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ENERGETIC AND EXERGETIC PERFORMANCE EVALUATION OF A SOLAR DISH BASED DUAL RECEIVER COMBINED CYCLE

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ABSTRACT

Solar Thermal Power Plants with concentrating technologies are important for providing a major share of the clean and renewable energy needed in the future. Dish based system is a promising option for power generation since it has high concentration ratio and it does not require a large space. This paper presents a proposed solar dish based combined cycle: air based Brayton cycle as the topping cycle with a bottoming steam cycle. This paper also proposes a dual receiver, which can be utilized for heating the air for topping Brayton cycle as well as to produce the superheated steam required for the bottoming cycle. Program coding on C language has been developed and the detailed thermodynamic parametric analysis of the combined cycle has been carried out energetically as well as exergetically for the varying pressure ratio of the compressor. Component wise exergy analysis has been done to identify the component where the exergy loss is maximum. The thermodynamic analysis shows that for the topping gas turbine cycle only, both the work obtained and the thermal efficiency maximizes at very high pressure ratio. But the incorporation of a simple downstream HRSG drags the point to lower pressure ratio range. It has also been found that at pressure ratio 9, where the overall thermal efficiency is maximum, solar receiver shares the maximum exergy destruction among all the other components. Exergetic performance of the solar receiver, dish, HRSG and stack has also been carried out in the present study.

Keywords: Solar Energy; Combined Cycle; Dish System; Dual Receiver; Second Law.

1. INTRODUCTION

Demand for electricity is increasing day by day due to the higher standards of living, increasing number of consumers, increasing demand for desalination, cooling etc. Most of this demand is met by the burning of the fossil fuels like coal, petroleum oil and natural gas. But it is a well-known fact that these non-renewable energy sources are limited and will be depleted within few decades or centuries depending on the reserve of the corresponding fuel. So the fossil fuel supply gap (FFSG), which is nothing but the difference between the energy demand and supply, will go on increasing but the demand must meet the supply. In order to minimize the gap between the supply and demand, either the demand should drop strongly which is next to impossible or the alternate renewable energy option for power generation should be opted. Among all the renewable energy sources, solar energy can itself meet the total energy demand since it is a very abundant and inexhaustible source.

Solar energy is attracting the attention since it is clean, carbon neutral and climate-friendly energy resource to mankind, relatively well-spread over the globe.

The costs of solar energy have been falling rapidly and are entering new areas of competitiveness. Solar thermal electricity (STE) allows shifting the production of solar electricity to peak or mid-peak hours in the evening, or spreading it to base-load hours round the clock, through the use of thermal storage. STE today is based on concentrating solar power (CSP) technologies, which can be used mostly all around the globe. Producing electricity from the energy in the sun's rays is a straightforward process: incident solar radiation is concentrated and collected by a range of Concentrating Solar Power (CSP) technologies (like parabolic trough, power tower, parabolic dish etc.) to provide medium to high temperature heat. The heat energy gained by the fluid can be utilized to run a Brayton or a Rankine cycle or a Stirling engine.

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Depending on the configuration of each square metre of CSP concentrator surface, for example, is enough to avoid annual emissions of 200 to 300 kilograms of carbon dioxide, which is a by-product of fossil-fuelled power plants. Most of the CSP solar field materials can be recycled and used again for further plants **[1]**.

A lot of works have been carried out on Solar dish/engine systems, which convert solar energy to mechanical energy and then electrical energy. In the late 1970s and early 1980s modern solar dish/engine technology was developed by Advanco Corporation, United Stirling AB, McDonnell Douglas Aerospace Corporation (MDA), the US Department of Energy (DOE), and NASA's Jet Propulsion Laboratory **[2]**. The Advanco Vanguard system, 25-kWe nominal output module, using the United Stirling Power Conversion Unit (PUC), obtained a solar-to electric conversion efficiency of 29.4% **[3, 4]**. The project report by Diver et al. **[2]** cited the development and validation of a 9-kWe dish/Stirling solar power system for remote power markets. Davenport et al. **[2]** reported the operational results and experiences from a prototype of the SunDish system at the Salt River Project (SRP).

The motivation behind this work i.e. to consider a dish-Brayton cycle as the topping cycle is that many limited papers have appeared on the solar dish based Brayton cycle. The report by Brayton Energy, LLC **[5]** cited the system configuration of a solar dish based Brayton cycle for power generation with compressed air energy storage and also demonstrated the excellent part load efficiency of Brayton engine. Few projects have tested gas turbines on solar towers with heliostat mirror fields. There have only been two truly Dish-Brayton demonstrations. The first, in 1984, was led by Sanders Associates, using a micro turbine, a Lajet 460 dish, and a Sanders receiver. On August 30, 2011, Brayton and SST successfully tested the world's second dish-Brayton. This system employed the SST 320 Sq. meter dish, with a purpose-built gas turbine and receiver designed and built by Brayton Energy. The Dish-Brayton is believed by many engineers to offer substantially better reliability and a long (60,000 hour) maintenance free life as compared to the Dish-Stirling **[6]**.

Nowadays, combined cycle power plants are becoming more and more attractive option for power generation because of its outstanding plant efficiency and its environmental friendliness to match the both clients' and governmental requirements. In the combined cycle power plants, the heat remaining in the working fluid coming out from the gas turbine is utilized to run a steam turbine plant. The performance study of such systems is not complete without the exergy analysis since the performance study of the system based on the second law of thermodynamics overcomes the limit of an energy-based analysis. It assesses the magnitude of exergy destruction, identifies the location, magnitude and source of thermodynamic inefficiencies in the thermal system.

