Sudan University of Science and Technology
College of Graduate Studies

A Design of Pre Cooler for Gas Turbine at Garri (1&2) Power Station Using an Absorption Solar Refrigeration System

تصميم مبرد قبلي للتوربين الغازي في محطة كهرباء قري الحرارية (1&2) بإستخدام نظام التبريد بالدورة الإمتصاصية العاملة بالطاقة الشمسية

A Thesis Submitted in Partial Fulfillment of the Degree of M.Sc. in Mechanical Engineering (POWER)

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الاستهلال

بسم الله الرحمن الرحيم

قال تعالي:

(لَا يُكَلِّفُ اللَّهُ نَفْسًا إِلَّا وُسْعَهَا لَهَا مَا كَسَبَتْ وَعَلَيْهَا مَا اكْتَسَبَتْ رَبَّنَا لَا تُؤَاخِذْنَا إِنْ نَسِينَا أَوْ أَخْطَأْنَا رَبَّنَا وَلَا تَحْمِلْ عَلَيْنَا إِصْرًا كَمَا حَمَلْتَهُ عَلَى الَّذِينَ مِنْ قَبْلِنَا رَبَّنَا وَلَا تُحَمِّلْنَا مَا لَا طَاقَةَ لَنَا بِهِ وَاعْفُ عَنَّا وَاغْفِرْ لَنَا وَارْحَمْنَا أَنْتَ مَوْلَانَا فَانْصُرْنَا عَلَى الْقَوْمِ الْكَافِرِينَ)

سورة البقرة – الآية (286)

Dedication

To my father who gives me direction to the sky

To my mother who gives me lovely life

To my brothers and sisters who give me support

To my wife who gives me wormed life

Acknowledgement

First of all I would like to submit my best greeting to my supervisor Dr. Abdallah Mokhtar Mohammed for his acceptance to supervise my research and for his directions during preparing this research. Also my full respect and appreciation to my all friends, colleagues and co-worker enhanced me in this research and helped me in data collection stage. Finally, a lot of thanks to Garri Power Station Directorate which gave me permeations to apply this research and analysis in the station and has been supplying me by all data I needed throughout the research.

Abstract

Several gas turbines are being widely used for power generation in several countries all over the world. Obviously, many of these countries have a wide range of climatic conditions, which impact the performance of gas turbines. Problems arise when a gas turbine is used in a geographic location with hot summers. Hot inlet air results in a gas turbine's generating less power, during summer season, when the demand for electricity is possibly higher. In such conditions, power augmentation techniques are highly desirable. Indeed, a little increment of thermal efficiency could result in a significant amount of fuel being saved and a higher level of power being generated. The climate of sudan is the semi desert climate influenced by the north-south movement of dry northerly winds and moist southerly winds that produce a wet summer and a dry winter .Due to high ambient temperature in the area at summer average of 42°C the gas turbines operate at 75% of their rated capacity at the standard ISO conditions as GE Product (the air inlet conditions are: air temperature 15 °C, relative humidity 60%, absolute pressure 101.325 kPa at sea level power out put 42MW).The objectives of this research an attempt to design a chiller that works in turn absorbed solar energy to cool the cooling medium (water) that cools the air at the inlet of the turbine to operate the gas turbine on standard conditions. Through numerical analysis, the temperature of the air entering the gas turbine dropped to 15 ° C and the power out put was increased by 40%. As well as efficiency increased to 32%.the important recommendation is Study the effect of the cooler to the related systems like steam turbine and it's auxiliary, The design and construction of the new plants should be also take into account local environmental conditions so that the cost of the pre cooler can be included in the overall project cost.

المستخلص

يتم استخدام التوربينات الغازية على نطاق واسع لتوليد الطاقة في العديد من البلدان في جميع أنحاء العالم. العديد من هذه البلدان لديها مجموعة متعددة من الظروف المناخية، والتي تؤثر على أداء توربينات الغاز .تنشأ المشاكل عند استخدام التوربينات الغازية في موقع جغرافي مع صيف حار. ينتج عن هواء المدخل الساخن توليد طاقة أقل للتوربينات الغازية خلال موسم الصيف، عندما يكون الطلب على الكهرباء أعلى. في مثل هذه الظروف ، تكون تقنيات زيادة الطاقة مرغوبة للغاية. في الواقع، يمكن أن تؤدي الزيادة البسيطة في الكفاءة الحرارية إلى توفير كمية كبيرة من الوقود وتوليد مستوى أعلى من الطاقة. مناخ السودان هو مناخ شبه صحراوي يتأثر بحركة الشمال والجنوب للرياح الشمالية الجافة والرياح الجنوبية الرطبة التي تنتج صيفًا رطبًا وشتاءً جافًا.نتيجة الي ارتفاع درجة الحرارة المحيطة في المنطقة في الصيف الي 42 درجة مئوية ، تعمل التوربينات الغازية بنسبة 75٪ من قدرتها االتصميمية في ظروف ISO القياسية لمنتجات GE (درجة حرارة الهواء 15 درجة مئوية، الرطوبة النسبية 60٪، والضغط المطلق 101.325 كيلو باسكال في مستوى سطح البحر تكون القدرة المنتجة 42 ميقاوات). الغرض من هذا البحث هو محاولة لتصميم مبرد يعمل بدوره امتصاصية مستخدمة الطاقة الشمسية لتبريد وسيط التبريد (الماء) الذي يبرد الهواء عند مدخل التوربينات لتشغيل التوربينات الغازية في الظروف القياسية.من خلال التحليل العددي ، انخفضت درجة حرارة الهواء الداخل التوربينة الغازية إلى 15 درجة مئوية واذدادت الطاقة الناتجة بنسبة 33٪. وكذلك الكفاءة إلى 32٪.التوصية الاهم هي دراسة تأثير المبرد على الأنظمة ذات الصلة مثل التوربينات البخارية وملحقاتها، ايضا يجب الاخذ في الاعتبار عند تصميم وانشاء المحطات الجديدة مراعاة الظروف البيئة المحلية حتي يتسني تضمين تكلفة مبرد الهواء القبلي ضمن التكلفة الكلية للمشروع.

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Symbols:

$\dot{m}_{ch}$	Mass flow rate of chilled water	$^{kg}/_{sec}$
T_{chi}	Inlet temperature of chilled water	^{O}C
T_{cho}	outlet temperature of chilled water	^{O}C
T_{ai}	Inlet temperature of air	^{O}C
T_{ao}	outlet temperature of air	^{O}C
cp_a	Specific Heat of air	kJ/kg k
cp_{ch}	Specific Heat of chilled water	kJ/kg k
Q_a	Heat lost by air	Kw
Q_{ch}	Heat gained by chilled water	Kw
V_{ch}	Volume flow rate of chilled water	$m^3/_{sec}$
A_o	Total Area for Heat Exchanger	m^2
U	Overall Heat Transfer Coefficient	w/m^2k
F	Corrector Factor	%
LMTD	Log Mean Temperature Difference	^{O}C
Do	Outer Diameter of large pipe	Mm
Di	Inter Diameter of large pipe	Mm
Do	Outer Diameter of small pipe	Mm
Di	Inter Diameter of small pipe	Mm
a_i	inlet Area of small pipe	m^2
a_o	Outlet Area of small pipe	m^2
a_s	Surface Area of small pipe	m^2
L	Length of small pipe	M
N_{row}	Number of rows for Heat Exchange	rows
q_{supply}	Volume flow rate of chilled water for large pipe	m^3/sec
v_{tube}	Velocity of chilled water for small pipe	m/sec
R_T	Total Heat Resistance	K/W
ST	Transverse pitch	Mm
SL	Longitudinal pitch	Mm
N_{tubes}	Total Number of Tubes	Tubes
Nst	Number of cooling stages	Stages

List of Abbreviations

h	specific enthalpy (kJ/kg)
hfg	latent heat of vaporization of water (kJ/kg)
QCL	total cooling load (kW, RT)
T	temperature (1C)
W	overall heat transfer coefficient (W/m2 K)
V	volume of stored water (m3)
U	humidity ratio (kg of moisture/kg of dry air)
Subscripts a,b,c, d	state points
Amb	ambient
CHW	chilled water
CHWR	chilled water return
CHWS	chilled water supply
CL	cooling load e electrical
ISO	international standards organization
th	thermal
AC	absorption chiller
CC	combined cycle
Comp	air compressor
ECON	economizer
EVAP	evaporator
ST	generator
GT	gas turbine
HE	heat exchanger
HRSG	heat recovery steam generator
LHV	low heating value
NG	natural gas
RT	ton-refrigeration
G	steam turbine
k	thermal conductivity of tube material
K1	factor
kl	factor
K2	thermal conductivity of liquid (W/m K)
U	Nusselt number
Pr	Prandtl number
Prs	solution Prandtl number
Re	solution Reynolds number for vertical tube
Red	Reynolds number
NuD	average overall heat transfer coefficient (W/m2 K)

CHAPTER I

INTRODUCTION

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INTRODUCTION

1.1 Introduction:

Sudan began the electricity industry in 1908 the first 100KW generator was installed by English electric company and then raised to 500KW at Burri power plant in Khartoum.

Due to the demand growth in 1925 a contract is signed with group of British companies for development of electricity through a period of thirty years. As a result steam generation was started in 1956 at Burri power station (30 megawatts) with 4steam turbine.

In 1960 the government established central electricity and Water Corporation unit which began to extend electricity and water production and its services extended in many cities in the country.

Gas turbine generation began in 1968 at Kilo (10) power station (15megawatts) erected by Fiat Gas Turbine Company.

1.2 Statistical data:

Installed capacity

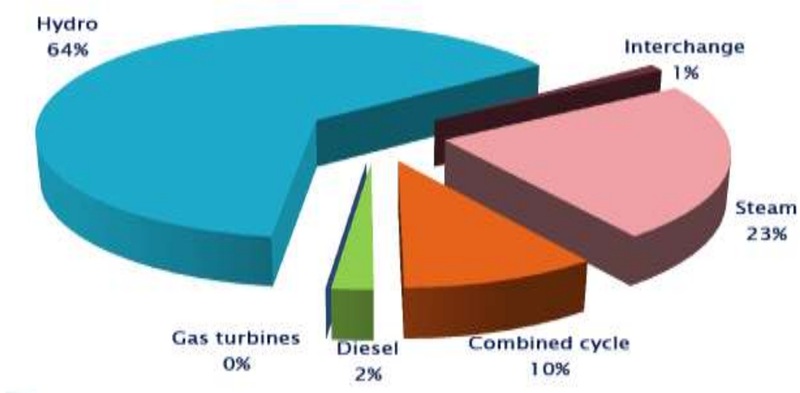

Figure (1.1) Installed capacity

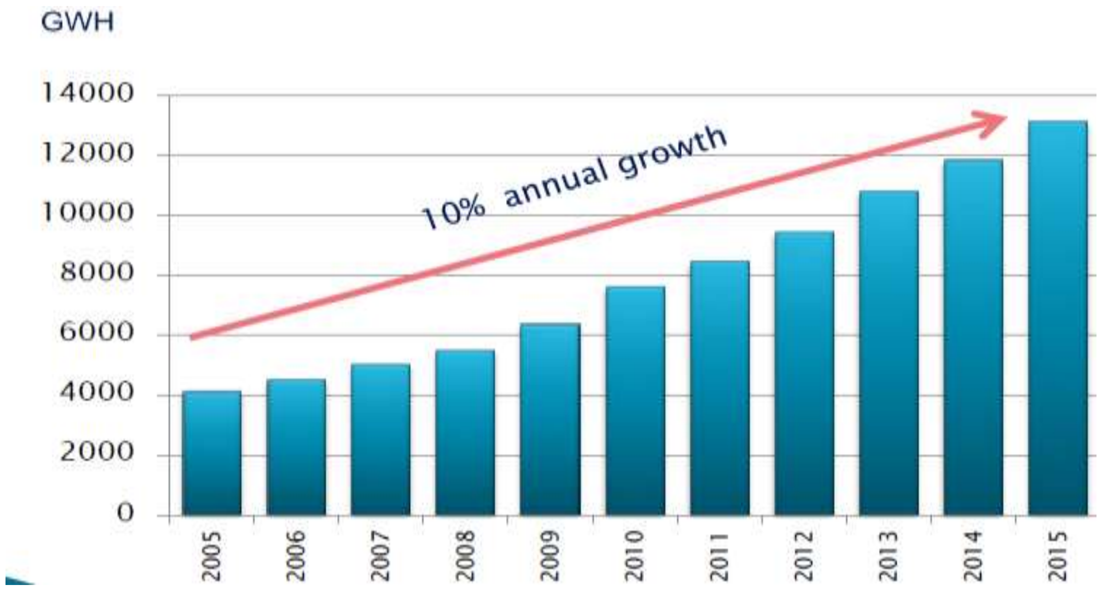

Figure (1.2) Electricity generation growth 2005 -2015

1.3 Thermal power generations:

There are 13 thermal power plants with different types of steam turbine generation; gas turbines combined cycle and diesel.

The total installed capacity (grid + off grid) is about 1650MW

1.4 Hydro power generation:

In1962 it was a first hydro power plant consists of two units to generate electricity from Sinnar Dam with capacity of 15 MW. Nowadays Sudan has five hydro power plants with a total capacity of 1593MW

Then Algirba power plant consists of three turbines and two units with a total capacity of 17.8MW (1964).

Rosaries power plant consists of seven units with total capacity of 280MW (1971).

Jabal Awlia power station consists of 80 units with total capacity of the plant is 30MW(2005).

Merowe dam power plant it is recent and bigger dams which erected in 2009, it is consists of (10) units with total capacity of 1250MW. Merowe Dam power evacuated via transmission of high voltage (500kv) for the first time in Sudan. So the total Hydro power generation now 1593MW

1.5 Thermal &Hydro generation planned projects:

Due to the electrical energy demand growth the ministry of water resources &electricity plans to increase the generation capacities with different sources as well as conventional and renewable energy.

1.5.1 Hydro Generation planned projects:

To implement the following planned hydro power projects:

No	Dam	Capacity (MW)
1	Up Atbara - Sitat	320
2	Kajbar	360
3	Dagash	312
4	Total	982

Figure (1.3) planned projects dames

No	Project	Capacity (MW)
1	Red Sea Combined cycle	1,500
2	Red sea fired coal –steam turbines	600
3	Garri3 simple cycle –phase1	510
4	Garri3 combined cycle phase2	240
	Total	2,850

Figure (1.4) planned projects thermal

1.5.2 Thermal ongoing projects:

Alfula thermal power project: consists of three units each with capacity 135 MW with total capacity of 405MW.

1.5.3 Renewable Energy:

Electrical power generation is an important factor for development of people. Electrical power from energy sources ease of use and represent the backbone for growth, development, economic, social progress and cultural also measured progress nation's average per person consumption of electrical energy. The electrical generation in Sudan beginning in 19centry, about 80 kilowatt hours a year per person compare to other nation that over 1,000 kilowatt hours. Electrical energy has an important role in moving the national economy, Due to the demand growth of electricity government contract signed with group of Chinese company for period of time to development of electricity

1.6 Garri power station: GT

Garri power station is located 70 km north of Khartoum. The station consists of eight GE frame 6 gas turbines. Four of them (located in Garri 1) are of type PG6581, which are equipped with heat recovery and connected to two steam turbines as combined cycles. Three of the other four turbines (located in Garri 2) are type PG6551, which were commissioned in 2003 and used to work as open cycles. In 2007, a fourth gas turbine, type PG6581 was added together with two steam turbines to Garri 2 block so as to become similar to Garri 1,Garri 4fired coal steam turbine is consists of two steam turbine and Garri 3 in ongoing project consists of two gas turbine equipped with heat recovery and connected with one steam turbine.

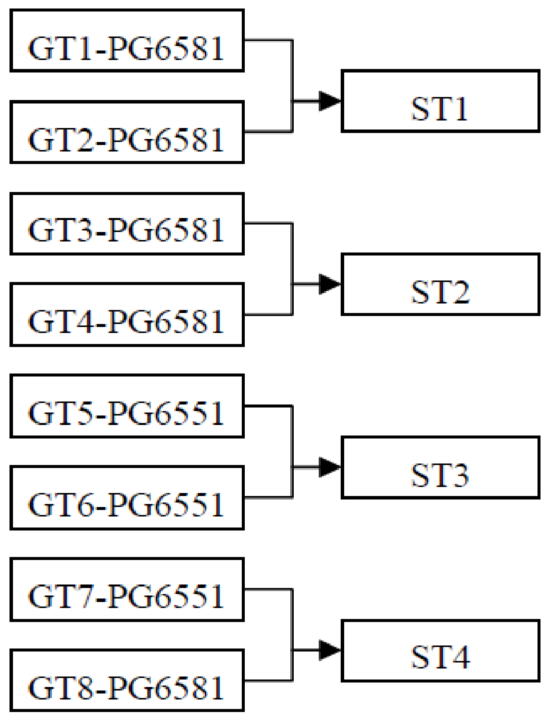

Figure (1.5): Schematic Layout of Garri (1&2) power plants.

The climate of the area is the semi desert climate influenced by the north-south movement of dry northerly winds and moist southerly winds that produce a wet summer and a dry winter. Figure 3.2 shows the year's average maximum and minimum daily temperature and rainfall .Due to the high ambient temperature, there is a drop of 14 MW in each turbine of type PG6581 and 12 MW in each turbine of type PG6551. The total loss of power is 92 MW in the eight gas turbines. If recovered, this will add needed power to the national grid, which suffers from power shortage. From Figure 3.2, there is a potential of 15 C for cooling the inlet air of the gas turbines at Garri power station.

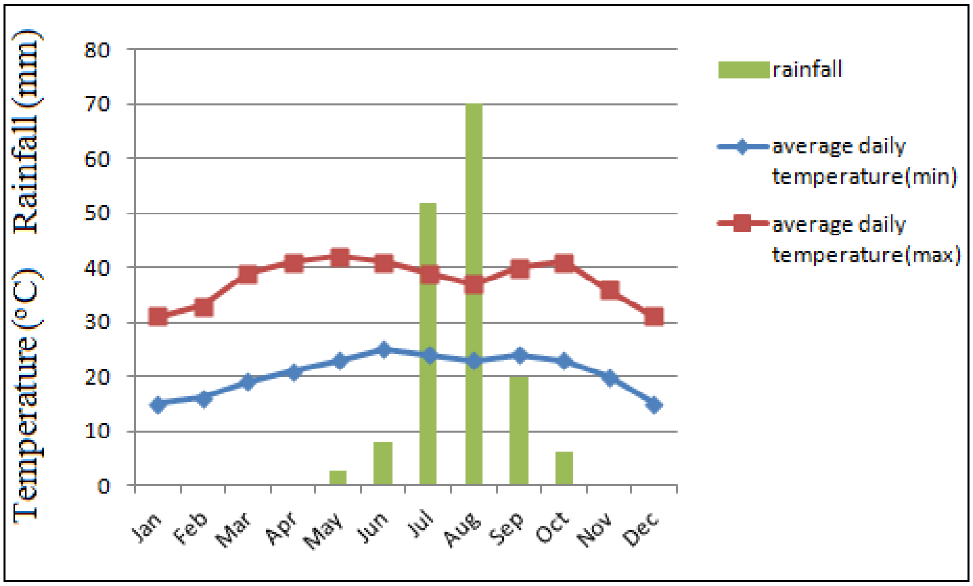

Figure (1.6) the years' maximum and minimum temperature and rain fall at Garri area.

1.7 Problem statement:

To meet the growing demand for electricity, power utilities have to continuously expand their generation capacities. Not only is the installation of new generation plants expensive, but finding energy sources for these new stations is also a problem. The installation of a new power station also raises more concerns about environmental pollution. The need for installing new power projects can be delayed by enhancing the production capacity of existing power stations, which is very much needed considering the tough challenges facing many countries nowadays. In Sudan, a considerable part of the thermal generation of the Sudan comes from the gas-turbines located at Garri Power Station. Due to highambient temperature in the area at summer average of 42°C the gas turbines operate at 75% of theirrated capacity at the standard ISO conditions as GE Product table , Garri Power Station contains eight General Electric frame 6 gas turbines, of which of type PG6581.

From the table find that gas turbine fram6 output (PG6581 (B)) is 42 MW under ISO condition but the actual power produced at Sudan not more than 30 MW at summer session. (Fig 1-1) show the effect of ambient temperature to the efficiency.

