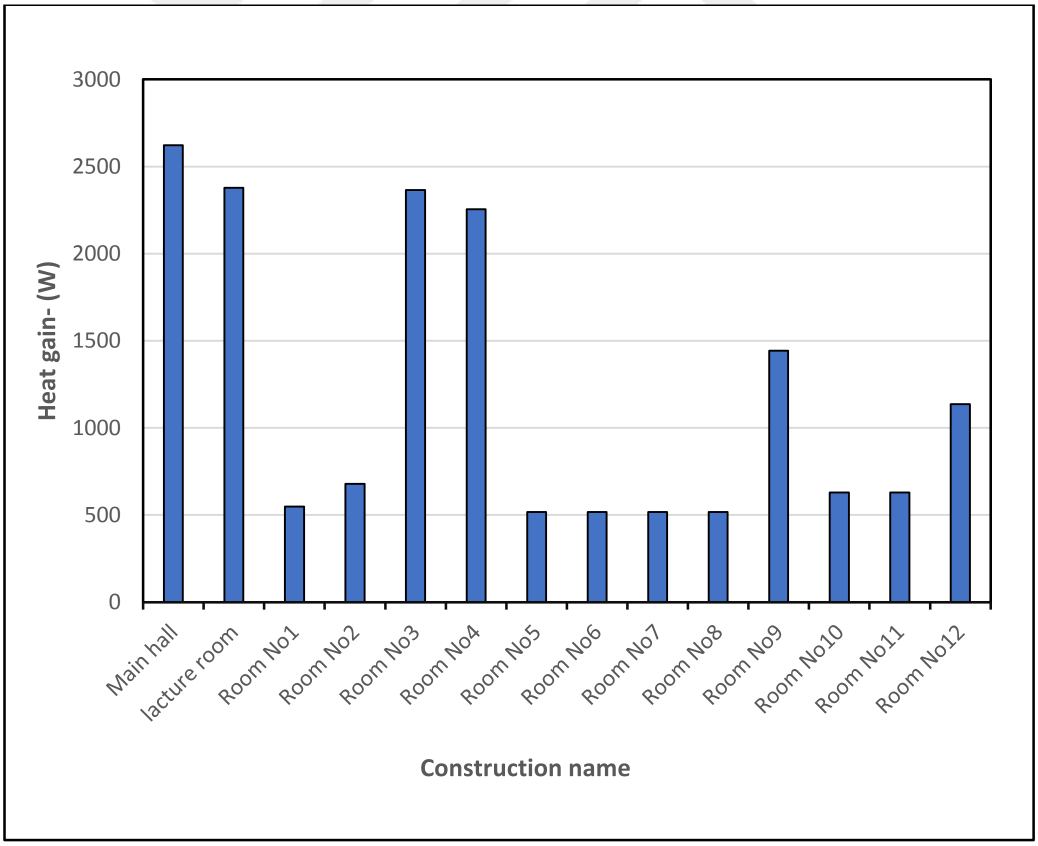


Figure 4.2: Wall heat gain of ground floor.

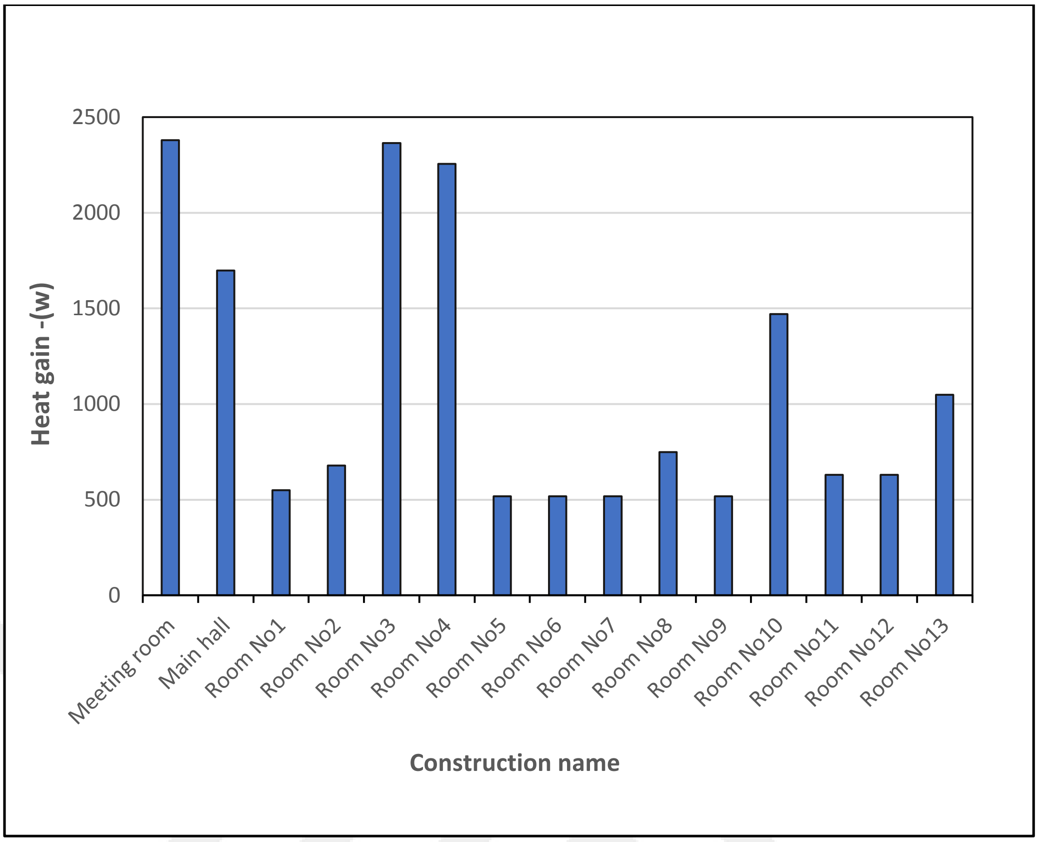


Figure 4.3: Walls heat gain of first floor.

4.2.5.2 Roofs Heat Gain of the Building

Figure 4.4 shows the heat variation of the heat gained through the roofs of the office building. Note in the figure below that it has varying heat gain quantities, the highest value of which is found in the main hall as it has the largest roof area. The minimum heat loads is are found in rooms 5, 6, 7, 8, 9 and 11. The heat gained through the roofs does not depend on the direction of the building; however, it does depend on location the building, the area of the roof and the materials of the roof. The heat load calculation of the roofs are choosing with spending roof. The roofs of the ground floor have no heat gain through their roofs because they have identical temperatures between the ground floor and first floor. The results of heat value through roofs agree with the results of research [1][11].

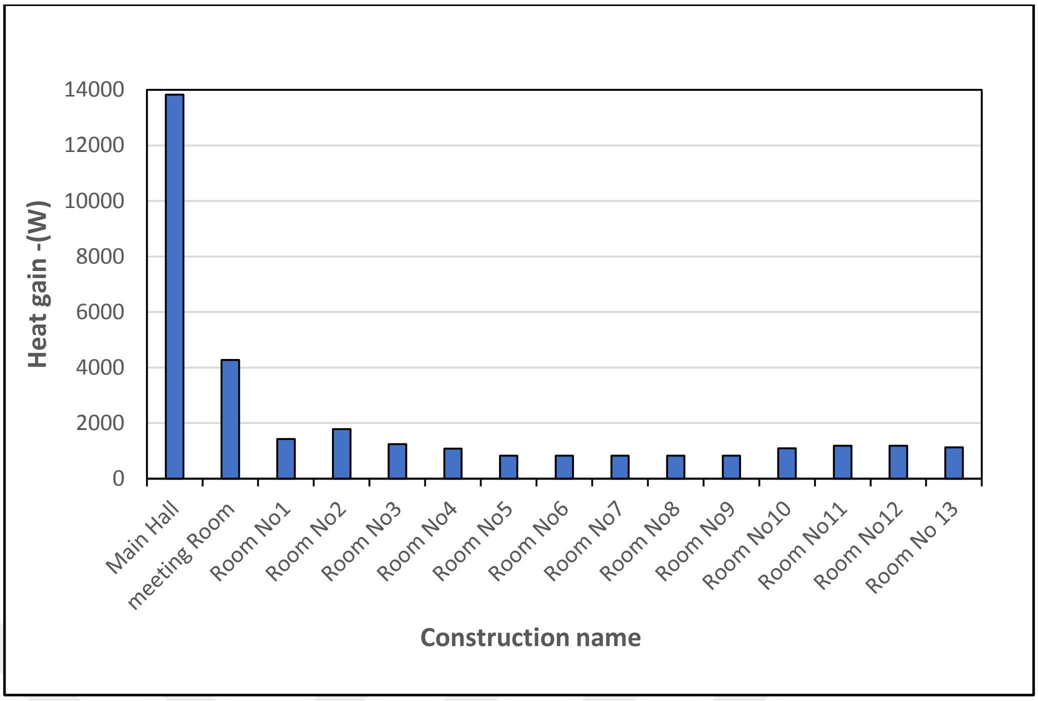


Figure 4.4: Roofs heat gain of first floor.

4.2.5.3 Windows Cooling Load Calculation of the Building

Graphs 4.5 and 4.6 show the heat gain of conduction and convection through the window glass of the building. This situation shows that heat gain through external windows from different directions such as the west, north, east and south. The heat gain value through the windows is an estimation. Note that the heat gain quantity differs among the directions. The maximum value of solar heat gain in the rooms occurs from the east and west windows as the maximum value **SHG** occurs in those directions of the building. The area of the window is again an important part of heat gain. The results of heat value through the windows agree with the results of research [1][11].

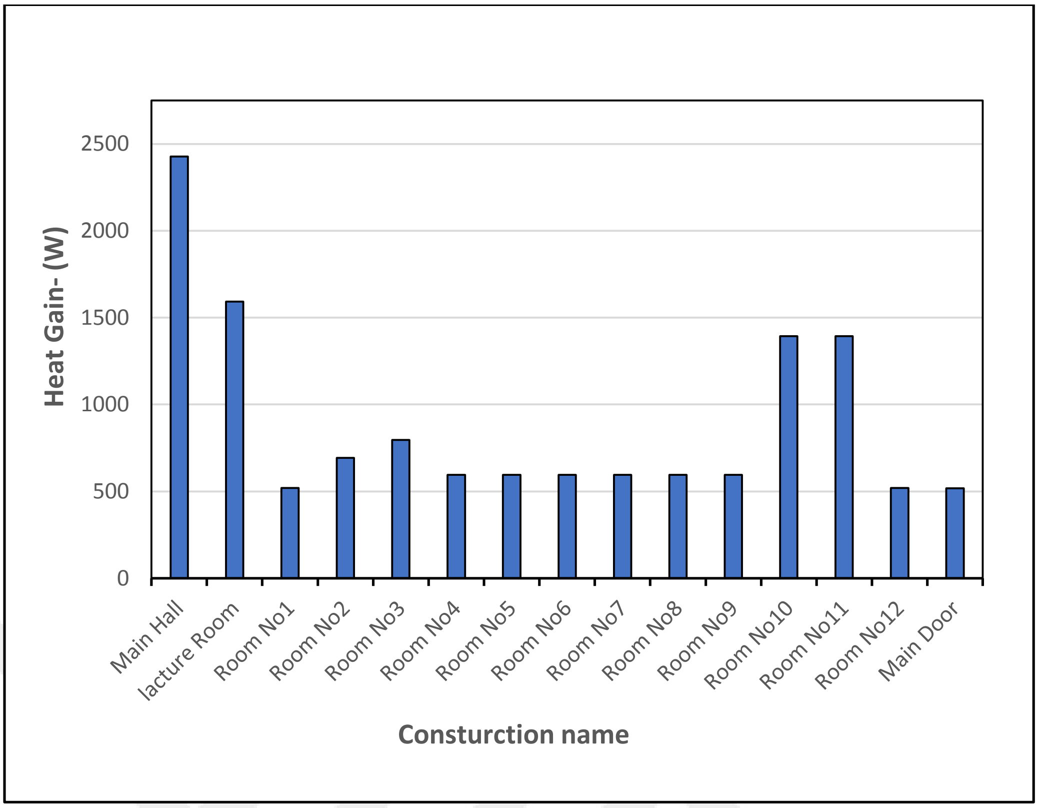


Figure 4.5: Windows heat gain for ground floor.

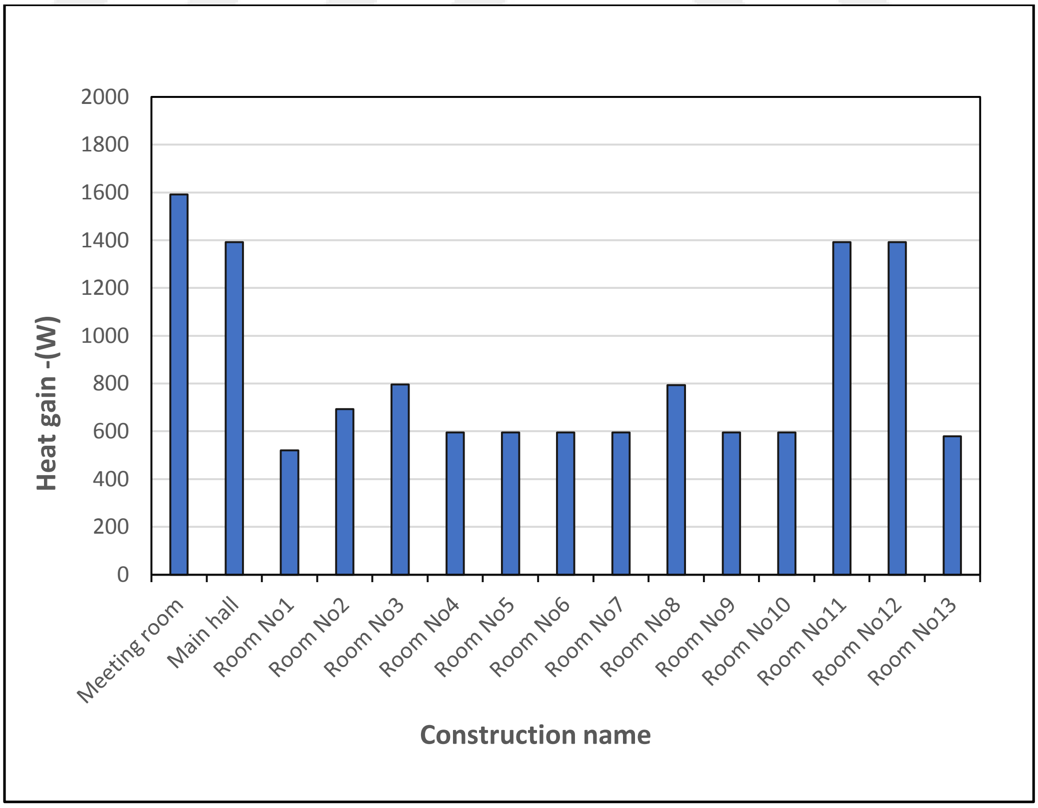


Figure 4.6: Windows heat gain for first floor.

4.2.5.3 Occupancy Heat Load Calculation of the Building

Figures 4.7 and 4.8 show the heat gain generation by people inside those spaces. Note that in the figures below, various values of the occupancy heat load inside the spaces, the lecture room and the main halls have maximum values of the heat gain and heat load. The heat gain value differs from room to room because it depends on the number of people and activity inside the space. The results of heat generation due to occupancy agree with the results of research [1][11].

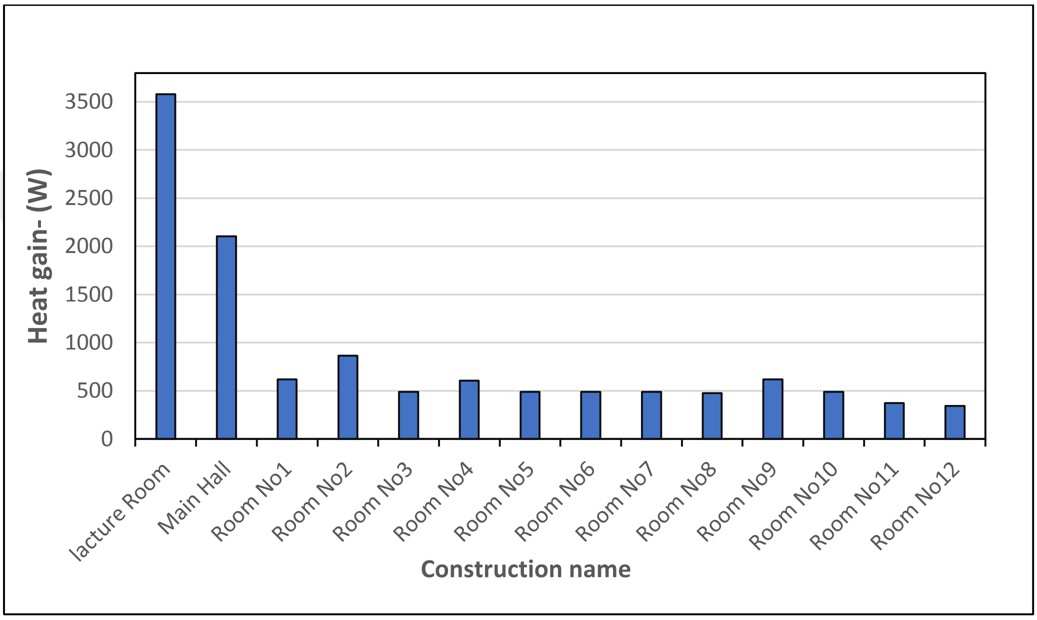


Figure 4.7: Occupancy heat gain for ground floor.