In the present paper, dish based combined cycle has been proposed, using Directly Irradiated Annular Pressurised Receiver (DIAPR) with Brayton cycle as the topping cycle with air as the working fluid and in the bottoming, a Rankine cycle has been considered. The detailed thermodynamic analysis of the combined cycle has been carried out energetically as well as exergetically. Component wise exergy analysis has been done to identify the component where exergy loss is maximum.

2. PROPOSED MODEL OF DISH BASED COMBINED CYCLE POWER PLANT

Figure 1 depicts the schematic of a dish based solar powered gas turbine combined cycle considered in the present study. Since this paper only considers the energy and exergy performance of the system at steady state, the thermal storage subsystem has not been discussed and not shown in **Fig. 1**. Without considering the thermal storage subsystem, the dish based combined cycle system can be considered as consisting of the subsystems like dish collector subsystem, receiver subsystem, gas turbine subsystem, steam generation subsystem (**HRSG**), and the steam turbine subsystem. The sun's rays fall on a dish and are reflected into the receiver, located at the focal point of the dish. The concentrated rays onto the receiver result in high temperature of the receiver. Ambient air (298 K and 1.01325 bar pressure) at point **1** enters the compressor (**COMP**) and leaves at point **2**. Compressed air then enters the solar receiver (**R**) and the receiver raises the air temperature to a final temperature of 1000^{O}C. Then heated air at 1000^{O} C enters into the gas turbine (**GT**) at point **3** and expands in the turbine. This exhaust from GT is passed through a heat recovery steam generator (**HRSG**) to produce saturated steam. In the HRSG, air is first used to change the phase of the saturated water at 65 bar and 553.85 K (which is the saturation temperature of water at 65 bar) to saturated steam at Evaporator (EVAP) i.e. from state point **11** to state point **12** and finally, the heat energy remained in the exhaust air from the gas turbine is utilized to heat the feed water in the economiser (**ECO**) from point **10** to **11**. Then the air after passing through the economiser is discharged into the atmosphere. Saturated steam from evaporator i.e. from state point **12** is passed to the receiver again to convert it into superheated steam at 733K by the concentrated solar energy i.e. from state point **12** to point **7**. The superheated steam at point **7** (65 bar pressure and 733K) enters into the steam turbine (**ST**), and after doing work in the steam turbine, steam is exhausted in the condenser (**COND**) at point **8** at 0.07 bar pressure. After being condensed in the condenser, water is pumped from point **9** to point **10** i.e. at the boiler pressure by a feed pump (**P**). The system as well as the coding in C language is developed in such a way that the temperature difference between the evaporator exit temperature of air and the saturation temperature of water (i.e. $T_{11}=T_{12}$) does not fall below 15^{O}C for better heat transfer.

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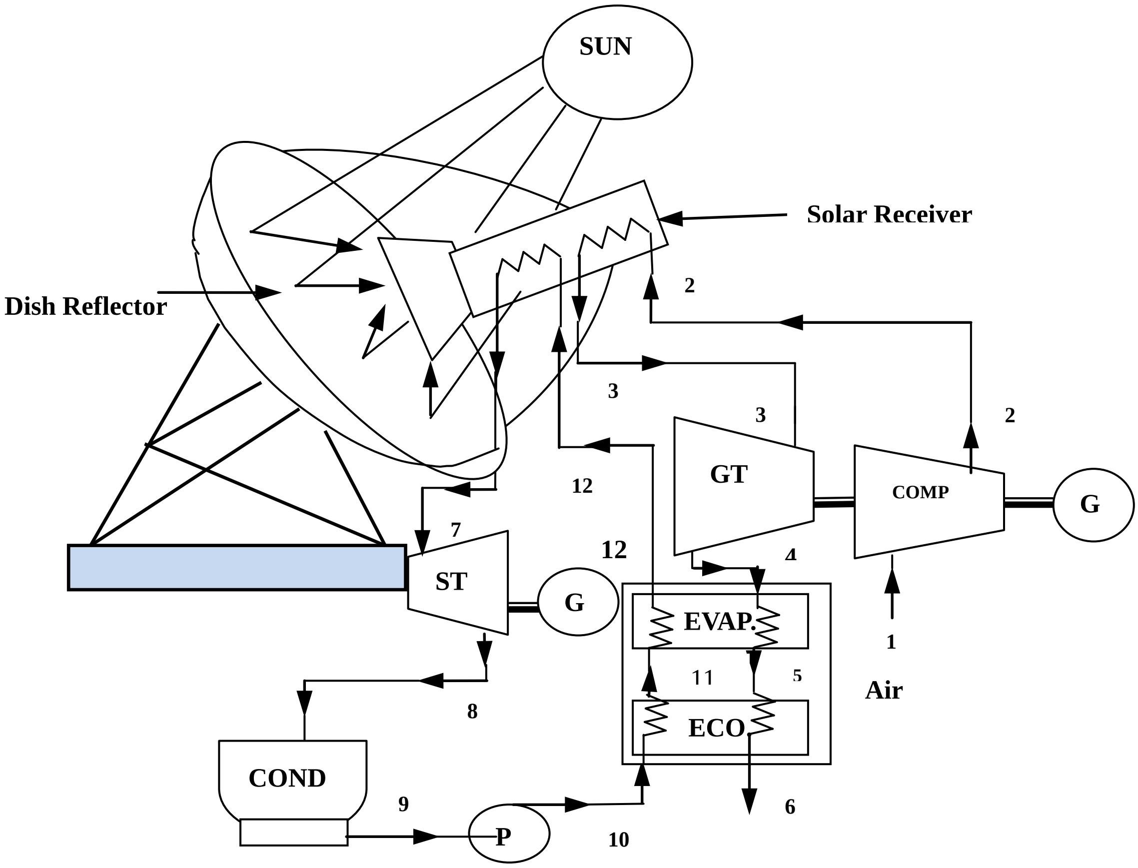

