Diesel Engines and Solar Energy for Electric and Cooling Applications

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Vorwort

Diese vorliegende Arbeit entstand während meiner Tätigkeit als wissenschaftlicher Mitarbeiter am Institut für Energietechnik der Universität Hannover.

Mein besonderer Dank gilt Herrn Professor Dr.-Ing. M. Gietzelt, der die vorliegende Arbeit angeregt, betreut und letztlich kritisch begutachtet hat. Professor Gietzelt war für mich nicht nur ein Doktorvater sondern ein echter Vater, der mir unglaublich viel geholfen hat. Herrn Professor Dipl.-Ing. G. Gütschow and Frau PD Dr. A Luke danke ich für die Übernahme des Korreferates und Herrn Prof. Dr.-Ing K. Popp für die Übernehme des Vorsitzes der Prüfungskommission.

Des Weiteren möchte ich allen Kollegen und Freunden, die mir während der Fertigstellung dieser Dissertation mit Rat und Tat hilfreich zur Seite gestanden haben, meinen herzlichen Dank.

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Hannover, im 09 September 2004

Keywords

Diesel engines, Energy management, Absorption refrigeration, Solar collector

Abstract

The coupled production of the earth energies power and heat from the assigned primary energy is a measure for the reduction of the carbon dioxide emission state of the art of the power supply. This technology of the power- heat coupling can take place in central or decentralized systems. The power- heat -cooling coupling has been increasingly improved, where cold out of heat in absorption refrigeration machine are operated either in single-stage or in two- stage. The power- heat-cooling coupling participates in particular for the building air conditioning in subtropical or tropical countries. These systems are the basis of frequent decentralized application as energy transformation system of the diesel engine and also the fuel cell in the future.

It offers at the same time to produce heat in solar thermal systems (solar heat collectors) since thus the different cooling load demand can be compensated between day and night.

The engineer-scientific investigations must be both extended to the design and the performance of such plants. As evaluation criterion, primary energy saving (fuel consumption) and the working reliability (engine cooling water temperature) are consulted.

The diesel engines are characterized by characteristic values of the heat losses. The conception of the absorption refrigeration (single-stage and/or two-stage) leads to different characteristic diagrams, which affect on one hand the necessary cooling water temperatures and on the other hand the efficiency of the solar heat collectors. From this requirements, the thermal connection between cooling water heat exchanger , exhaust-gas heat exchanger and solar collectors should be studied.

For different ambient temperatures and solar radiation conditions, whereby significant cities in Egypt are consulted, the connection between fuel expenditure and collector surface is quantitatively occupied. By means of a modified power characteristic number as quotient from produced power and cooling the power feed- into the net and/or the power purchase takes-out the net in dependence of the demand. Management strategies for the engine cooling water system are developed, which serve at the same time as basis for the mass flow control system.

Schlagworte

Dieselmotoren, Energiemanagement, Absorptionkältemaschine, Solarkollektor

Kurzfassung

Die gekoppelte Erzeugung der Erdenergien Strom und Wärme aus der eingesetzten Primärenergie ist als Maßnahme zur Reduktion der Kohlendioxid-Emission Stand der Technik der Energieversorgung. Diese Technologie der Kraft-Wärme-Kopplung kann in zentralen oder dezentralen Systemen erfolgen. In zunehmendem Maße gewinnt in Erweiterung die Kraft-Wärme-Kälte-Kopplung an Bedeutung, bei der die Kälte aus Wärme in Absorptionskältemaschinen, die einstufig oder zweistufig betrieben werden, erzeugt wird. Die Kraft-Wärme-Kälte-Kopplung eröffnet sich insbesondere für die Gebäudeklimatisierung in subtropischen oder tropischen Ländern. Diesen Systemen liegen wegen der oftmals dezentralen Anwendung als Energiewandlungssystem der Dieselmotor und zukünftig auch die Brennstoffzelle zugrunde.

Zugleich bietet es sich an, Wärme in solarthermischen (Solarkollektoren) zu erzeugen, da somit der unterschiedliche Kältebedarf zwischen Tag und Nacht kompensiert werden kann.

Die ingenieurwissenschaftlichen Untersuchungen müssen sich sowohl auf die Auslegung als auch den Betrieb derartiger Anlagen erstrecken. Als Bewertungskriterium werden die Primärenergieeinsparung (Brennstoffverbrauch) und die Betriebssicherheit (Kühlwassertemperatur) herangezogen.

Die Dieselmotoren werden durch Kennwerte der Abwärmeströme charakterisiert. Die Konzeption der Absorptionkältemaschine (einstufig bzw. zweistufig) führt zu unterschiedlichen Kennfeldern, die sich einerseits auf die erforderlichen Kühlwassertemperaturen und andererseits auf die Effizienz der Solarkollektoren auswirken. Hieraus ergeben sich zugleich Anforderungen an die Verschaltung zwischen Kühlwasser, Abgaswärmetauscher und Solarkollektoren.

Für unterschiedliche Umgebungstemperaturen und solare Einstrahlungsverhältnisse, wobei signifikante Städte in Ägypten herangezogen werden, wird der Zusammenhang zwischen Brennstoffaufwand und Kollektorfläche quantitativ belegt. Mittels einer modifizierten Stromkennzahl als Quotient aus erzeugter elektrischer Energie und Kälte ergeben sich in Abhängigkeit des Bedarfs die Stromeinspeisung in das Netz bzw. der Strombezug aus dem Netz. Es werden Betriebsführungsstrategien für das Motorkühlsystem entwickelt, die zugleich als Basis für das Massenstrom-Regelsystem dienen.

Contents

Nomenclatures

A	Area
A/F	Air fuel ratio
ARU	Absorption refrigeration units
bmp	Brake mean effective pressure
bsfc	Brake specific fuel consumption
COP	Coefficient of performance
C_p	Specific heat at constant pressure
CRU	Compression refrigeration unit
CV	Calorific value of fuel
CW	Water specific heat
D	Difference
E	Radiation coefficient
I	Solar radiation
m	Mass
P	Power
PE	Primary energy
Q	Heat rate
Sc	Solar collector
S_c	Power-cooling ratio
SF	Solar to engine heat energy factor
S_h	Power-heating ratio
T	Temperature
α	Collector coefficient
η	Collector thermal efficiency

ρ	Density
δ	Part load
Δ	Difference
ω_c	Fuel utilization of cooling
ω_h	Fuel utilization of heating

Subscripts:

a	Ambient
ab	Absorber
ARM	Absorption Refrigeration Machine
b	Back
BHKW	Total heat and power
br	Brake
c	Cooling
CHP	Combined Heat and Power
co	Condenser
col	Cooling losses
coll	Collector
con	Convection
cus	Customer
CRM	Compression Refrigeration Machine
cw	Cooling water
d	Demand
de	Diesel engine
e	Evaporator

eff	Effective
el	Electrical
g	Generator
h	Heating
HB	Heating Boiler
in	Inlet
kw	Cooling water
lost	Losses
m	Mechanical
N	Night
o	Reference value
out	Outlet
PP	Power plant
ratio	Ratio
rad	radiation
ref	Reference
s	Supply
sol	Solar
tr	Transmission
th	Thermal
t-out	Take out
ven	Ventilation
w	Water

Chapter 1

Introduction

1 Objectives of the Research

Among various alternative energy sources to fossil fuels, a great advance has been achieved in the use of solar collector to convert solar energy to heat energy or cooling energy by using absorption refrigeration units. The coupled production of the earth energies power and heat from the assigned primary energy is measured for the reduction of the carbon dioxide emission state of the art of the power supply. This technology of the power- heat coupling can take place in central or decentralized systems. The power- heat -cooling coupling at meaning wins increasingly in extension, with which cold out of heat in absorption refrigeration machine, which are operated single-stage or in two- stage, is produced. The power- heat-cooling coupling participates in particular for the building air conditioning in subtropical or tropical countries. These systems are the basis because of often decentralized application as energy transformation system the diesel engine and in the future also the fuel cell.

Thermally driven absorption refrigeration machine (ARM) can contribute for power supply with cogeneration systems a better utilization in summer months. Clear ecological advantages are to be expected by the cogeneration of electric and heat in relation to other versions of the refrigeration supply. The supply temperature of (ARM) controls the type of this machines which varied from single –stage can not be used diesel engine cooling water heat reason for the high supply temperature range and two-stage can be used diesel engine cooling water cause of lower supply temperature range.

The air conditioning defined as the temperature control humidity control, air filtering, air purification ,air management and circulation. The end energy demand is varying from electric ,heating and cooling. The heating energy used in winter in Europe is very wide to use but in the middle east and Africa the weather in winter don't need to use heating energy. The large demand in this area is air conditioning in summer months to overcome the high temperature weather in this world area. Complete air conditioning provides automatic control of these conditions for both summer and winter. The important point to be discussed in the current study is the temperature control for summer which is the very important factor in the area of the high temperature range in the summer (Middle east and Africa). Therefore, the current study is concerned in this point the summer air conditioning. Temperature control for summer cooling conditions requires automatic control for the refrigeration system to mention the desired room temperatures. It is a very economical and ecological advantage to use the sun to provide this cooling load, but the problem is if the solar energy can deliver electric load or not. It is found many systems of supply electric heating and cooling loads, these system will be illustrated in the next chapter to through the light of these systems.

Using diesel engines, electric efficiency can be changed by changing the ambient temperatures referring to the designed engine performance. The engine electric efficiency should be decreased by increasing ambient temperature. Therefore, the engine electric efficiency should be changed by changing the ambient temperature. The fuel consumption rate comparison will be studied in the current work, which takes into consideration that both electric and cooling load must be the same for the selected cases mentioned before. Difference engine cooling water rates, engine loads, solar collector areas, solar radiations and refrigeration machines will be studied to produce a constant both cooling and electric load. The engineer-scientific investigations must extend both to the design and the performance of such plants. As valuation criterion primary energy saving and the working reliability (engine cooling water temperature) are consulted. Both engine and solar collector performance changed by changing the weather performance. Therefore, the change of the installation place of the plant causing the change of the performance of the plant. Temperature control is very important point, where the change of it produces a change of both engine and refrigeration units as illustrated in the current study.

In the current study, the cooling load was studied at four cases namely: Compression refrigeration units connected to electric power station, Compression refrigeration units connected to diesel engine power plant, Absorption refrigeration units connected to diesel engine power plant, Absorption refrigeration units connected to both diesel engine power plant and solar collector. It is common in practice to assume that diesel engines and electric power stations should behave in a fuel consumption similar. This assumption is justified only when the engines are operating as electric power unit only, therefore both exhaust gas losses and cooling water losses are not used . Generally, the interference between diesel engines and electric power stations are taken in our considerations. These interference of the different cases are systematically described and analyzed in the present work. The diesel engine performance to use cooling water heat and control the cooling water temperature range (inlet and outlet) to obtain the efficient performance of the engine.

All the possible arrangements of the named cases are grouped into two sections: compression and absorption refrigeration units. The possible compression refrigeration units may be classified in two categories: with and without diesel engines. These two categories are divided by an interference fuel consumption. The interference may be either partial, when only one the electric load is affected, or combined, when both electric and cooling load are constant.

The aim of the present work is to analyze and study the problem of the fuel consumption for the case of use absorption refrigeration units, at constant heat load and cooling load, as well as using solar energy to reduce fuel consumption . Engine parameters as well as solar parameters are considered. In addition, results will be shown when the interference between the use of engine losses and/or solar heat energy. The following information is specifically required:

Reference electric power station as a function of both electric and cooling load, Compression refrigeration unit coefficient of performance, Absorption refrigeration unit coefficient of performance and solar collector

In the current study , two cooling demand loads will be considered:

a- Night cooling load   b- Day cooling load

The connection between solar collector and diesel engine was studied in four cases: parallel, series –A, series –B and part –series to obtain the more efficient connection for both low solar collector area and the minimum fuel utilized. The analyses were considered the different parameters varying from solar radiation (500 and 1000 W/m^2) and the ambient temperature varied from (20°C to 40 °C). The more efficient connection ,temperature and solar radiation will be studied in the current study.

All the above questions will be answered in the upcoming chapters. The investigation was extend to include the variation of engine, collector and refrigeration units parameters. As stated in the introduction, only absorption refrigeration units have the advantages of using both engine losses and solar energy to achieve the cooling load demand.

Chapter 2

Supply Systems of Power, Cooling and Heating

Diffrent systems of power- heat units are often divided into classes: central energy systems and decentral heat – power systems. In many cases, particularly central power systems produce the electric loads, which are used for domestic application such as lighting, industry and cooling through compression refrigeration systems. The heating load in this case can be obtained by separated heating boiler. The main types of this reference systems are namely:

- Steam turbine power station
- Gas turbine power station
- Combined gas and steam power station
- Large engine power station
- Nuclear power station

The primary energy used in this large systems are coal, oil, natural gas and nuclear primary energy. The electrical efficiency this systems varies from 0.3 to 0.6. The high efficiency occurs with the case of combined gas and stem turbine power plant[1].

Figure 2.1 shows the different two types of operation to feed electric, cooling and heating loads. The cooling loads supplied in this case through the compression cooling unit using the electrical power. In case a) the heat supplied through a separate heat boiler with additional separate primary energy (PE_{HB}) but the electric and cooling loads can be obtained directly in the reference power station which used a primary energy of (PE_{PP}). On the other hand, when considering combined heat and power or cooling and power systems as shown in case b). Three primaries energy are considered (PE_{PP}),(PE_{CHP}) and (PE_{HB})[1,2].

Cogeneration systems are not limited to produce heat and power. Many systems have also been developed that deliver chilled water for cooling loads. Absorption refrigeration units are used in range of (+6 °C), and operate with water /Li Br. If temperature below about (6°C) are required, absorption based on ammonia / water are available. This machines allow cooling temperature of range down to (-40°C), which are often required in industry. The range and performance of difference stages of absorption cooling units will be discussed in chapter 4.

Figure 2.2 considers the combined heat and power systems which used absorption cooling machines (ARM) to produce the cooling load. The small size implies that power production reference systems power station can be considered in connection to the small combined heat and power (CHP) to take- out or feed –in depending on the demand loads.

Many decentralized (CHP) plants are furthermore equipped with one or more boilers. The main function of these boiler is to provide heat in the case of high heat load. The operation of boiler is determined only in the case of special heat loads.

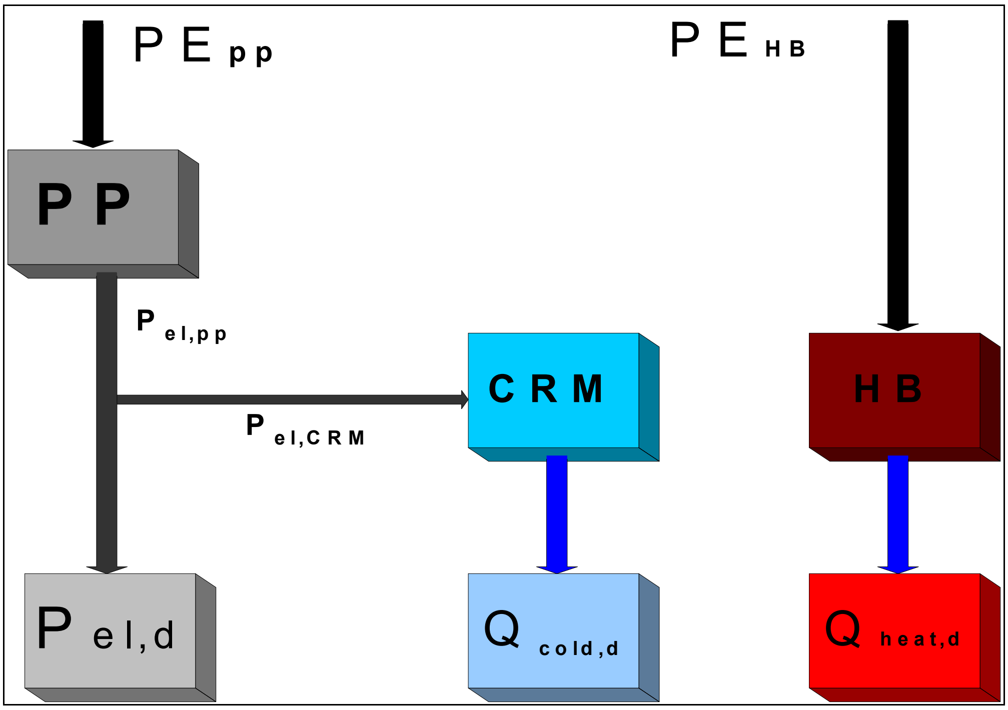
The many forms of total energy systems(BHKW) matrices are

- Engine +heat boiler
- Gas turbine +heat boiler

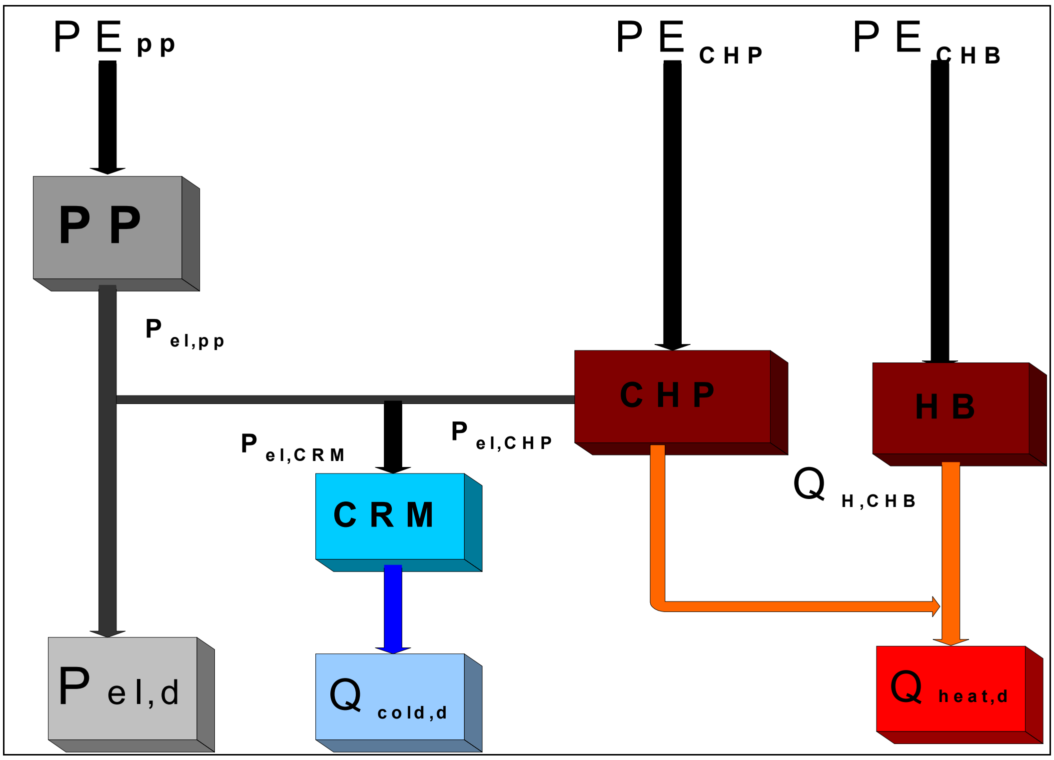
In the current study , it can be used a solar energy system to produce both heat and cooling. In this case to the last two cases can be added the following:

- Engine +Solar collector
- Engine +solar collector +heat boiler
- Heat boiler +solar collectors
- Gas turbine +solar collectors
- Gas turbine +solar collectors +heat boilers

The system will be considered in the current study is engine + solar collector + ARM to produce cooling load.

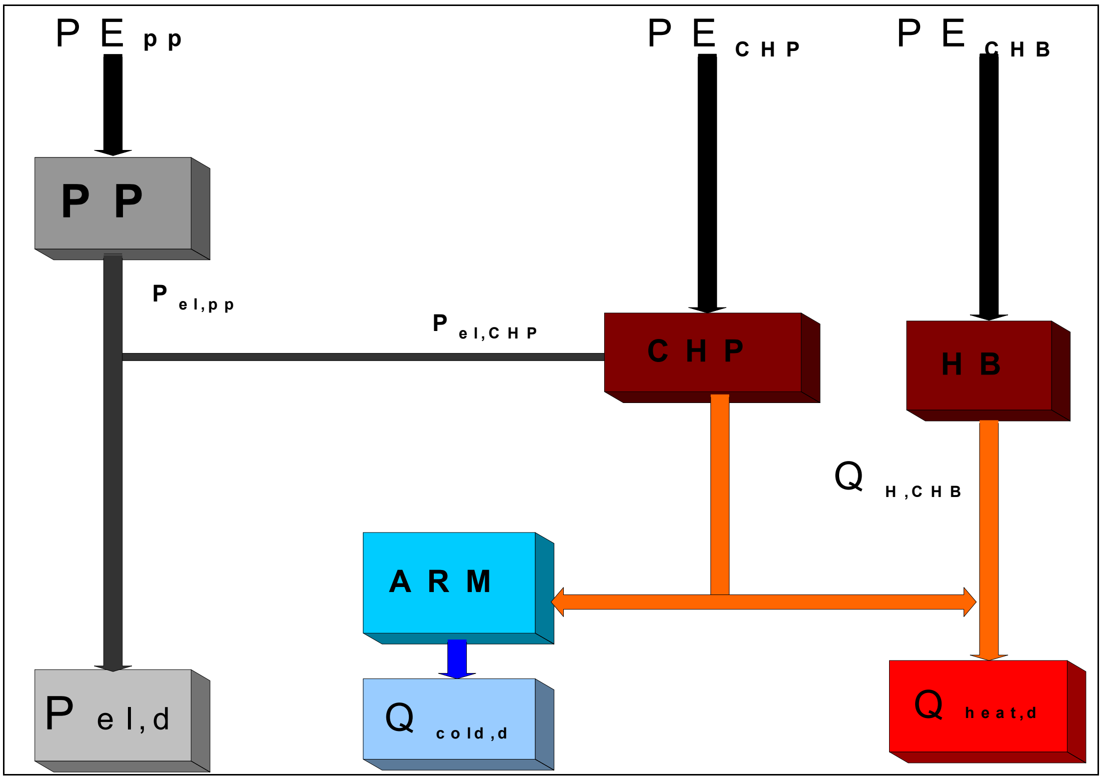


a)Reference varieties (Separate feed)

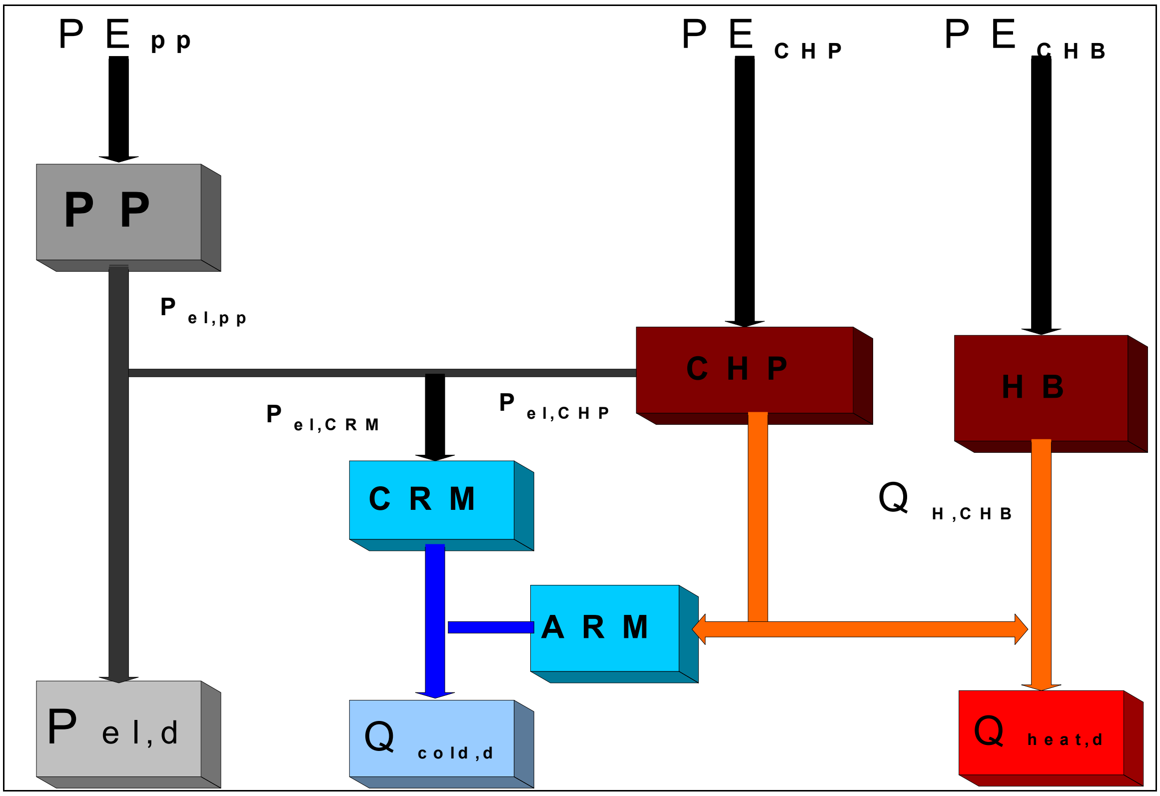


b)CHP +CRM

Figure2.1 Different Supply verities of electricity and thermal energy (cold and heat)



a) CHP +ARM



b)CHP+ARM+CRM

Figure2.2 Different Supply varities of electricity and thermal energy (cold and heat)

Chapter 3

Diesel engines

Diesel engines are one of the most efficient types of heat engines and are widely used as a prime mover for many applications such as marine diesel engine, marine generators, stationary diesel engines and in the transportations. From this application, it is find different forms in construction and maintenance. In the range of power more than 0.3 MW, marine and stationary diesel engines are close together in several parts.

3.1 Construction and performance

3.1.1 Marine diesel engines power plant

Basically, marine diesel engine can be divided into two main types. Large slow-speed direct driven main engines operating exclusively on the two –stroke cycle and being of crosshead construction. This category refers to engines operating at between 55 to 100 rpm.

Medium speed generally means between 400 and 1000 rpm, most of these are designed to operate on the four-stroke cycle and are of trunk piston construction. They are much lighter and smaller than equivalent slow-speed engines[4,5]. The disadvantage of use medium speed diesel engine is that the engine must use a reduction gear. That means the reduction of cargo space.

Figure 3.1 shows that the main production of diesel engines two and four stroke This figure summaries the production of diesel engines in the world from 1990 to 2000 [10-17]. **Figure3.2** shows the different connection between engine and propeller . The main difference between the marine and stationary diesel engine is that the marine diesel operates at variable speed depending on the load of the propeller and the type of this propeller such as fixed pitch propeller or controllable pitch propeller. In case of fixed pitch propeller[3], power produced by the engine under any condition must be at any time be equal to the power absorbed by the propeller. The controllable pitch propeller, in this case the change in the load changes the propeller pitch therefore the engine speed governor is acting on the fuel pumps that leads to the speed of rotation being constant.

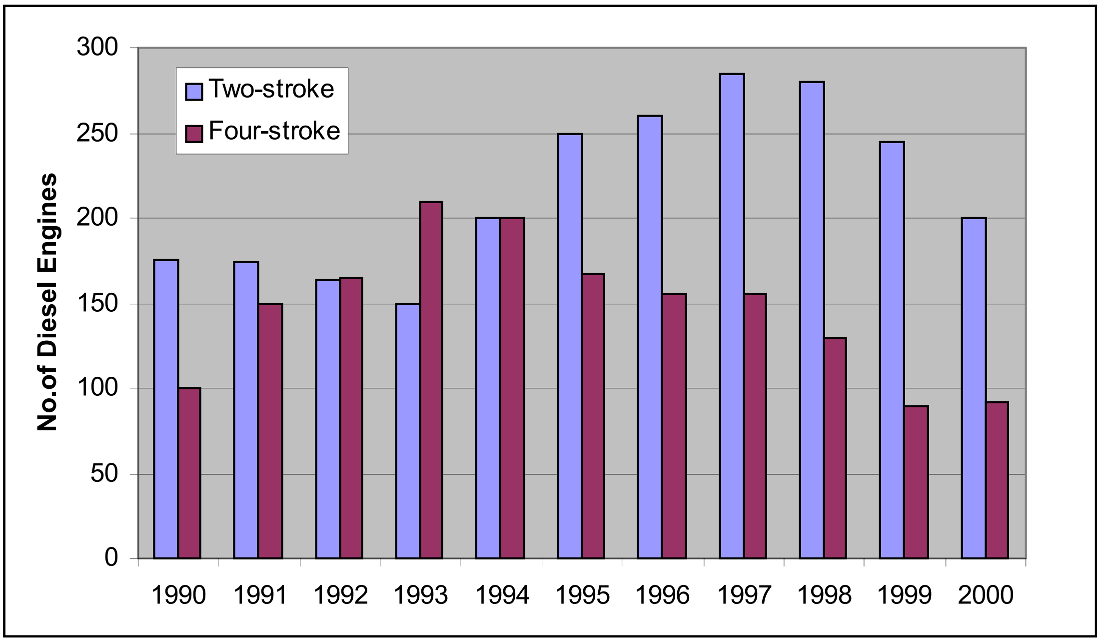


Figure 3.1 Two and four stroke marine diesel engines in the world

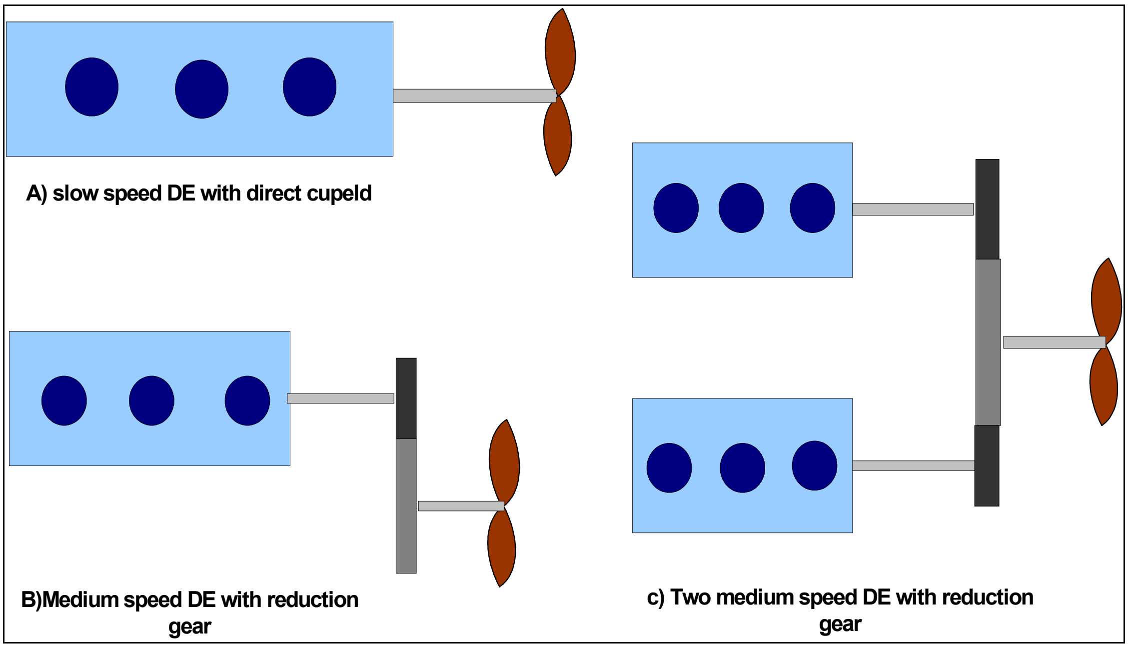


Figure 3.2 Different connections between engines and propeller

Marine diesel engines apply the same theory of the internal combustion engines but the application differs from other type of the engines reason for marine diesel engines has to couple with the marine use on board. The main features of the marine engines are the following[5]: Specific weight, engine speed, specific volume, operating at the sea, maintenance and construction, classification rules, starting systems and reversing systems.

Marine diesel engines of two-stroke or 4-stroke design are presently the most economical propulsion machinery because of their low fuel consumption rates compared with steam or gas turbine drives. The fuel oil will be illustrated in the next section.

The term fuel oil may conveniently cover a wide range of petroleum products, it may be applied to crude petroleum, to light petroleum fraction similar to kerosene or gas oil ,or heavy oil. Fuel remains one of the highest cost factors in running a ship and also the source of the most potent problems. Heavy fuels are the most used in the marine applications reason for the low cost of this type of fuel but with problems in this use.

The problems of the heavy fuel oil can be summarized as:

- Storage and handling
- Combustion quality and burn ability
- Contaminants, resulting in corrosion and/or damage to engine components.

The quality of a fuel oil is generally estimated by a number of specific parameters or proportions of metals or impurities in a given sample of the particular fuel. The most important parameters are the calorific value and viscosity.

The calorific value is measure of kJ/kg and the viscosity of an oil is measure of its resistance to flow which decreases with increasing of the temperature. Heating is necessary to the heavy fuels of high viscosity in current common use and ease their handling.The other properties of the oil fuel are Cetane number, Canradson carbon value, ash content, sulpher content, water content, cloud point, pour point, flash point and specific gravity [6,8].

Low, medium and high speed Diesel engines are designed to burn marine diesel oil while low speed and many medium speed engine can also accept commercial available heavy fuel oil with viscosity up to 700 cst. The fuel oil treatment system can be improved further by an automatic filtration unit installed as a monitoring/ safety device in the event of separator malfunction. The fuel oil heater may be of the tube or plate heat exchanger type [5,19].

The recent development of marine propulsion systems for merchant ships has been mainly influenced by the general energy situation and especially the costs, availability and quality of marine fuels. Subsequently the fuel cost went up to about from 30 to 50% of the total operating costs of merchant ships, in certain cases even higher. Due to this situation marine engineering research works had been intensified and concentrated on reducing the necessary propulsion power [21-22] and improving thermal efficiency [24].

The improving thermal efficiency of the marine diesel engine power plant can be done by the following:

- Improving the brake thermal efficiency
- Waste heat recovery systems

The improving of the brake thermal efficiency can be done by improving the engine performance such as improving the combustion efficiency, or increasing the compression pressure. In the recent researches [25], the brake thermal efficiency is improved by using long stroke engines. MAN-B&W diesel are now introducing a new generation of ship propulsion engines with electronic injection and electronic control of all main parameters. The new technology improves the fuel economy which improves thermal efficiency[26-27].

A considerable improvement of the overall diesel engine plant efficiency is achieved by waste heat recovery. With an exhaust gas boiler a relatively high percentage of the exhaust gas energy can be utilized for steam generation. Larger plants of this kind , as described for example by Geisler and Gietzelt [7,9,21,22,28,29] in many research works not only provide steam for heating purposes but also by means of turbo alternator ,the required electrical energy for the ship's operation.

The common utilized of the waste heat recovery can be done by:

1. Using part of the cooling water losses

The cooling water losses as mentioned before ranges between 20 to 30% of the total heat added ,thus it is recommended to use part of this heat can be used for: Fresh water heating, Air conditioning(use absorption refrigeration units), Fresh water generation, Heating of fuel and Domestic use of heating on board

2. Using the exhaust gas losses

The exhaust gas losses are mentioned before ranges between 25 to 35% of the total heat added

- Exhaust gas boiler with natural circulation

This type of boilers is very simple and is generally used to produce steam for domestic uses heating and services on board the ship, the most use of this boiler are the fire tube boilers

- Exhaust gas boiler with forced circulation

This type of boilers is more complicated with forced circulation. The use of such boiler is to produce steam not only for domestic use but also for power generation [18,20].

3.1.2 Electrical power generation on ships

To a great extent the total efficiency of a marine diesel engine plant also depends on the economical generation and distribution of the electric energy on board. The following different modes of power generation have therefore, been carefully investigated by [30]:

- Heavy fuel operated diesel generators
- Shaft generators
- Exhaust gas boilers with turbo generators

A special problem in calculating the power demand is the rating of the time variant electrical load resulting from discontinuous operation and load fluctuations of several units. So far the load share of this units has been considered .

The measured course of the electrical load, shown in **figure 3.3**, indicates the range of time variant power alterations on a container ship during different operation conditions. These measurments carried by Geisler in TUHH [9] .

The shaft generators connection can be shown in **figure 3.4**, from this figure can be summarized that the both cases a and b is used without reduction gears but the other two cases use a reduction gears which increases the cost of installations . The reason for using shaft generators is the economically point of view of the primary energy utilization. The classifications of ship do not accepte the shaft generator as a main generator, only as an auxiliary generators.

Figure 3.5 shows the exhaust gas boiler with turbo generator connected to the main engine. This figure illustrated that the electric load generated on ship board need a complete steam cycle which is not found in the stationary diesel electric power plant as mentioned in the next sections.

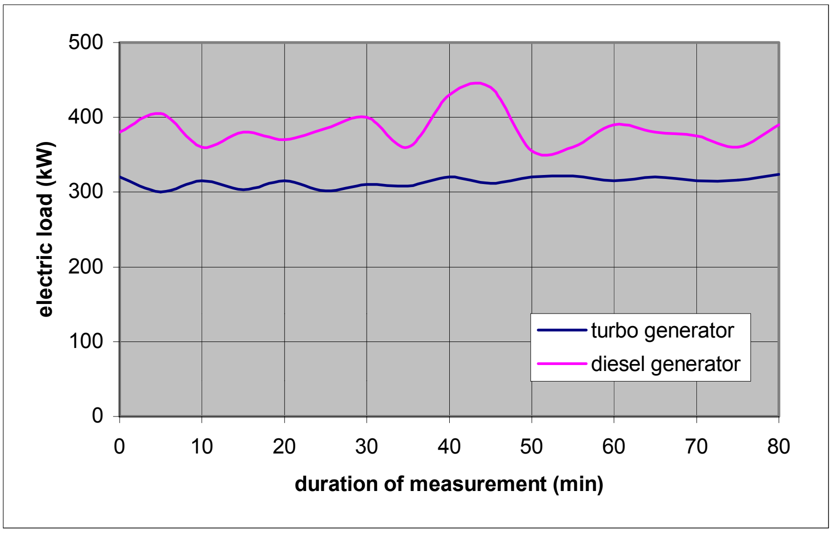


Figure 3.3 Measured time –variant electric load on a container ship

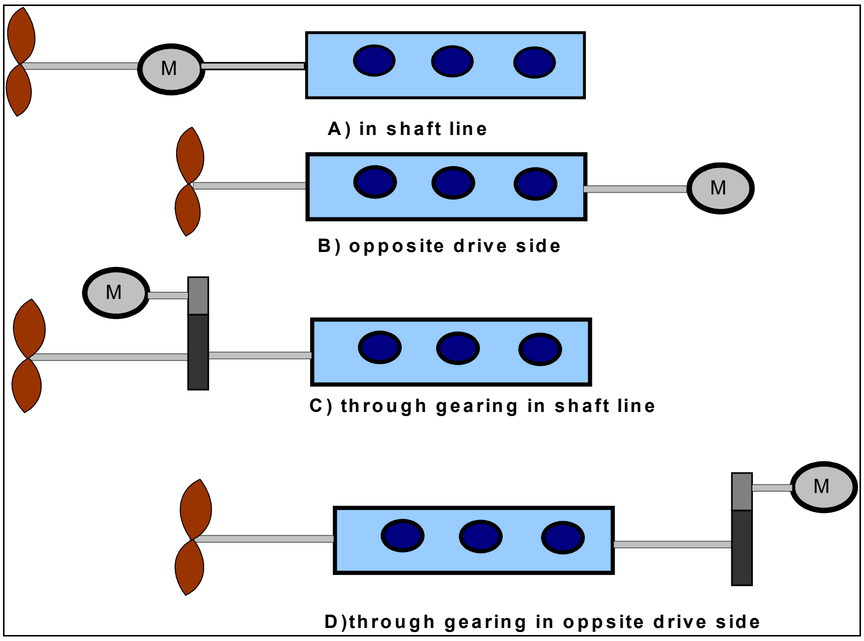


Figure3.4 Schema of shaft generators

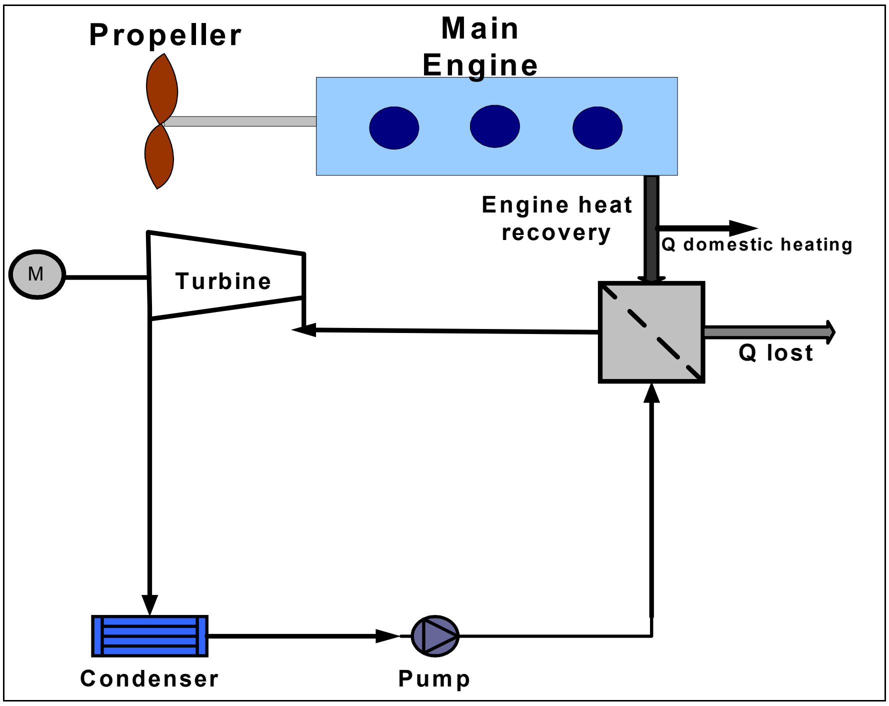


Figure3.5 Exhaust gas boiler with turbo generator

Figure 3.6 shows that the heat recovery system can be economical for power generation at about 15 MW main engine, where at power less than this range, the installation will not be economical. At this power range the generated power from the waste heat recovery system will be enough to cover the electric load requirements on the ship board. At main engine power more than 19 MW ,the turbo generator develops excess power, this power can be used for the propulsion of the ship.

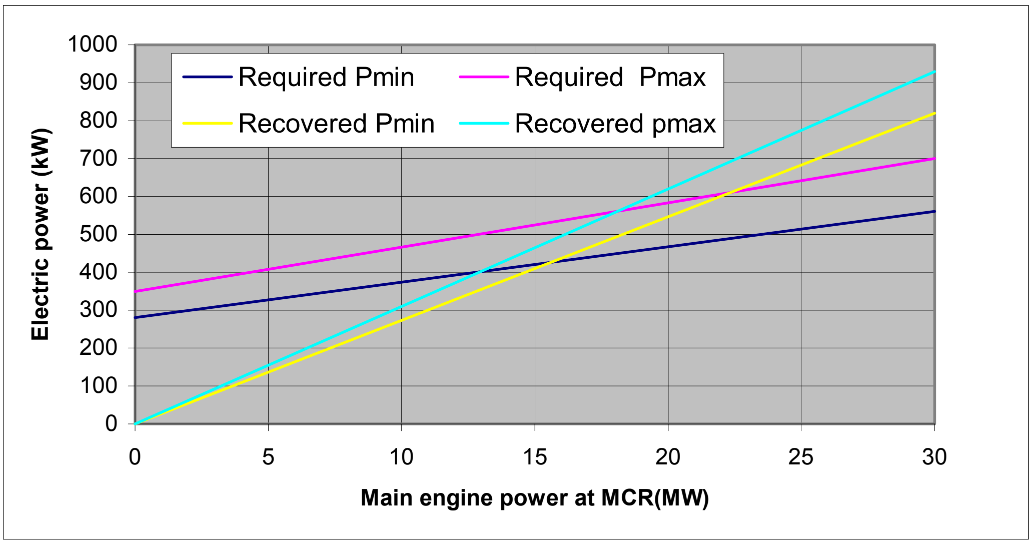


Figure 3.6 Electric power recovered from exhaust gas for different ships

The operating costs for electric power generation depend , however, on the actual fuel oil price. Thus the exhaust gas boiler/turbo generator system in use at a present certainly will not be as economical as some years ago. Furthermore in many cases today the available exhaust gas energy will not be sufficient for a single pressure exhaust gas boiler/generator system.

