A Dissertation on

PERFORMANCE STUDY OF CARBON-DIOXIDE AS AN ALTERNATIVE WORKING FLUID IN EVACUATED TUBE COLLECTOR UNDER DIFFERENT THERMODYNAMIC CYCLES

Submitted in the partial fulfillment of the requirement for the award of the degree

MASTER OF TECHNOLOGY

IN

Renewable Energy Technology

Submitted By

Aadish Jain

ROLL. NO. 2K13/RET/12

UNDER THE GUIDANCE OF:

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(Department Of Mechanical Engineering)

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DELHI TECHNOLOGICAL UNIVERSITY, BAWANA ROAD, DELHI - 42

MAY 2015

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DECLARATION

This is to certify that the work presented in this dissertation entitled "**Performance study of Carbon-dioxide as an alternative working fluid in Evacuated tube collector under Different Thermodynamic Cycles**" towards the partial fulfillment of the requirements for the award of the degree of **Master of Technology** with specialization in **Renewable Energy Technology**, from Delhi Technological University, Delhi is an authentic record of my own work carried out under the supervision of **Mr. P. K. Jain**, Associate professor, Department of Mechanical Engineering at Delhi Technological University, Delhi.

To the best of my knowledge, the content of this dissertation report has not been submitted by me for the award of any previous degree to anyone else.

Aadish Jain
2K13/RET/12
Place: Delhi

This is to certify that the above statement made by the candidate is true to the best of my knowledge.

Mr. Pradeep Kumar Jain
Associate Professor
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Delhi Technological University

CERTIFICATE OF APPROVAL

It is certified that the contents and form of the project entitled **"Performance study of Carbon-dioxide as an alternative working fluid in Evacuated tube collector under Different Thermodynamic Cycles"** submitted by Aadish Jain (2K13/RET/12) is hereby approved as a creditable study of research topic and has been submitted in a satisfactory manner for its acceptance as prerequisite to the degree for which it has been submitted.

It is understood that by this approval, the under signed do not necessarily endorse any conclusion drawn or opinion expressed there, but approve the Submission for the purpose for which it is submitted.

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Abstract

Evacuated tube solar collectors as an energy delivering device for low grade heat employment have been investigated for years. In current ages, some researchers have given additional concentration to engaging non-conventional fluids such as Nano-fluids, various refrigerants as an absorbing medium to employ solar energy for its enormous and sustainable reserve. Typically Evacuated tube solar collectors have been comprehensively exercised in low temperature energy technology. A number of projects of solar collectors have been established over the years with the intention of improving and investigating their performances. Thermal Energy performance of a solar collector is studied with naturally available Carbon-dioxide gas as an alternative working fluid. The main aim of this study is to develop a thermodynamic cycle which could increase the usability of the proposed fluid within the parameters already achieved for the collector. A Possible thermodynamic cycle is deduced and proposed with primary heat addition at constant volume, turbine work output, multistage compression and intercooling as heat recovery. This system is expected to handle much more amount of energy, uniform heat recovery over a wide range of solar irradiation, requires much less compression work and better net-work output as compared to Conventional cycles with primary heat addition at constant pressure employing Evacuated tube Collector with Combined Heat and Power Cycle.

Important Notation

	Notation	Definition
1.	CO_2	Carbon Dioxide
2.	CNG	Compressed Natural Gas
3.	CST	Concentrated Solar Thermal
4.	MNRE	Ministry of New and Renewable Energy
5.	λ	Wavelength
6.	°	Degree
7.	k	kilo (10^3)
8.	M	Mega (10^6)
9.	µ	micro (10^{-6})
10.	W	Watt
11.	m	Meter
12.	K	Kelvin
13.	C	Celsius
14.	π	*pie*
15.	J	Joule
16.	Pa	Pascal
17.	AC	Alternating Current
18.	DC	Direct Current

Chapter: 1 INTRODUCTION

1.1 Introduction to Renewable Energy Technology:

Renewable Energy Technology is a wide area of Research and Development that is being investigated. It has true interdisciplinary character which find its roots in various basic subjects such as Physics, Chemistry, Biology, Thermodynamics, Heat transfer, Power Electronics, Geography, Economics etc., research in this area essentially requires expertise of more than one form of Science and/or Engineering.

A sensible renewable technology must be simple to understand, reliable in functioning and economical in production. Along with these properties if it helps in addressing some major problems of the urban areas of developing world then the technology becomes lucrative and can be more easily accepted. Now Electricity demand is a serious problem in the urban areas of a fast developing nation like India. With the rapid increase in urban population the problem is becoming more severe with every passing day. However the solution to the problem on the same scale is still at large.

1.2 Important Aspects: Renewable Energy Technology

Renewable energy is recognized as vital input for sustainability and hence encouraging its use, and promoting it commercially is significant [4]. Sources of energy will include: Wind, Water, Biomass, Sunshine and other such natural sources which either has the tendency to replenish its energy by naturally occurring process without any human effort or it lifespan is for several million years and can be considered as inexhaustible source, and several biological wastes.

Application of such sources to utilization processes ranges from very small to large units, and four questions are asked for practical application:

1. Amount of energy which is available in the immediate environment and what is the resource?
2. What are the possible applications of the available energy – what is the end-use?
3. What could be the possible effects on environment due to the employment of technology under consideration – is it sustainable?
4. Economics of the available energy – is it cost-effective?

1.3 List of Various Renewable Energy Technologies:

1. Solar Energy
    a. Solar Photovoltaic
    b. Solar Thermal Energy
2. Wind Energy
    a. On Shore
    b. Off Shore
3. Bio-mass
4. Biofuel
    a. Bio-CNG
    b. Bio-Diesel
    c. Ethanol
5. Small Hydro
6. Geothermal
7. Fuel Cell

1.4 Realizing the Potential of Non-Conventional Energy Sources:

The importance of renewable energy sources was realized in March, 1981, when the government of India made a Commission for Additional Sources of Energy (CASE) in the Department of Science and Technology. In 1982, CASE was taken under the Department of Non-Conventional Energy Sources (DNES), Ministry of Energy. IREDA (Indian Renewable Energy Development Agency Ltd.) got established in 1987. The DNES was converted to MNES (Ministry of Non- Conventional Energy Sources) in 1992. Finally in 2006 the ministry was named as the Ministry of New and Renewable Energy (MNRE) [4].

The cost of the energy is going up with the depletion in the current energy resources. The industrial sector is a major contributor towards this trend. So, an overall need arises to reduce down the energy costs which can be considerably achieved by enhancing the working efficiency of the various processes, equipment and operations involved in the industry. The energy efficiency can be further improved by switching over to the renewable sources of energy, creating awareness and a dedicated monitoring of the systems. The Government of India has passed various legislations and amendments to shift the focus towards energy efficiency. Bureau of Energy Efficiency (BEE) was made for defining certain energy standards, providing guidelines, accrediting various organizations for energy audits, carrying out the projects related to energy efficiency and carrying out the research activities related to the energy efficiency equipment [6].

The demand for power is increasing day by day. It's a high time we should stop our dependence on the conventional sources of energy and adopt renewable sources of energy to meet our energy demands. The emission of Carbon dioxide (CO_2) is almost unstoppable as a result of dependence on conventional sources of energy. Hence using fossil fuels is directly related to the rising environmental problems [5].It can be estimated that with the increasing concentration of CO_2 the earth can get warmer by (1-5°C) with the advent of the next century [2].The oil crisis of 1973 and the rising environmental concerns attributed to the use of fossil fuels have resulted in certain outstanding efforts in adopting alternative energy sources [4]. India has a tremendous potential for solar power. A $6 billion to $7 billion capital equipment market for grid-connected solar generators is estimated [5]. In a typical Indian scenario, where there is a tremendous increase in the population in the cities, a need arises to develop clean energy technologies and innovations have to be adopted to ensure the healthier habitat for this population [3].

These non-conventional, alternative and renewable sources of are continuously gaining importance and publicity, technologically as well as politically. Renewable energy was utilized to produce electricity as early as 75 or 100 years ago. Sources such as Solar, Wind, Hydro and Geo-thermal have all been utilized with varying levels of success. The most widely utilized sources are hydro and wind power, with solar power being moderately used worldwide and in current scenario with the advancement of the technology solar photovoltaic is taking big leaps in terms of adaptation .[2]

1.5 Solar Energy on Earth

As for Sun as source of renewable energy, Solar radiation reaches the Earth's surface at a maximum flux density of about 1 kW/m^2 in a wavelength band between 0.3 and 2.5 µm. This is called *short wave radiation* and includes the visible spectrum.

Table:1 Solar spectrum[11] Division:

S.No	Spectra	Wave Length band	Percent of Irradiance
1	Ultraviolet region	$0.3 < \lambda < 0.4$µm	~5% of the irradiance
2	Visible region	0.4µm $< \lambda <$0.7µm	~43% of the irradiance
3	Infrared region	$0.7 < \lambda <$ µm	~52% of the irradiance

For inhabited areas, this flux varies from about 3 to 30MJ/m^2 per day, depending on place, time and weather. The temperatures of the Earth's atmosphere, at about 230 K, and the Earth's surfaces, at about 260–300 K, remain in equilibrium at much less than the 6000K temperature of the Sun. Therefore the outward radiant energy fluxes emitted by the Earth's atmosphere and surfaces are also of the order of 1kW/m^2, but occur in an infrared wavelength band between about 5 and 25 µm, called *long wave radiation*, peaking at about 10µm.

- Solar constant = 1366 W/m^2
- Radius of Earth = $(2/\pi) \times 10^7$ m
- Total power of solar radiation reaching Earth :
  - $1366 \times (4/\pi) \times 10^{14}$ W
    $= 1.73 \times 10^{17}$ W
- Each day - 86,400 s
- Each year - 365.24 days.
- The total solar radiation reaching Earth surface per annum is:
  - $1.73 \times 10^{17} \times 86400 \times 365.24$
    $= 5.46 \times 10^{24}$ J.

Source of data [4,11]

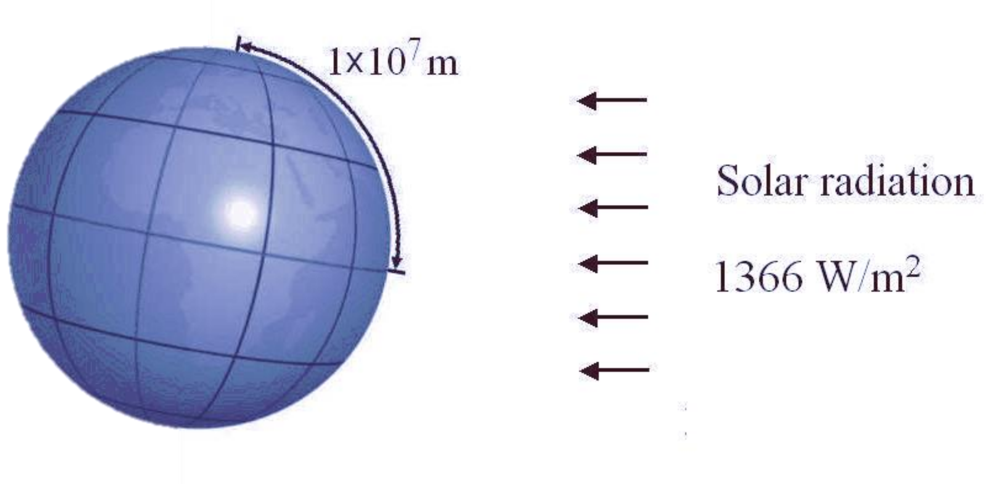


Figure 1. Solar Irradiation.
Source : [4]

Solar power has a low density per unit area (1kW/sq.m to 0.1 kW/sq.m.), therefore requires a large ground area by the collectors to harness it [4]. The collectors can be concentrating type or non-concentrating type. In the present discussion we will focus on the non-concentrating types of solar collectors.

1.6 Solar Photovoltaic

Photoelectric effect, mechanism and subsequent development along with continuous advancement of the solar cell technology, extracting useable electricity from the sun was made possible. Photoelectric effect is shown by semi-conductive material that converts visible range of energy i.e. light into a Direct Current (DC). By designing solar arrays according to the need, a series of solar cells electrically connected and DC voltage is generated which can be physically used on a load. Solar PV energy is being increasingly employed and popular in remote areas where placement of electricity lines is not economically viable. [2]

From the source of recorded history of mankind, it can be easily concluded that only two methods are employed till date to produce significant electric power. The first is due to the relative interaction of magnetic field and conductor which was discovered by Michael Faraday in 1821 and in commercial production by 1885. The second is by photovoltaic effect using *solar cells* (more technically called *photovoltaic cells*). The devices produce electricity directly from electromagnetic radiation, especially light, the technology does not employ any moving part during operation.

The photovoltaic effect was discovered by Becquerel in 1839 but not developed as a power source until 1954 by Chapin, Fuller and Pearson[1] using doped semiconductor silicon. Photovoltaic electricity in the past decade has been one of the fastest growing renewable energy source and technology: annual production of the solar pv cells grew tenfold, with this growth still continuing. With the active support of various government agencies worldwide to shift generation to carbon free technologies demand has seen an exponential upsurge. There are several other factors such as modular character, standalone and grid-linked opportunities, reliability, ease of use, lack of noise and emissions, and reducing cost per unit energy produced supporting popularization of demand.[1]

Solar panels convert incident energy from the rays of the sun into electrical current in the solar cells. The maximization of the rays of the sun incident upon the solar panels, in turn maximizes the output of electricity. A practical way of achieving this is by positioning the panels such that the rays of the sun fall perpendicularly on the solar panels by tracking the movement of the sun. This can be achieved by means of using a solar panel mount which tracks the movement of the sun throughout the day. Energy conversion is efficient when the rays' falls at an angle near to perpendicular onto the solar panels.

1.6.1 Solar Tracking Systems

In **solar tracking systems**, movement of the sun is tracked throughout the day and solar panels are mounted on a structure which moves accordingly.

The 3 methods of Sun Tracking:

- Active Tracking
- Passive Tracking
- Chronological Tracking.

These methods can also be classified as single axis or dual-axis solar trackers.

In **active tracking**, sensors are used to determine the position of the sun in the sky during the day. The sensors will are linked to the circuit controlling the motor or actuator to move the mounting system as required so that the solar panels always face the sun throughout the day.

Passive Tracking unlike active tracking which determines the position of the sun in the sky, a passive tracker moves in response to an imbalance in pressure between two points at both ends of the tracker.

A **chronological tracker** is a time-based system whereby the structure is moved at a fixed rate throughout the day. The logic is the movement of the sun across sky during a day time is at fixed rate. Thus the motor or actuator is programmed to continuously rotate at a “slow average rate (can be customized according to the need). This method of sun-tracking is very accurate.

Azimuth angle and **Hour angle** in the ***Horizon System*** for sun positioning is much more useful, as this gives the portion of the sun according to the observer. However Equatorial system of sun positioning helps in exchange of data with researchers at different locations which are practically so far that angle of horizon system may vary vastly.

Utilizing this knowledge of sun position and solar path following system can be designed which would be much lower in cost but competitive to Sun Trackers for net power output.

Sun position remains nearly same for a time moment of every year.eg. the position of the sun on 2: 20 pm , 21st April of every year will nearly be the same.

Similarly, the path followed by the sun for a day of every year will not deviate much. eg. The path followed by the sun on 21st April of every year will more or less be the same.

Although average length of day is 12 hrs but to practically utilize this positioning for Solar Energy purposes and for ample amount of Irradiation, it is suggested to design the system for sun tracking in between 8am to 5pm.

The solar hour angle h_S is based on the nominal time of 24hrs required for the sun to move 360° around the earth or 15° per hour or 4 mins per degree.

Longest day for 13.83 hrs on 21st June

Shortest Day for 10.17 hrs on 21st December

Maximum Azimuth Angle = 84.70 on 21st June

Minimum Azimuth Angle = 37.80 on 21st December

1.7 Various Solar Thermal Technologies and Applications

1.7.1 Flat Plate Collector (Non Concentrating Type Collector)

A flat-plate collector is a heat exchanger device which converts Solar radiation energy to Thermal energy by absorbing energized radiation from a distant object (Sun) and delivering absorbed heat to fluid (Air , Gas , Water etc.). Flat plate collector absorbs both direct and diffuse radiation, has very low maintenance and no sun tracking is required. Temperatures of around 80°C to 100°C can be achieved using this type of solar collector.[19]

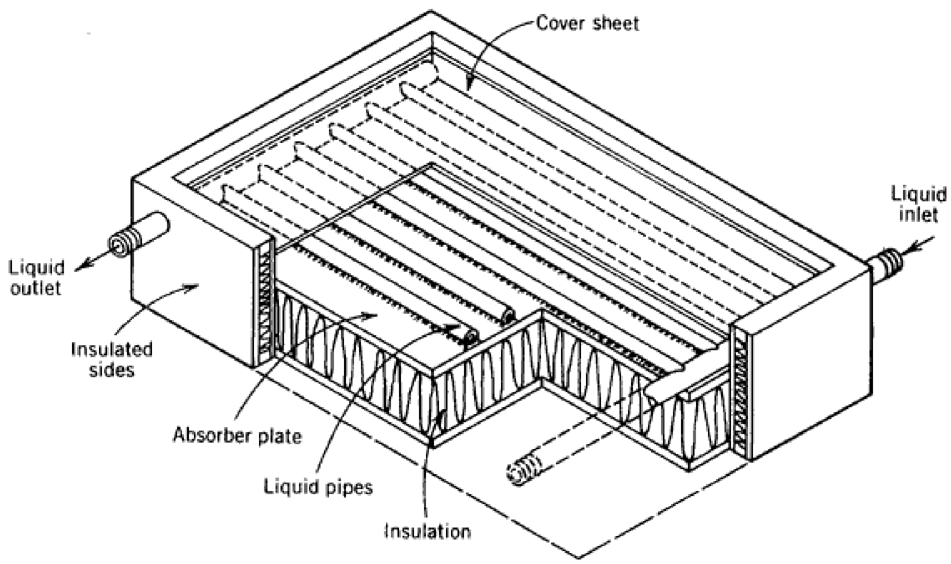


Fig : 2a Flat plate Collector.