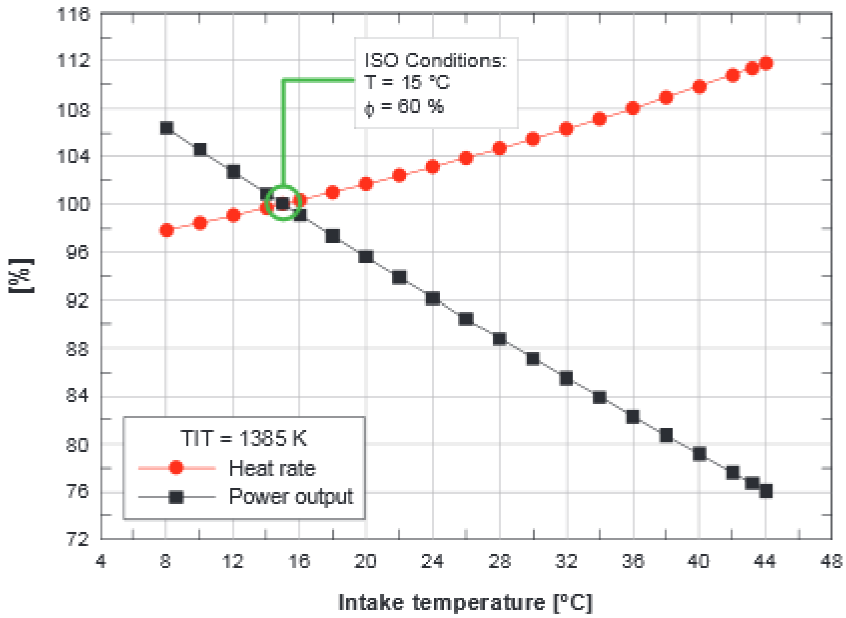

Fig (1.7) Effect of intake temperature on the gas turbine performance in comparison with ISO conditions

1.8 The objectives:

The main objectives of this study are:

- To Increase output power produced from gas turbine.
- To Increase the efficiency of the gas turbine.
- To Decrease fuel consumption and environmental impact.
- To Use economical system to get these objectives.

1.9 Methodology:

The study is using the existing Garri power plant as a case study and providing it by a design of an absorption cooling system to reduce the ambient temperature, parameter.

1.10 Expected results:

- Control the inlet air temperature all over the year at the bellow15 °C.
- Increase the output power.
- Increase the efficiency.

CHAPTER II

LITERATURE REVIEW

CHAPTER II

LITERATURE REVIEW

2.1 Basic Gas Turbine Operation

A schematic diagram of a simple cycle, single shaft gas turbine is shown in Figure 2.1 Air enters the axial flow compressor through point 1 which is at ambient conditions. Then it is compressed up to a higher pressure level. During this process there is no heat addition to the air, however the compression of air raises the temperature. So that the compressed air at the compressor discharge is at high pressure and temperature. The compressed air then enters the combustion chamber at point 2, where fuel is injected and combustion takes place. Either gaseous fuel (e.g. methane, natural gas, etc.) or liquid fuels (e.g. diesel, heavy fuel) can be used here. The combustion process occurs at constant pressure. The combustion chamber is design to provide mixing, burning and dilution effects to have proper combustion .The combustion product which is the flue gas leaves the combustion chamber at very high temperature.

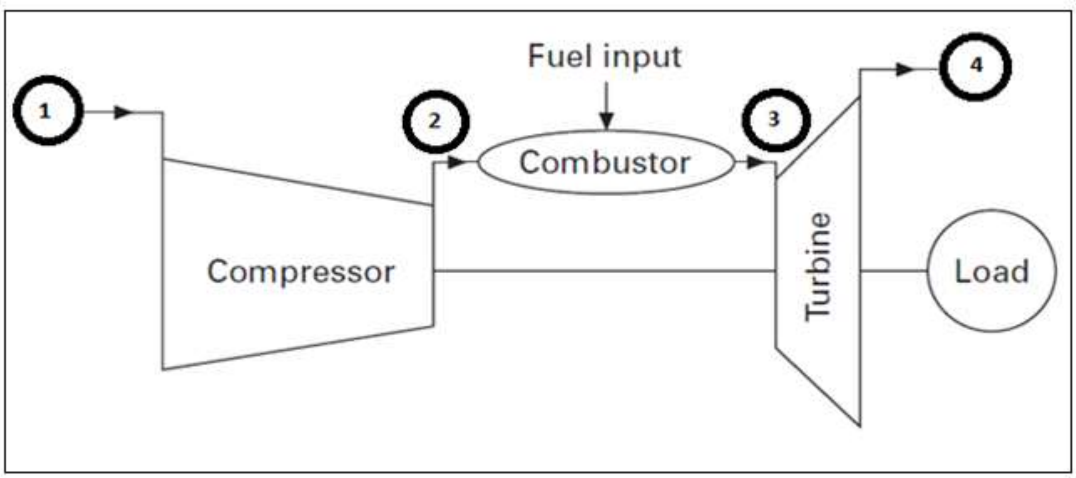

Figure 2.1 Schematic layout of a single-shaft gas turbine.

At point 3 the flue gas enters the turbine. In the turbine section, energy of the hot gas is converted into work. This conversion happens in two steps. In the first step, hot gases are expanding through the nozzle by converting the thermal energy to the kinetic energy. Then the turbine blades are converting this kinetic energy to mechanical work. The work delivered by the turbine blades is used to run the compressor and the remaining power is available for useful work at the output shaft of the turbine. As shown in the Figure 2.1 the output shaft can be coupled to an electrical generator to produce electricity. After the expansion through the turbine, exhaust gas leaves the system to the atmosphere at point 4. Still the exhaust gas is at a very high temperature and therefore has considerable amount of energy content in it. An energy recovery unit can be used to recover this energy for useful work. Heat recovery steam generator (HRSG) is a component which can be used to capture the energy of exhaust gas from the gas turbine. Recovered energy can be used to run a steam turbine and generate electricity, [1]

2.2 Brayton cycle: The Ideal Cycle for Gas-Turbine Engines

The Brayton cycle (or Joule cycle) was first proposed by George Brayton for use in the reciprocating oil-burning engine that he developed around 1870. Today, it is used for gas turbines only where both the compression and expansion processes take place in rotating machinery. Gas turbines usually operate on an open cycle, as shown in Figure 2.2. Fresh air at ambient conditions is drawn into the compressor, where its temperature and pressure are raised. The high pressure air proceeds into the combustion chamber, where the fuel is burned at constant pressure. The resulting high-temperature gases then enter the turbine, where they expand to the atmospheric pressure while producing power. The exhaust gases leaving the turbine are thrown out (not re circulated), causing the cycle to be classified as an open cycle. The open gas-turbine cycle described above can be modeled as a closed cycle, as shown in Figure 2.3, by utilizing the air-standard assumptions. Here the compression and expansion processes remain the same, but the combustion process is replaced by a constant-pressure heat-addition process from an external source, and the exhaust process is replaced by a constant pressure heat-rejection process to the ambient air. The ideal cycle that the working fluid undergoes in this closed loop is the Brayton cycle, which is made up of four internally reversible processes:

- ✓ 1-2 Isentropic compression (in a compressor)
- ✓ 2-3 Constant-pressure heat addition
- ✓ 3-4 Isentropic expansion (in a turbine)
- ✓ 4-1 Constant-pressure heat rejection

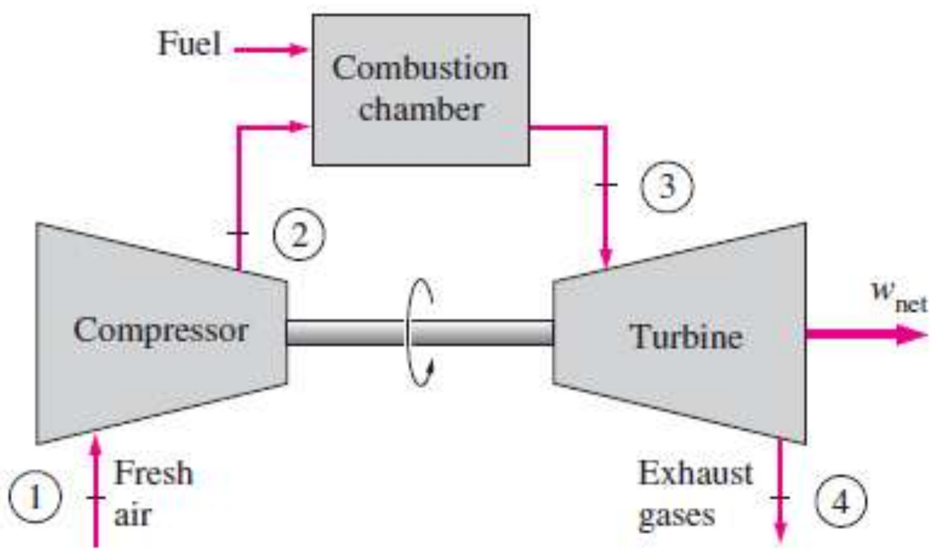

Figure 2.2 an open-cycle gas-turbine engine

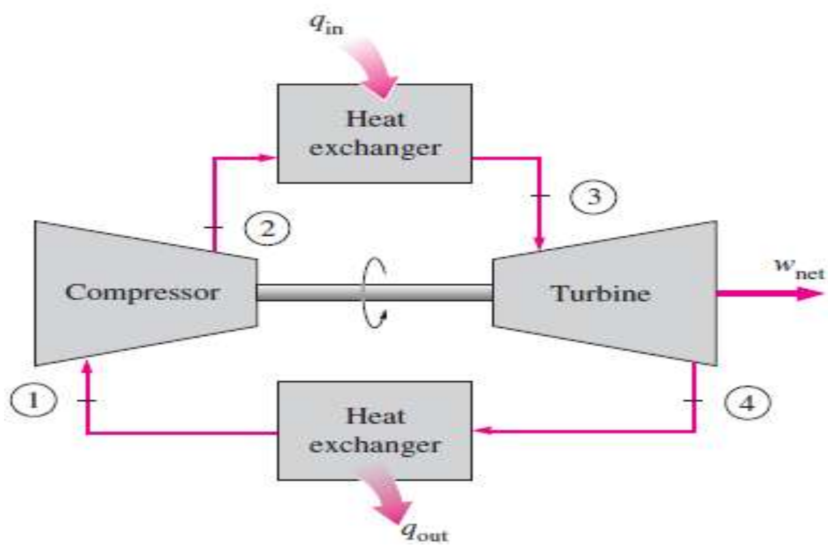

Figure 2.3 a closed-cycle gas-turbine engine.

The T-s and P-v diagrams of an ideal Brayton cycle are shown in Figure 2.4a Figure 2.4b

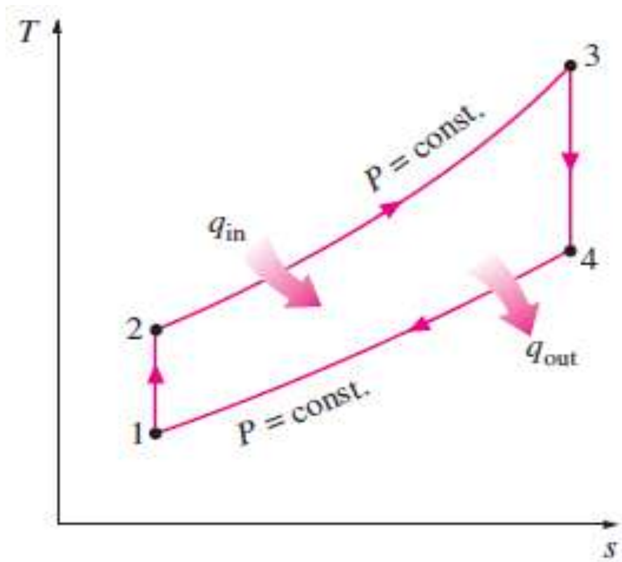

Figure 2.4 a T-s diagram

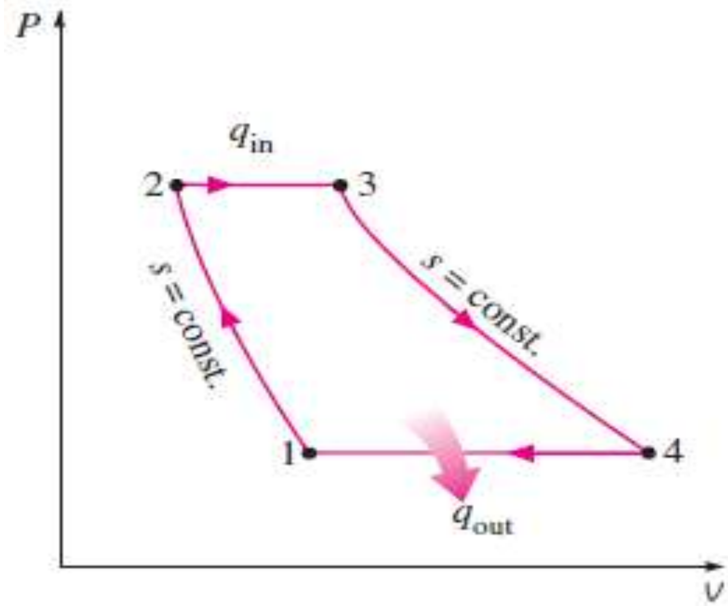

Figure 2.4b P-v diagram.

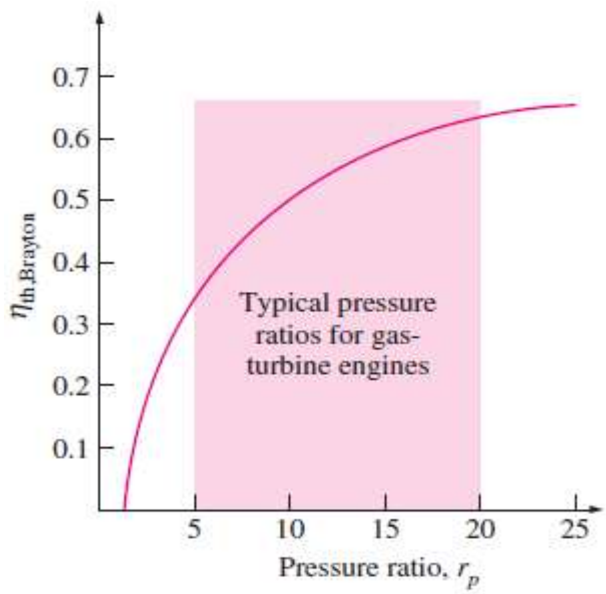

Figure 2.5 Thermal efficiency of the ideal Brayton cycle as a function of the pressure ratio.

In gas-turbine power plants, the ratio of the compressor work to the turbine work, called the back work ratio, is very high Figure 2.6 Usually more than one-half of the turbine work output is used to drive the compressor. The situation is even worse when the isentropic efficiencies of the compressor and the turbine are low. This is quite in contrast to steam power plants, where the back work ratio is only a few percent. This is not surprising, however, since a liquid is compressed in steam power plants instead of a gas, and the steady-flow work is proportional to the specific volume of the working fluid. A power plant with a high back work ratio requires a larger turbine to provide the additional power requirements of the compressor. Therefore, the turbines used in gas-turbine power plants are larger than those used in steam power plants of the same net power output.

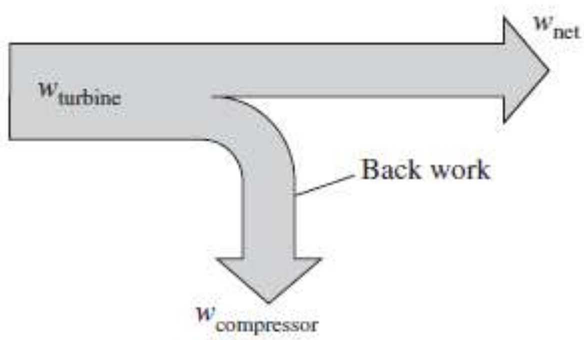

Figure 2.6 the fraction of the turbine work used to drive the compressor

2.3 Deviation of Actual Gas-Turbine Cycles from Idealized Ones

The actual gas-turbine cycle differs from the ideal Brayton cycle on several accounts. For one thing, some pressure drop during the heat-addition and heat rejection processes is inevitable. More importantly, the actual work input to the compressor is more, and the actual work output from the turbine is less because of irreversibility. The deviation of actual compressor and turbine behavior from the idealized isentropic behavior can be accurately accounted for by utilizing the isentropic efficiencies of the turbine and compressor as

$$\eta_c = \frac{w_s}{w_a} = \frac{(h_{2s} - h_1)}{(h_{2a} - h_1)} \qquad (2.1)$$

$\eta_c \equiv$ Isentropic Efficiency of the Comperssor .

and

$$\eta_T = \frac{w_a}{w_s} = \frac{(h_3 - h_{4a})}{(h_3 - h_{4s})} \qquad (2.2)$$

$\eta_T \equiv$ Isentropic Efficiency of the Turbine .

where states 2a and 4a are the actual exit states of the compressor and the turbine, respectively, and 2s and 4s are the corresponding states for the isentropic case, as illustrated in Figure 2.7.

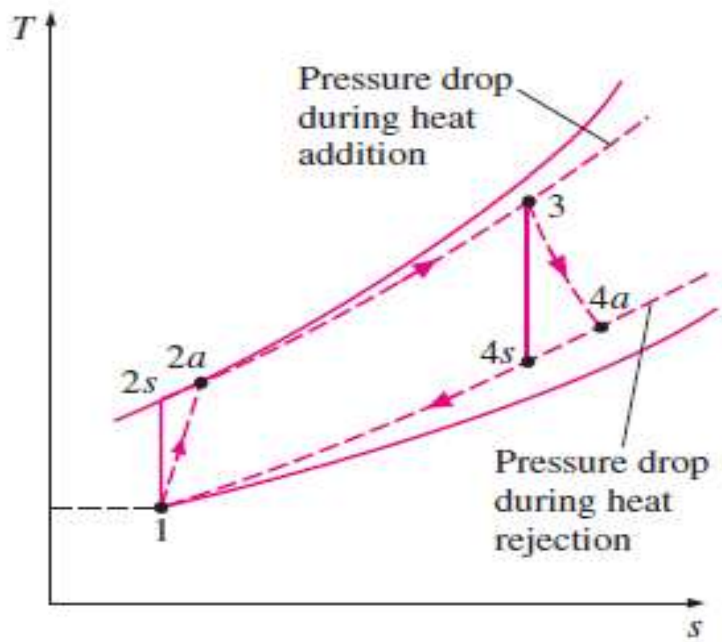

Figure 2.7 the irreversibility in the brayton cycle.

2.4 The Brayton Cycle with Regeneration

In gas-turbine engines, the temperature of the exhaust gas leaving the turbine is often considerably higher than the temperature of the air leaving the compressor. Therefore, the high-pressure air leaving the compressor can be heated by transferring heat to it from the hot exhaust gases in a counter-flow heat exchanger, which is also known as a regenerator or a recuperator. A sketch of the gas-turbine engine utilizing a regenerator and the T-s diagram of the new cycle are shown in Figure 2.8 and Figure 2.9, respectively.

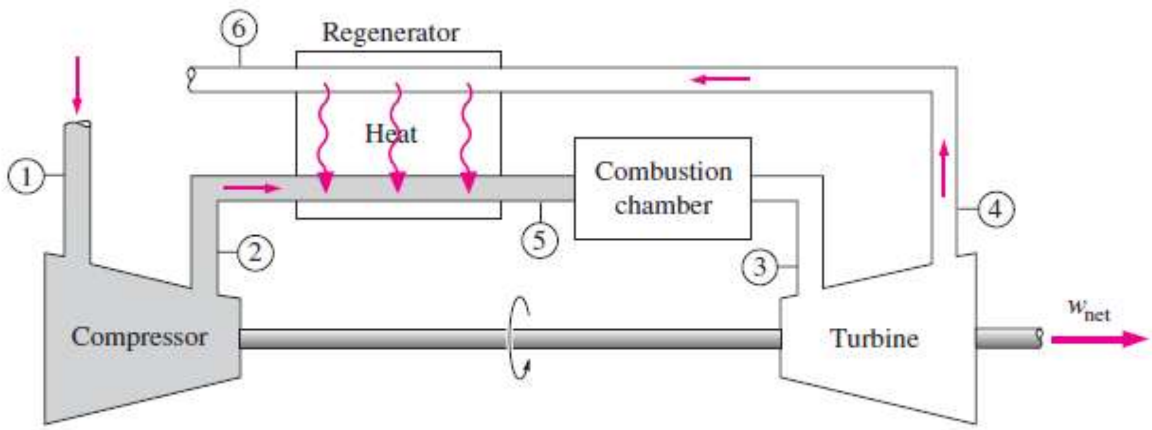

Figure 2.8 a gas-turbine engine with regenerator.

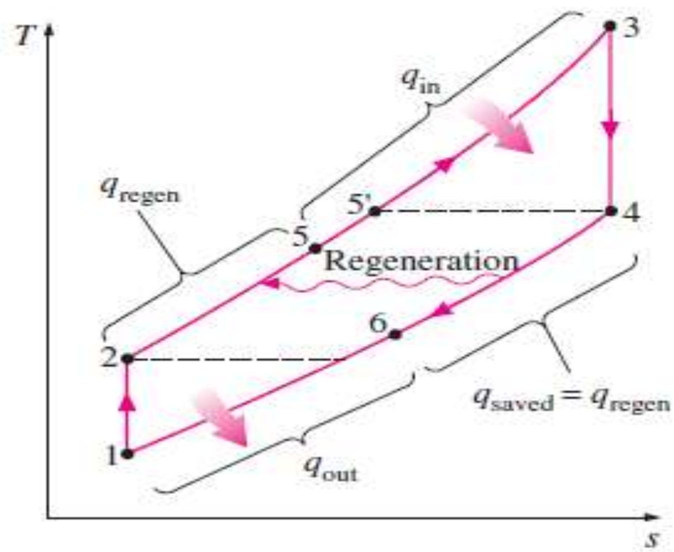

Figure 2.9 T-s diagram of a Brayton cycle with regeneration.