3.1.3 Stationary diesel engines power plant

In conventional power station only electric power is produced. The electric efficiency of approximate 45% was obtained. The reminder waste heat is transferred to the environment. In contrast ,in combined heat and power station ,the waste heat developed to use during the generation of the electricity. Thus the overall efficiency increases up to 90% with total energy systems (BHKW). They offer themselves everywhere, the requirement of electricity and heat occurs at the same time.

An engine can operate with natural gas, liquid gas, diesel or rapes oil. Gas-diesel technology allows to medium speed engines to be operated on heavy fuel and diesel oils, crude oil directly from the well or natural gas[5,31]. Supply can be switched automatically from one fuel to another with out shutdown. Conversions of existing engines from normal heavy fuel mode to natural gas/diesel oil operation can be executed with small modifications[36,37].

Under certain conditions small combined heat and power can contribute to energy saving and CO_2 reduction[32]. **Figure 3.7** shows that the comparison between the gas and diesel engine. From this figure it can be notified that diesel engine has electric efficiency more than gas engine but the thermal efficiency of gas engine is more. **Figure 3.8** illustrates both thermal and electrical efficiency of many engines of different loads[33-35].

The main features of the stationary electric generators which is generally medium speed diesel engines and four stroke is the constant speed operation. Referring to the generator number of pairs of poles , it can be found that the electric motor must run at 50, 25, 12.5 and 6.25 HZ in the power net work in 50 HZ systems.

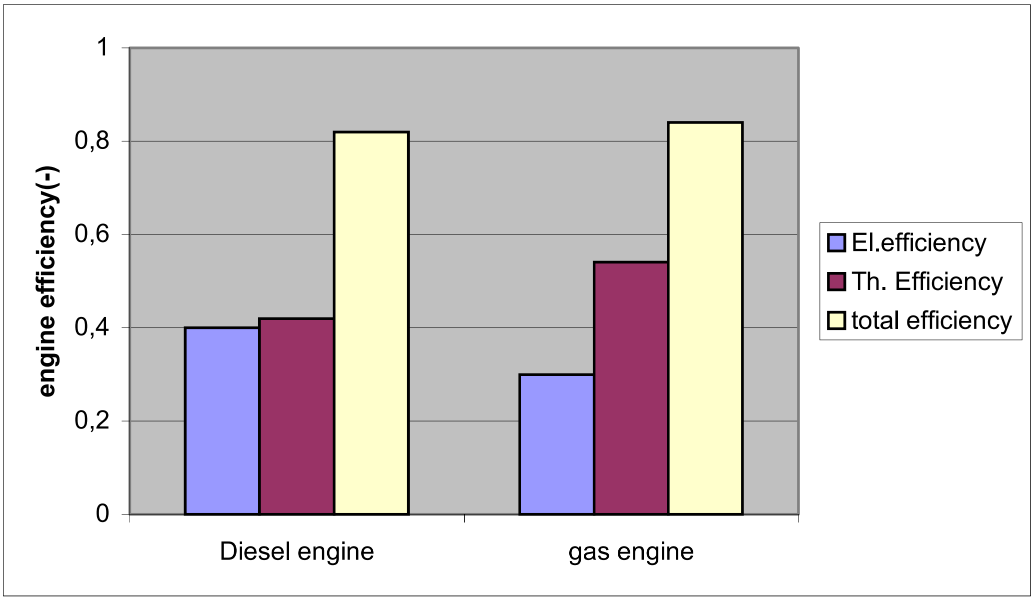


Figure 3.7 Efficiency comparison between diesel and gas engines (BHKW)

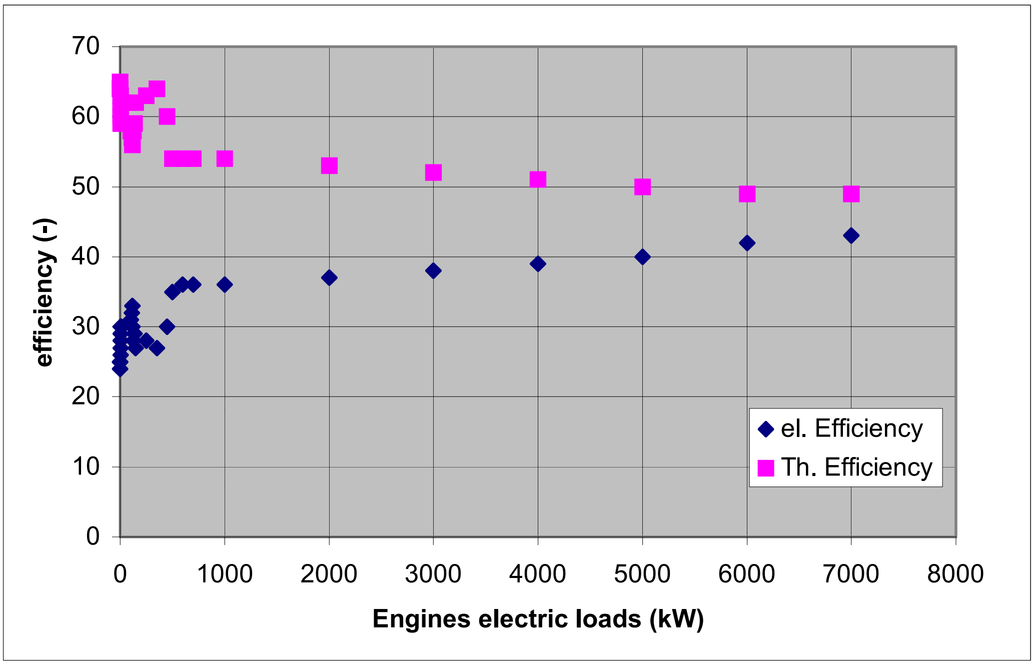


Figure 3.8 Efficiency of engines total energy systems (BHKW)

3.2 Energy system efficiency

3.2.1 Heat balance

The power obtained from an engine is most frequently called brake power(Pbr) and shaft power, delivered power or mechanical power. The total power actually delivered on the pistons in the engine is called indicated power (Pin). The relation between the both powers can be written as:

$$P_{in} = P_{br} + P_{fr} \tag{3.1}$$

The relation between the total delivered power and brake power can be written as:

$$\eta_m = \frac{P_{br}}{P_{in}} \tag{3.2}$$

The electric power P_{el} as a function of fuel consumption can be also written as:

$$\eta_{el} = \frac{P_{el}}{m_f * CV} \tag{3.3}$$

Therefore ,the electric power can be written as a function of heat added as:

$$P_{el} = CV * m_f - Q_{lost} \tag{3.4}$$

The heat lost (Q_{lost})is a function of both cooling water and exhaust gas [38,39].

To increase the engine thermal efficiency, both cooling and exhaust losses should be used in the power plant to produce heat or cooling load as investigated in chapter 5.

The relationship between heat added (Q_{add}) and the engine losses can be illustrated from **figure 3.9-a.** From this figure, it can be notified that the Sankey diagram of an engine which gives the useful electric load and the both effective exhaust and cooling energy.

The total thermal energy of an engine is around 82% as illustrated in the previous sections in this chapter, The difference between added energy and the thermal energy is the lost energy which is illustrated in Sankey diagram as Q_{lost}.

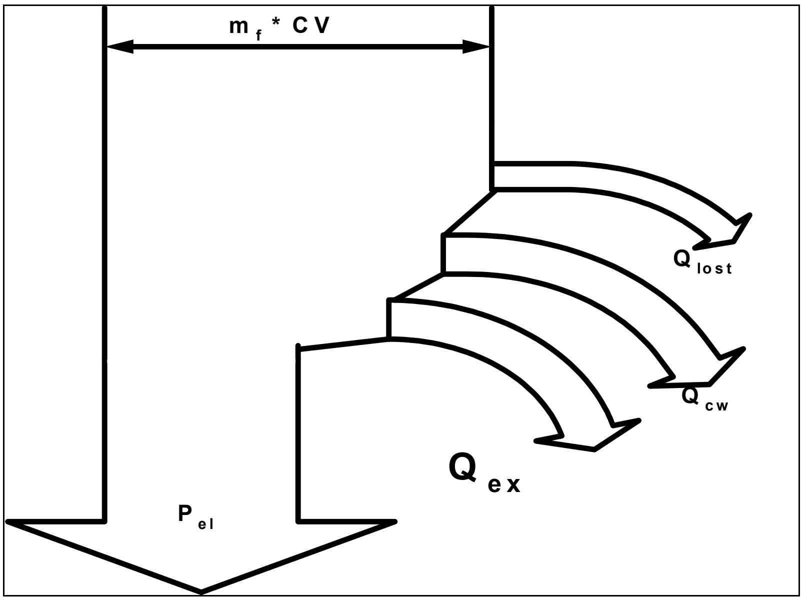


Figure 3.9-a Sankey diagram of diesel engine

A heat balance is a necessary tool in the design of a marine power plant. Once the shaft horsepower, engine losses, combustion conditions and basic cycle will be established in the current study, a heat balance calculations can be made. From it, the various fuel, air, exhaust, feed and condensate flows can be determined and used in selecting a suitable machinery and equipment. Heat balance may also be prepared to determine optimum engine conditions and cycle design or to analyze the performance of a power plant on both trials and service.

Heat balances should be a vital part of the economic evaluation of the power plant, many variables such as fuels, loads and ambient conditions affect the calculation result.

The simple heat balance can be illustrated as:

$$\eta_{bth} = \frac{BHP}{CV * m_f} \tag{3.5}$$

The remaining heat lost is the function of both cooling and exhaust losses can be written as:

$$Q_{lost} = (1-\eta_{bth}) * m_f * CV = Q_{col} + Q_{ex} + Q_{rad} \tag{3.6}$$

The cooling water losses heat (Q_{col}) will be:

$$Q_{col} = m_{col} * cw * \Delta t_{col} \tag{3.7}$$

The exhaust losses heat (Q_{ex}) follows:

$$Q_{ex} = m_{ex} * c_p * \Delta t_{ex} \tag{3.8}$$

The radiation heat losses (Q_{rad}) can be taken as a ratio of heat added to diesel engine range of (2-5%). The heat balance depends on the type of diesel engine as well as the loading conditions. The exhaust gas losses depends on the loading factor as well as the excess air factor . In general the engine heat losses increase by reducing the loading of the engine as shown in the previous section engine performance [21-24].

The exhaust gas losses represent about 30%of the total heat added. To improve the performance of the diesel engine ,it should be use this gas losses. This gas losses weight is a function of both fuel oil and air weight .The temperature of exhaust gases depend on the engine type, it can be summarized this values as follows:

- High speed diesel engine 330-600 °C
- Medium speed diesel engine 300-400°C

The exhaust temperature of slow speed marine diesel engines are depend on the two or 4-stroke engine. In general the exhaust gasses are used in a total energy system, which have a high temperature content [40-42].

The low temperature rise of cooling water around 10 °C, make the use of cooling water to produce steam for power generation is negligible. The cooling water losses as mentioned in section 3.1.3ranges between 20 to 35 % of the total heat added. In marine diesel engines, it is difficult to use this energy losses to improve the system efficiency in the normal conditions, but in the stationary total energy system this energy used beside exhaust gas energy to operate absorption heat pump. In chapter 5,this model can be investigated to produce cooling out of engine heat losses[43-47].

As shown in **figure 3.9-b** the cooling losses effective energy is more than the exhaust effective energy but with low temperature rise . from this figure ,it can be also obtained the supply temperature of heating output from this engine (Ts),which used in chapter 5 to operate the absorption refrigeration unit at this conditions.

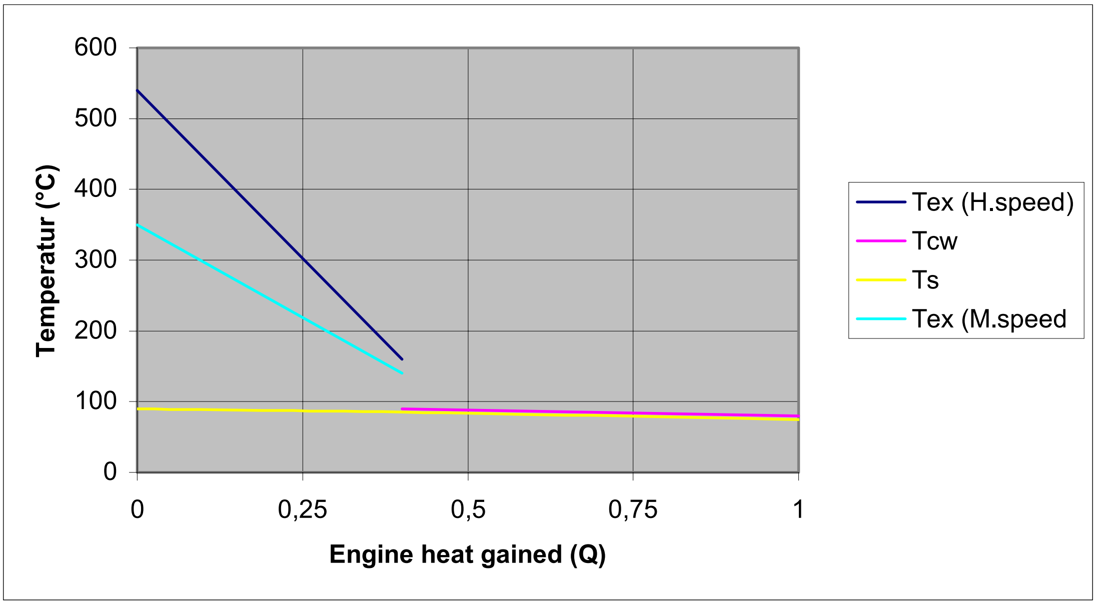


Figure 3.9-b Engine exhaust and cooling water temperature

In conventional power stations 10% of the fuel input are lost in combustion, 59 % in cooling towers or other cooling facilities,3% in transmission and only 28 % reach the consumer as electrical power. Overall efficiency of total energy plant can be up to 85%.

The electric power generated from diesel engine as mentioned before around 40 % of the heat added. The other 60% of the heat added is considered as waste heat in cooling and exhaust losses. The system efficiency is improved by utilized the engine waste heat recovery system.

In the total energy system (BHKW) both cooling and exhaust losses are used. This system is used to produce beside the electric load both heating and cooling by using absorption refrigeration units[48-49] .

The efficiency of an energy unit supply should be analyzed in each individual case. Cogeneration should not simultaneously result in the neglecting of economically prudent rationalization of final energy consumption[32].

3.2.2 Standardized ambient temperature effect

Using diesel engines, electric efficiency can be changed by changing the ambient temperatures referring to the designed engine performance. The engine electric efficiency should be decreased by increasing ambient temperature.

Figure 3.10 shows that the affect of increasing of ambient temperature from 20 to 40 °C on the standardized electric efficiency. It can be notified that the increase of the ambient temperature from 20 to 30°C decrease the standardized electric efficiency from 1.0 to 0.975 at the full load performance. The same effect can be obtain at the part load performance as shown in the mentioned figure.

Figure 3.11 shows the performance of the standardized electric load, exhaust temperature and exhaust gas weight at the study range of the ambient temperature. It can be from this figure illustrated that both standardized electric load and exhaust weight decreases from 1 to0.94 and 0.96 respectively by increasing the ambient temperature from 20 to 40 °C but the standardized exhaust gas temperature increases from 1 to1.04 by increasing the ambient temperature. Both engine and solar collector performance changed by changing the weather performance. Therefore, the change of the installation place of the plant causing the change of the performance of the plant. Temperature control is a very important point, where the change of the temperature produces a change of both engine and refrigeration units as illustrated in the current study. It should be taken in our consideration that the place of the installations unit where as affect on the performance. This factors will be study in chapter 5 with the total energy system with solar collector and refrigeration units.

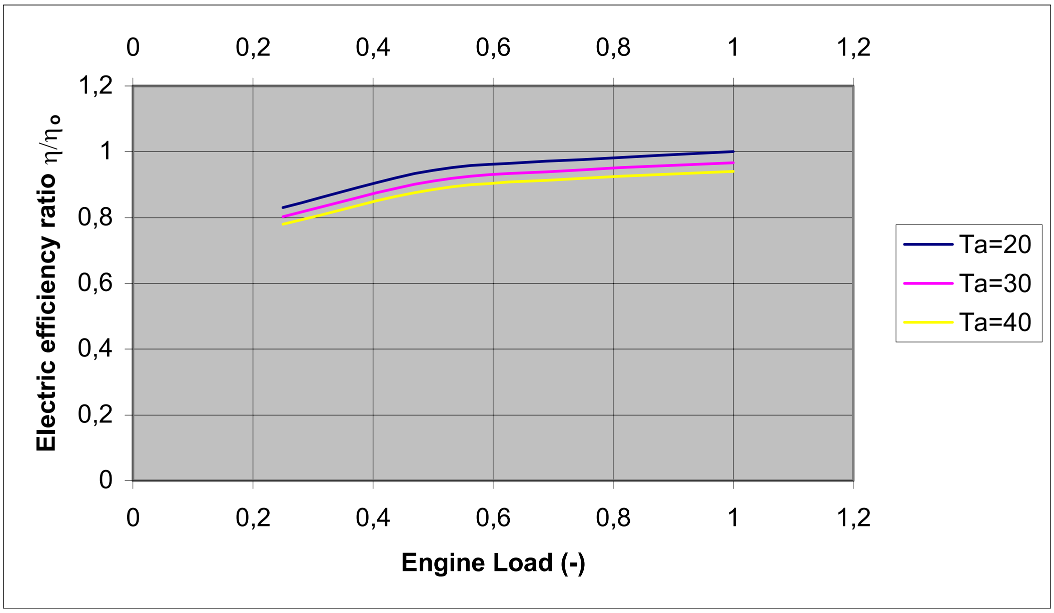


Figure 3.10 Standardized affect of ambient temperature on engine electric efficiency

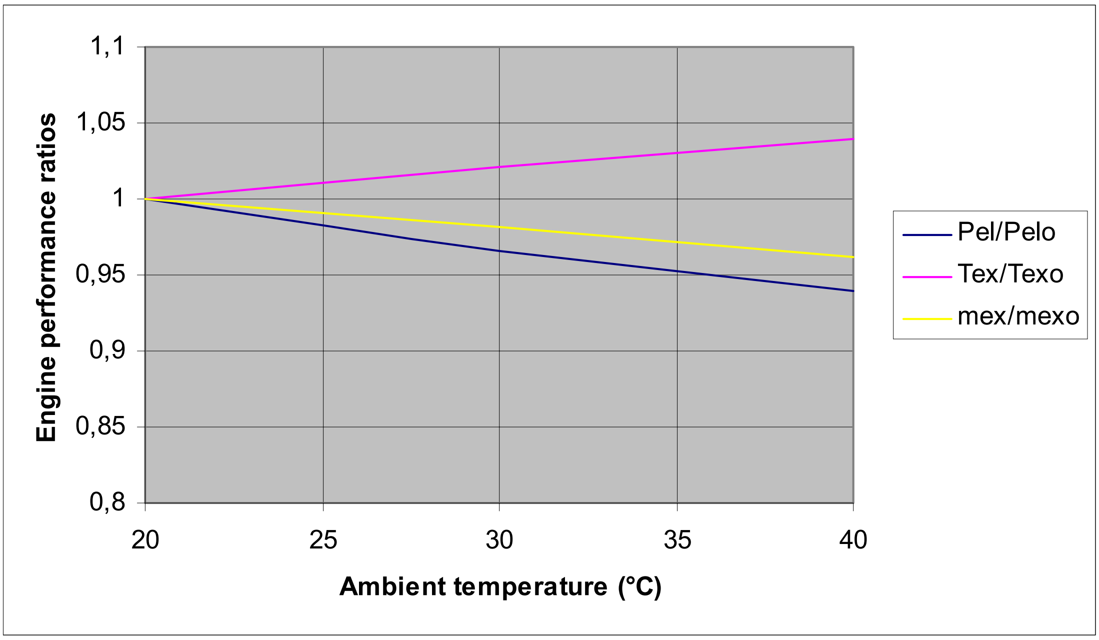


Figure 3.11 Standardized affect of ambient temperature on engine electric load, exhaust gas temperature and exhaust gas mass flow.

3.2.3 Part load performance

A two- stroke marine diesel engine(11MW) has been tested of the inflows scavenging and constant pressure supercharging type in Technical University of Hamburg.

Figure 3.12-a shows that some of the result of this engine performance. It can be notified from this figure that the engine rpm increases from82 to 110 by increasing the engine load from 50 to 100% load and remain constant by increasing the engine load to 110% of the maximum continuous rating load (MCR). Both the specific fuel consumption and the exhaust gas temperature have the same behavior. The specific fuel consumption reduces from 190 to 180 g / kW.h by increasing the load from 50 to 75 %MCR but increases from 180 to 190 by increasing the load to 110% MCR. The same result obtained for the exhaust gas temperature (290-280 and 320 °C). **Figure 3.12-b** shows the performance of medium speed diesel engines at constant speed of 750 rpm.

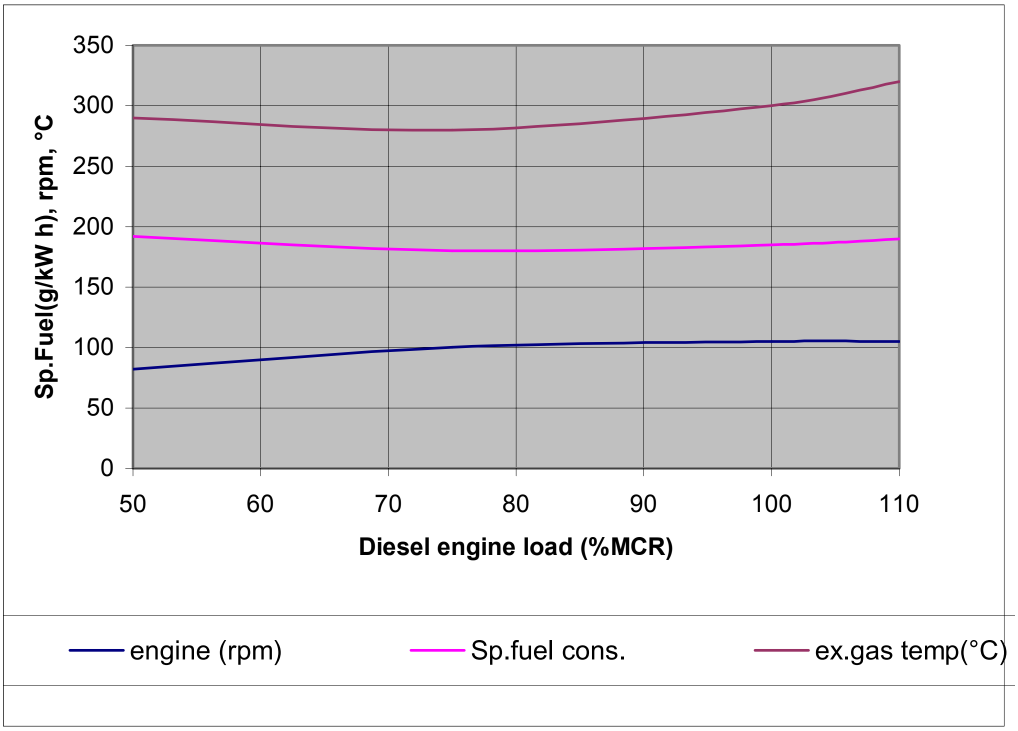


Figure 3.12-a Slow speed diesel engine performance at different loads

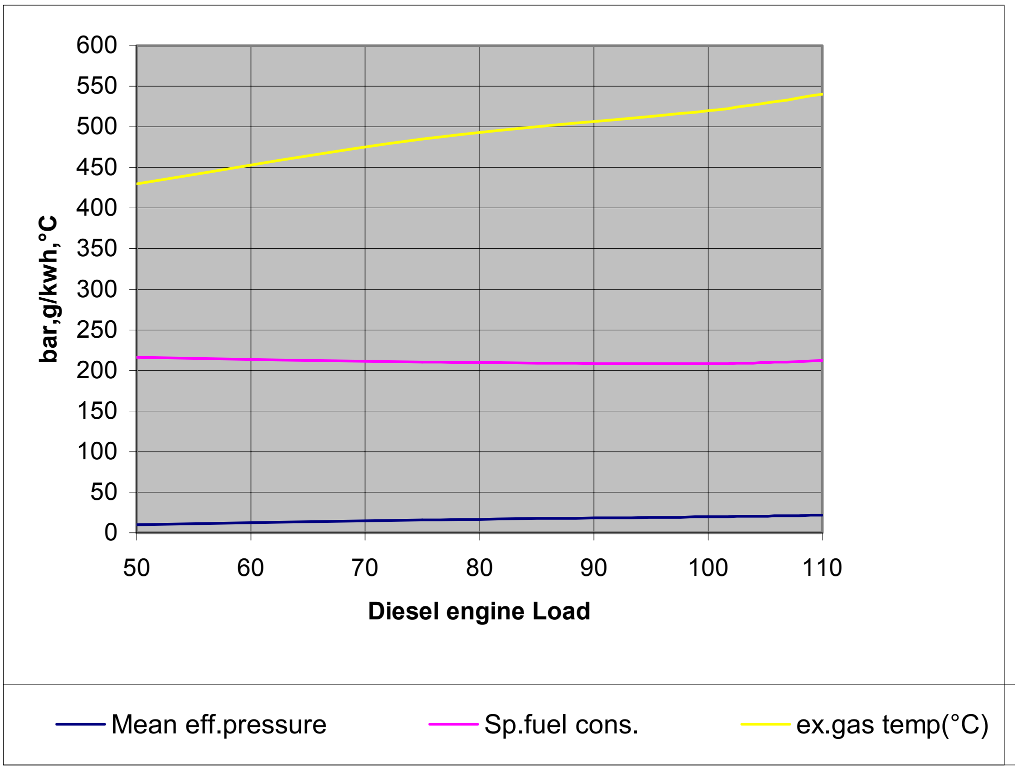


Figure3.12-b Medium speed diesel engine performance at constant rpm of 750

Figure 3.13 shows the diesel engine losses behavior at part loads . From this figure it can be notified that the engine losses decreases from 1.6 to 1.4 for the first engine of Q_{ex} to Q_{cw} of 1.4 and from o.9 to o.8 of the engine of ratio of losses 0.8. This two selected engines are illustrated in chapter 5 and analyses both electric and fuel consumption at different loads which are used to generat the cooling load with the use of absorption refrigeration units[9].

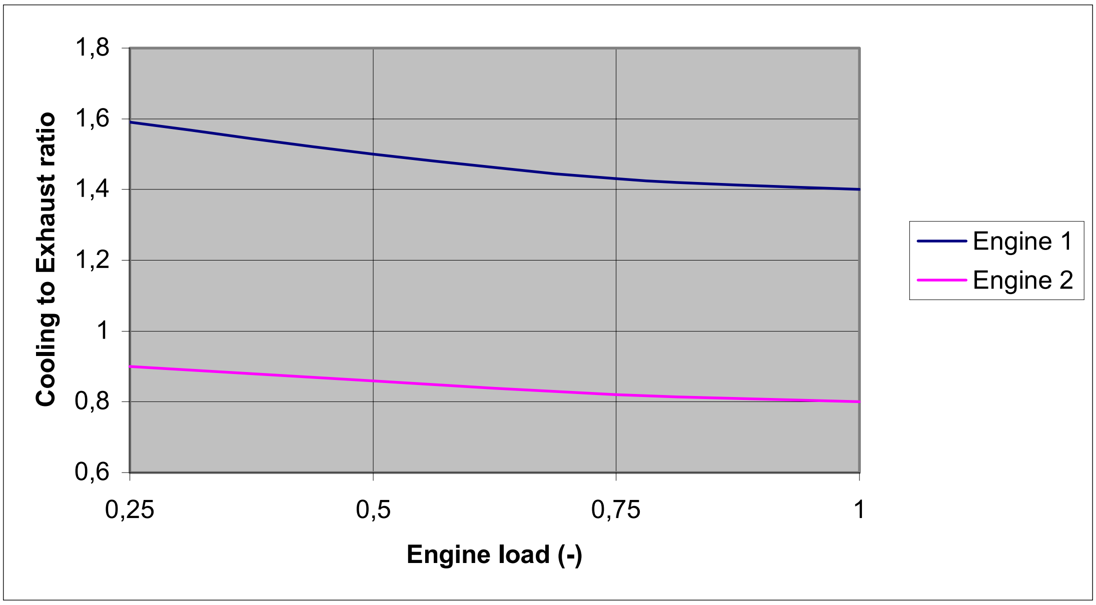


Figure 3.13 Diesel engines heat losses performance at part load

Chapter 4

Absorption Machine Simulation

4.1 Compression Cycle Performance

As illustrated in **figure 4.1**, the compression refrigeration cycle, the difference between the compression refrigeration cycles and the absorption refrigeration cycle, is the source of power. In compression systems, compressors activate the refrigerant by compressing it to the higher pressure and higher temperature level after it has produced its refrigeration effect. **Figure 4.2** show that the performance of actual compression cycle shown in figure 4.1, in this cycle it can be notified that the difference between the ambient temperature and the evaporation temperature. The same difference is between the demand and condensate temperature, the difference in two cases in the rang from 1 to 2 K.

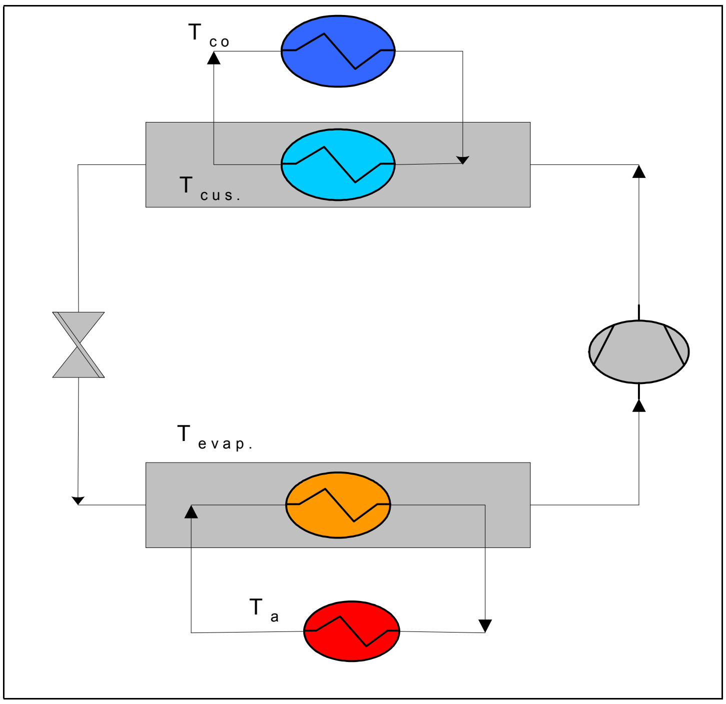


Figure 4.1 Actual compression cooling unit with variable of both source and sink Temperatures.

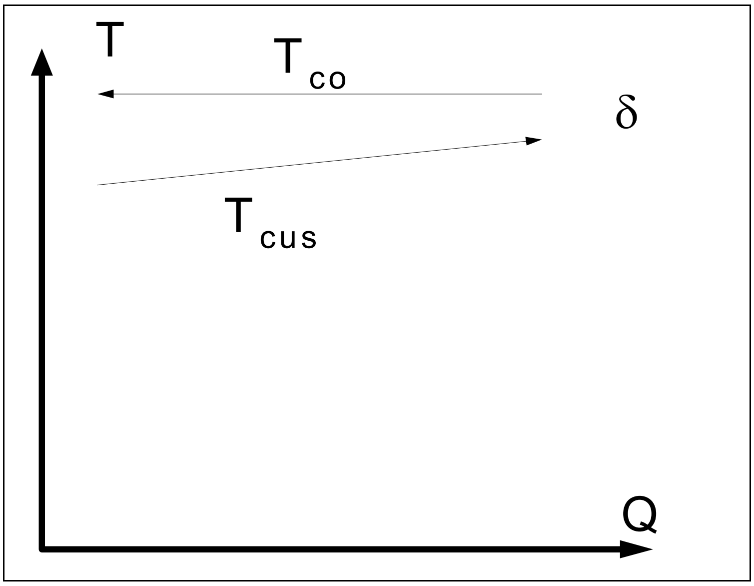


Figure 4.2-a Condenser (quantitative) T-Q diagram.

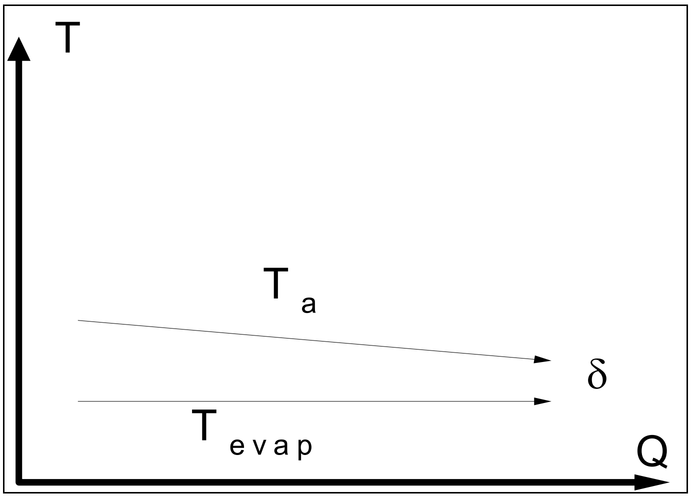


Figure 4.2-b Evaporator (quantitative) T-Q diagram.

Figure 4.3 shows that the effect of the evaporation temperature of the coefficient of performance at constant condensate temperature of 8 °C. It can be notified from this figure that the increase the evaporation temperature from 20 to 40 °C reduce the coefficient of performance from 6.3 to 4 but the increase of the temperature to 40 °C decrease the coefficient of performance only to 3.25. This reduction of the coefficient of performance will be take in our consideration in the comparison of the different systems arrangement in chapter 5 to obtain the optimum energy management of the selected power and cooling load

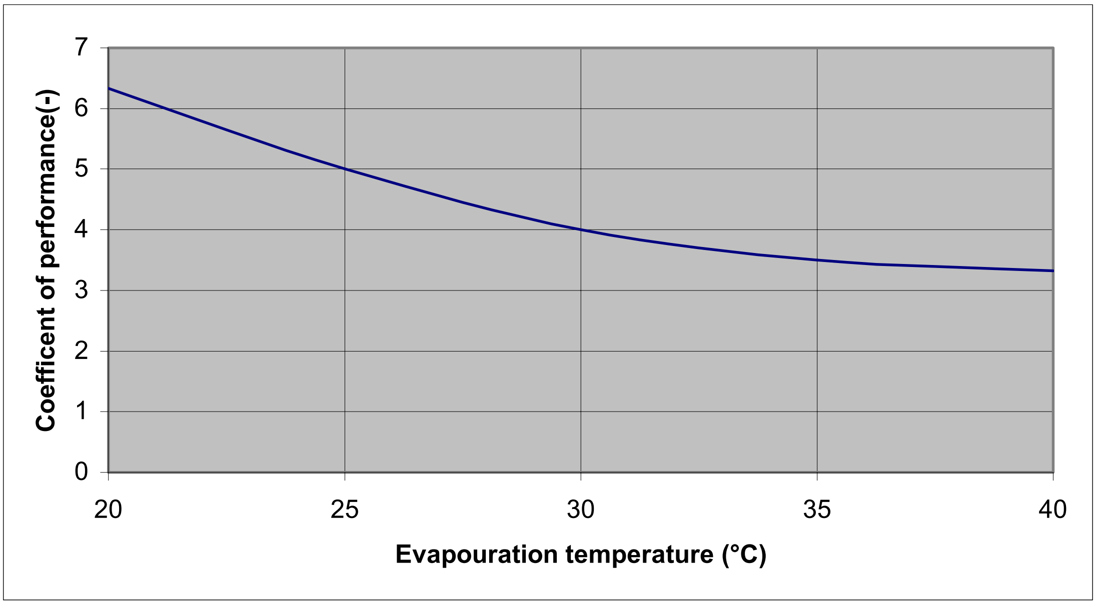


Figure 4.3 Coefficient of performance of compression cooling unit at constant condensate temperature of 8 °C

4.2 Absorption machine

4.2.1 Principle of operation

In the1950s and 1960s, both absorption chillers using steam as heat input to provide summer cooling and centrifugal chillers driven by electric motors were widely used in central refrigeration plants but the absorption process of producing cooling was first discovered by Faraday in 1824[71].

After the energy crisis in 1973,the price of the natural gas and oil used to fuel steam boilers drastically increased. The earliest single-stage indirect fired steam absorption chillers had a coefficient of performance of only 0.6 to 0.7.They required more energy and could not complete with electric centrifugal chillers. Many absorption chillers were replaced by centrifugal chillers in the late 1970s and 1980s. In recent years two- stage direct fired absorption chillers have been developed with a good coefficient of performance[72,73].

In many plants and buildings a ready supply of steam or hot flue gases are available as a by-product of some process or as used capacity of heating system which is inoperative in off-seasons. Absorption refrigeration systems have enjoyed renewed and growing in recent years due to their potential in heat pumping applications such as refrigeration, heating and air conditioning [77-80].

The essential difference between absorption and vapor -compression heat pump is that the absorption systems converts heat of a given temperature to heat of another temperature without any intermediate use of work as shown in **figure (4.1).** Absorption units used heat rather than mechanical energy to produce cooling. Heat sources can be a number of types including solar, natural gas, steam, hot water, or electrical energy [74,75,80].

4.2.2 Advantages of absorption refrigeration

The advantages of using absorption refrigeration units include the flowing:

1) **Environment :** Absorption units use refrigerants that have a zero ozone-depletion factor

2) **Power :** units use heat rather than electrical energy, and therefore can be added without increasing the power load which could be too expensive or impractical ,as in the present study in chapter fife.

3) **Service :** Absorption units have few moving parts only.

4) **Operating Cost :** In areas where the ratio of gas cost to electrical cost is favorable, units are purchased based on a lower operating cost.

In most areas, the larger sizes are more advantages than the smaller sizes. the characteristics that make absorption units less attractive are:

1) absorption units are heavier than electric units of the same capacity.

2) Absorption unit costs more than electric unit having the same capacity.

4.2.3 A single- stage absorption refrigeration unit

Constant cold water temperature of 6/12 °C are considered in many studies, such as the researches by FFI and institute of energy technology in Lower Saxony are carried by Baumann[82], Gietzelt [83-84] and Kruse [89-90]. Utesch [85] studied the use of natural gas performance with absorption cooling units and the advantages of use Li/Br pairs with block heat and power plants and Ewe [86] studied the total energy systems (BHKW) powered by absorption cooling units for the cooling of a sheet fed offset printing machine. Dittman [87-88] studied the performance, restriction and the chance to use the absorption cooling units.

The performance and design of small single effect absorption chiller units for district heating system are analyzed for the ambient temperature range only 29 to 32 °C, the maximum coefficient of performance of 0.56 [89,90].Sager [91] studied the integration of absorption refrigeration plants in district heating substations, but Margen [92] studied the developments of the concept of the production of cooling energy district heat driven absorption chillers in Sweden . Temperature control of district heating and cooling systems with absorption chiller units are studied by Saether [93].Greiter [94] studied a 500 kW absorption heat pump for heating at two temperature levels , Alefeld , Ziemann , Ahachad, Berkner and Salvako [95-99] studied the absorption heat transformer applications to absorption refrigeration machine.

Figure 4.4 shows that the single-stage absorption refrigeration system which consists of condenser, generator, evaporator and absorber. The high pressure level contains condenser and generator and the low pressure level contains evaporator and absorber.

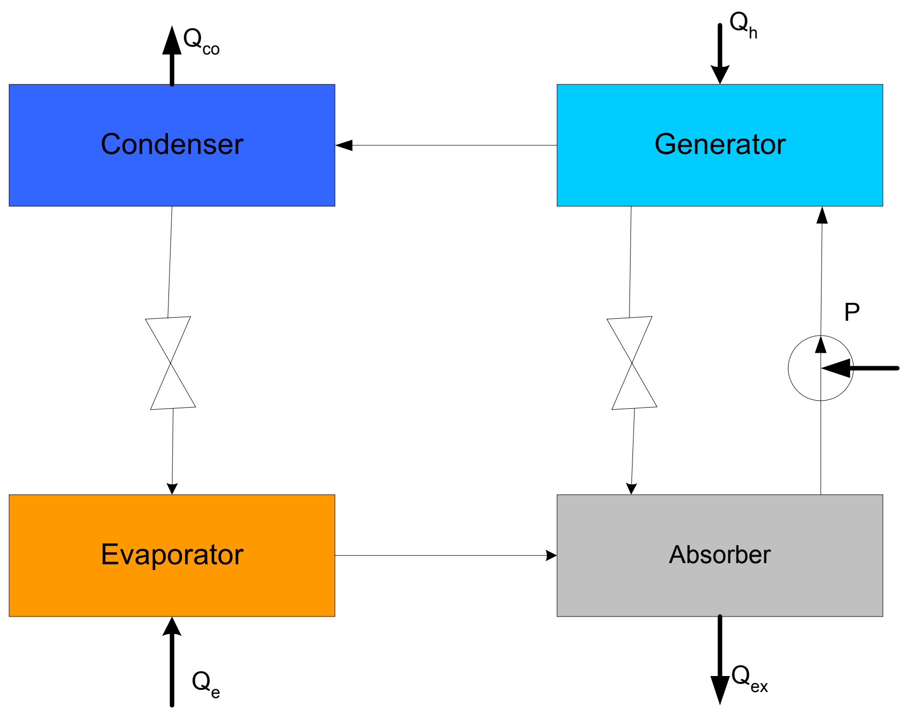


Figure 4.4 principle of single-stage absorption refrigeration machine

4.2.4 Two Stage Absorption Refrigeration Unit

Langel and Schweigler [100,101] studied cold out of heat with use the two-stage absorption refrigeration maschine. Xiaosng [102] studied the development of hest pump and Kahn [103] studied an ammonia-water absorption cycle with high temperature lift. Bassios and Mohamed [104,105] studied a 250 kW absorption pump with gas fired, thermal energy storage and co-generation, Schwenger [106] and Baksas [107] studied absorption heat pumps and high temperature absorption /compression heat pump for industrial waste heat. The use of ammonia water pairs are only in the case of single effect refrigerator. Ustsch [85] represented the advantages to use Lithium /Bromide in block heat and power system with single and double effect and illustrated the disadvantages to use Ammonia water by the higher effect stages absorption refrigeration systems.

In the current work, from this literature review, Lithium /Bromide water are the used pairs as a refrigerant in single and two stage absorption with the range of supply and return temperature of 8/12°C.

The use of two-stage absorption refrigeration unit is more efficient than a single-stage one ,while the possibility to use the lower supply temperature. **Figure (4.5)** shows the diagram of the two-stage cycle which includes an absorber, an evaporator, a condenser, high pressure generator, low pressure generator and solution pump.