Image source: [39]

Applications: Flat plate collector can be used for space heating applications by using air heaters, can be used for heating water for various purposes. This can also provide source for low temperature applications.

1.7.2 Modified Non Concentrating Type Collector

1.7.2.1 Modified flat-plate collector: A flat-plate collector can be modified by incorporating plane reflectors at the edge of the flat plate collector so that the additional radiation can be reflected into the receiver [4]. This type of arrangement increases the concentration of solar radiation. The plane reflectors (mirrors) are also called booster mirrors. The concentration ratio so obtained has a maximum value of 4.

Applications: Modified flat plate collectors can be used for water heating applications, grain and vegetable drying purposes to reduce the moisture content and for space heating applications with incorporation of rock-bed storage.

1.7.2.2 Evacuated tube collectors

Evacuated or vacuum tube collectors are used to heat pipes for their core instead of passing working fluid directly through them. Evacuated heat pipe tubes (EHPTs) are composed of multiple evacuated glass tubes each containing an absorber plate fused to a heat pipe. The heat is transferred to the heat transfer fluid which is then used for domestic hot water applications or hydronic space heating system in a heat exchanger. The vacuum inside of the evacuated tube collectors have been proven to last more than 25 years, the reflective coating for the design is encapsulated in the vacuum inside of the tube, which will not degrade until the vacuum is lost. Convection and conduction heat losses are greatly reduced by the Vacuum that surrounds the outside of the tube, therefore achieving greater efficiency than flat-plate collectors, especially in colder conditions.

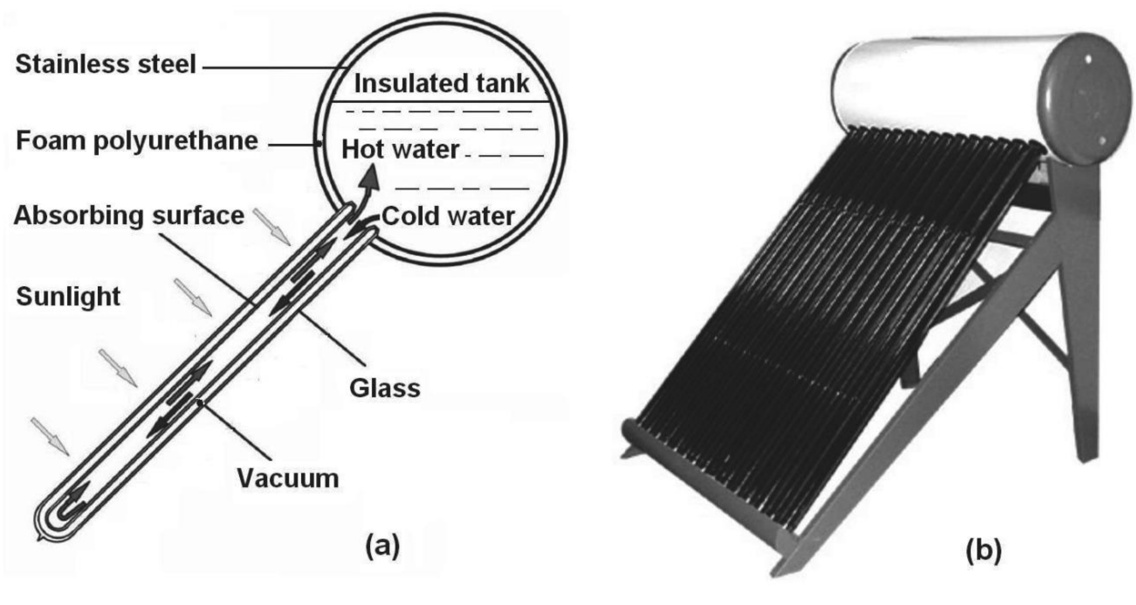


Fig : 2b: Evacuated-tube solar water heater. (a) Schematic of evacuated-tube solar water heater. Each heat collector is a double-walled glass tube. The system works automatically under the principle of natural convection. (b) Photograph of system. Source : [4]

Applications: Evacuated tube collectors can be used for water heating applications and for space heating applications. These can also be employed for various other solar thermal applications.

1.7.3 Cylindrical Trough Concentrator (CTC)

It CTC is a cylindrical trough with spherical profile reflector is very similar to Parabolic Trough with a metal tube (blackened) receiver placed at its focal line. These types of concentrators can be rotated about an axis to track the sun. The black coating of the tube increases the absorption. However after the development of the parabolic concentrators the use of spherical concentrator is reduced as with same space parabolic concentrators can deliver more energy and temperature. The concentration ratio ranges from 5-25.

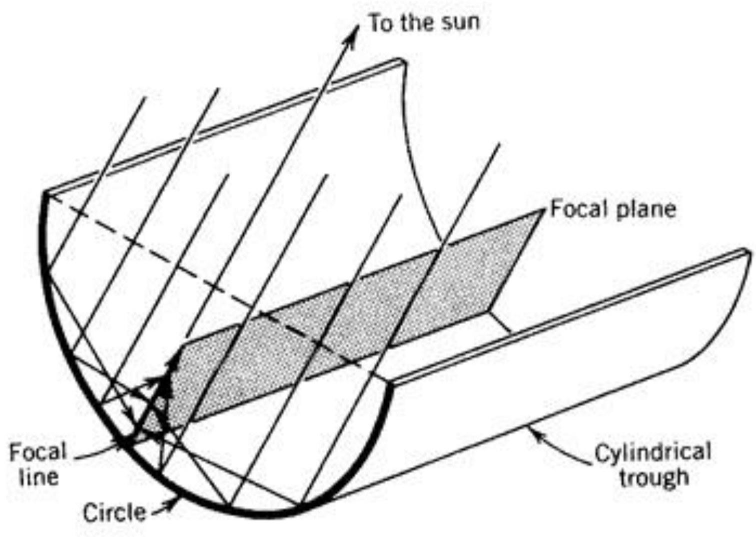


Fig : 3 Cylindrical Trough Collector
Image source: [39]

Applications: This technology can be used for water purification, steam generation purposes and solar chemistry.

1.7.4 Parabolic Trough

It is the most employed concentrated technology accounting a major share in the CSTs [7]. This technology is based on the fact that when sun rays falls on the parabolic mirrors they are directed towards the receiver placed at the focal line. The receivers have a special coating ensuring maximum energy absorption and minimum infrared re-radiation. The heat produced in the receiver carried by the heat transfer fluid which is transferred to the steam generator producing steam which runs the turbine. Single axis tracking of the sun in sufficient thereby producing long collector modules [8] .The temperature of around 400°C can be produced with this technology [10].

Applications: Parabolic trough can be used for desalination, water purification and steam generation purposes [36].

1.7.5 Parabolic Dish Reflector

It has a point focus and tracks the sun in two axes, concentrating the solar energy at the focal point. The receiver placed at the focal point stores the thermal energy which is absorbed by the heat transfer fluid. The temperatures of around 1500°C can be achieved with this arrangement [9]. The concentration ratio is in the range of 600-2000[8].

Applications: On the account of high temperature produced, parabolic dish can be used for power generation, cooking, high temperature water for laundry, solar furnaces and desalination purposes.

1.7.6 Compound Parabolic Concentrator

The booster mirrors in case of a modified flat plate collectors are replace by parabolic mirrors in this case. These mirror segments are oriented such that the focus of one segment lies at the base of the other segment which is in contact with the receiver. The acceptance angle is large for this arrangement. With this type of arrangement the rays which falls in the central region reaches the absorber directly whereas those falling near the edges undergo certain reflections. The concentration ratio that can be achieved ranges from 3-7[4].

Applications: It has applications in integrated collector storage, direct circulation, indirect water heating systems, heat pumps, absorption systems and solar chemistry systems [9]

1.7.7 Fixed Mirror Solar Concentrator

There is a practical difficulty in manufacturing a large cylindrical parabolic shape mirror, therefore long narrow mirror strips are used in this type of concentrator. These mirror strips are arranged on a circular reference cylinder while the receiver tube is tracked. The receiver tube rotates about the center of curvature of the reflector module to track the sun. The concentration ratio is approximately equal to the number of mirror strips [4].

Applications – This technology finds use in water heating, drying and space heating requirements.

1.7.8 Fresnel Concentrator

1.7.8.*1 Linear Fresnel Lens Refractor*

This type of collector consists of fine, linear grooves on the surface of a refractory material (generally optical quality plastic) on one side and flat on the other side. The angle of the groove is such that it behaves similar to a spherical lens. Temperature ranges between 150°C to 300°C with a concentration ratio of 10 to 30. The incident radiation is converged to a focal line where a receiver tube is provided [4].

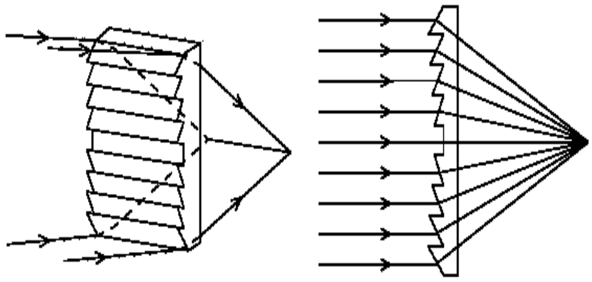


Fig : 4 a Fresnel lens

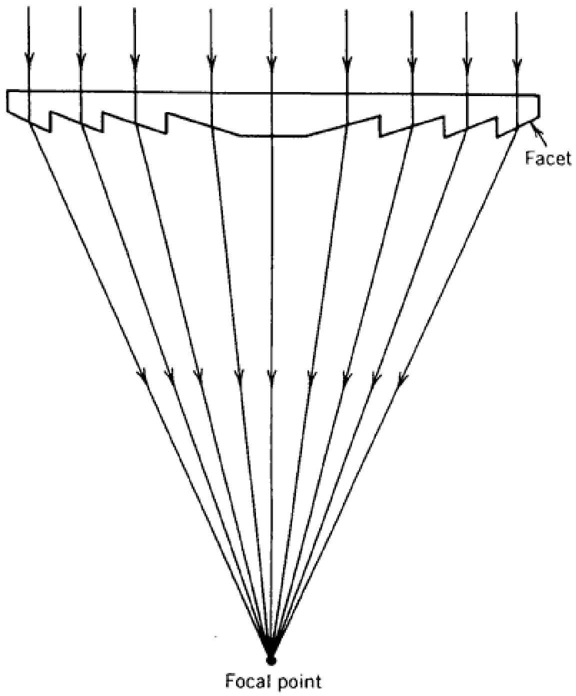


Fig : 4 b Fresnel lens
Image source: [39]

1.7.8.*2 Linear Fresnel Mirror Reflector*

This type of concentrator is an array of linear mirror strips which have a same focus and concentrate light on to a fixed receiver mounted on a tower, large absorbers can be constructed as absorbers are generally stationary. However the reflectors are moving about it’s linear axis to focus the sunlight to the fixed absorber. The greatest advantage of this type of system is that it uses flat or elastically curved reflectors which are cheaper compared to parabolic glass reflectors[9].

Additionally, these are mounted close to the ground, thus minimizing structural requirements. The first to apply this principle was the great solar pioneer Giorgio Francia, who developed both linear and two-axis tracking Fresnel reflector systems at Genoa, Italy in the 60s.

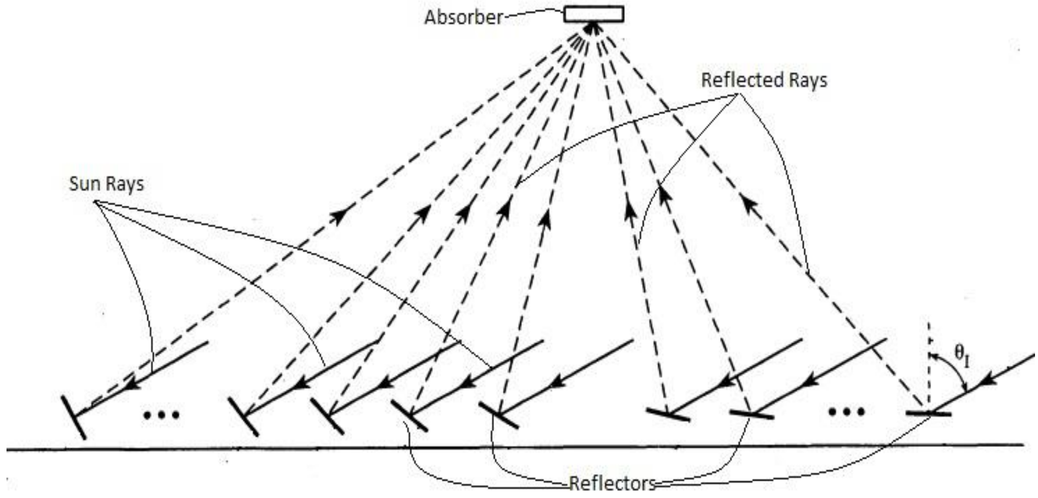


Fig : 4 c Fresnel Mirror Reflector
Image source: [39]

1.7.8.3 Circular Fresnel lens Concentrator

These lenses are used where high flux is required, such as with silicon solar cells or with gallium arsenide solar cells. The Fresnel lens here is divided into thin circular zones. The tilt of each zone is adjusted to make the lens approximately a spherical one. The concentration ratio of 2000 can be achieved. Fig 7

Applications- Fresnel reflector can be used for various thermal applications including power generation by steam and solar chemistry systems.

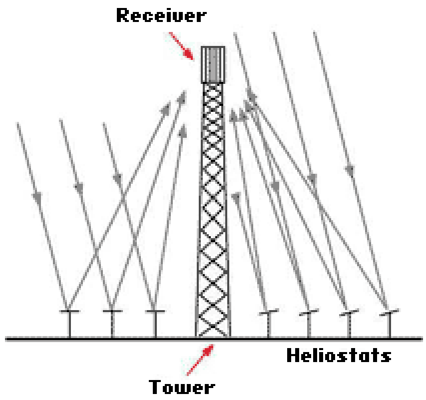
1.7.9 Hemispherical Bowl Mirror Concentrator

It is an example of linear focal optics and uses two-axes tracking. In this arrangement hemispherical bowl is fixed while the receiver tracks [11]. The rays entering the hemisphere crosses the paraxial line after the reflection at a point between the focus and the mirror surface. The absorber is driven around a polar axis at a constant angular speed or adjusted periodically.

Applications: This technology has applications in cooking, process heat and solar drying purposes.

1.7.10 Tower Receiver

A tower receiver or a solar power tower has an arrangement of heliostats (dual-axes tracking reflectors) which are used to concentrate the rays falling on them to the top of a tower. The temperature generated at the receiver is 500°C to 1000°C in which working fluid is used as a heat source for power generation or for storage systems [12].



Elevation

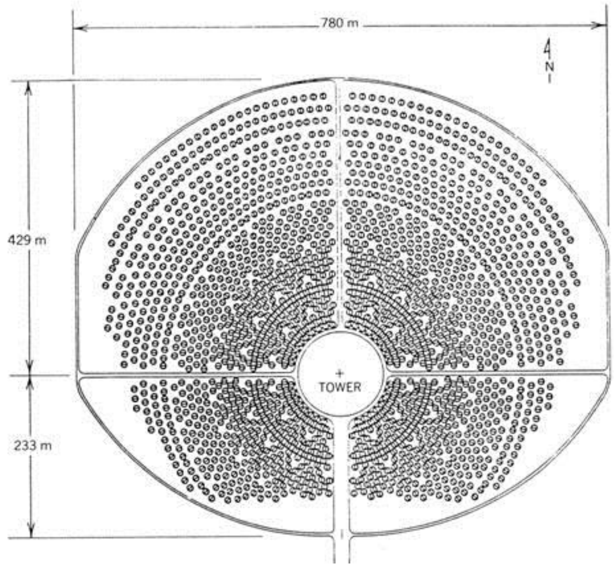


Fig : 5 Plan of tower receiver Power Plant
Image source: [39]

Applications: Tower receiver has exclusive applications in solar furnace and generating power by producing steam in a solar power plant on the account of very high temperatures .The waste heat from a solar power plant can be used to run a solar desalination process.

1.7.11 Dish Stirling

Dish Stirling has a large reflective dish to concentrate the incoming solar radiation at the focal point of the reflector. It also uses a two-axes tracking system. The working fluid in the receiver has a temperature between 250°C to 700°C and is used by Stirling engine for the generation of power [12].Dish Stirling has the highest efficiency of all the solar technologies available. It has an efficiency of 30% compared to 15% efficiency of solar photovoltaic [13].