2.5 The Brayton Cycle with Intercooling, Reheating, and Regeneration

The net work of a gas-turbine cycle is the difference between the turbine work output and the compressor work input, and it can be increased by either decreasing the compressor work or increasing the turbine work, or both, the work required to compress a gas between two specified pressures can be decreased by carrying out the compression process in stages and cooling the gas in between Figure 2.10 that

is, using multistage compression with inter cooling. As the number of stages is increased, the compression process becomes nearly isothermal at the compressor inlet temperature, and the compression work decreases.

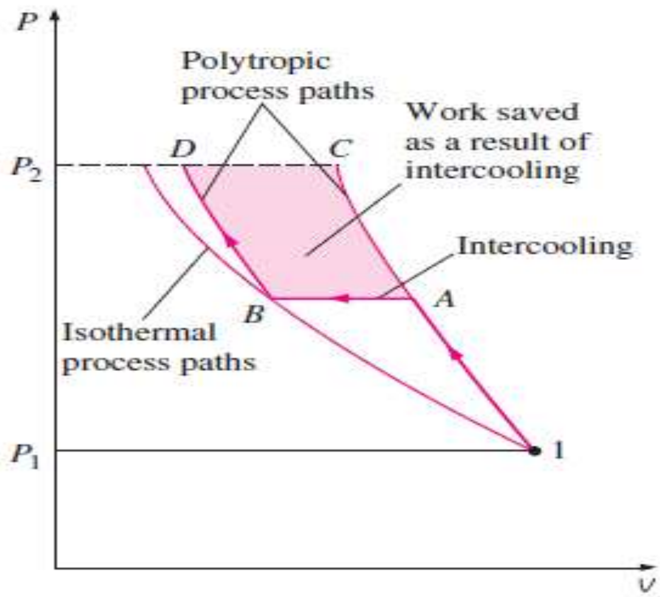

Figure 2.10 Comparison of work inputs to single-stage compressor and two-stage compressor with inter cooling

Likewise, the work output of a turbine operating between two pressure levels can be increased by expanding the gas in stages and reheating it in between—that is, utilizing multistage expansion with reheating. This is accomplished without raising the maximum temperature in the cycle. As the number of stages is increased, the expansion process becomes nearly isothermal. The foregoing argument is based on a simple principle: The steady-flow compression or expansion work is proportional to the specific volume of the

fluid. Therefore, the specific volume of the working fluid should be as low as possible during a compression process and as high as
possible during an expansion process. This is precisely what intercooling and reheating accomplish.

A schematic of the physical arrangement and the T-s diagram of an ideal two-stage gas-turbine cycle with intercooling, reheating, and regeneration are shown in Figure 2.11 and Figure 2.12, The gas enters the first stage of the compressor at state 1, is compressed isentropically to an intermediate pressure P_2, is cooled at constant pressure to state 3 ($T_3 = T_1$), and is compressed in the second stage isentropically to the final pressure P_4. At state 4 the gas enters the regenerator, where it is heated to T_5 at constant pressure. In an ideal regenerator, the gas leaves the regenerator at the temperature of the turbine exhaust, that is, $T_5 = T_9$. The primary heat addition (or combustion) process takes

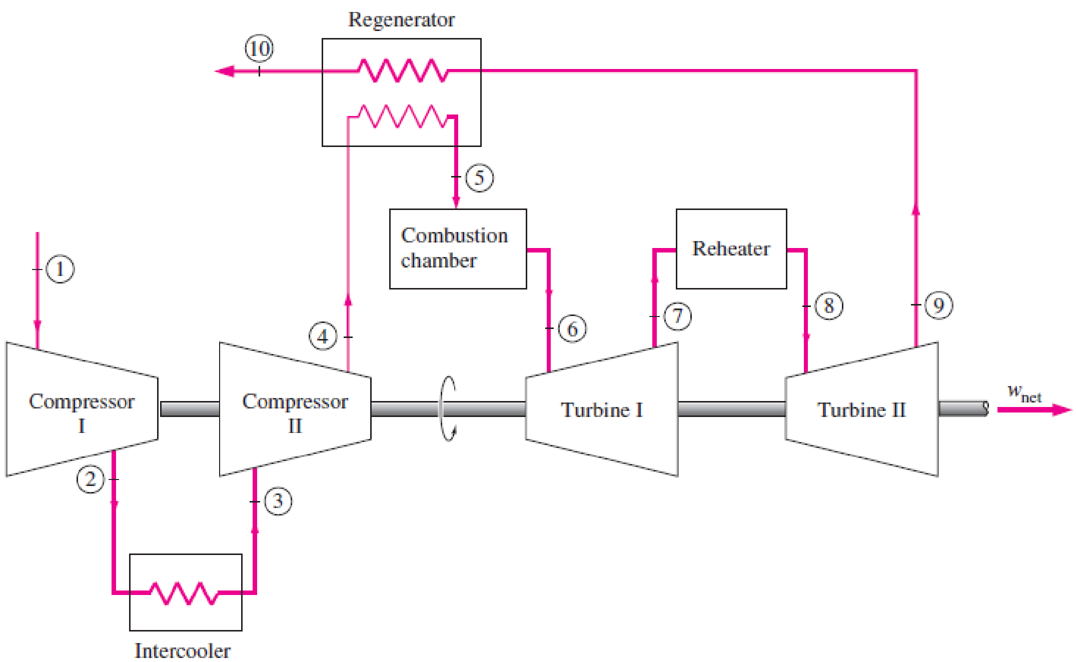

Figure 2.11 a gas-turbine engine with two-stage compression with inter cooling, two-stage expansion with reheating, and regeneration.

place between states 5 and 6. The gas enters the first stage of the turbine at state 6 and expands isentropically to state 7, where it enters the re heater. It is reheated at constant pressure to state 8 ($T_8 = T_6$), where it enters the second stage of the turbine. The gas exits the turbine at state 9 and enters the regenerator, where it is cooled to state 10 at constant pressure. The cycle is completed by cooling the gas to the initial state (or purging the exhaust gases).

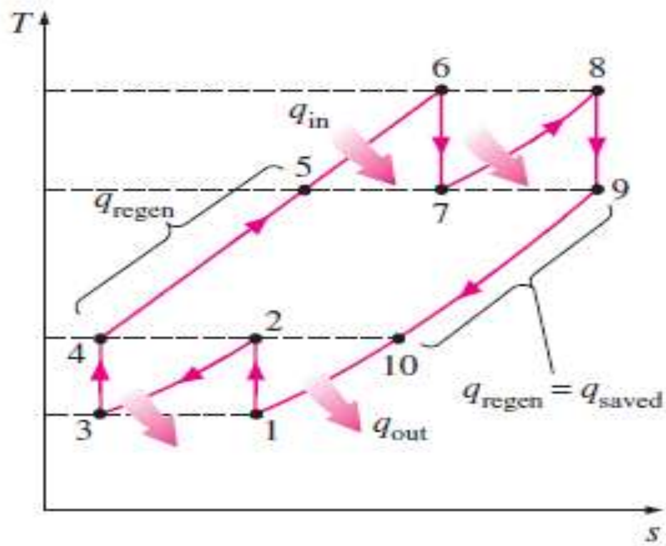

Figure 2.12 T-s diagram of an ideal gas-turbine cycle with inter cooling, reheating, and regeneration [2].

2.6 heat exchanger design:-

Heat lost by air Given by

$$Q_a = m_a * cp_a * (T_{ai} - T_{ao}). \qquad (2.3)$$

m_a =mass flow rate of air

cp_a =specific air ratio

the heat exchanger area need to occur heat transfer given by

$$Q = U * A_o * LMTD * F \qquad (2.4)$$

U =over all heat transfer

A_o=heat exchanger area need

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln(\frac{\Delta T_1}{\Delta T_2})} \quad (2.5)$$

$$P = \frac{t_2 - t_1}{T_1 - t_1} \quad (2.6)$$

$$R = \frac{T_1 - T_2}{t_2 - t_1} \quad (2.7)$$

Flowrate thro heat exchanger can be determine by

$$\dot{m} = \rho_c * A * V \quad (2.8)$$

The number of pipes at single row

$$\text{Number of rows} = \frac{L}{d_{\text{(both sides)}} + \text{space}_{\text{(both sides)}}} \quad (2.9)$$

Reynolds number given by

$$Re_{cw} = \frac{V_{cw} * d_i}{\nu_{cw}} \quad (2.10)$$

$$Nu_{cw} = 0.023Re^{0.8} * pr^{0.3} \quad (2.11)$$

$$Nu_{cw} = \frac{h_{cw} * d_i}{K_{cw}} \quad (2.12)$$

$$A_{min} = (S_T - 2 * D) * L * N_{row} \quad (2.13)$$

S_T transverse pitch

A_{min} =minimum free area available

$$U_{max} = \frac{S_T/_2}{\sqrt{(S_T{}^2 + (S_T/_2)^2)} - d_o} * \text{free flow velocity} \quad (2.14)$$

U_{max}=maximum velocity

S_T=transveres pitch

$$R_T = \frac{1}{h_i * a_i} + R_{wall} + \frac{1}{h_o * a_o} \quad (2.15)$$

$$A_o = n_{tubes} * \pi * d_o * l \quad (2.16)$$

2.7 Hot Water Solar collectors:

Useful gain can given by

$$q_u = F_r * (w - D)L\left[S - \frac{U1}{C}(T_r - T_a)\right] \tag{2.17}$$

q_u = Useful gain (kw)

$F_{r=\text{Correction factor}}$

L = length of collecter (m)

W = width of collecter (m)

D = diameter of cover (m)

U_l = Loss coefficient from receiver or tube (kw/m^2)

S = Absorber solar energy (kw/m^2)

T_r = Recive surface temperature(°C)

T_a = Ambient temperature (°C)

C=concentration ratio

$$S = \text{Absorber solar energy}\left(\frac{kw}{m^2}\right)$$

$$S = IbRb * \rho\tau(\tau\alpha) + IbRb(\tau\alpha)b\frac{D}{W-D} \tag{2.18}$$

$I_bR_b = Average\ solar\ radiation\ (\frac{KJ}{m^2 day})$

$$U_l = \left[\frac{1}{h_{wind}} + \frac{1}{h_r}\right]^{-1} \tag{2.19}$$

$$h_{wind} = 5.7 + 3.8V\left(\frac{w}{m^2}°C\right) \tag{2.20}$$

$$C = \frac{W - D}{\pi D} \tag{2.21}$$

$$h_r = 4\sigma\varepsilon T^{-3}\left(\frac{w}{m^2}°C\right) \tag{2.22}$$

$$T^- = \frac{Tr+Ta}{2} \tag{2.23}$$

$$hr = 4\sigma\epsilon T^{-3} \tag{2.24}$$

2.8 previous studies:

GAS turbines are used for power electric generation, operating airplanes and for several industrial applications ,The gas turbine engine consist of a compressor to raise combustion air pressure, a combustion chamber where the fuel/air mixing is burned, and a turbine that through expansion extracts energy from the combustion gases ,These cycles operates according to the open Brayton thermodynamic cycle and present low thermal efficiency and are referred as combustion turbines. Usually, the rated capacities of combustion turbines are based on standard ambient air, and zero inlet and exhaust pressure drops, as specified by the International Organization for Standardization (ISO) Therefore, the air inlet conditions are: air temperature 15 °C, relative humidity 60%, absolute pressure 101.325 kPa at sea level. Combustion turbines are constant-volume engines and their power output is directly proportional and limited by the air mass flow rate entering the engine. Combustion turbines are constant-volume engines and their power output is directly proportional and limited by the air mass flow rate entering the engine. As the compressor has a fixed capacity for a given rotational speed and a volumetric flow rate of air, their volumetric capacity remains constant and the mass flow rate of air it enter into the gas turbine varies with ambient air temperature and relative humidity, The performance of a gas turbine power plant is commonly presented in function of power output and specific fuel consumption, and it is sensible to the ambient conditions.

Thermodynamic analyses from literature show that thermal efficiency and specific output decrease with the increase of humidity and ambient temperature, but the temperature ambient is the variable that has the greatest effect on gas turbine performance.

The temperature ambient rise results in decrease in air density, and consequently, in the reduction of the mass flow rate. Thereby, less air passes through the turbine and the power output is reduced, at a given turbine entry temperature. Moreover, the compression work increase due the augmentation of the volume occupied by the air. According to, the net power output produced by gas turbine is directly proportional to the air mass flow, it that decreases when ambient temperature 4 increases. The work of Ibrahim, shown that an increment of 1 °C in the compressor air inlet temperature decreases the gas turbine power output by 1 %.

Gas turbines have been used for power generation in several places in the world, and each region have different climatic conditions. Furthermore, the periods of the peak electricity demand occur during the summer, when the ambient temperature is high. For example, in Arabian Gulf region the average ambient temperature presents a variation by more than 30 °C from summer to winter and this factor generate a large drop in power output during the summer.

Due to these severe ambient conditions, the turbine inlet air cooling is one of many available technologies to improve the performance of the gas turbine power plants by cooling the air at the compressor entry. Thus, the interest in the intake air cooling techniques for gas turbines has augmented in the last years, due the increasing requirement for power to a low specific investment cost.

Two different methods are frequently employed to obtain turbine inlet air cooling: the evaporative cooling and inlet chilling systems. Several works has been studied these cooling technologies as below detailed. Presented a comparison between two usual inlet air cooling methods, evaporative cooler and mechanical chiller, and one new technique that uses turbo-expanders to improve performance of a gas turbine located at the Khangiran refinery in Iran. Their results showed that turbo-expander method has the better cost benefit, because it offers the greatest increase in net power and a lower payback period. Performed a review of inlet air cooling methods that can be used for enhancing the power production of the Saudi Electric Company's gas turbine during summer peak hours. They concluded that the evaporative cooling system and the high-pressure fogging require a large amount of water this factor limits its use in the desert climate, the absorption chiller is an expensive system and its cost of investment isn't justifiable if it used only to improve the power output in the hour's peak. Mechanical refrigeration requires large electric power demand during the peak times, and thermal energy storage methods necessitate low electric power, but these Systems need a very large storage volume. The favored alternative choose for these authors is refrigeration cooling with chilled water or ice thermal storage, the last option can produce lower inlet air temperature and requires a smaller Storage volume. Presented a thermodynamic assessment of some inlet 5 air cooling system for gas turbine power plants in two different regions of Oman, and the considered techniques are evaporative cooling, fogging cooling, absorption cooling using both LiBr–H2O and aqua-ammonia, and vapor-compression Cooling systems. These different cooling techniques were compared with relation their electrical energy production augmentation, as well as their impact on increasing the on peak capacity of the considered gas turbine. modeled and evaluated an evaporative cooling system installed in gas turbines of the combined cycle power plant in Fars (Iran). Their results showed that the power Output of a gas turbine, at ambient temperature of 38 °C and relative humidity of 8%, it presents an increment by 11 MW for temperature drop of the intake air of about 19 °C. At this context, the present work focuses on the comparison of two inlet air cooling technologies. Evaporative cooling and absorption chiller are tested at different ambient temperature and humidity conditions, and the gas turbine power output and thermal efficiency are compared, Ana Paula, Claudia Edson [2].

Gas turbine plants are used for electricity production in many countries around the world because of their low capital cost, short synchronization time of 30 minutes (time required for gas turbine to reach the base load from zero speed), stability with regard to the electricity grid and availability in many countries, including in this case Egypt. Total electricity generated by gas turbine plants in Egypt is about 7001 MW ,from deferent gas turbine models and capacities varying from 25 MW to 260 MW. On hot days in summer, the ambient temperature reaches 40°C and the gas turbine power output decreases by18% from rated capacity, leading to total power lost from gas turbine plants of about 1440 MW while the electric load increases to maximum due to air conditioning and ventilation. It is therefore necessary to enhance the achievable gas turbine power output by cooling the inlet air. There are two main inlet cooling types: (i) evaporative or fogging cooling, and (ii) chiller cooling— electrical or absorption.

Evaporative systems cool the inlet air by evaporating water into the air stream. The water evaporation causes the air temperature to drop. Low humidity climates are suitable for use of this cooling technology. Two considerations must be taken into account. Firstly, the maximum relative humidity that it is possible to reach with an evaporative system is just over

90%. Secondly, the deference between wet and dry bulb temperatures in the outer section of the evaporative system is recommended not to be under 1°C. Two 6 working fluids are used in the chiller system; the first one is the refrigerant in the refrigeration machine which consists of a compressor, evaporator, condenser and expansion device. The refrigerant is used in this cycle to cool secondary fluid, usually chilled water, which is pumped through an air-water heat exchanger located at the gas turbine inlet to cool air coming into the compressor.

The chiller system has the advantage of being able to reduce the air temperature to 5°C, but it involves very high capital cost. The first application of combustion turbine inlet air cooling (CTIAC) was a direct air conditioning system for a plant in Battle Creek, Michigan (USA)in 1987–88, the second was an o-peak ice harvester system in Lincoln, documented the performance of a 36MW CT plant in which a chilled water-based storage refrigeration system was tasked with cooling inlet air. The cooling system was able to reduce the air temperature from an ambient 35°C to 7°C, thus enhancing plant performance by 10%. Examined the potential use of employing CTIAC refrigeration systems in the United Arab Emirates Nebraska (USA) in 1992 Ali, Abdalla [3]. They used wet-bulb and dry bulb weather data to determine characteristic design conditions of three Emirates: Al-Ain, inland arid, very hot and relatively dry; Abu Dhabi, coastal Arabian Gulf, very hot and humid; and Fujairah, coastal Oman Gulf, hot and very humid. For given inlet air temperatures, they determined annual gross energy Increase, average heat-rate reduction, cooling load requirement and net power increase. For viability, they recommended an inlet air temperature of 15°C – 25°C. They recommended that evaporative cooling be used where a peak-power increase between 8% and 15% is required at high temperatures, and refrigeration cooling where a sustainable increase of 10– 25% is required Depending on the weather. An exergy,economic and environment analysis was done M. A. Ehyaei, S. Hakimzadeh, N. Enadi, P. Ahmadi [4] for a 4900 kW absorption chiller integrated with a 159 MW gas turbine unit located at Bushr-Iran. The analysis shows that the gas turbine’s power increases from 137 MW to 153 MW during the hottest month (August) when the inlet air was cooled from 37°C to 15°C. Moreover, efficiency rose from 33.4% to 43.2% 7 Seyed ,Mahbod [5] discussed the following four cooling methods:

- Prevalent evaporative cooling systems: In this method, in the entrance of filter chamber, a hive-shaped humid field is installed that cools the inlet air by circulating water and vaporizing it in the field of action. Generally such system is effective to go down the compressor inlet air temperature to the ideal conditions of the environment (85% to 90%).
- Fog pumping systems: The organic water is sprayed directly into inlet air current and water drop vaporization in the inlet air causes cooling it. This sort of system regulates compressor's air temperature to the degree, which is higher partially than wet temperature of the environment.
- Mechanical chilling: To cool the inlet air by series of fan-tubes in the entrance filter, a chilling system used. Generally such system guaranties for certain dry inlet
- Temperature and based on the maximum conditions of environmental design. These kinds of systems may have a secondary circulating cycle of Glycol-water in the fan-tubes and also the filter chamber.
- Thermal storage: For thermal storage, the chilling system is applied. However, a great amount of glycol-water the inlet air circulates into the fan-tubes or pipes in order to decrease the internal power consumption of chilling system during climate load periods.

Generally, for co generative applications, the gas turbine is designed to operate in standard conditions, established by the International Standards Organization and defined as ISO conditions: 15°C, 1.013 bars and 60 % humidity. During summer season air temperature rises and its density decreases, leading to a decrease in the intake air mass flow; consequently decreases power output because it is proportional to the intake air mass flow rate. Without taking supplementary measures, both gas turbine output power and efficiency drop.

With the operation during summer season, the increase of ambient temperature leads to a decrease in gas turbine plants power output with25-35%, also leading to an average increase of the consumption of fuel 6%. The effect of intake air temperature over the performances differs from one gas turbine to another, but, generally, aero derivative gas turbines are more sensitive to this phenomenon than the industrial gas turbines Abam F., Ugot I., Igbong D. [6] During summer season, when the days are long and hot, the power requirements increase for the residential spaces ventilation, offices, store rooms, etc. Additional energy consumption can be ensured by starting other backup groups, or compensating the loss of power through various other methods. The usual compensation methods of power loss are compressor inlet air cooling (pre-cooling) 8, For industrial gas turbine manufacturers searching on for new materials that meet the requirements imposed by the higher strains of the gas turbines and also the development of new technologies, including technological transfer from aviation domain to power generation domain. Thus, in aviation, afterburning is used in order to increase traction of supersonic engines. The introduction of afterburning into co generative applications leads to an increase in flexibility and global efficiency of the cogeneration group.