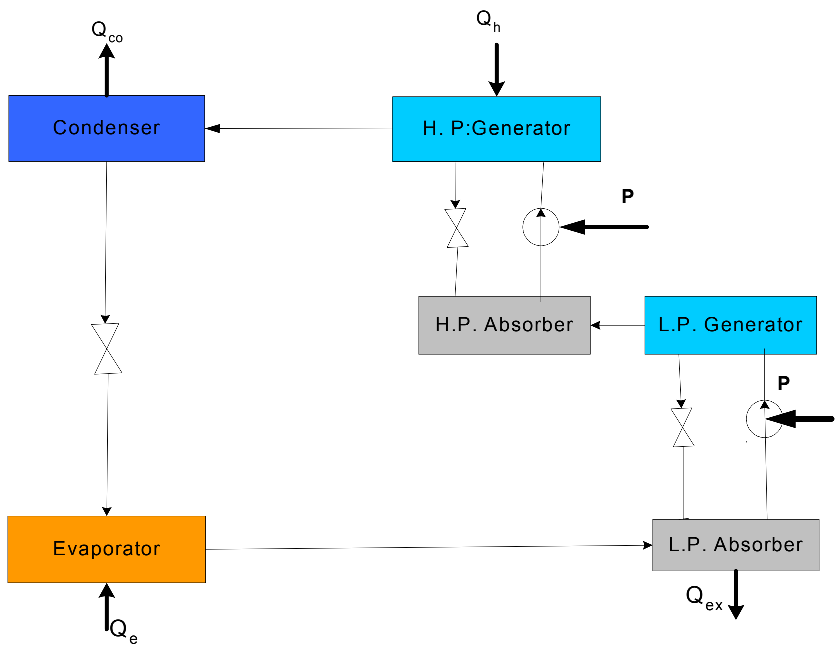


Figure 4.5 Principle of two-stage absorption refrigeration machine

4.2.5 Heat balance

Heat is supplied from a high –temperature heat source and from a low temperature heat source, and all this heat is released to a medium temperature heat sink. The heat flows of generalized absorption cycle are shown in **figure 4.5.**

As shown in **figure 4.6,** the top part of the unit is the driving cycle, where heat is supplied to the generator to boil off refrigerant that considered in the condenser. The main heat exchanges with the environment are :

- High temperature heat source (generator)
- Medium-temperature heat sink (absorber and condenser)
- Low temperature heat source (evaporator)

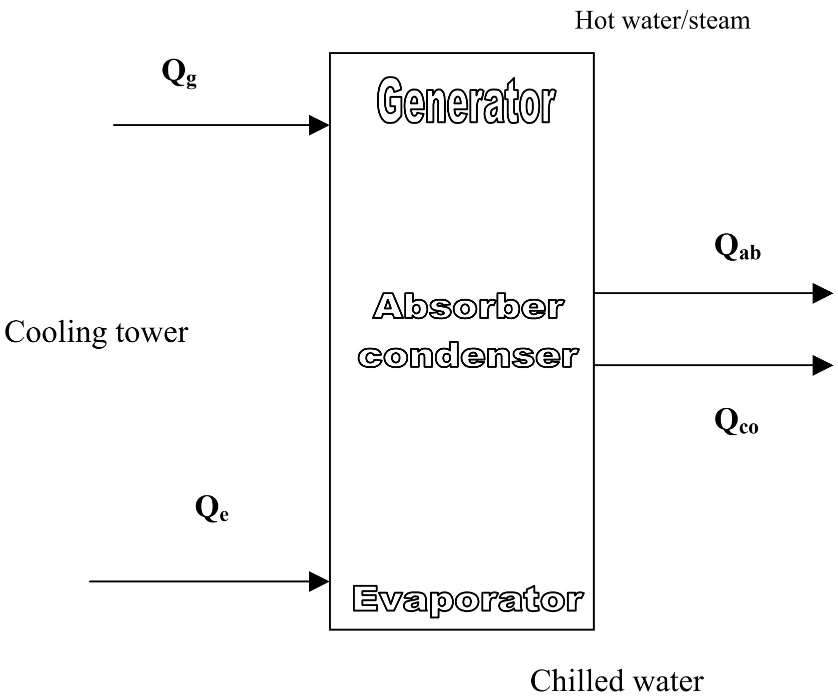


Figure 4.6 Heat flows of an absorption cycle

By applying the first law of thermodynamics, The heat balance is:

$$Q_e + Q_g = Q_{co} + Q_{ab} \tag{4.1}$$

If the second law of thermodynamics is applied, the total generation of entropy will be zero as in the case of ideal cycle:

$$\frac{Q_e}{T_e}+\frac{Q_g}{T_g}=\frac{Q_{co}}{T_{co}}+\frac{Q_{ab}}{T_{ab}} \tag{4.2}$$

By the following assumptions[76,109,110]are:

- Saturated liquid specific heat is negligible
- Heat of evaporator is constant
- Refrigerant expansion is isotropic; and
- Superheated vapor specific heat is negligible

Then the vapor heat will be equal to the condenser heat

$$Q_e = Q_{co} \tag{4.3}$$

from equations 4.1 and 4.3 the absorber heat will equal to the generator heat

$$Q_{ab} = Q_g \tag{4.4}$$

The coefficient of the performance can be written as:

$$COP = \frac{Q_e}{Q_g} \tag{4.5}$$

from the last three equations ,the coefficient of performance as a function of the four temperature can be written as[76]:

$$COP = \frac{\frac{1}{T_{ab}}-\frac{1}{T_g}}{\frac{1}{T_e}-\frac{1}{T_{co}}} \tag{4.6}$$

In many studies by Alefeld and others [75-80]have been to assume that the absorber and condenser have the same temperature, therefore the last equation can be simplified as:

$$COP = \frac{T_g - T_{ab}}{T_g} * \frac{T_e}{T_{ab} - T_e} \tag{4.7}$$

Therfore, it can be written the coficient of performance as:

$$COP = COP_{carnot} * \eta_{carnot} \tag{4.8}$$

From the previous equation, it can be concluded that the coefficient of performance of absorption refrigeration unit are dependent on Carnot efficiency and compression refrigeration cycle operated at the same levels of temperature, that means coefficient of performance of absorption should be less than compression refrigeration one.

4.2.6 Working Pairs in Absorption Refrigeration Units

The working fluid pairs to be used must be selected on the bases of the suitability under the conditions in question, e.g. its thermodynamic performance, chemical and thermal stability and toxicity[85,95].Examples of working fluid pairs and applications which have already been extensively developed are as follows:

a)Aqua- ammonia has found application in air conditioning and refrigeration, Ammonia is the cooling agent and water the solvent. The performance value of the process dependent on the temperature, with temperature level arrive to (- 45 °C). The main area of utilization of ammonia water was the large scale use in MW-area.

b)Aqua ammonia – hydrogen combination has found application domestic refrigerators

c)Aqua-LiBr solution has found application for domestic and commercial air-conditioning, a performance value of 0.65 (single stage) and 1.10(two stage) can be reached the required heating temperature lies between 90 and 115 °C the small temperature level arrive to (6 °C).

One of the two types of absorption chillers is normally applied, mostly depending on the required cooling temperature[76]:

a. for cooling temperature more than 5 °C, water/LiBr absorption machine is most frequently used, which must be water-cooled.

b. An ammonia /water machine can be used for cooling temperature less than 5 °C. It can be either air or water-cooled.

According to Perez. Blanco, the thermodynamic properties of a pair used in an absorption cooling unit are[76,81]:

- high latent heat of vaporization of refrigerant,
- low vapor pressure of the refrigerant but above atmospheric level,
- heat mixing or reaction is low or negative,
- low sensible heat capacity of absorbent solutions compared to the latent heat of the refrigerant,
- no crystallization or freezing with in the range of operating conditions,
- high solubility of refrigerant in the absorber .

The classification of absorption refrigerators and heat pump represented in [107] are shown in **table 4.1**, from this table, it can be illustrated that the mostly use of refrigerant pairs is water Lithium Bromide for both single and double effect

Classification		Cooling	Heating	H_2O LiBr	NH_3 H_2O
Refrigerator	Chiller	Single effect	___	●	●
		Double effect	___	●	
	Chiller/Heater	Single effect	Without Abs. cycle	●	●
		double effect	Without Abs. cycle	●	___
Heat pump	Chiller/Heater	double effect	Single effect	●	___
	Heat Amplifier	___	Single effect	●	___
		___	double effect	●	___
	HeatTransformer	___	Single effect	●	___

Table 4.1 Absorption machines for heating and cooling

The way energy is managed has major impact on environment. The combined heat and power plays an important role by promoting further development for combined heat and power systems. Sorption machines have gained increased interest in the recent decades. Sorption systems are not only viewed as energy efficient cooling and heating methods when installed in a properly systems but also as environmentally alternative to CFC,HFC or HCFC based technology. Sorption systems can use almost any heat source, including cogeneration systems power plants and solar energy systems [76,108]., as in the case of the current work use absorption refrigeration systems cogeneration with diesel power plant and solar collectors, will be illustrated in chapter 5.

4.2.7 Coefficient of performance

The coefficient of performance for a single –stage absorption refrigeration unit at various supply and ambient temperatures are shown in **figure 4.4.** In this figure **,**it can be noticed that the increasing of the ambient temperature decreasing the coefficient of performance and the same effect of the coefficient of performance with the supply temperatures.

Figure 4.7 shows the relation between the coefficient of performance and the chilled water supply temperature at various of the ambient temperatures. The range of the coefficient of performance in this type located between (0.25 and 0.5) lower than the coefficient of performance of a single-stage unit, but the advantages is the use of low supply temperature which is located between (55 and 100 °C).

Figure 4.8 shows that the relation between the temperature difference and the inlet temperature at various ambient temperature. The temperature difference in this case is greater than the case of a single-stage one which in the range of (18 to 50 k) at 20 °C and 40 °C ambient temperature respectively.

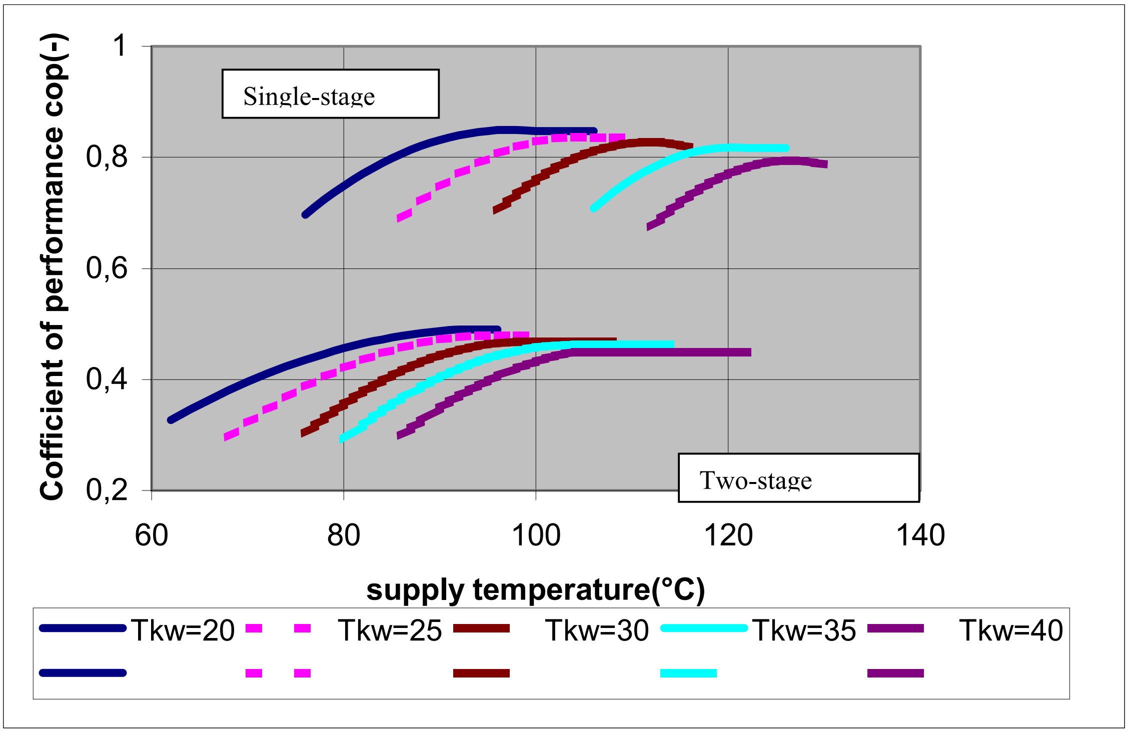


Figure4.7 Coefficient of performance of both single and two-stage absorption refrigeration machines

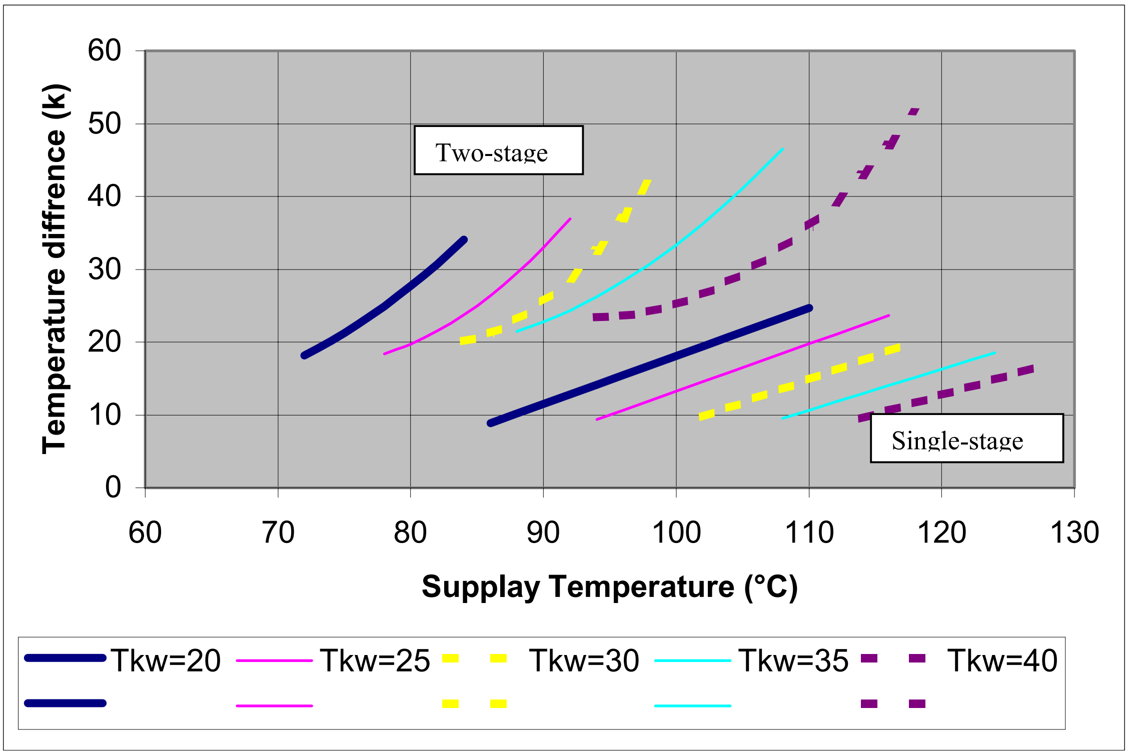


Figure 4.8 Temperature difference Comparison between single and two-stage absorption refrigeration machines

Chapter 5

Combined Power – Cold Plant - A Parametric Analysis

5.1 Reference system

It is aimed here to demonstrate the behavior of the compression cooling units connected to the electric power station. In order to compare the fuel utilized in the electric power stations results obtained in the next three cases . The cooling load was supplied with compression refrigeration unit, the electric load demand in this refrigeration unit, can be obtained from the stationary power station in the range of 150 to 300 MW.

The electric load delivered from the power station can be written as:

$$P_{el} = \eta_{pp} * m_f * CV \qquad (5.1)$$

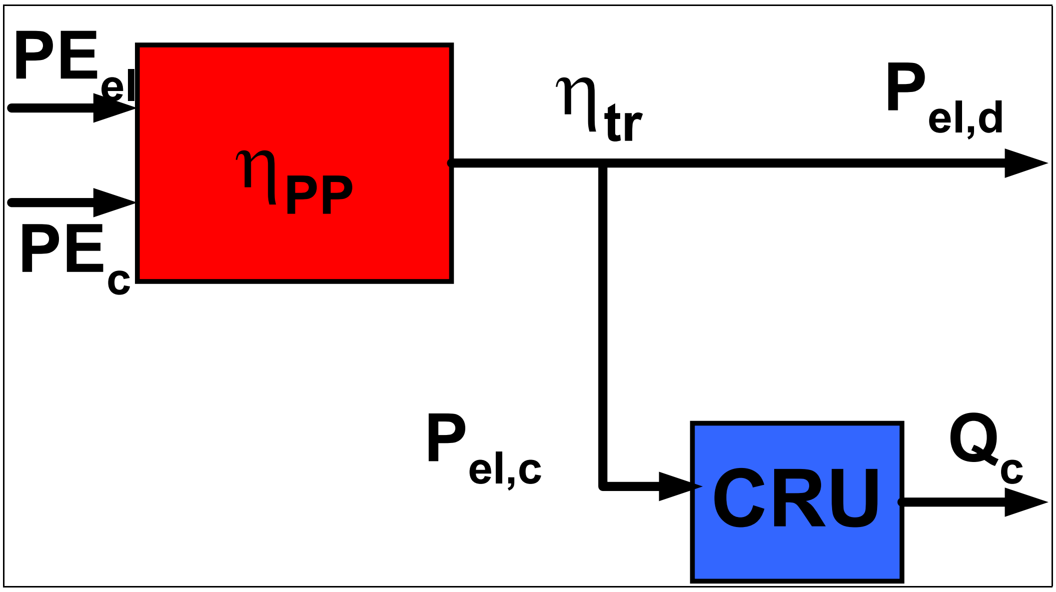


Figure 5.1 Primary energy supplied to both power station and compression refrigeration unit (CRU)

Figure 5.1 shows the electric and cooling load flow diagram delivered from the stationary power station and compression refrigeration unit. The demand electric load can be written as:

$$P_{el,d} = PE_{el} * \eta_{pp} * \eta_{tr} \tag{5.2}$$

Therefore, the electric primary energy can be written as:

$$PE_{el} = \frac{P_{el,d}}{\eta_{pp} * \eta_{tr}} \tag{5.3}$$

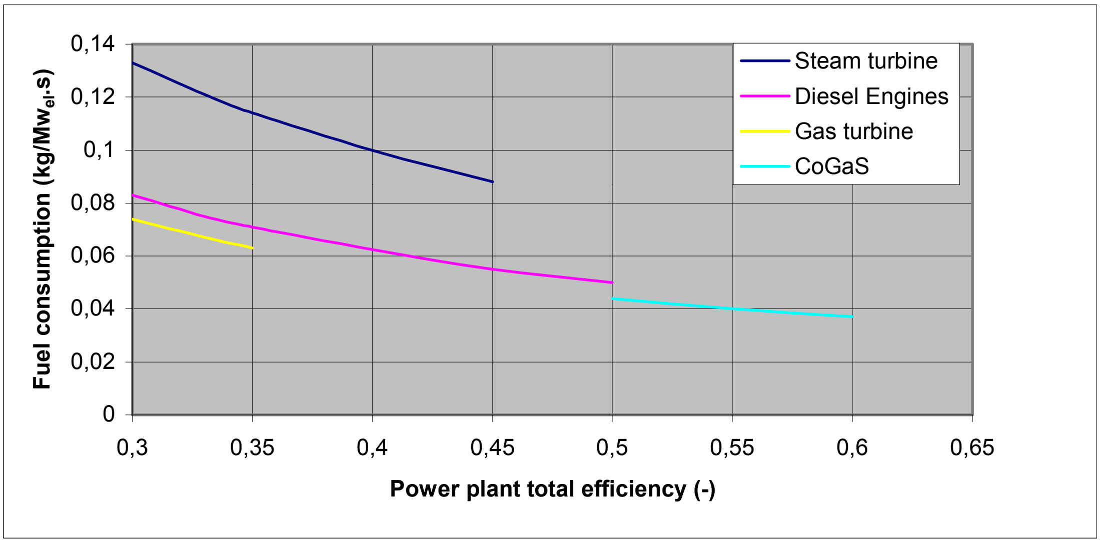


Figure 5.2 Power station efficiency affected on fuel consumption

The electric to cooling load ratio(Sc) defined as :

$$S_c = \frac{P_{el,d}}{Q_c} = \frac{PE_{el}}{PE_c * COP_{cru}} \tag{5.4}$$

Therefore , the cooling primary energy demand can be written as:

$$PE_c = \frac{P_{el,d}}{\eta_{pp} * \eta_{tr} * S_c * COP_{cru}} \tag{5.5}$$

From the last equations the term of total primary energy for electric and cooling load to the electric primary energy can be written as:

$$\frac{PE_{el} + PE_c}{PE_{el}} = (1 + \frac{1}{S_c * COP_{cru}}) \tag{5.6}$$

The study of the compression cooling load will be discussed for two cases namely power station and Diesel power plant. The estimation of the fuel utilized in the case of steam power station used coal as a fuel will be calculated at electric efficiency varied from 0.3 to 0.45 and in the case of gas and steam power station used gas as a fuel will be estimated at electric efficiency varied from 0.3 to 0.35 for gas turbine power station and from 0.5 to 0.6 for combined gas and steam power station. The Diesel engine power station will be taken in the range of 0.3to0.5. The different primary energy can be illustrated in **figure 5.2.** The coefficient of performance of the compression cooling unit used in this section is varied from 3 to 6.

It can be notified that the power station efficiency depend upon ambient temperature . In the case of steam power station, the ambient temperature affected on the condenser pressure. The increase in ambient temperature of 10 K decreases the efficiency around 3%. In the case of gas turbine, ambient temperature affected on the compressor performance. The increase of ambient temperature of 10 K decreases the efficiency around 5%. The same effect can be notified in the case of combined gas and steam power station (CoGaS). In Diesel engines, the ambient temperature affected on the air flow rate, the engine efficiency decreases around 2% by increases the air temperature of 10K.

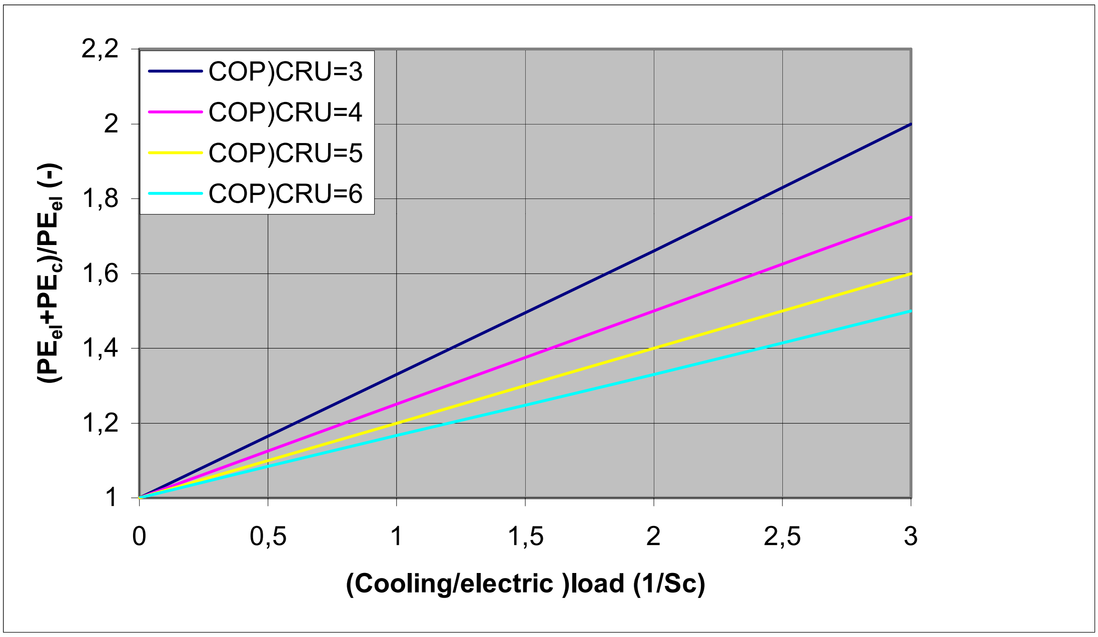


Figure 5.3 primary energy ratio as a function of both coefficient of performance (COP_{cru}) and cold to power ratio (S_c)

Figure 5.3 represents the fuel utilized ratio of both electric and cooling load to fuel utilized to electric load against the variation of the cold – power ratio at different compression coefficient of performance. It can be notified that the maximum fuel consumption ratio of 2.00 can be decreased to 1.33 by increasing the coefficient of performance from 3 to 6 . the minimum value at ratio of 0 that means the total electric regenerated used as a power load as shown in this figure.

5.2 Diesel power plant

5.2.1 Diesel engine and compression cooling unit

In this section , Diesel engine connected with the same compression cooling units in the previous section will be analyzed and studied. The main difference between the diesel power plant and the stationary power plant discussed in the last section that the performance at part load. The engine performance at part load change the performance efficiency . That is the reason for analyzing the next equations.

The full load performance of the engine to produce both electric and cooling load can be defined as (X_0) and can be written as:

$$X_0=X_{10}+X_{20} \tag{5.7}$$

Where

$X_{10} = P_{el,,d}$ and the electric load in this case:

$X_{20}= Q_c / COP_{cru}$

Therefore the cold- power ratio at full load can be written as:

$$S_0 = \frac{X_{10}}{X_{20}} * \frac{1}{COP_{cru}} \tag{5.8}$$

The cold- power ratio at part load can be written as:

$$S = \frac{X_1}{X_2} * \frac{1}{COP_{cru}} \tag{5.9}$$

From **figure 5.4**, it can be defined the total part load as:

$$\delta = \frac{X_1 + X_2}{X_{10} + X_{20}} \tag{5.10}$$

Therefore the total engine load as a function of both power and cold load can be written as:

$$\delta = \frac{S_0 * \delta_{el} + \delta_c}{S_0 + 1} \tag{5.11}$$

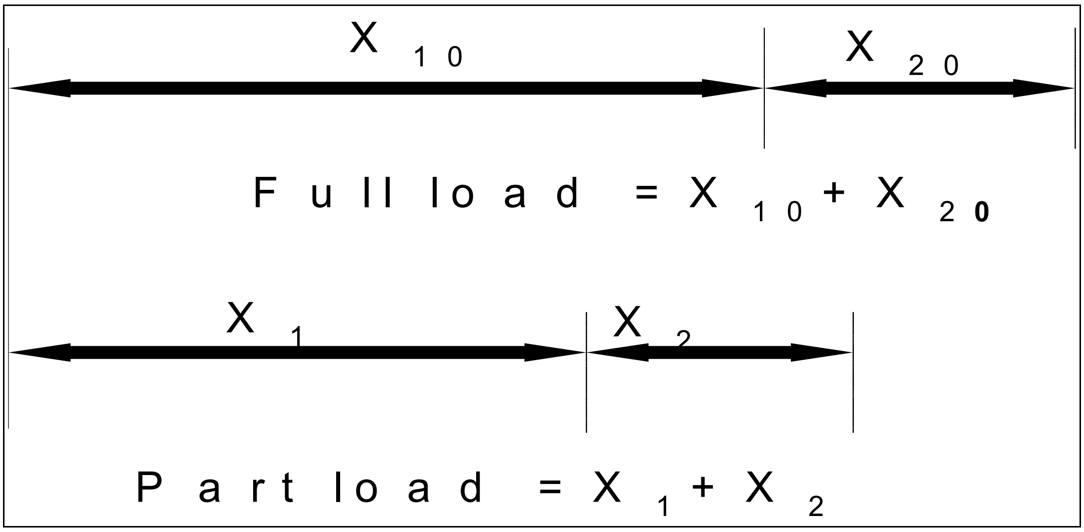


Figure 5.4 Part load chart of total engine plant

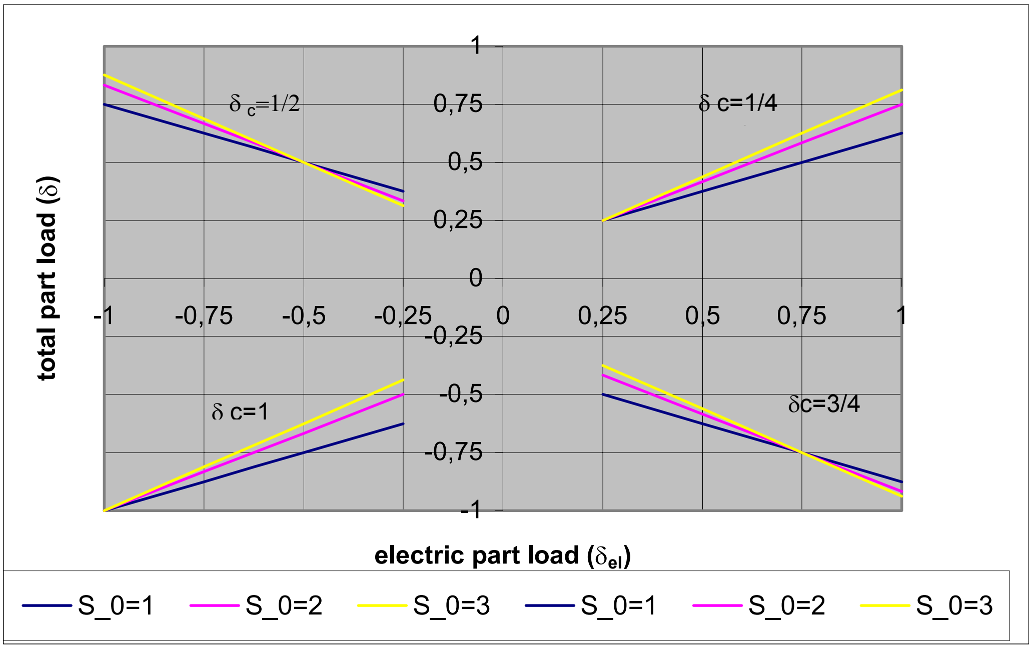


Figure 5.5 Part load performance of both diesel and compression refrigeration unit

From the last equation , the relation between the engine and cooling part load can be illustrated in **figure 5.5**. From this figure can be obtained the engine total load and therefore from **figure 5.6** can be obtained the efficiency ratio (η / η_0).

The actual primary energy used at this part load can be obtained from the equation:

$$m_f = \frac{1}{\eta_{total} * \eta / \eta_0} * \frac{P_{el,d} + Q_c / COP_{cru}}{CV} \tag{5.12}$$

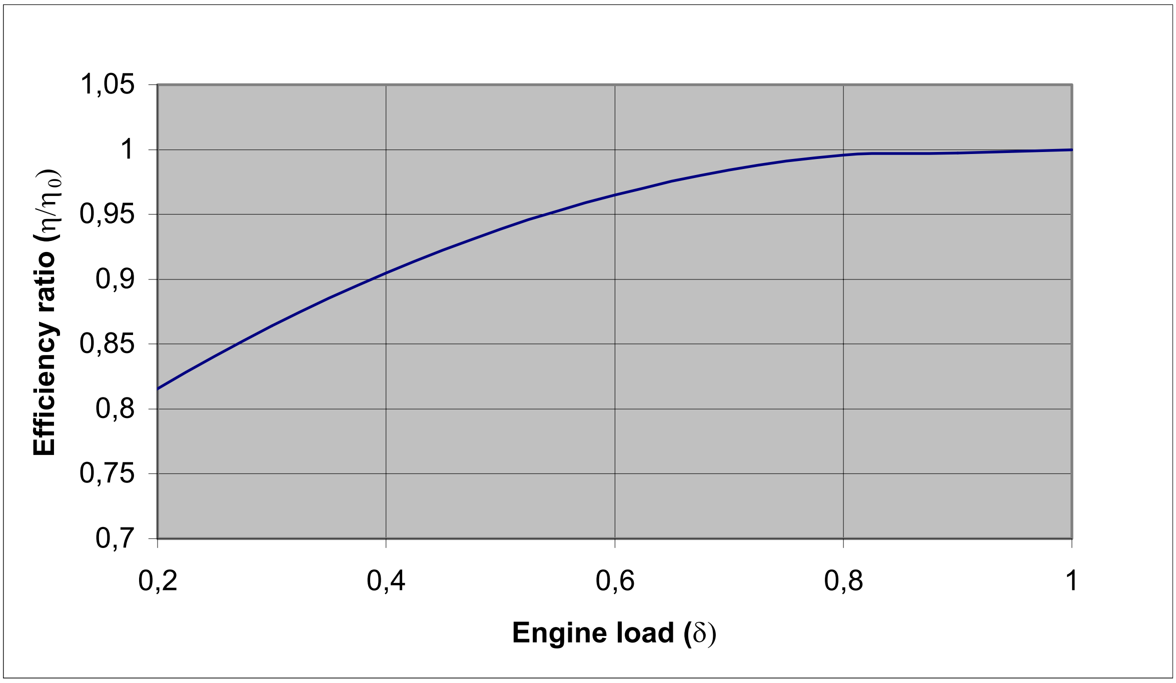


Figure 5.6 Efficiency ratio of diesel engine part load performance

The cooling and electric load are the same at 350 and 400 kW respectively as the case of the electric power stations . It can be obtained from the last three figures the performance at full load which has (1/S0) of 0.875 ,this load at the day performance. At the night performance which operated at engine part load both the engine power and the cooling load operate at difference part loads as discussed in the last figures.

The night part load will be chosen as:

- The engine electric load is 240 kW
- The cooling load of 100 kW

From these loads it can be obtained both part load of power as($\delta_{el} = 0.6$) and the cooling part load ($\delta_c = 0.286$) from this values and **figure 5.4** ,it can be obtained the total part load performance of (0.44 ,0.495 and 0.52) at the cooling – power ratio (1/S0) of (1 ,2 and 3) respectively. From **figure 5.6,** it can be obtained the range of efficiency ratio from 0.92 to 0.945) and therefore from equation (5.11) can be also obtained the range of primary energy used at this part load case of range from (0.0233 to 0.0240 kg/MW$_{el}$.s).

The relation between engine losses and heat added to the diesel engine can be expressed as a function of fuel consumption and calorific value were be discussed in chapter 3.

The full load performance of both reference and diesel power plant with compression refrigeration units can be summarized as:

- Reference system primary energy utilized is 0.0244 kg/s
- Diesel engine primary energy utilized is 0.0400 kg/s

The part load performance of 240 kW electric and 100 kW cooling can be obtained from both reference system and diesel engine as:

- Reference system primary energy utilized is 0.0132 kg/s
- Diesel engine primary energy utilized is 0.0233 kg/s

The effective cooling water losses and exhaust losses Q_{cw} and Q_{ex} as combined to heat losses Q_h. The heat balance depends on the type of the engine as well as the engine load. Generally, both cooling losses and exhaust losses increase by decreasing the engine load. It is important here to study the effect of the different engines on the performance of the absorption cooling units. The selected engines in this study has a different cooling to exhaust losses ratio. The first group of engines involves a cooling to exhaust ratio of 2:3, the second group involves the ratio of 1:1 but the third group has a ratio of 3:2. It is assumed that the cooling machines are constant when operate with the three groups of engines to study the effect of temperature difference supply and return temperature on both cooling water losses and average one.

5.2.2 Diesel engine and absorption cooling unit

In this section, the performance of single and double-stage absorption cooling units connected to Diesel engines as shown in **figure 5.7** will be studied. The demand cooling load will be changed from 200 to 400 kW. Because of the high complexity of the problem, the effective heat balance ratios will be fixed in this section at $\eta_{el} = 0.3$, $\eta_{ex} = 0.225$ and $\eta_{cw} = 0.3$ .The ambient temperature and the absorption cooling units type will be the parameters under consideration . The effect of changing effective heat balance ratios (Sankey diagrams) will be discussed in the later sections in this chapter. It is a very complex aim to investigate the all parameters affecting in the plant in one study. A detailed study is required to determine the temperatures and maximum coefficient of performance of the named system to obtain an optimal system for fuel utilization. Normally heating supply temperatures are varied in the summer and winter time. The temperature control of the heating systems is very important point, when the system operated at constant temperature process as the case of the two-stage absorption cooling units.

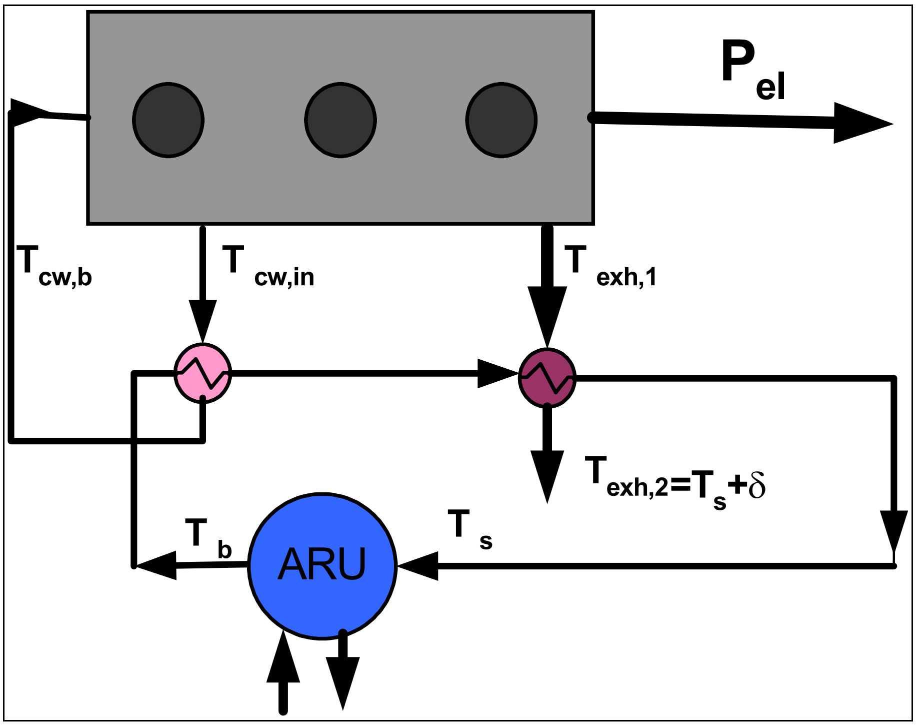


Figure 5.7 An absorption cooling unit connected to diesel engine system

The main difference between ARU and CRU that in the case of ARU the cooling load depend up on the electric load **Figure 5.8** shows that the Sankey diagram operation of both diesel engine and absorption cooling unit The fuel utilized in this figure can be divided in two loads Electric and cooling load .Therefore, it will be obtained from **figure 5.9** three cases between engine load and cooling load. In this figure it is assumed that the main demand is the cooling load. In the first case the engine developed both electric and cooling loads, in the second case the engine electric load is less than the demand electric load in this case the rest electric load can be obtained directly from the power station and in the third case the engine developed excess electric load ,it can be transferred to the power station .

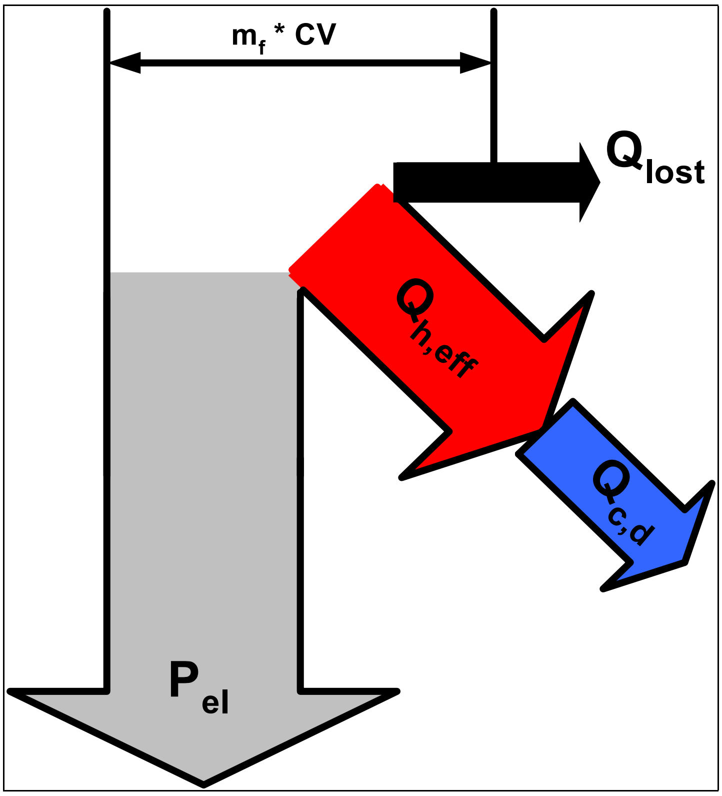


Figure 5.8 Sankey diagram of quantity diesel engine and Absorption cooling unit

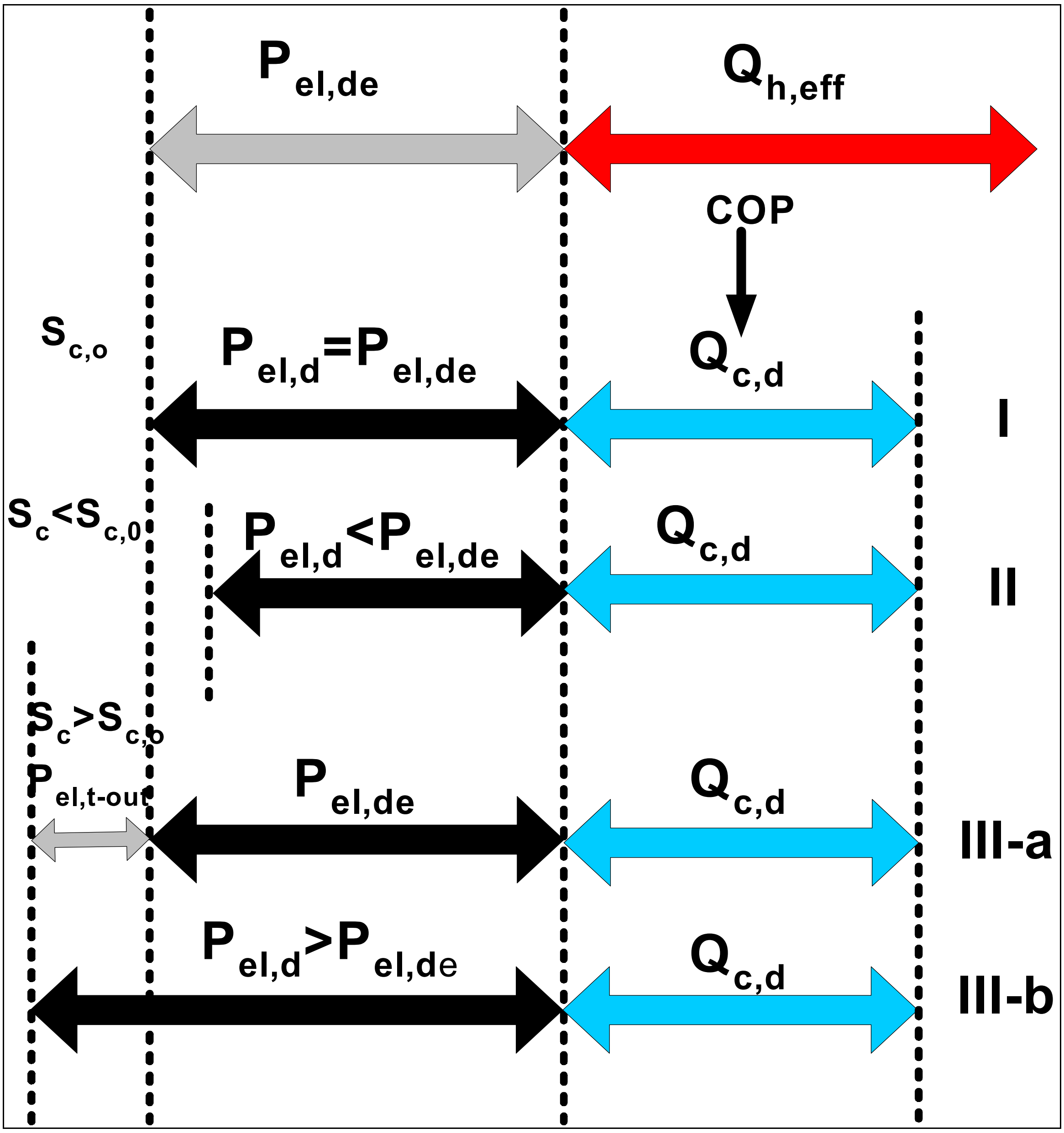


Figure 5.9 constant cooling load with difference electric engine loads (III-a Net work take out, II Network feed-in)

To discuss this three customers possibility can be difference engine to select to compare the different cases.

The fuel utilization ratio for heating can be written as follows:

$$S_h = \frac{P_{el}}{Q_h} \tag{5.12}$$

and the fuel utilization ratio for cooling can be written as follows :

$$S_c = \frac{P_{el}}{Q_c} \tag{5.13}$$

To illustrate the difference engine performance the fuel ratio of cooling can be written as:

$$S_c = \frac{\eta_{el,d}}{COP_{aru} * \eta_{eff}} \tag{5.14}$$

Figure 5.10 shows that the performance of diesel-absorption systems with different power-cooling ratio, which affected on the engine electric loads. From this figure ,it can be obtained that in case I the engine electric load and cooling load were equal to the demand loads. This is an ideal case and difficult to obtained for different load performance day and night. The case II which occurred diesel electric load more than the demand electric load this difference of power can be feed- in network. In case III the demand electric load is more than the engine load ,in this case the difference power can be taken out network and the comparison with reference system was illustrated in **figure 5.11**.