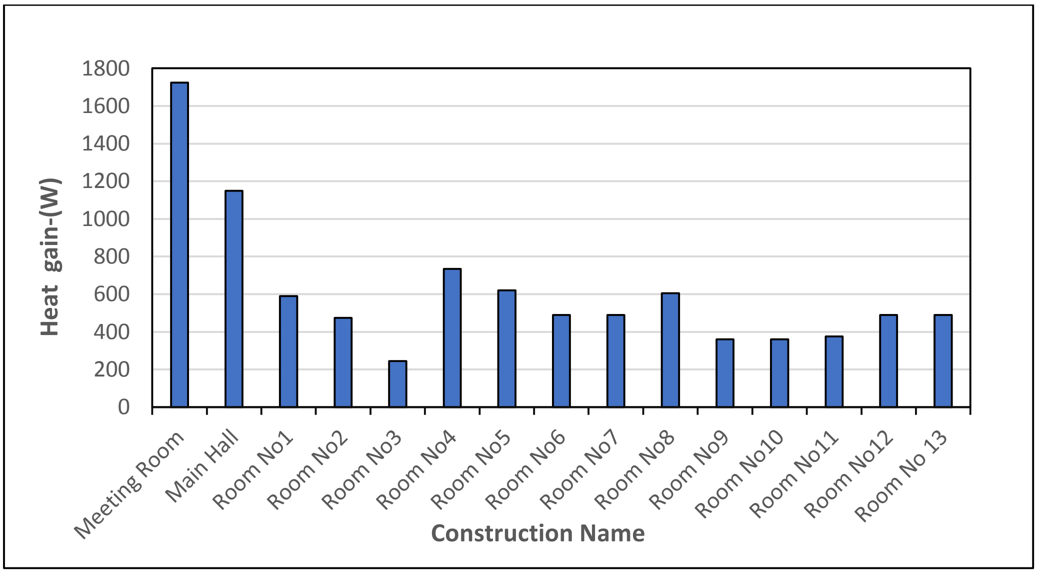


Figure 4.8: Occupancy heat gain for first floor.

4.2.5.4 Lighting Heat Load Calculation

Figures 4.9 and 4.10 show the heat gain from the lighting load for each space of the building. The lighting load depends on activity inside the space and it depends on the area of the space. The office building has several activities, such as office work, standing and study. Note that the maximum heat gains occur in the main halls, the lecture room and the meeting room for the floors. The maximum value for heat gain is shown in the main halls because they have a large value for their respective areas. The results for heat value from lighting are decide from the results of research [1][11].

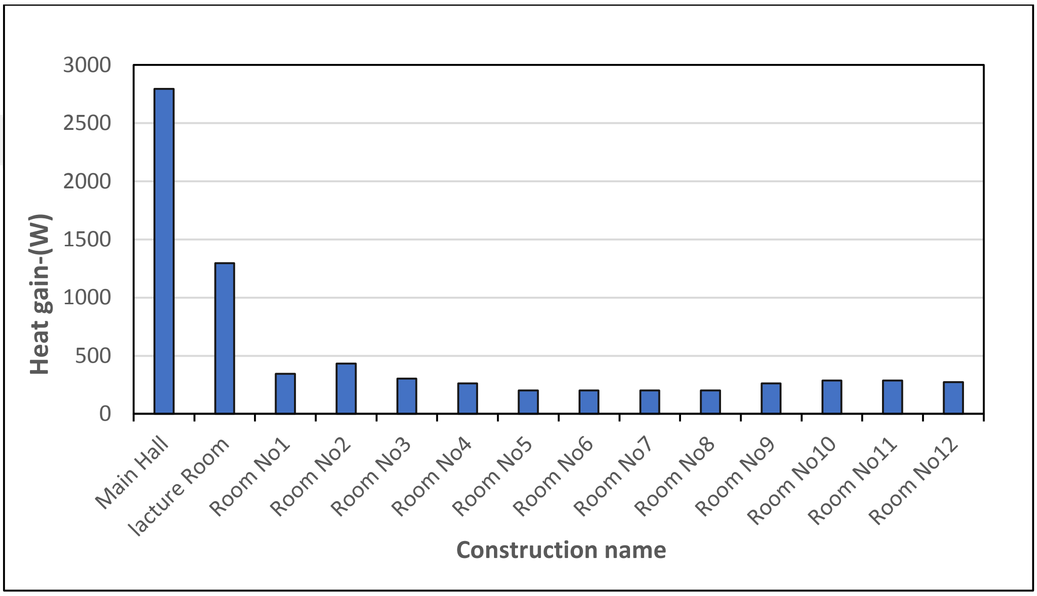


Figure 4.9: Lighting heat gain for ground floor.

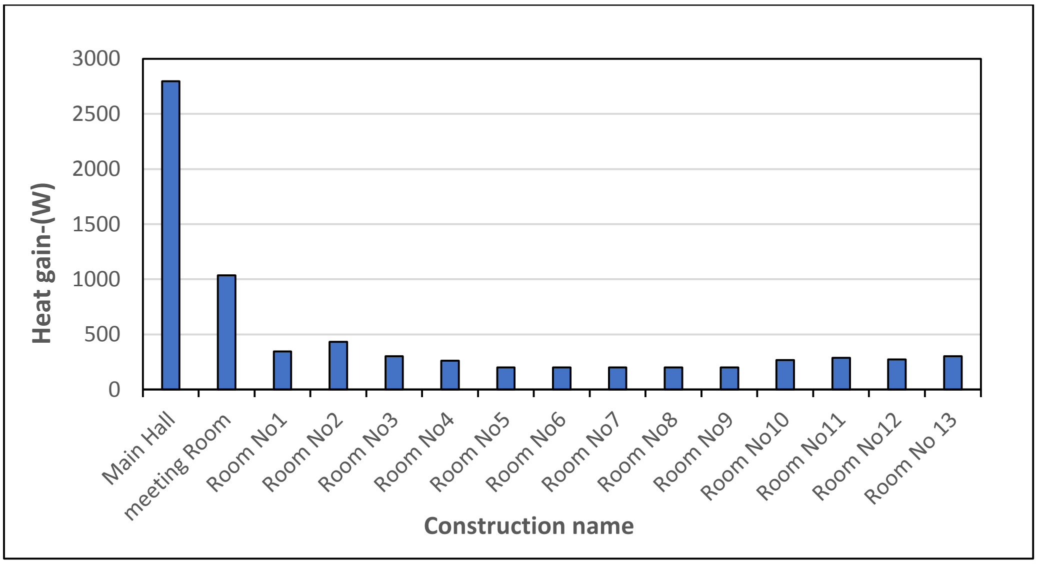


Figure 4.10: Lighting heat gain for first floor.

4.2.5.5 Equipment Heat Load Calculation

Figures 4.11 and 4.12 show heat loads in rooms from equipment. The maximum heat load value is found in the meeting room on the first floor because it has contain a large amount of equipment such as computers. The other rooms have multiple heat load values. The equipment in the office building, such as computers and printers, generate sensible heat only in the space. The figures below show the various amounts of heat generated inside the rooms commensurate with the amount and power capacity of the equipment inside the space. The results of heat generation from equipment agree with results of research [1][11].

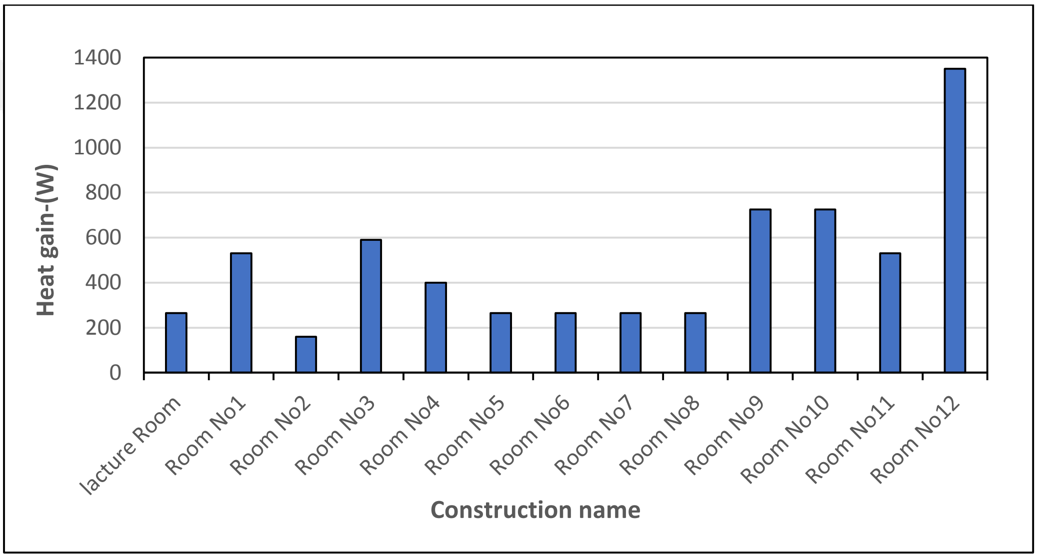


Figure 4.11: Equipment Heat gain of ground floor.

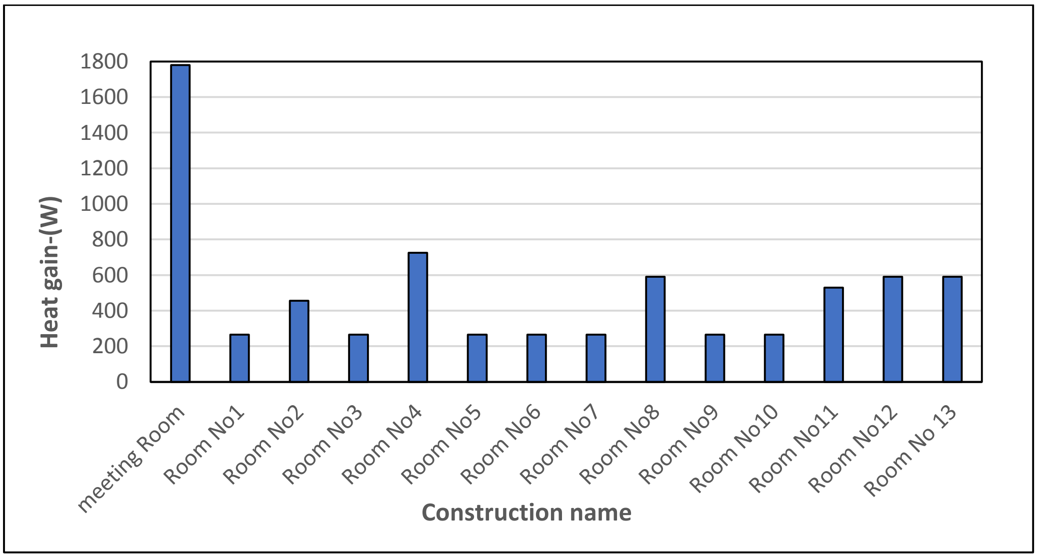


Figure 4.12: Equipment heat gain of first floor.

4.2.5.6 Infiltration Load of the Building

Figures 4.13 and 4.14 show the heat load gain from infiltration The heat gain value differs for each space because the load depends on the number of walls and windows exposed outside and the volume of the space. We note in the figures below the maximum values of heat loads in the main halls, meeting room and lecture room of the floors as they have various walls exposed outside and the largest volume. The other spaces have different heat load gain values because they have a multiple number of walls exposed outside with differing volumes of space. The results of the infiltration heat load value agree with the results of search [1][11].

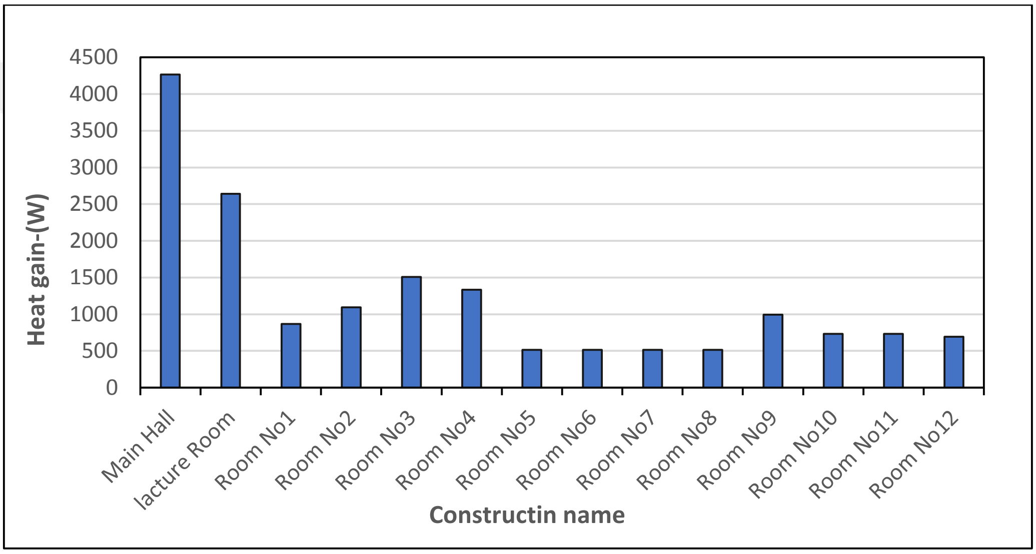


Figure 4.13: Infiltration heat gain of ground floor.

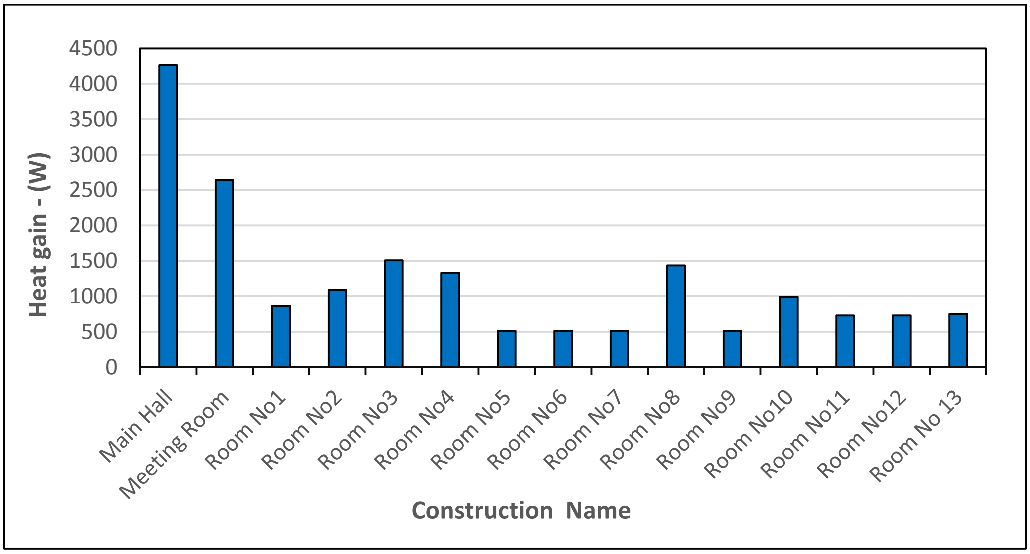


Figure 4.14: Infiltration heat gain for first floor.

4.2.5.5 Ventilation Loads of the Building

Figures 4.15 and 4.16 show the values of the heat ventilation load of the rooms. Note below the various ventilation heat load values inside the spaces as they have different numbers of people inside the spaces. The ventilation heat load depends on a number of people inside the space and the type of people. For the figures below, the maximum values of the ventilation heat load are found in the lecture room and meeting room due to the fact that these rooms can hold many people. The other spaces on each floor have multiple ventilation heat loads as they have varying numbers of people and activities. The results of the ventilation heat load values agree with results of research [1][11].