Cogeneration group's research focused specifically on increase in burning temperature; increase in compression ratio; improving the methods of design, cooling and burning technologies, and also advanced materials; technological transfer from the aviation domain in the industrial gas turbine domain and conversion of aviation gas turbine.

Ambient parameters humidity, pressure, temperature can vary significantly depending on geographic location and season, affecting air density and implicitly the gas turbine co generative group performances. In the past, the effect of air humidity was neglected but the increase in gas turbine co generative group's power and the introduction of water/steam in the combustion chamber made this effect to be reconsidered. Thus, some authors consider that air relative humidity even at temperatures higher than 10 °C has a neglectable influence over the gas turbine output power as the other performance parameters. This leads to the fact that in some calculus (especially when the results are presented in correlation to ISO conditions) the variations in atmospheric humidity and pressure to be neglected. Others consider that due to the fact that water content modifies thermodynamic properties of inlet air (density, specific heat), at certain gas turbines (depending on specific processes) the performances may increase when humidity rises and in the case of some gas turbines the performances may decrease in the same conditions Kurz R. [7]. However, the increase in relative humidity leads to a significant reduction of NOx emissions.

Ambient pressure is defined by the conditions from plant location, altitude modification leading to air density modification and implicitly to power output variation. Thus, 3-4% losses occur for each 304.8 m (1000 ft) rise in altitude. Pavri R., Moore G. [8].

plant.

The objective of his paper is to present a method to evaluate the characteristics and performance of a single stage lithium bromide (LiBr)–water absorption machine. The necessary heat and mass transfer equations and appropriate equations describing the properties of the working fluids are specified. These equations are employed in a computer program, and a sensitivity analysis is performed. The difference between the absorber LiBr inlet and outlet percentage ratio, the coefficient of performance of the unit in relation to the generator temperature, the efficiency of the unit in relation to the solution heat exchanger area and the solution strength effectiveness in relation to the absorber solution outlet temperature are examined. Information on designing the heat exchangers of the LiBr–water absorption unit are also presented. Single pass, vertical tube heat exchangers have been used for the absorber and for the evaporator. The solution heat exchanger was designed as a single pass annular heat exchanger. The condenser and the generator were designed using horizontal tube heat exchangers. The calculated theoretical values are compared to experimental results derived for a small unit with a nominal capacity of 1 kW. Finally, a cost analysis for a domestic size absorber cooler is presented. [9]

Designed chiller system to improve Garri combined cycle power plant by air intake cooling using compression refrigeration system but this system is very expensive and high operating and maintenance cost. [11]

CHAPTER III

PRACTICAL PART

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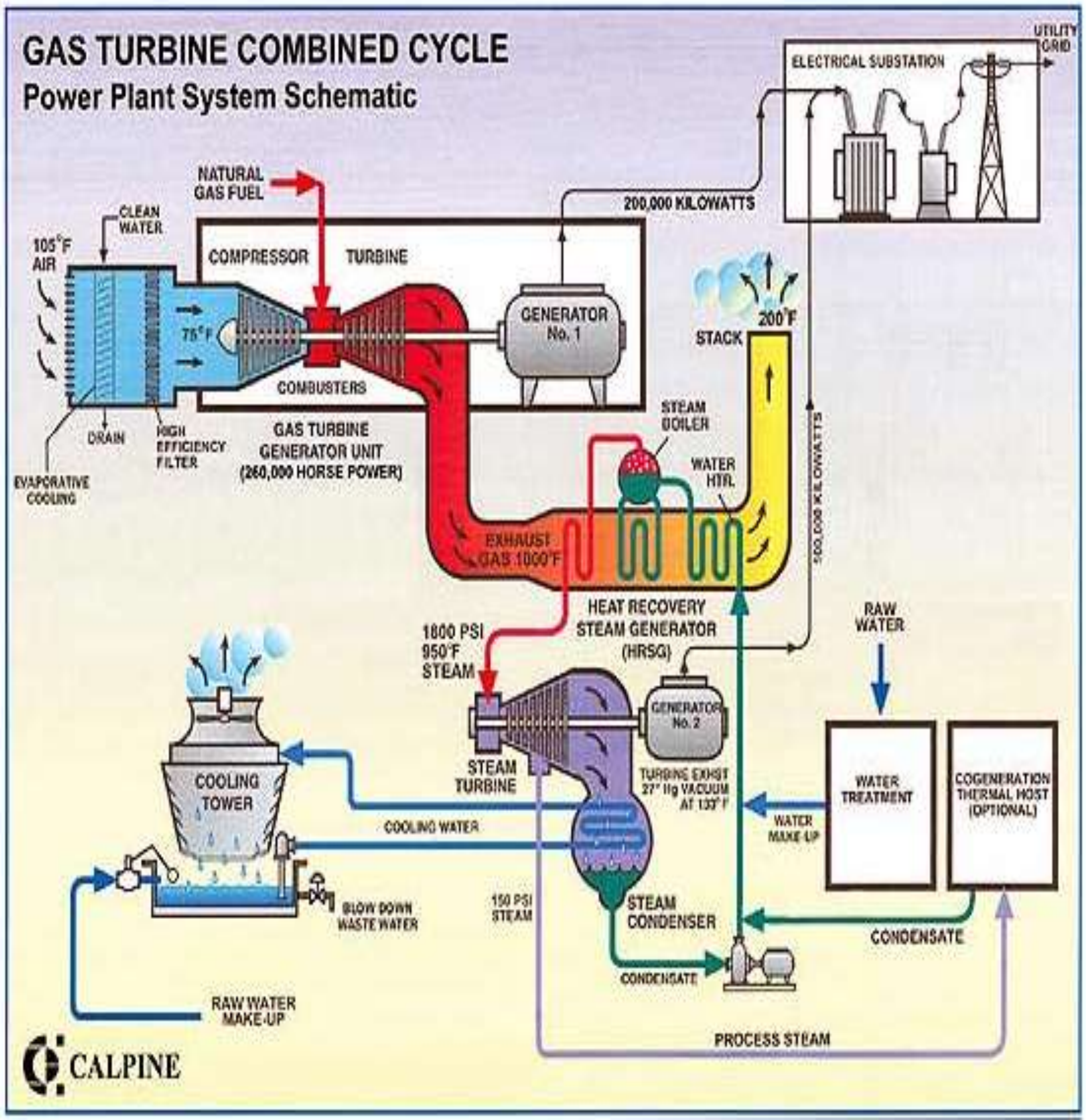
PRACTICAL PART

3.1 Overview:

The Garri complex power station is located about 70 km north of Khartoum city, the capital of Sudan. The contract titled Sudan Garri power Plant 1 was generally executed by Harbin Power Engineering Co .Ltd (HPE). The supervisor committee (consultant) was Lahmeyer International Germany. The Garri complex power station belongs to National Electrical Corporation (NEC) of Sudan. The complex consist of four plants, three existing (plant 1, plant2and plant 4), one is under construction (plant 3) .

Plant one consist of eleven units comprising of four sets of combined cycle units, including eight gas turbines with type of PG6581B and rated capacity of 42MW, four units of steam turbine with rated capacity of 36MW. Total installed capacity is 480 MW. The Heat Recovery Steam Generator (HRSG) is made by Harbin Boiler Works (China). The steam turbine type L36-6.70 is also the product of Nanjing Turbine & Electrical Machinery Group Co .Ltd. Each has a capacity of 36MW equipped with brushless excitation and digital excitation regulator SVR-2000A the water used for all plant comes from NILE River, where a water pre-treatment station is installed to purify the dirty water by chemical dosing, and then water is delivered to the Site through a common 15 km pipeline. The waste water after treatment through a new waste water treatment plant is going back to NILE River with an acceptable quality. For Light Diesel Oil (LDO) tanks have been installed and are receiving LDO by existing pipes from Khartoum Refinery Company (KRC) or unloading station. There is a big electrical substation with a voltage degree of 220kV, and 6 bays as well as one coupler are ready to send electricity to grid. There are two control centers, one located in substation where communication is available with Local Dispatching Center Garri Power Plant 37 (LDC) through Supervisory Control And Data Acquisition (SCADA) system and local control & protection systems are also put in. Another control room called Central Control Room (CCR) is on the 3rd floor of steam turbine hall. The general control system Distributed Control System (DCS) is Freelance 2000 produced by ABB Company, it has an ability to communicate with other controllers, such as Programmable Logic Controller (PLC) servicing for River-side and water demineralization Station, as well as controller (MARK VI). So all the running parameters including Gas Turbine, Steam Turbine, LDO system, Demine -water plant & Rive-side can be shown on the 10 sets of Operation Station (OS). For electrical arrangement, the power generated by gas turbine steps up through main transformer to the 220kV grid, same time steps down through a unit auxiliary transformer to 6.3kv medium voltage for house load system. The power generated by steam turbine is directly stepping up through main transformer to 220kVgrid. An emergency diesel generator is also available here in this plant for any emergency case in order to ensure the essential load. A newly -installed demin -water station is controlled by PLC Siemens Simapic S7-300 to generate enough capacity of demineralised water for HRSG, steam turbine and other customers. Garri Power Plant Other main ancillary systems consist of air compressor system, firefighting system, potable water generation plant, waste water treatment plant, heating ventilation and air conditioning (HVAC) system, DC system, Uninterruptible Power Supplies system (UPS), etc.[12]



Figures (3.1) Garri combined cycle power station1&2:

3.2 Parameters of Main Equipments:

Gas turbine generator unit The gas turbine -generator unit was manufactured by Nanjin Turbine Group Company limited. The power output is 32551KW under the following design conditions: Ambient temperature: 40º C

Atmosphere pressure: 0.966 bar

Ambient humidity: 38%

Inlet air pressure drop: 100 mmH2O

Exhaust pressure drop (under combined cycle): 350 mmH2O

Fuel: Distil (LDO)

Power factor: 0.80

Rated frequency: 50Hz

Heat recovery steam generator

The HRSG was supplied by Harbin marine boiler & turbine research institute. Chapter Four Garri Power Plant 39 It has two steam pressure levels one of which is used for deaerator and it has No duct firing. The HRSG output parameters are:

Maximum continues output: 63.78 t/h

Output steam pressure: 6.9 MPa(g)

Output steam temperature: 468 (+5-/l0)º C

Exhaust gas temperature: 154

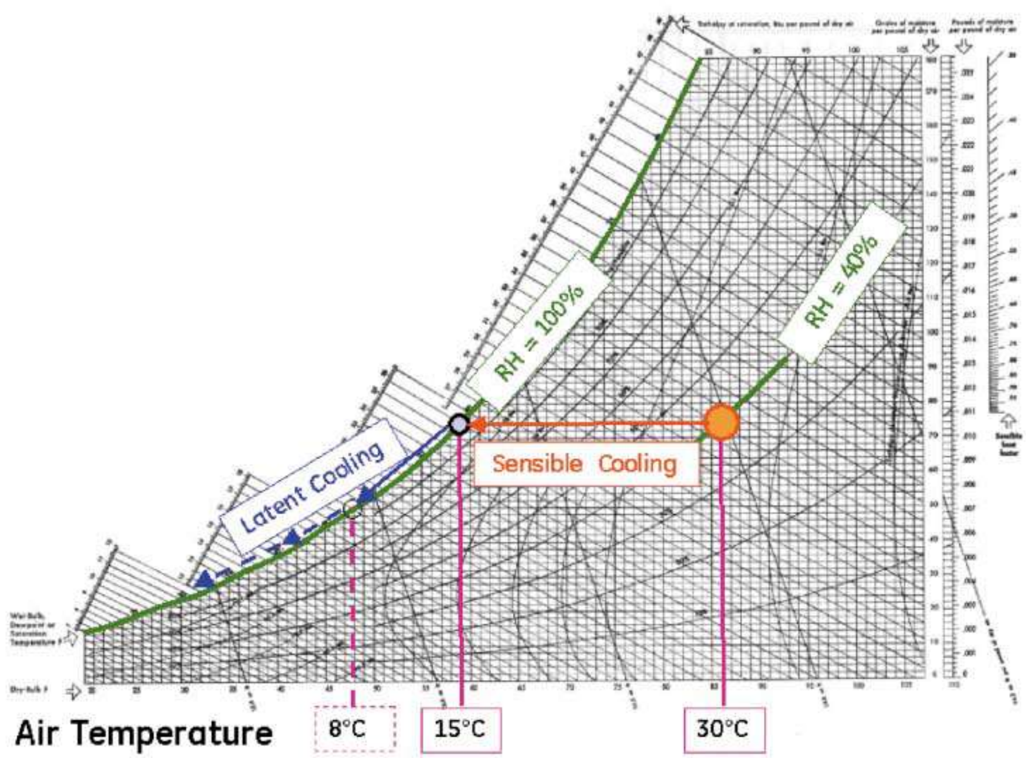


Figure (3.2) sensible and latent heat removed from the air psychometric chart.

The coolant is usually a mixture of water and glycol with the proportions determined to avoid icing. The coolant supplied to the heat exchangers can be supplied by several methods, including mechanically actuated refrigeration cycles, absorption chillers, vapor compression cycle, and ice storage systems. In the research expectation, the cooling would have an effect of up to a 20% increase in power and a reduction in heat rate of as much as 6.5% even considering the additional 1 inch of H2O pressure loss introduced by the coils and the mist eliminator. The actual power increase may be limited so as not to exceed the driver and driven equipment ratings. The effect of 1 inch of H2O permanent inlet pressure loss has to be taken into account when the unit operates with the chiller off.

The chilling coils are normally placed downstream of the filter cartridges in the clean-air path inside the filter house itself. This will result in a zero fouling factor for the chilling coils and a reduction of maintenance requirements. A mist eliminator is provided downstream of the coils to prevent condensed water droplets from entering the inlet duct and causing erosion or damage to the axial compressor. Moreover, the transition duct between the filter house and the inlet duct is symmetrical in order to ensure flow uniformity over the surface of the coils. This feature, together with the placement of the coils inside the filter house, ensure an airflow speed over the surface of the coils low enough to avoid the risk of water carry-over.

The heat exchangers are usually of the plate-and-fin type. These exchangers are characterized by tubes running through plates that cover the entire height of the coil set. The tube material is usually copper and the plate material is aluminum. The mist eliminator is of the inertial type and will capture most of the condensed water droplets that may have been carried over by the air stream. In Garri power plant gas turbine existing units, the filter house could be modified by inserting the chilling module and modifying the supporting structures in order to accommodate the transition duct between the filter house and the inlet duct. Due to the continuous presence of saturated air, a stainless steel inlet duct and plenum are usually needed.

Scope of Supply:

The chilling system design depending on site specific and must be engineered on an individual basis. But the basic scope of supply includes the following:

Chilling module consisting of:

- Heat exchangers.
- Mist eliminator.
- Drain system.

In order to speed-up field activities a complete new filter house could be supplied. The system requires a suitably sized refrigeration system to feed the coils with appropriate coolant flow at the correct temperature; therefore the absorption chiller system must be installed if not currently available.

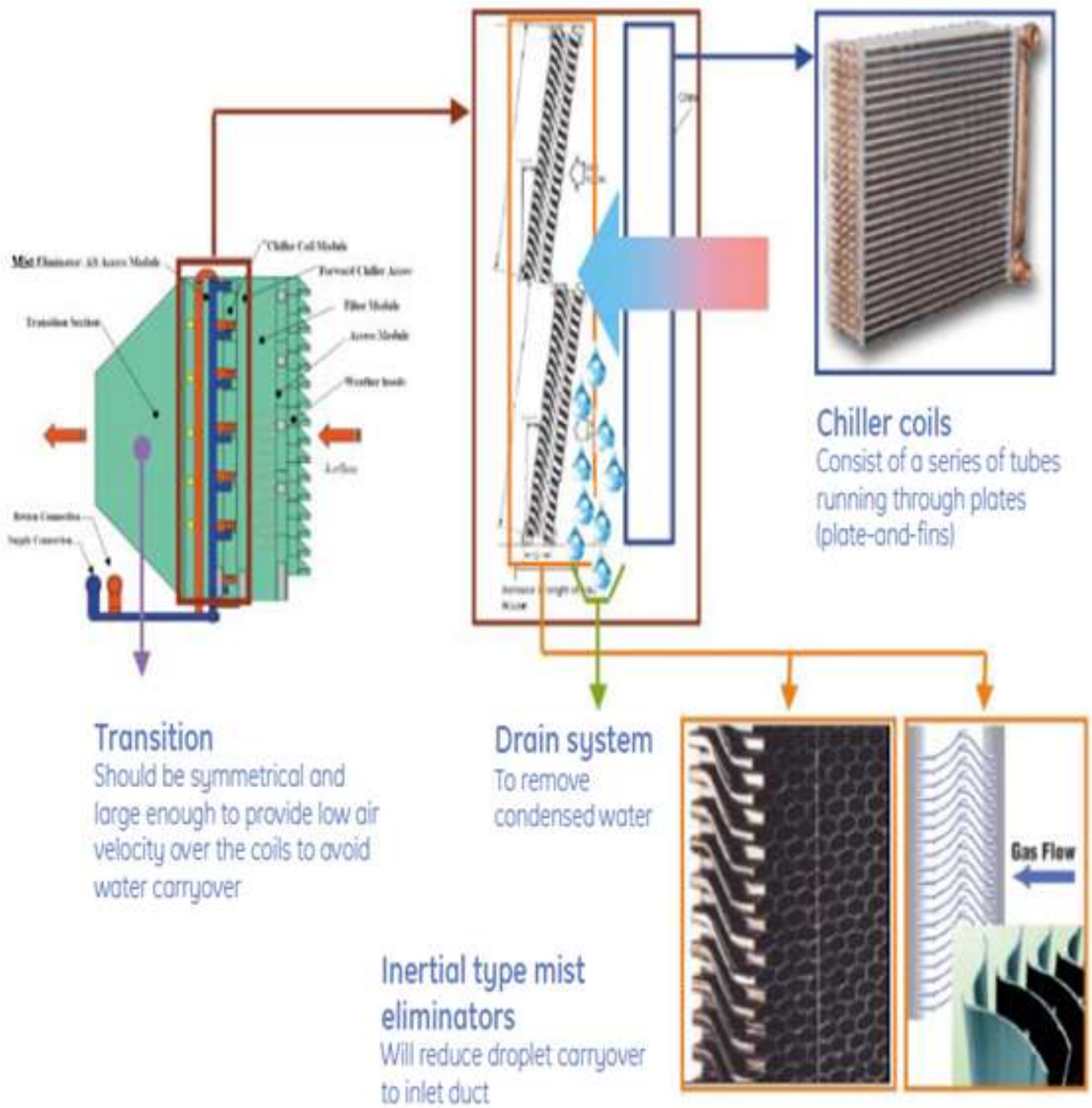


Figure (3.3) Main components of chilling system

3.3 Gas Turbine Inlet Air Cooling Available Technologies:

There are many cooling air technologies used at cooling inlet air for the gas turbine. Briefly explanation ofmain technologieswith the advantages and disadvantages for these technologies,the main types are:

1. Evaporative cooler.
2. Fogging system.

3. Mechanical refrigeration system.
4. Lithium Bromide Absorption chiller.

3.3 1Evaporative cooler:

This method is same the evaporative air cooler used at houses and its very effective at areas where RH and wet bulb temperature is rather low but the air flow is too big and used large amount of water the cooling cycle and main component are shown at schematic bellow:

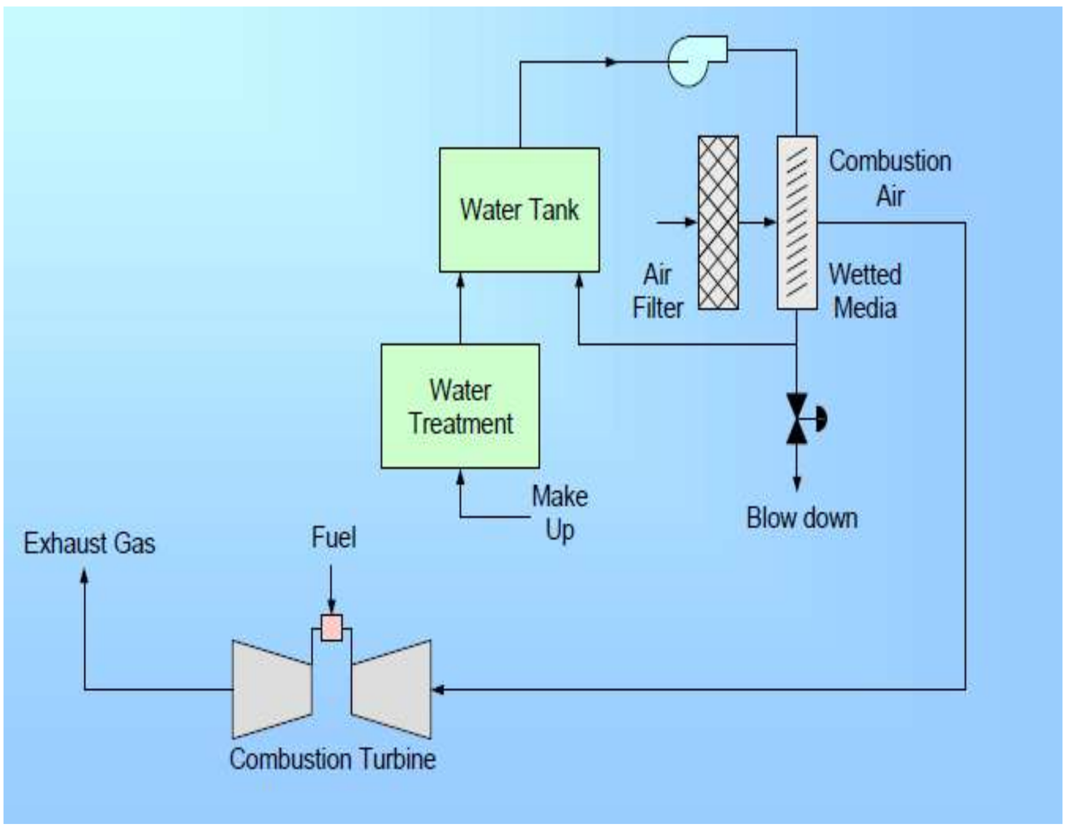


Figure (3.4) schematic shows the evaporative cooler

Advantages of the evaporative air cooling system are:

- Lowest capital cost.
- Lowest Operation &Maintenance cost..
- Can operate on raw water.
- Quick delivery and installation time.
- Operates as an air washer and cleans the inlet air.