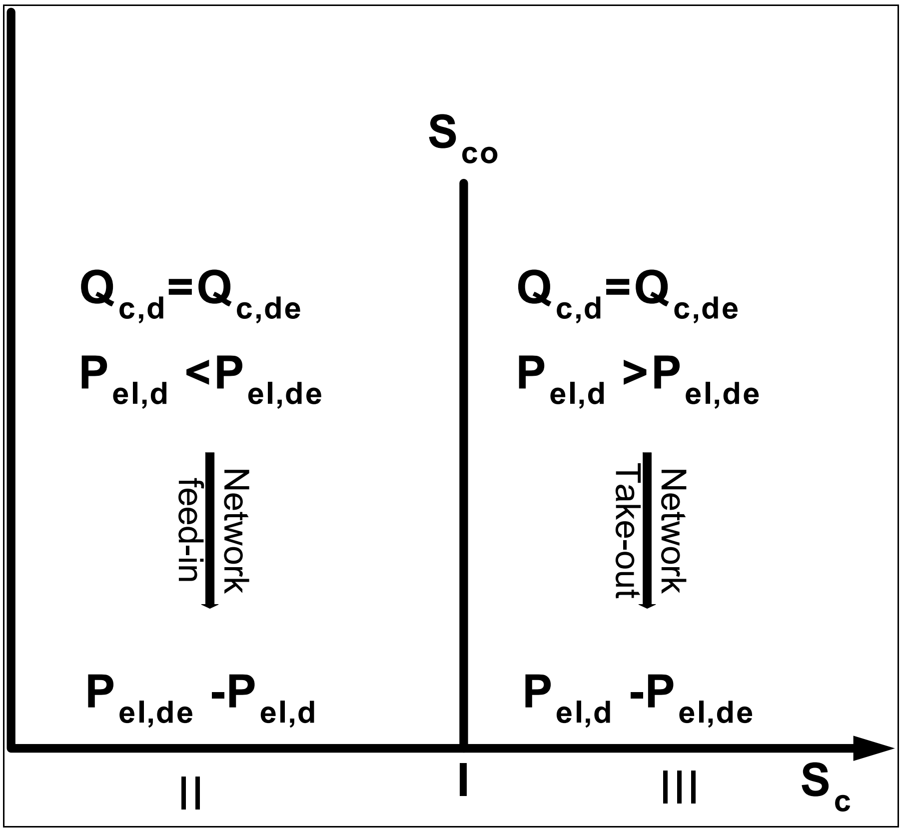


Figure 5.10 The affect of power –cooling ratio on the performance of both diesel and network power station

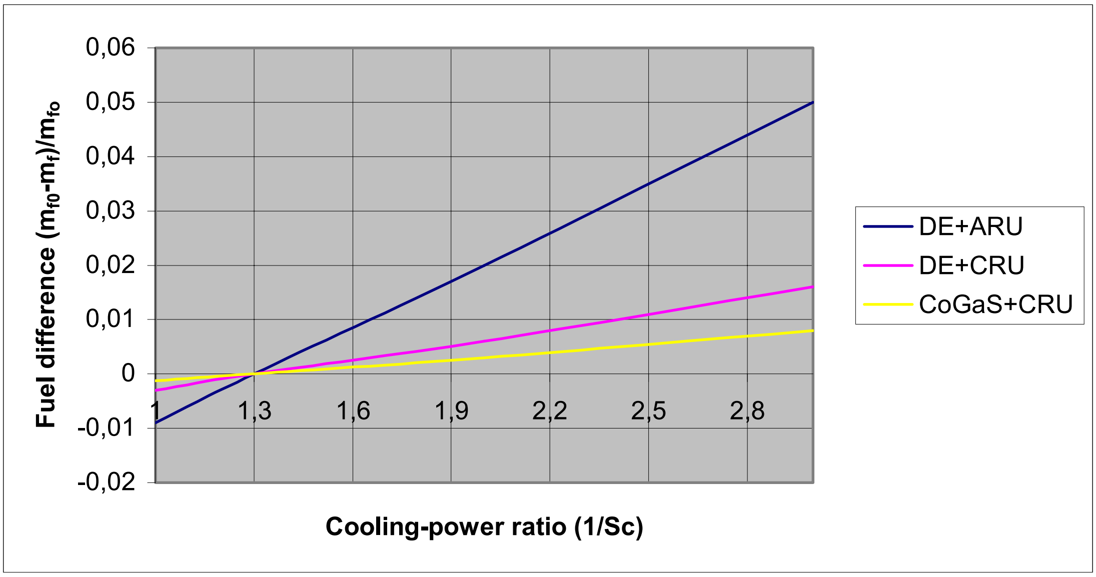


Figure 5.11 The affect of power –cooling ratio on the performance of ARU and CRU

It is aimed here to study the Sankey diagram effect in both electric load and fuel utilization ratio at the same cooling load illustrated in the previous sections which will be study the same effect with the solar collector. The selected engines have a variation of electric efficiency, exhaust gas losses and cooling losses . The different engines are illustrated in **table 5.1**. It can be notified that the reference engine was studied in the previous sections named engine I, has electric efficiency of 0.3 and effective heat losses ratio 0.525 . The selected engines have a variation of both electric and effective heat losses ratio, therefore the electric efficiency varied from 0.3 to 0.45 and the effective heat losses varied from 0.45 to 0.575 . The selected engines are compared with both types of absorption cooling unit at both ambient temperature 20 and 40 °C. Engine I at ambient temperature 20 °C was considered as a reference case for two-stage absorption cooling unit.

Figure 5.12 shows the variation 0f fuel utilization ratio for different engines ratio (Sankey diagram). From this figure ,it can be shown that for engines I and II (have the same electric efficiency) the fuel utilization ratio in Engine I is more than engine II, that means the increasing of the effective heat losses decrease the fuel utilization ratio. The same effect can be done but in another cases engine III and IV. The performance of single-stage absorption cooling machine will be taken at maximum coefficient of performance.

It can also be notified that for the sex cases the electric load and the fuel utilization ratio of single-stage is approximately the halve of the two-stage absorption cooling unit. **Figure 5.13** shows that the fuel utilization ratio at ambient temperature 40°C . From this figure ,It can be notified that the increase of temperature from 20 to 40 °C increase both electric load and fuel utilization ratio, for example at engine I at two-stage absorption cooling unit from 1 to1.2 and for engine IV from 1.8 to 2.15. the same result obtained for the another cases. It can be illustrated that the same engine performance of engine I and two-stage absorption cooling unit at 20 °C can be obtained by engine IV at ambient temperature 40°C and single stage cooling unit and when two-stage cooling unit will be used at 40 °C the load increased to 2.15.

Engine	Electric efficiency	Effective heat losses	Rest losses
I	0.3	0.525	0.175
II	0.3	0.575	0.125
III	0.35	0.525	0.125
IV	0.35	0.55	0.10
V	0.4	0.50	0.10
VI	0.45	0.45	0.10

Table 5.1 Effective engine heat losses for different engines

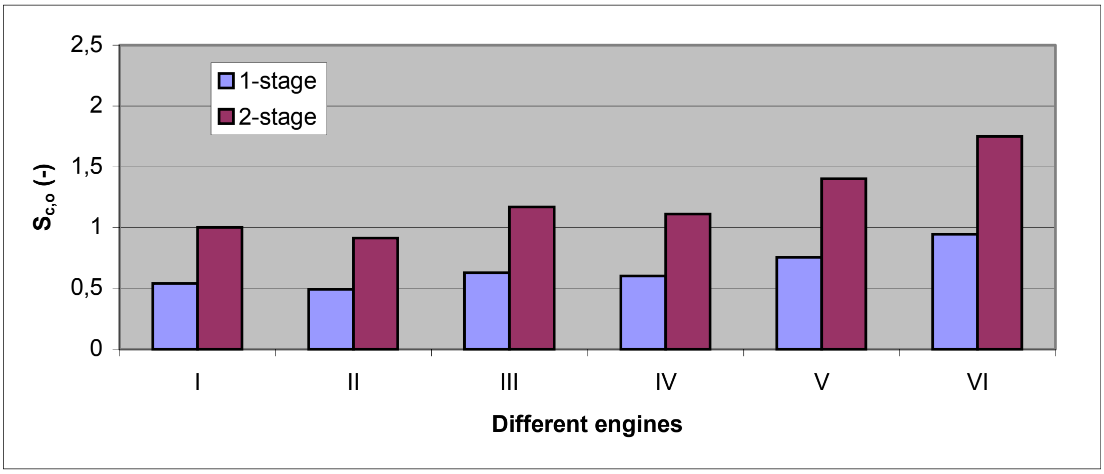


Figure 5.12 Power-cooling ratio for different engines at 20 °C ambient temperature of engine selected in table 5.3

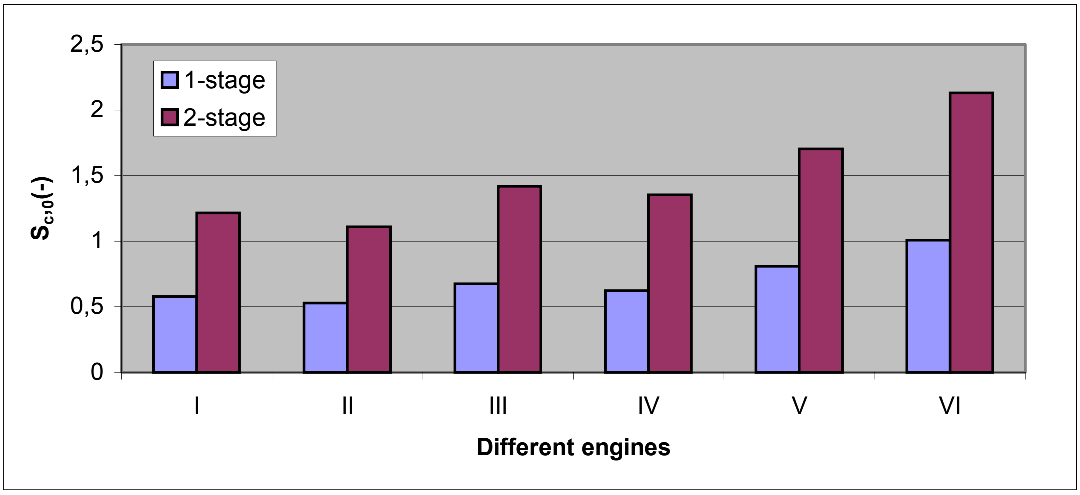


Figure 5.13 Fuel utilization for different engines at 40 °C ambient temperature of engine selected in table 5.1

From the heat transfer flow chart shown in **figure 5.7**, it can be illustrated in a heat-temperature diagram as shown in **figure 5.14**, which shown that the different temperatures an heat engine losses affected on both supply and back temperatures of absorption refrigeration machine.

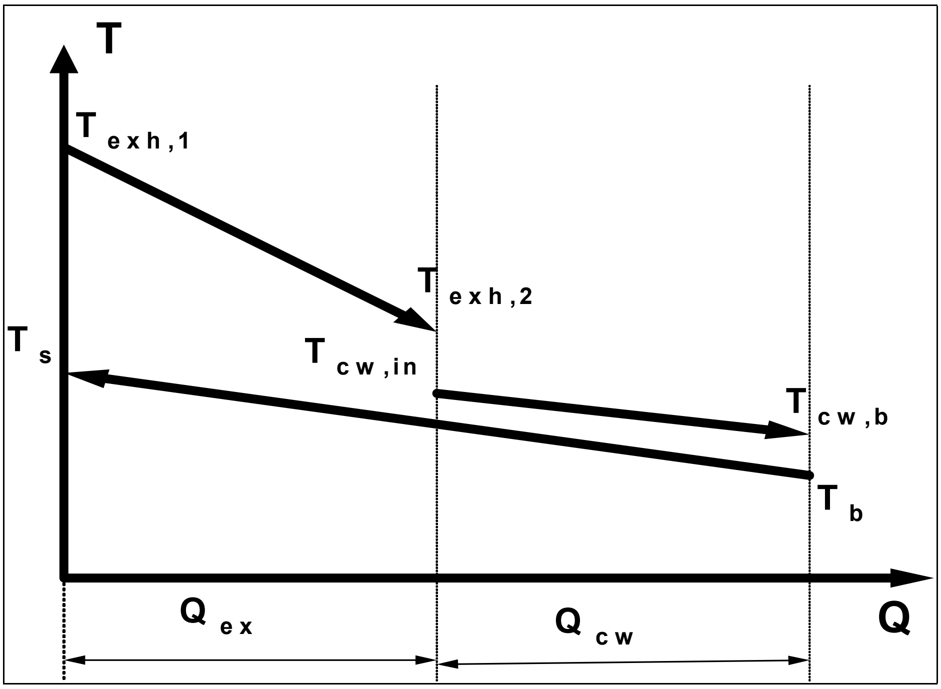
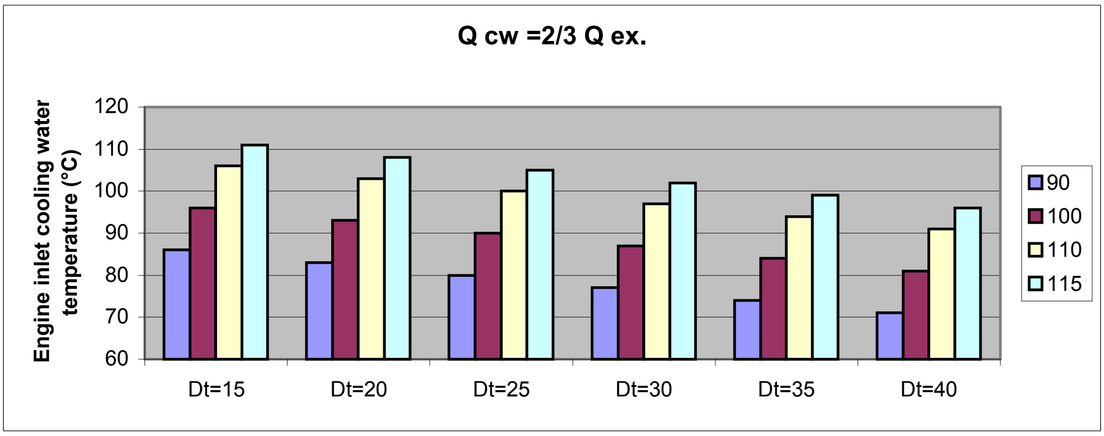
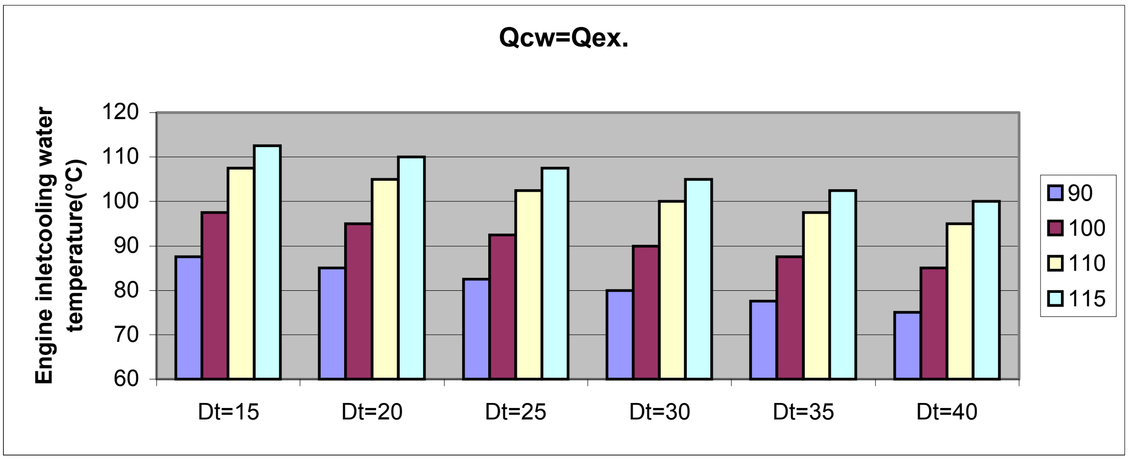


Figure5.14 T-Q diagram of Diesel engine heat losses temperatures and an absorption cooling machine performance temperatures.

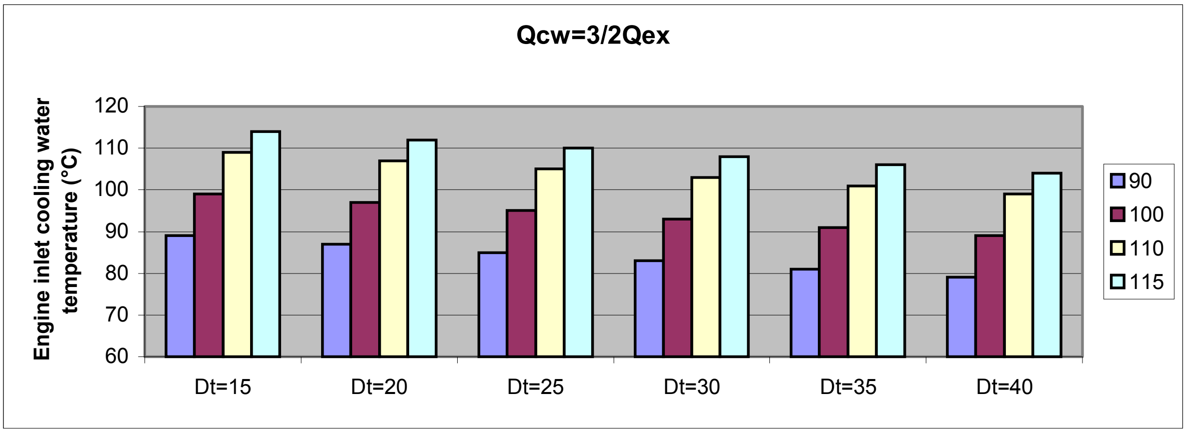
Figure 5.15-a illustrates the engine cooling water inlet temperature as a function of absorption supply temperature and absorption temperature difference at cooling exhaust engine losses ratio of 2:3. From this figure, it can be notified at 90 °C supply temperature that the increasing of absorption temperature difference from 15 to 40 k decrease the cooling water supply temperature from 86 to 71 °C. The same result can be obtained at 115 °C supply temperature but in this case decreases the cooling water temperature from 11 to 96 °C. It is important here to notify that the increase of the supply temperature increase the engine cooling temperature. To control the engine supply cooling water temperature at 90°C can be obtained the supply temperature as follows: 90 °C at $\Delta T = 15.90$ and 100°C at $\Delta T = 25$ and 35 °C. **Figure 5.15-b** shows the same effect at another engine cooling to exhaust losses ratio of 1:1 and **figure 5.15-c** illustrates the cooling water supply temperature for cooling to exhaust losses ratio of 3:2 . From the last three figures, it can be obtained the range of performance of absorption cooling unit for the controlled engine water temperature range.



a) Effective cooling heat losses =2/3 of Effective exhaust heat losses



b) Effective cooling heat losses = Effective exhaust heat losses

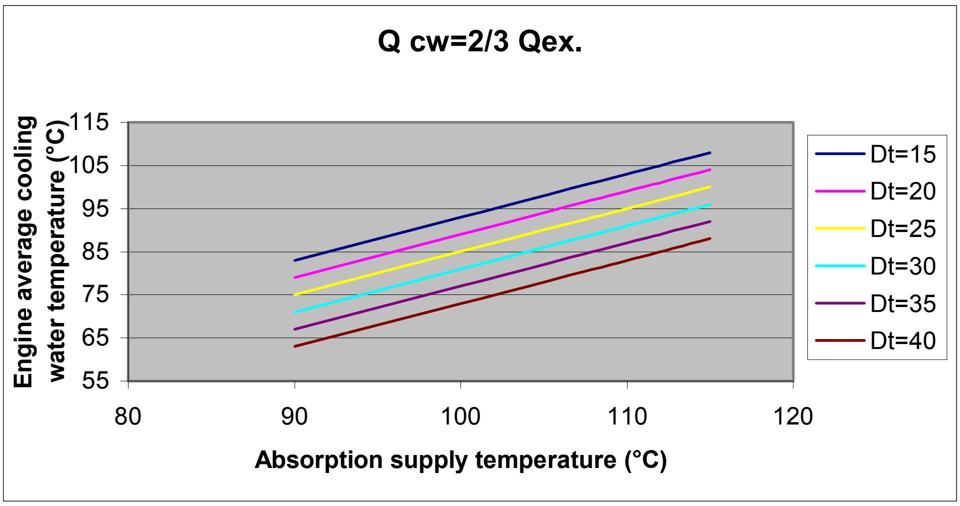


c) Effective cooling heat losses =3/2 of Effective exhaust heat losses

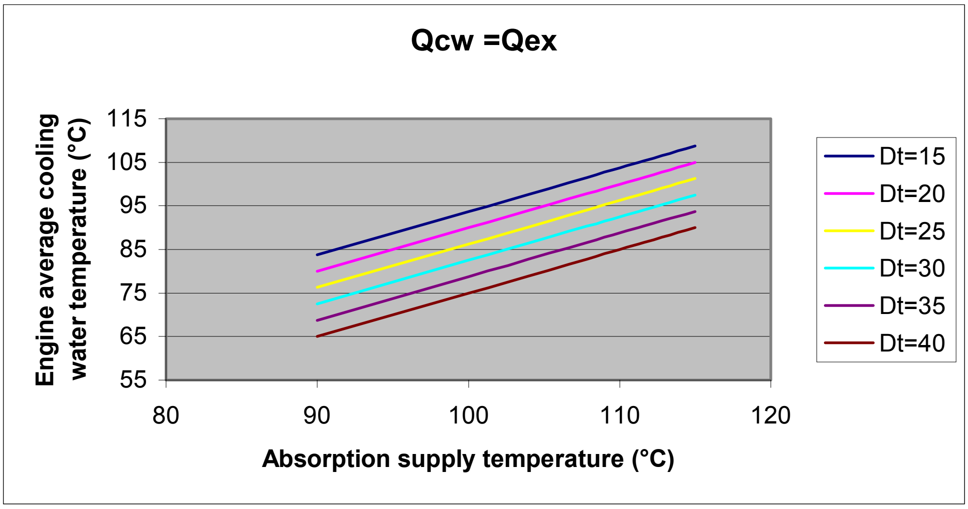
Figure 5.16 Absorption temperature difference affect on engine supply cooling water temperature at difference absorption supply temperature

Figures 5.17 represents the engine average water temperature defined as $((T_{cw,in}+T_{cw,b})/2)$ for the cooling to exhaust losses of 2:3,1:1 and 3:2 respectively. The controlled average water temperature of 85 °C gives tat at 3:2 engine losses and absorption supply temperature 90 °C , the temperature difference are available from15 to 40 K ,at 1:1 engine losses ratio and supply temperature 100 °C the temperature difference are available from 30 to 40 K but at the ratio of 2:3 and absorption temperature of 110 °C only the available value is 40 K.

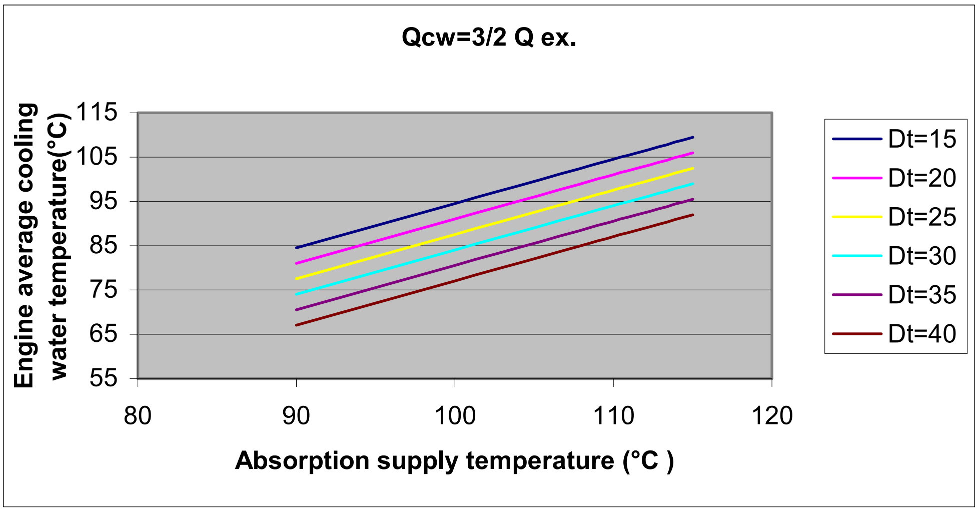
It is difficult to operate the two-stage absorption unit at 115 °C while for the three engine select groups the average temperature of 85 not occurred.



a) Effective cooling heat losses =2/3 Effective exhaust heat losses



b) Effective cooling heat losses = Effective exhaust heat losses



c) Effective cooling heat losses =3/2 of Effective exhaust heat losses

Figure 5.17 Absorption temperature difference affect on engine average cooling water temperature at difference absorption supply temperature

Figure 5.18 illustrates the affect of increasing of absorption supply temperatures on the inlet cooling water at difference ambient temperatures of 20 and 40 °C. From this figure it can be obtained the range of performance of both single and two-stage absorption refrigeration machine. In the case of single stage the performance engine cooling water only operated at 20°C ambient temperature but in the case of two-stage it can be operated till 107 °C supply temperature at ambient temperature of 40 °C.

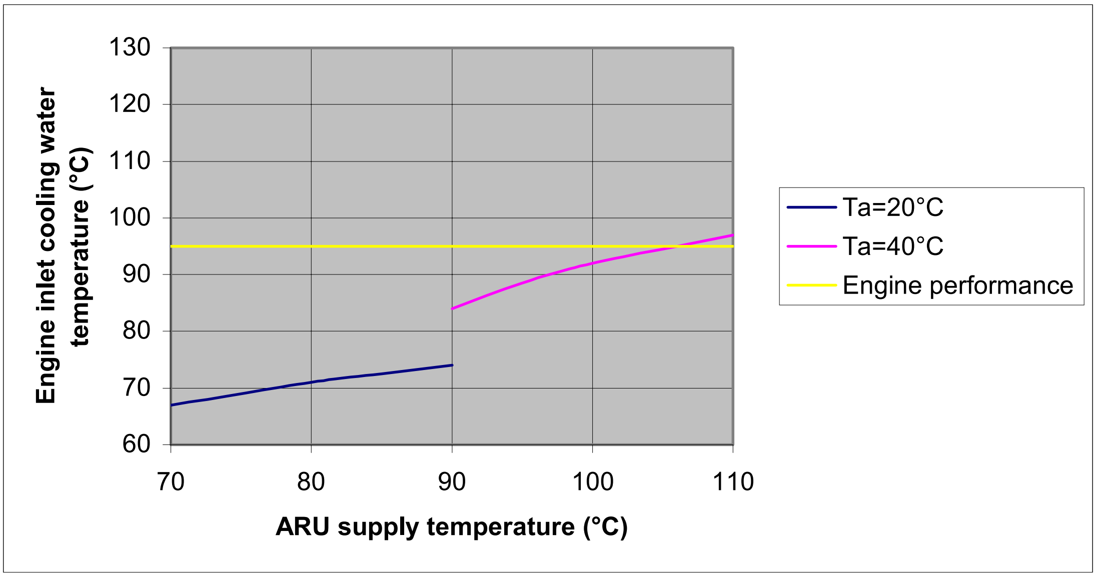


Figure 5.18-a Engine inlet cooling water as a function of both supply and ambient temperature of Two-stage ARU

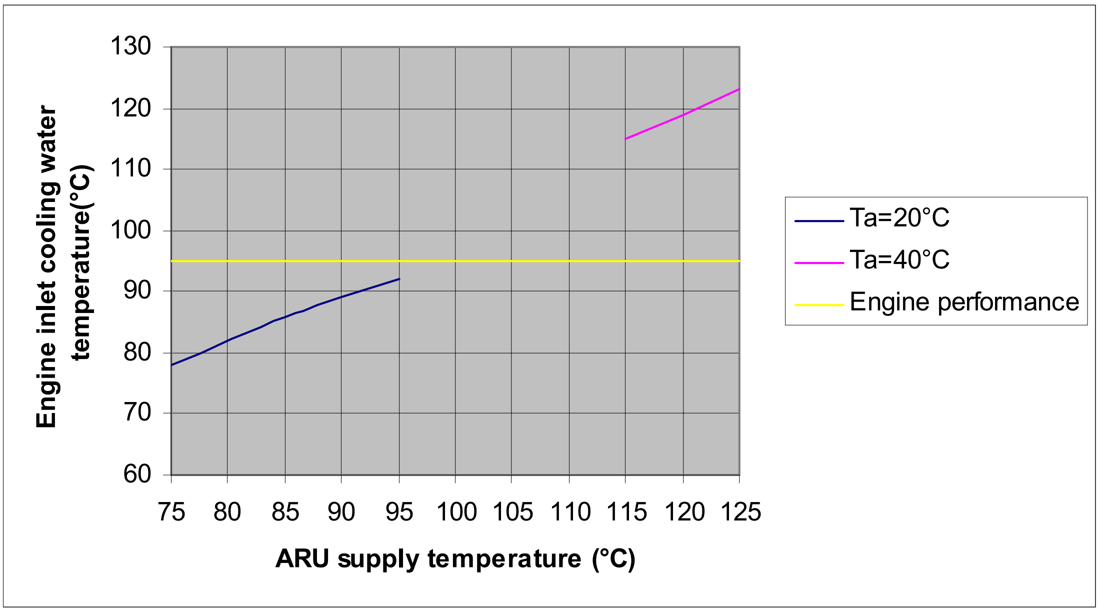


Figure 5.18-b Engine inlet cooling water as a function of both supply and ambient temperature of single-stage ARU

Figure 5.19-a shows the effect of the engine losses ratio of the two-stage absorption cooling machine. The variation of the engine losses to produce a cooling water supply temperature of maximum value 75 °C is shown in figure 5.18 ,in this case the variation of refrigeration machine supply temperature from 70 to 85 °C corresponding to the engine losses ratio (0.66,1 and1.5) respectively . It can be notified from this figure that the increasing the losses ratio

from 0.66 to 1.5 increases the water losses temperature. Figure 5.18-b shows the increasing of engine inlet cooling water temperature by increasing the engine heat ratio for the single-stage absorption cooling unit. From **figure 5.19** ,it can be obtained that the performance of the cooling unit at maximum coefficient of performance can be obtained at the high temperature range of engine cooling water. To use the performance engine cooling water values the cop should be decreased, which increases the fuel consumption.

Figure 5.20-a shows that the increase the ambient temperature from 20 to 40 °C produce an increasing in both absorption supply temperature and cooling water losses temperature. For example, it can be indicate at 115 °C supply temperature that the increasing of cooling exhaust ratio from 0.66 to 1.5 increase the cooling water temperature from 98 to 106.5 °C. From **figure 5.20-b**,it can be notified that the inlet cooling water temperature reach to 124 °C at 40 °C ambient temperature. This range of temperature is higher than the engine performance value. In addition, to control the water temperature of range between 85 to 95 °C of two stage ARU, the maximum supply temperature should be not exceed than100 °C. The different cooling water temperature obtained at the case of equality between cooling water and exhaust losses for different supply temperature90,100,110 and 115 °C are 84.5,92.25,96.75 and 102.25 respectively. It is important that to estimate the range of cooling water rang before selection the type of absorption cooling machine to obtain the optimum performance of the machine.

Figure5.21 shows the effect of ambient temperature on the engine cooling water supply temperature. It can be notified from this figure that the increasing of ambient temperature from 20 to 40 °C increase the engine cooling water temperature and the maximum cooling water temperatures occurred at ambient temperature 40 °C as 106.25,100 ,96 at engine losses ratio of 1.4, 0.8 and 0.66 respectively. To controlled the water losses temperature at 90 °C, the ambient temperature increased from 30 °C for the case of 0.66. The maximum and the minimum cooling water temperatures at constant ambient temperature of 30 °C, are 100 and

83 respectively.

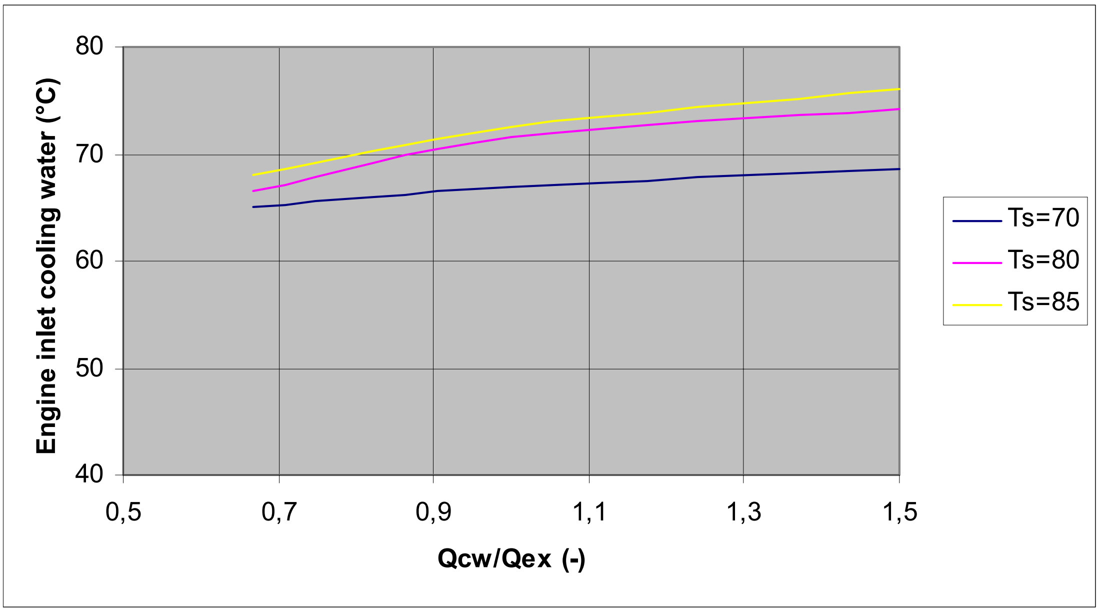


Figure 5.19-a Relationship between engines heat losses ratio and engine cooling supply temperature at difference absorption supply temperature and 20°C ambient temperature (Two-stage ARU)

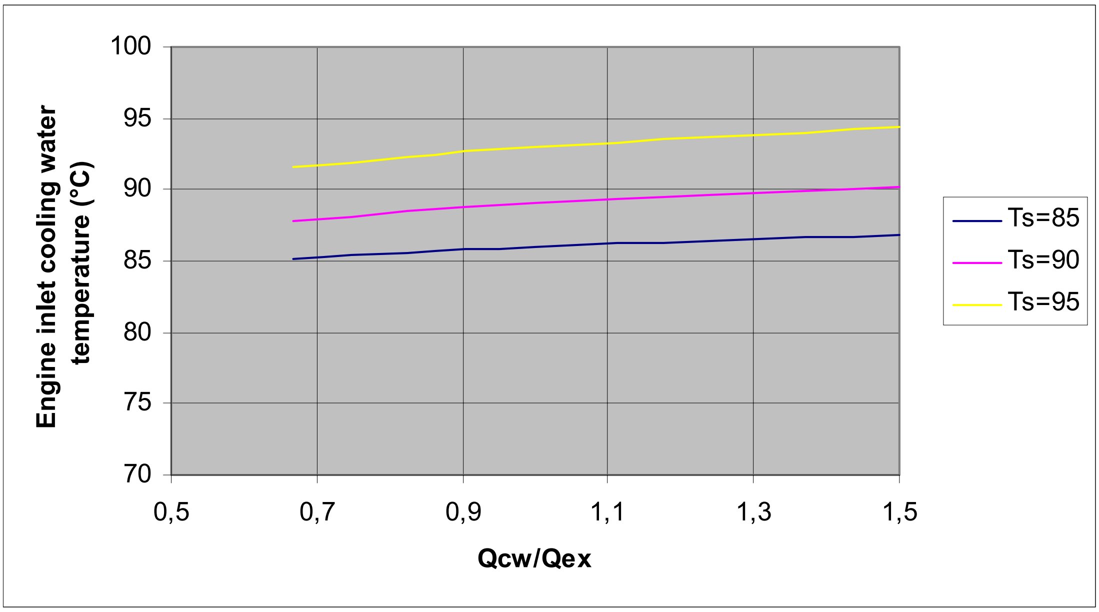


Figure 5.19-b Relationship between engines heat losses ratio and engine cooling supply temperature at difference absorption supply temperature and 20°C ambient temperature (Single-stage ARU).

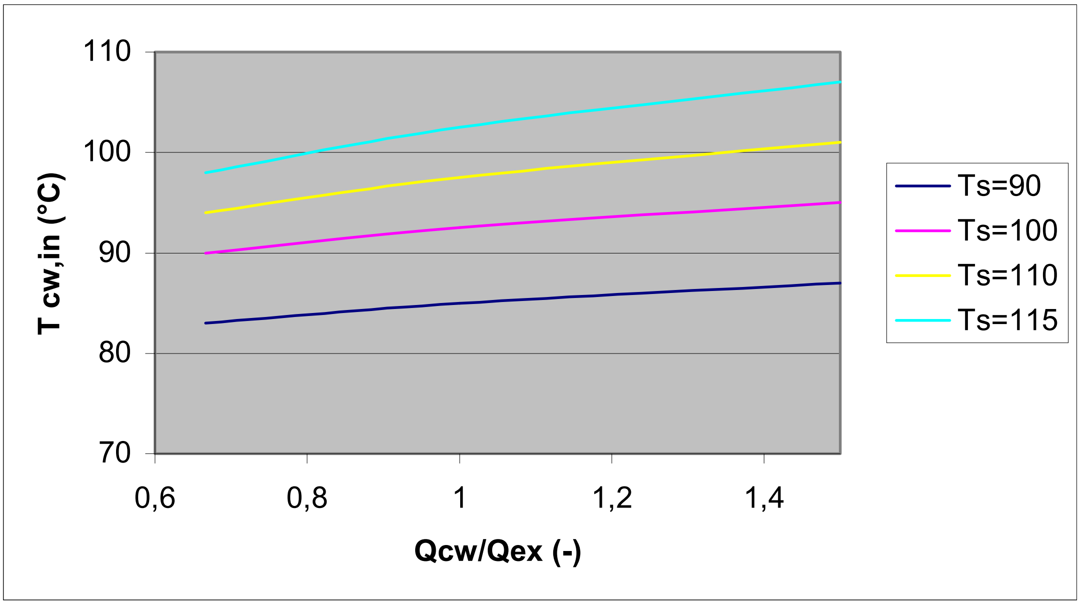


Figure 5.20-a Relationship between engines losses ratio and cooling supply temperature at difference absorption supply temperature and 40°C ambient temperature (Two-stage ARU).

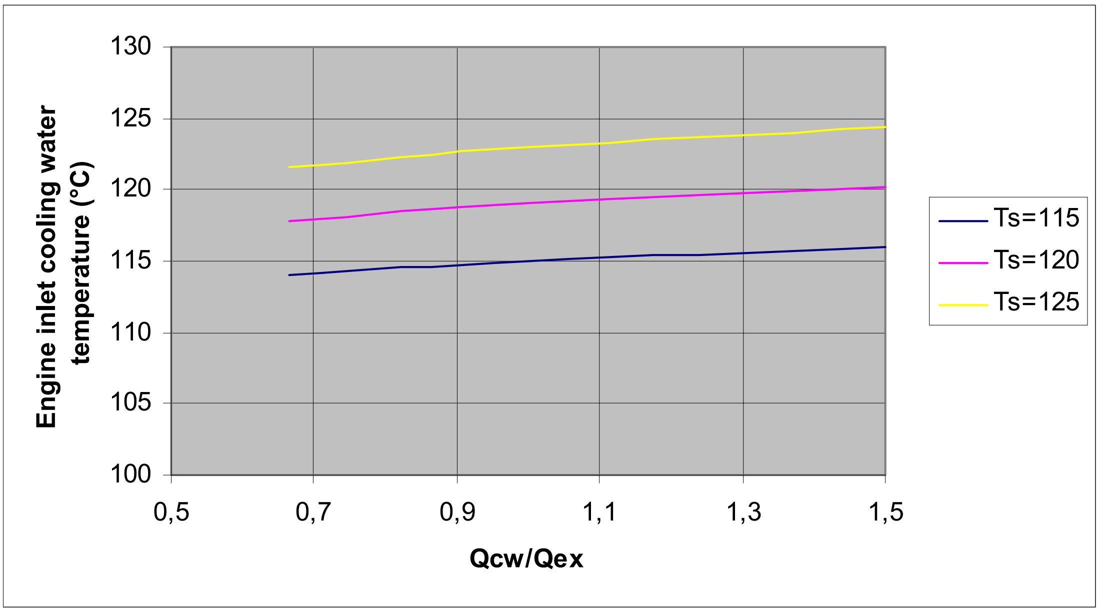


Figure 5.20-b Relationship between engines losses ratio and cooling supply temperature at difference absorption supply temperature and 40°C ambient temperature (Single-stage ARU).

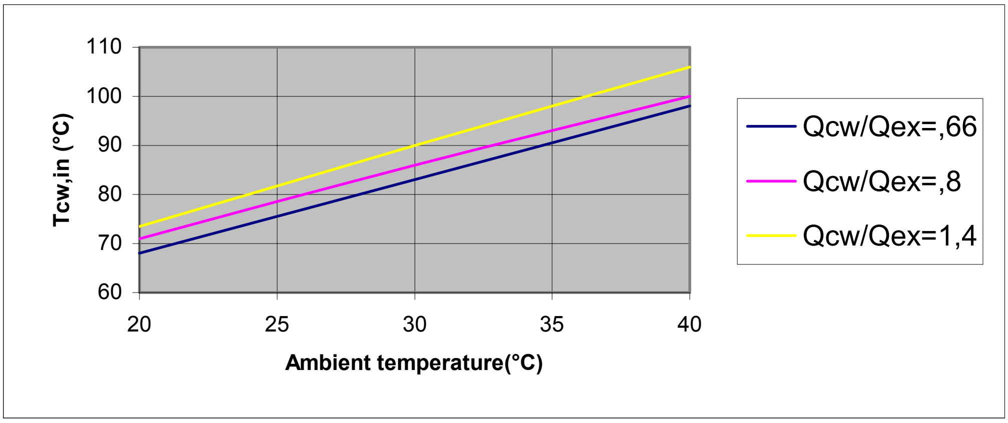


Figure 5.21 Performance area of engines cooling water supply temperature as a function of ambient temperature (Two-stage ARU).

Figure 5.22 represents the controlled supply temperature at constant range of cooling water inlet temperature from 75 to 90 °C. At the engine losses ratio of 1.4 the maximum supply temperature is 93.5 °C, for the engine losses ratio of 0.8 is 96.5 and for the engine losses ratio of 0.66 is 100 °C. **Figure 5.23** shows the effect of controlled supply temperature on the absorption machine coefficient of performance. In the case of 0.66 losses ratio the coefficient of performance decreased from 0.44 to 0.42 at 20 and 40 °C ambient temperature, decreased from 0.44 t0 0.415 in the case of 0.8 and from 0.44 to 0.41 in the case of 1.4. from this result, it can be obtained that the control of the cooling water temperature decreases the coefficient of performance for the all range of losses studied ratio. The best condition of performance of the two stage absorption machine is at the low ambient temperature range of 20 °C. the same machine operating in Alexandria has advantages than operate in Cairo or Aswan.