Applications: The Stirling engine has many a application from heating and cooling to underwater power systems. A Stirling engine can function in reverse as a heat pump for heating or cooling. Other uses include: combined heat and power, solar power generation, Stirling cryo-coolers, heat pump, marine engines, and low temperature difference engines

1.7.12 Enclosed Trough

An enclosed trough consists of a solar thermal system within a greenhouse-like glasshouse. The purpose of the glasshouse is to make sure that the solar thermal system is from negative impacts and its efficiency is not effected [14]. Curved shaped reflecting mirrors are used which are suspended by wires from the ceiling. A single axis tracking system is used and the sunlight is concentrated on a network of steel pipes which are also suspended from the glasshouse structure [15]. Water is made to flow through these pipes to produce steam. Creating a protecting environment helps the mirror from the effects of wind and higher temperature is achieved with prevention from dust deposits on the mirrors [14].

Applications: These systems have applications in the process heat requirement and power generation purposes.

1.8. Integration of CST with Other Technologies

1.8.1 Concentrated Solar Thermoelectricity Generation

It is a well-known fact that when junctions between two dissimilar metals and exposed to different temperatures they can deflect a compass [16]. In other words a potential differ an electric potential is set up and a current flows if the closed circuit. This Seebeck effect can be initiated by using a concentrating mirror arrangement which directs the incoming solar radiation at a fixed point where we want to make the junction hot for a given pair of metals [18]. Thus a suitable temperature difference can be generated which further creates the potential difference across the junction.

1.8.2 Solar Vapour Refrigeration

The compressor of an air-conditioner raises the temperature of the refrigerant fluid. If a concentrator is incorporated with the existing vapour compression system, the rays falling on the solar concentrator can be used to provide a portion of the temperature increase thereby reducing the workload on the compressor considerably. This will enable the system to consume less electricity.

1.9. Concentrated Solar Thermal (CST) Applications in India

India has an immense potential of solar energy. A fairly large amount of fossil fuels, especially electricity & fuel oil are being consumed for process heat, space cooling and various other thermal applications. In the majority of the cases, heat is required in the form of high pressure steam/air/oil between 90°C to 350°C.

Solar water heating is a well-established technology and is used exclusively but is limited to the temperature of 90°C. Whereas CST can provide temperatures in the range of 100°C to 450°C [17].

1.9.1 Status of CST Technology

In India there are mainly three types of CSTs.

i) Dish Solar Cooker to cook food for 10 to 40 people which are manually tracked.
ii) Fixed focus E-W automatically tracked elliptical dishes (Schaffer) for direct indoor cooking for about 50 to 100 people and generation of steam for laundry, space cooling of any capacity.
iii) Fully tracked dual-axes Fresnel dishes.

All the above mentioned technology are in operation with the support of MNRE in the recent past years.

1.9.2 Limitations and Hindrances with the CST technology

Lack of awareness about the technology, problems in large-scale promotion of CSTs, deficiency of information on successful projects, non-availability of evacuated tube receiver for CSTs, fool-proof technology, requirement of large space, performance data on CSTs with varying DNI is not available. So efforts are required to promote CST on a bigger scale and technological advancements are needed to overcome the existing difficulties.

Chapter 2: LITERATURE REVIEW:

While doing an exhaustive literature review, I came across many new developments in the field of Solar thermal Energy. Here I intend to present few of all the literatures that have been studied. The literature consist research papers mainly from the solar collectors and application of alternative working fluid in solar energy. The literatures based on solar energy collectors are reviewed and the variations of efficiency to some parameters e.g. solar radiation, mass flow rate, temperature difference etc are studied along with various workable thermodynamic Cycles. Below you can read some of the most influential material came across with observations and conclusions pointed out:

1. According to the studies conducted in 2012 by **Farzad et al** [22]

➢ Energy And Exergy Efficiencies show behavioral conflicts in many Cases.

- While an increase in inlet temperature of fluid leads to a reduction in energy efficiency of collector, but an overall gain in exergy efficiency and even to its maximum.
- Similarly, while an increase in mass flow rate leads to gain in energy efficiency of the collector, it shows inverse effect on exergy efficiency.
- Most of losses in exergy occur during the absorbing process in the collector's absorber plate.
- Increasing water temperature at inlet and decreasing water mass flow rate can be effective on decreasing these destructions

2. **Madhukeshwara. N and E. S. Prakash**[19], investigated in 2012 the performance characteristics of solar flat plate collector with different selective surface coatings. And concluded, that

➢ Absorber coatings has influence on the performance of flat plate collectors:

- Absorber coating influences the absorber temperature and hence the water temperature in the collector.
- Maximum temperature of hot water in the storage tank is obtained for black chrome coating followed by the matt black and solchrome coatings.
- Difference in temperature of hot water outlet and cold water inlet is maximum for black chrome during experimentation for a collector tilted at an angle.
- Performance of solar thermal absorber can be improved by change of absorber materials and coating thickness.

3. **Choi** in 1995 introduced the term Nanofluid as the suspension of metallic or non-metallic Nano-particles in a base fluid [2].

- A substantial increase in thermal conductivity, viscosity and heat transfer coefficient are the unique characteristic of nanofluid [26, 28].
- Solar collector is a well-established technology. However the thermal efficiency of these collectors are limited by the absorption properties of the working fluid, which is very poor for typical conventional solar collector.
- Heat transfer enhancement in solar devices is one of the key issues of energy saving and compact design. The poor heat transfer properties of these conventional fluids are the primary obstacles for high compactness and effectiveness of the system.

Nanofluids has attracted many researchers for analyzing there performance in flat plate collectors.

4. **M.A. Alim, Z. Abdin** in 2013-14 theoretically analyzes entropy generation, heat transfer enhancement capabilities and pressure drop of an absorbing medium with suspended nanoparticles (Al_2O_3, CuO, SiO_2, TiO_2 dispersed in water) inside a flat plate solar collector.[26, 35]
   - Steady, laminar axial flow of a nanofluid is considered.
   - The nanofluids considered have different nanoparticles volume fractions and volume flow rates in the range of 1–4% and 1–4 L/min, respectively.
   - Based on the analytical results they concluded the CuO nanofluid could reduce the entropy generation by 4.34% and enhance the heat transfer coefficient by 22.15% theoretically compared to water as an absorbing fluid.
   - It also has a small penalty in the pumping power by 1.58%.

5. **S.C.Vijayakumaar** conducted experiments in 2013[28]

- Found out that using 0.5 wt.% CNT nanofluid increases the efficiency of collector in comparison with water as working fluid by 39%.

  - They used Commercial single walled CNT of average diameter 1 nm with thermal conductivity of 3500 W/mK with Polysorbate 80 (Polyethylene Sorbitan Monooleate) as surfactant for dispersion of CNT.
  - The double distilled water was used throughout the studies as base fluid.
  - Using Sorbitan 80 as the dispersant Using ultrasonic vibration (with CYBERLAB ultrasonic steri-c1eaner).
  - CNT nano powder of wt. 10 gram and suitable amount of surfactant was added to double distilled water. The solution was kept in ultra-sonicator bath for 45 minutes.
  - After preparation of the fluid test were conducted in accordance with ASHRAE standard was followed to test the thermal performance of solar collector. It was determined by obtaining the values of instantaneous efficiency.

6. **Brian D. Iverson** et al reported that Supercritical CO_2 (s CO_2) Brayton cycle has shown significant efficiency benefits especially as solar-thermal power plants increase their operating temperatures.[27]

- Recommended that Supercritical CO_2 (sCO_2) Brayton cycles continue to be pursued for solar- thermal energy applications.

  - Due to thermal capacitance in the system and piping variable Heat load operation which is a common phenomenon to a solar resource, appears manageable especially for short durations.
  - For large capacity factors and indirect systems, heat exchange between CO_2 and a secondary fluid amenable to solar is also required.
  - This represents a significant challenge in terms of material selection as well as heat exchanger design.

7. **Paschalia Mavroua, Athanasios** in 2014 investigated the performance of binary working fluid mixtures in a low temperature solar Organic Rankine Cycle (ORC) system including heat storage.

➢ Results indicate that mixtures at different compositions and concentrations may have a significantly different performance in terms of parameters such as generated work, required collector aperture area and so forth.

- They investigated system performance for a real solar radiation profile for an entire year of operation.
- Inclusive, steady-state mathematical models are used for the simulation of both the solar collectors and the ORC.
- The effects of different mixtures on important operating parameters were also investigated.
- Neopentane- based mixtures appear promising for high overall performance for solar ORCs.

8. **Zhong Ge, Huitao Wang** et al used the first and second law of thermodynamics to study the performance of a flat plate solar collector.

- They observed the effects of ambient temperature, solar irradiance, fluid inlet temperature, and fluid mass flow rate on useful exergy rate and exergy loss rate .
- The useful heat rate and useful exergy rate have conflicting behavior in many cases.
- Thus, selecting an appropriate evaluation criterion (energy or exergy) for the collector according to specific conditions is recommended.
- Solar irradiance considerably affects both the useful heat rate and useful exergy rate.
- High performance is based on appropriate solar irradiance.
- The optimum fluid inlet temperature varies, and it is mainly affected by heat loss because environmental parameters change during the day.

9. **Mouna Hamed, Ali Snoussi, Ammar Ben Brahim** performed a study, a detailed simulation model for a flat plate solar collector considering the transient properties of its different zones was developed. This is done by deriving governing equations for each layer in the solar collector. general assumptions are taken for the establishment of the mathematical model. They concluded that, with variations of various

parameters such as the water flow rate and the inlet temperature affect the performance of the collector. From the dynamic simulation that had been carried out, the following conclusions can be drawn: The main cause of exergy destruction in collector is the difference between the temperature of the absorber plate and the temperature of solar radiation while the effect of exergy destruction by pressure drop is very small. Increasing of inlet water temperature can be effective on decreasing the total exergy but it has an inverse effect on energy efficiency. An increase in the water flow rate leads to an increase in the energy efficiency.

10. In studies conducted by **X.R. Zhang** along with other researchers (while working on Project on ‘‘Next Generation Zero-Emission Energy Conversion System’’ of Ministry of Education, Culture, Sports, Science and Technology, Japan) observed:[13]

➢ Proposed a solar collector using supercritical CO_2 as working fluid in order to investigate and estimate the CO_2-based solar collector characteristics.
   - Properties such as temperature and pressure in the collector, flow rate, and collector performances for CO_2 working fluid under various weather conditions.
   - The results show that the CO_2 temperature, CO_2 pressure and mass flow rate increase with the solar radiation, which is different from those of traditional solar collector using liquid as working fluid. The solar radiation has influence on the CO_2 states, being liquid, liquid-gas or supercritical state in the test, furthermore, affects the CO_2 mass flow rate. They also observed overall higher collector efficiency in the case of supercritical CO_2 as working fluid, which is much higher than that of water-based solar collector.
   - They supported the idea of potential of the supercritical CO_2-based solar collector in the field of solar thermal utilization.

11. **Pardeep Garg, Pramod Kumar, Kandadai Srinivasan**[38], investigated Supercritical carbon dioxide based Brayton cycle for solar power applications and compared it with trans- and sub-critical operations of the same fluid.
   - While the thermal efficiency increases almost linearly with low side pressure in the sub- and trans-critical cycles, he reported that supercritical cycle is capable of producing power even at a lower source temperature (820 K).
   - The reasons for lower efficiency than in an ideal cycle are extracted from an irreversibility analysis of components, namely, compressor, regenerator, turbine and gas

cooler. Low sensitivity to the source temperature and extremely small volumetric flow rates in the supercritical cycle could offset the drawback of high pressures through a compact system.

12. **Emmanuel Cayer and Nicolas Galanis**: analysed in detail for a trans-critical power cycle based on carbon dioxide, using an industrial low-grade stream of process gases as its heat source is presented.

    - The results have been calculated for fixed temperature and mass flow rate of the heat source, fixed maximum and minimum temperatures in the cycle and a fixed sink temperature by varying the high pressure of the cycle and its net power output.
    - It has been reported that the augmentation of the net power output produced from the limited energy source has no influence on the results of the energy analysis, decreases the exergetic efficiency and increases the heat exchangers' surface.

Changing the net power output has no significant impact on the high pressures optimizing each of the four steps.

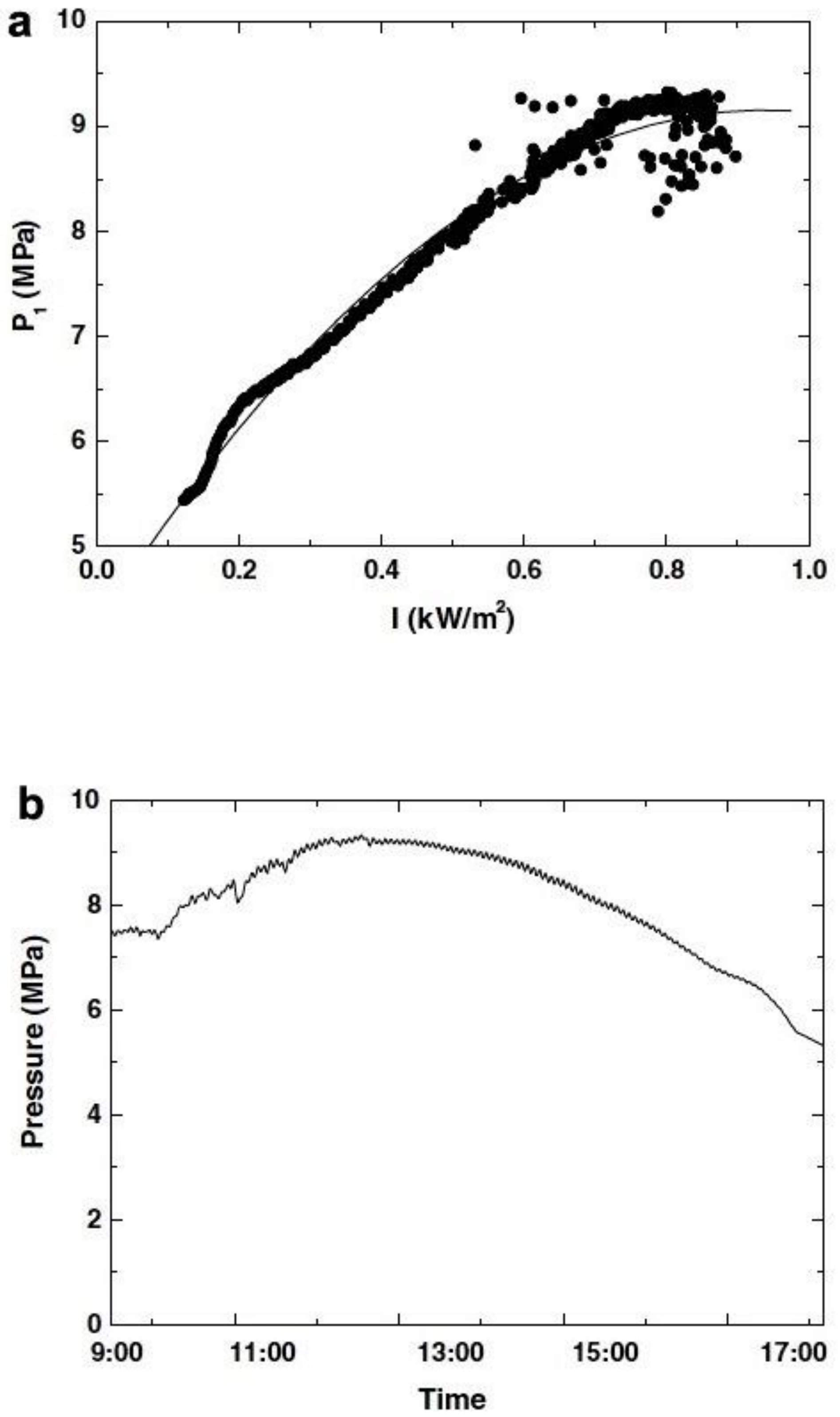


Figure: 6 Variations in the CO_2 Pressure Measured with the Solar Radiation and Time as studied by X.R. Zhang along with other researchers. Source [13]