Figure1: Schematic of Dish Based Solar Powered Combined Cycle Plant

3. RECEIVER MODEL

Various designs are available for the solar receiver used in solar power tower plants, but very limied designs of the receiver are available, which can be used in dish based Brayton cycle. In the present study, the authors have proposed modified Directly Irradiated Annular Pressurized Receiver (DIAPR), which can be used for dish based Brayton cycle as reported by Kribus et al. **[7]**.

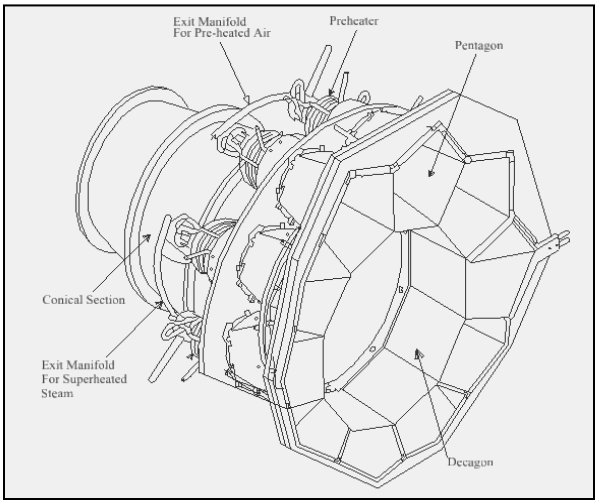

Figure 2: Dual Receiver Construction

This receiver was successfully tested at Weizmann Institute, and the air exit temperatures in this test series reached up to 1000^{O}C. It is capable of supplying hot air at a pressure of 10–30 bar and air exit temperature of up to 1300°C **[7]** is possible to achieve with this type of receiver. An array of secondary concentrators accepts the incident radiation falling on the dish and channels parts of the radiation into separate receivers as shown in **Fig. 2**.

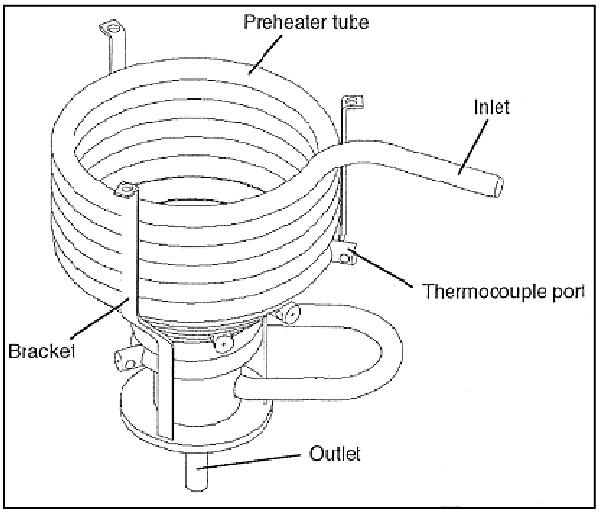

Figure 3: Cavity tubular receiver

The receiver consists of two parts: low temperature receiver used to pre-heat the air and to produce superheated steam and the high temperature receiver (DIAPR) used to increase the temperature of air. The low temperature stage receivers are designed as cavity tubular receivers. The cavity walls are composed of the absorber tube, which spirals around the cavity, as shown in **Fig. 3**. This type of arrangement can accommodate total ten numbers of cavity tubular receivers **[8]**, out of which four numbers will be utilized for preheating the air and the remaining six numbers have been proposed to be utilized for producing superheated steam required in the bottoming cycle. The preheated air will then go to the next high-temperature receiver stage and the superheated steam will then be passed to the steam turbine from the exit manifold.

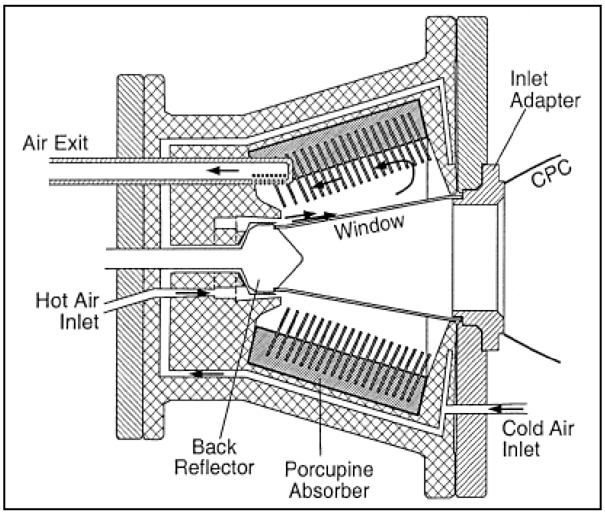

Figure 4: High Temperature Receiver

The high-temperature receiver stage, which accepts radiation from the central, high-flux region of the aperture plane, is the Directly Irradiated Annular Pressurized Receiver (DIAPR). The DIAPR contains a Porcupine volumetric absorber, which absorbs the concentrated sunlight and transfers its energy as heat to the working fluid i.e. air. The DIAPR cavity is closed by the Frustum-Like High-Pressure (FLHiP) window made of fused silica. The window separates the receiver cavity from the ambient air and allows operation at high pressure. An additional inlet for the preheated air flow from the low-temperature stage has been included. The cold air introduced through cold air inlet path, bypassing the preheaters serves for cooling of the window. Preheated air stream is discharged in an annular region surrounding the 'cold' stream, as shown in **Fig. 4 [8]**.