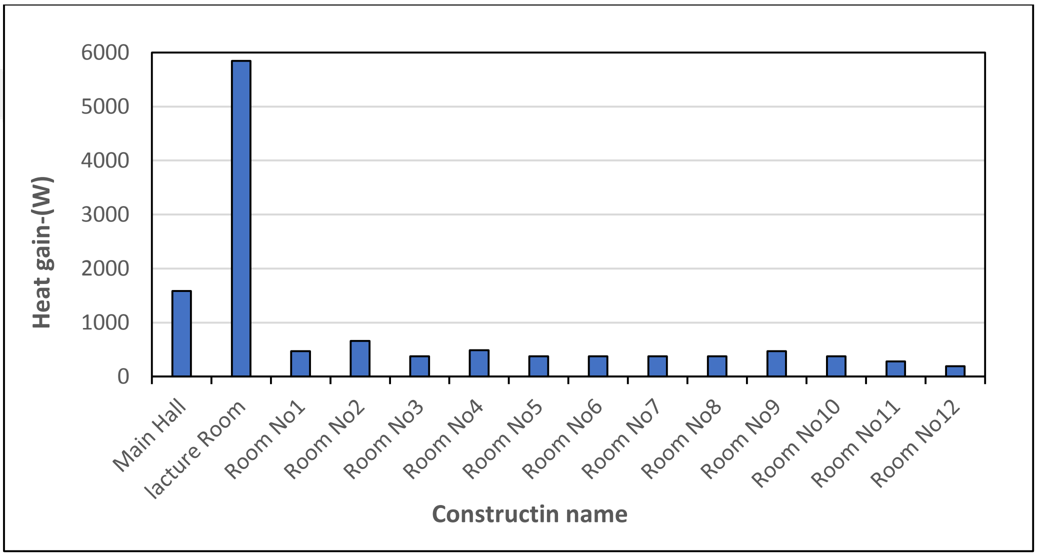


Figure 4.15: Ventilation heat gain of ground floor.

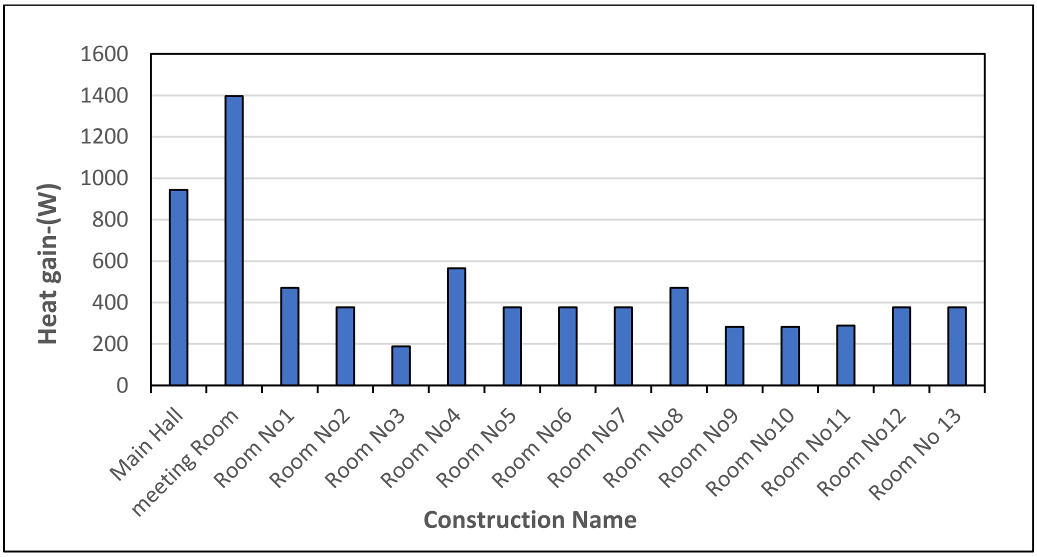


Figure 4.16: Ventilation heat gain for first floor.

4.2.5.6 Total Heat Load of the Building

Figures 4.17 and 4.18 show the total heat loads of spaces for the office building. Note that the figures below the heat load values differ from one space to another space because the value of the heat load depends on various factors such as orientation, occupancy, equipment, infiltration and window area. The maximum heat load is found at the main halls, lecture room and meeting room because they have the highest sources of heat gain in the building. Figures 4.1 to 4.15 show detailed graphs of the heat gain during a particular time. From eight types of heat gain into each space, they can be categorized into two main types of heat gain: internal heat gain and external heat gain. Internal heat gain comes from people, appliances and lighting. Exterior heat gain is due to heat radiation, heat conduction and heat gain due to ventilation.

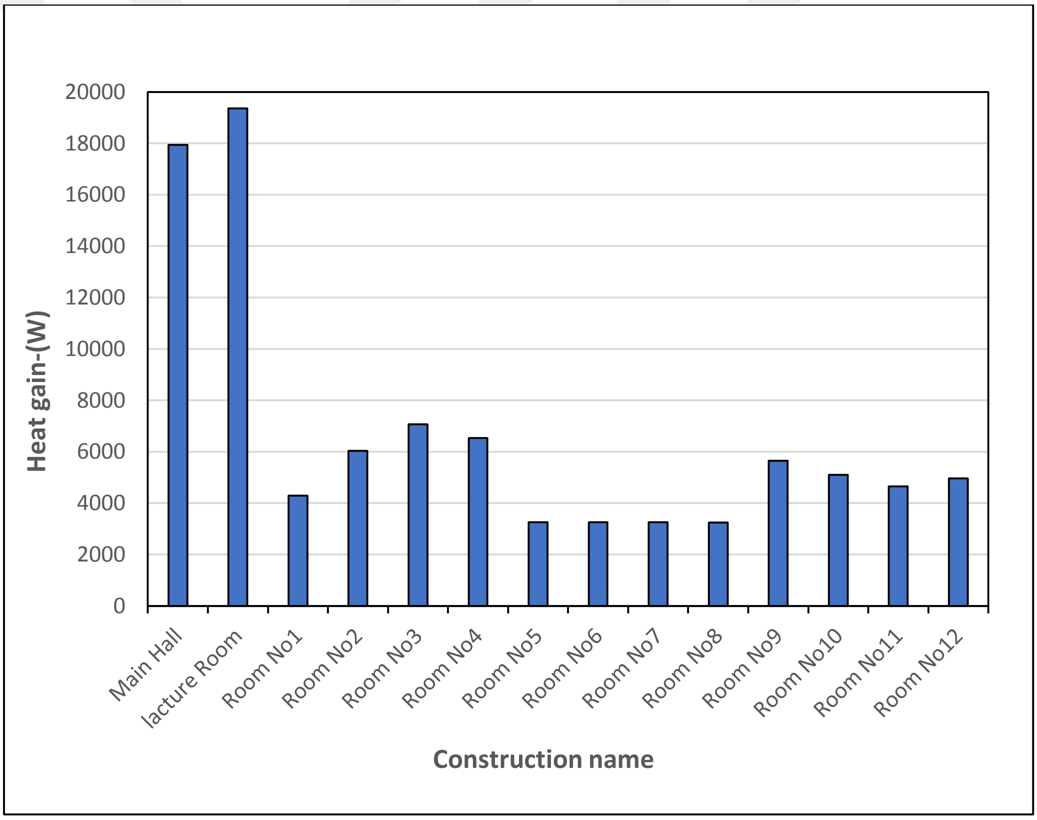


Figure 4.17: Total heat load of rooms for ground floor.

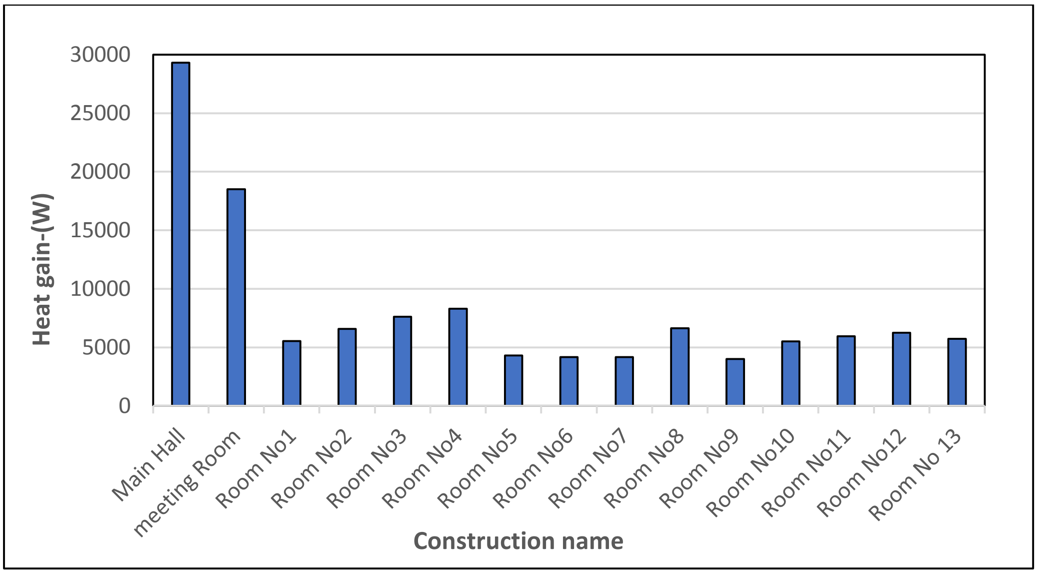


Figure 4.18: Cooling load gain of rooms for first floor.

Figure 4.19 shows the total cooling load of the ground floor, the first floor and the total load of the building. The total cooling load of the building is represented by the summation load of each part or room of the building and a 10-percent safety factor is added to the cooling load. This factor comes from the ASHRAE group. The calculation load is increased by a factor of 10 % to allow for possible discrepancies between the design criteria and actual operation.

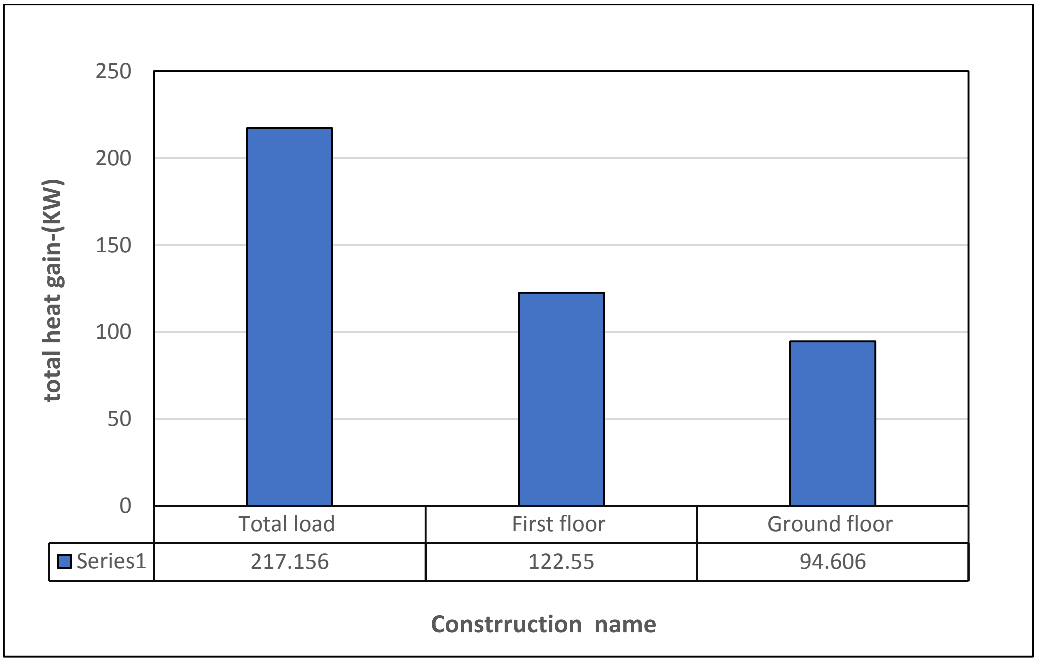


Figure 4.19: Total cooling load of the building.

4.2.5.7 Sensible Heat Factor of the Building

Figures 4.20 and 4.21 show the values of the **S.H.F**. They are used to find the conation of the space. Note that there are different values for each space because they depend on which space has a high value of latent heat load. The maximum values occur in rooms with minimum values of latent heat load, and the minimum value of **S.H.F** in spaces with high values of latent heat, such as a lecture room and room No. 5 of the first floor as they have high values of latent heat load for their sensible heat load. These results agree with the sources [1].

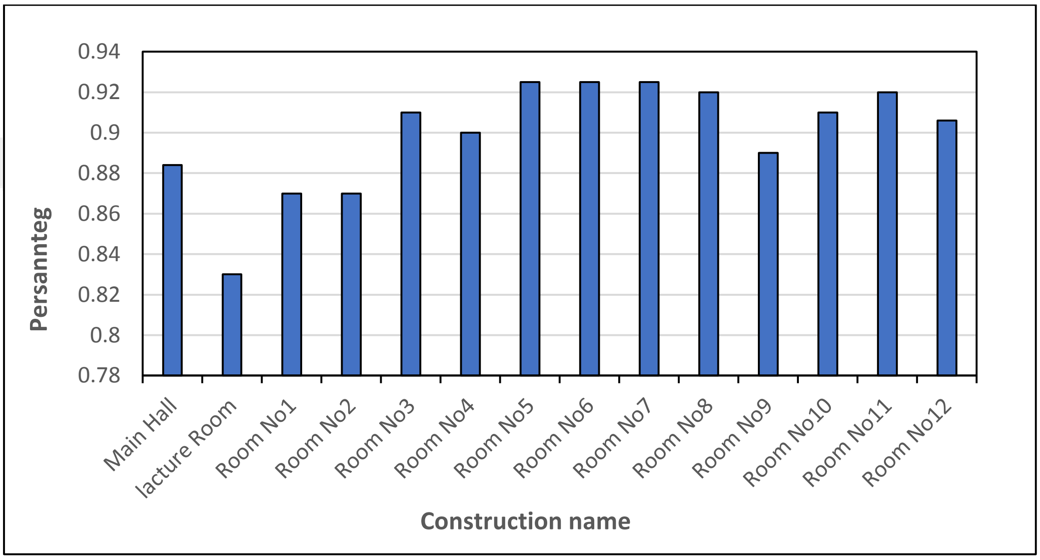


Figure 4.20: S.H.F of ground floor.

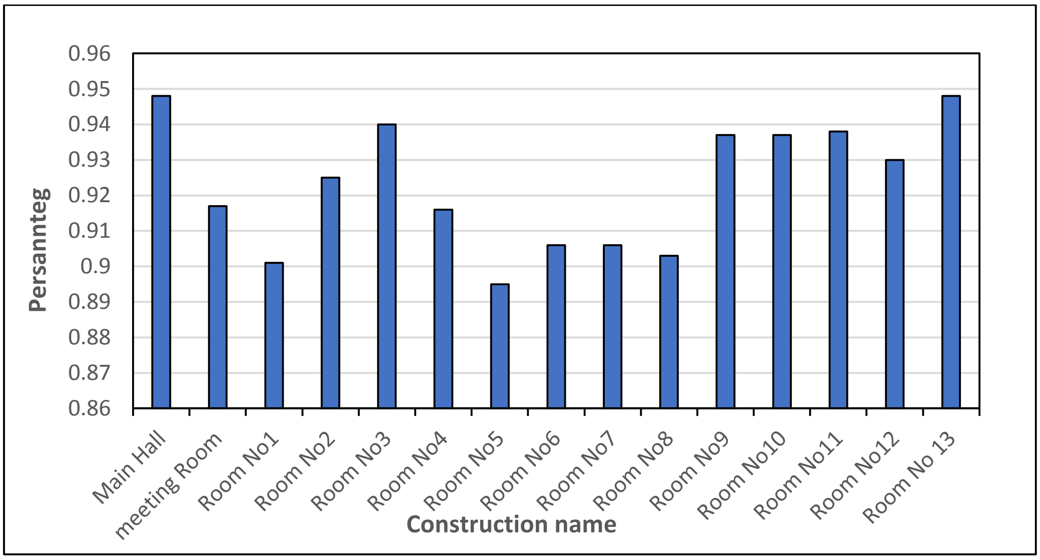


Figure 4.21: S.H.F of first floor rooms.