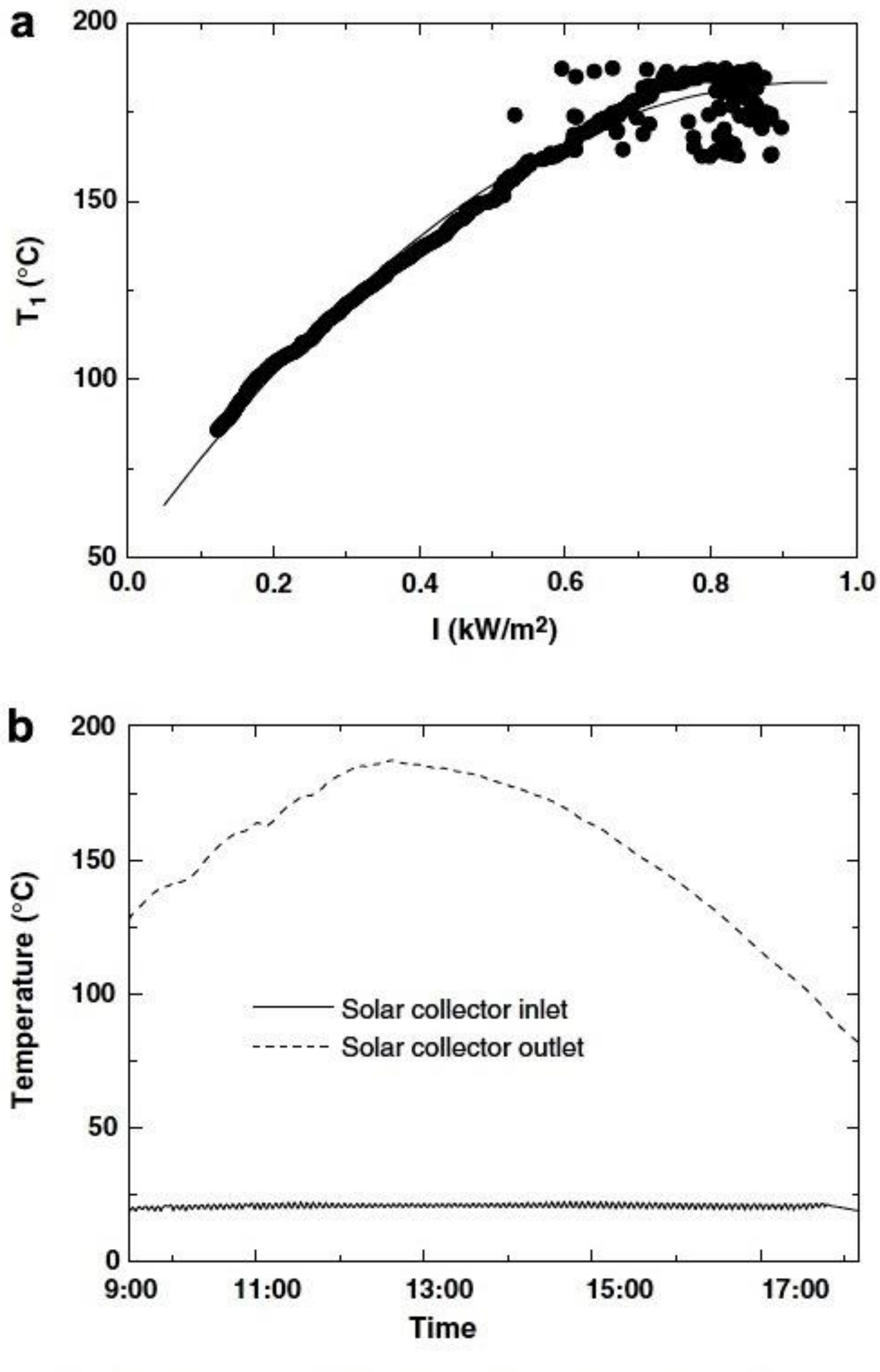


Figure: 7 Variations in the CO_2 temperature measured with the solar radiation and time as studied by X.R. Zhang along with other researchers. Source [13]

Important observation from literature review (summary):

- Carbon dioxide has been studied on Superheated Rankine and Superheated Brayton Cycle by many researchers and most of them report the favorability of the gas for solar thermal applications.
- Changes in design of Evacuated tube collector have resulted in higher temperature and pressure delivery, i.e. Maximum Temperature 180°C at 9 bar , Average Temperature 160°C at 7 bar.
- **Factors effecting performance of solar collector**

  Performance of solar collector can be concluded by the net energy delivered for the delivered outlet Temperature on a variable mass flow rate. These factors depend on various parameters such as Thermodynamic and physical properties of material of construction, Construction design etc.

- Some of them are as follows
  1. Absorber Surface Material
  2. No. of Glazing covers
  3. Spacing between the Cover and Plate as well as with the second cover
  4. Effect of Shading
  5. Collector Tilt
  6. **Thermal properties of working Fluid used in collectors.**
- Researchers have studied various working fluids such as water, binary mixtures of water with other materials such as Nano-particles of both metallic and non-metallic in nature, Alcohols, and gases such as Helium, carbon-dioxide etc.

This study can however also be applied to various collectors.

Table:2 Solar Collectors and Concentrating Ratios

Collector	**Conc. Ratio**	**Temp. °C**
Non Concentrating Type Collector	1	< 70
High efficiency FPC / Evacuated Tube Collector	1	60-120
Fixed Concentrator	3-5	100-150
Parabolic trough	10-50	150-350
Linear Fresnel	<100	300-500
Parabolic dish collector	200-500	250-700
Central receiver	500>3000	500 >1000

Literature Review

S.No.	Topic	Year	Author	Analysis	Working Fluid
1	Energy and exergy analysis of flat plate solar collectors in transient behaviors,	2014	Mouna Hamed, Ali Snoussi, Ammar Ben Brahim	Energy and exergy analysis	Water
2	A comprehensive feasibility study of applying solar energy to design a zero energy building for a typical home in Tehran	2014	J. Eshraghi, N. Narjabadifam, N. Mirkhani, S.S. Khosroshahi, M. Ashjaee,	Energy study	Natural Air
3	Exergy Analysis of Flat Plate Solar Collectors	2014	Zhong Ge, Huitao Wang, Hua Wang , Songyuan Zhang and Xin Guan,	Exergy Analysis	Water
4	Assessment of Working Fluid Mixtures for Solar Organic Rankine Cycles	2014	Paschalia Mavroua, Athanasios I. Papadopoulosa, Mirko Stijepovicb, Panos Seferlisc, Patrick Linkeb, Spyros Voutetakis	Performance Analysis	Binary mixtures
5	Analyses of exergy efficiency and pumping power for a conventional flat plate solar collector using SWCNTs based nanofluid	2014	Z. Saida, R. Saidura, N.A. Rahimb, M.A. Alim	Exergy Efficiency	SWCNTs based Nanofluid
6	Analyses of entropy generation and pressure drop for a conventional flat plate solar collector using different types of metal oxide nanofluids	2013	M.A. Alima, Z. Abdinb, R. Saidura, A. Hepbaslid, M.A. Khairula, N.A. Rahim	Entropy Generation and Pressure drop	Metal oxide Nanofluid
7	Supercritical CO2 Brayton cycles for solar-thermal energy	2013	Brian D. Iverson , Thomas M. Conboy, James J. Pasch, Alan M. Kruizenga	Brayton cycle	sCO2

8	Effect of CNT-H20 Nanofluid on the Performance of Solar Flat Plate Collector-An Experimental Investigation	2013	S.C. Vijayakumaar, R. Lakshmi Shankar, Dr.K.Babu	Energy Analysis	CNT Nanofluid
9	An investigation on the performance characteristics of solar flat plate collector with different selective surface coatings	2012	Madhukeshwara. N, E. S. Prakash	Selective Surface Coatings	Water
10	Solar systems for heating and cooling of buildings,	2012	Energy Procedia	Energy Analysis	Natural Air
11	Investigation of thermal performance of flat plate and evacuated tubular solar collectors according to a new dynamic test method	2012	Weiqiang Kong, Zhifeng Wang, Jianhua Fan, Bengt Perers, Ziqian Chen,Simon Furbo, Elsa Andersen	Performance Analysis	Performance Analysis
12	Energetic and exergetic evaluation of flat plate solar collectors	2012	Farzad Jafarkazemi, Emad Ahmadifard	Energy and exergy analysis	Water
13	Role of nanofluids in solar water heater	2009	Natarajan E, Sathish R	Performance Analysis	Nanofluids
14	An experimental study on evacuated tube solar collector using supercritical CO_2	2008	X.R. Zhang, H. Yamaguchi	Performance Analysis	Carbondioxide
15	Renewable Energy Resources	2006	John Twidell and Tony Weir,	TEXT	BOOK
16	Principles of Solar Engineering		Dr. Yogi Goswami, Frank Kreith, Jan F. Kreider		
17	Solar Engineering of Thermal Processes	2006	Duffie and W. A. Beckman.		
18	Physics of Solar energy	1999	C. Julian Chen.		

Chapter 3: RESEARCH GAP AND PROPOSAL

Research gaps:

On the basis of the comprehensive literature review, comparison for net usable energy delivered by flat plate solar collectors using various fluids as an absorbing medium is the new area which is still wide open for investigation.

Some of the important observations are:

1. Carbon dioxide has been studied on superheated Rankine and Superheated Brayton Cycle, however a new cycle can be deduced and experimented for solar thermal applications for specific working fluids.
2. Nano-particles have been an area of interest for some time now and various concerns still remain unanswered :
    a. High Production Cost
    b. Difficult storage
    c. Problem of transportation during phase change process in a cycle.
    d. Various Health and Environmental Concers.

Possible Alternative Working fluids:

1. Helium gas
2. Carbon dioxide gas
3. Ammonia Water mixture
4. Ammonia vapors

Objective:

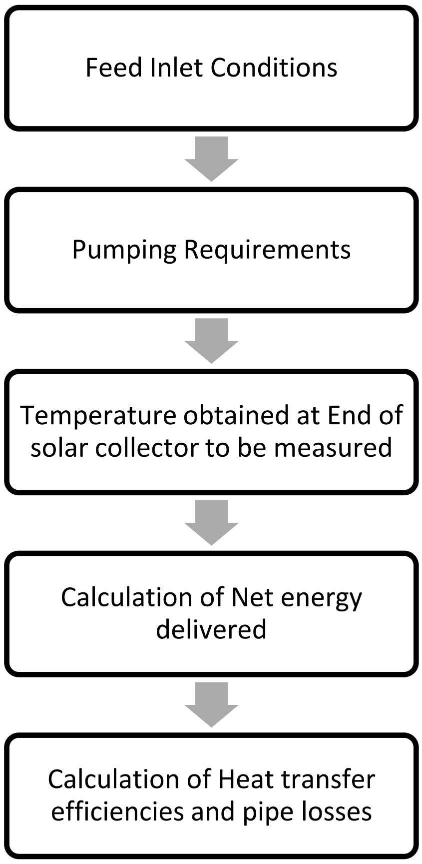
The main objective of this thesis is to study the Combined Heat and Power Delivered by Carbon dioxide while working in a Non-Concentrating type solar collector under Various Thermodynamic Cycles.

Probable advantages of this system:

- Considerable reduction in fuel consumption.
- Low maintenance cost associated.
- Reduction in cost of delivery energy.
- Reduction in emissions.

Approach for Analysis (Practical):

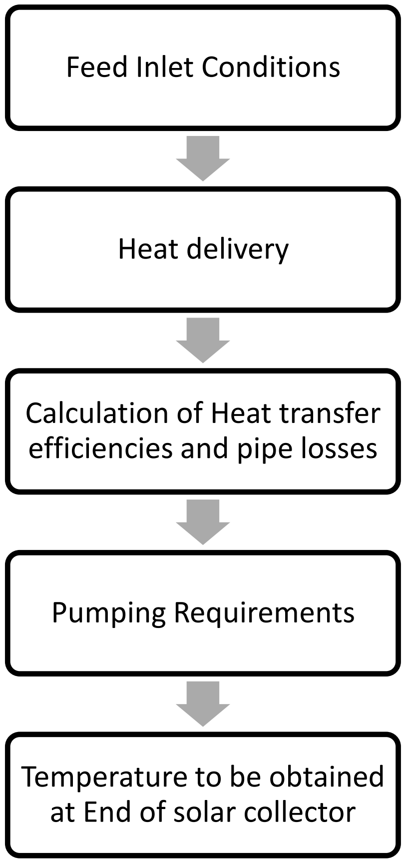
For Solar collector:



- **Feed Inlet Conditions:** At first we must know the inlet condition at the feed valve such as Temperature, Pressure.
- **Pumping Requirements:** Pumps and valves have to be provided to control the Mass / volume flow.
- **Temperature obtained at End of solar collector:** Based on the Data of solar Irradiance of the area selected, Area of the collector surface required would highly affect the net energy delivered.
- And the temperature might remain within a range but the flow rate of energy may depend upon the area of collector.
- **Calculation of Net energy delivered:** Based on the Data obtained we need to calculate and compare the availability for various fluids under consideration.
- **Calculation of Heat transfer efficiencies and pipe losses:** The desired output from solar collector must be equal to the sum of the conditions required at the inlet valve and the loss occurred during pipe flows and during heat transfer.

Approach for Analysis (Mathematical):

For Solar collector:



Feed Inlet Conditions: At first we must know the inlet condition at the feed valve such as Temperature, Pressure and Mass / volume flow rate etc.

Heat delivery: Heat must be delivered from collector surface to heat carrying medium (various fluids in our case).

Calculation of Heat transfer efficiencies and pipe losses: The desired output from solar collector must be equal to the sum of the conditions required at the inlet valve and the loss occurred during pipe flows and during heat transfer.

Pumping Requirements: Pumps and valves have to be provided to control the Mass / volume flow.

Temperature obtained at End of solar collector: Based on the Data of solar Irradiance of the area selected, Area of the collector surface required would highly affect the net energy delivered.

And the temperature might remain within a range but the flow rate of energy may depend upon the area of collector.

Chapter 4: Favorability for Co_2 as Working Fluid and Lenoir as Preferred cycle

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1. **Thermodynamic properties**
   - Critical temperature: 31°C
   - Critical Pressure: 7.38 MPa
   - Triple point : -56.6°C , 5.18 bar
   - Lower pressure drops
   - Good heat transferability

2. **Advantages of CO_2 over other working fluids**
   - Adaptability
   - Easy Natural availability
   - Easy handling
   - Good heat retention properties

3. **General Properties of CO_2**
   - Non Inflammable
   - Non toxic
   - Friendly to environment (Can be easily processed)
   - Chemically/ thermodynamically stable

Lenoir cycle

After studying basic fundamentals of thermodynamic cycles such a Carnot, Rankine, Brayton, Atkinson, Sterling cycle etc the purpose to choose Lenoir as preferred cycle as this cycle does not require any phase change process and alomg with that it also doest notrequire any compressor work to gain pressure. Heat Addition is the primary process which is also responsible for gain of pressure and hence reduces compressor work.

However, constant volume heating may pose a problem of intermittent operation, but while designing a system Solar Field of Evacuated tube collectors can be Designed accordingly.

Key Features of the solar field to be designed for this purpose:

1. Large solar field which can provide time to collectors for constant volume heating while Gas is being delivered to the turbine from a collector with ready properties of the gas.
2. Supply of gas can be controlled by pressure valves and firing order sequence.
3. Since only one process is constant volume heating and all other process are Flow process hence solar field can be designed to maintain constant supply to the turbine.

System Accessories

1. A small biogas based gas heating chamber can be installed with the first solar field to provide backup for the intermittent sun.
2. Solar concentrators can be designed to heat gas more quickly than non-concentrating type collector.

Temperature and pressure limits: Maximum temperature and pressure limits are decided on the basis of practically achieved parameters in an evacuated tube collector with the same working fluid.

Minimum Temperature and Pressure limits are based on atmospheric conditions.

While Temperature and Pressure for Intercooling and reheating are calculated for maximum efficiency.

T_{max} : 180°C P_{max} : 9 bar

T_{avg} : 160°C P_{avg} : 7 bar

T_{min} : 30-27°C P_{min} : 1.25 bar

CO_2 CYCLE AND DATA ANALYSIS

Lenoir Cycle

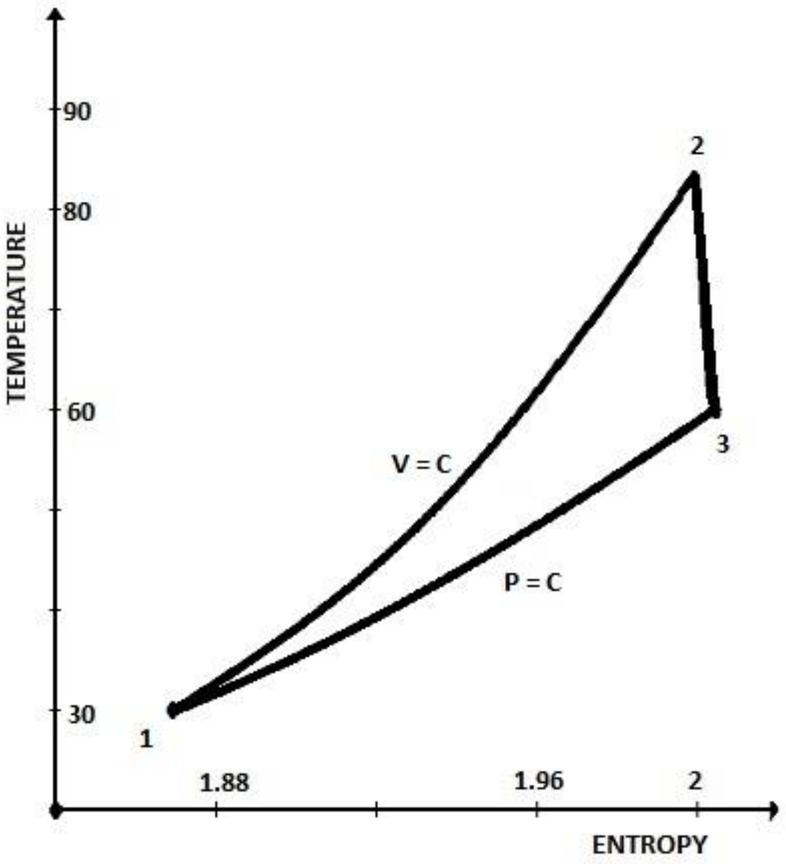

Figure:8 Lenoir Cycle on Temperature Entropy Chart

While trying a workable cycle for Carbon dioxide, we considered Lenoir Cycle, This cycle can utilize the Rise in pressure while Heating Carbon dioxide at Constant volume for producing Power in high Pressure turbine and then recovering heat for Water heating and other applications.