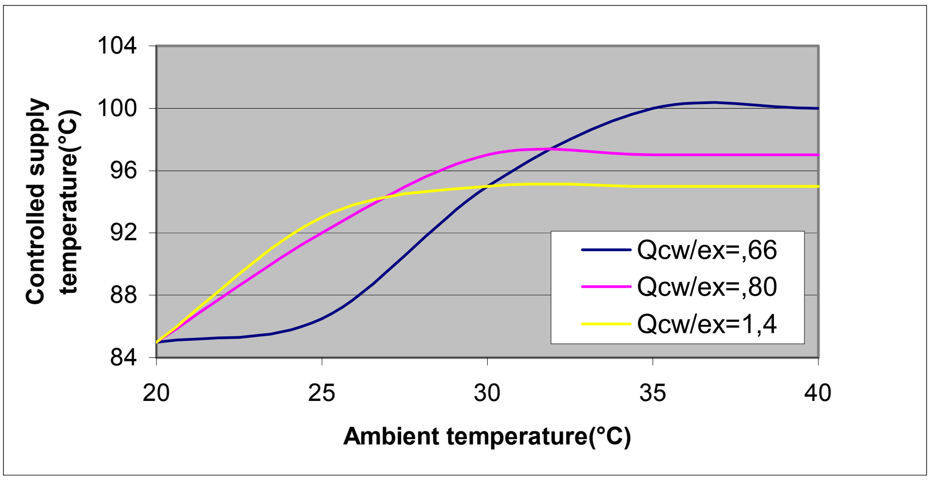


Figure 5.22 Controlled supply temperature as a function of difference engine ratios and ambient temperature (Two-stage ARU)

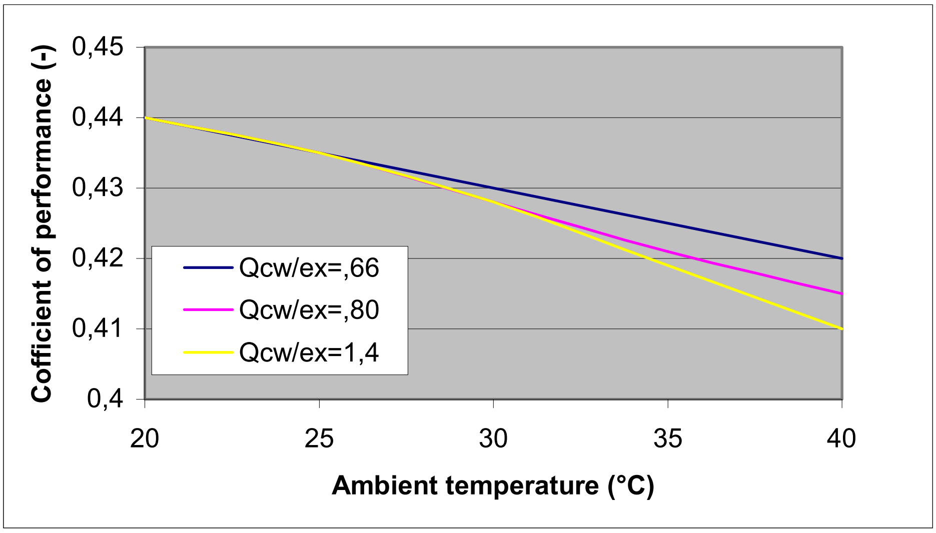


Figure 5.23 Coefficient of performance at controlled supply temperature of Two-stage absorption refrigeration unit (ARU)

The performance of single-stage absorption cooling unit is investigated in chapter 4 .**Figure 5.24** illustrates the supply temperature against the ambient temperature at maximum coefficient of performance ,while the supply temperature is considered to be not constant .The current study in this section carried out at the assumption for the temperature level shows in the named figure above . It can be shown that the supply temperature increased from 100 to 125 ° C when the ambient temperature increased from 20 to 40 °C and the required supply temperature increased from Alexandria to Aswan . According to this performance the ambient temperature and the coefficient of performance should be taken in our consideration in this section.

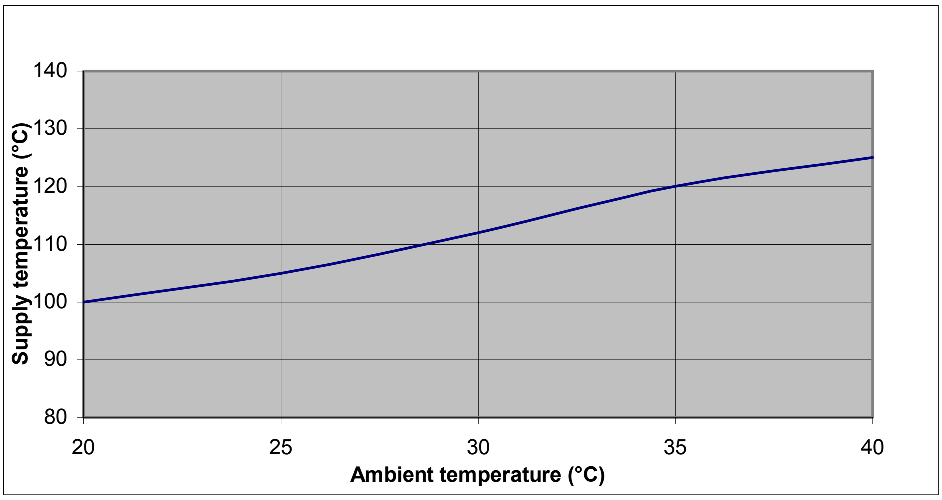


Figure 5.24 Single-stage supply temperature at maximum coefficient of performance

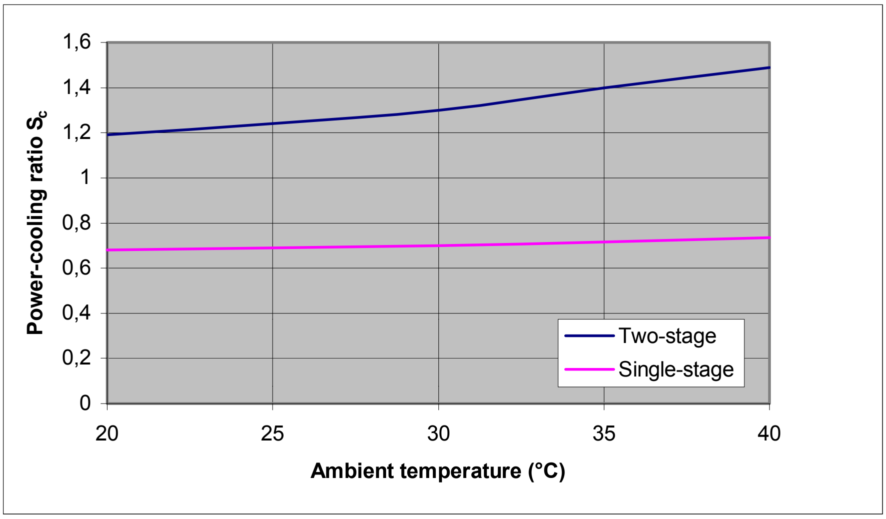


Figure 5.25 Power and cooling ratio comparison between single and two stage ACU(case of Two-stage supply temperature 90°C, single-stage maximum COP and engine I with Q_{cw}= 4/3 Q_{ex})

Figure 5.25 illustrates the comparison between single and two-stage absorption cooling unit, it can be noticed that the power-cooling ratio is 1.28 in the case of the two-stage but in the case of the single-stage one is 0.71 at the reference temperature 30 °C. The fuel utilization ratio for heating can be written as follows:

$$\omega_h = \frac{P_{el} + Q_h}{m_f.CV} \tag{5.15}$$

and the fuel utilization ratio for cooling can be written as follows :

$$\omega_c = \frac{P_{el} + Q_c}{m_f.CV} \tag{5.16}$$

therefore, the fuel utilization ratio of cooling can be written as:

$$\omega_c = (\eta_{el,d} + COP_{aru} * \eta_{eff}) \tag{5.17}$$

Figure 5.26 illustrates the comparison between single and two-stage absorption cooling unit, it can be noticed that the cooling utilization efficiency is more efficient in the case of the single-stage rather than in the case of the two-stage one. At the reference temperature 30°C the fuel utilization ratio is 0.525 for two-stage and 0.74 for single-stage but in the case of ambient temperature 40°C the vales of both cases are nearly the same difference of 0,24. It can also be notified from **figure 5.27** that single stage absorption cooling unit has allow temperature difference than the single stage one. This figure shows that the temperature difference are 17 and 26 K for both single and two stage at 30°C ambient temperature respectively.

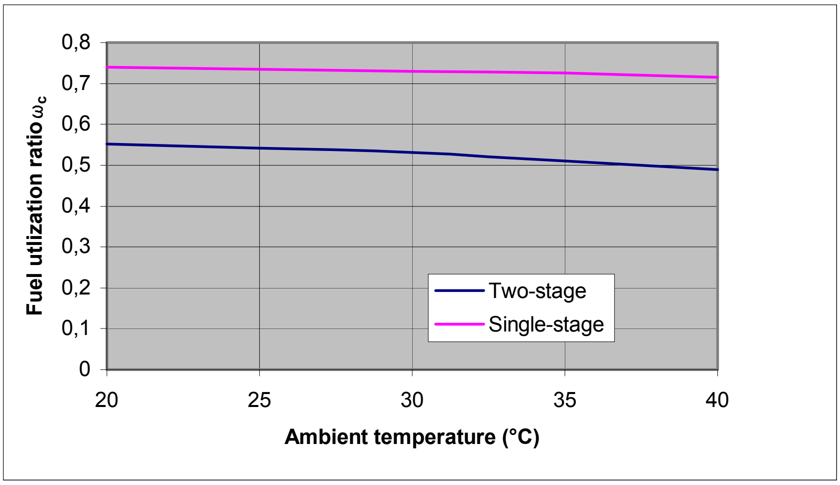


Figure 5.26 Fuel utilization ratio comparison between single and two stage ACU(case of Two-stage supply temperature 90°C, single-stage maximum COP and engine I with Q_{cw}= 4/3 Q_{ex})

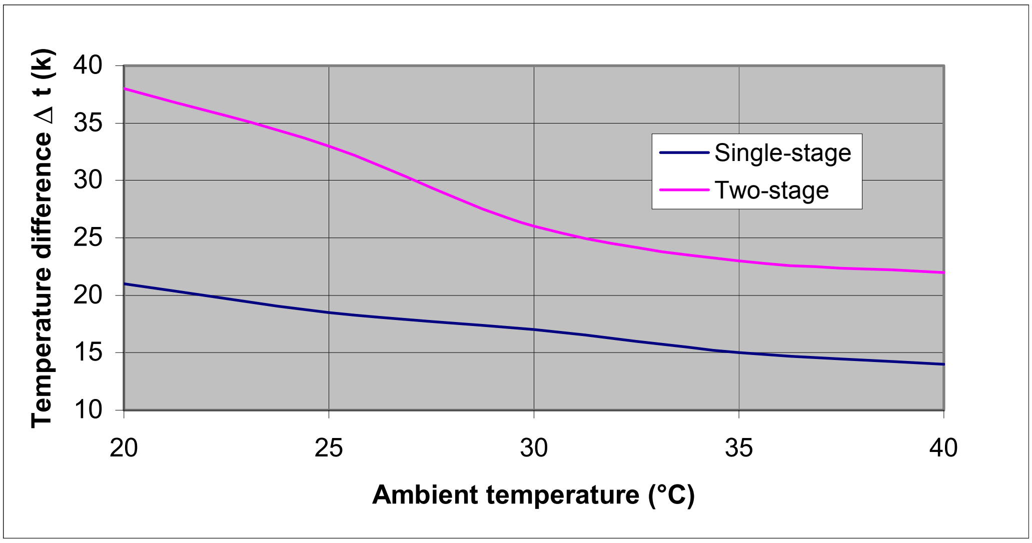


Figure 5.27 Difference of supply and return temperature comparison between single and two-stage absorption cooling units.

Figure 5.28 illustrates the affect of the change of supply temperature on power –cooling ratio. From this figure, it can be obtained that the increase of supply temperature of 10 k increase the power –cooling ratio from 0.68 to 0.72 at 20 °C ambient temperature by increasing the supply temperatures reach to 20 k the ratio increased to 0.816. The same affect can be notified at ambient temperatures 40 °C but the ratio increased from 0.72 to 0.84.

Figure 5.29 shows the restriction of use single stage absorption refrigeration machine for the selected engine input cooling water, from this figure ,it can be notified that the ambient temperature of 20 °C only can be used in this range of performance . Only the theoretical extrapolation of the performance of 40 °C can be used with supply temperature difference of 28 K but in this case the absorption machine is not operated. To overcome of this problem.

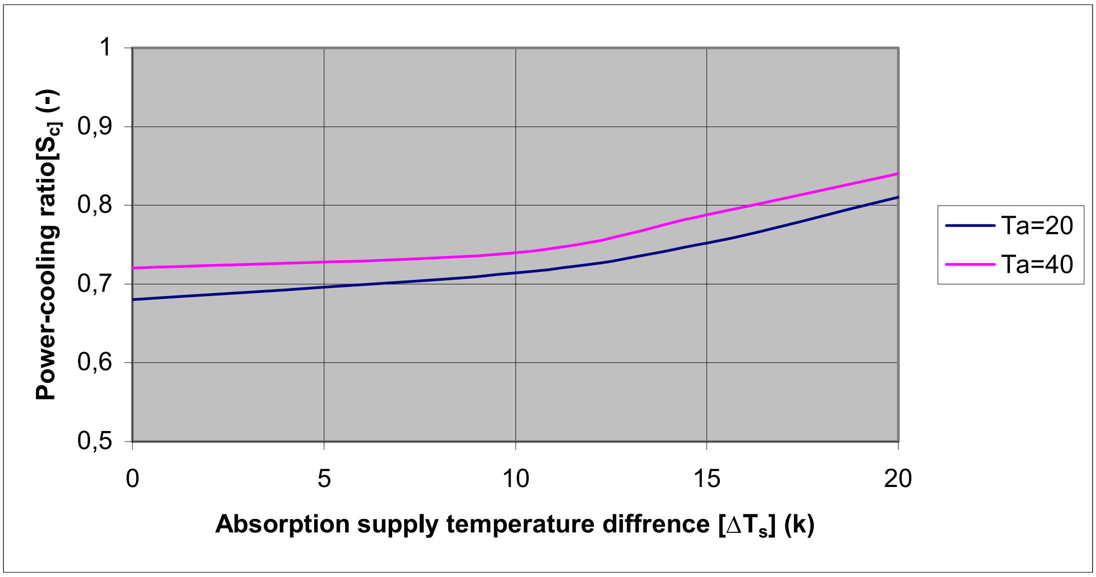


Figure 5.28 affected of change of absorption supply temperature on power –cooling ratio

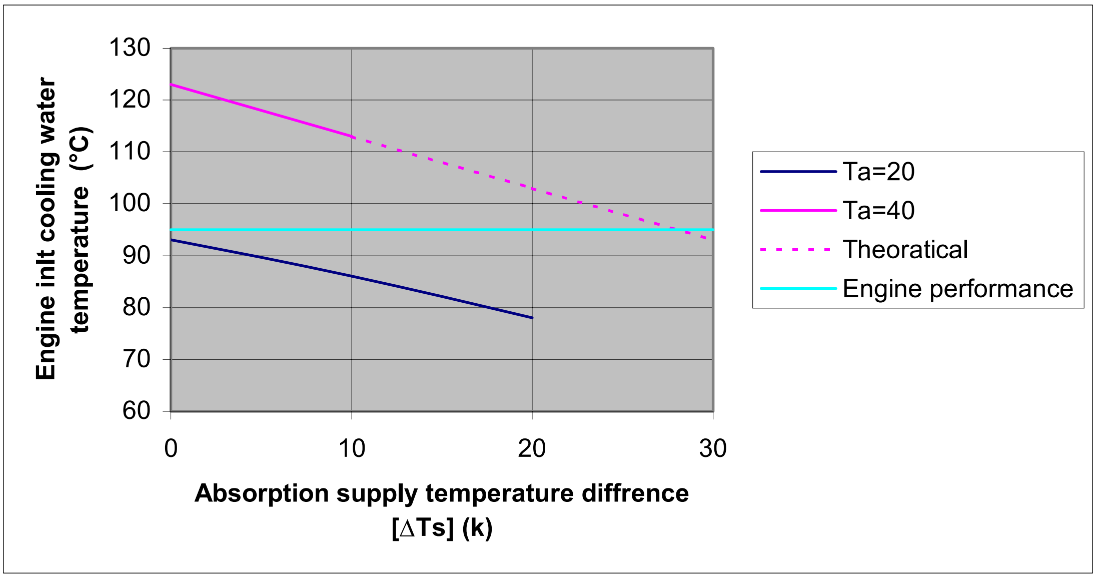


Figure 5.29 restrictions of used single-stage absorption cooling machines for a selected engine performance

Figure 5.30 shows that the range of performance of single –stage absorption cooling machine as a function of ambient temperature. From this figure , it can be obtained that at maximum COP only the machine operated in range of 22 °C (Alexandria winter). The range of ambient temperature increased to 28°C by decreasing the supply temperatures of 10 k and reach to 35 at supply temperatures decreasing of 20 k. the problem of the last to cases is the decreasing of COP, therefore increasing the primary energy used. This increased of primary energy can be obtained from **figure 5.31.** This figure shows that the increased of both supply temperatures and engine input cooling water increasing the primary energy. It can be obtained from this figure that the increase of engine cooling input temperatures of 10 k increasing the primary energy about (1.5 and 4.5 %) at ambient temperatures of (40 and 20 °C) respectively. The increase of temperatures difference of 20 °C increase this ratio to (16 and 20%) respectively.

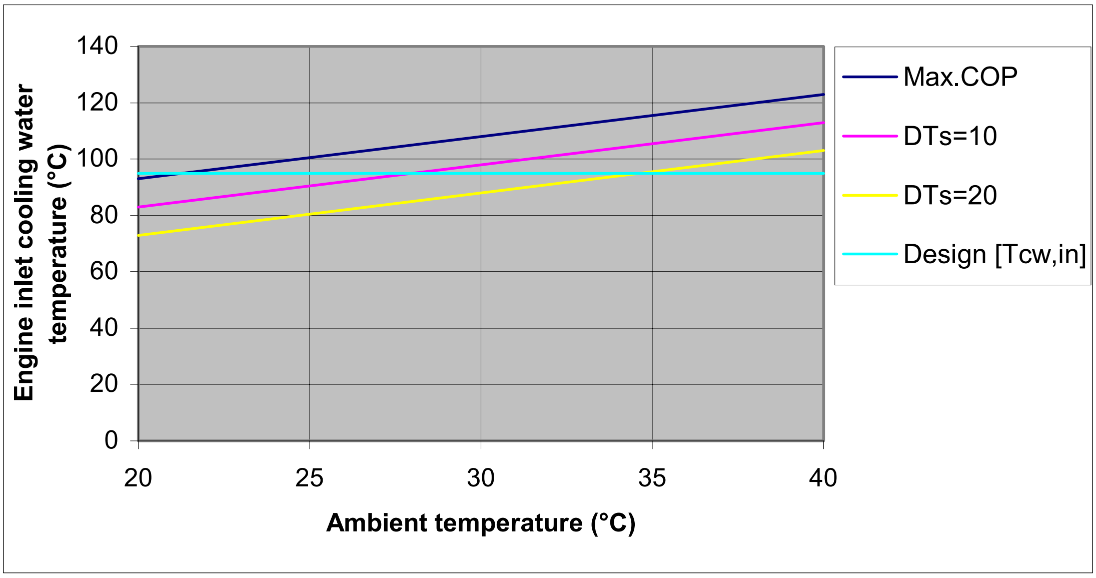


Figure 5.30 Range of performance of single-stage absorption cooling unit affected on changing supply temperatures.

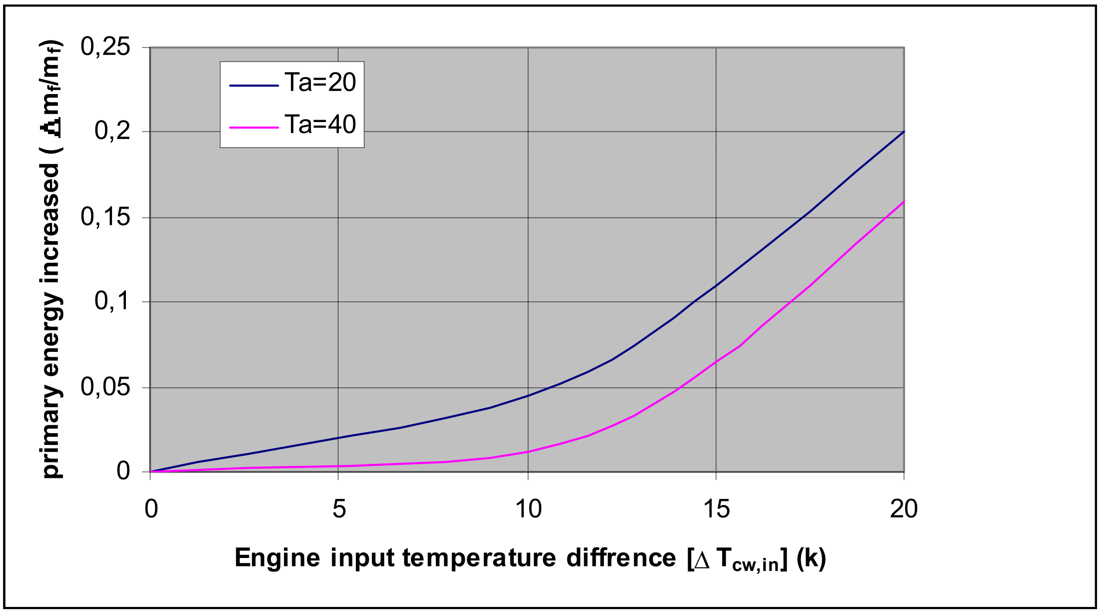


Figure 5.31 Effect of change input temperature cooling water on primary energy utilized in engine.

From the last figures ,it can be concluded that the single stage range is the low ambient temperatures, which is not practical to used in Aswan or Cairo, that is the introduction to study the another type two –stage absorption cooling machines.

It is aimed here to investigate the behavior of the single-stage cooling refrigeration units with Diesel Engines . In order to get the possibility to compare results obtained with two-stage absorption cooling units .The variation of cooling load from 200 to 400 kW concept will be applied. The electric load and fuel consumption will be discussed . **Table 5.2** summarizes the values of the electric engine load and the fuel consumption against the variation of the ambient temperature. Considering the numerical values of both electric engine load and fuel consumption of single stage absorption cooling units in table 5.2 the ambient temperature of 30 °C (Alexandria summer and Aswan winter)was used as a reference case in comparisons in the next sections.

Table 5.2 was discussed the affect of the ambient temperature for both fuel consumption and cooling- electric load ratio ($S_{c,o}$). It can be noticed that the increasing of the cooling load from 200 to 400 kW increasing the fuel consumption from 0.026 to .051 kg/s for the reference case Ta=30 °C .On the other hand, at constant fuel consumption 0.040 kg/s indicates that the increasing of the ambient temperature from 20 to 40 °C decreasing the cooling load from 320 to 300 kW. It can be also concluded that the increasing of the ambient temperature decreases the performance of the single-stage cooling unit and the performance of the single-stage is better in winter rather than summer. In this table, it was take in account that the case of Aswan summer (with maximum electric load of 590kW) as reference to compare with the another conditions, the reference system efficiency of ,55 was taken in account.

T_a (°C)	$P_{el,o}$ (kW)	$S_{c,o}$	m_f (kg/s)	Δ m_f (Reference system CogaS)
20 (Alexandria winter)	568	1,622	0,04544	,0009
25	572	1,634	0,04576	,0007
30(Alexandria summer and Aswan winter)	573	1,637	0,04584	,0007
35	575	1,643	0,0460	,0006
40 (Aswan summer)	590	1,686	0,0472	**Reference value**

Table 5.2 Results for single-stage absorption cooling unit at constant cooling load 350 kW and maximum COP operation.

The study of two-stage absorption cooling units were discussed in chapter 4 shows that the range of the coefficient of performance located between 0.25 and 0.50 and the advantages of this types are the use of low supply temperature which located between 55 and 100 °C .

The varying operating conditions chosen for the Diesel and absorption cooling units are namely: ambient temperature which varying from 20 to 40 °C , absorption cooling units types which two or single-stage and supply temperature of absorption cooling units .Besides, the coefficients of the absorption cooling units and Diesel engines presented in chapter 2 ,where the ambient temperature 30 °C (Aswan winter and Alexandria summer) is considered to be the reference temperature. Based on computer results illustrated, it can be noticed that at cooling load 350 kW the electric load demand varying from 425,5 kW at 20 °C to 540 kW at 40 °C . In addition the fuel consumption varied from 0,034 kg/s at 20 °C to 0,0432 kg/s at 40 °C. **Table 5.3** summarizes both the fuel consumption and electric load varied from winter to summer in Aswan and Alexandria. At constant electric Diesel engine load, it can be maintained that the cooling load decreased from 342 to 240 kW , when the ambient temperature increased from 20 to 40 °C . In addition , the performance of absorption cooling units load depend on the ambient temperature. The reference value of single –stage ACU (590 kW engine electric load) was be taken here too.

T_a (°C)	$P_{el,o}$ (kW)	$S_{c,o}$	m_f (kg/s)	Δ m_f (Reference system CogaS)
20 (Alexandria winter)	425,5	1,21	0,0340	,0066
25	435	1,24	0,0340	,0062
30(Alexandria summer and Aswan winter)	450	1,29	0,0360	,0056
35	525	1,50	0,0412	,0026
40 (Aswan summer)	540	1,54	0,0432	,0020

Table 5.3 Results for two-stage absorption cooling unit at constant cooling load 350 kW and 90 °C supply temperature

Table 5.3 was discussed the affect of the ambient temperature for both fuel consumption and engine electric load. It can be noticed that the increasing of the cooling load from 200 to 400 kW increasing the fuel consumption from 0.026 to .051 kg/s for the reference case Ta=30 °C .On the other hand, at constant fuel consumption0.040 kg/s indicates that the increasing of the ambient temperature from 20 to 40 °C decreasing the cooling load from 320 to 300 kW. It can be also concluded that the increasing of the ambient temperature decreasing the performance of the single-stage cooling unit and the performance of the single-stage is better in winter rather than summer.

The establishment of single and two-stage absorption cooling units in what concerns the fundamental principles for the determination of coefficient of performance and specification to suit a current study systems for a specified Areas (Aswan and Alexandria) has urged simulation to demonstrate both performance and temperature difference whose merits overweigh those of absorption cooling units . the interest was extend too to the investigation of comparison between single and two-stage absorption cooling units. It is aimed here to illustrate the comparison between the two named both fuel consumption and electric load.

When absorption cooling units are used for cooling depend on the engine losses energy, attention should be paid to the types of absorption cooling units .When covering the demand cooling load of 350 kW, the use of single-stage increased the electric engine load from 450 to 580 kW and the fuel consumption from 0,031 to 0,045 kg/s. On the other hand, the constant electric engine load at 400 kW illustrate that the cooling load at single-stage absorption cooling unit but at two-stage absorption cooling unit is 320 kW. **Figure 5.31** represent s the affect of change the ambient temperature more than the design temperature. From this figure it can be notified that the increase the ambient temperature more than the designed temperature decreases the cooling load. At Design temperature of 40 C° , there is no affect while it is the designed temperature. At design temperature of 30 °C , the affect appeared when the ambient temperature increases more than 30 °C, for example the increasing of ambient temperature to 40 °C decreases the cooling load to 85 % of the designed load. The same affect can be obtained at ambient temperature of 20 °C , the increasing the ambient temperature to 40 °C decreases the cooling load of 78 %. This affect should be taken in account when selected the Absorption cooling machines. **Figure 5.32** illustrates the same affect discussed in the last figure but for fuel effected by change the ambient temperature rather than the designed temperature . From this figure it can be seen that the affect of decreasing the ambient temperature decreases the fuel consumption ratio. The decreases the ambient temperature from 40 to 20 °C decreases this ratio to 72%.

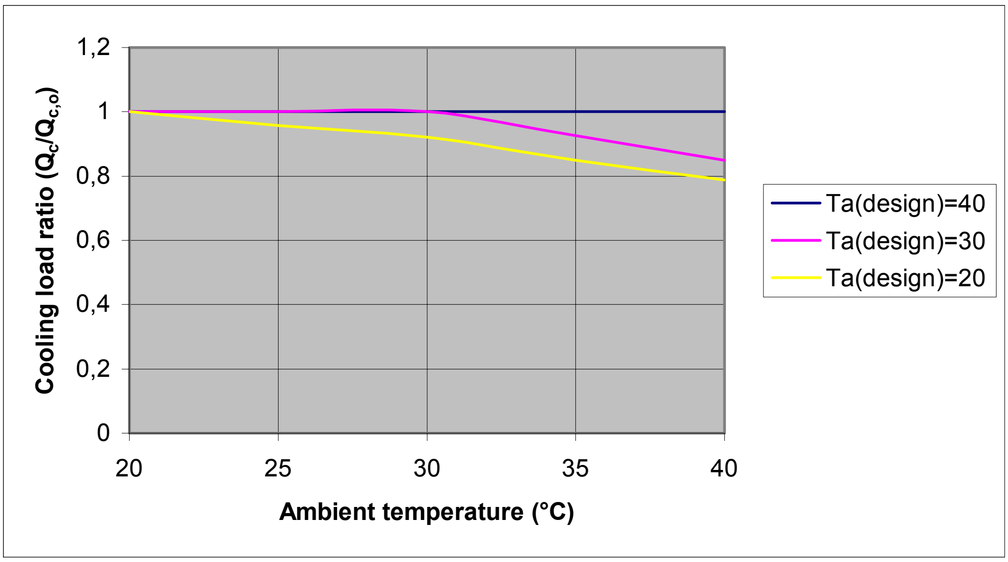


Figure 5.31 Operation of two-stage ACU depend on design temperature at constant cooling load (Supply temperature of 90°C)

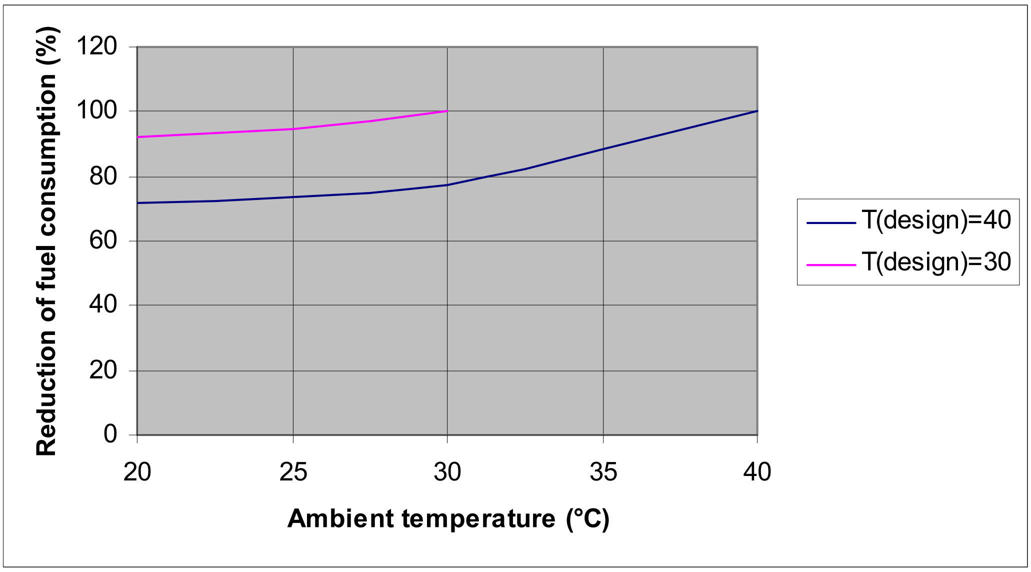


Figure 5.32 Reduction of fuel consumption depend on design temperature at constant cooling load

Chapter 6

Solar Energy Systems

6.1 Thermal solar energy utilization

Solar radiation incident upon the Earth is primary energy source by which the life of mankind has developed. The basic concepts of the utilization of solar radiation for the use of the present day's energy economy should be considered: to heat ,to electric ,to chemical energy carriers. Approximately 50 % of the gases responsible for global warming consist of carbon dioxide CO_2 .carbon dioxide is released during the combustion of fossil fuels such as coal, oil and gas. It can be notified that CO_2 ratio for the different fossil fuel as:

322.8 for coal energy source, 258.5 for oil energy source and 178 g/kWh for natural gas energy source The potential of solar energy systems for reducing the CO_2 emission in the conversion of primary energy to electrical power and refrigeration is strongly dependent on parameters like the conversion method, the efficiency of the power station and the coefficient of performance of the refrigeration system. Carbon dioxide is currently to be for more than 50% of the greenhouse effect [58,59].

Only CO_2 is considered above in relation to primary energy, but there are other emissions such as NO_x, SO_2, CH_4, particulate solids, nuclear waste ,etc, which also should be considered when used the solar energy systems. The data obtained in 2001 by IEA heat pump center for the approximate CO_2 emissions and efficiency of electricity generation for 1995-1996 can be concluded that the fossil energies should be decreased in the next decades.

Throughout the world in the current time and since 1980 , technology development will be done to utilized renewable energy. In the foreground to use both solar energy and wind energy and from the economic point of view, it is important to utilized of renewable energy. Renewable energy industries are covering numerous energy sources and scale of operation. While the increase in electricity generation may be significant in the past performance, the relative scale of renewable energy generation will be quit small growing. On the other hand , productivity of this renewable energy and their capacity should be taken in account in the foreground.

6.1.1 Power Generation

There are many ways of converting solar radiation to electrical power or heat energy. The different forms of utilized solar energy are shown in **figure 6.1.**

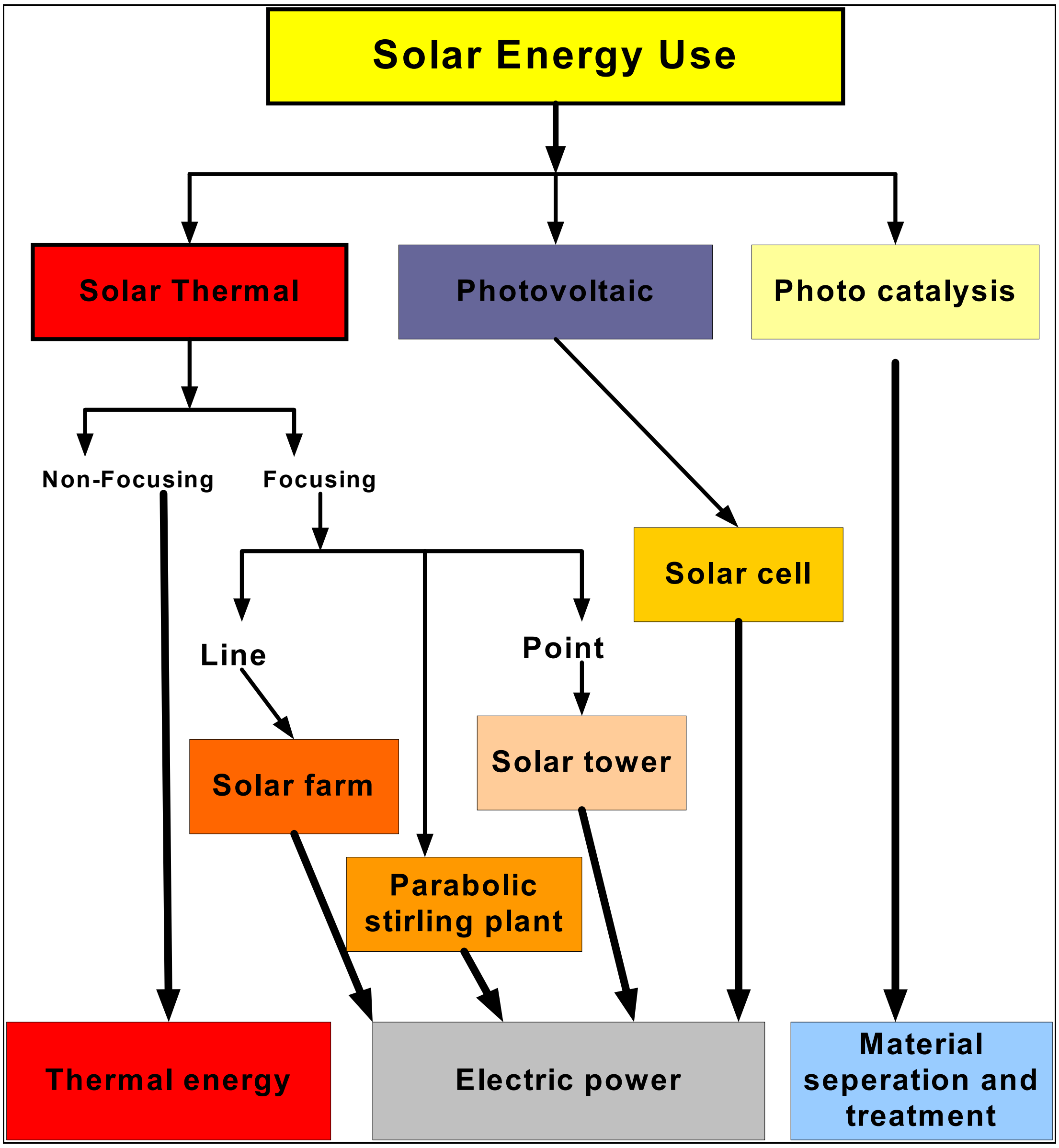


Figure 6.1 Energy chart of Solar energy used

6.1.1.1 Solar power-photovoltaic

Only a few photo voltaic dimension system in the MW range were built in the last decade **table 6.1**. The number of a new system will increase continuously within the next years resulting in decreasing costs [53].

The generation of electric power directly from the sun is possible in two ways:

- Solar cells using photoelectric effect (photovoltaic)
- Concentrators and heat engines (solar-thermal)

Both systems have their advantages and disadvantages ,the advantages of photovoltaic can be summarized as:

- Use both diffuse and direct radiation
- Long life time, maintenance-free

The disadvantages of the photovoltaic can be summarized as:

- Very expensive and low efficiency and Storage only in expensive and bulky batteries

Place of PV	Installed capacity	Start of operation
Toledo, Spain	1.0 MW	1994
Serre, Italy	3.3 MW	1994
Munich, Germany	1.0 MW	1998
Herne, Germany	1.0 MW	1999
Tudela, spain	1.2 MW	2001
Relzow, Germany	1.5 MW	2001
Relzow, Germany	3.5 MW	2002

Table 6.1 Examples of photovoltaic systems in the MW range.

6.1.1.2 Solar thermal power plants

Although solar thermal electricity is much more reasonable than photovoltaic electricity no more commercial power plants have been erected since 1991. The situation for solar thermal power plants is similar, although series production can reduce the electricity costs significantly below 10 eurocents/kWh. Due to the higher direct solar irradiation in the south they are most useful in southern Europe and north Africa where their potential is very high [52]. The advantages of the solar-thermal can be summarized as [54,55]:

- High efficiency with respect to photovoltaic
- No storage needed

The disadvantages of solar –thermal can be summarized as:

- Use only direct radiation
- Tracking necessary

The efficiency ranges for both photovoltaic and solar thermal power plant are 0.10 and 0.25 respectively[60-62] , for both system the electricity production is more expensive and this will be discussed in the economic section later.

The cost of solar energy should be estimated, the cost of power generation from solar systems still more expensive than the traditional types of power generation as shown in **table 6.2.**

Solar energy form	Investment cost Euro/kW	Electricity costs Euro/kWh
Solar thermal power plant	(3-5)*10^3	(0,12-0,20)
Photovoltaic	(5-15)*10^3	(0,30-0,70)

Table 6.2 Investment and electricity costs of difference solar energy systems

6.1.2 Heat generation

Solar radiation incident upon the Earth is primary energy source by which the life of mankind has developed. The basic concepts of the utilization of solar radiation for the use of the present day's energy economy should be considered: to heat ,to electric ,to chemical energy carriers [50]. Of the various types of solar thermal systems, solar collectors, photovoltaic and solar power plants are represents in last years.

The useful heat is related to flow rate, specific heat at constant pressure, and inlet and outlet temperature to the system. The efficiency may depend on many factors, collector temperature, ambient temperature, isolation, flow rate and solar radiation rate. In order to characterize a collector, one must therefore specify carefully the conditions under which the efficiency has been measured or calculated. In practice it is much easier to measure the temperature of the heat transfer fluid in the collector than the temperature of the absorber surface [50-52].

6.1.2.1 Active Solar heating systems

Several methods for collection and conversion of solar radiation into heat are feasible. Concentrating configuration in combination with liquids have been favored concepts in most of thermal solar plants which exist today. The type of the heating systems used depend upon the used of solar radiation, to use the all radiation direct and diffused solar collectors used but in the case of concentration the direct radiation only will be used. The high temperature heat achievable with radiation concentration is useful immediately for industrial process heat purposes, and is attractive for driving chemical reactions. Furthermore, efficient thermodynamic conversion becomes feasible in high temperature regimes, and prime movers can be used which are established technology in conventional power plants. There are basically three types of solar collectors:

1. Flat plate collector is still the most common type of collector

2. Evacuated-tube collectors, there are two types of this collector glass-glass or glass-metal. The first one is widely used rather than the second.

3. Concentrating collectors are used for high temperature range.

The assembly of collectors, consisting of either point-focusing or non-concentrating collectors are summarized in **table 6.3.**

Collector type	Concentration factor	Temperature	Power range
Flat plate collector	1	< 200 °C	<1.0 MW (thermal)
Solar pond	1	< 80 °C	<5.0 MW (thermal)
Solar chimney	1	< 150 °C	<2.0 MW (electric)
Vacuum tube collector	3	< 250 °C	<1.0 MW (thermal)
Parabolic trough collector	40-80	< 400 °C	<100 MW (electric)
Parabolic dish collector	100-250	< 1500°C	<150 kW/unit (thermal)
Heliostat collectors	200-700	< 1500°C	<200 MW (electric)

Table 6.3 Principle methods of solar radiation collection

6.1.2.2 Passive solar energy

Passive solar heating and cooling does not depend on pumps or fans or any other systems. It relies on the natural ebb and flow of the energy of the sun through a building. In order to obtain a substantial contribution from the sun and from persons in the building, three important elements were included in the plan:

1. small losses of energy
2. planned solar insulation
3. temperature equalization

the house was placed on the site in such a manner that was not shaded during the spring or autumn. Passive solar heating is improving with the following [56]:

1. house location
2. house orientation
3. the effect of heavy structure on energy consumption

It should be notified that when implementing solar energy features into building renovation, some of the solar technologies have toward becoming cost effective, because of high initial investment costs. It is important that solar concepts applied in building renovation serve other purposes besides energy saving.

Passive systems generally reliable, easier to maintain, and possibly longer lasting than active systems are often less expensive than active systems, but are also generally less efficient due to slower water flow rates through the systems.

6.2 Thermodynamics of solar collectors

Many concentrator types are possible for increasing the flux of radiation on receivers are summarized in **table 6.3**. From this table, it can be shown that, with concentration factor less than 3, total irradiation is converted into thermal energy or electricity. With concentration ratio more than 500, high temperature well above 1000 °C are achievable. Furthermore, efficient thermodynamic conversion becomes feasible in high temperature ranges, and prime movers can be used which are established technology in conventional power plant [50,57].

A simple flat plat collector consists of an absorber surface, a trap for radiation losses from absorber surface, a heat transfer medium such as air, water, etc.; and some thermal insulation behind the absorber surface. Flat plate collectors are used typically for temperature requirements up to 75 °C although higher temperatures can be obtained from high efficiency collectors. This collectors are of two basic types based on heat transfer fluid:

- Liquid type: where heat transfer fluid maybe water, mixture of water and antifreeze, or oil
- Air type: where the heat transfer medium is air

The air type used manly for drying and space heating requirements.

The main types of liquid solar collectors can be summarized as:

- Coated flat plat collector
- Simple flat plat collector
- Evacuated collector

Figure 6.2 shows the comparison among three types of collectors, flat plate collector, coated collector and vacuum collector. from this figure it can be illustrated that the evacuated tube collector can be used in the case of high temperature difference more than 140 °C because of the most expensive cost. Therefore, in the low temperature difference, the simple or coated flat plate collector are favorite, cause of the less expensive cost. It can also be notified that for the temperature difference less than 40 °C the simple flat plate is considered but by increasing the temperature difference rather than 80 °C the collector is useless.

At temperature difference of 60 K, the range of the current study in the next chapters, the coated collector efficiency are 0.46 and the simple flat collector are 0.25. For this reason, in the current study, the coated flat collector are be considered.

The economic analysis of the different types of collector will be discussed in the next sections, besides the cost analysis of the different solar systems illustrated in this chapter.