4. THERMODYNAMIC MODEL OF THE PROPOSED COMBINED CYCLE

The combined cycle has been analysed based on the following energy and exergy tools and the following assumptions have been considered in the analysis:

a) The system runs at steady state
b) Pressure drop in the solar receiver has been considered as 0.12 bar **[9]**

4.1 Gas Turbine Cycle

Since the maximum thermal power output obtained from the DIAPR is 60 kW as reported by Kribus et al. **[8]**, the thermal energy utilised to heat the air has been considered to be 50 kW and the rest is utilised to produce the superheated steam. The mass flow rate of air (m_a) is calculated from the following relationship

$$m_a \int_{T_2}^{T_3} C_{pa} dT = 50 \qquad (1)$$

Temperature of air after compression considering the isentropic efficiency of compression (=0.88) can be found out from the following relationship

$$T_2 = T_1 + \frac{1}{\eta_c}(T_{2'} - T_1) \qquad (2)$$

Compression work is given by

$$W_{comp} = m_a \int_{T_1}^{T_2} C_{pa} dT \qquad (3)$$

Gas turbine work is obtained from the following equation

$$W_{GT} = m_a \int_{T_4}^{T_3} C_{pa} dT \qquad (4)$$

Net work obtained from GT cycle

$$(W_{net})_{GT} = (W_{GT} - W_{comp})\eta_{mech}\eta_G \qquad (5)$$

Incident solar insolation required for heating air to the required turbine inlet temperature (**TIT**) is given by

$$Q_1 = \frac{m_a \int_{T_2}^{T_3} C_{pa} dT}{\eta_{dish}\eta_{receiver}} \qquad (6)$$

The energetic or first law efficiency η_{th} of a system and or system component is defined as the ratio of energy output to the energy input to system/component. So the thermal efficiency of the Gas Turbine cycle is given by

$$(\eta_{th})_{GT} = \frac{NetWorkOutputfromGTCycle(W_{net})_{GT}}{IncidentSolarRadiationOnDish(Q_1)} X100 \qquad (7)$$

4.2 HRSG Plant:

Approximate mass flow rate of steam in HRSG plant can be calculated from the overall energy balance equation in the HRSG:

$$m_w = \frac{m_a \int_{T_6}^{T_4} C_{pa} dT}{(h_{12} - h_{10})} \qquad (8)$$

Initially, T_6 is taken as 120^0 C. Later on T_6 is corrected.

T_5 can be found out from the energy balance in the evaporator:

$$m_a C_{pa4}(T_4 - T_5) = m_w(h_{12} - h_{11}) \qquad (9)$$

Based on equation **(8)**, first m_w is calculated and subsequently from equation **(9)**, T_5 is found out. T_5 should be minimum 15^O C more than the saturation temperature of water. If the calculated T_5 does not satisfy this condition, m_w is reduced by 0.0001 Kg/s and T_5 is recalculated. The process is repeated until T_5 satisfies the above condition and finally corrected T_6 is calculated from the following energy balance equation in economizer:

$$T_6 = [T_5 - \frac{m_w(h_{11} - h_{10})}{m_a C_{pa5}}] \qquad (10)$$

Heat energy required in the superheater to raise the saturated steam to the required superheat condition is given by

$$Q_2 = \frac{m_W(h_7 - h_{12})}{\eta_{receiver}\eta_{dish}} \qquad (11)$$

Overall thermal efficiency is given by

$$\eta_{th,overall} = \frac{(W_{net,GT} + W_{ST})}{Q} \qquad (12)$$

Where, $Q = Q_1 + Q_2$

The Temperature-Entropy diagram of the combined cycle is shown in **Fig. 5**.

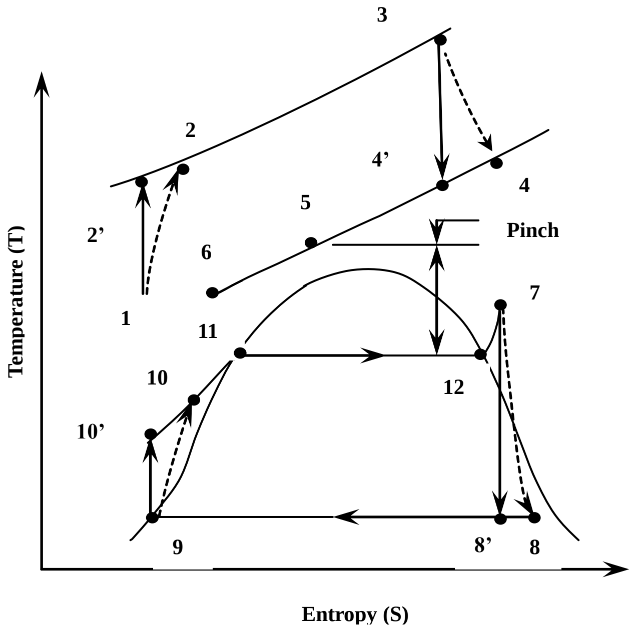

Figure 5: T-S Diagram of the combined cycle

4.3 Exergetic Analysis of the Proposed Combined Cycle

Exergy is a generic term that defines the maximum possible work potential of a system from a stream of matter and or heat interaction with respect to the state of the environment being used as the datum state p_0 =1.01325 bar and T_0 =298K. The total exergy (Ex_{total}) for a steady flow stream, is the sum of its component exergies, vis: physical exergy (Ex_{ph}), chemical exergy (Ex_{ch}), kinetic exergy (Ex_{kin}) and potential exergy (Ex_{pot}) .