Table 3 : Process Table for figure 8

Process	Description
1-2	Constant Volume Heat Addition
2-3	Isentropic Expansion
3-1	Constant Pressure Heat Rejection

Table 4 : Properties on points for figure 8

	Unit	1	2	3
state of aggregation :		Gas		
Pressure (P) :	Bar	50	67	50
Temperature (T):	Celsius	30	83	60
Density (ρ) :	kg / m^3	124	124.928	98.3
Specific Enthalpy (h):	kJ / kg	451.4	507.709	494.2
Specific Entropy (s) :	kJ / kg K	1.869	1.999	2.004
Specific isobar heat capacity (C_p) :	kJ / kg K	1.69	1.326	1.267
Specific isochor heat capacity (C_v):	kJ / kg K	0.8363	0.804	0.7828
Heat conductance	10^{-3} (W / m * K)	23.7	26.667	23.48
Thermal diffusivity :	10^{-7} m^2 / s	1.131	1.633	1.885
Coefficient of compressibility Z :		0.704	0.799	0.8082
Cp - Cv		0.854	0.522	0.484
Cp / Cv		2.021	1.649	1.619

Assumptions:

- All Processes are Reversible Process.
- Kinetic Energy of the working fluid at inlet and outlet of turbine is same.
- Radiation and mechanical losses are neglected.
- Arbitrary Values to identify positive work output while working with Lenoir cycle.

Cycle Parameters:

Maximum Temperature (T_{max}) : 83 °C

Pressure at T_{max} (P) : 67 bar

Minimum Temperature (T_{min}): 30 °C

Pressure : 50 bar

Now Enthalpy Calculations :

- Process: 1-2 Enthalpy change for Constant Volume Heat Addition

$$\Delta \mathbf{h_1 = h_2 - h_1} = 507.709 - 451.4 = 56.31 \text{ kJ / kg}$$

- Process: 2-3 Isentropic Expansion

$$\Delta \mathbf{W}_{1} = \mathbf{h}_{2} - \mathbf{h}_{3} = 507.7 - 494.2 = \mathbf{13.5\ kJ / kg}$$

- Process: 3-1 Constant Pressure Heat Rejection

$$\Delta \mathbf{h}_{2} = -(\mathbf{h}_{3} - \mathbf{h}_{1}) = -(494.2 - 451.4) = \mathbf{-42.8}\ \text{kJ / kg}$$

According to First law :

$\sum Q = \sum W$ (for a closed cycle)

$\Delta \mathbf{h} + \Delta \mathbf{h}_{2} = \Delta \mathbf{W}_{1}$

56.31 - 42.8 = 13.51 kJ / kg

Heat added to cycle = ΔQ_{1} = 56.31 kJ / kg

Work done obtained from turbine: ΔW_{1} = 13.5 kJ / kg

Heat Recovery : ΔQ_{2} = - 42.8 kJ / kg

Ist Proposed Cycle for CO_2

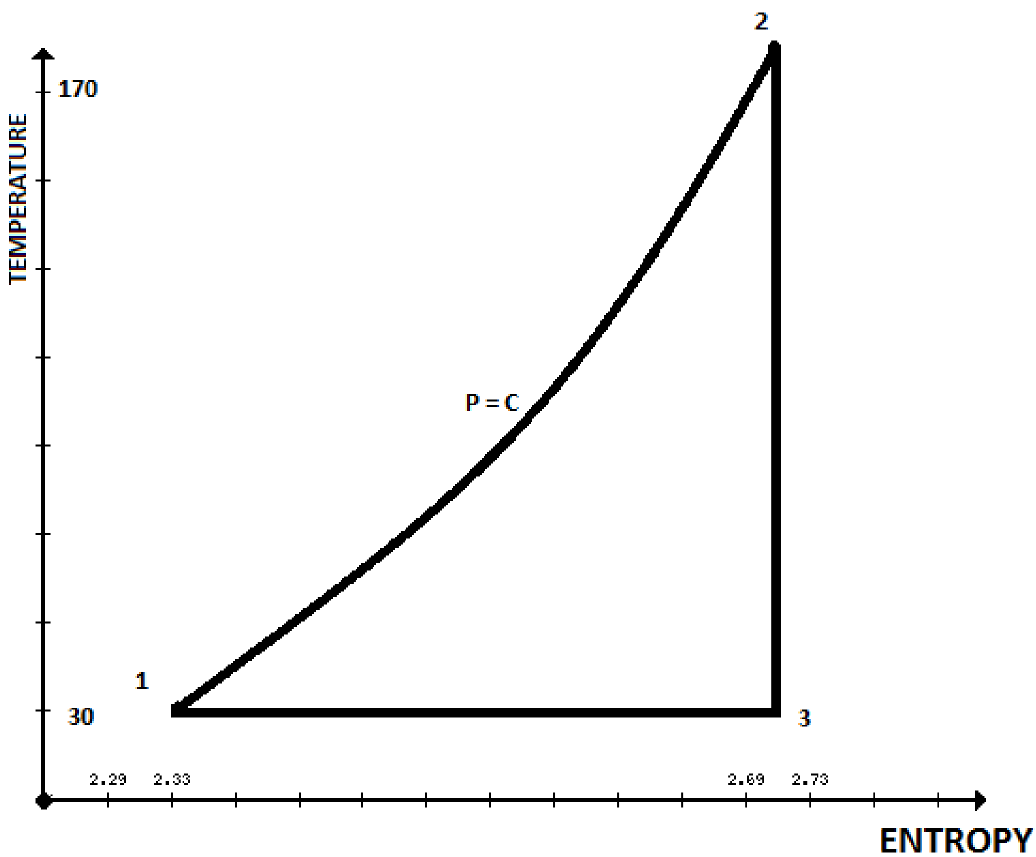

Figure 9: Ideal Power Cycle for Carbon dioxide with primary heat addition at constant pressure

Three process Cycle can be ideal cycle which can be used to produce work using Working fluid which remains in gaseous phase throughout the cycle.

Three process Cycle:

1-2 Constant Pressure Heat Addition

2-3 Isentropic Expansion

3-1 Isothermal Heat Rejection and Compression

Table 5 : Properties on Points for figure 9

Property	Unit	1	2	3
state of aggregation :		**Gas**	**Gas**	Gas
Pressure (P) :	bar	**9**	**9**	1.25
Temperature (T):	Celsius	**30**	**180**	27
Density (ρ) :	kg / m^3	**16.4688**	**10.6582**	2.2268
Specific Enthalpy (h):	kJ / kg	**502.66**	**645.48**	507.32
Specific Entropy (s) :	kJ / kg K	**2.3274**	**2.7075**	2.7257
Specific isobar heat capacity (C_p) :	kJ / kg K	**0.9132**	**0.9967**	0.8544
Specific isochor heat capacity (C_v):	kJ / kg K	**0.68146**	**0.7956**	0.6600
Heat conductance	10^{-3} (W / m * K)	**17.444**	**29.8756**	16.7990
Thermal diffusivity :	10^{-7} m^2 / s	**12.642**	**30.5808**	105.5450
Coefficient of compressibility Z :		**0.95642**	**0.9898**	0.9939

Assumptions:

- All Processes are Reversible Process.
- Kinetic Energy of the working fluid at inlet and outlet of turbine is same.
- Radiation and mechanical losses are neglected.

Cycle Parameters:

Maximum Temperature (T_{max}) : 180 °C

Pressure at T_{max} (P) : 9 bar

Minimum Temperature (T_{min}): 27 °C

Pressure : 1.25 bar

Now Energy Calculations:

- Process: 1-2 Enthalpy change for Constant Volume Heat Addition

$$\Delta \mathbf{h}\ \Delta \mathbf{h} = \mathbf{h}_{\mathbf{2}} - \mathbf{h}_{\mathbf{1}} = 645.48 - 502.66 = 142.82 \text{ kJ / kg}$$

- Process: 2-3 Isentropic Expansion

$$\Delta \mathbf{W}_{\mathbf{1}} = \mathbf{h}_{\mathbf{2}} - \mathbf{h}_{\mathbf{3}} = 645.48 - 507.32 = \mathbf{138.16 \text{ kJ / kg}}$$

- Process: 3-1 Isothermal Compression and Heat Rejection

$$\Delta \mathbf{Q}_{\mathbf{2}} = -(\mathbf{h}_{\mathbf{3}} - \mathbf{h}_{\mathbf{1}}) = -(507.32 - 502.66) = -4.66 \text{ kJ / kg}$$

According to First law:

$\sum Q = \sum W$ (for a closed cycle)

$\mathbf{\Delta Q_1 + \Delta Q_2 = \Delta W_1}$

142.82 - 4.66 = 138.16 kJ / kg

Net Heat added to cycle = ΔQ_1 = 142.82 kJ / kg

Net Work done obtained from turbine: ΔW_1 = 138.16 kJ / kg

Heat Recovery: ΔQ_2 = - 4.66 kJ / kg

Cycle with Reheat

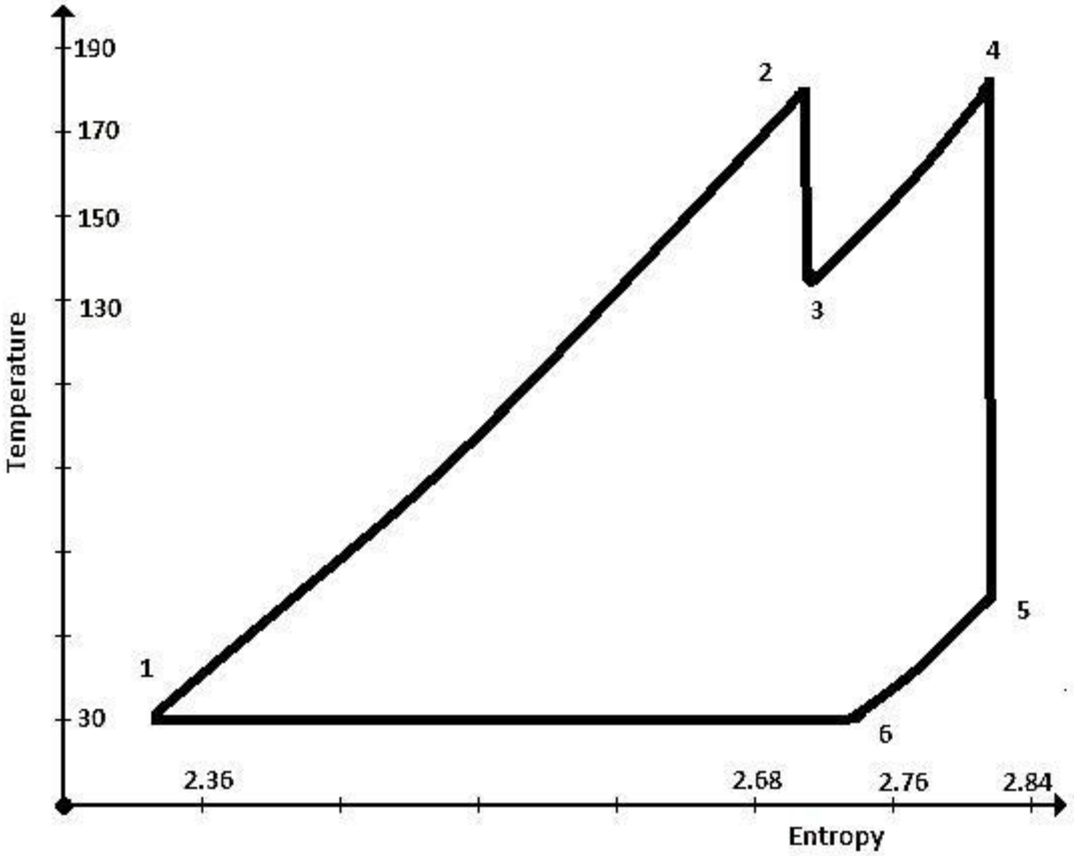

Figure 10: Ideal Power Cycle for Carbon dioxide with primary heat addition at constant pressure and reheating

Table 6 : Process Table

Process	Description
1-2, 3-4	Constant Pressure Heat Addition
2-3, 4-5	Isentropic Expansion
5-6	Constant Pressure Heat Rejection
6-1	Isothermal Heat Rejection and Compression

Table 7: Properties on Points

Property	Unit	1	2	3	4	5	6
state of aggregation :		**Gas**	**Gas**	Gas	Gas	Gas	Gas
Pressure (P) :	Bar	**9**	**9**	5	5	1.25	1.25
Temperature (T):	Celsius	**30**	**180**	135	180	60	27
Density (ρ) :	kg / m^3	**16.4688**	**10.6582**	6.5628	5.8914	1.9990	2.2268
Specific Enthalpy (h):	kJ / kg	**502.66**	**645.48**	603.22	647.08	536.01	507.32
Specific Entropy (s) :	kJ / kg K	**2.3274**	**2.7075**	2.7130	2.8150	2.8165	2.7257
Specific isobar heat capacity (C_p) :	kJ / kg K	**0.9132**	**0.9967**	0.9579	0.9893	0.8849	0.8544
Specific isochor heat capacity (C_v):	kJ / kg K	**0.68146**	**0.7956**	0.7599	0.7938	0.6921	0.6600
Heat conductance	10^{-3} (W / m * K)	**17.444**	**29.8756**	25.9740	29.7540	19.5106	16.7990
Thermal diffusivity :	10^{-7} m^2 / s	**12.642**	**30.5808**	41.5640	51.2960	131.4343	105.5450
Coefficient of compressibility Z :		**0.95642**	**0.9898**	0.9914	0.9943	0.9957	0.9939

Assumptions:

- All Processes are Reversible Process.
- Kinetic Energy of the working fluid at inlet and outlet of turbine is same.
- Radiation and mechanical losses are neglected.

Cycle Parameters:

Maximum Temperature (T_{max}) : 180 °C

Pressure at T_{max} (P) : 9 bar

Minimum Temperature (T_{min}): 27 °C

Pressure: 1.25 bar

Reheating:

$P_3 = (P_1 \ P_5)^{½}$

$= (9 \text{ X } 1.25)^{½} = 3.35$ bar

But Reheating Pressure is kept at 5 bar as average pressure output is 5 Bar from the Evacuated Tube collector

Calculations for Enthalpy:

- Process: 1-2 Constant Pressure Heat Addition

$$\Delta \mathbf{Q}_1 = \mathbf{h}_2 - \mathbf{h}_1 = 645.48 - 502.66 = \mathbf{142.82\ kJ / kg}$$

- Process: 2-3 Isentropic Expansion

$$\Delta \mathbf{W}_1 = \mathbf{h}_2 - \mathbf{h}_3 = 645.48 - 603.22 = \mathbf{42.26\ kJ / kg}$$

- Process: 3-4 Constant Pressure Heat Addition (Reheating)

$$\Delta \mathbf{Q}_2 = \mathbf{h}_4 - \mathbf{h}_3 = 647.08 - 603.22 = \mathbf{43.86\ kJ / kg}$$

- Process: 4-5 Isentropic Expansion

$$\Delta \mathbf{W}_2 = \mathbf{h}_4 - \mathbf{h}_5 = 647.08 - 536.01 = \mathbf{111.07\ kJ / kg}$$

- Process: 5-6 Constant Pressure Heat Rejection

$$\Delta \mathbf{Q}_3 = \mathbf{h}_6 - \mathbf{h}_5 = 507.32 - 536.01 = \mathbf{-\ 28.69\ kJ / kg}$$

- Process: 6-1 Isothermal Compression and Heat Rejection

$$\Delta \mathbf{W}_3 = \mathbf{h}_1 - \mathbf{h}_6 = 502.66 - 507.32 = \mathbf{-\ 4.66\ kJ / kg}$$

According to First law of Thermodynamics:

$\sum Q \approx \sum W$ (for a closed cycle)

$\Delta Q_1 + \Delta Q_2 + \Delta Q_3 = \Delta W_1 + \Delta W_2 + \Delta W_3$

142.82 + 43.86 - 28.69 = 42.26 + 111.07 - 4.66

Heat added to cycle = $\Delta Q_1 + \Delta Q_2$ = 142.82 + 43.86 = 186.68 kJ / kg

Work done obtained from turbine: $\Delta W_1 + \Delta W_2$= 42.26 + 111.07 = 153.33 kJ / kg

Heat Recovery: ΔQ_2 = - 28.69 kJ / kg

Real Cycle with Reheat and Intercooling

As isothermal Compression is a very slow process. A qausi static process is inherently and is practically not feasible for a cycle. Hence Isothermal compression is replaced be isentropic Compression and Constant pressure cooling.

CYCLE 1.