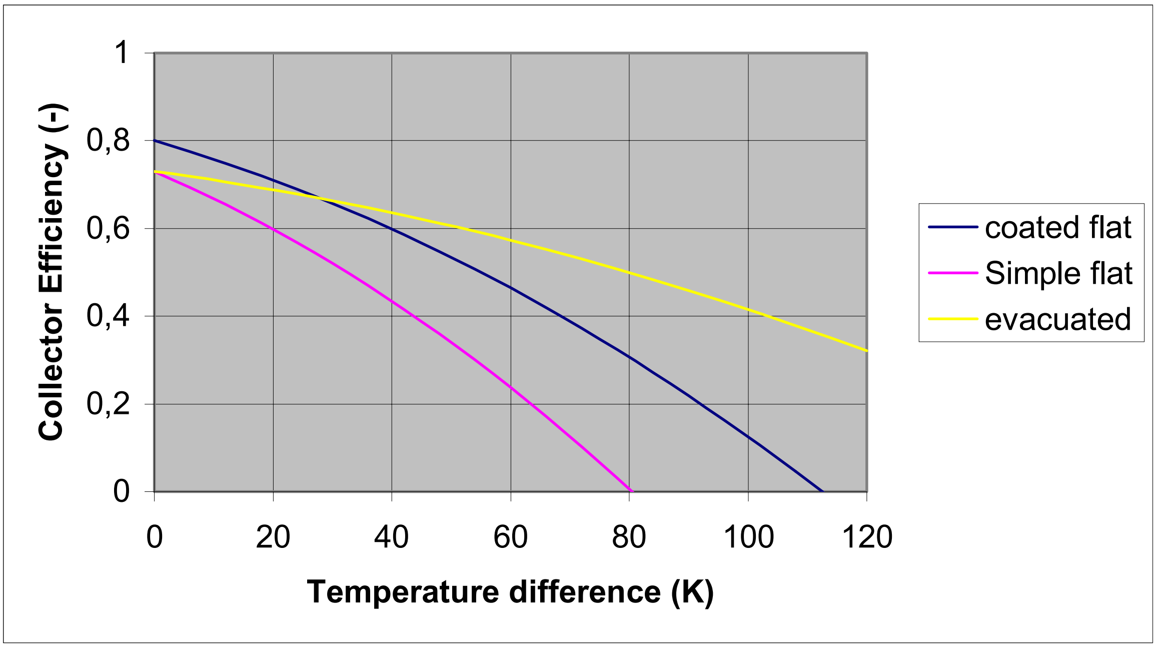


Figure 6.2 Performance of different collectors at solar radiation of 1000 W/m^2

The flat plate collector and the coated flat collector with concentration factor of 1.0 can be simulated in this section. In order to determine the efficiency of a solar collector, the rate of heat transfer to the medium fluid must be calculated, the rate of heat transfer to the fluid through a collector depends on the temperature of the collector surface from which heat is transferred by convection to the fluid, and the heat transfer coefficient between the collector and the fluid. The absorption heat of the solar collector can be written as:

$$Q_{abs} = m_w * Cw * (T_{out} - T_{in}) \tag{6.1}$$

In order to construct a model for thermal analysis of a flat plate collector ,the following assumption will be taken in our consideration [55]:

- The collector is thermally in steady state
- The temperature drop is negligible
- Heat flow is one-dimensional through both the cover and the back insulation
- The irradiation on the collector plate is uniform
- The headers connecting the tubes cover only a small area
- The sky can be treated as through it were a black-body source for infrared radiation at an equivalent sky temperature.

The efficiency of the collector is simply the ratio of the useful energy delivered to the total incoming energy. The collector efficiency in a function of solar radiation, convection losses and radiation losses can be written as:

$$\eta_{coll} = \eta_0 - (q_{rad} + q_{con}) / I_0 \tag{6.2}$$

q_{rad} can be written as:

$$q_{rad} = \sigma * E * (T_{abs}^4 - T_a^4) \tag{6.3}$$

where E is a function of collector temperature difference (δt_{abs}) which equal to

$$\delta t_{abs} = T_{abs} - T_a \quad \text{and} \tag{6.4}$$

σ =5.7605*10^{-8} (W)

The convection collector losses can be written as:

$$q_{conv} = \alpha_{conv} * (T_{abs} - T_a) \tag{6.5}$$

where η_0 can be written as:

$$\eta_0 = \frac{Q_{abs}}{I_0 * A_{coll}} \tag{6.6}$$

The collector heat losses consists of convection, radiation and conductance, in order to obtain an understanding of the parameter determining the thermal collector efficiency, it is important to develop the concept of collector heat balance . The collector heat balance can be written as:

$$Q_{sol} = Q_{abs} - Q_{rad} - Q_{conv.} - Q_{tr} \tag{6.7}$$

Where Q_{sol} is the total solar energy radiation on the solar collector, can be written as:

$$Q_{sol} = I_0 * A_{coll} \tag{6.8}$$

Limits of solar thermal collector efficiency and operating temperature are discussed in section 6.1 but the concentration ratio and the range of used power are shown in **table 6.3.**

Figure 6.3 shows that the collector efficiency against the temperature difference of coated flat plat at different solar radiation range from 500 to 1000 W/m^2. It can be notified that from this figure, The same collector efficiency can be obtained at the difference solar radiation but at high temperature difference range, for example, at collector efficiency of 0.40, the temperature range increased from 40 to 70 K by increasing the solar radiation from 500 to 1000 W/m^2.

At the current study the temperature difference range will be around 60 k the collector efficiency covered by this collector are 0.13, 0.24, 0.32, 0.38, 0.42, and 0.46 at the solar radiation coefficient of 500, 600, 700, 800, 900, and 1000 W/m^2 respectively . this means that the decreasing of the solar radiation from 1000 to 500 W/ m^2 decreases the efficiency from 0.46 to 0.13. Therefore the performance of the solar collector reduced by 70%. For this reason, it should be add the solar collector to fossil energy systems, in the current stud y the solar collector add to the diesel engine power plant.

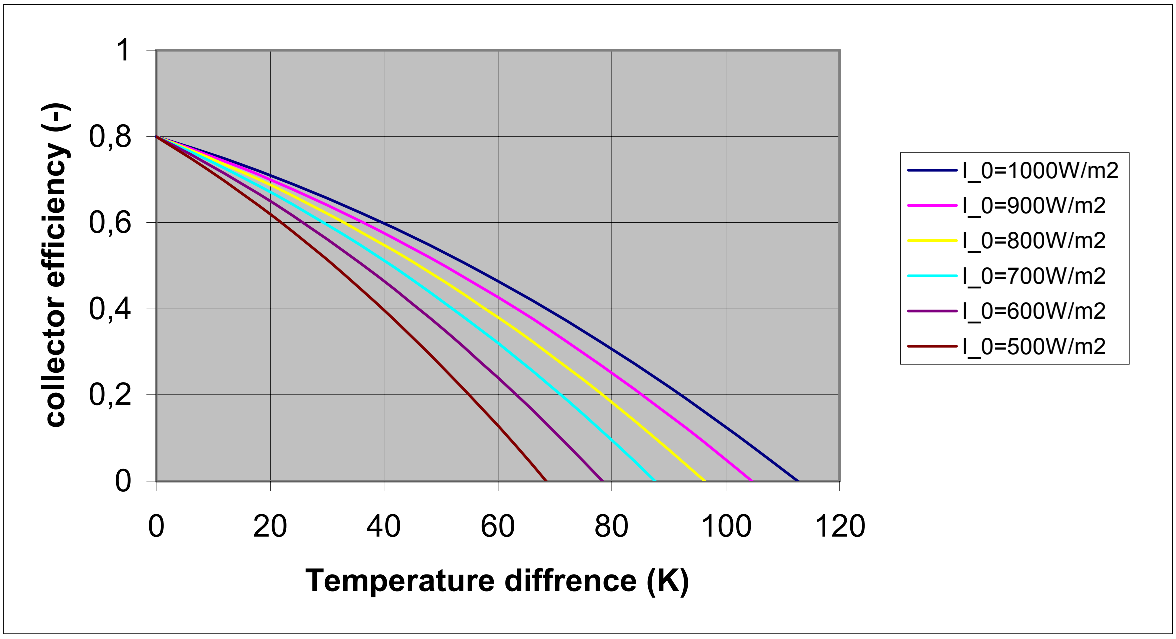


Figure 6.3 Performance of coated flat plate collector at different solar radiation.

6.3 *Thermal solar utilization systems*

Solar energy storage is an important term in the future utilization of solar energy. Without storage, solar energy has to be used as soon as it is obtained. Solar energy cannot be stored in electromagnetic form. It must be converted to another form. **Figure 6.4** summarized the possible forms of solar storage, the used form in the current study is the heat in storage systems at low or medium temperature. The solar heat as a function of ambient temperature was discussed for Hannover city [63] as an empirical equation which depend on building place. A solar heating system for production of domestic hot water consists of collector connected by a piping system to a storage tank. In the solar collector the solar radiance transferred to heat and absorbed by a liquid in the solar collector channel as mentioned in the previous sections [64]. This liquid transports the solar heat or the waste heat recovery heat through a piping system to the storage tank.

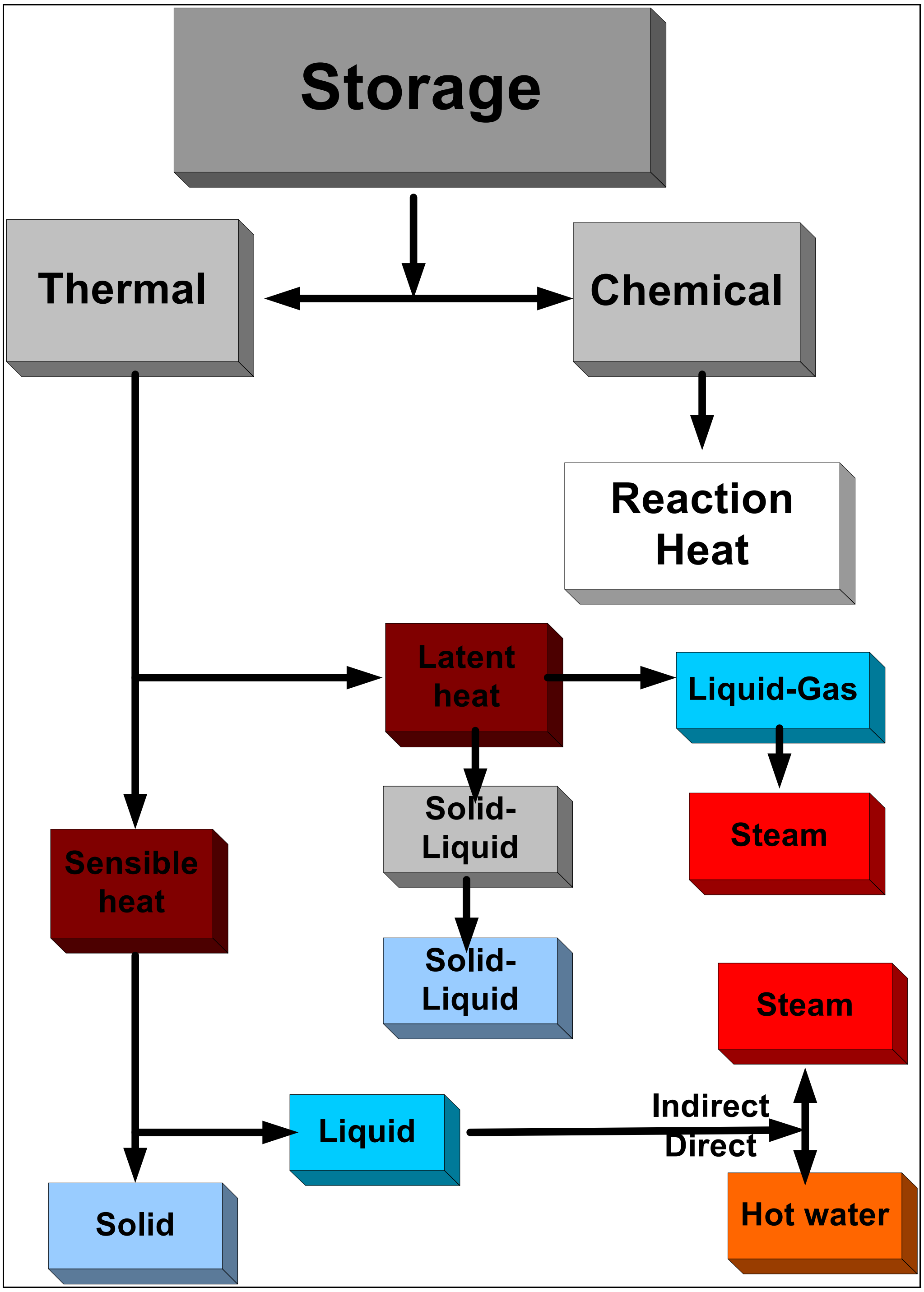


Figure 6.4 Different forms of storage heat energy

There are basically two ways of achieving this. **Figure 6.5** shows the first one, open loop system ,where all of the fluid in the tank runs through the panels. **Figure 6.6** shows the second way , closed loop systems use a heat exchanger to separate the fluid from the potable water and second pump to circulate the water in the storage tank .This systems are typically filled to pressure of about 2 bar . Because they are pressurized they can use smaller pumps than the open loop systems.

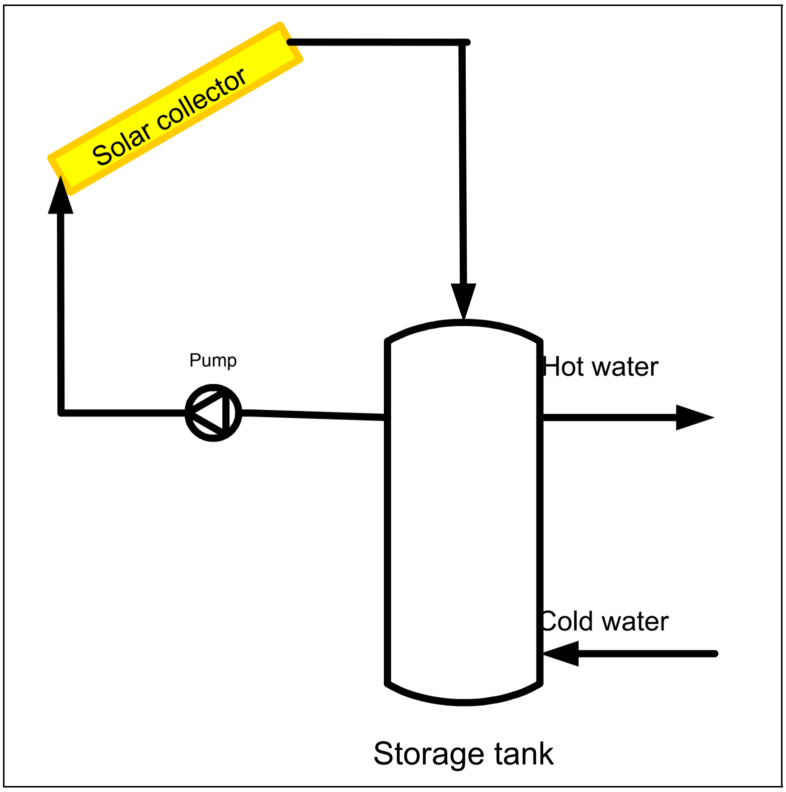


Figure 6.5 Open loop storage tank diagram.

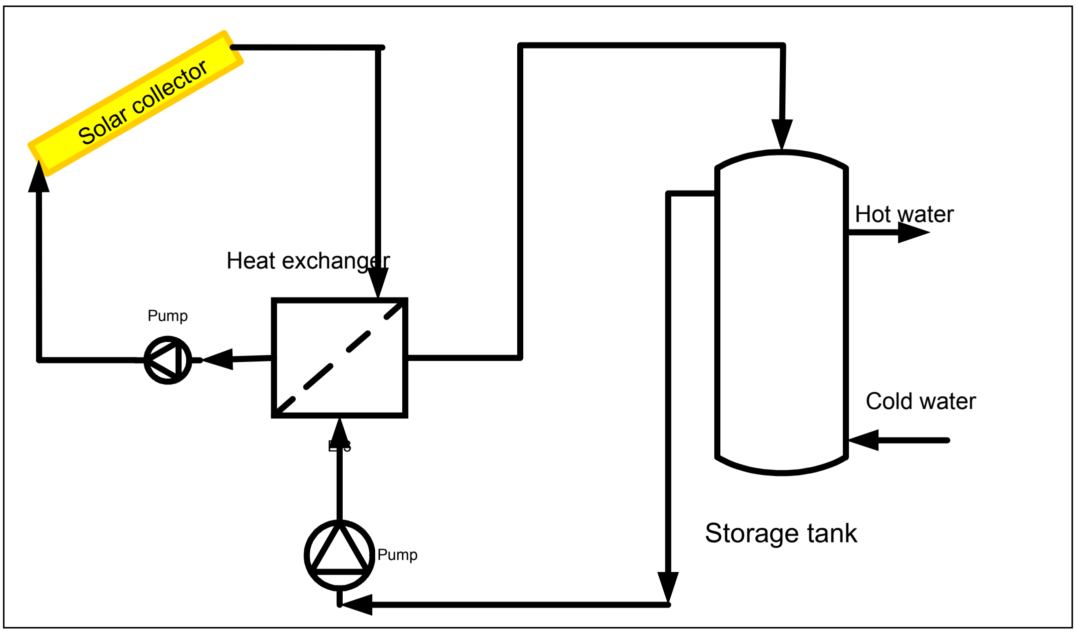


Figure 6.6 Closed loop storage tank diagram.

The different types of long term storage are used in Germany. The aquifer storage stores the heat in the under ground, supported by two wells and applicable only in approximate 5% of the regions in Germany. It is suitable for large volumes and relatively low temperatures [65-68].

The duct heat store, U-pipes are placed under ground and circulating water heats the soil. This storage system is suitable for temperature up to 80°C.

For the Gravel water storage a pit is excavated and lined ,this type of storage requires 50% more in the volume than pure water tank due to the lower heat store capacity of the gravel in comparison to water.

The most effective type of storage is storage in pure water, because of excellent heat storage capacity. Water is also cheap and presents no environmental hazard.

The main requirement for current study that water store range up to 95°C

6.4 Solar Energy and cold generation (ARU)

Water/ Lithium-bromide Absorption cooling units are most often operation in combination with heat and power generation in Germany, but can also be assisted by vacuum tube solar collectors to operate in range of temperature more than 80 °C. With coefficient of performance range from 0.5 to 1.2 for single and two stage type. Because solar cooling is based on thermally processes instead of the normal electrical cold production, the costs for the used heat plays a central role: a fundamental produced by fossil fuel systems or waste heat.[69,70]. **Figure 6.7** shows this basics of cold generation out of solar energy used absorption refrigeration systems. As mentioned before in chapter 4, the use of absorption refrigeration machines decreases the ozone affected. Furthermore , the use of solar energy to generate cooling load decreases carbon dioxide emissions.

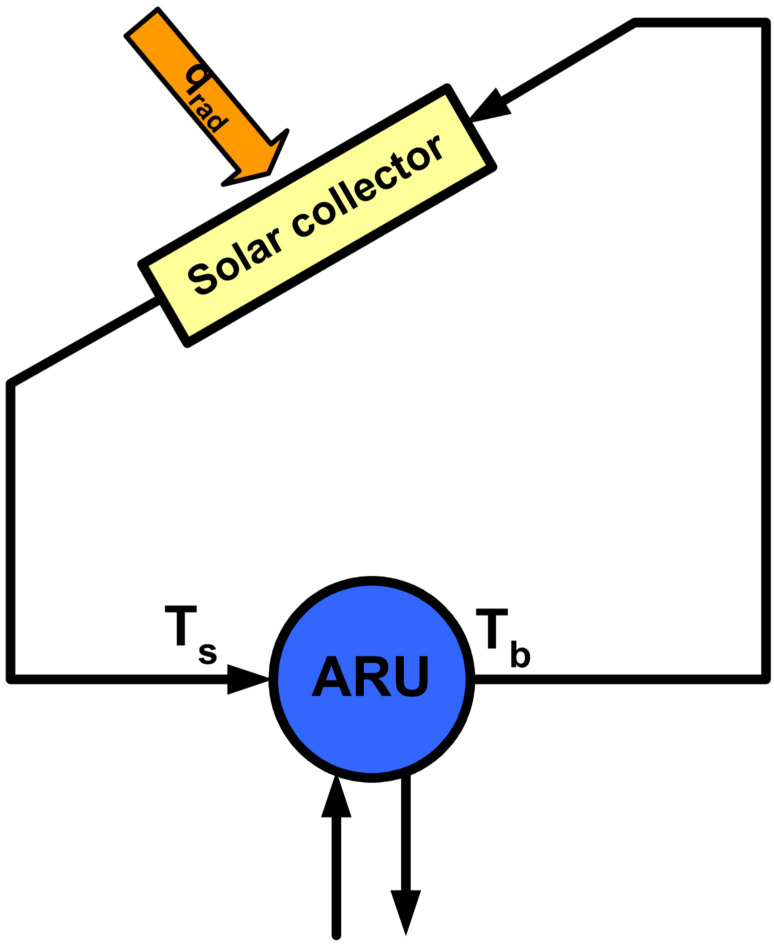


Figure 6.7 Cold out of solar Energy (ARU).

With the input of solar energy systems under special consideration of the heat, cooling and power industry, there are interesting aspects which are serviceable for extension of the co-generation , district heat and cooling supply.

It is important that in this section to study the specific collector area as a function of ambient temperature for both types of cooling systems.

Figure 6.8 -a represents the advantages of the use of two-stage absorption cooling units at low temperature 20°C, are the most advantages at the high temperature 40 °C at constant radiation factor 500 and 1000 W/m^2 respectively.

Figure 6.8-b represents the advantages of the use of single -stage absorption cooling units at low temperature 20°C, are the most advantages at the high temperature 40 °C at constant radiation factor 1000 W/m^2 respectively. From the last figure it can be concluded that the use of two-stage for low radiation and the single is more efficient for the high solar radiations.

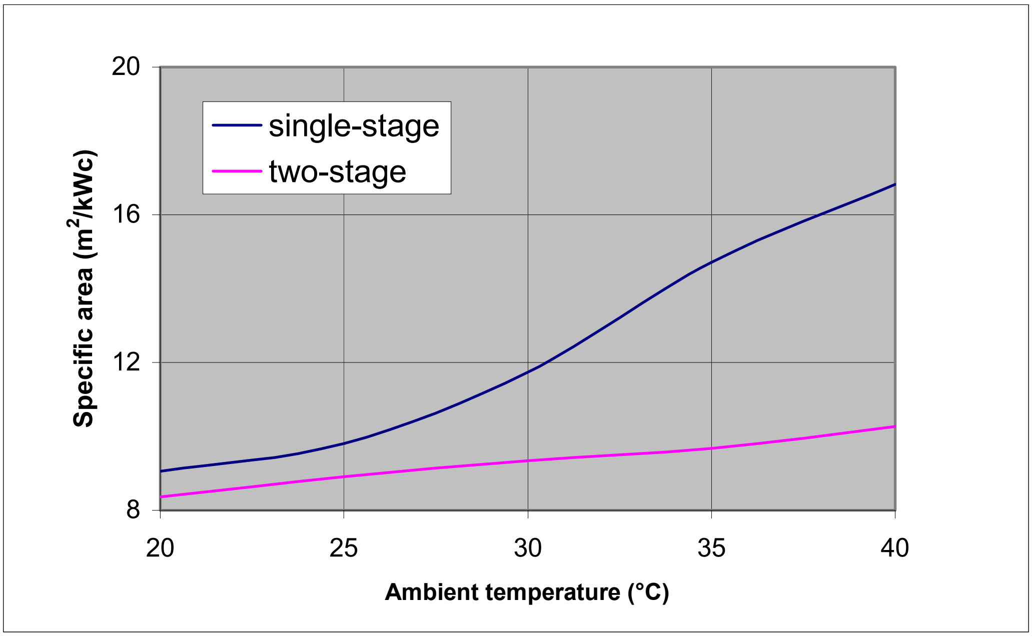


Figure 6.8-a Specific collector area Comparison between single and two-stage absorption cooling unit type at solar radiation 500 W/m^{2}.

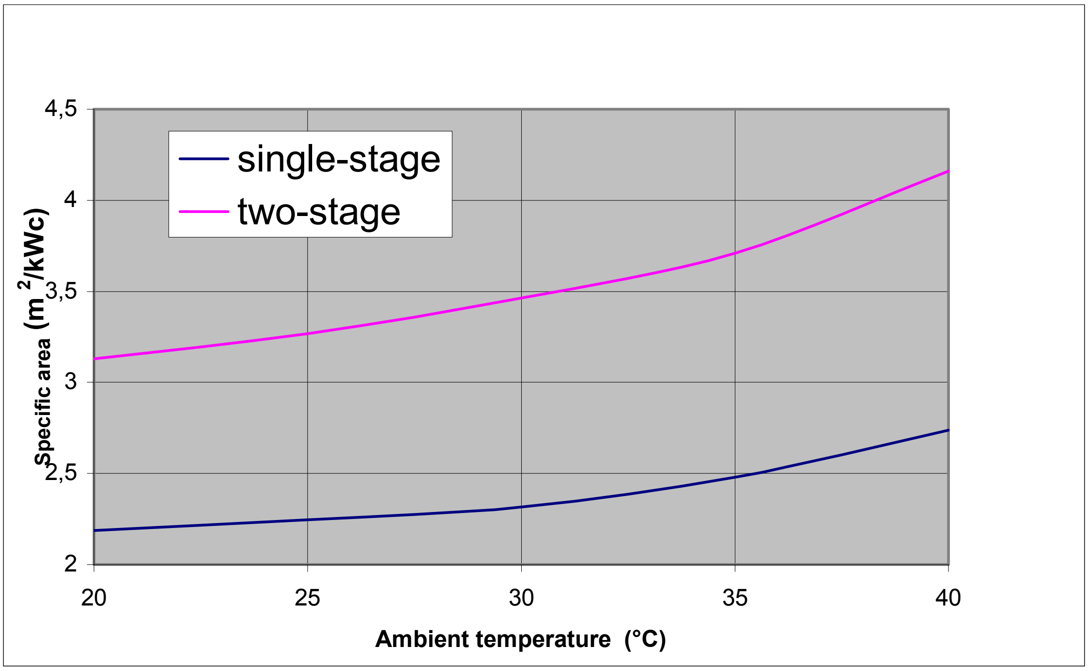


Figure 6.8-b Specific collector area Comparison between single and two-stage absorption cooling unit type at solar radiation 1000 W/m^{2}.

Figure 6.9 represents the specific collector area of single and two-stage absorption cooling units. The obtained result are for single-stage at maximum coefficient of performance and at constant supply temperature of 90 °C for the two-stage case. It can be notified that for the two-stage the increasing of the solar radiation from 500 to 1000 W/ m^2 decreases the specific collector area from 10.2 to 4.2 m^2/kWand from 8.3 to 3.2 m^2/kW at ambient temperature 40 and 20 °C respectively. It can also be illustrated that the decreasing of the ambient temperature will decrease the specific area at constant solar radiation. It can also be obtained the fuel saved in the reference system from this figure. It shows the advantages of use solar energy to reduced the fuel consumption at the reference system discussed in chapter 5. from this figure it can be obtained that the value of fuel saved in the reference systems according to the use of constant solar collector area of 500 m^2. the comparison is carried out for two-stage absorption refrigeration unit at supply temperature of 90 °C . the value of fuel saved obtained from **figure 5.2** for the case of combined gas and steam power station (CoGaS)as reference case. The collector area can be obtained from this figure with the help of chosen the absorption cooling unit.

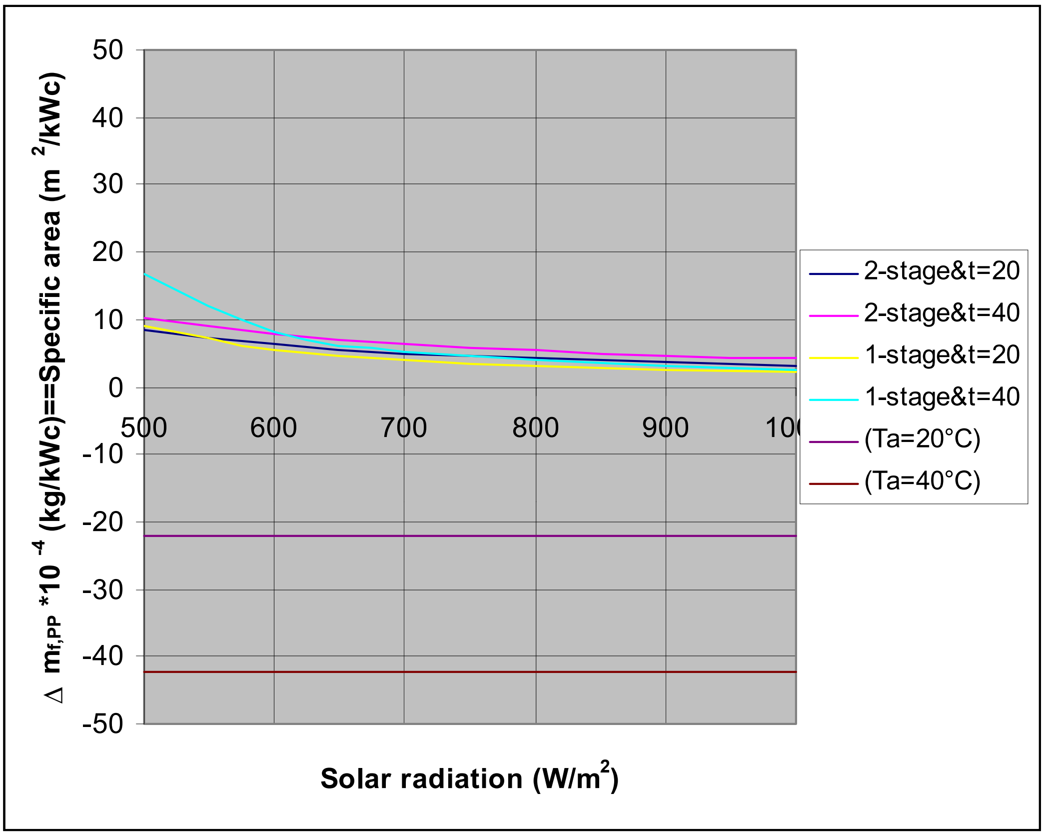


Figure 6.9 Solar energy affected on fuel saved at the Reference power systems at different ambient temperatures

Chapter 7

Combined Power- Cold Plant with Thermal Solar Energy - A parametric Analysis

7.1 General aspects

With the input of solar energy systems under special considerations of heat, cooling and power industry, there are interesting aspects which are serviceable for extension of co-generation , district heat and cooling supply. The addition of solar energy was tried to be maximum utilization value. **Figure 7.1** shows that the addition of solar energy decreases the utilized fuel which discussed in chapter 5.

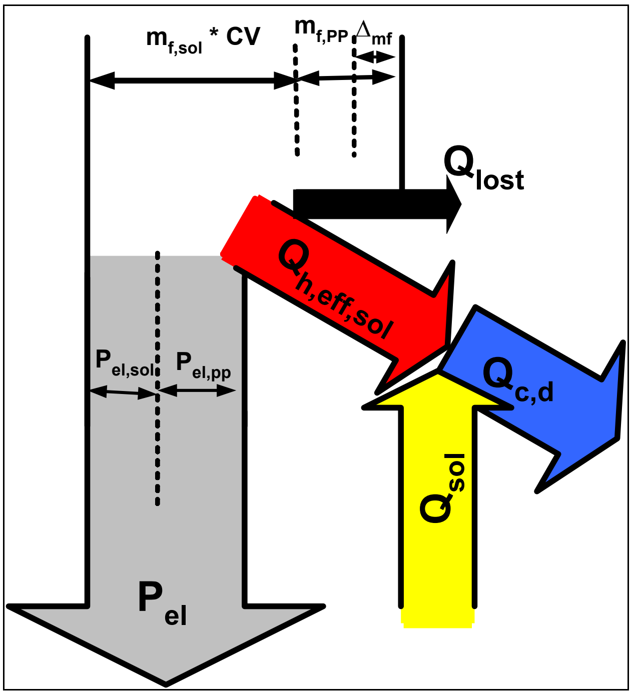


Figure 7.1 Sankey diagram of cold generation of both diesel engine and solar energy using absorption cooling machine (ARU)

The input of the solar thermal energy into combined diesel power system and absorption cooling units will be discussed in two cases : parallel connection and series connection. It is aimed in the next sections to investigate the advantages of using solar energy in diesel engine-absorption cooling machine illustrated in chapter 5 to obtain the useful range of solar energy to generate cooling load. The parallel and the series connections will be discussed in the next section respectively. The power-cooling ratio estimated in chapter 5 in equation 5.13,can be improved after adding the solar energy to the combined system and can be written as:

$$S_{c,sol} = \frac{P_{el,sol}}{Q_c} \tag{7.1}$$

where the $P_{el,sol}$ is defined as the engine electric load after adding the solar energy and the Q_c is the same cooling load estimated in chapter 5 in equation 5.13 as a function of effective engine heating. In this section, cooling load is defined as a function of both solar heat and engine effective heating and can be written as:

$$Q_c = (Q_{h,eff} + Q_{sol}) * COP_{aru} \tag{7.2}$$

The solar energy was estimated and discussed in equation 6.1. The heat balance after the addition of solar energy can be written as:

$$P_{el,sol} + Q_{h,eff} + Q_{lost} = m_{f,sol} * CV \tag{7.3}$$

Therefore, engine electric load after adding solar energy can be written as:

$$P_{el,sol} = (m_{f,sol} * CV) * \eta_{el,de} \tag{7.4}$$

The fuel utilized in diesel engine after adding solar energy can be written as:

$$m_{f,sol} = \frac{P_{el,sol}}{\eta_{el,de} * CV} \tag{7.5}$$

The reduction of electric load can be obtained from the reference system discussed in chapter 5, the fuel utilized in this reference system can be estimated directly from **figure 5.2 .**

Therefore, the reduction in primary energy can be obtained from this equation as:

$$\Delta m_f = m_f - m_{f,sol} - m_{f,PP} \tag{7.6}$$

The connection between diesel engine, absorption cooling unit and solar collector can be obtained in the following ways:

- Series A (Solar collector between cooling and exhaust heat)
- Series B (Solar collector before cooling heat)
- Parallel (Solar collector and engine heat parallel connection)

The last three ways can be illustrated in the next sections to obtain the maximum benefit of solar heat energy and minimize the use of primary energy. The analysis in this chapter will use two-stage absorption cooling machine to make use of the cooling water energy as concluded in chapter 4 and for the security reason for using the water energy at maximum temperature of 90°C. The use of single-stage made the cooling water inefficient to use. The heat gained in this case can be defined as:

$$Q_{heat} = Q_{h,eff,sol} + Q_{sol} \tag{7.7}$$

From the last equation , a new factor can be defined as the ratio of solar heat to the engine effective heat as:

$$SF = \frac{Q_{sol}}{Q_{h,eff,sol}} \tag{7.8}$$

The value of this factor illustrates the ratio of using solar energy, this factor could be ranged from (0 to ∞). The first case has been discussed in chapter 5 and the other case has been discussed in chapter 6. The other values of this case can be discussed in this chapter. To discuss this factor in determined values ,it can be written as:

$$S_{Ratio} = \frac{Q_{sol}}{Q_{h,eff,sol} + Q_{sol}} \tag{7.9}$$

This ratio can be varied from 0.0 to 1.0 as shown in the next figures.

Figure 7.2-a shows the engine electric load decreasing after the addition of solar energy to the engine and absorption cooling unit. From this figure, the ratio of solar energy to operate

the engine at a certain load can be estimated. **Figure 7.2-b** shows the affect of addition solar energy to the engine on the electric –cooling load ratio discussed in chapter 5.

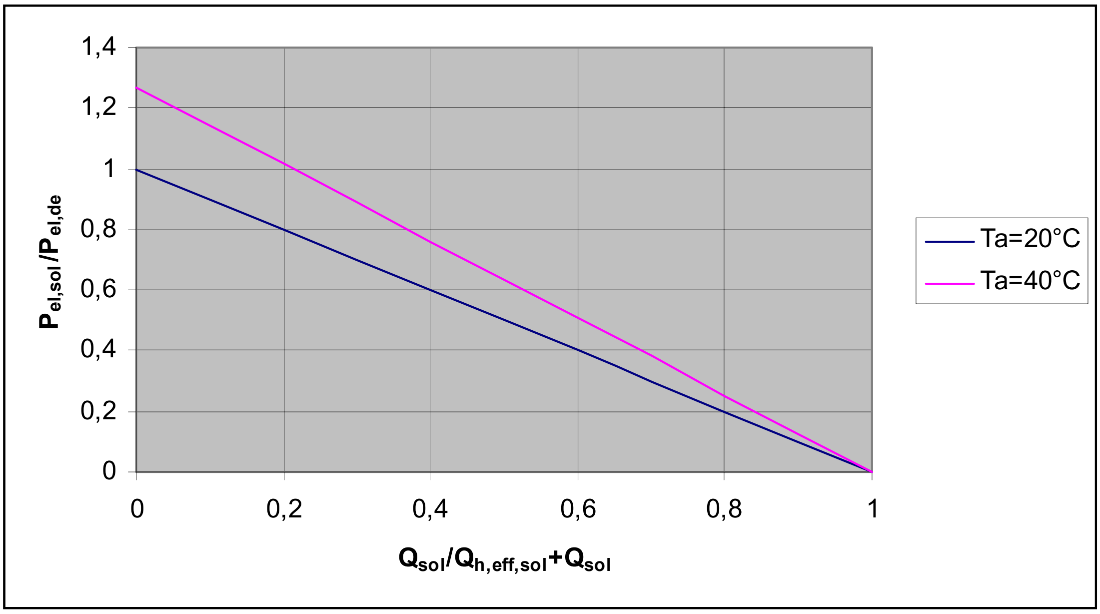


Figure 7.2-a Engine electric load after addition solar energy to the engine load to produce the same cooling load.

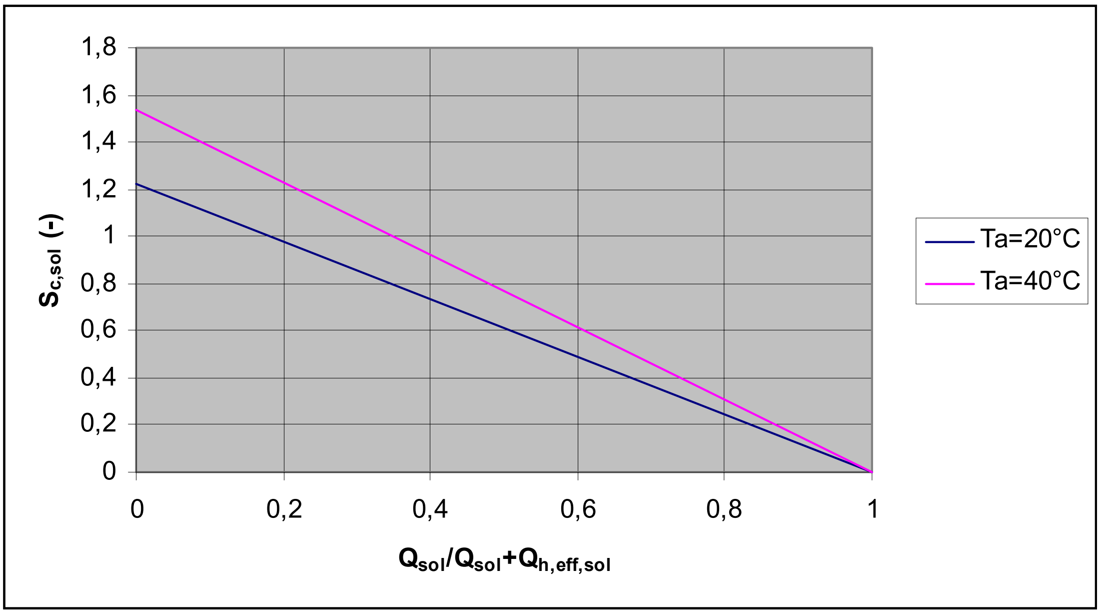


Figure 7.2-b Engine electric load –Cooling load ratio after addition solar energy to the engine load to produce the same cooling load.

Figure 7.3 shows the primary energy saved after adding the solar energy. This figure indicates the fuel difference shown in figure 7.1 after electric load corrections from the reference system discussed in chapter 5 in figure 5.2.

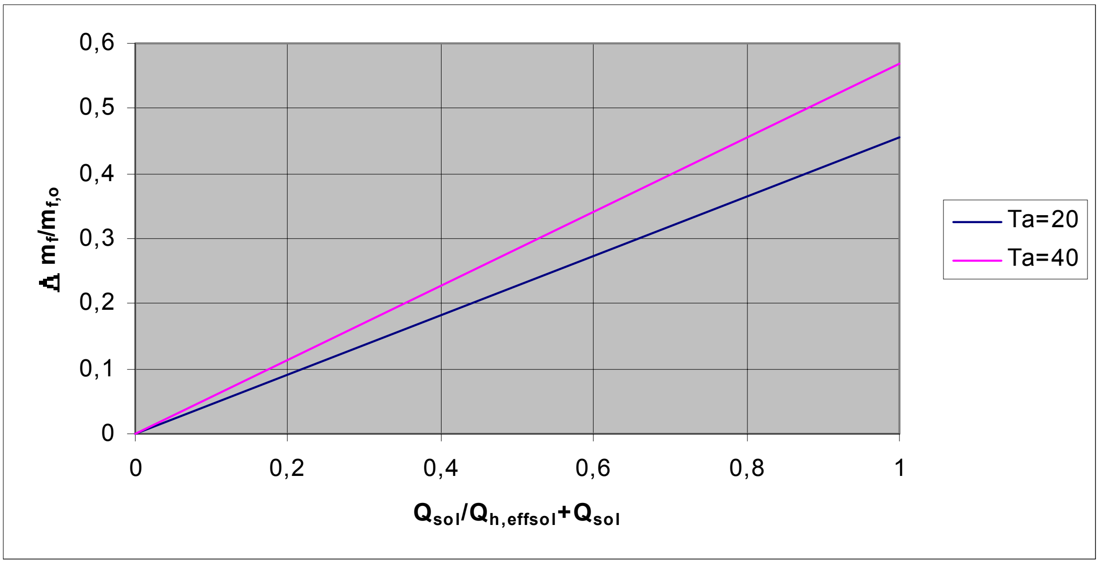


Figure 7.3 Fuel consumption saved after addition solar energy to produce the same electric and cooling load

7.2 Solar collector series connected to the Diesel engines

7.2.1 Series -A

In this section there are two connections will be suggested . The first illustrates that the solar collector maintained between the engine water cooling system and absorption cooling unit as shown in **figure 7.4**.

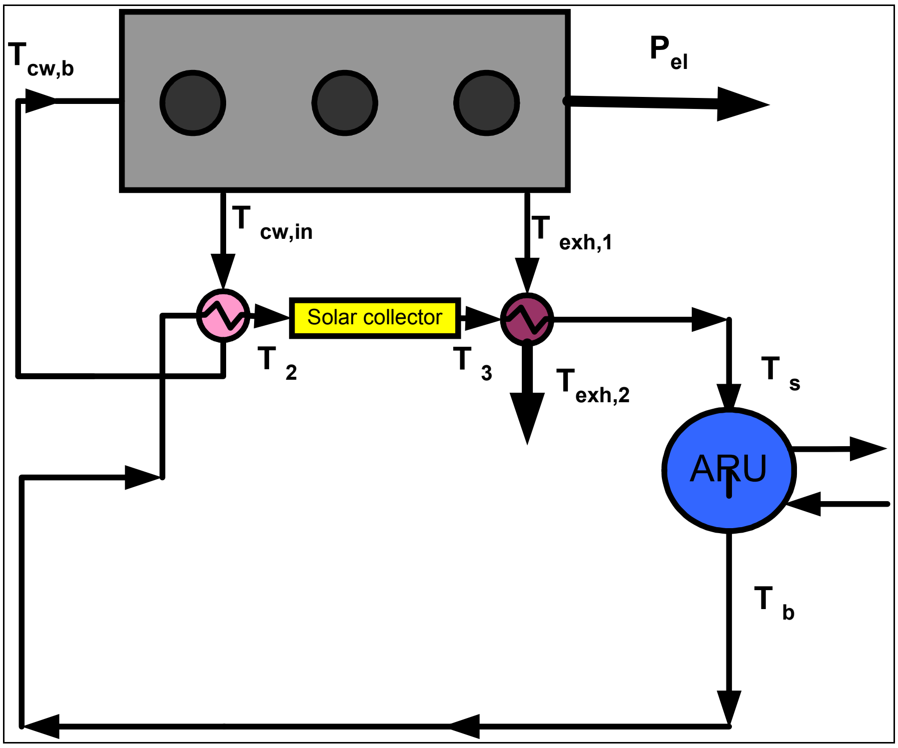


Figure 7.4-a Series-A connection between solar collector and Diesel engine connected with absorption cooling unit

From this figure, the absorber temperature in this case can be define as:

$$T_{abs} = \frac{T_2 + T_3}{2} \tag{7.10}$$

The solar heat gained in this case can be written as:

$$Q_{sol} = m_w * C_w * (T_3 - T_2) \tag{7.11}$$

The engine effective heat can be written as:

$$Q_{h,eff,sol} = m_w * C_w * (T_2 - T_b + T_s - T_3) \tag{7.12}$$

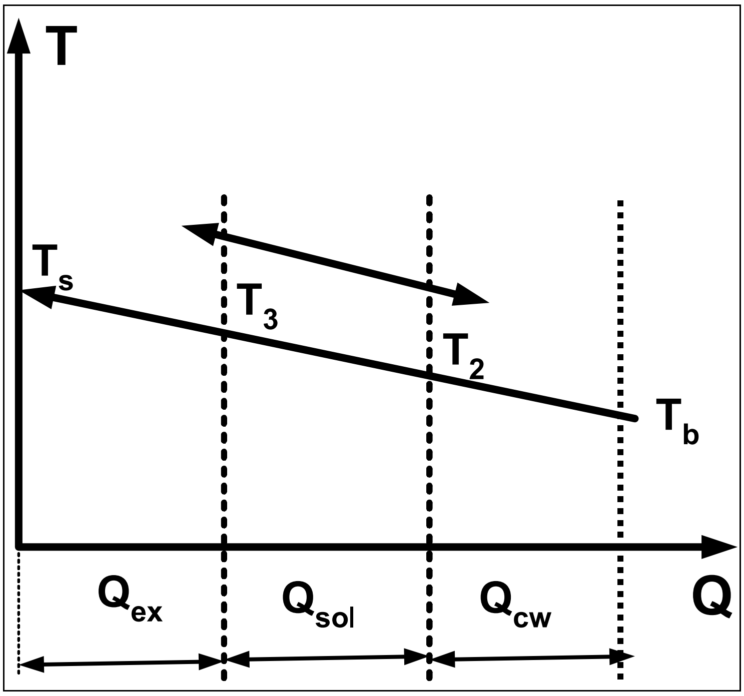


Figure 7.4-b T-Q diagram of Series-A connection between solar collector and Diesel engine connected with absorption cooling unit.