$$Ex_{total} = Ex_{ph} + Ex_{ch} + Ex_{kin} + Ex_{pot} \qquad (13)$$

Neglecting the chemical, kinetic and potential exergy, physical exergy for a given stream can be expressed as **[10]**

$$Ex = m\int_{T_0}^{T} CpdT - mT_0 \int_{S_0}^{S} dS \qquad (14)$$

4.3.1 Dish Sub-System

If Ex_Q is the solar radiation exergy and the Ex_{rec} is the exergy delivered to the receiver, then the exergy balance for the dish sub-system is given by

$$Ex_Q = Ex_{rec} + Ex_{dest.} \qquad (15)$$

where,

$$Ex_Q = Q(1 - \frac{T_0}{T_{sun}}) \qquad (16)$$

$$Ex_{rec} = Q_{rec}(1 - \frac{T_0}{T_{sun}}) \qquad (17)$$

$$Q_{rec} = \eta_{dish} Q \qquad (18)$$

Then the exergetic efficiency of the dish sub-system is given by the following equation:

$$\eta_{ex,dish} = \frac{Ex_{rec}}{Ex_Q} \qquad (19)$$

4.3.2 Solar Receiver Sub-System

The solar receiver, which is installed at the focal point of the dish, has been proposed as dual receiver. The input exergies are the exergies of air coming out from compressor (Ex_2), saturated steam (Ex_{12}) and the solar radiation reaching to receiver (Ex_{rec}). The output exergies are the exergies of hot air entering to airt-turbine (Ex_3) and the superheated steam (Ex_7). Then, exergy destroyed in receiver can be computed from the following equation

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$$Ex_{dest,dish} = Ex_{rec} + (Ex_2 - Ex_3) + (Ex_{12} - Ex_7) \quad (20)$$

The exergetic efficiency of the receiver is given by

$$\eta_{ex,rec} = \frac{(Ex_{rec} + Ex_2 + Ex_{12} - Ex_{dest,dish})}{(Ex_{rec} + Ex_2 + Ex_{12})} X100 \quad (21)$$

4.3.3 HRSG Sub-System

For the HRSG, the input exergy is the exergy corresponding to the air exergy at GT exhaust (Ex_4) and the exergy corresponding to the water at economiser inlet (Ex_{10}). Output exergy is the exergy of air at final exit (Ex_6) from HRSG and the exergy of saturated water at evaporator outlet (Ex_{12}). Then the exergy destroyed in the HRSG is given by

$$Ex_{dest,HRSG} = (Ex_4 - Ex_6) + (Ex_{10} - Ex_{12}) \quad (22)$$

The exergetic efficiency of the HRSG sub-system is given by

$$\eta_{ex,HRSG} = (1 - \frac{Ex_{dest,HRSG}}{Ex_4 + Ex_{10}}) \quad (23)$$

5. RESULTS AND DISCUSSIONS

Computer codes using **C** were developed based on the thermal models as discussed in earlier sections. This section describes the base case performance and the energetic as well as exergetic performance of the solar dish based combined cycle power plant at varying pressure ratios of the compressor. The following parameters have been used during the dish based Brayton combined cycle power plant analysis:

a)Efficiency of dish has been considered as 90% **[14]** and of receiver as 82% **[9]**.
b)Gas turbine inlet temperature of air is assumed as 1000^{O}C **[8]**.
c) Steam turbine inlet temperature and pressure of superheated steam as 460^{O}C and 65 bar.
d)Condenser pressure has been taken as 0.07 bar.

5.1 BASE CASE PERFORMANCE

Table 1 shows the base case performance of the combined cycle at pressure ratio 4. At this pressure ratio, the plant delivers a net power of 25.16 kJ, shared by air turbine and steam turbine almost equally. While air turbine shares 13.17 kJ, rest is provided by the bottoming ST block.

Table 1: Base Case Performance of the combined cycle at pressure ratio 4

Parameter	Unit	Value
Air Mass Flow Rate	kg/hr	200.46
Net Air Turbine Work	kJ	13.17
Thermal Efficiency of Topping Cycle	%	19.44
Steam Rate	kg/hr	41.90
Final Exit Temperature of Air	K	401
Net Steam Turbine Work	kJ	11.99
Total Work Obtained from Combined Cycle	kJ	25.16
Overall Thermal Efficiency of Combined Cycle	%	33.06
Exergy Destroyed in Dish	kJ	7.22
Exergy Destroyed in Receiver	kJ	30.37
Exergy Destroyed in HRSG **(38)**	kJ	3.526
Stack Exergy	kJ	0.825

To generate the total power, the plant is required to absorb solar energy of about 76.12 kJ which is equivalent to about 95.15 sq. m. of collector surface exposed to an insolation level of 800 W/m^2. It is also seen in the table that exergy destruction in the solar receiver is the highest among all the other sub-systems.

5.2 Parametric Analysis Based on Energetic Performance

This sub-section describes the energetic analysis of the combined cycle at varying pressure ratios and the pressure ratio has been varied from 2 to 28. The net work obtained from the gas turbine cycle initially increases; takes a maximum value 20.55 kJ at pressure ratio 23 and then decreases with the increase of pressure ratio.