Disadvantage of the evaporative air cooling system are:

- Limitation on capacity improvement at Garri power plant after used this technology the output power increase only by 10% and still there is 10% of rate power.
- Highly influenced by the site wet bulb temperature.

3.3.2 Fogging system

in this system spray treated water at the compressor bill moth and this system although suitable for the high ambient temperature and low relative humidity areas, and main component are shown at Figure 2.14

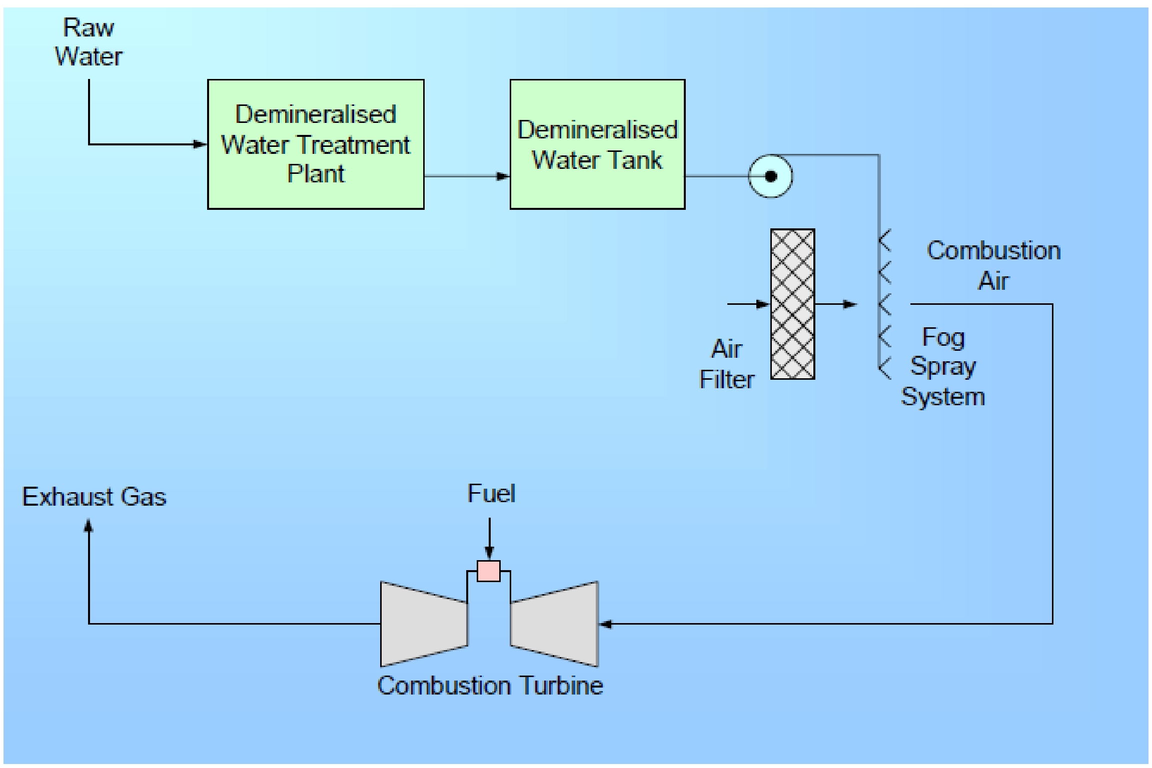


Figure (3.5) schematic shows the Fogging system.

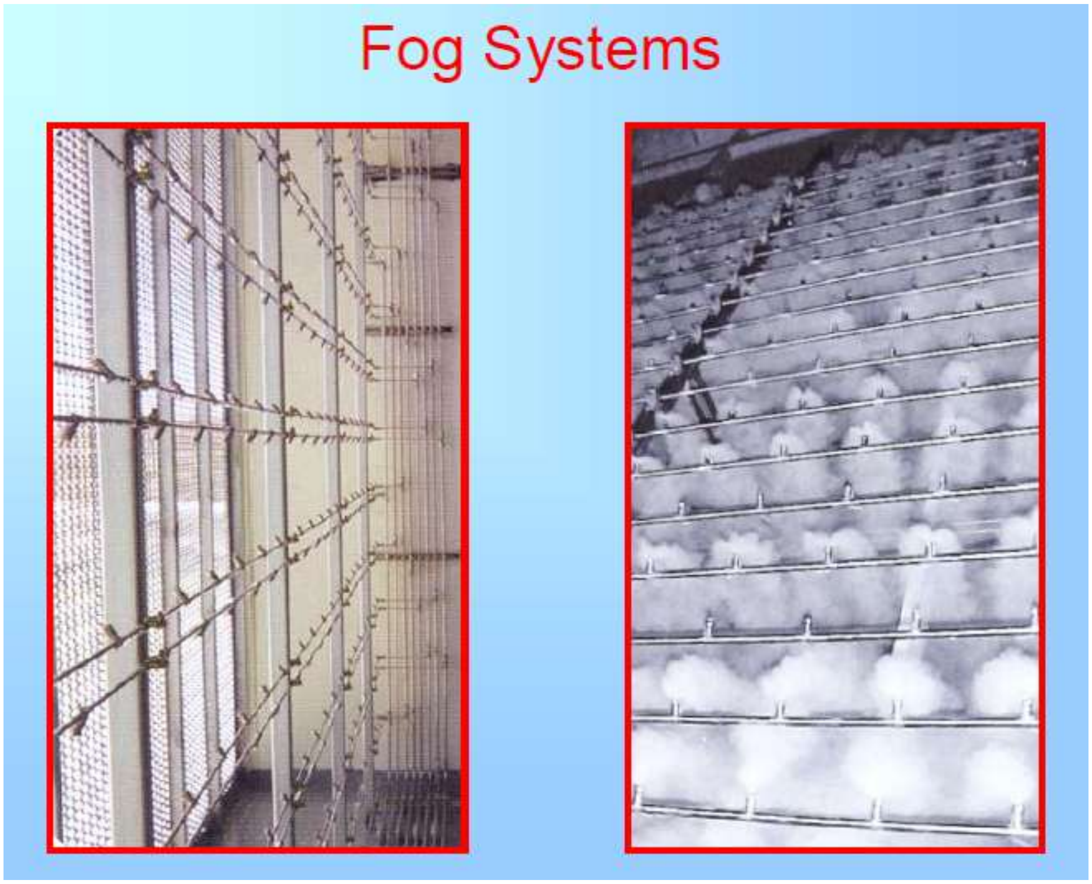


Figure (3.6) Fogging system.

Advantage of fogging system:

- Low capital cost.
- Low Operation &Maintenance cost.
- Can increase gas turbine performance better than evaporative cooling.
- Quick delivery and installation time.

Disadvantage of fogging system:

- Limitation on capacity improvement
- Highly influenced by the site wet bulb temperature.

3.3.3 Mechanical refrigeration system

This technology used mechanical devises to produced chilled water to cool inlet air at heat exchanger installed in side inlet air duct, this technology suitable for the high ambient temperature and high relative humidity areas, and main component are shown at Figure 2.16

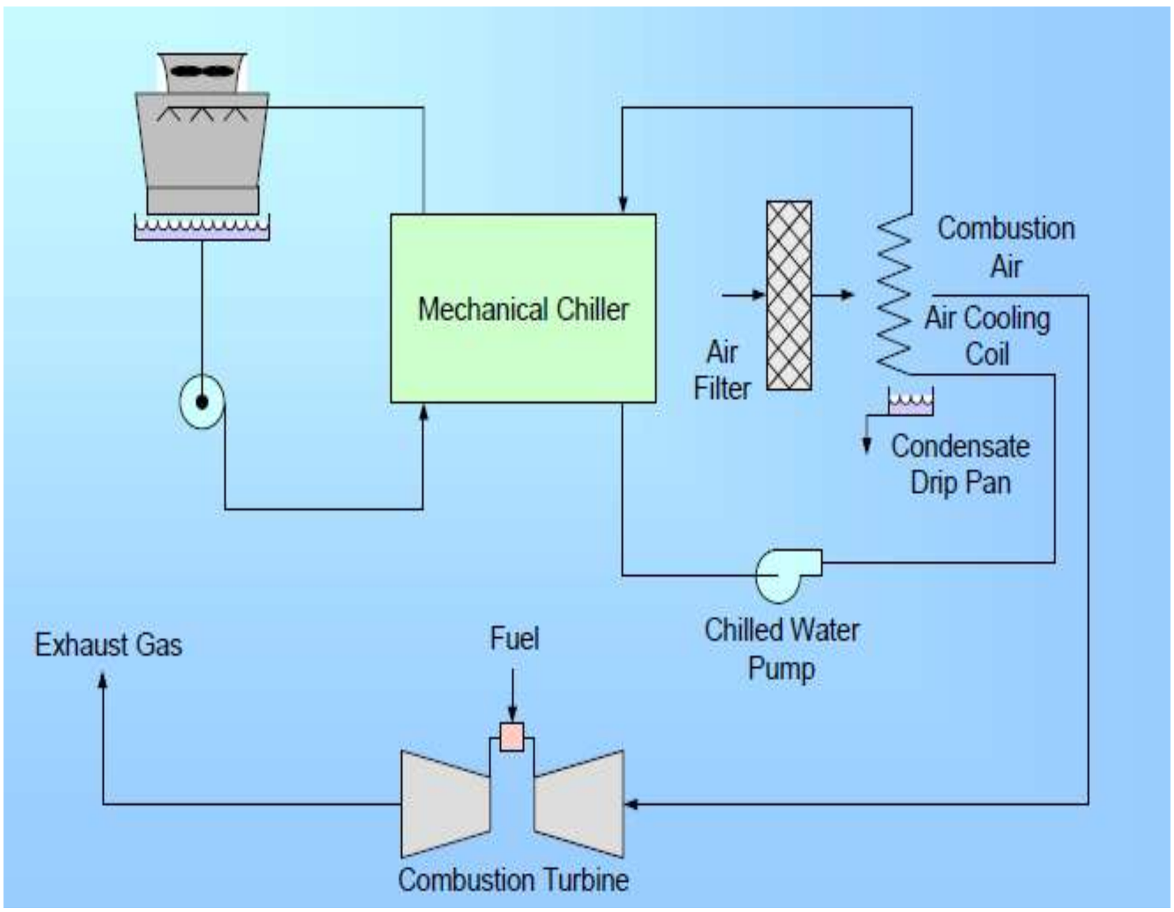


Figure (3.7) Schematic System Using a Mechanical Chiller.

Advantage of Mechanical Refrigeration System:

- Can increase gas turbine performance better than evaporative cooling, and fog system.

- Not very sensitive to ambient air wet bulb temperature.

Disadvantage of Mechanical Refrigeration System:

- High initial capital cost.
- High Operation &Maintenance cost.
- Long delivery and installation time.
- Requires chilled water cooling circuit.
- Higher parasitic load than direct type.
- Higher energy input compared to evaporative cooler.

3.3. 4 Lithium Bromide Absorption chiller:

An absorption refrigerator is a refrigerator that uses a heat source solar energy, a fossil-fueled flame, waste heat from factories, and district heating systems which provides the required energy to drive the cooling process. Cooling inlet air at gas turbine by using the heat from gas turbine exhaust this use is very efficient. Since the gas turbine then produces electricity, the main components to produced chilled water are shown at schematic bellow:

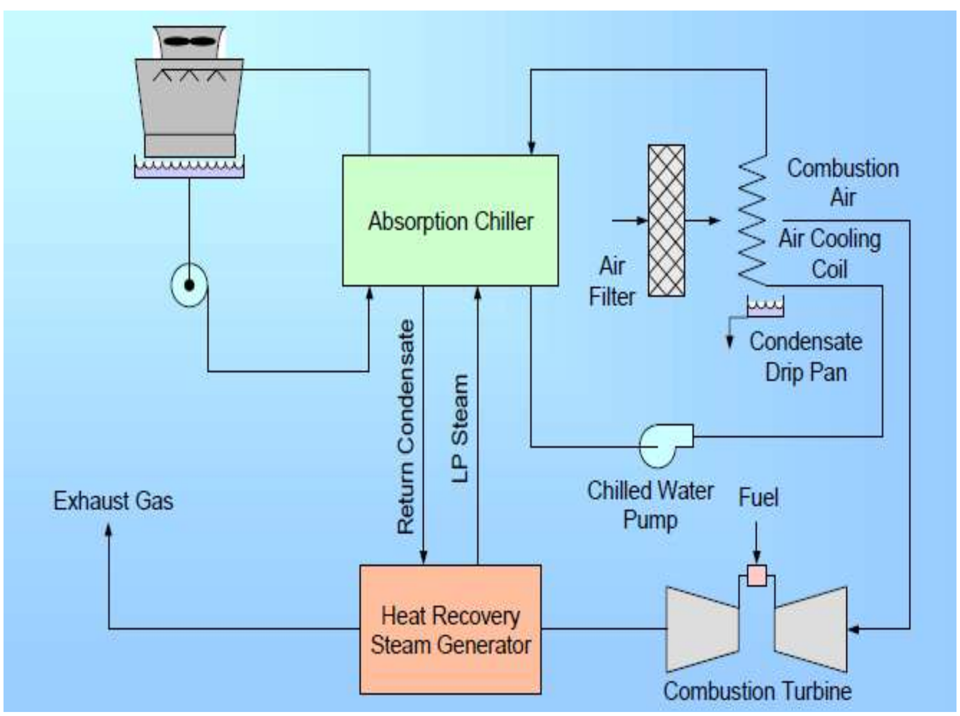


Figure (3.8) Absorption Chiller Inlet Air Cooling System Schematic

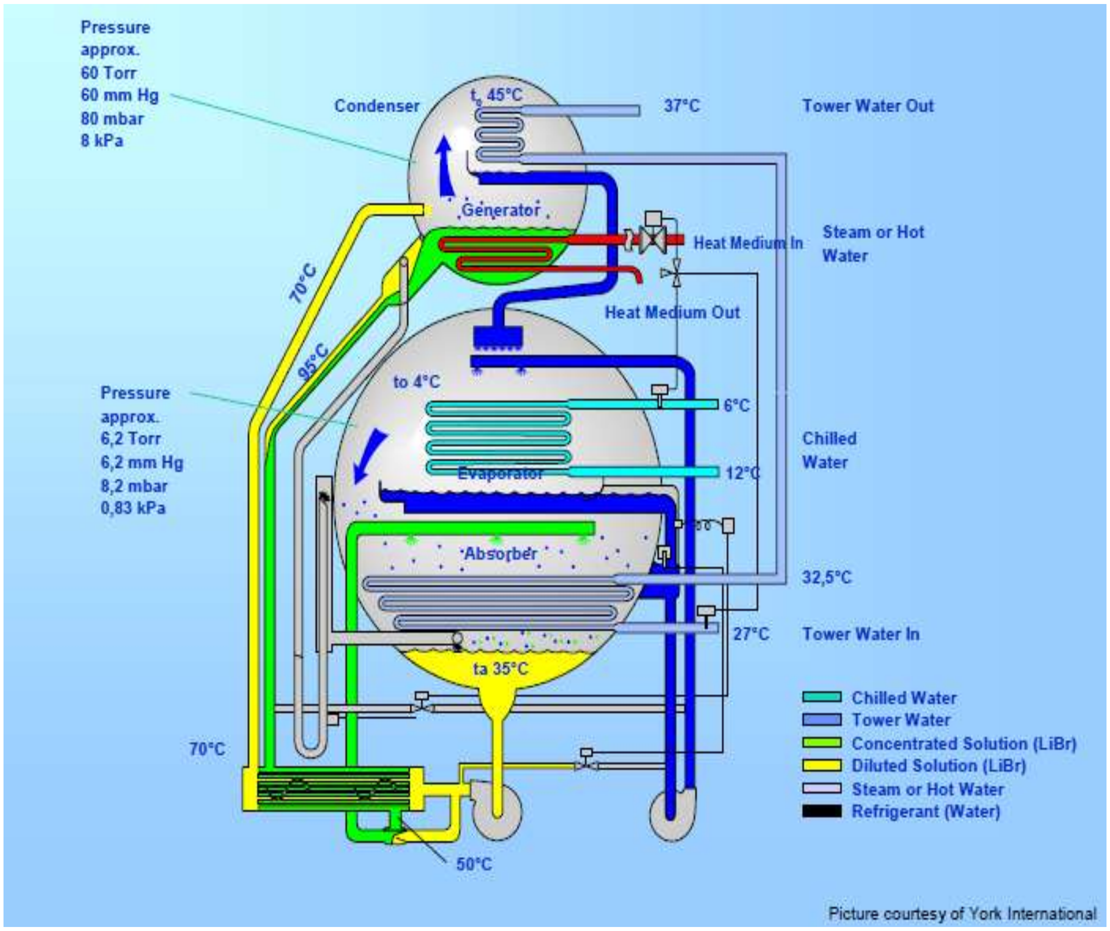


Figure (3.9) Absorption Chiller Flow Diagram from York international

Single Stage Lithium Bromide Absorption Chiller used at areas whererelative humidity is rather high, and the plantis going to operate in a combined cycle or cogeneration mode and hasaccess to low pressure steam.

Advantage of Lithium Bromide Absorption Chiller:

- Can increase gas turbineperformance better thanevaporative cooling, andfogsystem.
- Not very sensitive to ambient air wet bulb temperature.

Disadvantage of Lithium Bromide Absorption Chiller:

- High initial capital cost.

- High Operation &Maintenance cost..
- Longer delivery and installation time.
- High expertise is needed to operate and maintain the plant, Bob Omidvar [10]

In case of a steam operated chiller, cannot be applied inan open cycle gas turbineplant

3.4 Calculation:

To produce chilled water for cooling inlet air at General Electric (GE) Gas turbine frame 6 at Garri power plant to get design condition A(2) 145kg/s of chilled water must be cooled from maximum temperature at Khartoum 43°C to Design Condition 15 °C .and then calculate the following :-

- Quantity of heat must be removed from air and absorbed by chilled water.
- Quantity of chilled water need.
- Heat exchanger area.
- Number of tubes needed to install intake air filter house the exits dimensions is 7*7 meter.
- The Chiller System Selection.

  Assumptions
- Steady operating conditions exist.
- Changes in the kinetic and potential energies of fluid streams are negligible.
- Fluid properties are constant.
- The material of heat exchanger is pure copper.
- LMTD method is used to analyze the heat exchanger (Temperatures are known)

3.4.1 Quantity of heat to be removed from air:

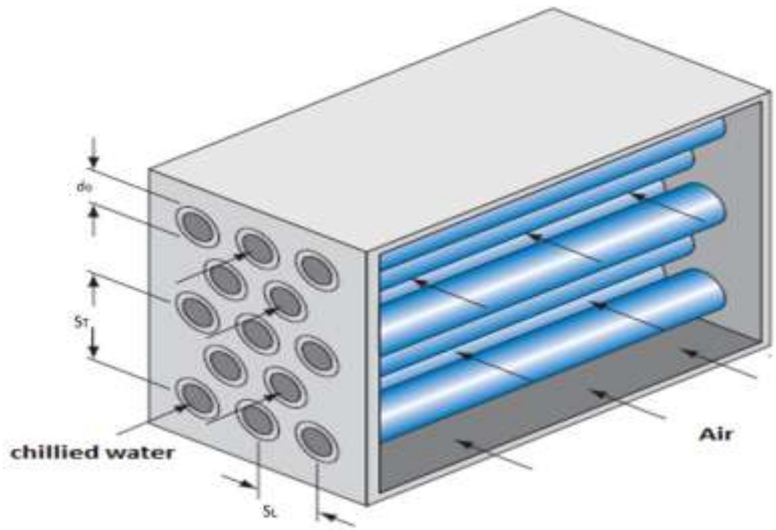


Figure 3.10 Cross flow heat exchanger (Chilled water & Air)

Table 3.1 Problem characteristics

$\dot{m}_a$ (kg/sec)	T_{chi}(°C)	T_{cho}(°C)	T_{ai}(°C)	T_{ao}(°C)
145	7	12	43	15

- From Thermo physical Properties of Air at atmospheric pressure at film temperature (T_f) figure (7)

$T_f = \frac{Tao+Tai}{2} = \frac{15+43}{2} = 29$ °C

$\rho_f = 1.1694$ kg/m^3. $cp_f = 1.007$ kJ/kg k,$\nu_f = 1.5942 * 10^{-5} m^2/sec$,

$K_f = 0.026352$ w/m^{2}k, $Pr_f = 0.7122$, $Pr_s = 0.71$

-From Thermo physical Properties of water at Average temperature (T_{avg}) figure (8)

$T_{avg} = \frac{T_{chi}+T_{cho}}{2} = \frac{7+12}{2} = 9.5$ °C

$\rho_{cw} = 999.7$kg/m^3.,$pr_{cw} = 9.62 m^2/sec$,$\nu_{cw} = 13.37 * 10^{-7} m^2/sec$,

$K_{cw} = 0.57$ W/m^{2}k ,$cp_{cw} = 4.197$kJ/kg k.