4.2.5.5 Supply Air of the Building

Figures 4.22 and 4.23 show the quantity of air supply to the spaces. The variation in the quantity of the air supply of different rooms is shown below. Note that the maximum value of the air supply to the spaces, the halls spaces, the lacquer room and the meeting room have high values of sensible heat load. The air supply to the space is an air treatment and control system. It treats the incoming air to ensure that the air quality is suitable for sufficient amounts and for clean environments such as office buildings or other spaces. The results show that the air supplied to the spaces depends on the sensible heat value. The results of the air supply agree with research results[42].

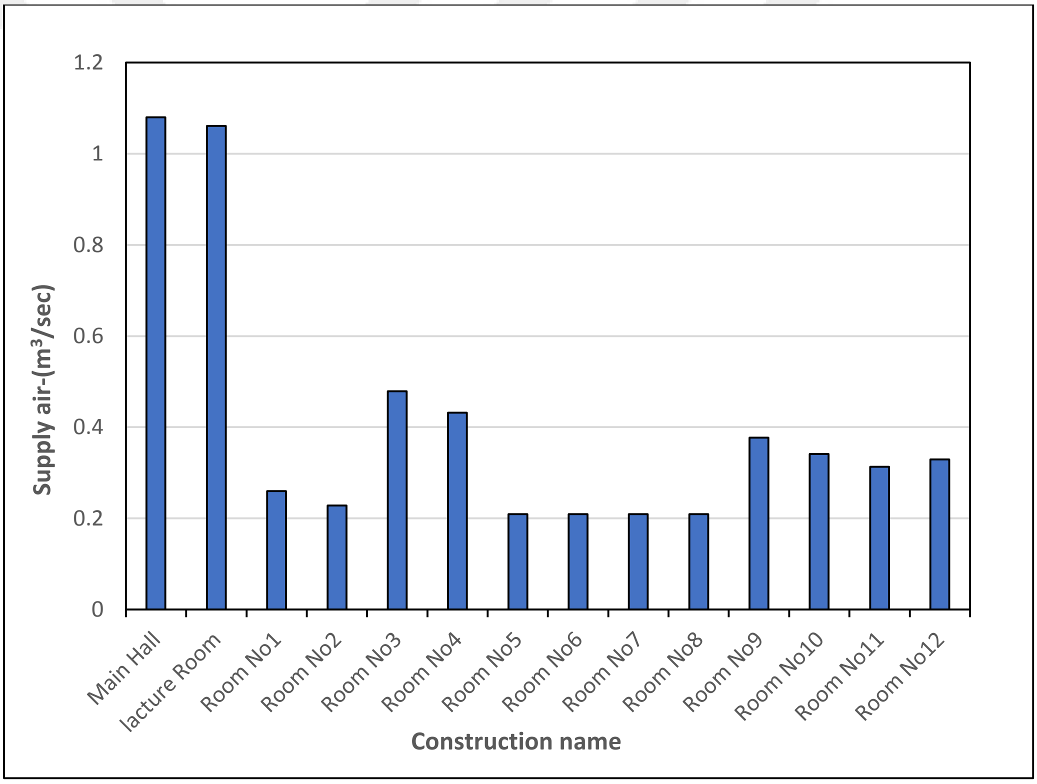


Figure 4.22: Supply air of ground floor room.

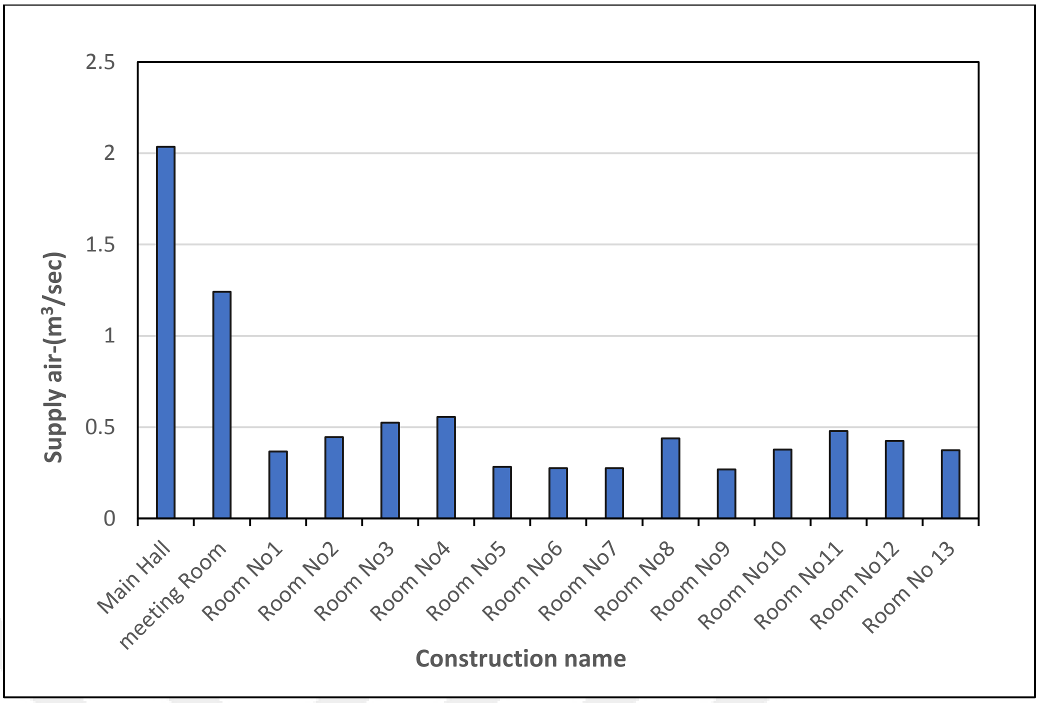
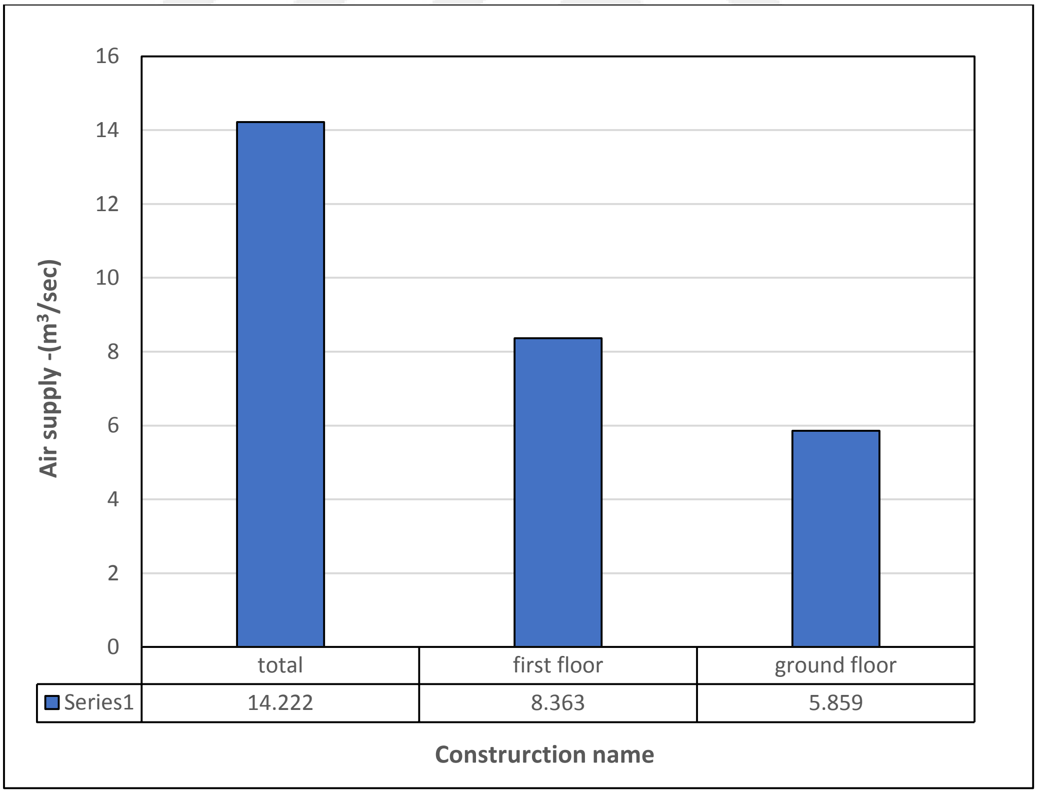


Figure 4.23: Supply air of first floor rooms.



	total	first floor	ground floor
Series1	14.222	8.363	5.859

Figure 4.24: Total supply air of building.

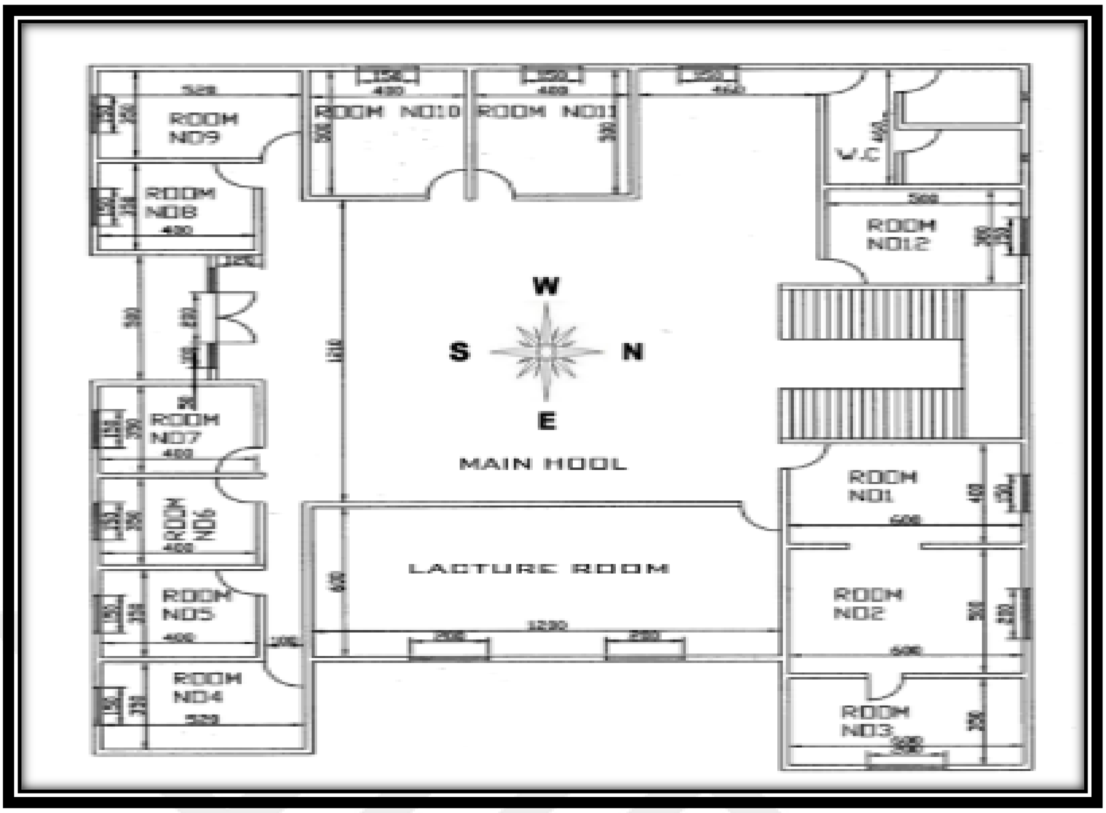


Figure 4.25: Ground floor diagram of the building**.**

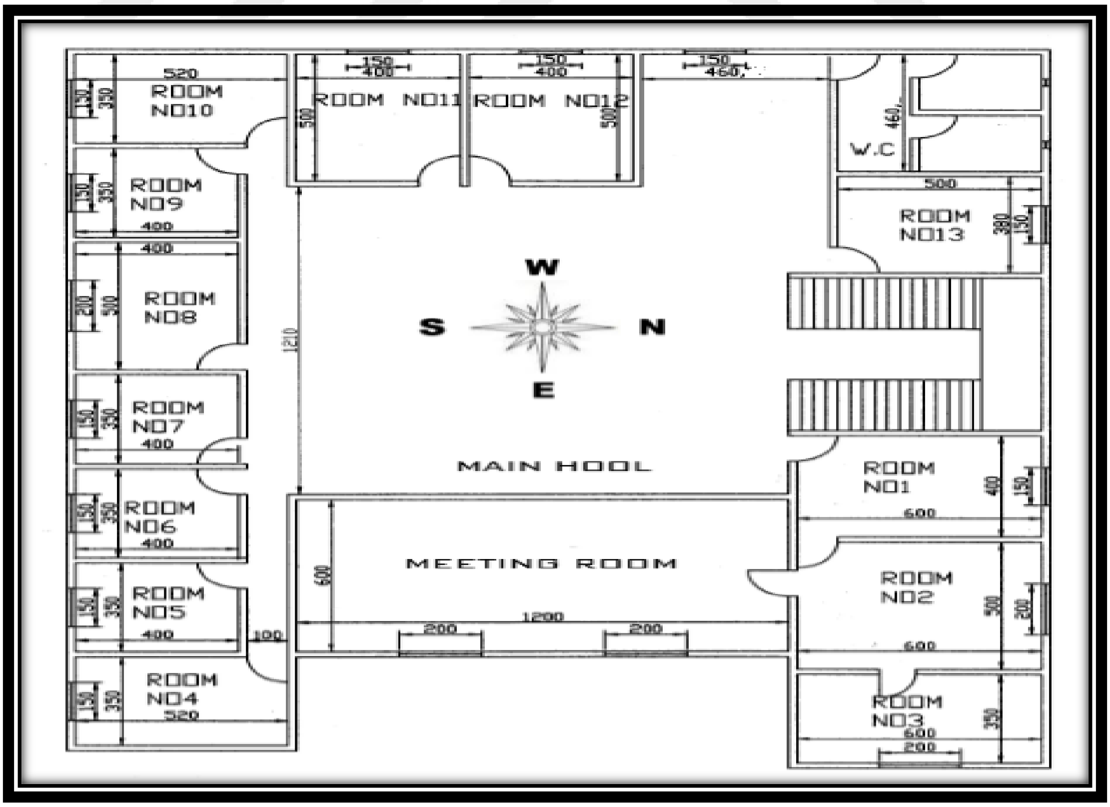


Figure 4.26: First floor diagram of the building.

4.3 Absorption System Cycle

One single effect of the absorption refrigeration system will be its modeling and design the size of which will depend on the cooling heat load of the building. Waste heat gases from the gas turbine exhaust will be a source of energy to operate the system. One waste gas turbine power system will estimate and calculate the COP of the system. Taking this approach, the absorption cycle machine will be powered by waste gases from the exhaust gas from the turbine engine. The absorption machine is set to cool a stream of water at a refrigeration load of 217.156 kW.

4.3.1 Single Effect Lithium Bromide-Water Absorption System

The design model of a single stage lithium bromide-water machine has pressure levels and it will be occupied within the cycle. The generator and condenser are under same pressure value. The evaporator pressure level and the absorber pressure value are under the same pressure level. The value of the higher pressure level data is also resolute from the pressure of the saturated liquid at the outlet of the condenser. The lower pressure level data value is read from the refrigerant temperature at the outlet of the evaporator.

4.3.2 Results and Dissection of the Study Absorption System

The results model for the thermodynamic properties and heat transfer rates of each element are shown in three main cases in various conditions. The various conditions represent the generator temperature and condenser temperature.

4.3.2.1 Case One

The results of the model for the thermodynamic properties and heat transfer rates of each element are shown in Table 4.7. In this simulation, calculations were achieved for a 217.561-kilowatt cooling load of the office building and the parameters were taken as the highest COP of the system $T_c = 35°C$, $T_e = 8°C$, $T_a = 35°C$, $T_g = 85°C$, $\varepsilon = 0.71$ (ε = effectiveness of heat exchanger). We calculated four cases using Excel program to design of the absorption system and we selected the highest COP of cases that represented a suitable design of the system in this case study. Table 4.6 shows the mass flow rates. The chemical composition is provided along with temperature, concentration, pressure and enthalpy values of the working fluids.

Table 4.6: Thermodynamic properties of state points**.**

P.No	T (ċ)	P (kpa)	m (kg/sec)	X (%)	h (kj/kg)
1	85	5.643	0.091640	0	2660.8
2	35	5.643	0.09164	0	146.56
3	35	1.072	0.09164	0	146.56
4	8	1.072	0.09164	0	2516.2
5	35	1.072	0.67383	54	75
6	35	5.643	0.67383	54	75
7	-	5.643	0.67383	54	135.48
8	85	5.643	0.582189	64.5	210
9	49.5	5.643	0.582189	64.5	140
10	49.5	1.072	0.582189	64.5	140

Table 4.7: Heat transfer rates of the system parts.

components	**Duty**
Evaporator (Qe)	217.156 KW
Generator (Q_g)	261.55613 KW
Condenser (Q_c)	274.80741KW
Absorber(Q_a)	230.40727KW
Work Pump (WP)	0.00169 KW
COP	0.79

Figure 4.22 shows the results of the heat loads in different components of the system with varying generator temperature. The generator temperature increases by 50ºC, which causes a reduction in generator heat load. The increase or decrease of the condenser temperature causes multiple heat loads while the evaporator heat load remains constant since evaporator is at a constant temperature.