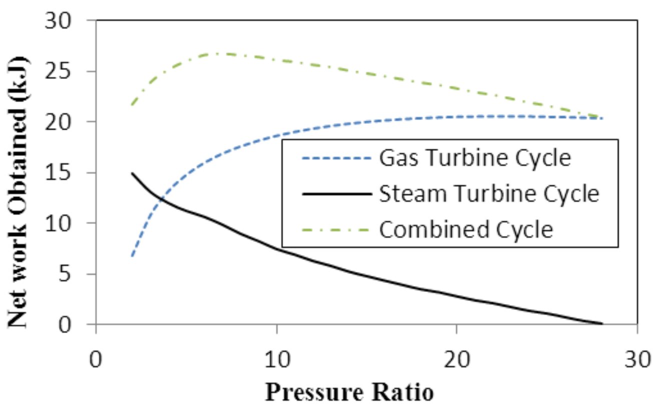

Figure 6: Variation of work obtained from gas turbine, steam turbine and combined cycle

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For the steam turbine cycle, work obtained from the cycle gradually decreases with the increase of pressure ratio since the mass flow rate of water decreases with the increase of pressure ratio as explained in **Fig. 7**. Since the gas turbine work takes the major share in the total work obtained from the combined cycle from pressure ratio 4, the total work obtained from the combined cycle also takes initially the upward trend and then a downward trend with the increase of pressure ratio, which is evident in **Figure 6**.

From the **Fig. 7**, it is evident that air temperature at gas turbine exit is decreasing with the increase of pressure ratio. For the combined cycle, since the pinch point temperature difference is set at minimum 15^{O} C for better heat transfer and temperature after gas turbine expansion is decreasing with the increase of pressure ratio, the evaporation rate for the HRSG is decreasing

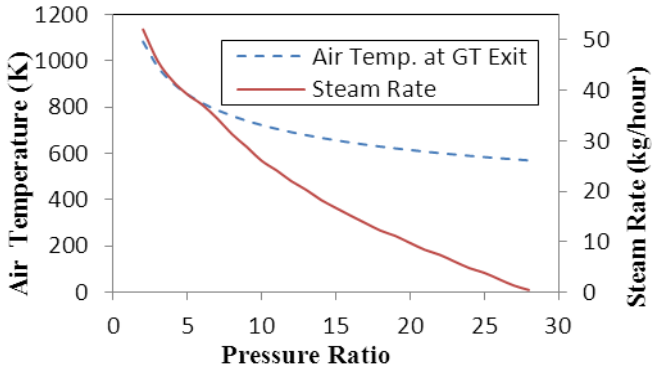


Figure 7: Variation of air temperature at GT exit and steam rate with pressure ratio

with the increase in pressure ratio while the mass flow rate of air is increasing since temperature of air after compression is increasing with the increase of pressure ratio and a fixed amount of heat energy (50 kJ) has been assumed to heat the air to reach a fixed gas turbine inlet temperature of 1000^{O}C, as shown in **Fig. 8**.

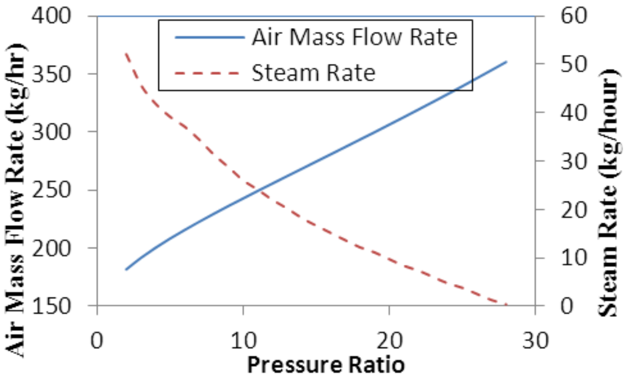


Figure 8: Variation of air mass flow rate and steam rate with pressure ratio

Figure 9 shows the variation of solar insolation required with pressure ratio.

The amount of solar insolation falling on the dish assumes a straight line for Brayton cycle at varying pressure ratios since a constant amount of solar energy has been utilised in the dual receiver to reach the air to constant turbine inlet temperature of 1000^{O}C. Since the mass flow rate of water decreases with the increase of pressure ratio and since the steam turbine inlet temperature and pressure is fixed, the solar insolation required to convert the saturated steam to superheated steam also gradually decreases with the increase of pressure ratio. Hence, the bottoming cycle dominates for the downward trend of the solar energy required for the combined cycle at varying pressure ratios.

Figure 10 shows the variation of thermal efficiency of gas turbine as well as the combined cycle with varying pressure ratio. Initial increase and consecutive decrease of thermal efficiency of gas turbine cycle is attributed to the fact that solar insolation required for the topping cycle has been assumed constant and work obtained from the topping cycle also initially increases and then decreases.