Heat lost by air $=$ Heat gained by chilled water

From equation (2.3)

$$Q_a = 145 * 1.007 * (43 - 15) = 4088.42 \text{ kw.}$$

10% factor of safety is assumed for better design.

$Q_{design} = (1 + 0.10) * Q_{calc}$

$Q_{design} = (1 + .10) * 4088.42 = 4497.262$ kw.

There for heat to be removed is 4497.262 kW.

3.4.2 Quantity of chilled water need:

Referring to first law of thermodynamics the heat removed from the air absorbed by the chilled water when neglecting loses at surrounding so the quantity absorbed by chilled water is 4497.262 kW.

From equation (2-3)

$m^{\cdot}_{ch} = \dfrac{4497.262}{4.197 * (12 - 7)} = 214.308 \text{kg/sec}$

$V_{ch} = \frac{m^{\cdot}_{ch}}{\rho_{ch}} = \frac{214.308}{999.7} = 0.2144 \frac{m^3}{sec} = 771.89 \frac{m^3}{hr}$.

3.4.3 Number of tubes need to install inside the intake air filter house:

To calculate the number of heat exchanger tubes first calculate the heat exchanger area need to occur heat transfer, at filter house cross section area is 7*7 m, and7 m long .

$\Delta T_1 = \quad 15 - 7 = 8°C$

$\Delta T_2 = 43 - 12 = 31°C$

From equation(2-5)

LMTD=$\frac{8-31}{\ln(\frac{8}{31})}$=16.979°C or K.

P $= \frac{12-7}{43-7} = 0.13889.$ equation (2.6)

R $= \frac{43-15}{12-7} = 5.6$ equation (2.7)

At P=0.13889, R=5.6 .figure 6

F=0.7844.

Table 3.2 Heat Exchanger parameters

D_o(mm)	D_i(mm)	d_o(mm)	d_i(mm)	L(m)	$m^{\cdot}_{ch}$ ($^{kg}/_{sec}$)
107.6	101.6	10	8	7	214.308

3.4.3.1 Volume flow rate of chilled water through the large pipe:

$A_{cw} = \frac{\pi * D_i{}^2}{4} = \frac{\pi * 0.1016^2}{4} = 0.008107\ m^2.$

$$V_{cw} = \frac{214.308}{999.7 * 0.008107} = 26.4428\ m/\sec.$$

$q_{supply} = 0.008107 * 26.4428 = 0.2143 m^3/sec$ From equation(2-8)

3.4.3.2 Number of rows:

To find the number of rows divide the heat exchange length (7m) to the pipe diameter plus space between the pipe assume that the pipe diameter is 0.01m space equal to the pipe diameter (0.02m) from above

The number of pipes at single row:

$d_{(both\ sides)}$=0.02 m

$space_{(both\ sides)}$= 0.02m

$$\text{Number of rows} = \frac{7}{0.04} = 175 \text{rows}$$

$N_{row} = 175$ rows .

3.4.3.3 Volume flow rate of chilled water through the small pipe:

$q_{tube} = \frac{q_{supply}}{N_{row}} = \frac{0.2143}{175}$=0.0012249 m^3/sec.

$$A_{tube} = \frac{\pi * d_i{}^2}{4} = \frac{\pi * 0.008^2}{4} = 5.026 * 10^{-5} m^2.$$

$$v_{tube} = \frac{q_{tube}}{A_{tube}} = \frac{0.0012249}{5.026 * 10^{-5}} = 24.373 \text{ m/sec}$$

3.4.3.4 Overall heat transfer coefficient:

$Re_{cw} = \frac{24.373*.008}{13.37*10^{-7}} = 145836.6$ equation (2-10)

145836.6> 4000

The flow of chilled water is turbulent.

$Nu_{cw} = 0.023 * Re_{cw}^{0.8} * Pr_{cw}^{0.4} = 0.023 * (145836.6)^{0.8} * (9.62)^{0.4} =$ 613.44equation(2-11)

$h_{cw} = \frac{613.44*0.57875}{0.008} = 44378.7193 W/m^2k$equation(2-12)

Fouling factor of chilled water

$R_{f,cw}$ =0.0001figures 13

- For outside tube (air)

$A_a = 7 * 7 = 49 \text{ m}^2$.

$V_a = \frac{m^{\cdot}_a}{\rho_a * A_a} = \frac{145}{1.1694*49}$=2.5305 m/sec.

Frank , Raj ,Mark [16]

$A_{min} = (0.04 - 2 * 0.01) * 7 * 175 = 24.5 \text{ m}^2$ .equation(2-13)

$U_{max} = \frac{0.04/2}{\sqrt{(0.04)^2 + (0.04/2)^2)} - 0.01} * 2.5305 = 1.457 \frac{m}{sec}$. equation (2-14)

$Re_a = \frac{1.457*0.01}{1.5942*10^{-5}} = 914$ equation(2-10)

50<914<1000

This is in the laminar regime

Since $S_T / S_L = 40/40 = 1 < 2$

For staggered bank with $S_T / S_L < 2$ figures 10

$$Nu_D = 0.9 * (908.29)^{0.4} * (0.7122)^{0.36} * (\frac{0.7122}{0.71})^{0.25} = 12.18$$

$h_a = \frac{12.18*0.026352}{0.010} = 32.11 \text{ w/m}^2\text{k}$.

Fouling factor of air

$R_{f,a}$ =0.0004 fingers 13

$a_i = \pi * 0.008 * 7 = 0.1759 m^2$

$a_o = \pi * 0.010 * 7 = 0.2199 m^2$.

From equation (2-15)

$$\frac{1}{U_o * a_o} = \frac{1}{44378.7193 * 0.1759} + 0 + \frac{1}{32.034 * 0.2199} + \frac{0.0001}{0.1759} + \frac{0.0004}{0.2199}$$

$U_o = 241.94 \text{ W/m}^2\text{k}$.

$A_o = \frac{4497.262*1000}{241.94 *16.979*0.7844} = 1395.65 \text{ m}^2$equation(2-4)

3.4.3.5 Heat Exchanger Area:

$A_o = 1395.65 \text{ m}^2$.

Number of Tubes

$N_{tubes} = \frac{A_o}{\pi * d_o * l} = \frac{1395.65}{\pi*0.010*7} = 6346.4 \approx 6347 \text{ tubes}$. equation (2-16)

3.4.3.6 The Number of cooling stages:

$$Nst = \frac{N_{tubes}}{N_{row}} = \frac{6347}{175} = 36.27 \text{ stages} \approx 37 \text{ stages.}$$

3.5 The Chiller System Selection:

From performance

At LWT= 44 °F, T.CAP = Q (k w)/3.5=4497.262/3.5=1283 Ton.

The chilled water flow rate (WFR) $= 771.89 \frac{m^3}{hr} = 3398.52 \frac{gallon}{min}$.

From product catalogue of absorption chiller we need there chiller two in services and one is standby.[13]

From table of technical parameter we select type 661H2.

- Cooling capacity =2330 kw
- Inlet and outlet temperature 12,7C

- Hot water Inlet and outlet temperature130,68C
- Consumption of Hot water40.8 ton/hr

3.6 Hot Water Solar Design:

Hot water Inlet and outlet temperature130,68C and Consumption of Hot water11.33kg/s we are use parabolic solar collector.

Solar radiation received in Khartoum-155N-325E-based-on-optimal-tilt-and-annual.png [14]

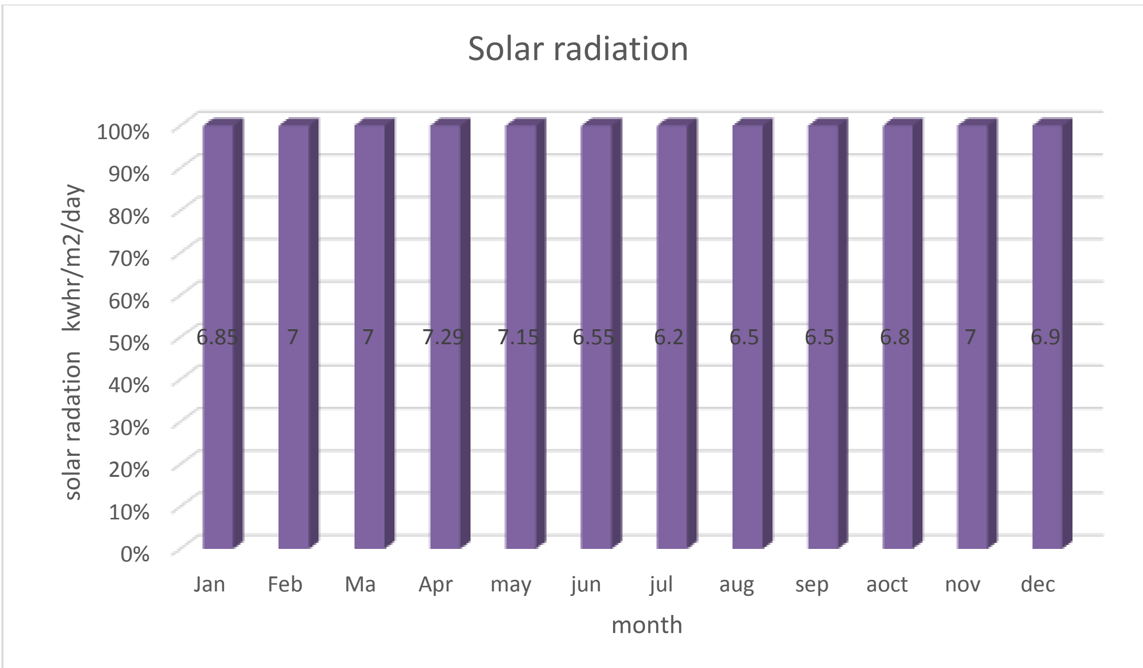


Figure no(3.11) solar radiation received in Khartoum

Collectors Characteristics:

Table (3.3) collectors characteristics

Length(m)	Width (m)	Cover diameter(m)	F_r	R	ρ	τα	Γ	$V_{wind}(\frac{m}{s}$	σ (J/m^{2}sK^4)	ϵ
4.06	2	0.65	0.84	0.9	0.85	0.77	0.94	4.2	$5.67 * 10^{-8}$	0.9

$Q = 11.33 * 4.18 * 62$ Equation (2-3)

$Q = 2936.3$ KW

C=$\frac{2.5\ 0.065}{3.14*0.065}$=11.93Equation (2-21)

S=295.83*0.85*0.93*0.78+295.83*0.78*$\frac{0.65}{2.5-0.65}$Equation (2-18)

S=263.45w/m^2

$h_{wind} = 5.7 + 3.8 * 4.2 = 21.66$ Equation (2-20)

Assumptions:

$T_a = \text{Ambient temperature} = 30℃$

$T_r = \text{Recive surface temperature } 150℃$

$H_{av} = \frac{7.1\text{kwhr}}{m^2\text{day}} = 295.8323\ w/m^2$

$h_{wind} = 5.7 + 3.8V = 5.7 + 3.8 * 4.2 = 21.66\frac{w}{m^2}℃$

$T^- = \frac{Tr + Ta}{2} = \frac{150 + 30}{2} = 90℃ = (90 + 273) = 363K$

$h_r = 4 * 5.67 * 10^{-8} * 0.9 * 363^3 = 9.763\ w/m^2 ℃$Equation (2-21)

$U_l = \left[\frac{1}{21.66} + \frac{1}{9.763}\right]^{-1} = 6.7296\ w/m^2 ℃$Equation (2-19)

$q_u = 0.84 * (2.5 - 0.65)9\left[263.47 - \frac{6.7296}{11.93}(150 - 30)\right]$Equation (2-17)

q_u =2738.4 kw

Number of collectors = $\frac{Q}{q_u}$ =$\frac{2936.2}{2738.4}$ =1.072≈2 collectors

For one unit we need 2*3=6 collectors

CHAPTER IV

RESULTS AND DISCUSSIONS

CHAPTER IV

RESULTS AND DISCUSSIONS

4.1 Results and discussions:

The power output of a gas turbine (GT) will increase by around 10 MW. Totally the combined cycle power may increase by about 12 MW From below table the average heat rate saved is 1138.9 KJ/KWh this means that we can save more than 1138.9 kJ of fuel energy per each kWh of generation to calculate the amount of fuel saved divided the heat rate per kw to the Lower Heat Value for diesel is 42612 kJ/kg found that Fuel saved $= \frac{1138.9\ \text{kJ/kWh}}{42612\ \text{kJ/kg}} =$ 0.02676 kg/kWh

Before installing chilling system				After install chilling system				Saved Heat rate
Unit	MW generated at 5 July 2008	Efficiency %	heat rate kJ/kWh	Unit	MW designed	Efficiency%	heat rate kJ/kWh	
Unit 1	30	30	11909.0	Unit 1	42	32	11225.7	
Unit 2	30	29.8	12048.7	Unit 2	42	32	11225.7	
Unit 3	29	30	11886.7	Unit 3	42	32	11225.7	
Unit 4	27.5	29	12353.7	Unit 4	42	32	11225.7	
Unit 5	28	28	12772.1	Unit 5	42	32	11225.7	
Unit 6	28	27	13287.0	Unit 6	42	32	11225.7	
Unit 7	28	28.5	12630.8	Unit 7	42	32	11225.7	
Unit 8	31	28	12767.0	Unit 8	42	32	11225.7	

For example unit 6 at 14 august2015 generated 715000 kW*h

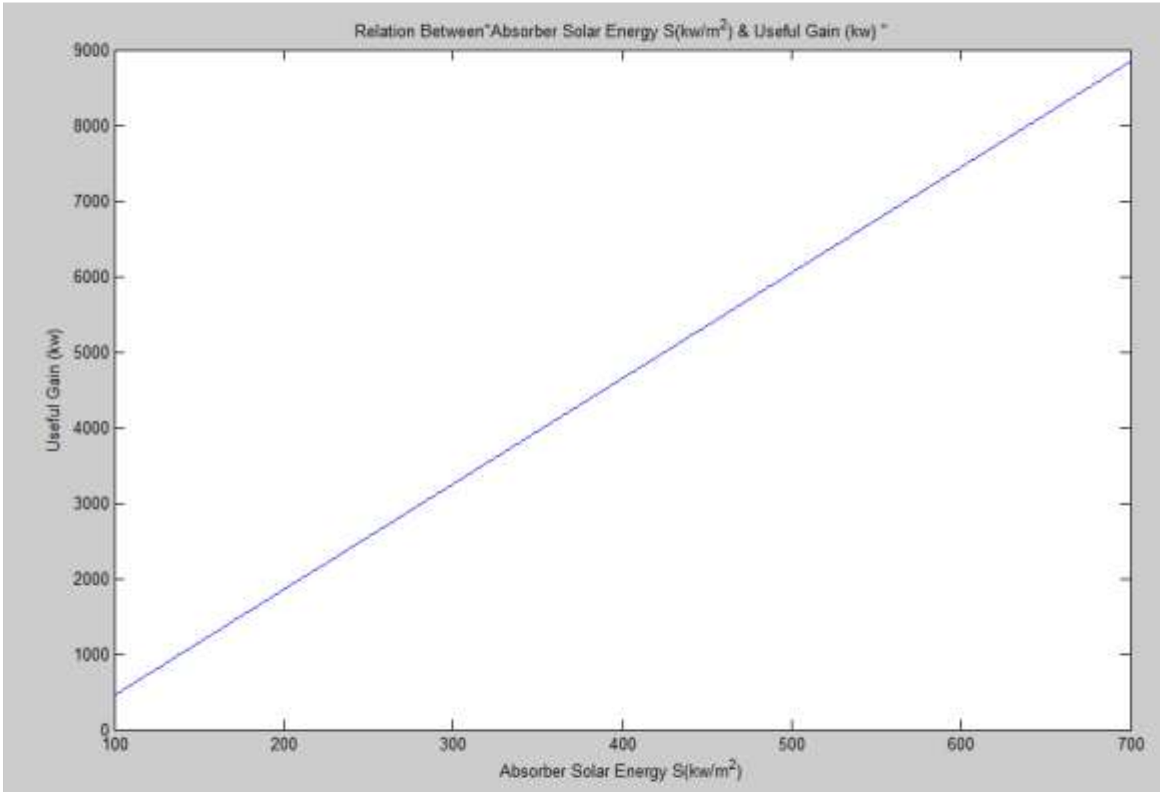
Fuel saved from unit6 =715000 kW*h ×0.02676 kg/kWh =19.133 ton.

I think this amount of fuel made the project attractive to do additional researched for installation and study the environment impact to area.

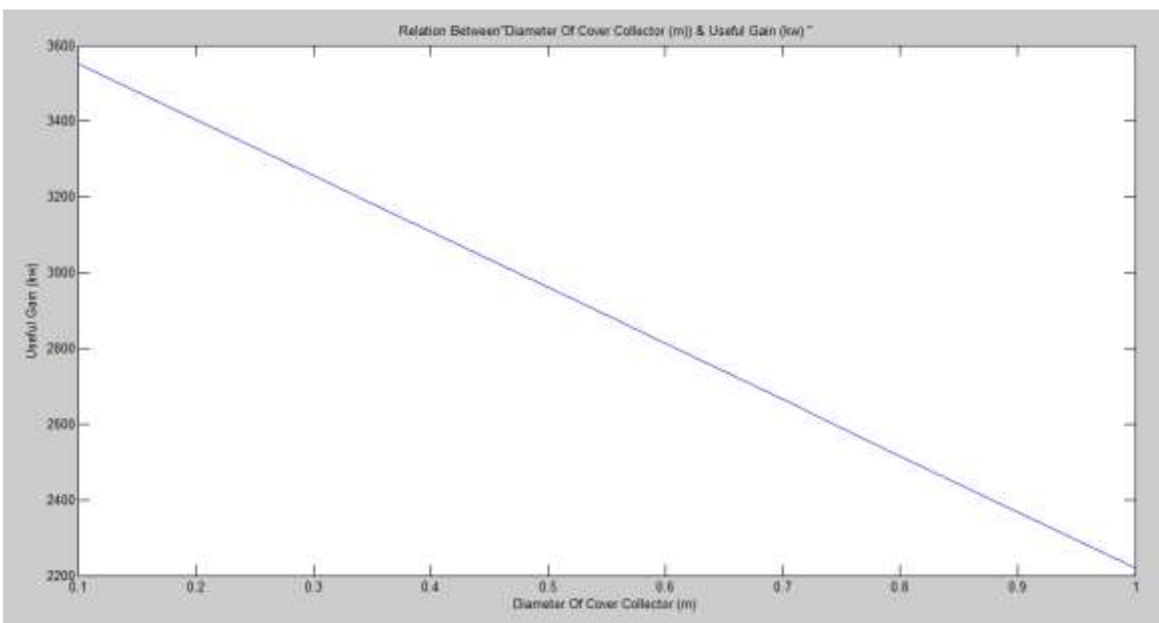
Additionally using cooling system at the inlet air to the gas turbine well increase the exhaust mass flow to the heat recovery steam generate (HRSG) Which help at increasing the amount of steam generated and increasing power from steam turbine at combined cycle need another research to study this effect but its visible because when using evaporative cooler at combined cycle the power produced from gas turbine increased by 4 MW and steam turbine increased by 1 MW and when we used chilling system the power increase at gas turbine by 10 MW expected 2MW increased at steam turbine.

At the table above we take reading from one day but the deviations from it not big and compare it with standard condition, the gas turbine efficiency not affected only by ambient temperature the engine internal parts condition, intake air filter condition, the location altitude and operation condition.