Figure 7.5 shows the collector efficiency of series –A connection at different solar ratios, this figure shows the solar collector efficiency at cooling to exhaust ratio of 1:1.

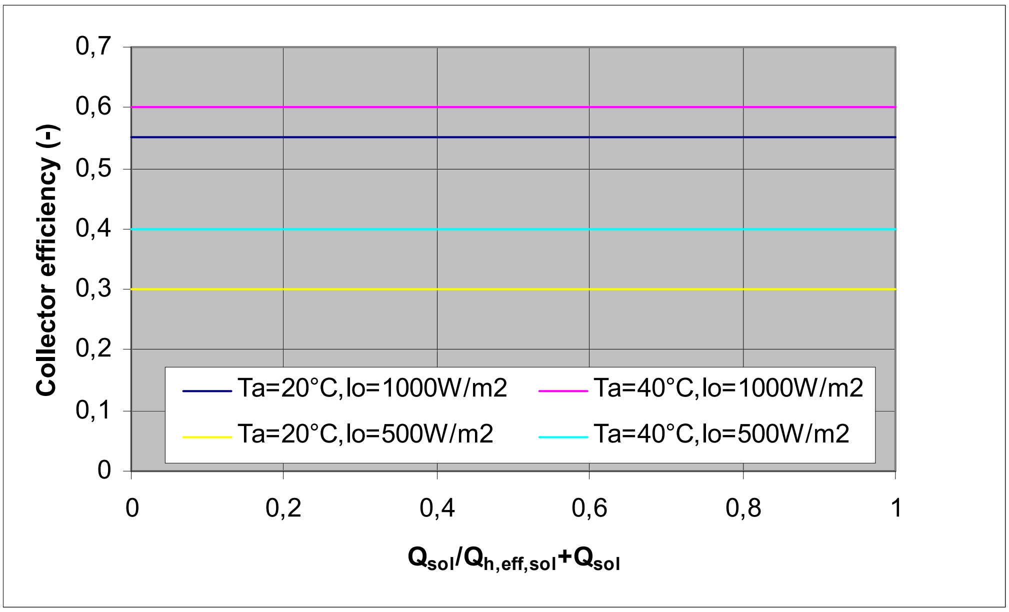


Figure 7.5 Solar energy ratio affected on solar collector efficiency at difference solar radiation and ambient temperatures

Figure 7.6 illustrates the ratio of collector area to cooling load related to this collector efficiency discussed in figure 7.5 at different solar radiation and ambient temperatures to obtain the collector area required to delivered the solar energy .

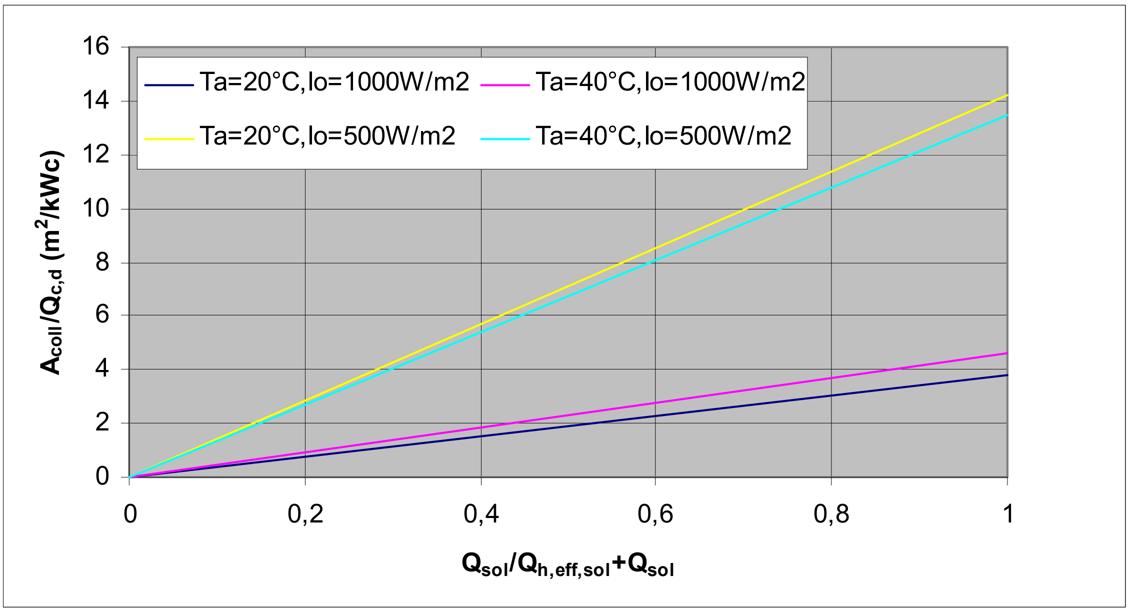


Figure 7.6 Solar collector efficiency affected on solar collector area to cooling load ratio at difference solar radiation and ambient temperatures.

The state of the art of absorption cooling units is introduced in chapter 4. it can be notified that two-stage absorption cooling unit are frequently used with diesel engine . The same results can be obtained from the previous section which introduced the advantages of use of two-stage absorption cooling units over the single-stage with and without solar collector. In chapter 5, that the use of two-stage rather than single stage which required more energy level of high temperature range was discussed.

As stated in chapter 5 ,the present work is concerned with the two-stage absorption cooling units rather than the single stage. For that reason, a computer results is developed for the simulation of two-stage absorption refrigeration units according to the presented solution in previous sections, it is extendable to make use of different collector area and solar radiation described in this section according to the engine used. The advantage of using the same engine models is the capability of varying the ambient temperature and the solar collector parameters ,expressed in the previous sections, during the simulations. Thus different parameters can be compared for specific fuel consumption and electric load reduction. **Figure**

7.7 shows the effect of solar energy to engine heat ratio on absorber temperature at differentengine heat ratios.

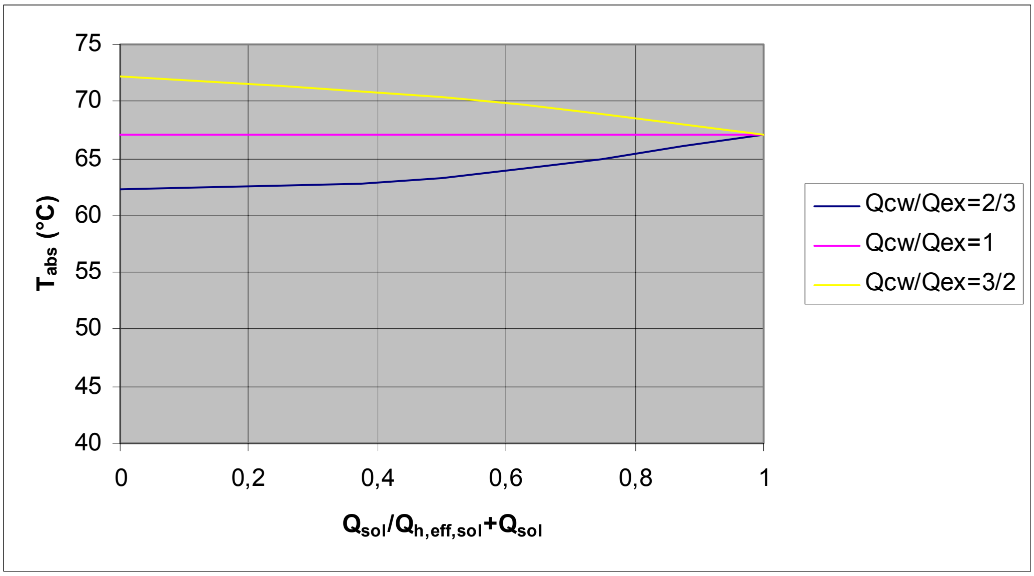


Figure 7.7-a Engine heat ratio affected on solar collector absorber temperature at 20 °C ambient temperature

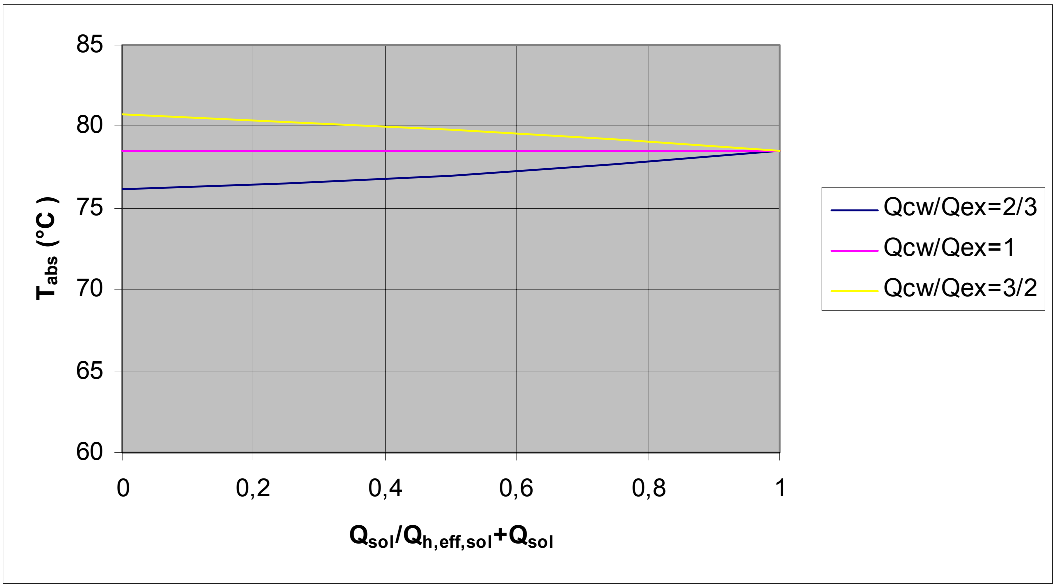


Figure 7.7-b Engine heat ratio affected on solar collector absorber temperature at 40 °C ambient temperature

Figure 7.8 shows the effect of solar energy to engine heat ratio absorber temperature of 40 °C, at this connection on the engine performance cooling water. From this figure it can be obtained that the engine average temperature can be operated at lower values than the designed one. This gives advantages of security to confirm that the engine cooling losses heat should be used in this case or operated the engine in other conditions as discussed in the last chapters. This effect increases with decreasing the ratio of cooling to exhaust engine heat as shown in figure 7.8-b, in this case the engine average temperature can reach to 70°C, which means long run of engine performance security to use the cooling heat. By increasing the cooling to exhaust ratio, the engine average temperature should be increased to reach 80 °C. However in this case ,it can be reduced the engine average temperature in range of 10 K.

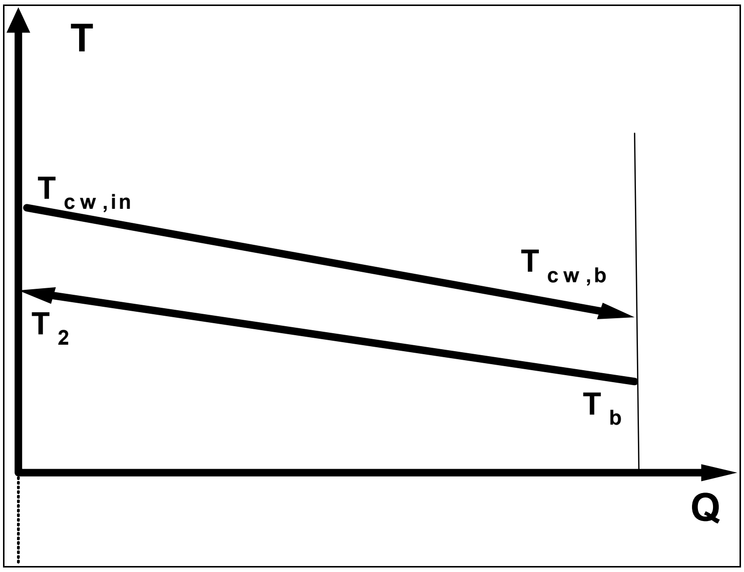


Figure 7.8-a Relationship between Engine cooling water and solar collector of series A.

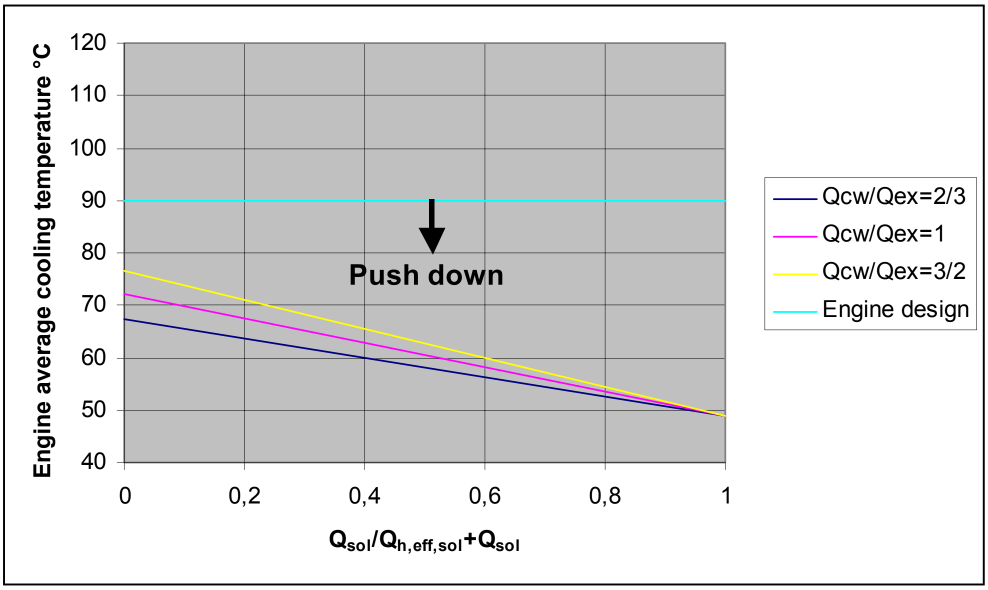


Figure 7.8-b Relationship between Engine average cooling water and solar collector absorber temperature at 20 °C ambient temperature.

7.2.2 Series –B

In this connection, the solar collector was maintained between cooling water and exhaust gas system as shown in **figure 7.9** , the solar energy temperature level should be estimated to obtain a more efficient system. The absorber temperature can be written as:

$$T_{abs} = \frac{T_2 + T_b}{2} \tag{7.13}$$

From figure 7.3, the solar heat gained in this case can be written as:

$$Q_{sol} = m_w * C_w * (T_2 - T_b) \tag{7.14}$$

The engine effective heat can be written as:

$$Q_{h,eff,sol} = m_w * C_w * (T_s - T_2) \tag{7.15}$$

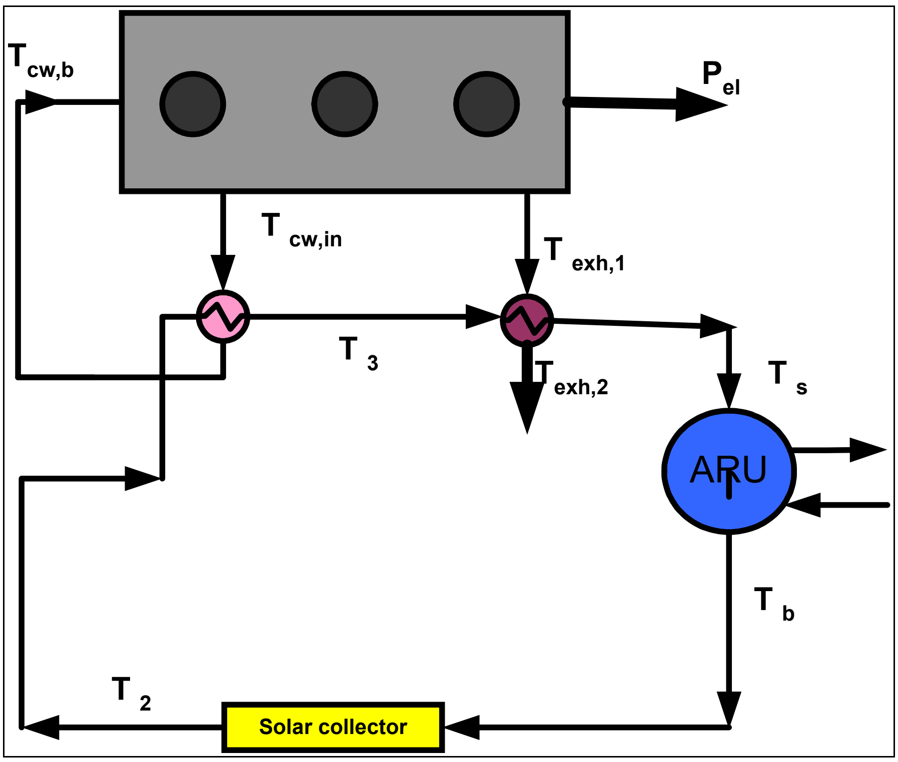


Figure 7.9-a Series-B connection between solar collector and Diesel engine connected with absorption cooling unit.

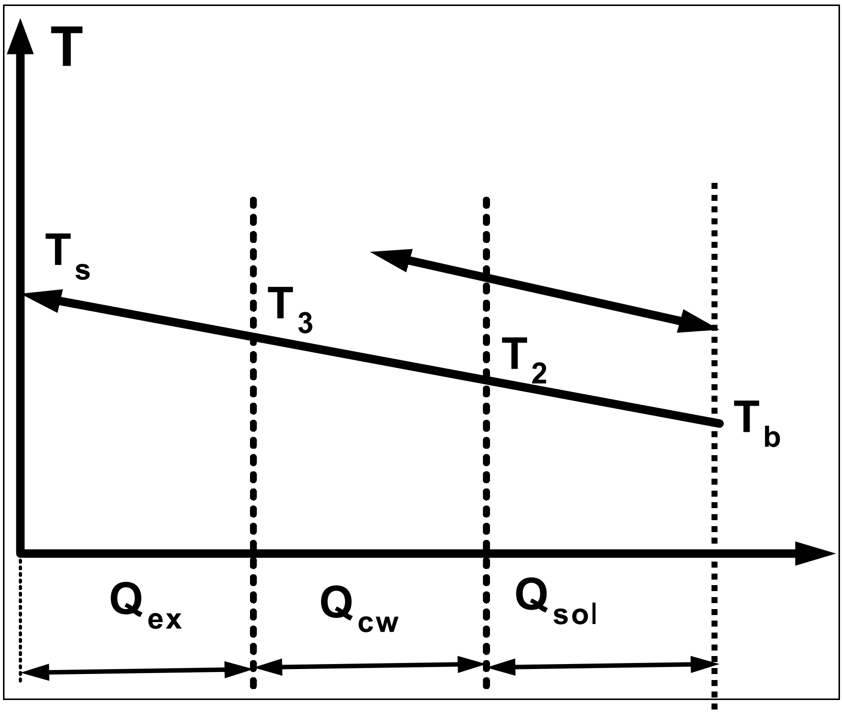


Figure 7.9-b T-Q diagram of Series-B connection between solar collector and Diesel engine connected with absorption cooling unit.

From **figure 7.9-b,** it can be shown that that the increased of solar energy ratio can be affected on the solar absorber temperature in the case of heat engine ratio of as mentioned in the next figures. The increasing of solar ratio increased the engine cooling water range and decreasing the use of cooling water heat. To obtain the best connections between engine and solar energy the same result obtained in the last section should be carried out in this section. For that reason , computer results of the two-stage absorption refrigeration units were simulated according to the presented solution in previous sections ,it is extendable to make use of different collector area and solar radiation described in this section according to the engine used. The advantage of used the same engine models is capability of varying the ambient temperature and the solar collector parameters ,expressed in the previous sections, during the simulations. Thus different parameters can be compared for specific fuel consumption and electric load reduction.

Figure 7.10 and **7.11** show the solar collector to cooling water demand as a function of both solar ratio and ambient temperature. From the comparison between this figure and the same figure in series –A ,it can be notified that the star and end points are the same while that is the same case discussed in chapter 5 and 6 respectively. From the last figure it can be concluded that at the fraction ratio of solar ratio series –A is more efficient than series -B in both collector efficiency and collector Area , therefore the performance of engine heat and engine fuel consumption.

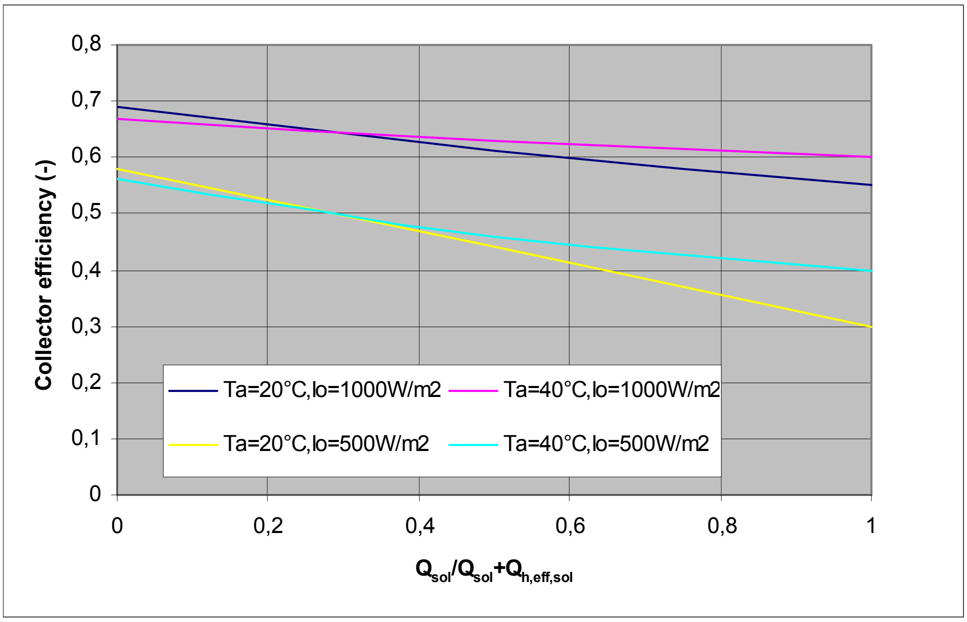


Figure 7.10 Solar energy ratio affected on solar collector efficiency at difference solar radiation and ambient temperatures

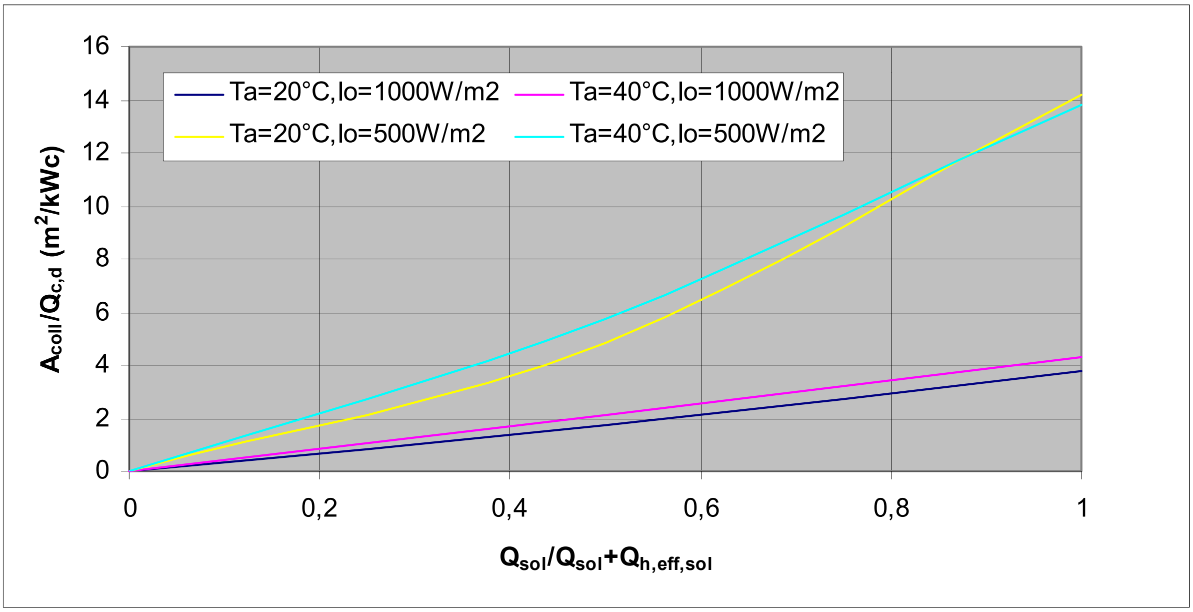


Figure 7.11 Solar collector efficiency affected on solar collector area to cooling load ratio at difference solar radiation and ambient temperatures.

Figure 7.12 shows that the series-A system is more efficient than series-B systems. It can also be notified that the increasing of the collector area decreases the use of cooling water. It is aimed here to estimate the critical solar collector area which cause the useless of the solar energy in this system connection between solar collector and diesel engine. The estimation carried out for two engine cooling water ranges namely (90/80) and (75/65).the second case cannot be used in this connections reason for the high temperature range required. The main reason of this disadvantage was decreasing the solar collector efficiency at high temperature range. It can be used this connection but with increasing the engine average temperature more than the designed vale as shown in this figure it should be pushed up. From this figure and the last two figures it can be obtained that the more the use of cooling water range the more the use of series –A connection. It can be compare this case with the third case in the next sections. This affect increases with decreases the ratio of cooling to exhaust engine heat as shown in figure 7.8-b, in this case can be the engine average temperature reach to 95°C this means the engine performance security to use the cooling heat is less efficient than the series -A. By increasing the cooling to exhaust ratio ,the engine average temperature should be increased reach to 100 °C but in this case ,it should be increased the engine average temperature in range of 10K.

From the last figure, it can be obtained that the use of this connection can be reduced the collector to cooling load ratio but the disadvantages was increasing the engine performance cooling water more than the designed value. Therefore, the use of engine and solar collector in series connection should be in the first form discussed in section 7.2.1 to obtain more security to use cooling engine heat in wide range of used.

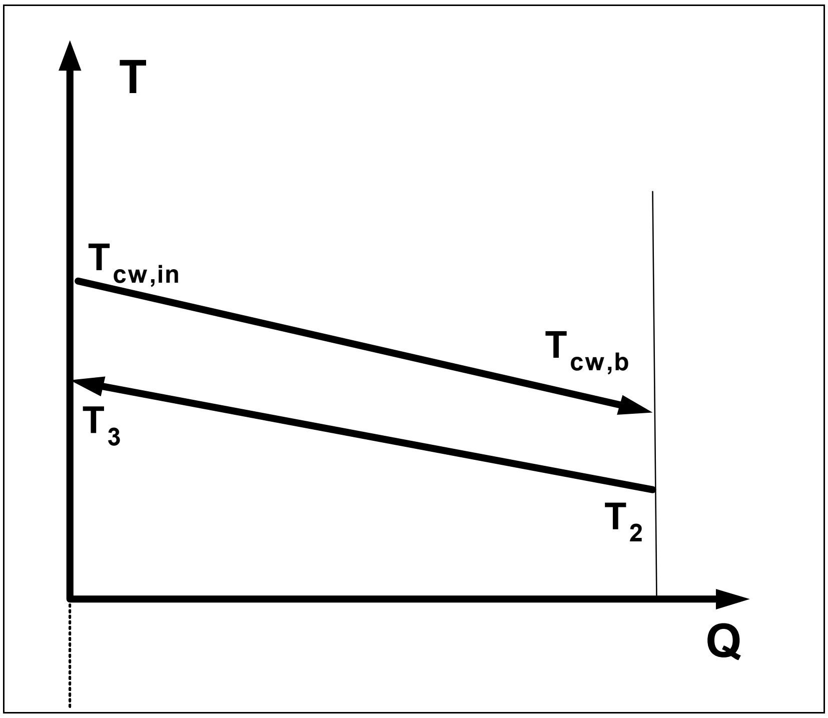


Figure 7.12-a Relationship between Engine cooling water and solar collector of series B

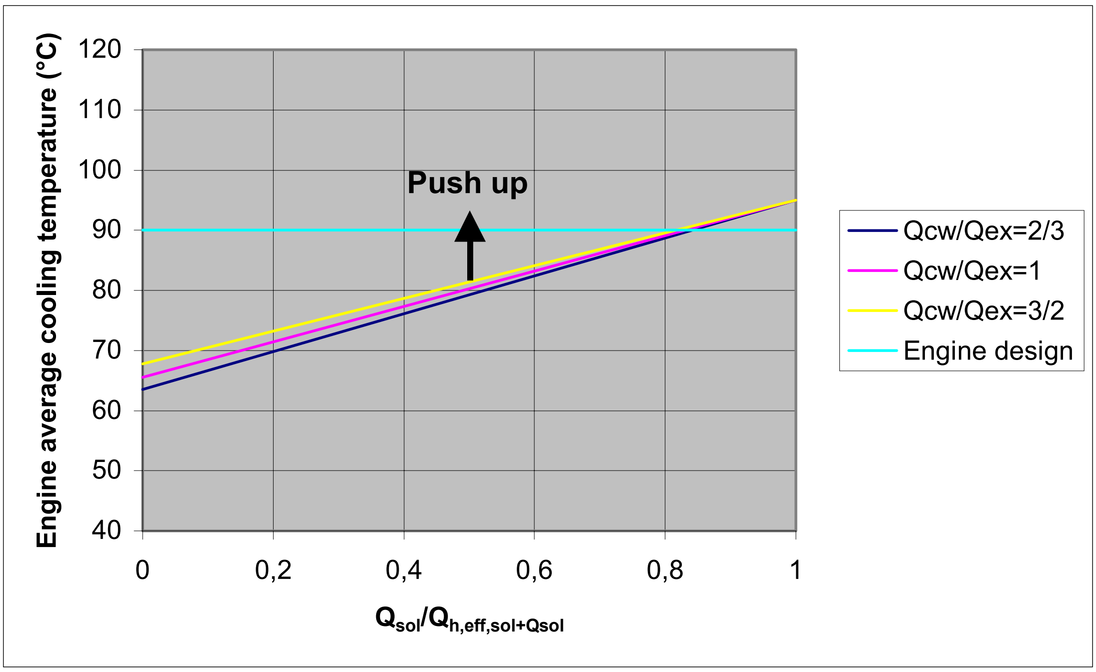


Figure 7.12-b Relationship between Engine average cooling water and solar collector absorber temperature at 20 °C ambient temperature.

7.3 Solar collector parallel connected to the Diesel engines

Figure 7.13 shows the parallel connection between solar collector and diesel engine connected with absorption cooling unit. In this case, the collector efficiency and collector area are discussed in section 7.2.1 as shown in **figures7.5** and **7.6** . The absorber temperature can be from figure 7.13 defined as:

$$T_{abs} = \frac{T_s + T_b}{2} \tag{7.16}$$

From figure 7.13, the solar heat energy can be written as:

$$Q_{sol} = m_{sol} * C_w * (T_s - T_b) \tag{7.17}$$

The engine effective heat in this case can be defined as:

$$Q_{h,eff,sol} = m_w * C_w * (T_s - T_b) \tag{7.18}$$

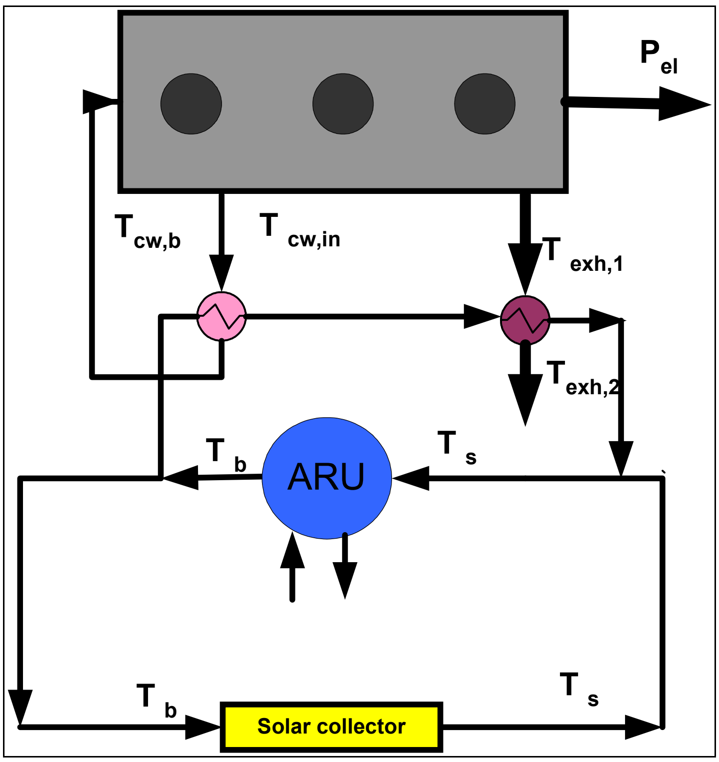


Figure 7.13 Parallel connection between solar collector and Diesel engine connected with absorption cooling unit.

Cooling load results are highly dependant on the types of the cooling units, solar collector area and diesel engine load . Therefore, a cooling load is a good verification of the solar collector area but not necessarily a guarantee. It becomes guaranteed when a good comparison with engine load and ambient temperature of the areas is established. In this section ,fuel consumption and electric load results for those cases studied in the previous section, will be represented but with solar collector. All this results will be taken for the cases studied in the previous section without solar collector.

7.4 Solar collector and Diesel engines part-series

In this section, the system illustrates that the solar collector maintained in parallel connection with engine water cooling system. The mass flow rate in this case is not equal as shown in **figure 7.14-a**. The total mass flow rate delivered in the engine exhaust gas system. **Figure 7.14-b** shows the T-Q diagram of this connection and the relationship between the different temperatures as shown.

The absorber temperature in this connection can be written as:

$$T_{abs} = \frac{T_2 + T_b}{2} \tag{7.19}$$

The solar heat in this connection and from last figure can be written as:

$$Q_{sol} = m_{sol} * C_w * (T_2 - T_b) \tag{7.20}$$

The engine effective heat in this case can be written as:

$$Q_{h,eff,sol} = m_{cw} * C_w * (T_2 - T_b) + m * C_w * (T_s - T_2) \tag{7.21}$$

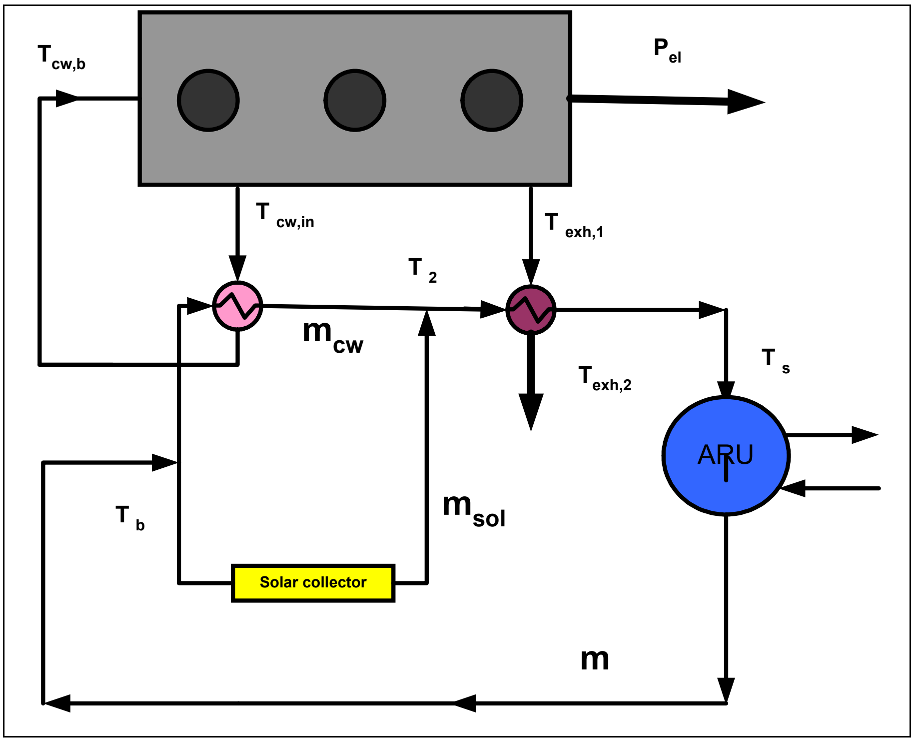


Figure 7.14-a Solar collector and Diesel engines connected as part-series

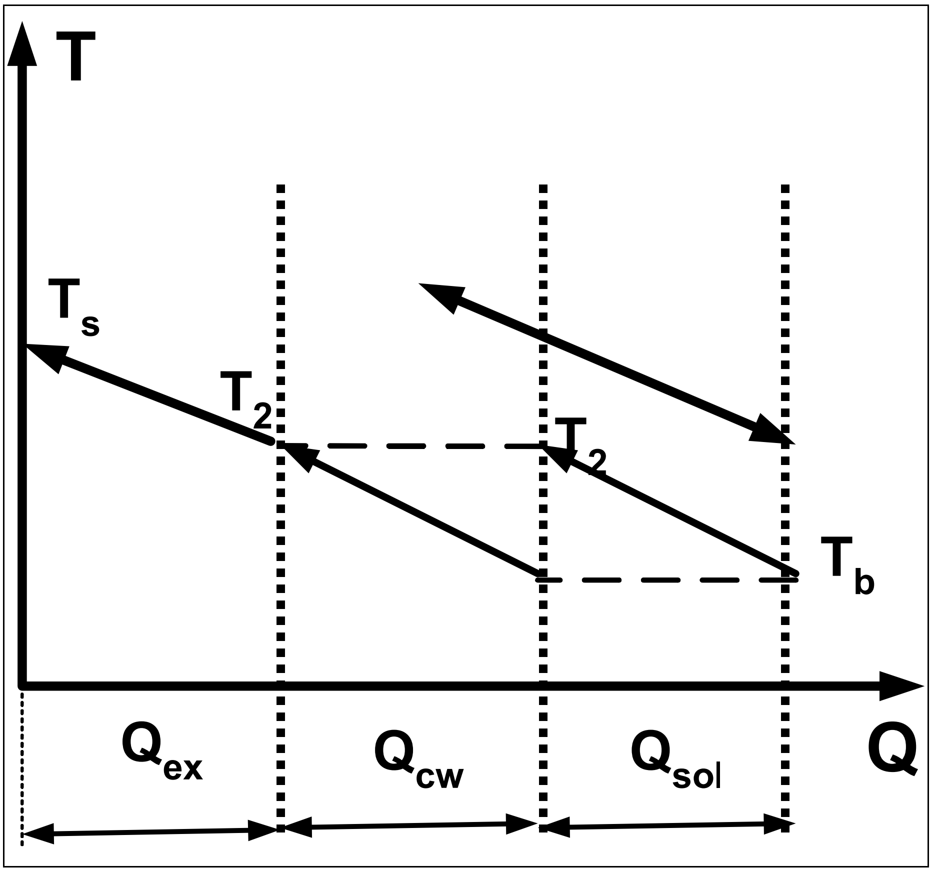


Figure 7.14-b T-Q diagram of part-Series-connection between solar collector and Diesel engine connected with absorption cooling unit

The relationship between the different mass flow rates can be written as:

$$m = m_{cw} + m_{sol} \tag{7.22}$$

It can be illustrated that variation of engine heat ratio on the mass flow rate ratio. **Figure 7.15** shows the effect of engine heat ratio on the solar mass flow rate as a function of the total mass flow rate. From this figure it can be obtained that the increasing of the cooling water heat ratio from 2/3 to 3/2 decreases the solar mass flow rate, for example, at solar ratio 0.5, decreasing from 0.714 to 0.625. the start and end points are constant as shown in the figure.

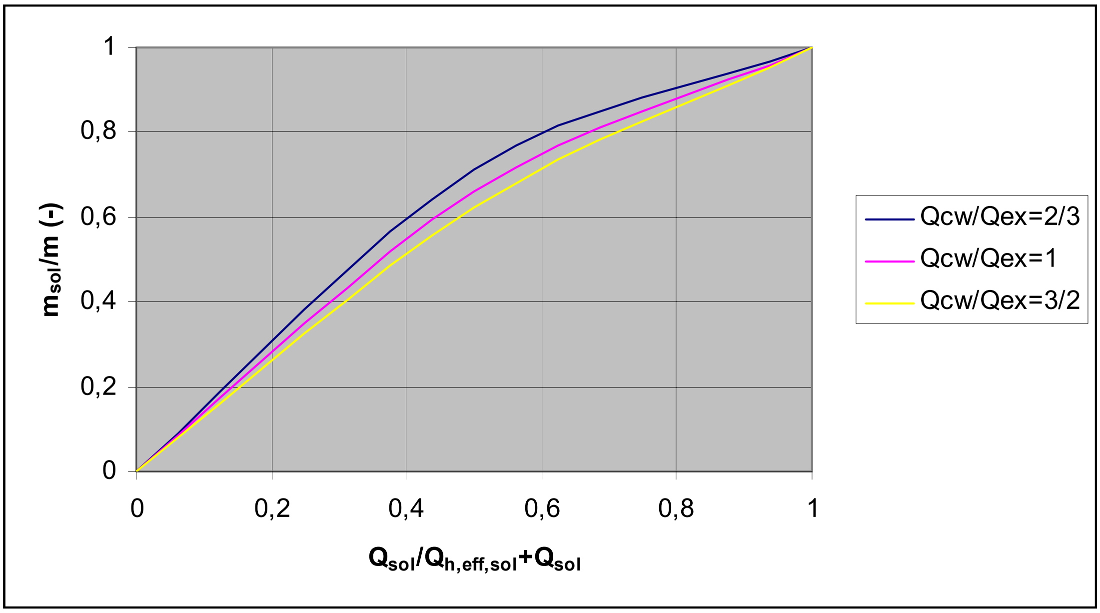


Figure 7.15 solar collector mass flow rate of part-Series-connection as a function of engine heat ratio

Figure 7.16 shows the collector efficiency of part-series connection at different solar radiations. This figure shows the solar collector efficiency at cooling to exhaust ratio of 1:1. From this figure it can be shown that the collector efficiency is more efficient than the series – A connection discussed in section 7.2.

Figure 7.17 illustrates the collector area ratio to cooling load related to this collector efficiency discussed in figure 7.16 at different solar radiation and ambient temperatures to obtain the collector area required to delivered the solar energy. This figure can be used with the help of the figure of fuel saved in section 7.1 to obtain the required collector area for the certain fuel values.