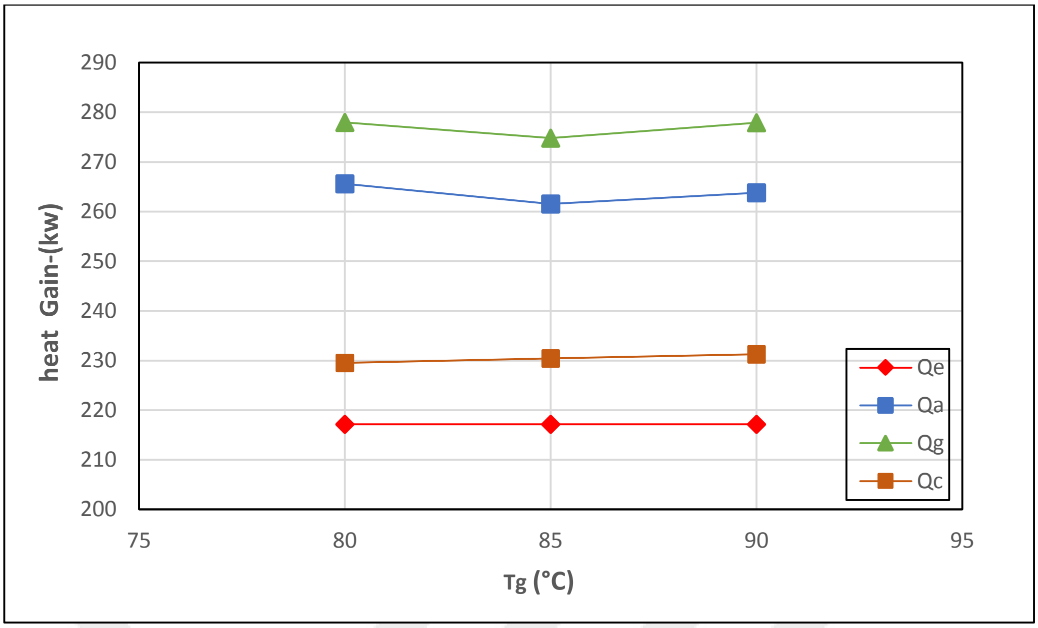


Figure 4.27: Variation heat load with generator temperature (Tg)**.**

Figure 4.28 shows the solution concentration of the refrigerant. The concentration of the strong solution increases with an increase in the generator temperature and causes degrees in the circulation ratio value. The weak solution remains constant at a constant absorber temperature. Note that the maximum value of the concentration occurs at a generator temperature of 90ºC and the circulation ratio has a maximum value of 80 ºC. The circulation ratio depends on the value of the string solution and its value is not constant.

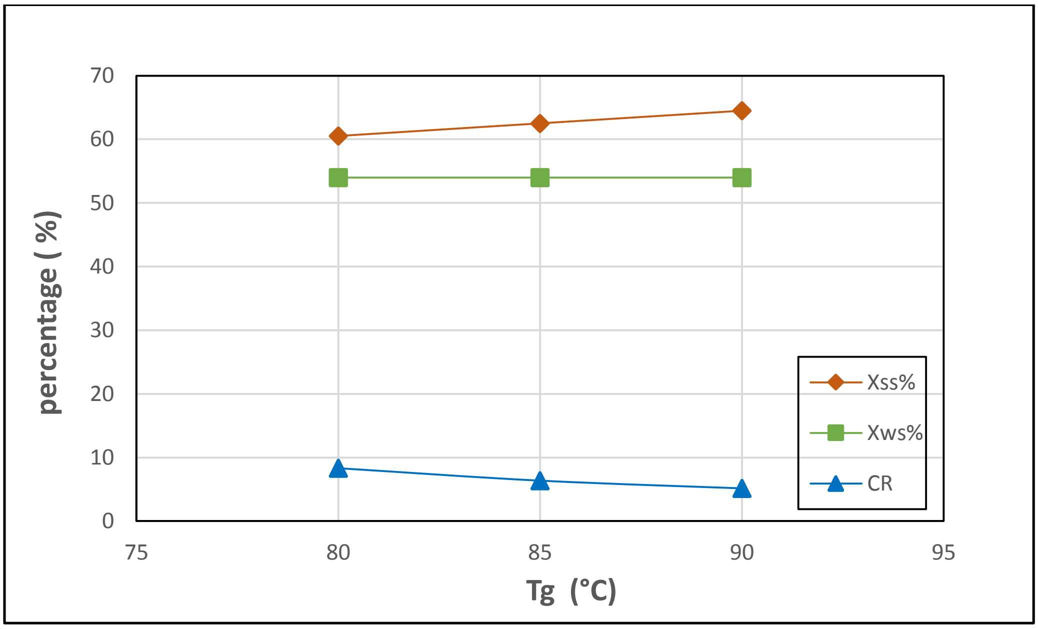


Figure 4.28: Solution concentration with generator temperature (Tg).

Figure 4.29 shows various generator temperatures with the mass flow rate of the refrigerant inside the system, where we note the values of the mass flow rate of the weak solution and strong solution of the system decrease with an increase in the generator temperature. When the condenser and evaporator temperatures of the system are constant, the mass flow rate inside the evaporator remains at a constant value because the evaporator is at a constant temperature of 8°C. The maximum value of the mass flowrate of strong solution and weak solution are at the 80°C of the generator temperature, while the minimum value of the mass flow for both the strong solution and weak solution are at 90°C because they depend on the circulation value.

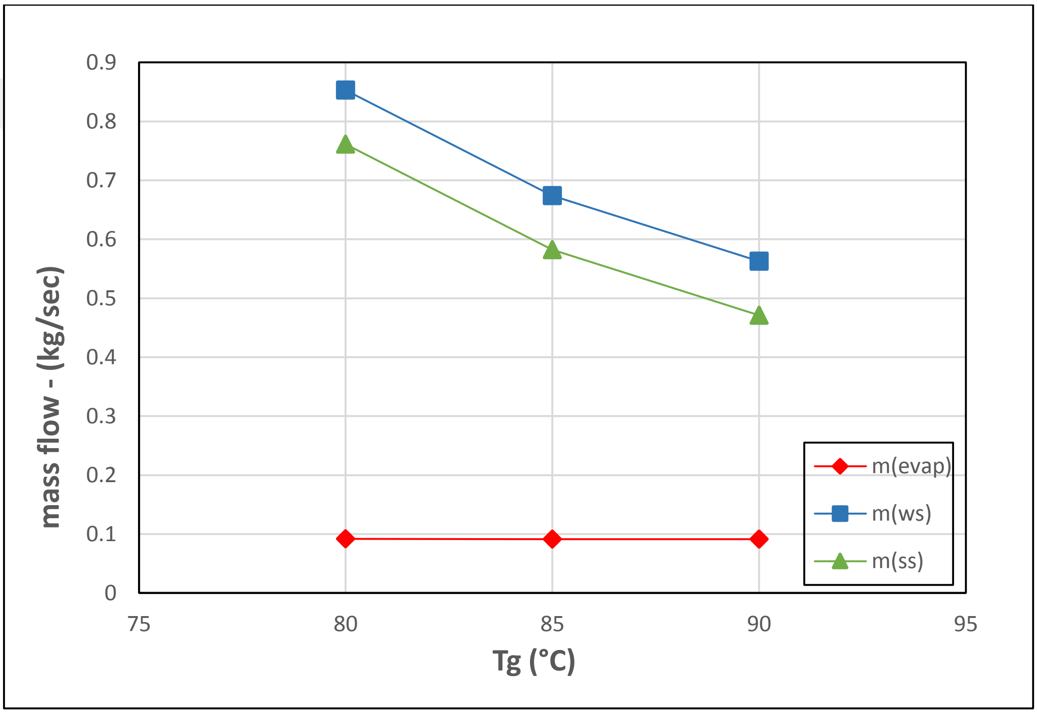


Figure 4.29: Variation mass flow with generator temperature **(Tg).**

Figure 4.30 shows the values of the COP that change with changes in the temperatures of parts of the system. Note that for each stated system setting, there exists a lower boundary of the generator temperature below which the absorption system will not operate. Above this temperature limit, the generator temperature increases the Coefficient of Performance of the system, which increases suddenly until the generator temperature reaches 90ºC depending on the condenser and absorber temperatures. The COP in this case reaches a maximum value of 85ºC and it decreases slightly after up to the crystallization point. The result and carves behavior of this case are agree with the search [2][36].

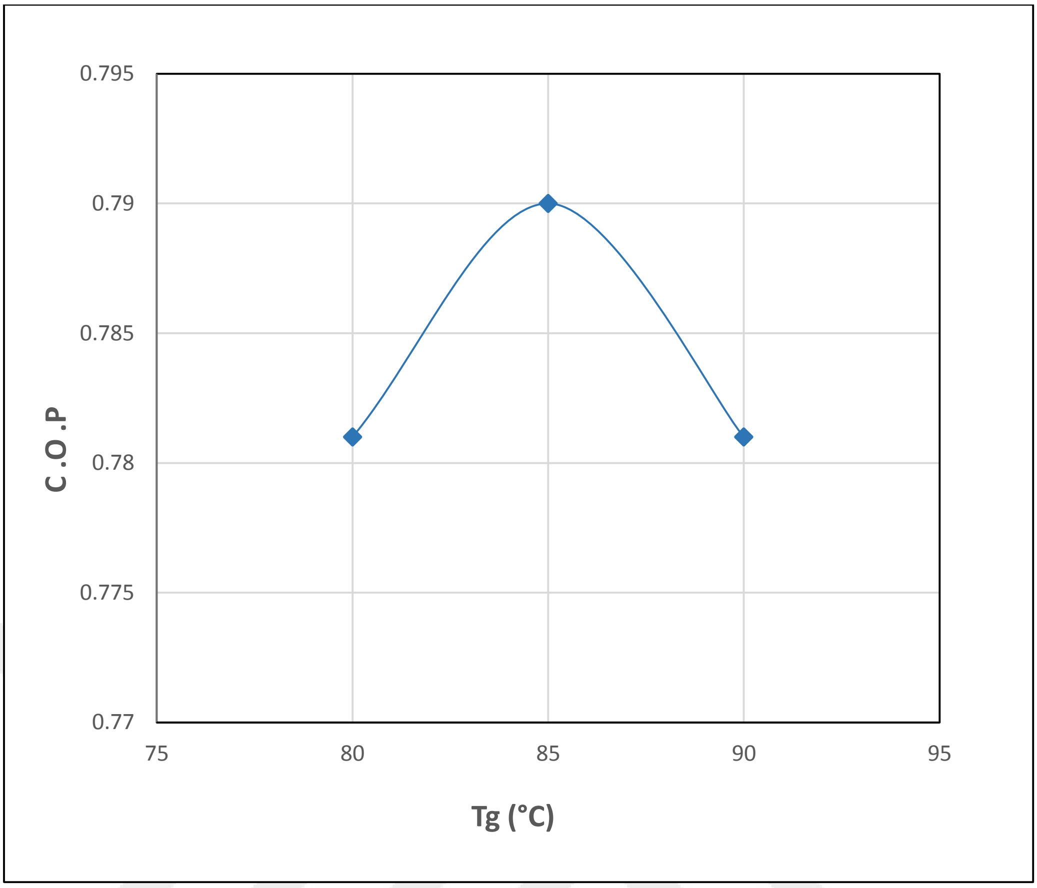


Figure 4.30: Variation COP with generator temperature (Tg).

4.3.2.2 Case Two

The analyses model for the thermodynamic properties and heat transfer rates of each element are shown in Table 4.9. In this simulation, calculations were achieved for a 217.561-kilowatt cooling load of the building and the parameters were taken on the highest COP of the system at Tc = 40 ºC, Te = 8 ºC, Ta = 35 ºC, Tg = 90 ºC, and ε = 0.71. In this state, we calculated five cases using the Excel program to design the absorption system. We selected the maximum COP for the cases that represented a suitable and design for the system. The value of the COPs differed between Cases 1 and 2 because they have different condenser temperatures, we tried three states, but in Case 2, tried four states, In Table 4.8, mass flow rates and chemical compositions are provided along with temperature, concentration, pressure and enthalpy values of the working fluids.

Table 4.8: Thermodynamic properties of state points.

P.No	T (Ċ)	P (kpa)	m (kg/sec)	X (%)	h (kj/kg)
1	90	7.375	0.092455	0	2689
2	40	7.375	0.092455	0	167.45
3	40	1.072	0.092455	0	167.45
4	8	1.072	0.092455	0	2516.2
5	35	1.072	0.716533	54	75
6	35	7.375	0.716533	54	75
7	-	7.375	0.716533	54	139.5
8	90	7.375	0.624077	66	222
9	50.95	7.375	0.624077	66	142
10	50.95	1.072	0.624077	66	142

Table 4.9: Transfer rates of the system parts.

components	Duty
Evaporator (Qe)	217.156 KW
Generator (Q_g)	285.499466 KW
Condenser (Q_c)	231.394222KW
Absorber(Q_a)	271.261244 KW
Work Pump (WP)	0.00176 KW
COP	0.77

Figure 4.31 shows the results of the heat loads in different components of the system with varying generator temperatures. A generator temperature increase of 50ºC causes a reduction in generator heat load. An increase or decrease of in the condenser and absorber temperatures causes multiple heat loads. The evaporator heat load remains constant since it has a constant temperature. The maximum value of the heat gain for the generator and absorber is 80°C and the maximum value of heat gain for the condensor is 95°C.

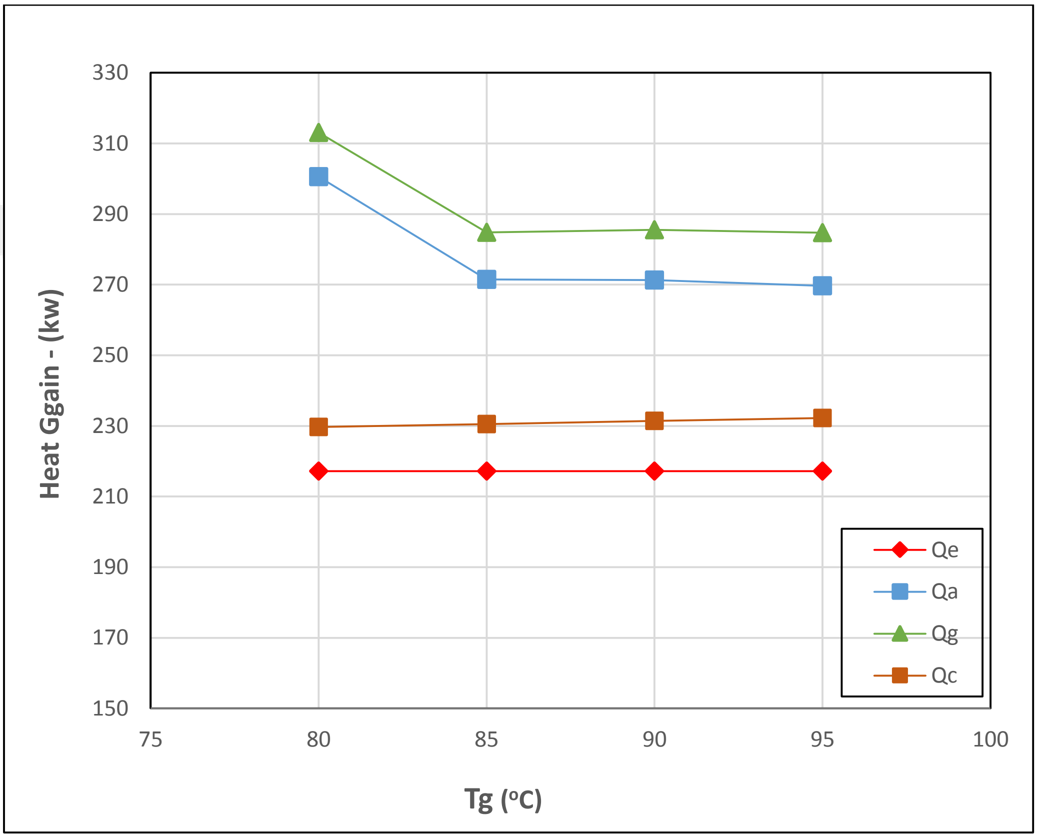


Figure 4.31: Variation heat load with generator temperature (Tg).