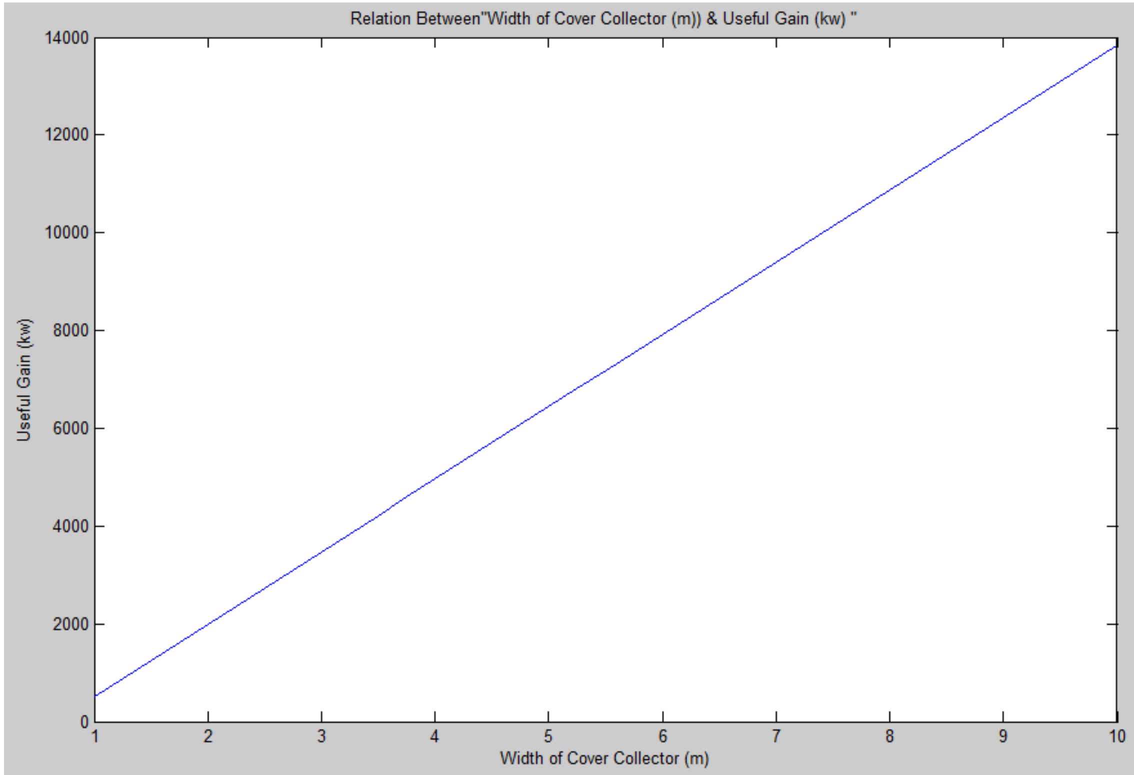
For solar collectors results by using matlab program below shows that the relationshipbetween useful gain and (Absorber solar energy, diameter of cover, recive surface temperature, width of collectors, length of collectors and ambient temperature) .



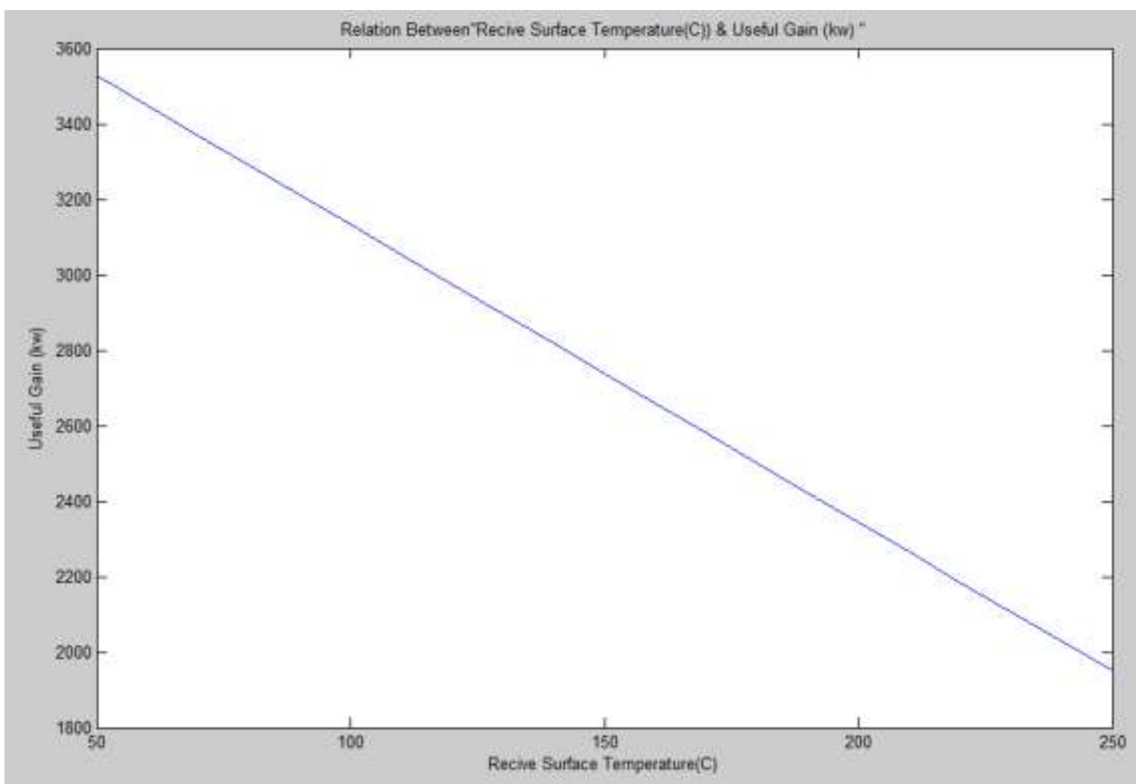
Relation Between"Absorber Solar Energy S(kw/m2) & Useful Gain (kw) "
Useful Gain (kw)
Absorber Solar Energy S(kw/m2)



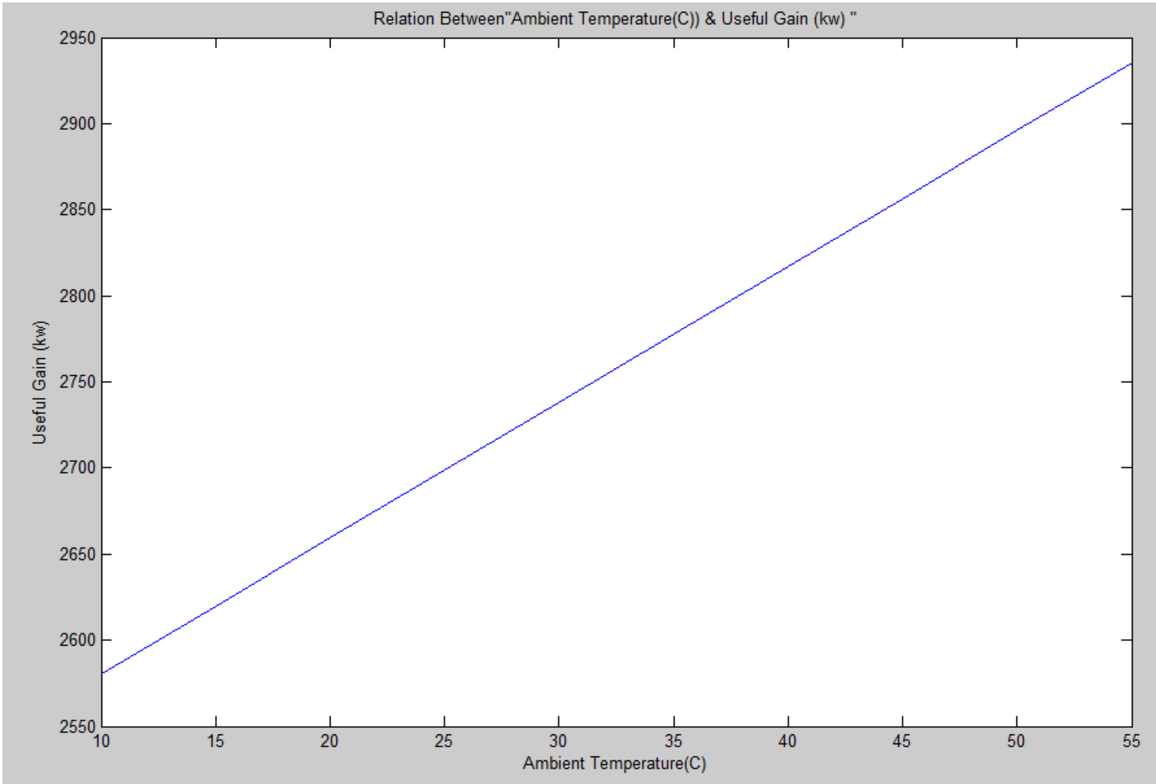
Relation Between"Diameter Of Cover Collector (m)) & Useful Gain (kw) "
Useful Gain (kw)
Diameter Of Cover Collector (m)



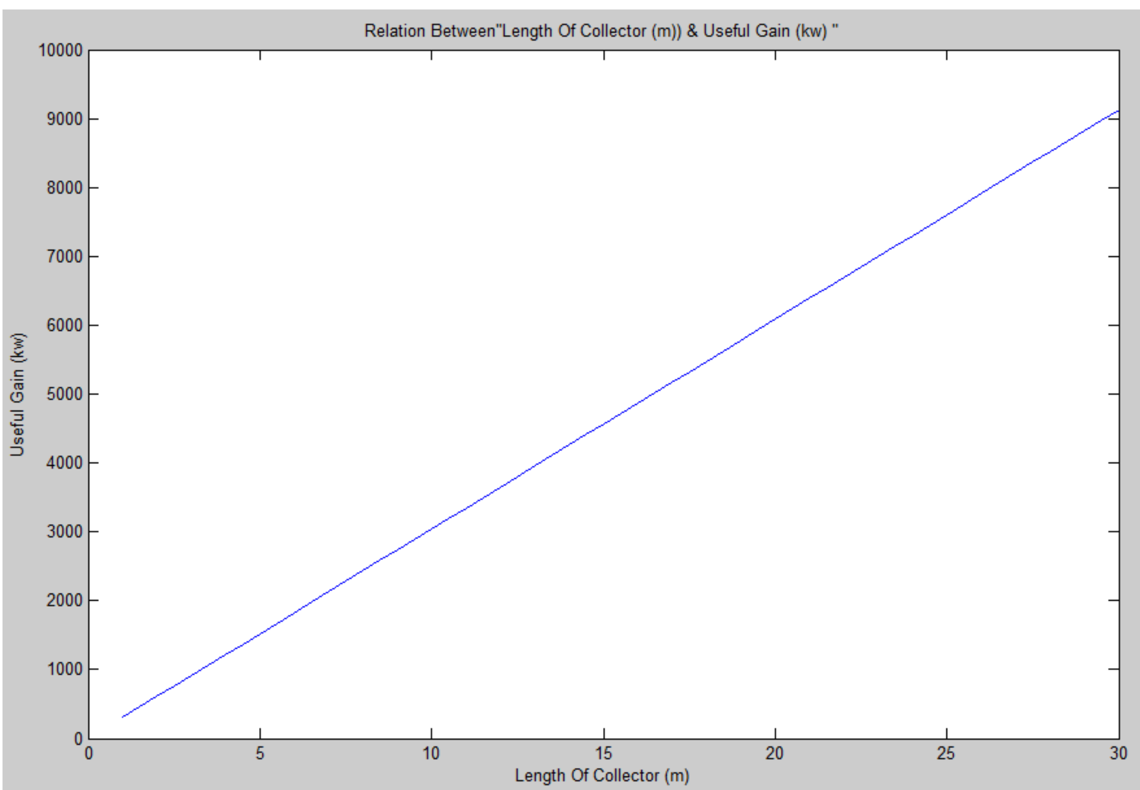
Relation Between"Width of Cover Collector (m)) & Useful Gain (kw) "
Useful Gain (kw)
Width of Cover Collector (m)
0
2000
4000
6000
8000
10000
12000
14000
1
2
3
4
5
6
7
8
9
10



Relation Between"Recive Surface Temperature(C)) & Useful Gain (kw) "
Useful Gain (kw)
Recive Surface Temperature(C)
1800
2000
2200
2400
2600
2800
3000
3200
3400
3600
50
100
150
200
250



Relation Between"Ambient Temperature(C)) & Useful Gain (kw) "
Useful Gain (kw)
2950
2900
2850
2800
2750
2700
2650
2600
2550
10
15
20
25
30
35
40
45
50
55
Ambient Temperature(C)



Relation Between"Length Of Collector (m)) & Useful Gain (kw) "
Useful Gain (kw)
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
0
0
5
10
15
20
25
30
Length Of Collector (m)

CHAPTER V

CONCLUSIONS & RECOMMENDATIONS

CHAPTER V

CONCLUSIONS & RECOMMENDATIONS

5.1 Conclusions:

A number of power-augmentation methods have been used in order to compensate for the effect of ambient conditions on the gas-turbine output. The most common power augmentation method increases the air mass flow-rate by cooling the inlet air. Inlet-air cooling also reduces the heat-rate of the plant Compared to other power augmentation methods, such as steam or water injection. By applying this reliable technology additional megawatts can be obtained from existing gas turbines at a fraction of the cost of installing new generation plants. There are many type of energy sources available to operate absorption chiller at Garri power station (Exhaust air - Steam –hot water) during night solar not available.

A simple comparison between the Mechanical refrigeration system and Lithium Bromide Absorption chiller We find that the Mechanical refrigeration system uses estimated amounts of electrical energy, which makes Lithium Bromide Absorption chiller more suitable for use in the station Garri power station. The new power plant can be delayed by enhancing the production capacity of existing power stations.

At this research we try to reflect this problem to get maximum power from gas turbine by using chilled water to reduced inlet air temperature to the design to get 25% losses load and the main advantage of this method is:

- Can increase gas turbine performance better than evaporative cooling, and fog system.
- Reduces the heat-rate of the plant to save fuel.
- Not very sensitive to ambient air wet bulb temperature.

From this research we get the maximum power from gas turbine but there are many disadvantages must take at account during design and construction

- High initial capital cost.
- High maintenance cost

5.2 Recommendations:

The recommendations are to decrease the disadvantages and further research is required such as:

- Site study at Garri power plant for water, power, space to install the equipment.
- Study the effect of the cooler to the related systems like steam turbine and it's auxiliary.
- For new power plant project Site study should be done to design the power plant machine and systems according to the local environment parameter.
- Study the available technology to get solar collectors operate 24 hours.

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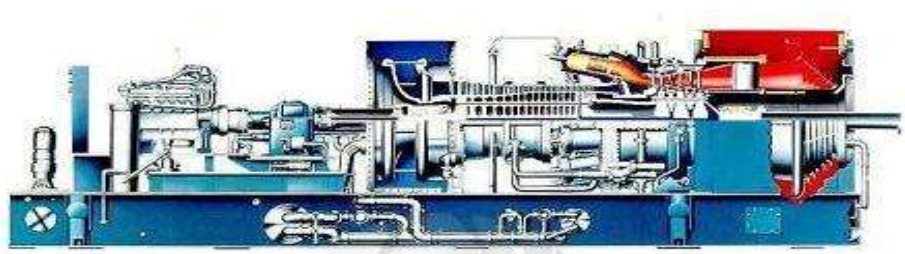
Appendix

Figure no (1) Garri power plant station

Figure (2) gas turbine

Figure (3) inlet air system



Operating Parameters (Base Load)

Performance	Natural Gas	Distillate
ISO Output (kW)	42100	41240
ISO Heat Rate Kcal / kW-hr,LHV	2683	2704
Exhaust Flow (kg/sec)	145.8	146.2
Exhaust Temperature ° C	552	552
Speed (RPM)	5163	5163

Construction Features

Compressor Stages: 17

Turbine Stages : 3

Number of Combustors : 10

Load Gear Box Design : Horizontal, offset, double helical gear

Dimensions, L x W x H mtrs: Turbine Base - 8 x 3.3 x 3.8 Accessory Base 7 x 3.3 x 3.8

Figure 4: operating parameters (Base Load).

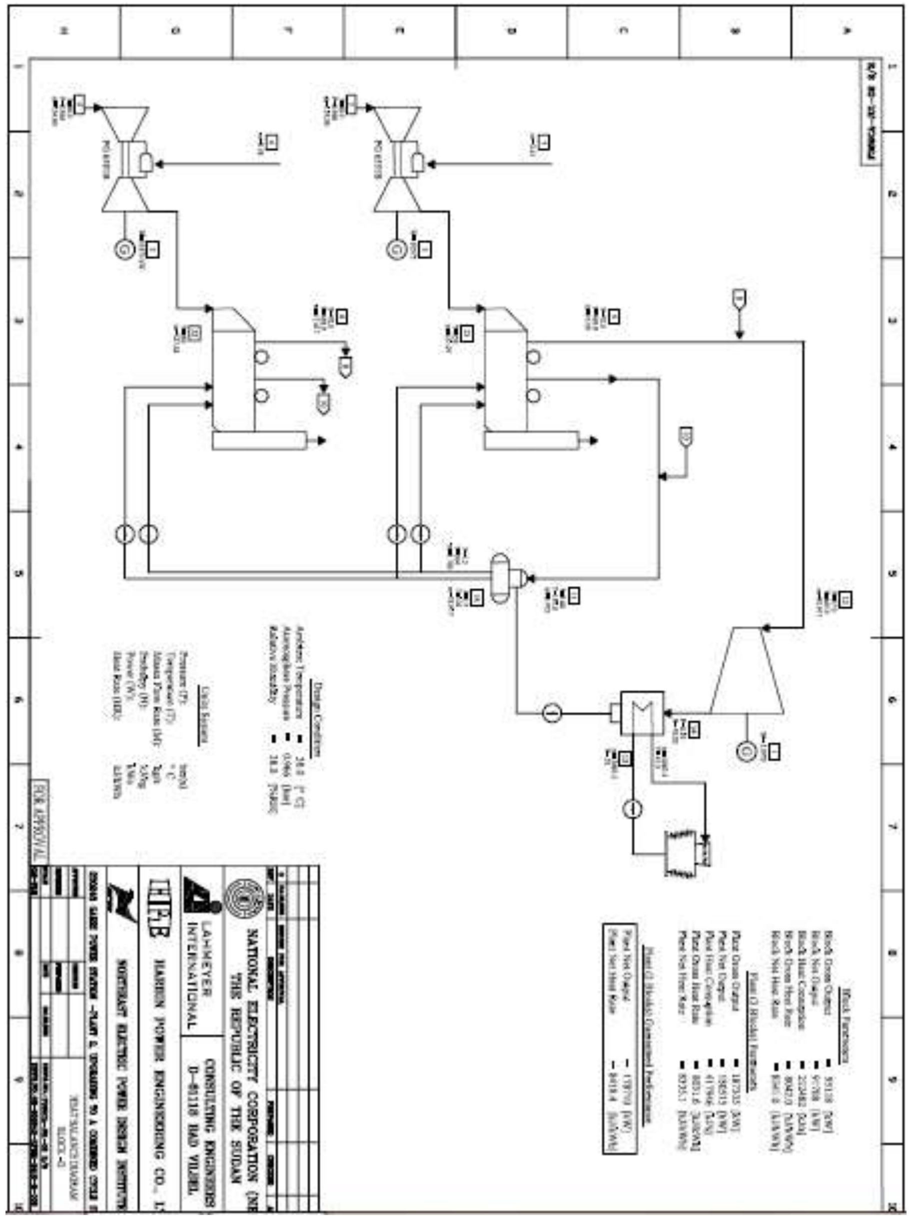


Figure5: Heat Balance of combined cycle.