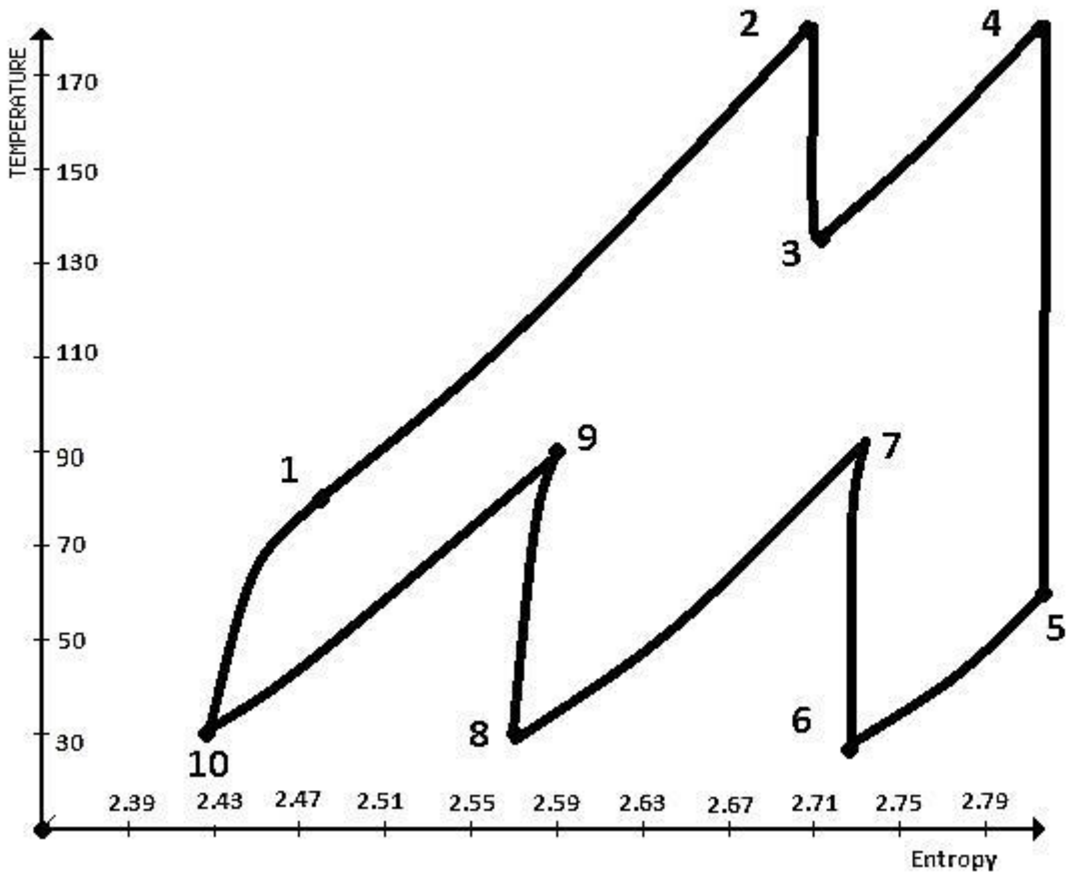

Figure 11: Combined Heat and Power Cycle for CO_2 with primary heat addition at constant pressure and reheating and Intercooling

Table 8: Process Table

Process	Description	Device
1-2, 3-4	Constant Pressure Heat Addition	Evacuated tube collector
2-3, 4-5	Isentropic Expansion	High/ Low Pressure turbine
5-6	Constant Pressure Heat Rejection	Condenser
6-7, 8-9,	Isentropic Compression	Compressor
7-8, 9-10	Constant Pressure Heat Recovery	Heat Exchanger (Thermal Water Heater)

Table 9: Properties on Points

Property	Unit	1-a	1-b	2	3	4	5	6	7	8	9	10
state of aggregation :		**Gas**	**Gas**	**Gas**	Gas	Gas	Gas	Gas	Gas	Gas	Gas	Gas
Pressure (P) :	bar	**9 (max)**	**9**	**9**	5	5	1.25	1.25	3.35	3.35	5.5	5.5
Temperature (T):	Celsius	**100**	**80**	**180**	135	180	60	27	90	30	90	30
Density (ρ) :	kg / m^3	**13.051**	**13.864**	**10.658**	6.563	5.891	1.999	2.227	4.944	5.978	8.148	9.885
Specific Enthalpy (h):	kJ / kg	**567.580**	**548.760**	**645.480**	603.220	647.080	536.013	507.319	561.702	507.985	560.340	526.020
Specific Entropy (s) :	kJ / kg K	**2.520**	**2.468**	**2.707**	2.713	2.815	2.817	2.726	2.731	2.570	2.590	2.427
Specific isobar heat capacity (C_p) :	kJ / kg K	**0.946**	**0.934**	**0.997**	0.958	0.989	0.885	0.854	0.919	0.872	0.927	0.887
Specific isochor heat capacity (C_v):	kJ / kg K	**0.736**	**0.720**	**0.796**	0.760	0.794	0.692	0.660	0.722	0.668	0.724	0.673

Assumptions:

- All Processes are Reversible Process.
- Kinetic Energy of the working fluid at inlet and outlet of turbine is same.
- Radiation and mechanical losses are neglected.

Cycle Parameters:

Maximum Temperature (T_{max}) : 180 °C

Pressure at T_{max} (P) : 9 bar

Minimum Temperature (T_{min}): 27 °C

Pressure : 1.25 bar

Reheating:

$P_3 = (P_1 P_5)^{1/2}$

$= (9 \text{ X } 1.25)^{1/2} = 3.35$ bar

But Reheating Pressure is kept at 5bar as average pressure output is 5bar from the Evacuated Tube collector.

Intercooling:

$P_3 = (P_1 P_5)^{1/2}$

$= (9 \text{ X } 1.25)^{1/2} = 3.35$ bar

$P_9 = (P_1 P_7)^{1/2}$

$= (9 \text{ X } 3.35)^{1/2} = 5.49$ bar

Calculations for Enthalpy:

- Process: 1-2 Constant Pressure Heat Addition

$$\Delta Q_1 = h_2 - h_1 = 645.48 - 548.76 = \mathbf{96.72\ kJ / kg}$$

- Process: 2-3 Isentropic Expansion

$$\Delta W_1 = h_2 - h_3 = 645.48 - 603.22 = \mathbf{42.26\ kJ / kg}$$

- Process: 3-4 Constant Pressure Heat Addition (Reheating)

$$\Delta Q_2 = h_4 - h_3 = 647.08 - 603.22 = \mathbf{43.86\ kJ / kg}$$

- Process: 4-5 Isentropic Expansion

$$\Delta W_2 = h_4 - h_5 = 647.08 - 536.01 = \mathbf{111.07\ kJ / kg}$$

- Process: 5-6 Constant Pressure Heat Rejection

$$\Delta Q_3 = h_6 - h_5 = 507.32 - 536.01 = \mathbf{-28.69\ kJ / kg}$$

- Process: 6-7 Isentropic Compression

$$\Delta \mathbf{W}_{3} = \mathbf{h}_{6} - \mathbf{h}_{7} = 507.32 - 561.7 = \mathbf{-\ 54.39\ kJ\ /\ kg}$$

- Process: 7-8 Constant Pressure Heat Rejection (Intercooling)

$$\Delta \mathbf{Q}_{4} = \mathbf{h}_{8} - \mathbf{h}_{7} = 507.98 - 561.70 = \mathbf{-\ 53.72\ kJ\ /\ kg}$$

- Process: 8-9 Isentropic Compression

$$\Delta \mathbf{W}_{4} = \mathbf{h}_{8} - \mathbf{h}_{9} = 507.98 - 560.34 = \mathbf{-53.73\ kJ\ /\ kg}$$

- Process: 9-10 Constant Pressure Heat Rejection (Intercooling)

$$\Delta \mathbf{Q}_{5} = \mathbf{h}_{10} - \mathbf{h}_{9} = 526.02 - 570.34 = \mathbf{-\ 54.32\ kJ\ /\ kg}$$

- Process: 10-1 Isentropic Compression

$$\Delta \mathbf{W}_{5} = \mathbf{h}_{10} - \mathbf{h}_{1} = 526.02 - 567.58 = \mathbf{-41.56\ kJ\ /\ kg}$$

According to First law of Thermodynamics:

$\sum Q \approx \sum W$ (for a closed cycle)

$\Delta Q_1 + \Delta Q_2 + \Delta Q_3 + \Delta Q_4 + \Delta Q_5$

$\approx 96.72 + 43.86 - 28.69 - 53.72 - 54.32$ **= 3.85 kJ / kg**

$\Delta W_1 + \Delta W_2 + \Delta W_3 + \Delta W_4 + \Delta W_5$

$\approx 42.26 + 111.07 - 54.39 - 53.73 - 41.56$ **= 3.65 kJ / kg**

Heat added to cycle =

$\Delta Q_1 + \Delta Q_2 = 96.72 + 43.86 = 140.58$ kJ / kg

Work done obtained from turbine:

$\Delta W_1 + \Delta W_2 = 42.26 + 111.07 = 153.33$ kJ / kg

Heat Recovery:

$\Delta Q_3 + \Delta Q_4 + \Delta Q_5 = -28.69 - 53.72 - 54.32 = -136.73$ kJ / kg

Compressor work:

$\Delta W_3 + \Delta W_4 + \Delta W_5 = -54.39 - 53.73 - 41.56 = -149.68$ kJ / kg

Net Work Output:

$\Delta W_1 + \Delta W_2 + \Delta W_3 + \Delta W_4 + \Delta W_5$ **= 3.65 kJ / kg**

CYCLE 2.

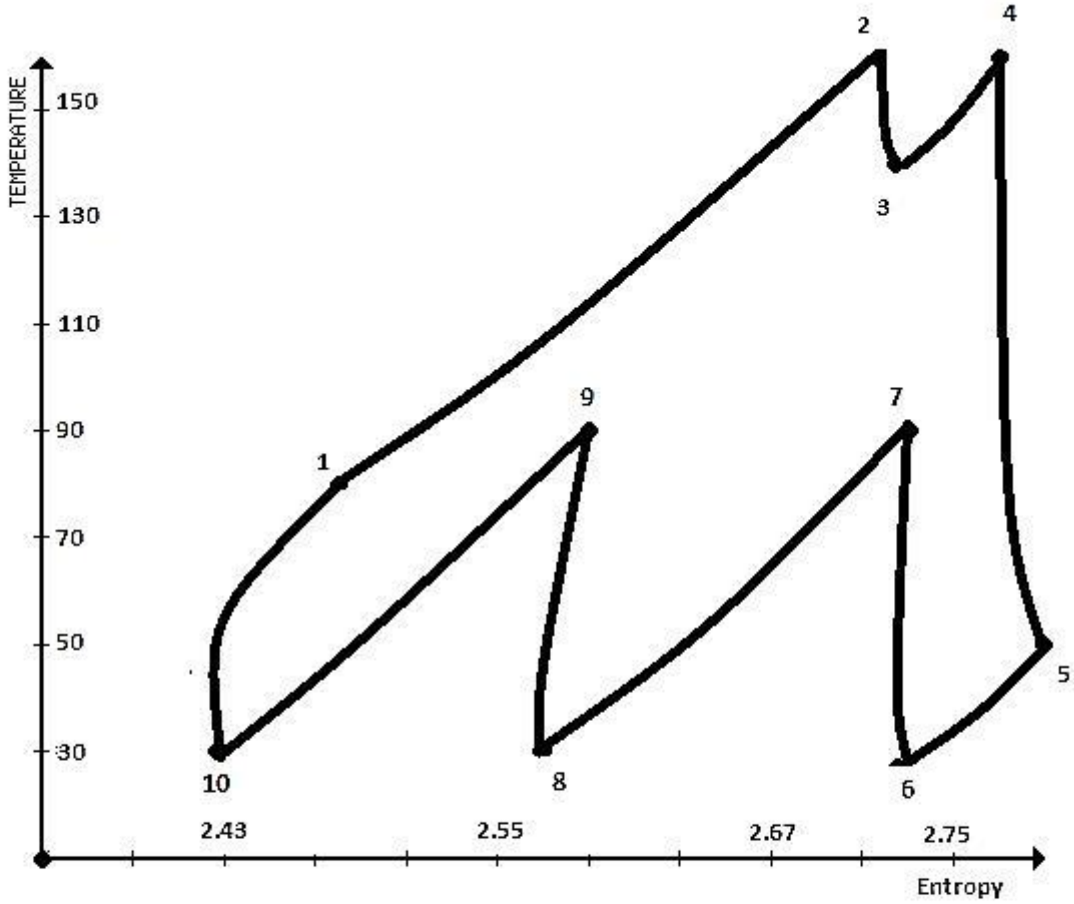

Figure 12: Combined Heat and Power Cycle for CO_2 with primary heat addition at constant pressure and reheating and Intercooling

Table 10: Process Table

Process	Description	Device
1-2, 3-4	Constant Pressure Heat Addition	Evacuated tube collector
2-3, 4-5	Isentropic Expansion	High/ Low Pressure turbine
5-6	Constant Pressure Heat Rejection	Condenser
6-7, 8-9, 9-10	Constant Pressure Heat Recovery	Heat Exchanger (Thermal Water Heater)

Table 11 Properties on Points

	1	2	3	4	5	6	7	8	9	10
state of aggregation :	**Gas**	**Gas**	Gas	Gas	Gas	Gas	Gas	Gas	Gas	Gas
Pressure (P) :	**7**	**7**	5	5	1.25	1.25	3.35	3.35	5.5	5.5
Temperature (T):	**60**	**160**	140	160	50	27	90	30	90	30
Density (ρ) :	**11.4344**	**8.662**	6.475	6.164	2.063	2.227	4.944	5.978	8.148	9.885
Specific Enthalpy (h):	**531.68**	**626.480**	607.980	627.360	516.831	507.319	561.702	507.985	560.340	506.020
Specific Entropy (s) :	**2.4694**	**2.717**	2.725	2.771	2.790	2.726	2.731	2.570	2.590	2.427
Specific isobar heat capacity (C_p) :	**0.91336**	**0.980**	0.962	0.976	0.876	0.854	0.919	0.872	0.927	0.887
Specific isochor heat capacity (C_v):	**0.70092**	**0.780**	0.764	0.779	0.683	0.660	0.722	0.668	0.724	0.673

Assumptions:

- All Processes are Reversible Process.
- Kinetic Energy of the working fluid at inlet and outlet of turbine is same.
- Radiation and mechanical losses are neglected.

Cycle Parameters:

Maximum Temperature (T_{max}) : 160 °C

Pressure at T_{max} (P) : 7 bar

Minimum Temperature (T_{min}): 27 °C

Pressure : 1.25 bar

Reheating:

$P_3 = (P_1 P_5)^{½}$

$= (9 X 1.25)^{½} = 3.35$ bar

But Reheating Pressure is kept at 5bar as average pressure output is 5bar from the Evacuated Tube collector.

Intercooling:

$P_3 = (P_1 P_5)^{½}$

$= (9 X 1.25)^{½} = 3.35$ bar

$P_9 = (P_1 P_7)^{½}$

$= (9 X 3.35)^{½} = 5.49$ bar

Calculations for Enthalpy:

- Process: 1-2 Constant Pressure Heat Addition

  $\Delta Q_1 = h_2 - h_1$ = 626.48 – 531.68 = **94.8 kJ / kg**

- Process: 2-3 Isentropic Expansion

  $\Delta W_1 = h_2 - h_3$ = 626.48 – 607.98 = **18.5 kJ / kg**

- Process: 3-4 Constant Pressure Heat Addition (Reheating)

  $\Delta Q_2 = h_4 - h_3$ = 627.360- 603.22= **19.38 kJ / kg**

- Process: 4-5 Isentropic Expansion

  $\Delta W_2 = h_4 - h_5$ = 627.360- 516.831= **110.53 kJ / kg**

- Process: 5-6 Constant Pressure Heat Rejection

$$\Delta \mathbf{Q}_3 = \mathbf{h}_6 - \mathbf{h}_5 = 507.319 - 516.831 = \mathbf{-9.511\ kJ / kg}$$

- Process: 6-7 Isentropic Compression

$$\Delta \mathbf{W}_3 = \mathbf{h}_6 - \mathbf{h}_7 = 507.319 - 561.7 = \mathbf{-52.39\ kJ / kg}$$

- Process: 7-8 Constant Pressure Heat Rejection (Intercooling)

$$\Delta \mathbf{Q}_4 = \mathbf{h}_8 - \mathbf{h}_7 = 507.98 - 561.70 = \mathbf{-28.69\ kJ / kg}$$

- Process: 8-9 Isentropic Compression

$$\Delta \mathbf{W}_4 = \mathbf{h}_8 - \mathbf{h}_9 = 507.98 - 560.34 = \mathbf{-52.73\ kJ / kg}$$

- Process: 9-10 Constant Pressure Heat Rejection (Intercooling)

$$\Delta \mathbf{Q}_5 = \mathbf{h}_{10} - \mathbf{h}_9 = 506.02 - 560.34 = \mathbf{-54.32\ kJ / kg}$$

- Process: 10-1 Isentropic Compression

$$\Delta \mathbf{W}_5 = \mathbf{h}_{10} - \mathbf{h}_1 = 508.02 - 529.68 = \mathbf{-20.66\ kJ / kg}$$

According to First law of Thermodynamics:

$\sum Q \approx \sum W$ (for a closed cycle)

$\Delta Q_1 + \Delta Q_2 + \Delta Q_3 + \Delta Q_4 + \Delta Q_5 \approx \Delta W_1 + \Delta W_2 + \Delta W_3 + \Delta W_4 + \Delta W_5$

$94.8 + 19.38 - 9.511 - 28.69 - 54.32 \approx 18.5 + 110.529 - 52.39 - 52.73 - 20.66$

Heat added to cycle =

$\Delta Q_1 + \Delta Q_2 = 94.8 + 19.38 = 114.18$ kJ / kg

Work done obtained from turbine:

$\Delta W_1 + \Delta W_2 = 129.029$ kJ / kg

Heat Recovery:

$\Delta Q_3 + \Delta Q_4 + \Delta Q_5 = -102.521$ kJ / kg

Compressor work :

$\Delta W_3 + \Delta W_4 + \Delta W_5 = -125.78$ kJ / kg

Net Heat Delivered:

$\Delta Q_1 + \Delta Q_2 + \Delta Q_3 + \Delta Q_4 + \Delta Q_5 = 21.65$ kJ / kg

Cycle 1 with Primary heating at Constant Volume

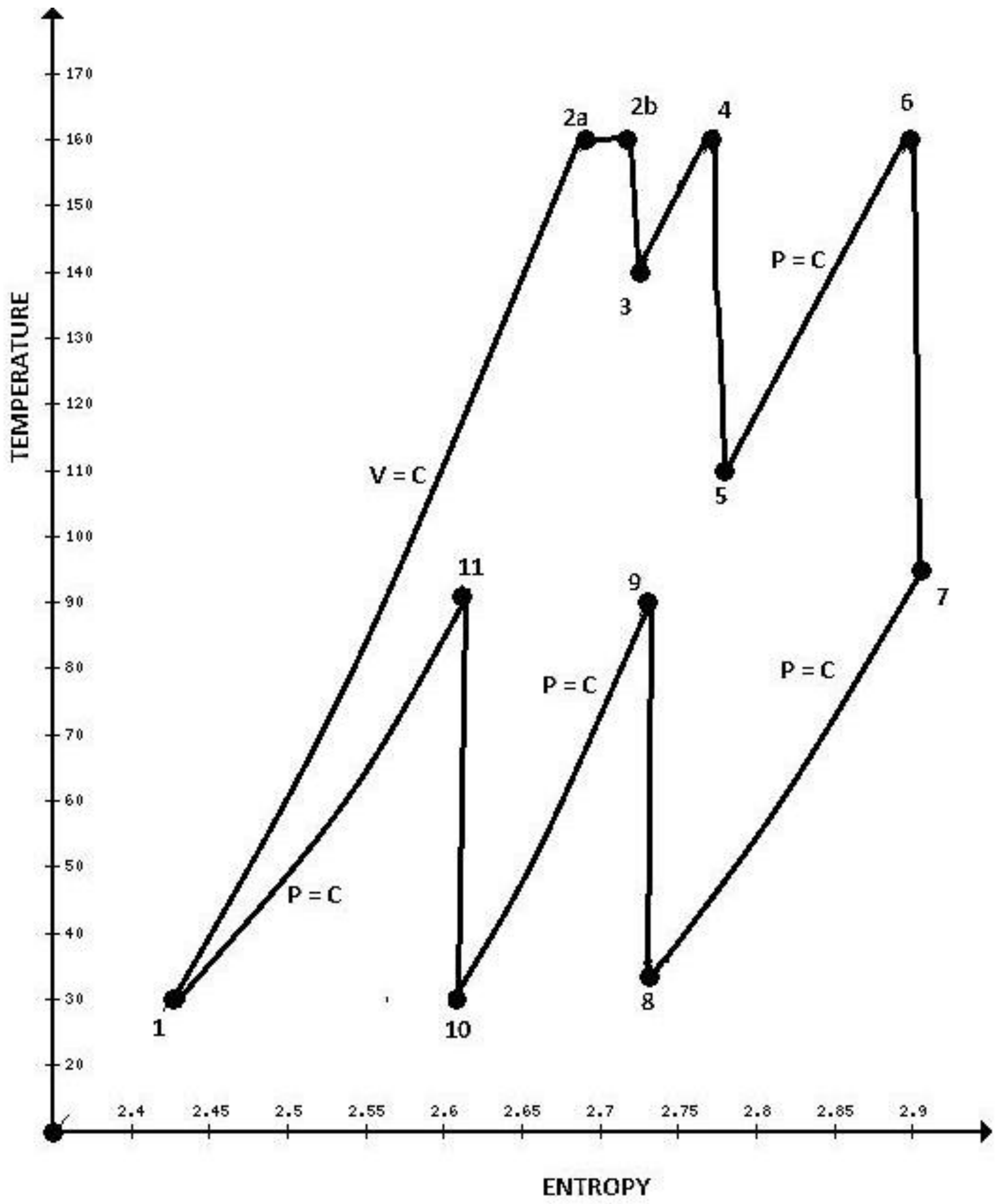

Figure 13: Combined Heat and Power Cycle for CO_2 with primary heat addition at constant Volume and reheating and Intercooling

Table 12 : Process Table (Modified Cycle)

Process	Description	Device
1-2	Constant Volume Heat Addition	Evacuated tube collector
3-4, 5-6	Constant Pressure Heat Addition	Evacuated tube collector
2-3, 4-5, 6-7	Isentropic Expansion	High/ Low Pressure turbine
7-8, 9-10, 11-1	Constant Pressure Heat Rejection	Condenser, Heat Exchanger (Thermal Water Heater)
8-9, 10-11	Isentropic Compression	Compressor

Table 13: Properties on Points (Modified Cycle)

C v Cycle		2-a	2-b	3	4	5	6	7	8	9	10	11	1
state of aggregation :		**gas**	Gas	Gas	Gas	Gas	Gas	Gas	Gas	Gas	Gas	Gas	Gas
Pressure (P) :	bar	7.95	7	5	5	3.35	3.35	1.25	1.25	3.35	3.35	5.5	5.5
Temperature (T):	Celsius	160	160	140	160	110	160	95	27	90	30	90	30
Density (ρ) :	kg / m³	9.849	8.662	6.475	6.164	4.686	4.127	1.807	2.227	4.944	5.978	8.148	9.885
Specific Enthalpy (h):	kJ / kg	626.062	626.480	607.980	627.360	580.361	628.070	567.595	507.319	561.702	507.985	560.340	506.020
Specific Entropy (s) :	kJ / kg K	2.691	2.717	2.725	2.771	2.780	2.898	2.906	2.726	2.731	2.570	2.590	2.427
Specific isobar heat capacity (C_p) :	kJ / kg K	0.982	0.980	0.962	0.976	0.935	0.972	0.916	0.854	0.919	0.872	0.927	0.887
Specific isochor heat capacity (C_v):	kJ / kg K	0.781	0.780	0.764	0.779	0.738	0.778	0.724	0.660	0.722	0.668	0.724	0.673

Assumptions:

- All Processes are Reversible Process.
- Kinetic Energy of the working fluid at inlet and outlet of turbine is same.
- Radiation and mechanical losses are neglected.

Cycle Parameters:

Maximum Temperature (T_{max}) : 180 °C

Pressure at T_{max} (P): 9 bar

Minimum Temperature (T_{min}): 27 °C

Pressure: 1.25 bar

Reheating:

$P_6 = (P_2\ P_7)^{½}$

$= (7 \text{ X } 1.25)^{½} = 3$ bar

$P_4 = (P_2\ P_6)^{½}$

$= (7.95 \text{ X } 3.35)^{½} = 5.00$ bar

Intercooling:

$P_6 = (P_2\ P_7)^{½}$

$= (7.95 \text{ X } 1.25)^{½} = 3.15$ bar

$P_4 = (P_2\ P_6)^{½}$

$= (7.95 \text{ X } 3.35)^{½} = 5.006$ bar

Calculations for Enthalpy:

- Process: 1-2 Enthalpy change for Constant Volume Heat Addition

  $\Delta h_1 = h_2 - h_1$ = 626.062– 506.020= **120.0 kJ / kg**

- Process: 2-3 Isentropic Expansion

  $\Delta W_1 = h_2 - h_3$ = 626.480- 607.980= **18.5 kJ / kg**

- Process: 3-4 Constant Pressure Heat Addition (Reheating)

  $\Delta Q_2 = h_4 - h_3$ = 627.360- 607.980= **19.38 kJ / kg**

- Process: 4-5 Isentropic Expansion

  $\Delta W_2 = h_4 - h_5$ = 627.360- 580.361= **46.999 kJ / kg**

- Process: 5-6 Constant Pressure Heat Addition (Reheating)

  $\Delta Q_3 = h_6 - h_5$ = 628.070– 580.361 = **47.709 kJ / kg**

- Process: 6-7 Isentropic Expansion

$$\Delta \mathbf{W}_3 = \mathbf{h}_6 - \mathbf{h}_7 = 628.070 - 567.595 = \mathbf{60.475\ kJ / kg}$$

- Process: 7-8 Constant Pressure Heat Rejection (Intercooling)

$$\Delta \mathbf{Q}_4 = \mathbf{h}_8 - \mathbf{h}_7 = 507.319 - 567.595 = \mathbf{-\ 60.27\ kJ / kg}$$

- Process: 8-9 Isentropic Compression

$$\Delta \mathbf{W}_4 = \mathbf{h}_8 - \mathbf{h}_9 = 507.319 - 561.702 = \mathbf{-54.383\ kJ / kg}$$

- Process: 9-10 Constant Pressure Heat Rejection (Intercooling)

$$\Delta \mathbf{Q}_5 = \mathbf{h}_{10} - \mathbf{h}_9 = 507.985 - 561.702 = \mathbf{-\ 53.717\ kJ / kg}$$

- Process: 10-11 Isentropic Compression

$$\Delta \mathbf{W}_5 = \mathbf{h}_{10} - \mathbf{h}_{11} = 507.985 - 560.340 = \mathbf{-52.355\ kJ / kg}$$

- Process: 11-1 Constant Pressure Heat Rejection (Intercooling)

$$\Delta \mathbf{Q}_6 = \mathbf{h}_1 - \mathbf{h}_{11} = 506.020 - 560.340 = \mathbf{-\ 54.32\ kJ / kg}$$

According to First law of Thermodynamics:

$\sum Q \approx \sum W$ (for a closed cycle)

$\Delta \mathbf{h}_1 + \Delta Q_2 + \Delta Q_3 + \Delta Q_4 + \Delta Q_5 + \Delta Q_6 = 18.782$

$\Delta W_1 + \Delta W_2 + \Delta W_3 + \Delta W_4 + \Delta W_5 = 19.236$

Heat added to cycle =

$\Delta \mathbf{h}_1 + \Delta Q_2 + \Delta Q_3 = 120.0 + 19.38 + 47.71 = 187.09$ kJ / kg

Work done obtained from turbine:

$\Delta W_1 + \Delta W_2 + \Delta W_3 = 18.5 + 46.99 + 60.475 = 125.96$ kJ / kg

Heat Recovery:

$\Delta Q_3 + \Delta Q_4 + \Delta Q_5 = -\ 60.27 - 53.717 - 54.32 = -168.3$ kJ / kg

Compressor work :

$\Delta W_3 + \Delta W_4 = -\ 54.39 - 52.355 = -\ 106.745$ kJ / kg

Net Work Output

$\Delta W_1 + \Delta W_2 + \Delta W_3 + \Delta W_4 + \Delta W_5 = \mathbf{19.236\ kJ / kg}$

Cycle 2 with Primary heating at Constant Volume

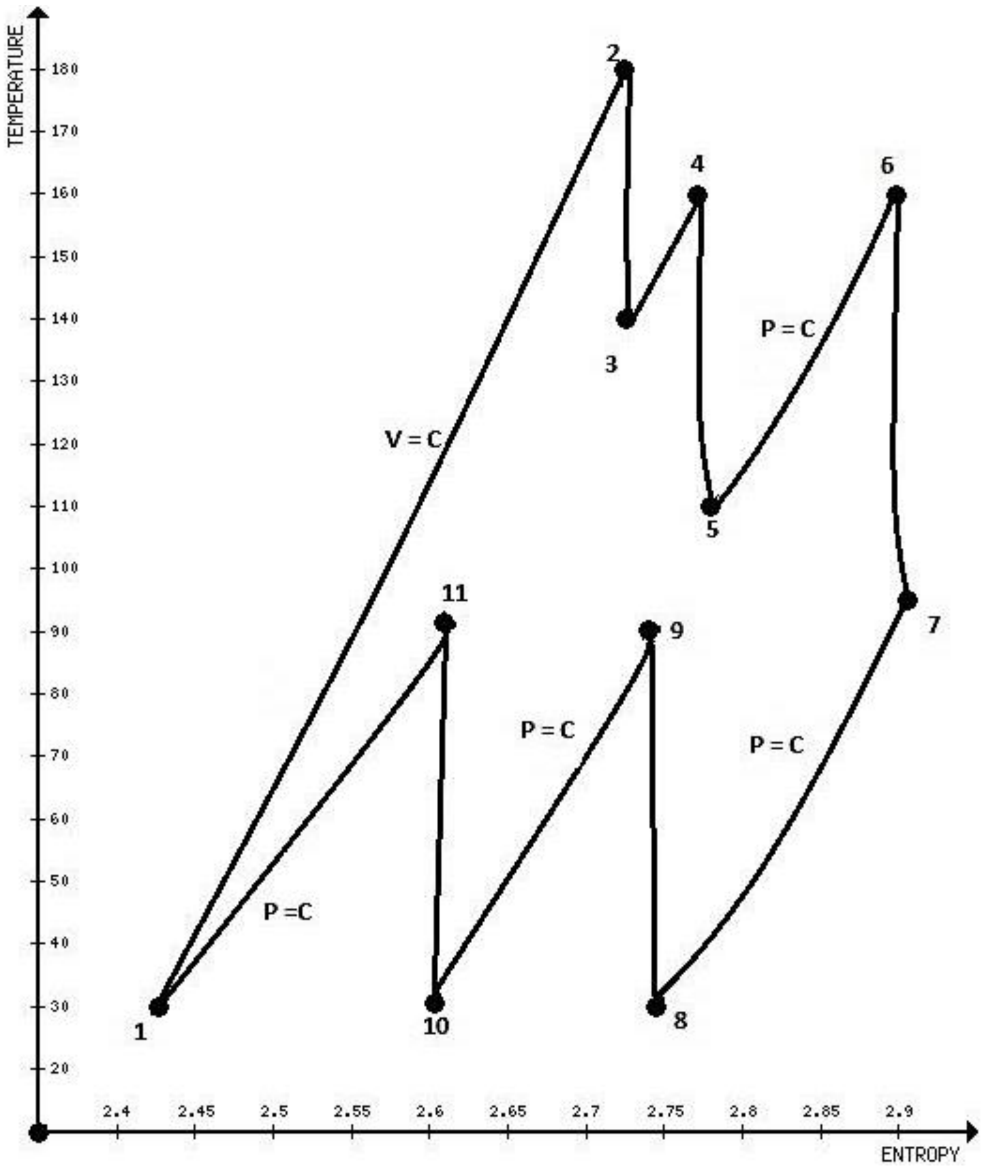

Figure 14: Combined Heat and Power Cycle for CO_2 with primary heat addition at constant Volume and reheating and Intercooling

Table 14 : Process Table (Modified Cycle)

Process	Description	Device
1-2	Constant Volume Heat Addition	Evacuated tube collector
3-4, 5-6	Constant Pressure Heat Addition	Evacuated tube collector
2-3, 4-5, 6-7	Isentropic Expansion	High/ Low Pressure turbine
7-8, 9-10, 11-1	Constant Pressure Heat Rejection	Condenser, Heat Exchanger (Thermal Water Heater)
8-9, 10-11	Isentropic Compression	Compressor

Table 15: Properties on Points (Modified Cycle)

C v Cycle		2	3	4	5	6	7	8	9	10	11	1
state of aggregation :		Gas	Gas	Gas	Gas	Gas	Gas	Gas	Gas	Gas	Gas	Gas
Pressure (P) :	bar	8.35	5	5	3.35	3.35	1.25	1.25	3.35	3.35	5.5	5.5
Temperature (T):	Celsius	180	140	160	110	160	95	27	90	30	90	30
Density (ρ) :	kg / m^3	9.884	6.475	6.164	4.686	4.127	1.807	2.227	4.944	5.978	8.148	9.885
Specific Enthalpy (h):	kJ / kg	645.740	607.980	627.360	580.361	628.070	567.595	507.319	561.702	507.985	560.340	506.020
Specific Entropy (s) :	kJ / kg K	2.725	2.725	2.771	2.780	2.898	2.906	2.726	2.731	2.570	2.590	2.427
Specific isobar heat capacity (C_p) :	kJ / kg K	0.996	0.962	0.976	0.935	0.972	0.916	0.854	0.919	0.872	0.927	0.887
Specific isochor heat capacity (C_v):	kJ / kg K	0.795	0.764	0.779	0.738	0.778	0.724	0.660	0.722	0.668	0.724	0.673

Assumptions:

- All Processes are Reversible Process.
- Kinetic Energy of the working fluid at inlet and outlet of turbine is same.
- Radiation and mechanical losses are neglected.