Figure 4.32 shows the solution concentration. The concentration of strong solution increases with an increase in the generator temperature and causes a decrease in the circulation ratio value. The weak solution remains constant at a constant absorber temperature. The maximum value of the concentration occurs at a generator temperature of 95ºC and the circulation ratio has a maximum value at 80ºC. The circulation ratio depends on the value of the strong solution, the value of which is not constant. The circulation ratio depends on the percentage of the solution on the high pressure side and low pressure side.

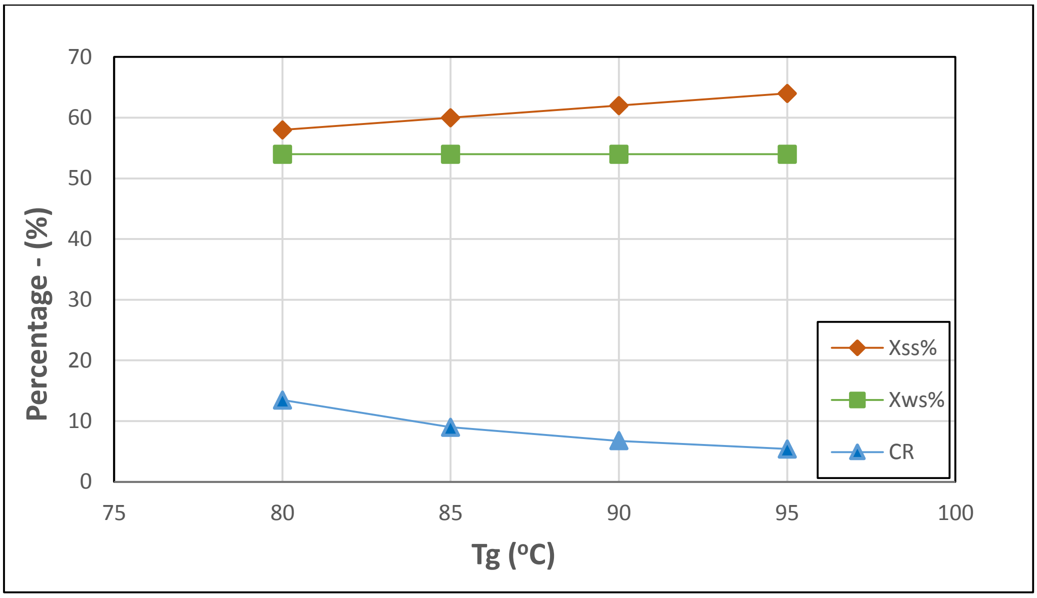


Figure 4.32: Solution concentration with generator temperature (Tg)**.**

Figure 4.33 shows the mass flow at various generator temperatures of the refrigerant inside the system. The mass flow rate of the weak solution and strong solution of the system decreases one more time with an increase in generator temperature. When we tried constant condenser and evaporator temperatures of the system, the mass flow rate inside the evaporator remained constant because the evaporator temperature was constant at 8 ºC. However, it had a different change value at different evaporator and condenser temperatures.

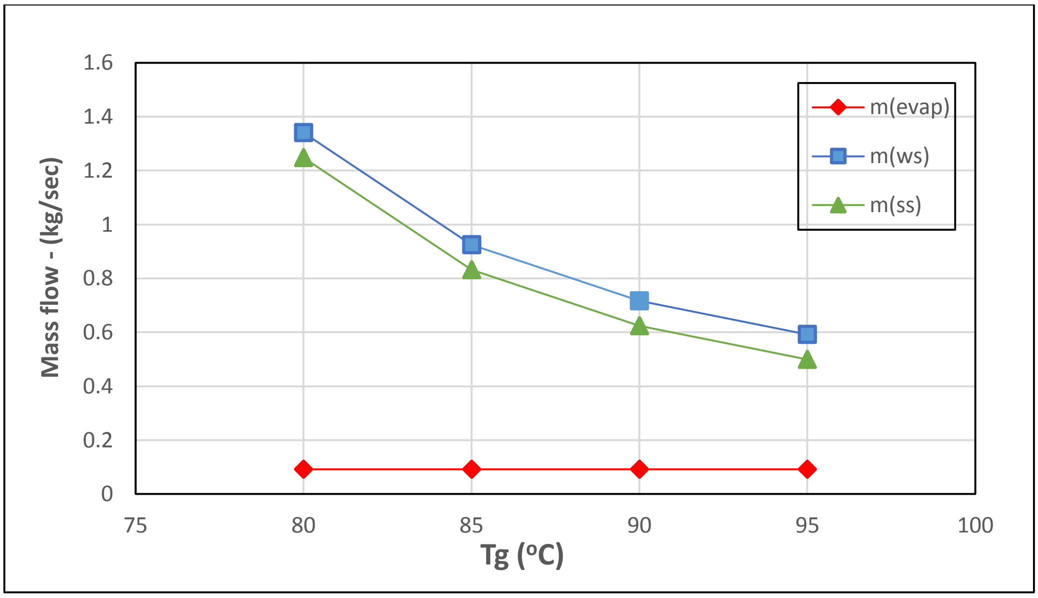


Figure 4.33: Variation mass flow with generator temperature (Tg).

Figure 4.33 shows that the COP depends on the generator temperature. However, the value of the COP changes with changes in temperature of the system parts. It can be noted that for each stated system setting, there exists a lower boundary of the generator's temperature below which the absorption system will not operate. Above this temperature limit, and as the generator temperature increases, the Coefficient of Performance of the system clearly increases until the temperature of the generator reaches 95 ºC. Depending on the condenser and absorber temperatures, the COP of the system reaches a maximum value at 90 ºC. While increasing the absorber and condenser temperatures, the COP values vary in the system. The results and the behavior of the curves of this case agree with the research [2][36].

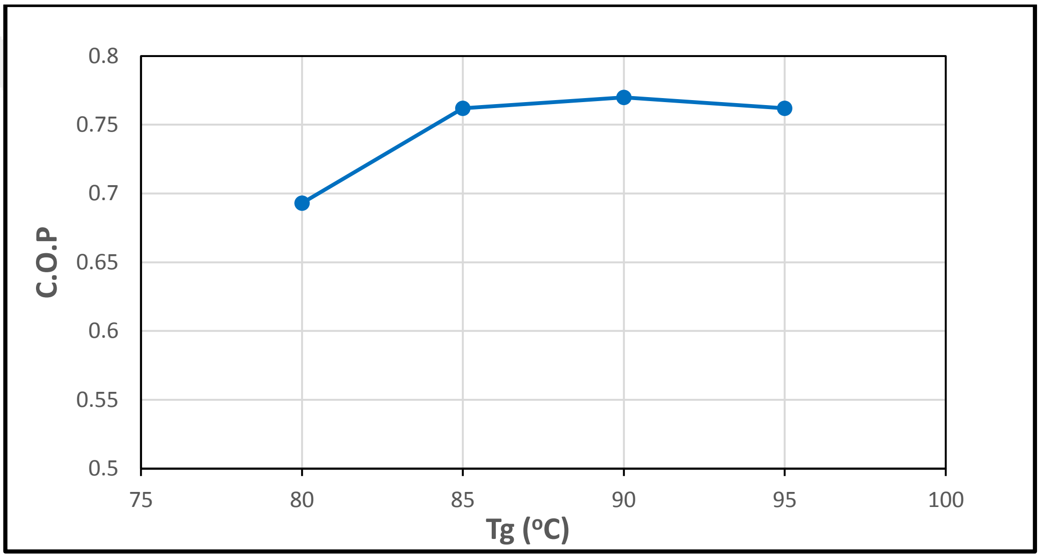


Figure 4.34: Variation COP with generator temperature (Tg).

4.3.2.3 Case Three

The results model for the thermodynamic properties and heat transfer rates of each element are shown in Table 4.30. In this simulation, calculations were achieved for a 217.561-kilowatt cooling load for the building and the parameters were taken on the highest COP of the system, namely Tc = 45 ºC, Te = 8 ºC, Ta = 35 ºC, Tg = 105 ºC, and ε = 0.71. We calculated six cases using the Excel program to analyze of the absorption system and we selected the highest COP of cases that represented a suitable design of the system. In Table 4.6, mass flow rates and chemical composition are provided along with temperature, concentration, pressure and enthalpy values of the working fluids.

Table 4.10: Thermodynamic properties of state points.

P.No	T (Ċ)	P (kpa)	m (kg/sec)	X (%)	h (kj/kg)
1	100	9.593	0.09328	0	2689
2	45	9.593	0.09328	0	188.35
3	45	1.072	0.09328	0	188.35
4	8	1.072	0.09328	0	2516.2
5	35	1.072	0.59703	54	75
6	35	9.593	0.59703	54	75
7	-	9.593	0.56293	54	148.406
8	100	9.593	0.47129	64	250
9	53.85	9.593	0.47129	64	163
10	63.85	1.072	0.47129	64	163

Table 4.11: Transfer rates of the system parts.

components	Duty
Evaporator (Qe)	217.156 KW
Generator (Q_g)	288.179357 KW
Condenser (Q_c)	233.27583 KW
Absorber(Q_a)	272.05952 KW
Work Pump (WP)	0.0027 KW
COP	0.753

Figure 4.34 shows the results of heat loads in different components of the system with varying generator temperatures. The generator temperature increases by 50 °C causing a reduction in the generator heat load. The increase or decrease in the condenser and absorber temperatures causes multiple heat loads while the evaporator heat load remains constant since the temperature of the evaporator is constant.

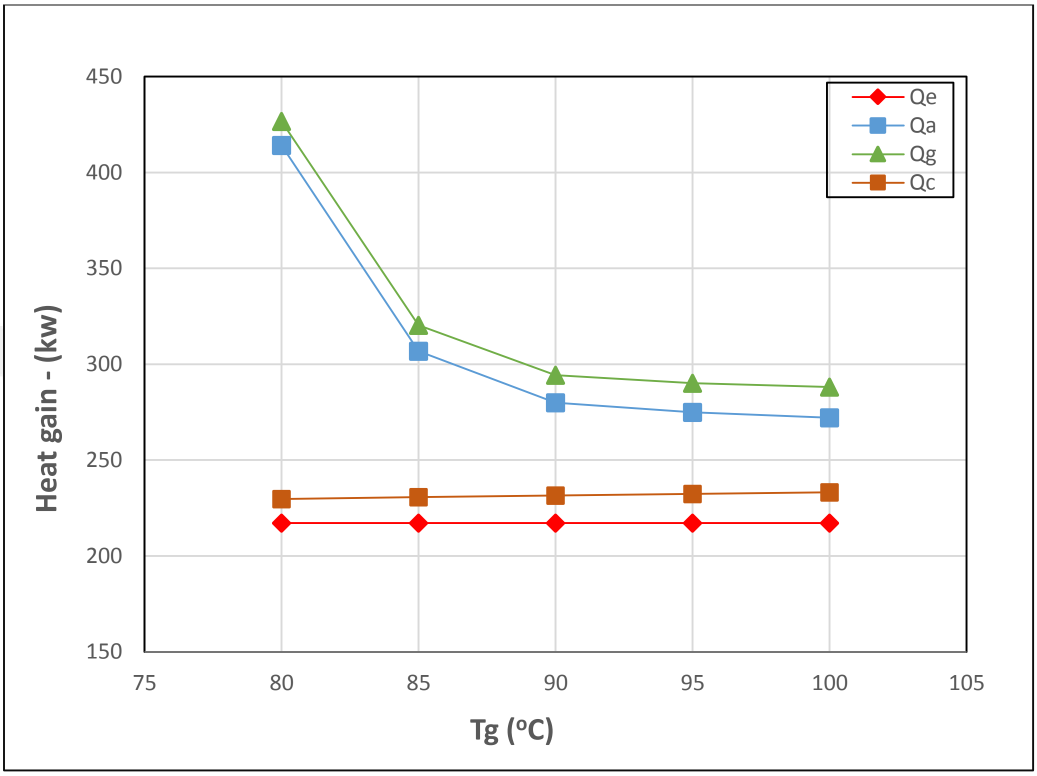


Figure 4.35: Variation heat load with Tg.

Figure 4.35 shows the solution concentration. The concentration of strong solution increases with an increase in generator temperature and it causes it causes a decrease in the circulation ratio value. The weak solution remains constant at the constant absorber temperature. Note that the maximum value of concentration at generator temperature is 100 °C and the circulation ratio has a maximum value at 80 °C. The circulation ratio depends on the value of the string solution and its value is not constant.

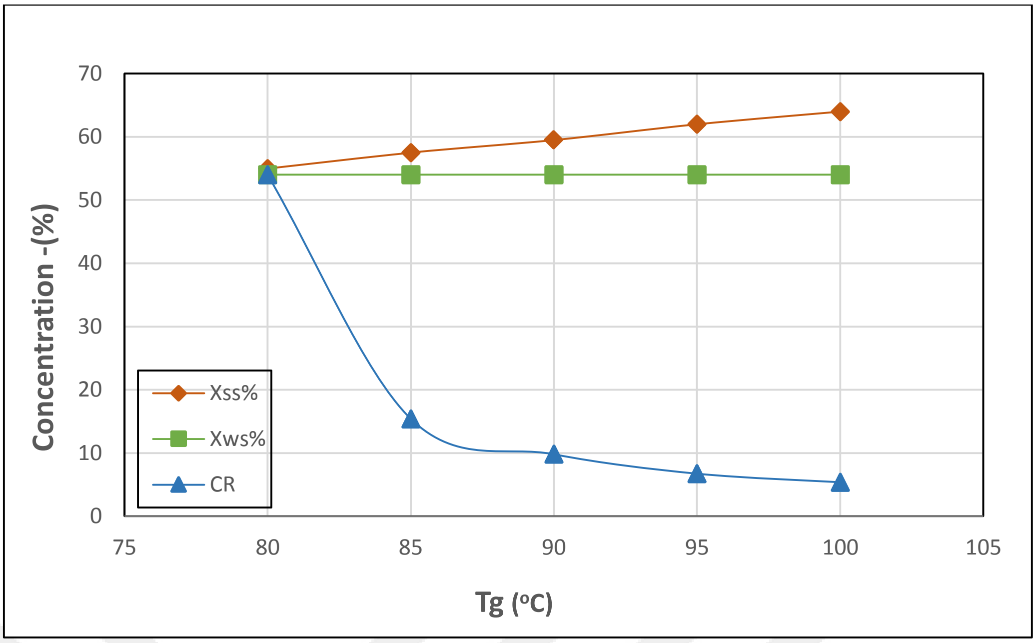


Figure 4.36: Solution concentration with generator temperature (Tg).

Figure 4.36 shows the mass flow at various generator temperatures of the refrigerant inside the system .The mass flow rate of the weak solution and strong solution of the system decreases with an increase in the generator temperature When the condenser and evaporator temperatures of the system are constant'. The mass flow rate inside the evaporator remains constant because the evaporator remains at a constant temperature of 8 °C. In this state, the mass flow reaches 100 °C for the generator reaches a temperature of 100 °C.

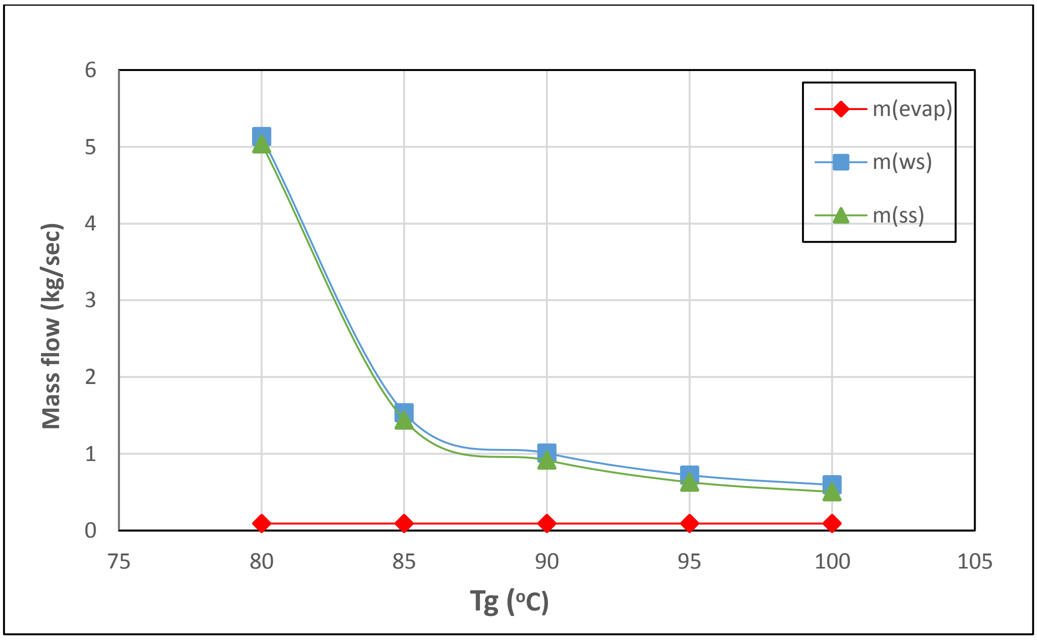


Figure 3.37: Variation mass flow with generator temperature (Tg).