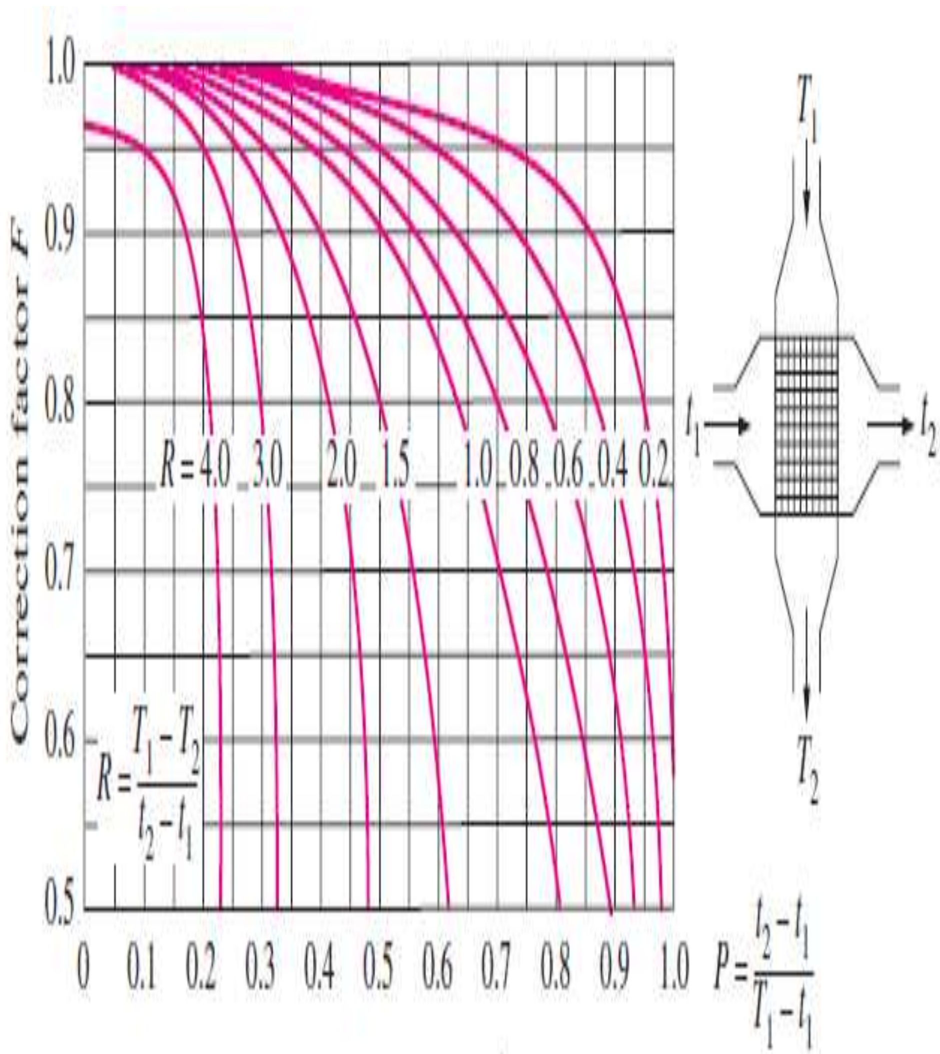


Figure 6: Single-pass cross-flow with both fluids unmixed.

T (K)	ρ (kg/m^3)	c_p (J/kg·K)	μ (kg/m·s)	ν (m^2/s)	k (W/m·K)	α (m^2/s)	Pr
Air							
100	3.605	1039	0.711×10^{-5}	0.197×10^{-5}	0.00941	0.251×10^{-5}	0.784
150	2.368	1012	1.035	0.437	0.01406	0.587	0.745
200	1.769	1007	1.333	0.754	0.01836	1.031	0.731
250	1.412	1006	1.606	1.137	0.02241	1.578	0.721
260	1.358	1006	1.649	1.214	0.02329	1.705	0.712
270	1.308	1006	1.699	1.299	0.02400	1.824	0.712
280	1.261	1006	1.747	1.385	0.02473	1.879	0.711
290	1.217	1006	1.795	1.475	0.02544	2.078	0.710
300	1.177	1007	1.857	1.578	0.02623	2.213	0.713
310	1.139	1007	1.889	1.659	0.02684	2.340	0.709
320	1.103	1008	1.935	1.754	0.02753	2.476	0.708
330	1.070	1008	1.981	1.851	0.02821	2.616	0.708
340	1.038	1009	2.025	1.951	0.02888	2.821	0.707
350	1.008	1009	2.090	2.073	0.02984	2.931	0.707
400	0.8821	1014	2.310	2.619	0.03328	3.721	0.704
450	0.7840	1021	2.517	3.210	0.03656	4.567	0.703
500	0.7056	1030	2.713	3.845	0.03971	5.464	0.704
550	0.6414	1040	2.902	4.524	0.04277	6.412	0.706
600	0.5880	1051	3.082	5.242	0.04573	7.400	0.708
650	0.5427	1063	3.257	6.001	0.04863	8.430	0.712
700	0.5040	1075	3.425	6.796	0.05146	9.498	0.715
750	0.4704	1087	3.588	7.623	0.05425	10.61	0.719
800	0.4410	1099	3.747	8.497	0.05699	11.76	0.723
850	0.4150	1110	3.901	9.400	0.05969	12.96	0.725
900	0.3920	1121	4.052	10.34	0.06237	14.19	0.728
950	0.3716	1131	4.199	11.30	0.06501	15.47	0.731
1000	0.3528	1142	4.343	12.31	0.06763	16.79	0.733
1100	0.3207	1159	4.622	14.41	0.07281	19.59	0.736
1200	0.2940	1175	4.891	16.64	0.07792	22.56	0.738
1300	0.2714	1189	5.151	18.98	0.08297	25.71	0.738
1400	0.2520	1201	5.403	21.44	0.08798	29.05	0.738
1500	0.2352	1211	5.648	23.99	0.09296	32.64	0.735

Figure 7: Thermo physical Properties of Air at atmospheric pressure (1.01325pa).

Temperature K	°C	ρ (kg/m^3)	c_p (J/kg·K)	k (W/m·K)	α (m^2/s)	ν (m^2/s)	Pr	β (K^{-1})
				Water				
273.16	0.01	999.8	4220	0.5610	1.330×10^{-7}	17.91×10^{-7}	13.47	-6.80×10^{-5}
275	2	999.9	4214	0.5645	1.340	16.82	12.55	-3.55×10^{-5}
280	7	999.9	4201	0.5740	1.366	14.34	10.63	4.36×10^{-5}
285	12	999.5	4193	0.5835	1.392	12.40	8.91	0.000112
290	17	998.8	4187	0.5927	1.417	10.85	7.66	0.000172
295	22	997.8	4183	0.6017	1.442	9.600	6.66	0.000226
300	27	996.5	4181	0.6103	1.465	8.568	5.85	0.000275
305	32	995.0	4180	0.6184	1.487	7.708	5.18	0.000319
310	37	993.3	4179	0.6260	1.508	6.982	4.63	0.000361
320	47	989.3	4181	0.6396	1.546	5.832	3.77	0.000436
340	67	979.5	4189	0.6605	1.610	4.308	2.68	0.000565
360	87	967.4	4202	0.6737	1.657	3.371	2.03	0.000679
373.15	100.0	958.3	4216	0.6791	1.681	2.940	1.75	0.000751
400	127	937.5	4256	0.6836	1.713	2.332	1.36	0.000895
420	147	919.9	4299	0.6825	1.726	2.030	1.18	0.001008
440	167	900.5	4357	0.6780	1.728	1.808	1.05	0.001132
460	187	879.5	4433	0.6702	1.719	1.641	0.955	0.001273
480	207	856.5	4533	0.6590	1.697	1.514	0.892	0.001440
500	227	831.3	4664	0.6439	1.660	1.416	0.853	0.001645
520	247	803.6	4838	0.6246	1.607	1.339	0.833	0.001909
540	267	772.8	5077	0.6001	1.530	1.278	0.835	0.002266
560	287	738.0	5423	0.5701	1.425	1.231	0.864	0.002783
580	307	697.6	5969	0.5346	1.284	1.195	0.931	0.003607
600	327	649.4	6953	0.4953	1.097	1.166	1.06	0.005141
620	347	586.9	9354	0.4541	0.8272	1.146	1.39	0.009092
640	367	481.5	25,940	0.4149	0.3322	1.148	3.46	0.03971
642	369	463.7	34,930	0.4180	0.2581	1.151	4.46	0.05679
644	371	440.7	58,910	0.4357	0.1678	1.156	6.89	0.1030
646	373	403.0	204,600	0.5280	0.06404	1.192	18.6	0.3952
647.0	374	357.3	3,905,000	1.323	0.00948	1.313	138.	7.735

Figure 8: Thermo physical Properties of water at atmospheric pressure.

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HEAT TRANSFER

TABLE 1–1

The thermal conductivities of some materials at room temperature

Material	k, W/m · °C*
Diamond	2300
Silver	429
Copper	401
Gold	317
Aluminum	237
Iron	80.2
Mercury (l)	8.54
Glass	0.78
Brick	0.72
Water (l)	0.613
Human skin	0.37
Wood (oak)	0.17
Helium (g)	0.152
Soft rubber	0.13
Glass fiber	0.043
Air (g)	0.026
Urethane, rigid foam	0.026

*Multiply by 0.5778 to convert to Btu/h · ft · °F.

Figure 9: The Thermal Conductivities of Some materials

TABLE 7.5 (Continued)

Geometry	Correlation Equation			Restrictions
Packed bed—heat transfer to or from containment wall, gas	$\overline{\mathrm{Nu}}_{D_p} = 2.58\,\mathrm{Re}_{D_p}^{1/3}\mathrm{Pr}^{1/3} + 0.094\,\mathrm{Re}_{D_p}^{0.8}\mathrm{Pr}^{0.4}$			$40 < \mathrm{Re}_{D_p} < 2000$ cylinderlike packing
	$\overline{\mathrm{Nu}}_{D_p} = 0.203\,\mathrm{Re}_{D_p}^{1/3}\mathrm{Pr}^{1/3} + 0.220\,\mathrm{Re}_{D_p}^{0.8}\mathrm{Pr}^{0.4}$			$40 < \mathrm{Re}_{D_p} < 2000$ spherelike packing
Tube bundle in cross-flow (see Figs. 7.21 and 7.22)	$\overline{\mathrm{Nu}}_D\mathrm{Pr}^{-0.36}(\mathrm{Pr}/\mathrm{Pr}_s)^{-0.25} = C(S_T/S_L)^n\,\mathrm{Re}_D^m$			
	C	*m*	*n*	
	0.8	0.4	0	$10 < \mathrm{Re}_D < 100$, in-line
	0.9	0.4	0	$10 < \mathrm{Re}_D < 100$, staggered
	0.27	0.63	0	$1000 < \mathrm{Re}_D < 2 \times 10^5$, in-line $S_T/S_L \geq 0.7$
	0.35	0.60	0.2	$1000 < \mathrm{Re}_D < 2 \times 10^5$, staggered $S_T/S_L < 2$
	0.40	0.60	0	$1000 < \mathrm{Re}_D < 2 \times 10^5$, staggered $S_T/S_L \geq 2$
	0.021	0.84	0	$\mathrm{Re}_D > 2 \times 10^5$, in-line
	0.022	0.84	0	$\mathrm{Re}_D > 2 \times 10^5$, staggered $\mathrm{Pr} > 1$
	$\overline{\mathrm{Nu}}_D = 0.019\,\mathrm{Re}_D^{0.84}$			$\mathrm{Re}_D > 2 \times 10^5$, staggered $\mathrm{Pr} = 0.7$
Flow over staggered tube bundle, gas or liquid ($\mathrm{Pr} > 0.5$)	$\overline{\mathrm{Nu}}_D = 0.0131\,\mathrm{Re}_D^{0.883}\mathrm{Pr}^{0.36}$			$4.5 \times 10^5 < \mathrm{Re}_D < 7 \times 10^6$ $S_T/D = 2$, $S_L/D = 1.4$
Liquid metals	$\overline{\mathrm{Nu}}_D = 4.03 + 0.228(\mathrm{Re}_D\mathrm{Pr})^{2/3}$			$2 \times 10^4 < \mathrm{Re}_D < 8 \times 10^4$, staggered

For in-line tubes in the laminar flow range $10 < \mathrm{Re}_D < 100$,

$$\overline{\mathrm{Nu}}_D = 0.8\ \mathrm{Re}_D^{0.4}\mathrm{Pr}^{0.36}\left(\frac{\mathrm{Pr}}{\mathrm{Pr}_s}\right)^{0.25} \tag{7.27}$$

and for staggered tubes in the laminar flow range $10 < \mathrm{Re}_D < 100$,

$$\overline{\mathrm{Nu}}_D = 0.9\ \mathrm{Re}_D^{0.4}\mathrm{Pr}^{0.36}\left(\frac{\mathrm{Pr}}{\mathrm{Pr}_s}\right)^{0.25} \tag{7.28}$$

Chen and Wung [32] validated Eqs. (7.27) and (7.28) using a numerical solution for $50 < \mathrm{Re}_D < 1000$.

Figure 10: Heat transfer correlations for external flow, Frank, Raj, Mark [15].

Technical Specifications

Table of Technical Parameters (SI)

HSC(130/68)- HSB(120/68)-			99H2	165H2	198H2	265H2	331H2	413H2	496H2	579H2	661H2	744H2	827H2	992H2	1157H2	1323H2	1488H2
Cooling capacity		KW	350	580	700	930	1160	1450	1740	2040	2330	2620	2910	3490	4070	4650	5230
		10^4kcal/h	30	50	60	80	100	125	150	175	200	225	250	300	350	400	450
		USRt	99	165	198	265	331	413	496	579	661	744	827	992	1157	1323	1488
Chilled capacity	Inlet/outlet temp.	℃	12→7														
	Flow	m^3/H	60	100	120	160	200	250	300	350	400	450	500	600	700	800	900
	Pressure drop	mH_2O	12	12	12	11	7	7	8	8	10	10	13	13	15	12	13
	Piping dia. (DN)	mm	100	125	125	150	150	200	200	200	250	250	250	300	300	350	350
Cooling water	Inlet/outlet temp.	℃	32→38														
	Flow	m^3/h	114	189	227	303	378	473	567	662	756	851	945	1134	1323	1512	1701
	Pressure drop	mH_2O	8	8	8	6	8	8	10	10	12	12	10	10	13	14	15
	Piping dia. (DN)	mm	125	150	200	200	250	250	300	300	300	350	350	400	450	450	450
Hot water	Outlet temp.	℃	68														
	Consumption: Inlet temp. 130℃	ton/h	6.1	10.2	12.2	16.3	20.4	25.5	30.6	35.7	40.8	45.9	51	61.2	71.4	81.6	91.8
	Consumption: Inlet temp. 120℃	ton/h	7.3	12.2	14.6	19.4	24.3	30.4	36.5	42.5	48.6	54.7	60.8	72.9	85.1	97.2	109.4
	Pressure drop	mH_2O	10	10	10	11	11	11	13	13	11	11	14	14	13	14	13
	Piping dia.(DN)	mm	40	50	50	65	80	80	80	80	100	100	100	125	125	150	150
Electrical data	Power supply		3Φ-380VAC-50Hz														
	Total current	A	21	21	24	25.8	27.1	28.2	28.8	32.3	33.3	36	38.7	40.5	44.2	47.1	51.6
	Electric power	KW	5.15	5.15	5.95	6.35	6.85	7.25	7.65	8.05	8.65	9.85	10.25	11.45	12.35	14.85	17.35
Overall dimensions	Length	mm	4150	4144	4242	4610	5070	5190	5595	5760	6147	6270	7110	7160	7860	8722	9542
	Width	mm	1950	2023	2086	2170	2275	2492	2430	2632	2700	2856	2912	3226	3268	3146	3176
	Height	mm	2690	2698	2852	2913	2857	3167	3295	3480	3654	3852	3852	4090	4225	4350	4350
Shipping weight		t	6.8	8.4	9.3	11.4	15.3	17.4	19.5	22.2	25.9	30.8	34.6	40.6	45.8	52.5	60.6
Operating weight		t	8.3	10.9	12.2	15.0	19.8	23.1	26.2	30	34.7	40.7	45.2	53.4	59.6	66.8	75.8

Figure 11: technical parameters for 661H2

5 Hot Water Operated Two Stage Lithium Bromide Absorption Chiller

About "Hot Water Operated Two Stage Unit"

Figure12 absorption chiller

TABLE 13–2

Representative fouling factors (thermal resistance due to fouling for a unit surface area)

(*Source:* Tubular Exchange Manufacturers Association.)

Fluid	R_f, $m^2 \cdot °C/W$
Distilled water, sea water, river water, boiler feedwater:	
Below 50°C	0.0001
Above 50°C	0.0002
Fuel oil	0.0009
Steam (oil-free)	0.0001
Refrigerants (liquid)	0.0002
Refrigerants (vapor)	0.0004
Alcohol vapors	0.0001
Air	0.0004

Figure 13: Representative fouling factors (thermal resistance due to fouling for a unit surface area).

TABLE 6.3 Heat transfer correlations for liquids and gases in incompressible flow through tubes and pipes

Name (reference)	Formula[a]	Conditions	Equation
Dittus-Boelter [35]	$\overline{Nu}_D = 0.23 Re_D^{0.8} Pr^n$ $n \begin{cases} = 0.4 \text{ for heating} \\ = 0.3 \text{ for cooling} \end{cases}$	$0.5 < Pr < 120$ $6000 < Re_D < 10^7$	(6.60)
Sieder-Tate [16]	$\overline{Nu}_D = 0.027 Re_D^{0.8} Pr^{0.3} \left(\frac{\mu_b}{\mu_s} \right)^{0.14}$	$6000 < Re_D < 10^7$ $0.7 < Pr < 10^4$	(6.61)
Petukhov-Popov [36]	$\overline{Nu}_D = \frac{(f/8) Re_D Pr}{K_1 + K_2 (f/8)^{1/2} (Pr^{2/3} - 1)}$ where $f = (1.82 \log_{10} Re_D - 1.64)^{-2}$ $K_1 = 1 + 3.4f$ $K_2 = 11.7 + \frac{1.8}{Pr^{1/3}}$	$0.5 < Pr < 2000$ $10^4 < Re_D < 5 \times 10^6$	(6.63)
Sleicher-Rouse [37]	$\overline{Nu}_D = 5 + 0.015 Re_D^a Pr_s^b$ where $a = 0.88 - \frac{0.24}{4 + Pr_s}$ $b = 1/3 + 0.5 e^{-0.6 Pr_s}$	$0.1 < Pr < 10^5$ $10^4 < Re_D < 10^6$	(6.64)

Figure 14: Heat transfer correlations for liquids and gases in incompressible flow through tubes and pipes, Frank , Raj ,Mark [16]

Matlab code:

```
clear all
clc
n=0;
for S=100:700
    n=n+1;
qu=0.84*(2.5-0.65)*9*(S-(6.7296/11.93)*(150-30));
A(n)=S;
B(n)=qu;
end
plot(A,B)
title('Relation Between"Absorber Solar Energy S(kw/m^2) & Useful Gain
(kw) "')
xlabel('Absorber Solar Energy S(kw/m^2)')
ylabel('Useful Gain (kw)')
figure
clear al,
clc
m=0;
for D=0.1:0.05:1
    m=m+1;
```

```
qu=0.84*(2.5-D)*9*(263.47-(6.7296/11.93)*(150-30));
A(m)=D;
B(m)=qu;
end
plot(A,B)
title('Relation Between"Diameter Of Cover Collector (m)) & Useful Gain
(kw) "')
xlabel('Diameter Of Cover Collector (m)')
ylabel('Useful Gain (kw)')
figure
clear all
clc
R=0;
for W=1:0.25:10
    R=R+1;
qu=0.84*(W-0.65)*9*(263.47-(6.7296/11.93)*(150-30));
A(R)=W;
B(R)=qu;
end
plot(A,B)
title('Relation Between"Width of Cover Collector (m)) & Useful Gain (kw)
"')
xlabel('Width of Cover Collector (m)')
ylabel('Useful Gain (kw)')
figure
clear all
clc
s=0;
for Tr=50:10:250
    s=s+1;
qu=0.84*(2.5-0.65)*9*(263.47-((6.7296/11.93)*(Tr-30)));
A(s)=Tr;
B(s)=qu;
end
plot(A,B)
title('Relation Between"Recive Surface Temperature(C)) & Useful Gain (kw)
"')
xlabel('Recive Surface Temperature(C)')
ylabel('Useful Gain (kw)')
figure
clear all
clc
s=0;
for Ta=10:5:55
    s=s+1;
qu=0.84*(2.5-0.65)*9*(263.47-((6.7296/11.93)*(150-Ta)));
A(s)=Ta;
B(s)=qu;
end
plot(A,B)
title('Relation Between"Ambient Temperature(C)) & Useful Gain (kw) "')
xlabel('Ambient Temperature(C)')
ylabel('Useful Gain (kw)')
figure
clear all
clc
s=0;
```

```
for L=1:0.5:30
    s=s+1;
qu=0.84*(2.5-0.65)*L*(263.47-((6.7296/11.93)*(150-30)));
A(s)=L;
B(s)=qu;
end
plot(A,B)
title('Relation Between"Length Of Collector (m)) & Useful Gain (kw) "')
xlabel('Length Of Collector (m)')
ylabel('Useful Gain (kw)')
figure
clear all
clc
s=0;
for IR=100:700
    s=s+1;
S=(IR*0.85*0.93*0.78)+(IR*0.78*0.65/(2.5-0.65));
A(s)=IR;
B(s)=S;
end
plot(A,B)
title('Relation Between"Average Solar Radiation (kj/m^2.day)) & Absorber
Solar Energy (kw/m^2) "')
xlabel('Average Solar Radiation (kj/m^2.day)')
ylabel('Absorber Solar Energy (kw/m^2)')
figure
clear all
clc
s=0;
for D=0.1:0.05:1
    s=s+1;
S=(295.83*0.85*0.93*0.78)+(295.83*0.78*D./(2.5-D));
A(s)=D;
B(s)=S;
end
plot(A,B)
title('Relation Between"Diameter Of Cover Collector (m)) & Absorber Solar
Energy (kw/m^2) "')
xlabel('Diameter Of Cover Collector (m)')
ylabel('Absorber Solar Energy (kw/m^2)')
```