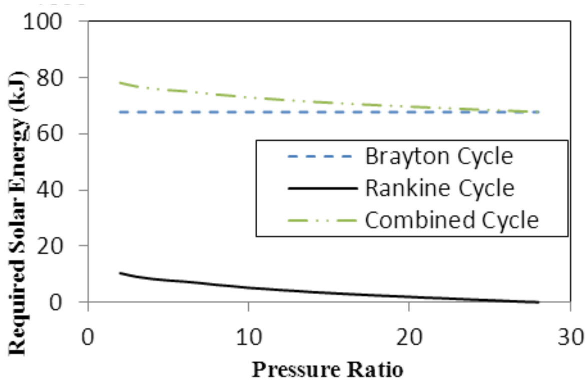
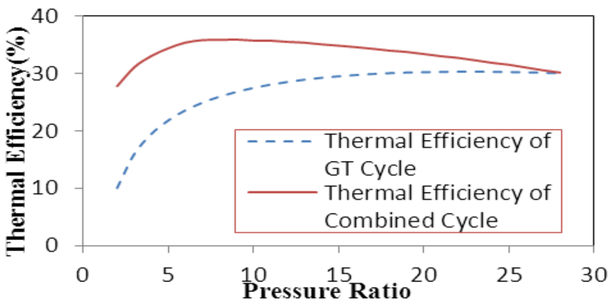


Figure 9: Variation of required solar insolaion with pressure ratio

Overall thermal efficiency of the combined cycle initially increases, maximises at pressure ratio 9 (value being 35.92%) but then goes on decreasing with the increase of pressure ratio. The initial increase in efficiency is attributed to the increased GT work in the lower pressure ratio range. Subsequent decrease in efficiency is due to the fact that the ST work reduces



significantly at higher pressure ratios.

Figure 10: Variation of overall thermal efficiency with pressure ratio

5.3 Parametric Analysis Based on Exergetic Performance

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By exergy analysis, irreversibility or the inefficiency of a process is found out. **Figure 11** shows the lost exergy for different components (compressor, dish, solar receiver, gas turbine, HRSG) and also the stack loss for the conceptualized plant at pressure ratio of 9, where the overall thermal efficiency of the combined cycle maximises. It is evident from the figure that at this pressure ratio, exergy destruction is highest for solar receiver and the exergy loss in case of solar receiver is very large because heat is transferred from a very high temperature of heat source to a comparatively low temperature solar receiver. Considerable amount of exergy is also destroyed at solar dish, HRSG and stack. That's why parametric analysis of these components have been discussed subsequently.

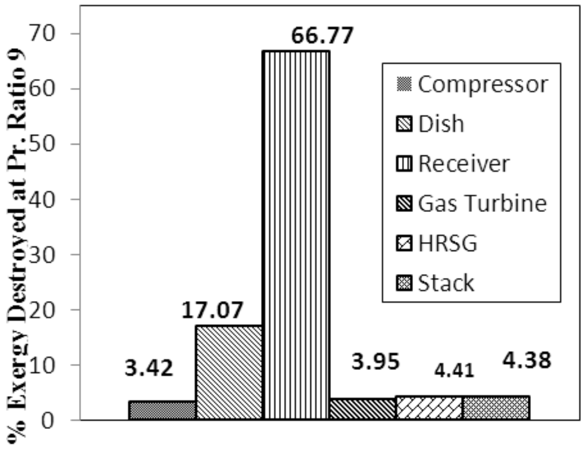

Figure 11: % Exergy destruction of different components at pressure ratio 9

The effect of varying pressure ratios on exergy destroyed and the exergetic efficiency of solar receiver is shown in **Fig. 12**. The decrease of exergy destruction with increased pressure ratio is due to the fact that steam rate is reducing with the increase of pressure ratio and hence amount of heat energy required to superheat the steam in solar receiver is also reducing, resulting in decrease in associated exergy.

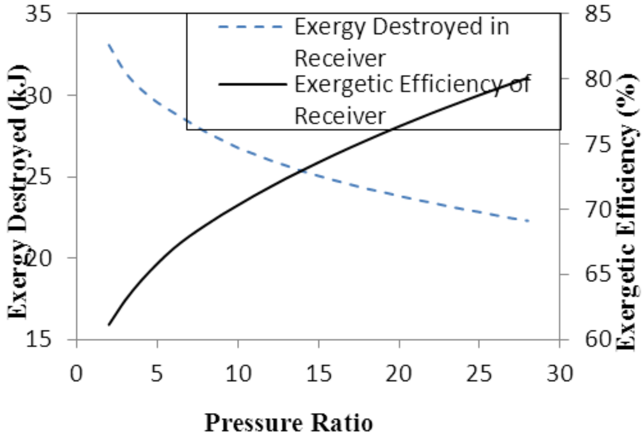

Figure 12: Variation of exergy destroyed and exergetic efficiency of solar receiver with pressure ratio

Since the exergy destruction is reducing with the increase of pressure ratio, exergetic efficiency of the solar receiver is increasing with the increase of pressure ratio.

For the dish sub-system, the exergetic performance is improving i.e. the exergy destruction is reducing with the increase of pressure ratio. This is due to the fact that with the increase of pressure ratio of the compressor, the amount of heat energy required in the solar receiver is reducing due to reduced steam rate, resulting in reduction in associated exergy. But the exergetic efficiency curve assumes a straight ine at 90% since the dish efficiency has been considered as 0.9. The variation of exergy destroyed and the exergetic efficiency of dish sub-system is shown in **Fig. 13**.

The variation of exergy loss in HRSG and the exergetic efficiency of HRSG with increasing pressure ratio are shown in **Fig. 14**.