Figure 4.37 shows that the pattern on which COP depends on the generator temperature is always the same. However, the value of the COP changes with changes in temperatures of parts of the system. Note that for each stated system setting, there exists a lower boundary of the generator's temperature, below which the absorption system will not operate. Above this temperature limit, as the generator temperature increase, the coefficient of performance of the system increases suddenly until the temperature of the generator reaches 100°C depending on the condenser and absorber temperatures. The COP of the system reaches a maximum value at 100 °C and decreases slightly after the crystallization point while increasing the absorber and condenser temperatures causing variations in the system COP values. The result and the behavior of the curves of this case agree with the research **[2][36]**.

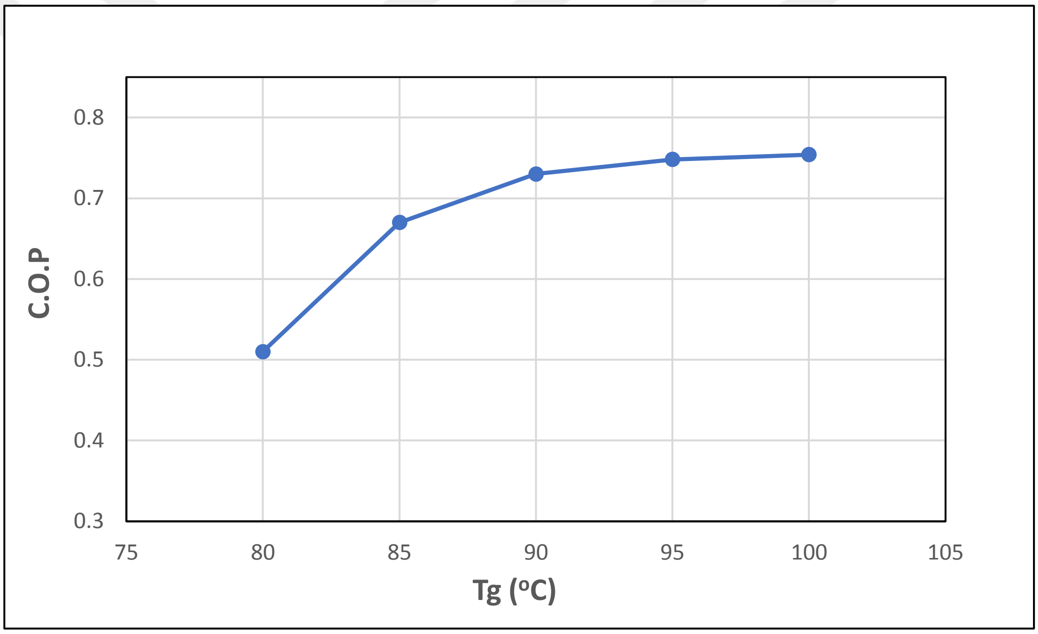


Figure 4.38: Variation COP with generator temperature (Tg).

4.4 Waste Gas Turbine Gases

In an average gas turbine engine, as we know, one-third of the energy is wasted as heat out to the atmosphere through the exhaust gas through the turbine side head. The experiment released heat through the exhaust in the Siemens Virgin V94.2 gas turbine at temperatures between 336 °C and 490 °C, which depended on the gas turbine load. This rich source of heat and the motion that is too high in speed are the most appropriate and suitable source for running and working the generator of a vapor absorption compression system. The absorption cooling system needs 287.93 kW of heat source at maximum COP operation. Table 3.16 shows the value of the heat masses exiting from gas turbine exhaust. It represents experimental values at multiple electrical generation loads of the gas turbine. The quantity of waste heat energy is enough to operate the system at each electrical load generation. The heat source from gas turbine exhaust has an advantage such that it reduces the emissions of heat gases to the atmosphere. When it uses this type of energy, the consumption of electric power is very low when another source drives the system.

4.4.1 Heat Exhaust Gases

Figure 4. 38 shows heat quantities from the exhaust gas turbine, where we note the figure below the value of the load generation is from 20 MW to 134.7 MW, and the value of the heat differs at each load. The value of the exhaust heat depends on the mass supply of air and the mass of full oil of those burning inside the compression chamber. The mass of the air is constant for loads between 1 MW and 70 MW with a variable mass of fuel which changes the mass after that value of the power load. We note the value of the heat is very high in comparison to the electric generation because the mass of air and the firing temperature are very high. The values of the results in the figure below agree with the research.

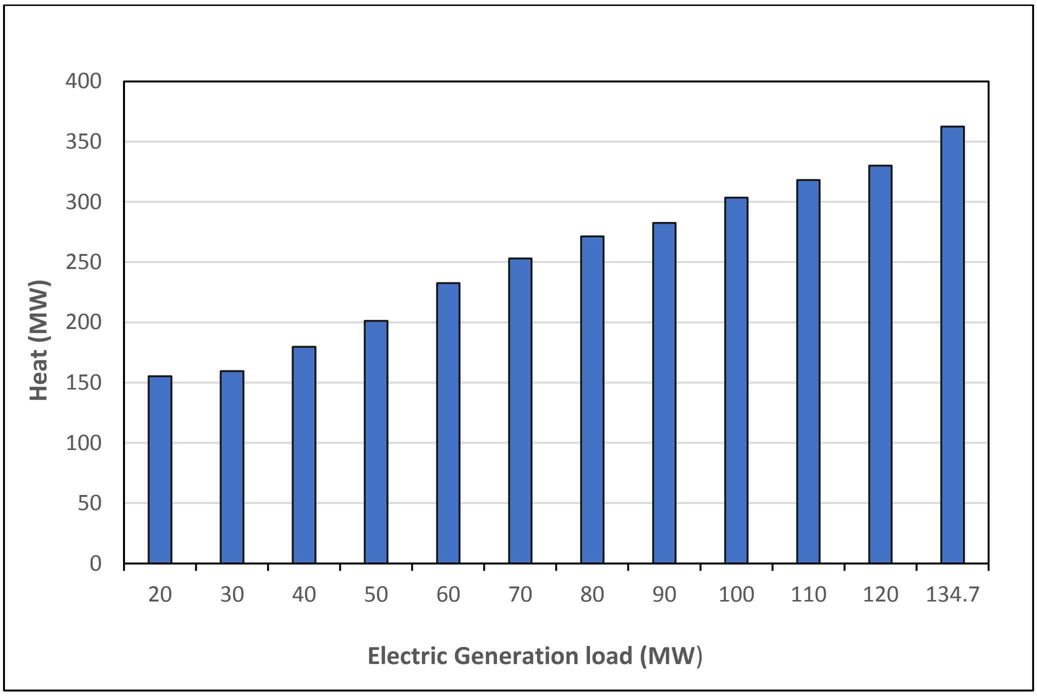


Figure 4.39: Variation heat from exhaust.

4.5 Comparison of Cost Operation between Vapor Compression Chiller System and Vapor Absorption Chillier System.

Table 4.8 shows a comparison of the operational costs between the vapor compressor and the vapor absorption chiller in Iraq and in Turkey .Note that the cost of operations of the electric chiller is higher than the absorption chiller because the part consuming maximum electrical power is the compressor of the compression chiller, while the absorption has no compressor because it uses waste heat to operate the cooling system. Another comparison of the cost of operations can been found between Iraq and Turkey such that both countries opt for industrial type of electrical cost operations.

Table 4.12: Cost comparison between the vapor compression chillers and exhaust gases absorption chiller

Cooling Capacity (TR)	Cost operation of Electric Chiller of 6 month	Cost operation of Exhaust Absorption Chiller of 6 month	Saving cost operation of 6 month	Cost by USD($)
62	6,725,565 (ID)	957726 (ID)	5,767,839 (ID)	4846.92 **[49]**.
62	22597.89 (TL)	3217.96 (TL)	19379.93 (TL)	5519.77 **[50].**

4.6 Sample of Calculation

4.6.1 Cooling Load of Room Number One for Ground Floor

To calculate the cooling load of room number one, we should find the data of the room.

Table 4.13: shows all data of the room that needs to estimate the cooling load.

Room number one	Data
Outside condition	49.5 Ċ & 15 % RH
Inside design condition	24 Ċ & 50 % RH
Room dimension of area	6 m*4 m
Room dimension volume	6 m*4 m*3.5 m
Wall area with N facing	13.75 m^2
Window area withe N facing	2.25 m^2
Number of people	5 people
Activity of the people	3 work office ,2 standing
Equipment inside the room	2 computer 19",2 laser printer
(Wo – Wi)	0.0022 kg of moisture/kg of dry air

1) **Heat Gain Through The Walls**

Equation 3.1 determines the heat through the wall. The overall heat transfer coefficient through the wall is 2.17 W/m^2 °C. From fig (4.1) and the table 3.5 the (CLTDc) is 18.4 at 3 pm. The wall area for the room No.1 is 13.75 m^2 and taken from the table 4.13.

$Q = 13.75 \times 2.17 \times 18.4$

$Q = 549.01$ watt

2) **Conduction of Heat Gain The Through Glass**

Equation 3.4 predicts heat conduction through the window glass. The heat transfer coefficient U for the window glass is taken as 6.13W/m^2. °C. From fig 4.1, the room had aluminum frame window with single glass and the $CLTD_C$ is 21.1 at 3 pm. The glass is 2.25 m^2 and taken from the table 4.13.

$Q = 2.25 \times 6.13 \times 21.1$

$Q = 291.1$ watt

3) **Solar Radiation Through Glass**

Equation 3.6 estimates solar heat radiation through the window glass. The shading coefficient SC is taken as 1. The solar heat gain factor SHGF is taken as 124 W/m^2 from the table 3.8 and the cooling load factor is taken 0.82 from the table 3.9. The area of the glass is taken from the table 4.13.

$Q = 2.25×1×0.82×124$

$Q = 228.78$ watt

4) **Heat Generation From Occupant**

Equations 3.7 and 3.8 estimate heat gain of the people both sensible and latent heat gain. The sensible heat gain (SHG) which produced by each people is 75W and 55W for latent heat of the working office. 70 W for sensible heat and 45W for latent heat of the sitting. Table 3.10 shows the heat rate for occupant. The cooling load factor CLF for one occupant is equal to 1. The occupant number and their activities data is taken from the table 4.13.

$Qs = (75×3×1) + (70×2×1)$

$Qs =365$ watt

$Ql= (55×3×1) + (45×2×1) = 155$ watt

5) **Heat Gain From Lighting Equipment**

Equation 3.9 gauges the heat gain from lighting equipment and from table 3.11 the value of the lighting power is 12 W/m^2. The fluorescent lights have its own value ballast factor. FC is 1.2. In this case the cooling load factor CLF is 1. All data of lighting equipment for the room No.1 is taken from the table 4.13.

$Q_{light} = 12×24×1.2×1$

$Q_{light} = 345.6$ watt

6) **Heat Gain From Office Equipment**

Equation 3.10 predicts the heat gain from the office equipment and from the table 3.11 the values of the generated heat for office equipment are 135 and 130 W. The case the cooling load factor CLF takes 1.0. Data of equipment for the room number one takes from table 4.13.

$Q_{equ} = [(2×135) + (2×130)] ×1$

$Q_{equ} = 530$ watt

7) Heat Load From Infiltration

Equations 3.12 and 3.13 calculate heat gain from infiltration load and from the table 3.13 the value of air change per hour for the space is 1.0. Data of infiltration heat gain of the room number one takes from table 4.13.

$Qs = 1.22\times0.023\times25.5\times1000$

$Qs = 715.53$ watt

$Q_L= 3010\times0.023\times0.0022\times1000$

$Q_L = 152.31$ watt

8) Heat Load From Ventilation

Equations 3.12 and 3.13 predict the heat gain from infiltration load and from table 3.14 the quantity of air required per second for the person is 0.0125 m^3/sec. Data of infiltration heat gain of room number one takes from table 4.13**.**

$Qs = 1.22\times0.0125\times25.5\times1000$

$Qs = 388.9$ watt

$Q_L = 3010\times0.0125\times0.0022\times1000$

$Q_L= 82.8$ watt

9) Total Sensible Heat Load

Equation 3.15 predicts the total sensible heat load of the room

$Qs = 549.01+ 291.1+ 228.78+365+ 345.6 + 530 + 715.53 + 388.9$

$Qs = 3413.81$ watt.

10) Total Latent Heat Load

Equation 3.16 predicts the total latent heat load of the room

$Q_L= 255+152.31+82.8$

$Q_L= 490.11$ watt.

11) Total Cooling Load

Equation 3.14 estimates the total cooling load of the room and multiply by 10% watch represents the safety factor.

$Q_T = (3413.81+490.11) \times10$ %

$Q_T = 4294.28$ watt.

12) Sensible Heat Factor (SHF)

Equation 3.17 predicts the sensible heat factor (SHF) of the room and the value of sensible heat factor depends on the sensible heat load and latent heat load of

$$SHF = \frac{3755.19}{4294.28}$$

SHF = 0.874

4.6.2 Absorption Cooling System Cycle

To analyze the cooling absorption system cycle, we should have some data to estimate the system.

Table 4.14: Shows all necessary data to study absorption cooling cycle system**.**

Part name	Data
Heat evaporator	217.156 kw
Generator temperature	100 °C
Condenser temperature	45 °C
Evaporator temperature	8 °C
Absorber temperature	35 °C
Effectiveness of heat exchanger	71%

The condenser has liquid water and the evaporator has water vapor, so the pressure values for the condenser and the evaporator of the system are taken from stem table which depends on Tcon and T_{eva}.

P1= P2 = P8 = P7 = P6 = P9 = 9.593 kpa at T_{con} = 45 °C

P3 = P4 =P5=P10= 1.072 kpa at T_{eva} = 8 °C

By drawing line between T_g and P_g on the water –lithium bromide chart the concentration of week solution value (Xws) is 54%. Whereas, the concentration of the strong solution value (Xss) is 64% by drawing line between T_{abs} and P_{abs} on water –lithium bromide chart.

The circulation ratio is produced by dividing the mass flow rate of the solution which comes from the generator to the mass flow rate of the working fluid.

$$\lambda = \frac{0.54}{0.64 - 0.54} = 5.4$$

Enthalpy value of pure water and of superheated water vapors at each temperature can be predicted from the steam table or by simple equation (3.23).

h_1 = 2501+1.88(100) = 2708 kj/kg at T_{gen} = 100 °C

$h_2 = h_3$ = 209 kj/kg at T_{con} = 45 °C

h_4 = 2516.2 kj/kg at T_{eva} = 8 °C

The mass of the refrigeration fluid in evaporator is fixed in this case, so we can find it by applying equation 3.24.

$m_1 = m_2 = m_3 = m_4$ = 217.156/ (2516.2 − 209)

= 0.093286 kg/sec

$m_{ws} = \lambda \times m$ = 0.59703 kg/sec

$m_{ss} = \lambda \times m$ = 0.503744 kg/sec

From the formula of effective heat exchanger, the temperature of exiting heat exchanger (T_9) and the strong solution exiting the generator (T_8) can be obtained since T_6 is almost equal to T_5 and T_8 is the same to the generator temperature Tg.

$$0.71 = \frac{100 - T9}{100 - 35}$$

T_9 = 58.85 °C

Enthalpies value of solutions are predicted from LiBr-Water Temperature-Concentration-Enthalpy (T - X -h) Chart appendix.

h_5 = 75 kj/kg at T_{ab} = 35 °C and Xws = 54%

h_6 = 75 kj/kg at T_{ab} = 35 °C and Xws = 54%

h_8 = 250 kj/kg at T_g = 100 °C and Xws = 54%

$h_9 = h_{10}$ =163 kj/kg at T_g = 53.85°C and Xws=54%

The enthalpy value at point 7 is predicted by heat balance across heat exchanger.

$h_7 = h_6$ + mss $(h_8 + h_9) / m_{ws}$

h_7 = 75 + 0.50374 (250 -163) / 0.59703

h_7 = 148.406 kj/kg.