Cycle Parameters:

Maximum Temperature (T_{max}): 180 °C

Pressure at T_{max} (P): 8.35 bar

Minimum Temperature (T_{min}): 27 °C

Pressure: 1.25 bar

Reheating:

$P_5 = (P_2\ P_7)^{½}$

$= (8.35 \text{ X } 1.25)^{½} = 3.23$ bar

$P_4 = (P_2\ P_6)^{½}$

$= (8.35 \text{ X } 3.23)^{½} = 5.19$ bar

Intercooling:

$P_9 = (P_2\ P_7)^{½}$

$= (8.35 \text{ X } 1.25)^{½} = 3.23$ bar

$P_{11} = (P_2\ P_9)^{½}$

$= (8.35 \text{ X } 3.23)^{½} = 5.19$ bar

Calculations for Enthalpy:

- Process: 1-2 Enthalpy change for Constant Volume Heat Addition

  $\Delta h_1 = h_2 - h_1$ = 645.74 – 506.020= **139.72 kJ / kg**

- Process: 2-3 Isentropic Expansion

  $\Delta W_1 = h_2 - h_3$ = 645.74- 607.980= **37.76 kJ / kg**

- Process: 3-4 Constant Pressure Heat Addition (Reheating)

  $\Delta Q_2 = h_4 - h_3$ = 627.360- 607.980= **19.38 kJ / kg**

- Process: 4-5 Isentropic Expansion

  $\Delta W_2 = h_4 - h_5$ = 627.360- 580.361= **46.999 kJ / kg**

- Process: 5-6 Constant Pressure Heat Addition (Reheating)

  $\Delta Q_3 = h_6 - h_5$ = 628.070– 580.361 = **47.709 kJ / kg**

- Process: 6-7 Isentropic Expansion

$$\Delta \mathbf{W}_3 = \mathbf{h}_6 - \mathbf{h}_7 = 628.070 - 567.595 = \mathbf{60.475\ kJ / kg}$$

- Process: 7-8 Constant Pressure Heat Rejection (Intercooling)

$$\Delta \mathbf{Q}_4 = \mathbf{h}_8 - \mathbf{h}_7 = 507.319 - 567.595 = \mathbf{-\ 60.27\ kJ / kg}$$

- Process: 8-9 Isentropic Compression

$$\Delta \mathbf{W}_4 = \mathbf{h}_8 - \mathbf{h}_9 = 507.319 - 561.702 = \mathbf{-54.383\ kJ / kg}$$

- Process: 9-10 Constant Pressure Heat Rejection (Intercooling)

$$\Delta \mathbf{Q}_5 = \mathbf{h}_{10} - \mathbf{h}_9 = 507.985 - 561.702 = \mathbf{-\ 53.717\ kJ / kg}$$

- Process: 10-11 Isentropic Compression

$$\Delta \mathbf{W}_5 = \mathbf{h}_{10} - \mathbf{h}_{11} = 507.985 - 560.340 = \mathbf{-52.355\ kJ / kg}$$

- Process: 11-1 Constant Pressure Heat Rejection (Intercooling)

$$\Delta \mathbf{Q}_6 = \mathbf{h}_1 - \mathbf{h}_{11} = 506.020 - 560.340 = \mathbf{-\ 54.32\ kJ / kg}$$

According to First law of Thermodynamics:

$\sum Q \approx \sum W$ (for a closed cycle)

$\mathbf{\Delta h_1 + \Delta Q_2 + \Delta Q_3 + \Delta Q_4 + \Delta Q_5 + \Delta Q_6 = 38.502}$

$\mathbf{\Delta W_1 + \Delta W_2 + \Delta W_3 + \Delta W_4 + \Delta W_5 = 38.496}$

Heat added to cycle =

$\mathbf{\Delta h}_1 + \Delta Q_2 + \Delta Q_3 = 139.72 + 19.38 + 47.71 = 206.81$ kJ / kg

Work done obtained from turbine:

$\Delta W_1 + \Delta W_2 + \Delta W_3 = 37.76 + 46.99 + 60.475 = 145.234$ kJ / kg

Heat Recovery:

$\Delta Q_3 + \Delta Q_4 + \Delta Q_5 = -\ 60.27 - 53.717 - 54.32 = -168.3$ kJ / kg

Compressor work :

$\Delta W_3 + \Delta W_4 = -\ 54.39 - 52.355 = -\ 106.745$ kJ / kg

Net Work Output

$\Delta W_1 + \Delta W_2 + \Delta W_3 + \Delta W_4 + \Delta W_5$ **= 38.496 kJ / kg**

Chapter 5: RESULTS AND DISCUSSION

Observations:

We have already seen various thermodynamic cycles in previous section and there individual performance. In this section all the previous cycles are compared with each other for analysis. Below is the table of the few Quantities of net energy exchanged within temperature and pressure limits and further some discussion regarding the expected performance of the proposed system over various parameters along with bar graph.

Table 16: Net Energy Exchanged Within Temperature and Pressure Limits

		1	2
	Temperature limits (°C)	**180 / 27**	**160 / 27**
	Pressure limits (bar)	**9 / 1.25**	**7 / 1.25**
Heat added (kJ / kg)	Constant pressure	114.18	140.58
	Enthalpy rise for Constant volume	187.09	206.81
Turbine work (kJ / kg)	Constant pressure	119.029	153.33
	Constant volume	125.96	145.234
Heat recovery (kJ / kg)	Constant pressure	102.521	136.73
	Constant volume	168.3	168.3
Compressor work (kJ / kg)	Constant pressure	133.78	169.68
	Constant volume	106.745	106.745
Net work (kJ / kg)	Constant pressure	3.65	4.78
	Constant volume	19.236	38.496

Enthalpy Gain: It can be easily deduced that within the same temperature and pressure limits heat absorbed by the cycle with primary het addition at constant volume is much higher as compared to heat addition at constant pressure.

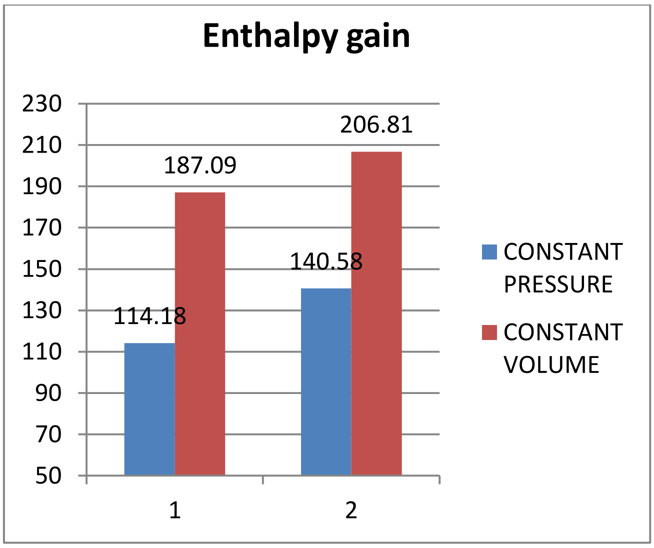

Figure 15: Graph showing amount of heat added during various cycles

Turbine Work: It is projected that when turbine work output is compared within the same temperature and pressure limits there was no significant difference with primary het addition at constant volume as compared to heat addition at constant pressure is observed.

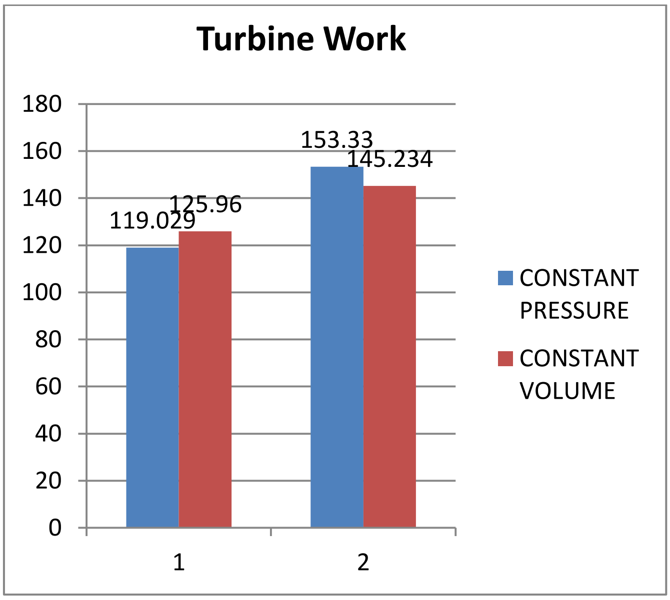

Figure 16: Graph showing Turbine Output during various cycles

Heat Recovery: It is projected that within the same temperature and pressure limits heat recovered by the cycle with primary het addition at constant volume is much higher as compared to heat addition at constant pressure. This can be attributed to higher heat absorption at constant volume and nearly equal turbine output.

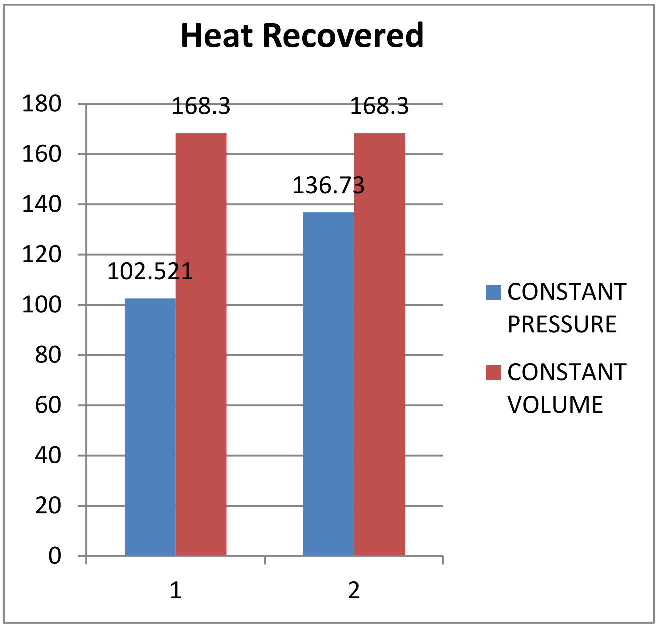

Figure 17: Graph showing Turbine Output during various cycles

Compressor Work: within the same temperature and pressure limits much higher compressor work is expected to be given in for the cycle with primary het addition at constant pressure as compared to heat addition at constant volume.

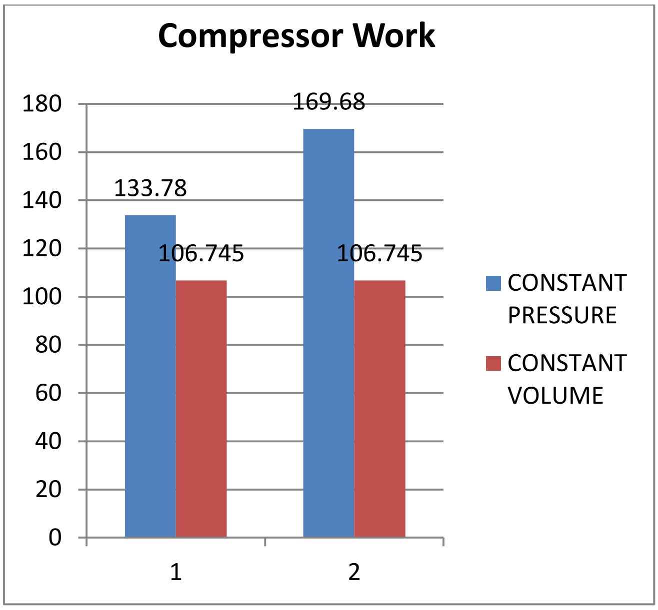

Figure 18: Graph showing Compressor work input during various cycles

Net Work Output: It is projected that within the same temperature and pressure limits Net Work obtained by the cycle with primary het addition at constant volume is much higher as compared to heat addition at constant pressure. This can be attributed to higher compressor work and low heat absorption in cycle with primary het addition at constant pressure.

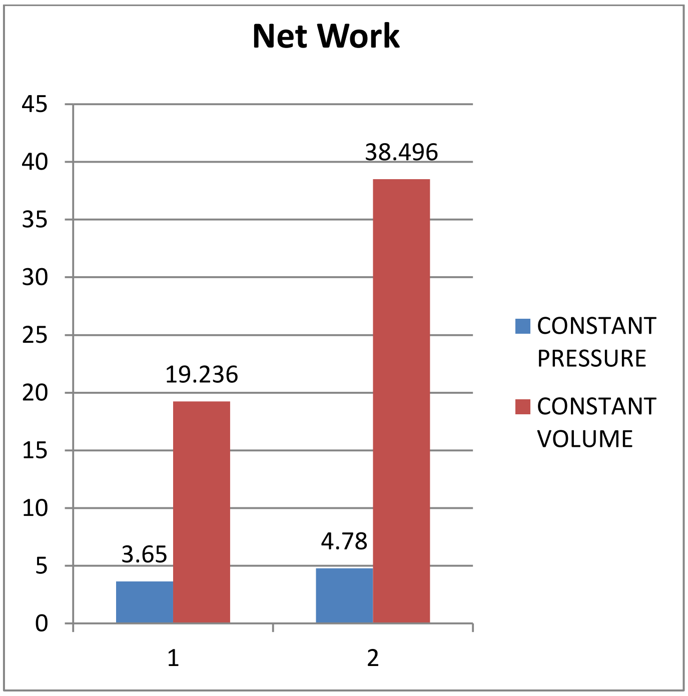

Figure 19: Graph showing Net Work output during various cycles

Chapter 7: CONCLUSIONS AND FUTURE SCOPE

Project Conclusions:

The purpose of this thesis was to do an exhaustive literature review, propose a Combined Heat and Power Cycle with CO_2 as working fluid and using non concentrator type solar thermal device i.e. Evacuated Tube Collector to absorb energy.

The concluding points are as follows:

1. Present Study validates the point of using CO_2 as working fluid for solar thermal applications.
2. **Energy Turnover** of the system: With Primary heating at Constant Volume net Energy handled by the system increases and there by largely increasing the prospects of higher efficiency.
3. A modified real cycle is also proposed with graphical validation.
4. Higher work output is obtained mainly due to reduced compressor work.
5. Lower Compressor work was required. This can be attributed to less number of Compressors used.

It can be concluded that CO_2 has a good potential in solar thermal application and it can be the answer for the heat transfer limitation of conventional heat transfer fluids.

Future Scope:

- This study can also be extended for concentrating type solar collectors and higher temperature and pressures can be achieved for higher work output as CO_2 is thermally stable up to 1200°C.
- Area of application of alternative and natural working Fluids in Solar Energy is still wide open research field.
- There is a necessity of further theoretical analysis according to Second law of thermodynamics as well as experimental work, investigations to enhance the efficiency of solar collector and to obtain firm and authenticate results.

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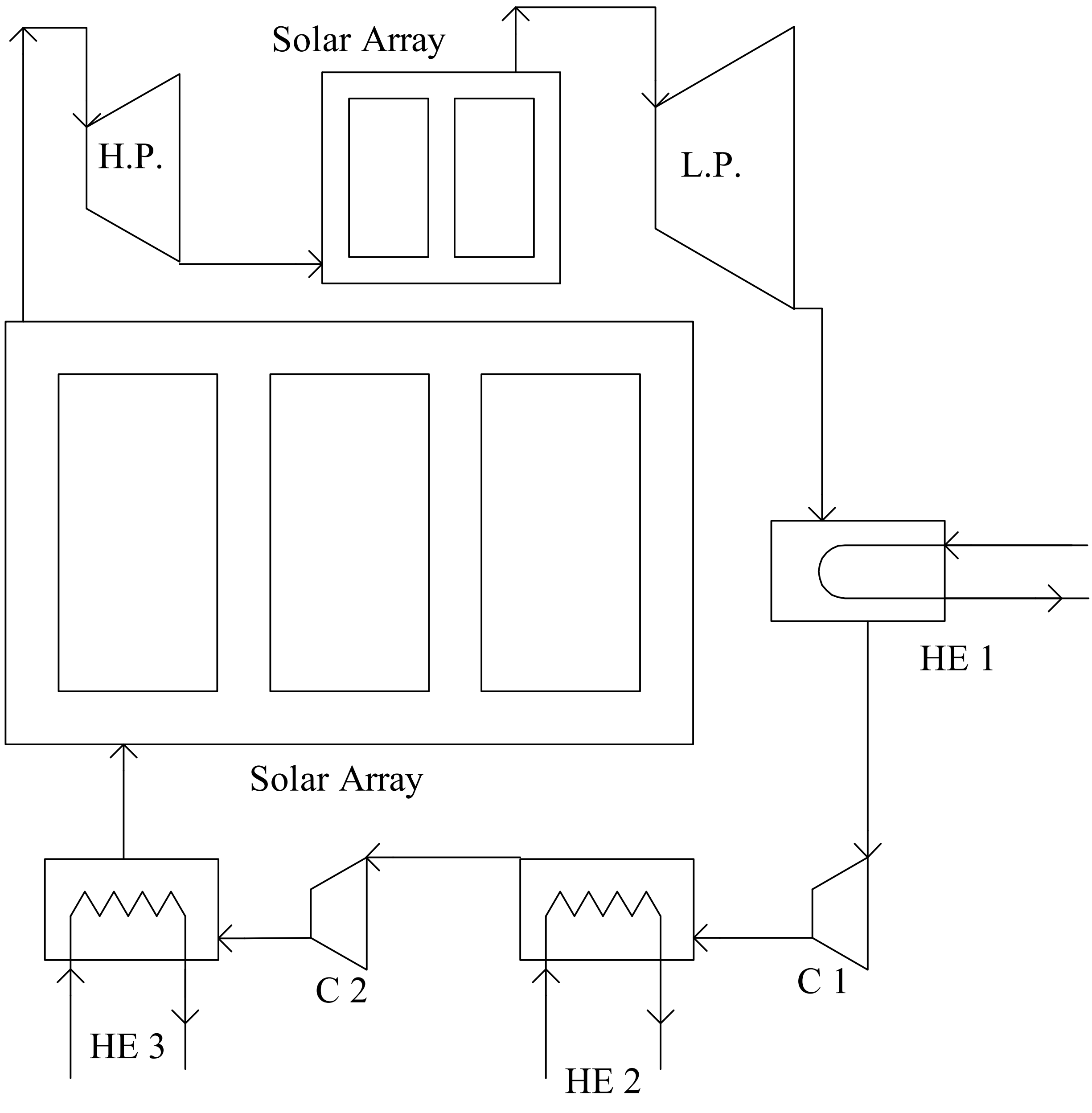
Appendix

FLOW CHART

Proposed flow chart for thermodynamic cycle with Primary Heating at Constant Volume



HE - Heat Exchanger

LP - Low Pressure Turbine

C - Compressor

HP- High Pressure Turbine