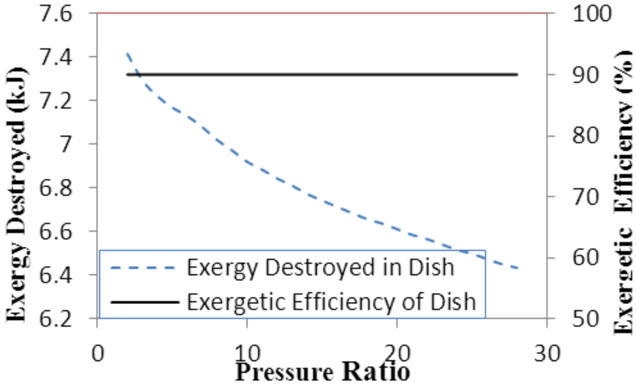

Figure 13: Variation of exergy destroyed and exergetic efficiency of dish sub-system with pressure ratio

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With the increase of pressure ratio, evaporation rate of the HRSG plant decreases since the minimum pinch point temperature difference is set at 15^{O}C, as explained in **Fig. 8**. Since the evaporation rate is decreasing, exergy associated with it is also decreasing and this is ultimately responsible for the downward nature of the exergy destruction curve with pressure ratio. Since the exergy destruction is decreasing with the increase of pressure ratio, the exergetic efficiency is increasing with pressure ratio.

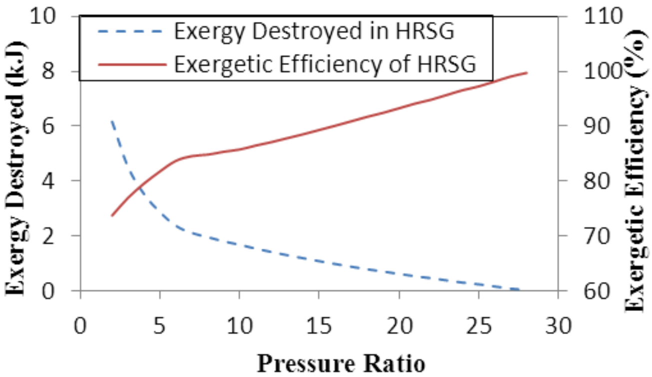

Figure 14: Variation of exergy destroyed and exergetic efficiency of HRSG with pressure ratio

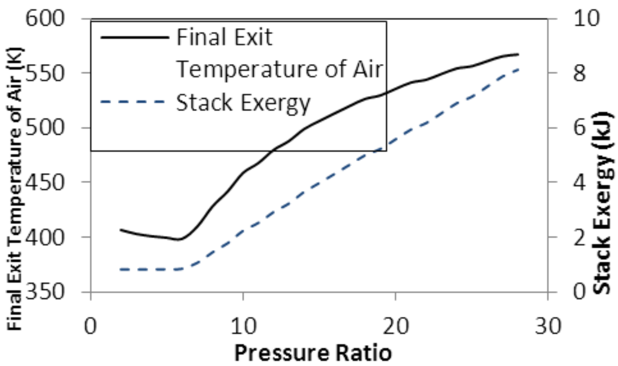

Figure 15: Variation of stack temperature and stack exergy with pressure ratio

Figure 15 shows the variation of stack temperature and stack exergy with varying pressure ratio. To maintain the pinch point temperature difference minimum 15^{O}C, mass flow rate of water is adjusted and this is responsible for the variation in the evaporator outlet temperature of air. The variations in the evaporator outlet temperature of air, mass flow rate of air and the water are combinedly responsible for the initial decrease and then increase of the stack temperature with pressure ratio. Since stack temperature is initially decreasing and then increasing, stack exergy is also initially decreasing and then increasing with the increase of pressure ratio.

6. CONCLUSION

In this present paper, a solar dish based Brayton-Rankine combined cycle has been proposed. The hot exhaust air of the topping cycle is utilized to run a downstream HRSG. This paper also proposes a dual receiver, which can be utilized for heating the air as well as to produce the superheated steam. The energetic as well as exergetic performance of the combined cycle has been carried out for the varying pressure ratio of the GT block, which has been varied from 2 to 28. From the energetic performance, it is found that the both the net work obtained from the gas turbine cycle and the thermal efficiency maximizes at pressure ratio 23, the value being 20.55 kJ and 30.33 % respectively. But for the combined cycle, both the maximum total work obtained and the maximum overall thermal efficiency are exhibited at much lower pressure ratio. The total work obtained from the combined cycle maximizes at pressure ratio 7, value being 26.74 kJ and the overall thermal efficiency maximizes at pressure ratio 9, value being 35.92%. From the exergetic performance, it is seen that solar receiver takes the major share for the exergy destruction among all the other sub-systems. Though for the components like solar receiver, dish and HRSG, the exergetic performance is improving i.e. the exergy destruction is reducing at higher pressure ratios, the total work obtained from the combined cycle as well as the overall thermal efficiency reduces at higher pressure ratios. Since the total work obtained from the combined cycle is maximum at pressure ratio 7 and the overall thermal efficiency is maximum at pressure ratio 9, the combined cycle should be operated at pressure ratio between 7 and 9.

NOMENCLATURE

T	Absolute temperature (K)
p	Absolute pressure (bar)
η_c	Isentropic efficiency of compression
m_a	Mass flow rate of air (kg/s)
W	Work (kJ)
C_{pa}	Specific heat of air at constant pressure (kJ/kg-K)
C_{va}	Specific heat of air at constant volume (kJ/kg-K)
Q	Heat energy (kJ)
η_{dish}	Dish efficiency
$\eta_{receiver}$	Receiver efficiency
η_{GT}	Isentropic efficiency of gas turbine
η_{mech}	Mechanical efficiency of GT-Comp and ST-Pump
η_G	Generator efficiency
η_{th}	Thermal efficiency
h	Enthalpy
v	Specific volume (m^3/kg)
η_{ST}	Isentropic efficiency of steam turbine
m_w	Mass flow rate of water (kg/s)
W_{ST}	Steam turbine work (kJ)
S	Entropy (kJ/K)

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