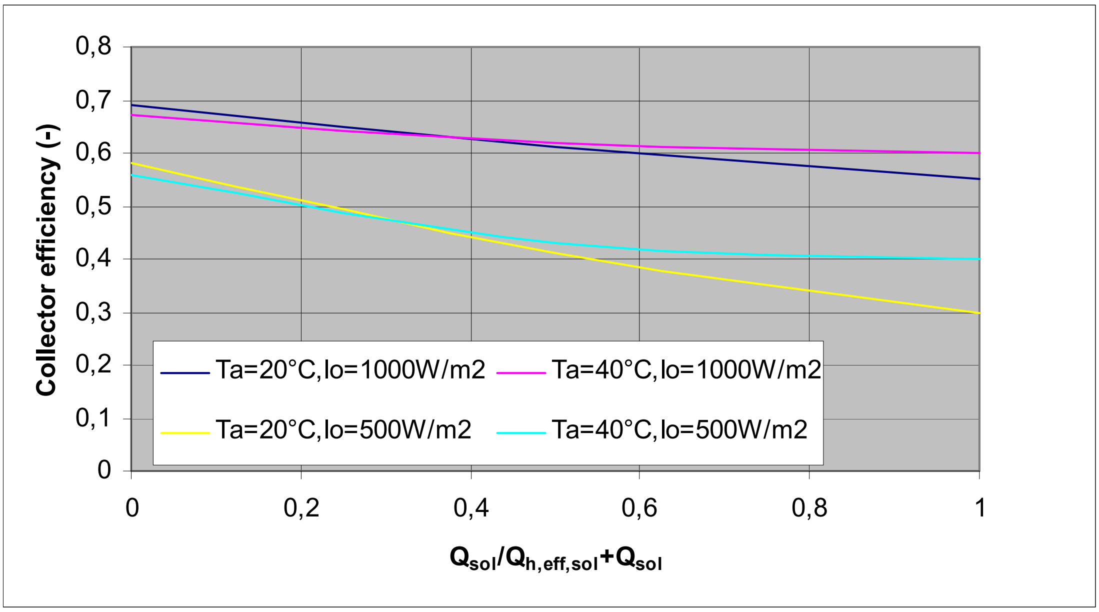


Figure 7.16 Solar energy ratio affected on solar collector efficiency at difference solar radiation and ambient temperatures.

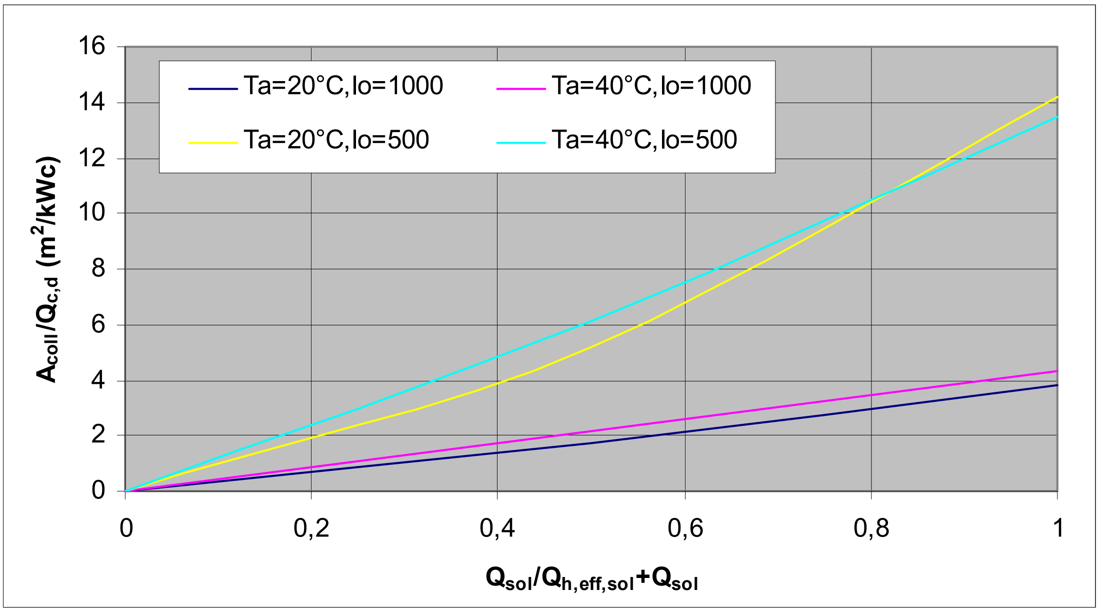


Figure 7.17 Solar collector efficiency affected on solar collector area to cooling load ratio at difference solar radiation and ambient temperatures.

Figure 7.18 illustrates the solar collector area at difference engine heat ratios. From this figure ,it can be shown that the effect of engine heat ratios is slight on collector area. It can be obtained from this figure the conclusion that the increasing the cooling heat to the exhaust heat increases the solar collector area. The affect on the engine average cooling water temperature has been discussed in the last section. The affect of this ratio can be maximum at the solar heat ratio equal to the engine effective heat which can be decreases by increasing and decreasing the solar heat ratio as shown in the figure.

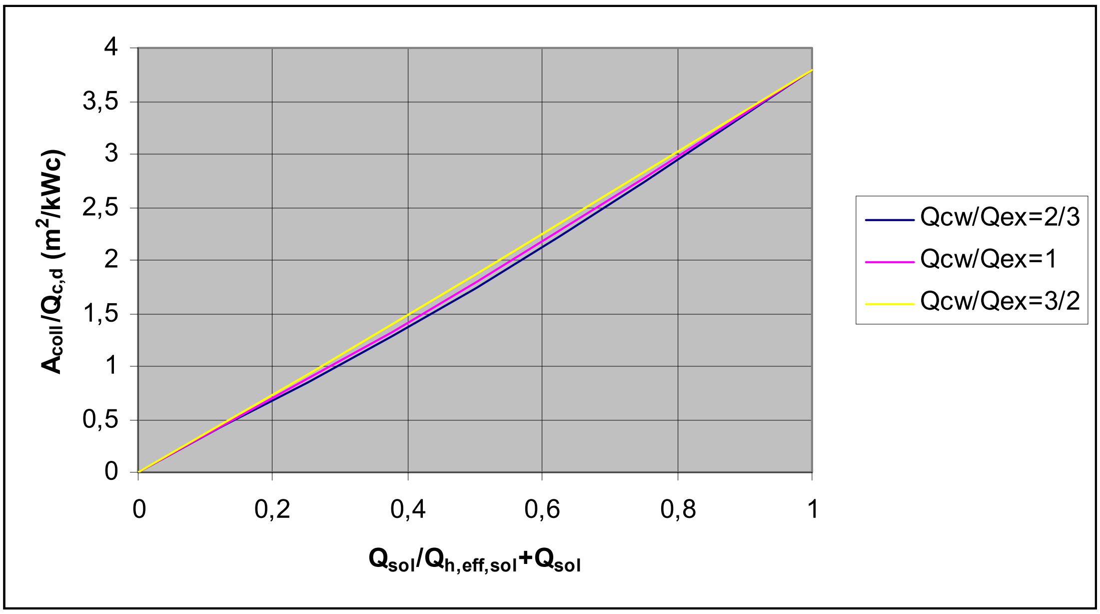


Figure 7.18 Engine heat ratio affected on solar collector area at solar radiation of 1000 W/m^2 and ambient temperature of 20°C.

7.5 Summing up

The different forms of connections between engine heats and solar energy can be summarized in the block diagrams shown in **figure 7.19**. The four case can be operated in the range of the absorption refrigeration machine (T_s and T_b) as shown in the block diagrams.

The different collector to cooling load ratio for this different forms discussed in the next figures to obtain the difference collector area required to produce the solar heat at different solar radiation of 500 and 1000 W/m^2 respectively. The main advantage of the part series connection that the reduce the average engine cooling water temperature than the series-B connection but the collector required area is more slightly than this connection.

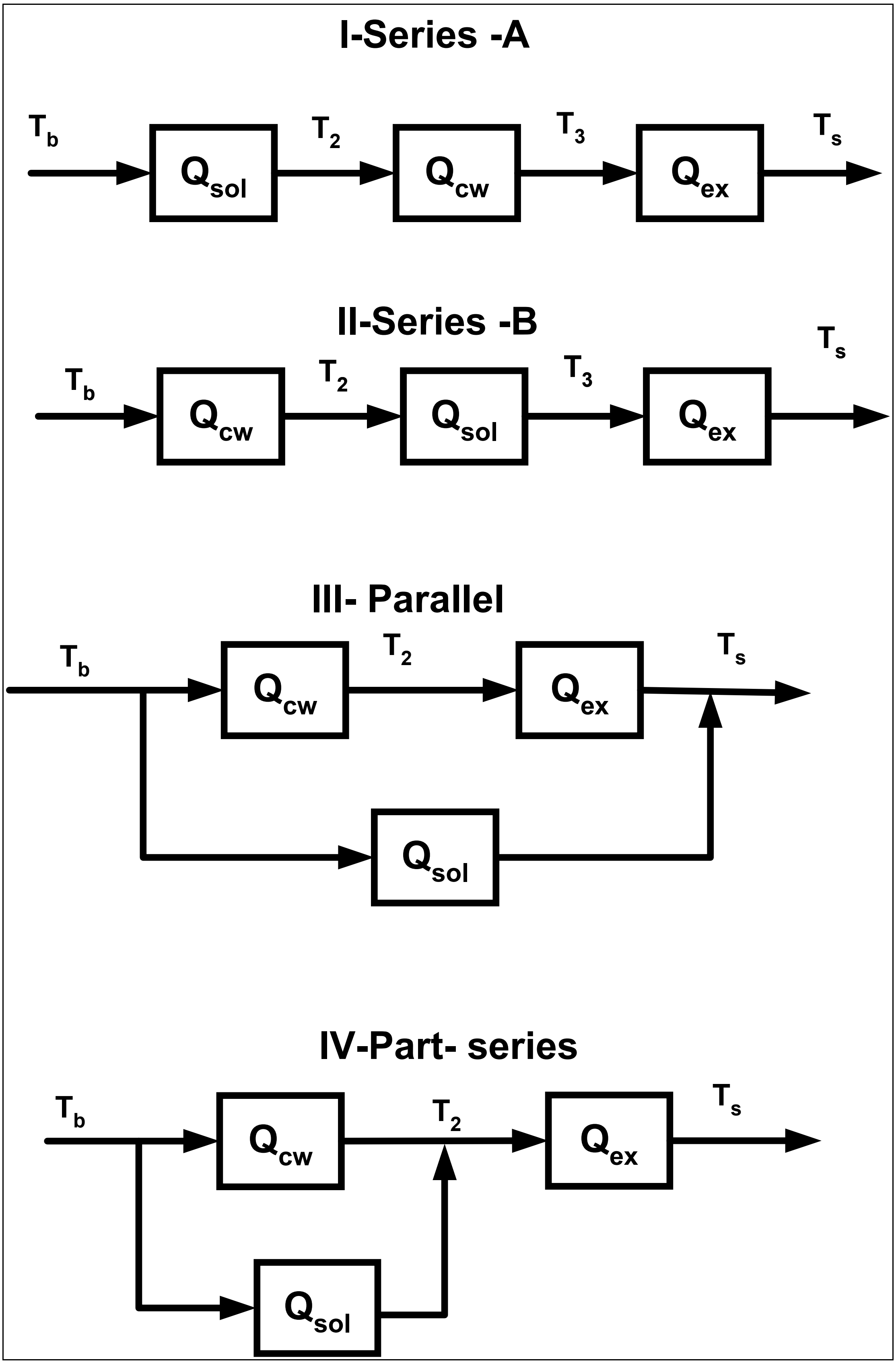


Figure7.19 Block diagrams of different engine and solar collector connections

The diesel engines without solar collector are discussed in the chapter 3, electric efficiency can be changed by changing the ambient temperatures referring to the designed engine performance as mentioned in the previous sections. Two fundamental basis should be taken into account, the engine fuel consumption to produce the cooling load and the solar collector area used. In order to build the correct comparison through the different electric plants and refrigeration systems , the same fuel consumption or electric load should be take into account.

Figure 7.20 shows the collector area to the cooling load for the four cases of connection discussed in figure 7.18 at constant radiation of 500 W/m^2. **Figure 7.21** discusses the same effect but at another solar radiation of 1000 W /m^2. In this case the difference is very small as shown in the figure. From these figures ,it can be illustrated the advantages of both series –B and part-series than the parallel and the series –A connection. To obtain the more efficient connection the engine cooling water should be taken into account which is better for the part-series than the series –B. it can be concluded that the best connection is the part-series for both engine and solar collector area.

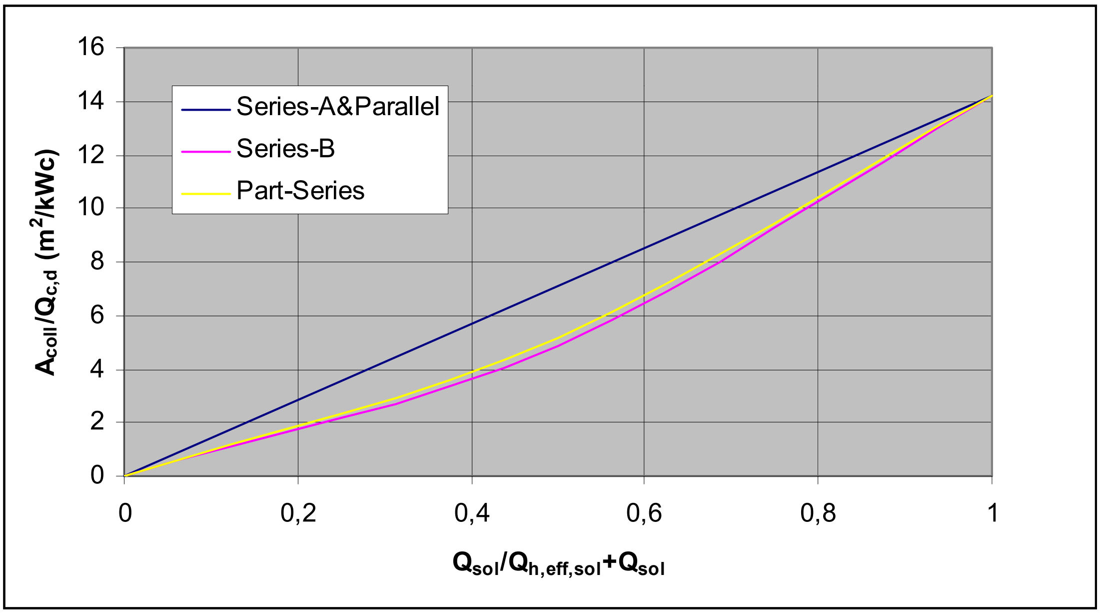


Figure 7.20 Solar collector areas for different connections at solar radiation of 500W/ m^2 and ambient temperature of 20 °C.

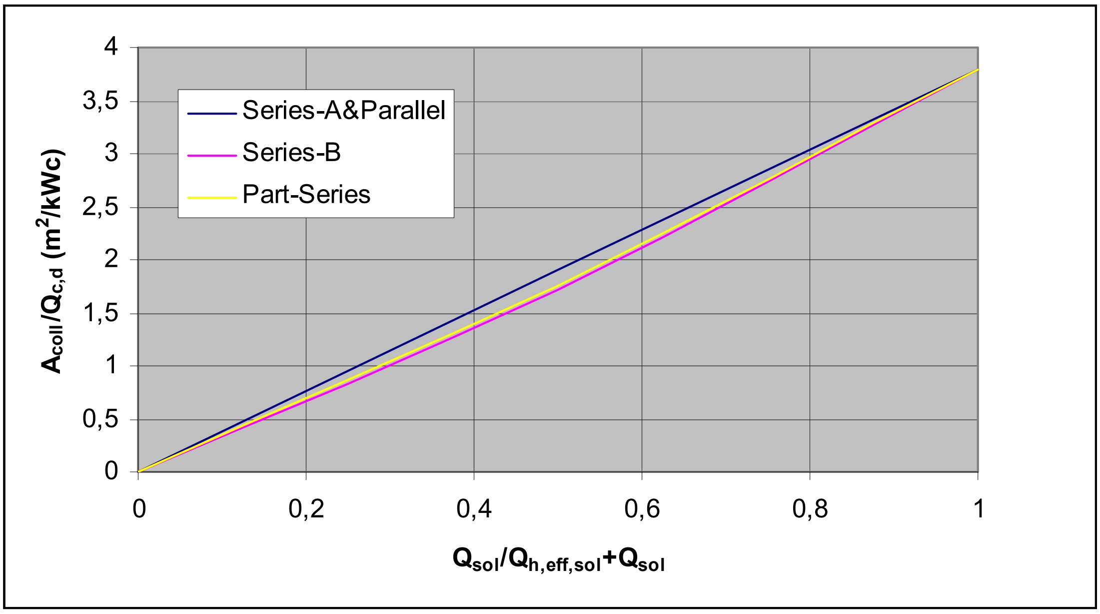


Figure 7.21 Solar collector areas for different connections at solar radiation of 1000 W/m^2 and ambient temperature of 20 °C

From **figure7.22**, the collector area required to obtain a certain saved fuel value for the part-series connection discussed in the last section which indicated the fuel saved in the diesel engine causing of using solar collector at the engine heat ratio of 1:1 was estimated.

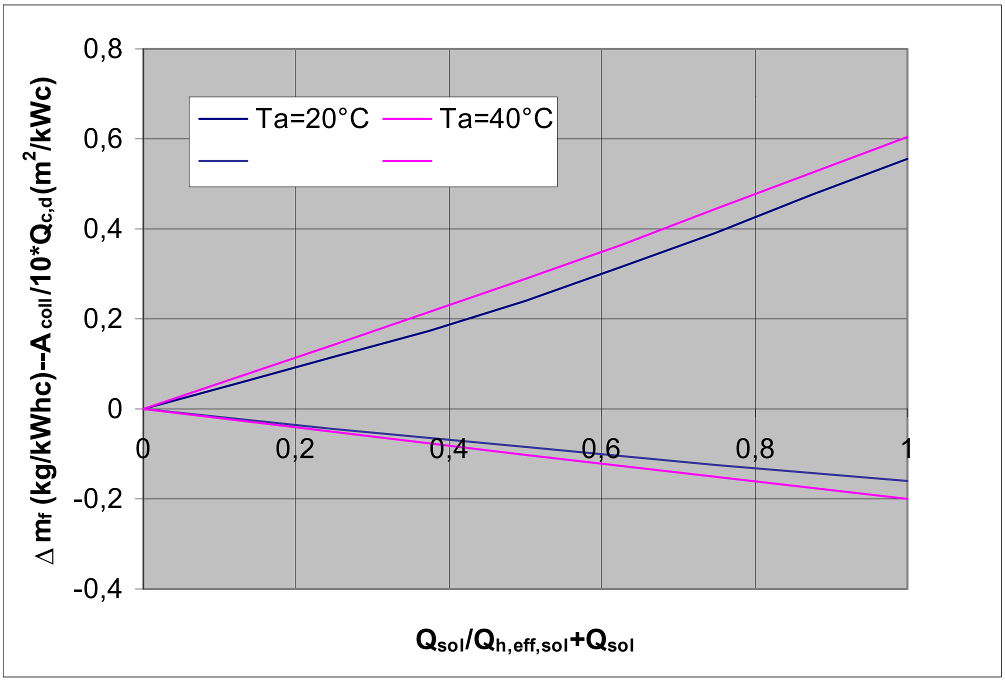


Figure 7.22 Fuel saved and collector areas by using part-series connection at solar radiation of 800 W/m^2.

Chapter 8

Conclusions and Future Work

8.1 Conclusions

The following conclusions demonstrates on the behavior of the different loads used to obtain the electric and cooling load from the total cooling and power systems used in the current study. It was surprising to find that only the total cooling and power systems of this problem took attention in the literature and no work concentrates on the solar energy side of the problem although it is actually of great interest.

The cooling loads demand have different forms, the cooling load demand at night and day:

- Night performance

The first case is the night performance, i.e., without both solar heat and solar collector. This case has been discussed in chapter 5 but the demand cooling load required in this case can be as a night cooling load which means the diesel engine is run alone without the solar collector.

The demand cooling load at night can be written as:

$$Q_{c,N} = Q_{tr} + Q_{in} + Q_{ven} \tag{8.1}$$

Figures 8.1 and 8.2 illustrates the relationship discussed in the last equation. It can be shown that the total heat changes from day to night in value of solar heat. The actual system curve can be obtained for different building systems which depend on the place of the building and the values of both ambient and solar heat as discussed in equation 8.3.

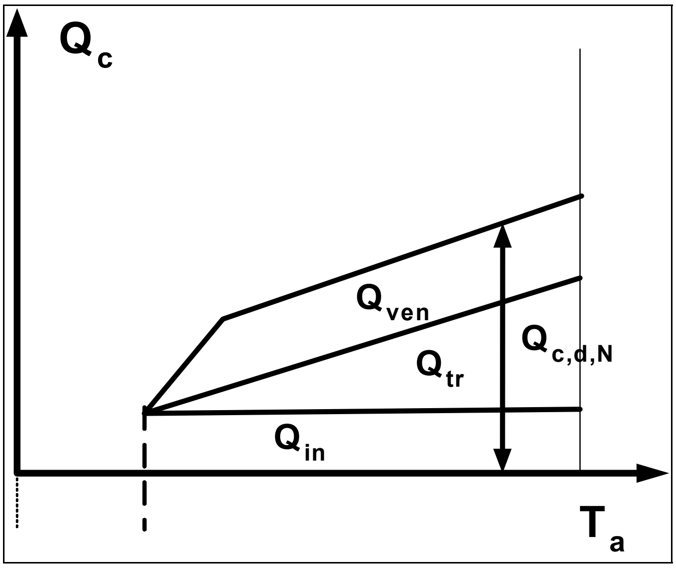


Figure 8.1 Diagram of building night heat balance

The air conditioning of the buildings should be taken into account referring to the different heat generated. The different forms of building heat can be consisted of transmission, internal, ventilation and solar energy. The value of the total heat can be obtained for different buildings depending on the area, ventilation and solar energy of this building.

The design of the conditioning systems of a certain building should be divided into a two main cases night and day operation of the system. The cooling –power ratio could be represented as follows:

$$S_c = \frac{P_{el,d,N}}{Q_{c,d,N}} \qquad (8.2)$$

for this ratio the fuel consumption should be minimized. As mentioned in chapter 5 ,the engine electric load can be in three forms:

1. Engine electric load = night electric load
2. Engine electric load > night electric load (Network feed-in)
3. Engine electric load < night electric load (Network take-out)

These three cases were illustrated in **figures 5.9** and **5.10.** The comparison between night and day performance can be shown in **figure 8.2.**

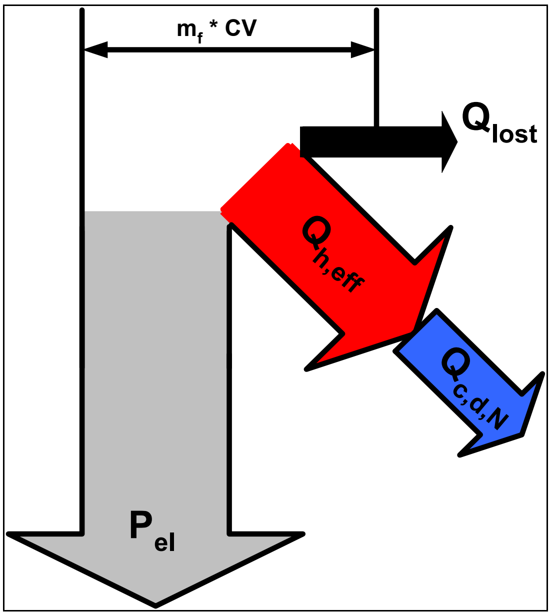


Figure 8.2 Night performance of diesel engine and absorption cooling machine

- Day performance

The solar radiation found at the day increases the cooling load demand at day more than the cooling load at night, in this case the cooling load increases with the value of solar energy and the cooling load demand can be written as after equation 8.1.

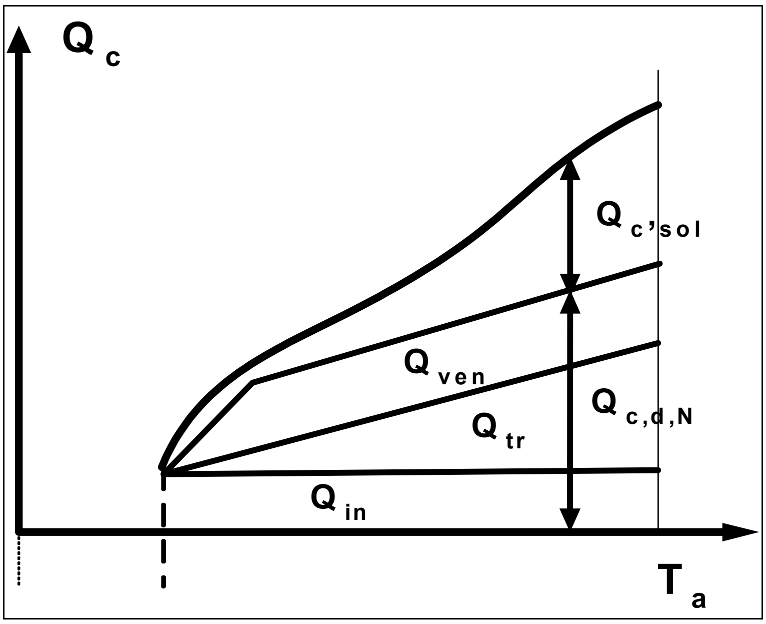


Figure 8.3 Diagram of building day heat balance

The total heat required can be changed between night and day and this difference should be taken into account. This relationship between the cooling load at day and night can be written as:

$$Q_{c,d,day} = Q_{c,N} + Q_{c,sol} \tag{8.3}$$

The day performance of the building can be divided into three cases depending on the solar energy values which can be equal, more than or less than the difference between the day performance and night performance as shown in **figure 8.3**. The three cases can be discussed as follows:

1. solar heat covers the difference between day and night cooling load

Figure 8.4 shows the affect of increasing day cooling load than night cooling load. In this case the solar energy cover the increasing in the cooling load demand at day than at night. The solar energy required in this case should be equal to:

$$Q_{sol} = \frac{(Q_{c,d} - Q_{c,d,N})}{COP_{aru}} \tag{8.4}$$

The required collector area is depend on the engine –solar collector connection and the solar energy radiation which discussed in chapter7 as a function of $Q_{sol} / (Q_{h,eff,sol} + Q_{sol})$.

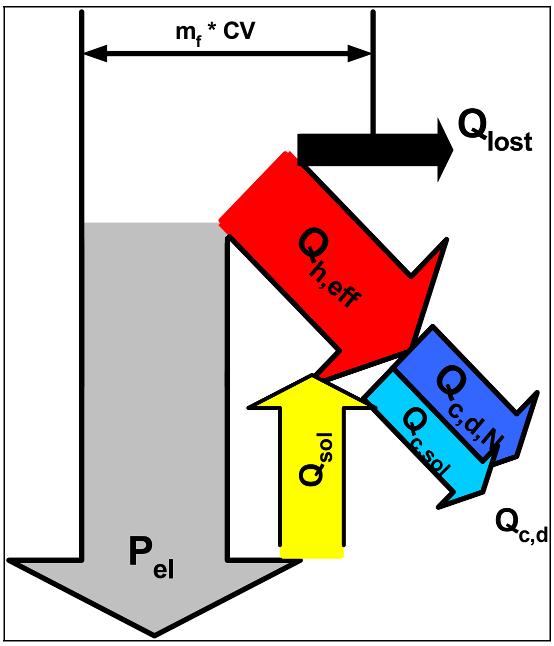


Figure 8.4 Day performance of diesel engine and absorption cooling machine

2. solar heat is less than difference between day and night cooling load

Figure 8.5 shows the affect of decreasing the solar heat than the required heat in figure 8.3 . In this case the difference of solar energy can be obtained by increasing the engine electric load over the electric load discussed in the last two cases which increases the engine effective heat .The solar energy in this case can be written as:

$$Q_{sol} < \frac{(Q_{c,day} - Q_{c,N})}{COP_{aru}} \tag{8.5}$$

The night and day cooling load illustrated in the last figures summarizes the performance of both engine and solar collector for a certain building which has the solar energy covers the increasing of the day cooling load ,less or more than the day cooling load. The first case can be carried out only for a few number of buildings ,the second case can be done when the solar radiation is more efficient than the designed one and the third case can be executed for some buildings has a certain solar heat generated from restricted solar collector area. The less solar energy rather than the difference of cooling load discussed in last equation, the more the both electric and engine fuel consumption as illustrated in chapter 7.

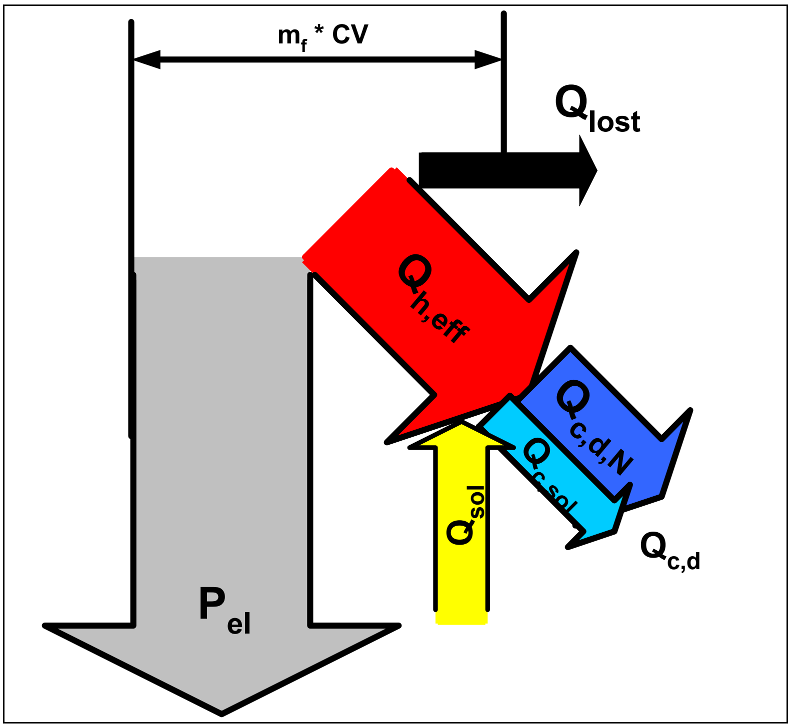


Figure 8.5 Day performance of diesel engine and absorption cooling machine (less solar energy generated)

The third case can be obtained between night and day, which has a zero solar energy and the demand cooling load is more than the night load. In this case, it is important to use a storage tank, that could be of different capacities depending on the difference between day and night load and the building type. The difference between night and day cooling load is a function of both the building capacity and the time as shown in **figure 8.6.**

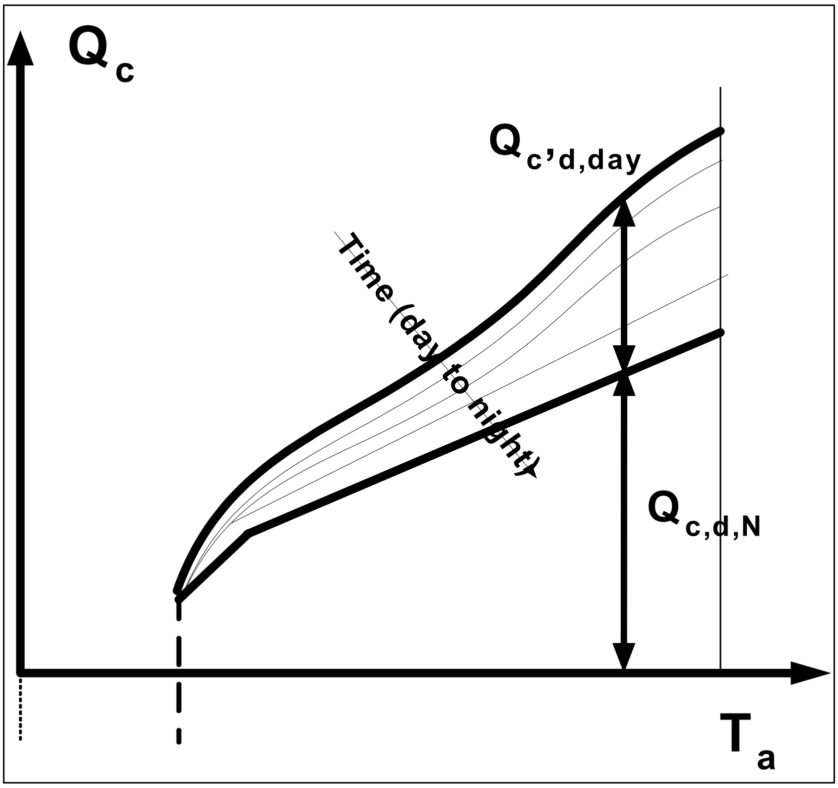


Figure 8.6 Diagram of cooling load depend on time variation

The charging of storage tank as discussed in chapter 6 can be in two forms

- **a.** Direct as shown in **figure 8.7-a**
- **b.** Indirect as shown in **figure 8.7-b**

The discharging of the storage tank used in the time between night and day when the solar energy is equal to zero but the air is still warm. The demand of the storage heat can be varied with the variation of time and the building capacity. **Figure 8.8** shows the discharging concept of the selected connection discussed in chapter 7 to compensate the difference between the actual cooling load and the night cooling load without any increasing in the fuel consumption.

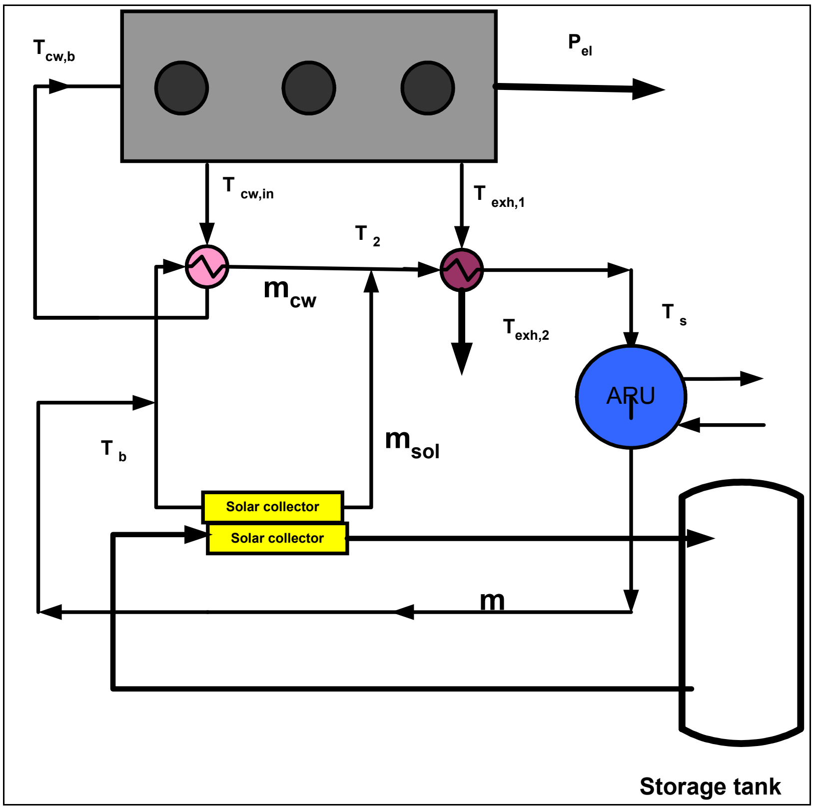


Figure8.7-a Direct storage tank charging from part-series engine connection.

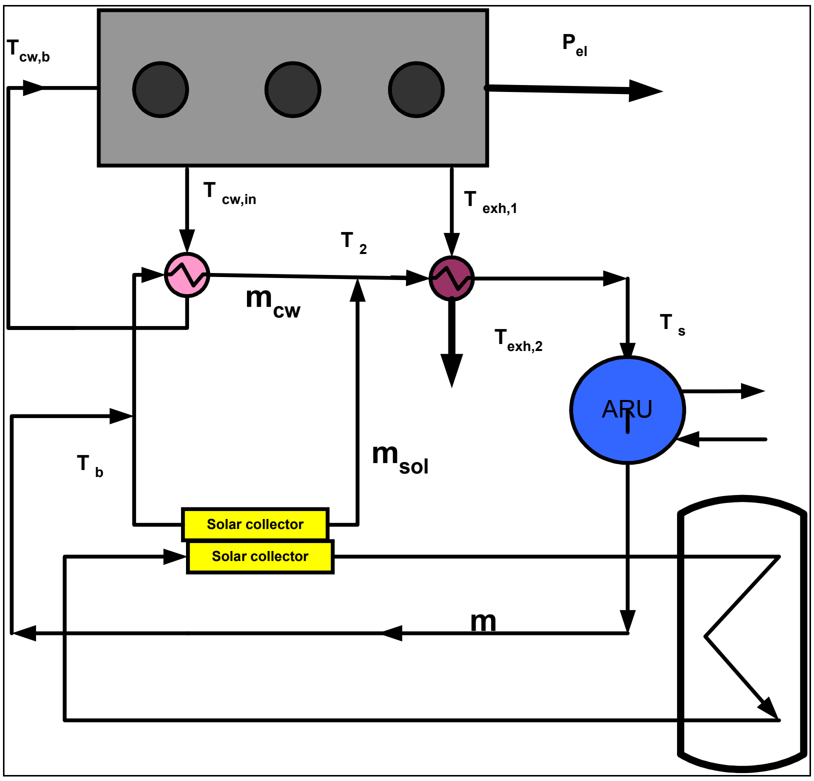


Figure 8.7-b Indirect storage tank charging from part-series engine connection.

The storage heat and the demand cooling load required for a certain building controlled the collector area and should be taken into account that the collector area.

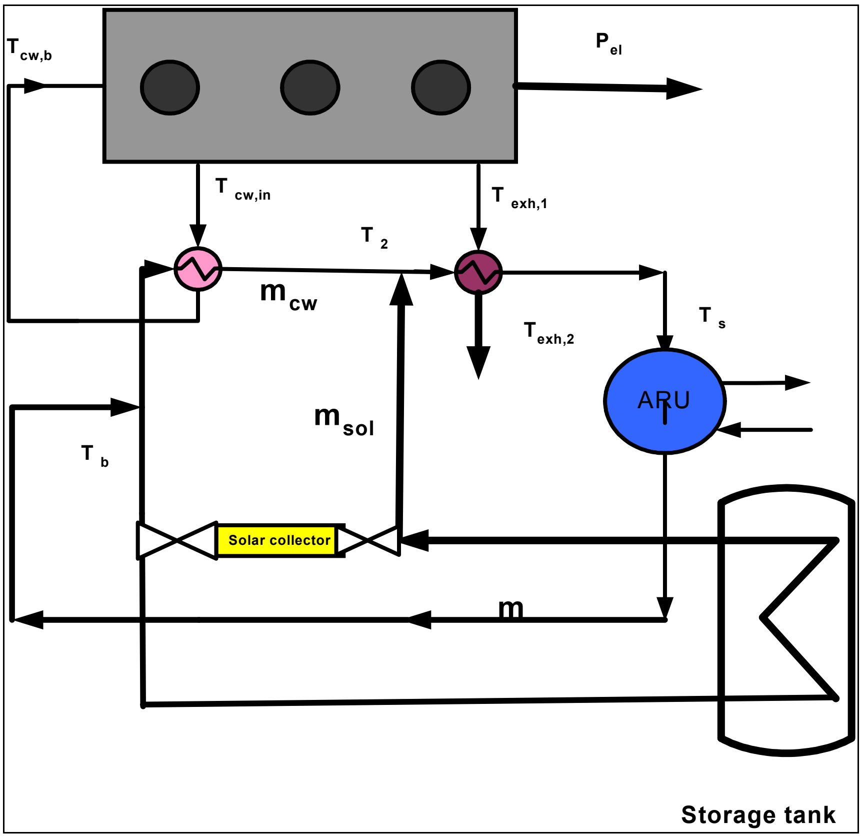


Figure 8.8 Discharging of indirect storage tank in part-series engine connection.

3. solar heat is more than difference between day and night cooling load

Figure 8.9 shows the effect of increasing the solar heat than the required heat in figure 8.4 . In this case the difference of solar energy can be saved in the storage tank to be used later on. The engine electric load in this case should be equal to the electric load in the previous case as shown in the last two figures. The solar energy balance can be in this case written as:

$$Q_{sol} > \frac{(Q_{c,day} - Q_{c,N})}{COP_{aru}} \tag{8.6}$$

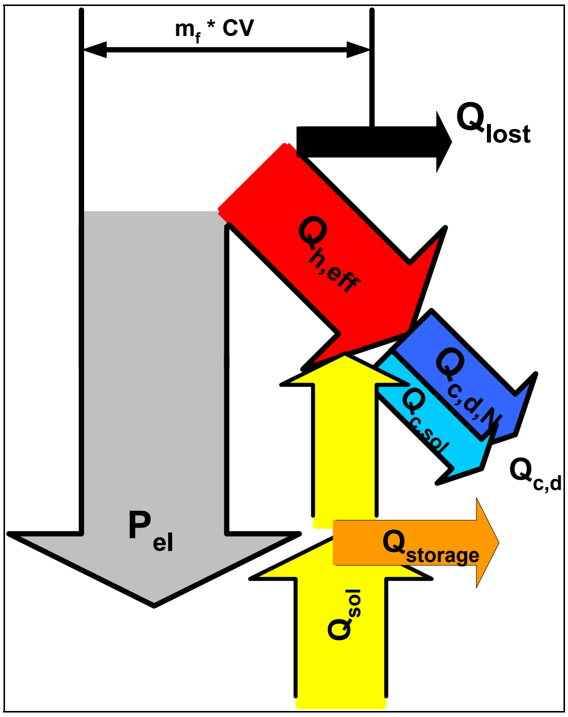


Figure 8.9 Day performance of diesel engine and absorption cooling machine (more solar energy generated).

8.2 Recommendations

The following recommendations of the current work can be summarized as follows:

- The principles of appropriate selection of marine diesel engine have been established through the last decades. Two fundamental basis should be taken into account; improving engine thermal efficiency by improving brake thermal efficiency or use of waste heat recovery systems. When solving similar problems, one should keep in mind the range of the main engine power. The heat recovery system can be economical for power generation more than 15 MW of main engine.

- According to the electric power, the used diesel engines (which are more efficient than gas engines) show an excellent ability in solving similar problems as the problem of the current study.

- The evacuated solar collector behavior is more efficient than the simple plate one that agrees well with the results found in the literature. From economical point of view, the simple plate collector was used in the current study.

- Water/ Lithium- Bromide absorption refrigeration units are the most used operation with heat and power generation. Two-stage Absorption cooling units are mostly used reason for low temperature range of engine waste heat recovery systems.
- Absorption cooling units are not only viewed as energy efficient cooling and heating methods but also as environmentally alternative to CFC, HFC or HCFC based technology.
- A mathematical model for diesel engine with absorption refrigeration units was derived. Two cases of connection between engine and solar collector were presented in the current study.
- Scanned solar and engine parameters were chosen as solar radiation, solar collector area, Sankey diagram and cooling units types where other engine performance characteristics vary accordingly. Results obtained, emphasize that the two-stage absorption refrigeration units are more efficient than the single stage one.
- The fuel consumption rate comparison studied in the current work, which takes into account that both electric and cooling load must be the same for the selected cases mentioned before.
- The addition of the solar collector improves both fuel utilization and power ratio. By increasing the solar radiation good result was obtained.
- The control of absorption refrigeration supply temperature in the range from 75 to 90 °C was studied in the current study.
- The use of the solar energy not only decrease the fuel oil used in the diesel engines but also improves the environmental affect by decreasing the CO_2 emission.
- The engine connection with the solar collector is decreased or increased both the collector area and the primary energy utilized
- The demand cooling load between night and day should be estimated to obtain the actual storage tank volume and solar collector area.

8.3 Future work

At the end of this work, it is useful to draw some lines for any extension research to the current study:

1. It would be very practical to extend the range of the collector types to cover different collectors or solar energy forms. It might be very interesting to study the validity of the result of the maximum average of utilized fuel for both gas turbines and fuel cell.

2. For the whole range of solar collector area under investigation, the best way to obtain the most efficient combined heat and power systems economical analysis should be carried out for the different types of solar energy. This result was found to be valid for both renewable energies used. It may be a great necessity to compare results with different power ranges in the future. Another renewable energy sources should be taken in consideration when combined heat and power systems to be studied.

3. Using fuel cell in heat and power combined systems for buildings and industry energy managements with solar energy as well as for the marine applications as a main and auxiliary engines.

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