1) **Condenser**

Equation 3.22 predicts the heat rejected in the condenser.

Q_{con} = 0.09328 (2689 – 188.35)

Q_{con} =233.275835 kw.

2) Absorber

Equation 3.25 predicts the heat rejected in the absorber.

Q_{abs} = 0.093286**[(**2516.2 − 75) + 5.4(163 – 75)].

Q_{abs} = 272.05952 kw.

3) Generator

Equation 3.20 predicts the heat additive in the absorber.

Q_{gen} = 0.093286 × [(2689 – 148.406) + 5.4 (250 -148.406)].

Q_{gen} = 288.17952 kw.

4) Co-efficient of performance (COP)

Equation 3.27 calculates the COP of the system.

COP = $\frac{217.156}{288.17952}$ = 0.753.

Then we can find the total value of added heat and rejected heat to the absorption cooling cycle.

Q_{add} = Q_{gen} + Q_{eva}.

Q_{add} = 288.17952 + 217.156

Q_{add} = 505.335 kw

Q_{rej} = Q_{con} + Q_{abs}

Q_{rej} =233.275835 + 272.05952

Q_{rej}= 505.335 kw.

So we can get equilibrium state of the cycle. In other word (Q_{add} = Q_{reg})

CHPTER FIVE

CONCLUSION AND FUTURE WORK

5.1 Conclusion

In this study, a building and a combined part of a research institution located in Salahuddin city (Iraq) was measured to calculate cooling loads. The Cooling Load Temperature Difference (CLTD) method was used to estimate the cooling load for the summer season (in the month of June). Cooling heat load components, such as people, light, ventilation and filtration, can be calculated with an Excel spreadsheet. Moreover, an Excel spreadsheet can also be used to estimate cooling loads through walls and roofs. The results of the calculations show that the cooling load values were different for the spaces as they were dependent on the direction of the space, any inside activity, and the volumes of the spaces. The total cooling load for the air conditioning required for the building was 62 tons for the summer (in the month of July). The average sensible heat ratio of the building is 0.911 for the summer season. The air supply was calculated for the building giving a result of 14.22 m^3/sec. The results showed that the cooling load calculation was properly performed for the latent heat coming from the people and filtration.

The lithium bromide water absorption cooling cycle was analyzed, with thermodynamics. The coefficient of performance (COP) of this cycle versus the generator temperature and condenser temperature was analyzed and it was noticed that the generator temperature was important factor at any moment to consider the optimum temperature at which an absorption refrigeration cycle operates. The variations of the generator temperature and condenser temperature with the mass flow rate ratio determine the maximum temperature that can be used at the generator in order to achieve the maximum COP value from the system. The analyses were performed for multiple temperatures and pressures at the generator and condenser and at a constant pressure and temperature for the evaporator and absorber. The study must continue in order to obtain operational maps that include the heat exchanger efficiency as a variable. The COP values and crystallization state depended upon the heat exchanger efficiency. Providing a minimum value of waste heat producing additional cooling for the absorption cooling system by recovering the waste heat from a 134.7-megawatt electricity generation process may save 217.156 KW of electricity consumption.

Most of the work energy used in electricity production and transportation is produced by burning fossil fuels such as coal, oil and natural gas. Increasing costs of fuels and concerns for climate change have sparked public interest in combined cooling, heating and power technologies which simultaneously provide electrical power, heat and cooling from a single energy source. The use of cooling system technologies increases the overall energy conversion efficiency and it consequently reduces emissions .On the other hand, the refrigerant of an absorption cooling chiller (water-LiBr) is not harmful to the ozone layer of the atmosphere, which directly prevents contamination by ultraviolet (UV) rays on the Earth. In this study, waste exhaust gases were analyzed as a potential application of an absorption cooling system. The water-LiBr powered by waste heat from the power-generating gas turbine was analyzed. The modeling results demonstrate that the single-effect water-LiBr could completely provide the required cooling load.

When utilizing waste heat recovered from gas turbine exhaust and as far as costs are concerned, electricity costs for the plant operation and its depreciation were considered. However, there may be other operational costs involved with the cooling tower, cooling water pump and chilled water pump. We compared the vapor compression chiller and exhaust (generator exhaust flue gas) driven absorption chiller and the analysis showed estimated savings on operation costs of $5,767,839 (ID) ($4846.92 USD per 6 months in Iraq) and 19,379.93 TL ($5519.77 USD) per 6 months (in Turkey). Where we note a comparison of cost operation values between Iraq and Turkey, it shows operational cost savings in Turkey that are higher than in Iraq when operating simultaneously under identical conditions.

5.2 Recommendation for Future Work

Several improvements could be made to make the following recommendations of this thesis.

- **Cooling of building**
  1. Use insulators to reduce heat transfer through the walls and roofs.
  2. Use double glazing for the windows to reduce heat transfer through the windows.
  3. When designing the building, its position should be taken into account when distributing windows and large area walls in directions of lower temperature.
- **Cooling absorption system cycle**
  1- This system can be used in a practical state.
  2- Any change can be applied to increase the COP value or an additional simple design technique.
  3- The effects of any other parameter can be studied.
- **Gas turbine exhaust gases**
  1- This system can be used in an experimental state.
  2- Surplus waste gases can be used with a compound system cycle to generate steam vapor.
  3- Surplus waste gases can be used in a vapor steam injection technique to improve gas turbine performance.

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APPEEDIX

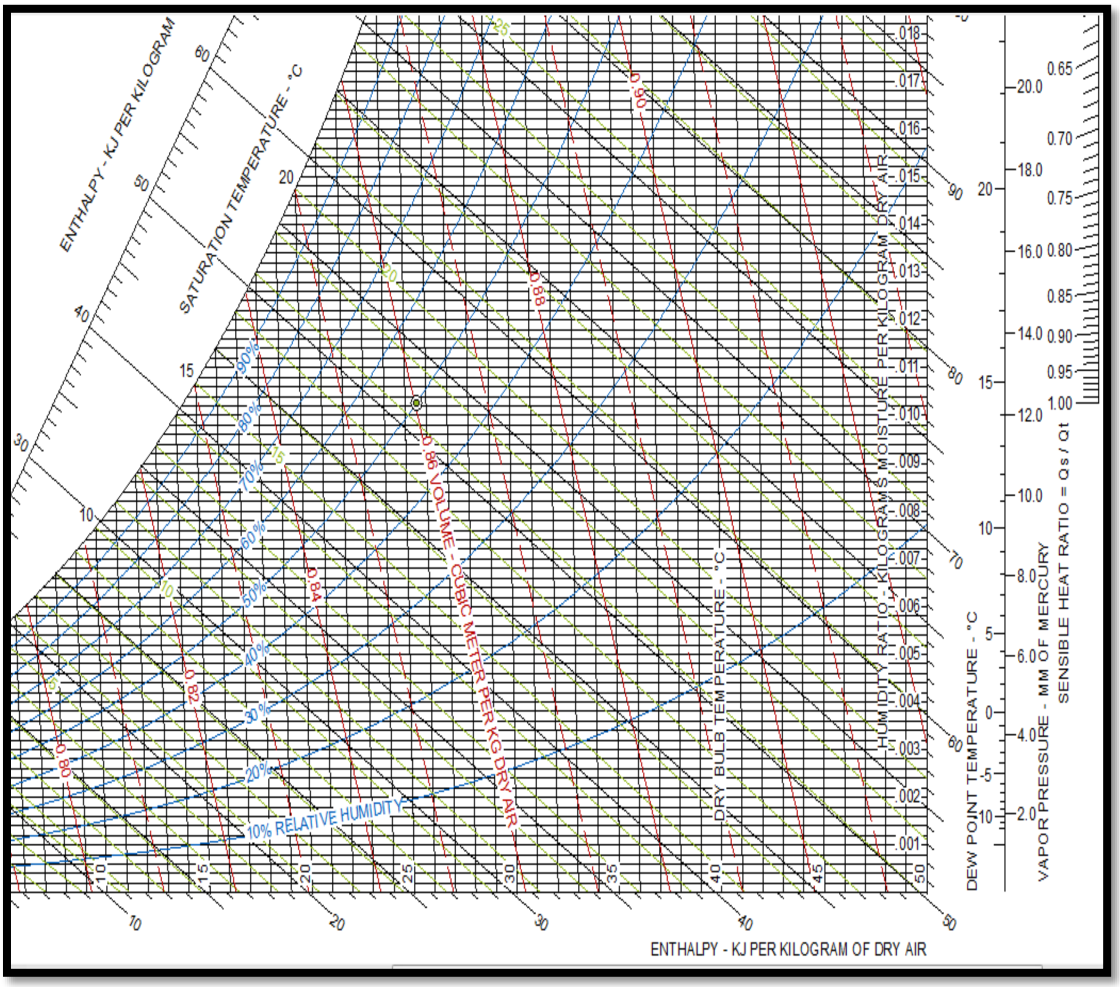

Fig A1. 1: Psychometric Chart.

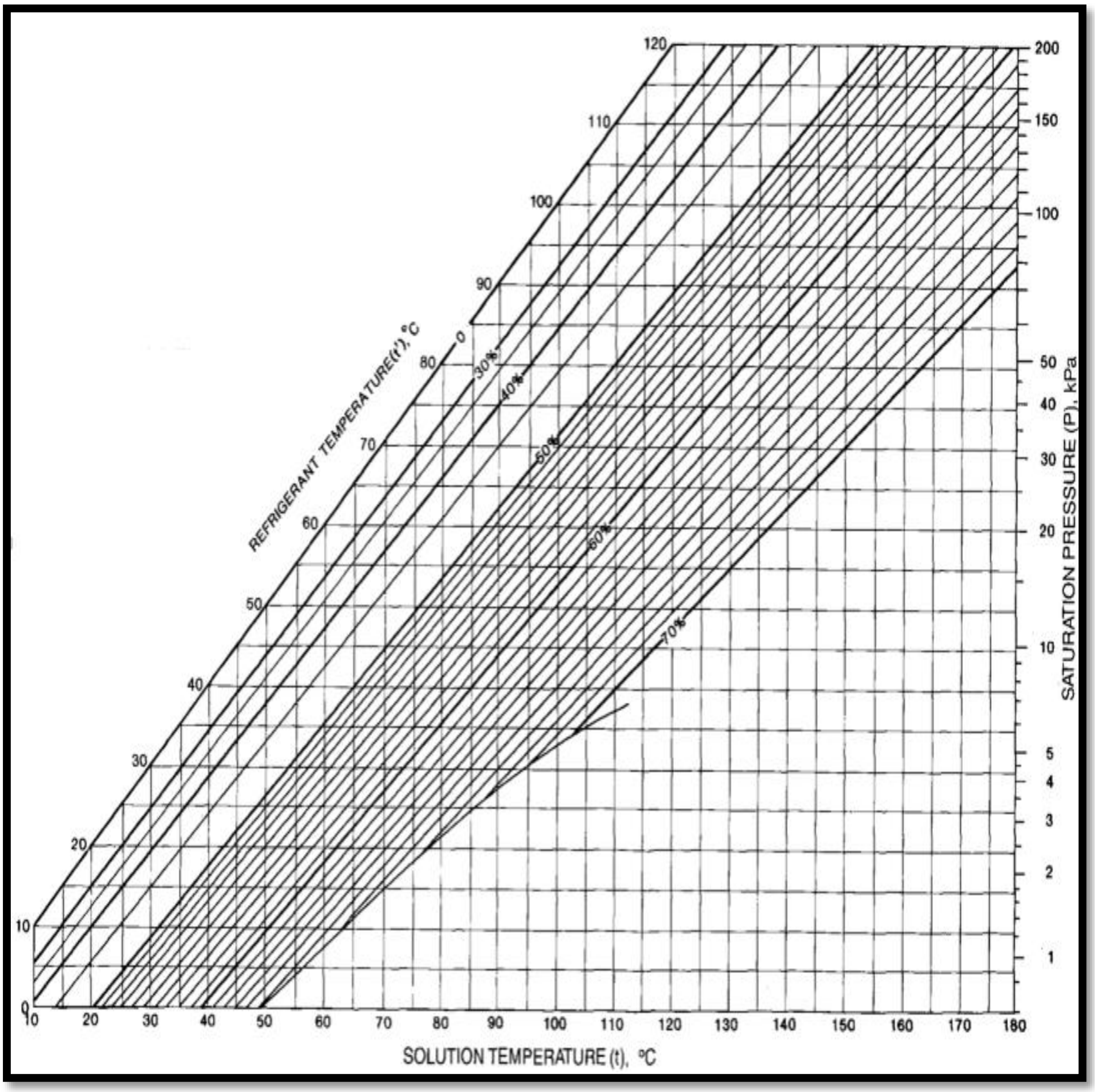

Fig A1. 2: Equilibrium chart for water lithium bromide solutions.

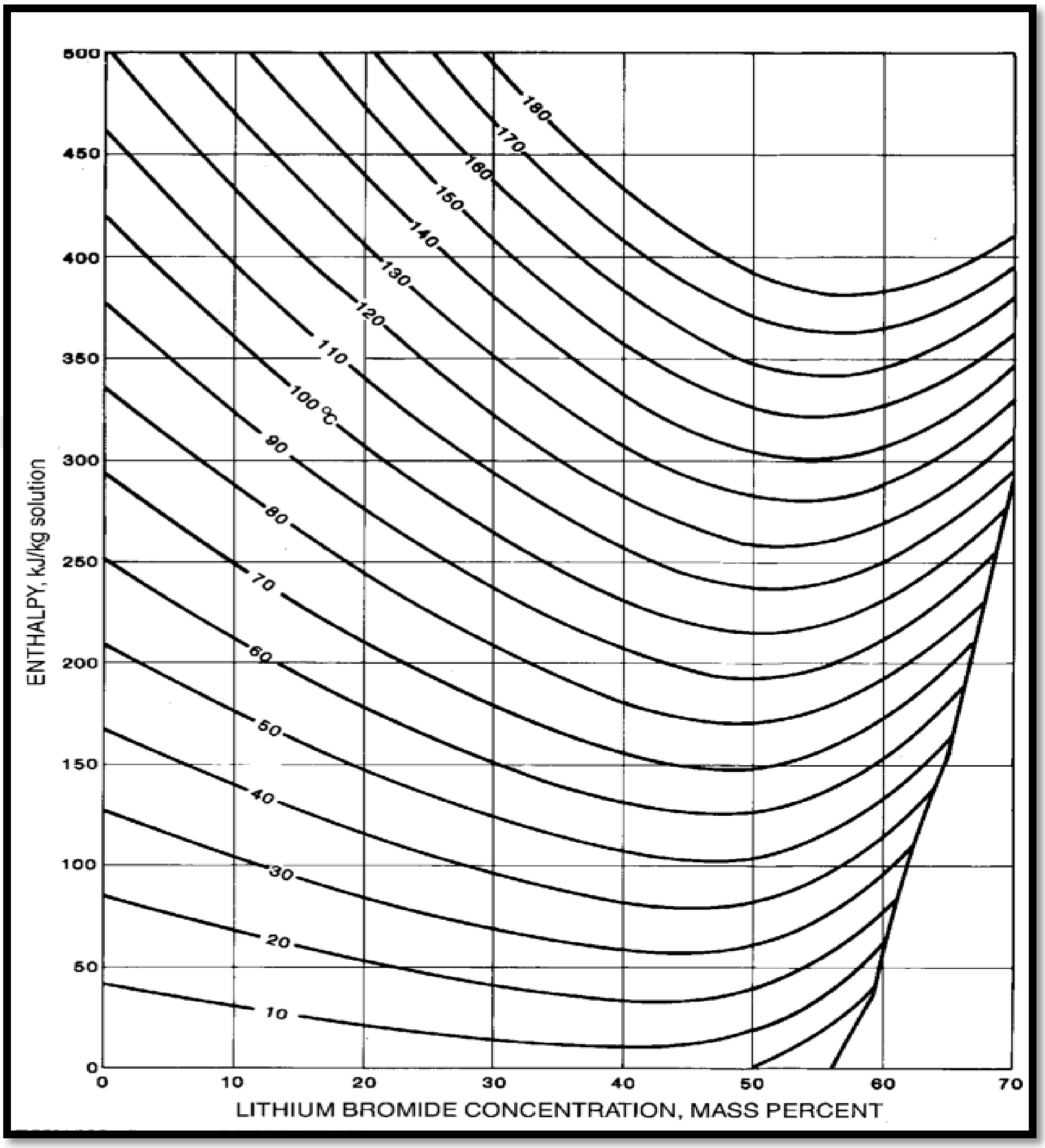

Fig A1. 3: Enthalpy-Concentration Diagram for Water-Lithium Bromide Solutions.