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MW LEVEL SOLAR POWERED COMBINED CYCLE PLANTS: PERFORMANCE PREDICTION USING THERMODYNAMIC METHODS

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ABSTRACT

This paper presents a conceptual configuration of a solar powered combined cycle plant with a topping gas turbine block and a bottoming steam turbine plant. The conventional GT combustion chamber is replaced by a high-temperature solar thermal air heating system. A simple downstream cycle with a heat recovery steam generator (HRSG) has been considered for integration with the solar GT plant. The power cycle has been analyzed using energy as well as exergy methods for a range of pressure ratio across the GT block. The solar receiver and the HRSG are the two main components responsible for exergy destruction. However, exergetic performance of both the components is found to improve at higher pressure ratio of the GT block.

From the thermodynamic analysis, it is found that thermal efficiency of the GT block is maximum (18.18%) at pressure ratio 14 whereas exergetic efficiency is maximum (35.54%) at pressure ratio 4. But by incorporating a simple downstream HRSG, the integrated system is found to give maximum overall thermal efficiency at a pressure ratio 6 (value being 26.16%), while still giving an exergetic efficiency of 29.24% (maximum exergetic efficiency 30.54% though occurring at pressure ratio 14). Thus the combined cycle can be designed to operate at pressure ratio of 6 to yield optimum performance.

INTRODUCTION

In the present century, energy market is the fastest developing sector for most of the countries. Annual demand for electricity is increasing day by day during recent years. Analyses of power generation systems are of scientific interest and also essential for the efficient utilization of energy resources. The most commonly used method for analysis of an energy-conversion process is the first law of thermodynamics. However, there is increasing interest in the combined utilization of the first and second laws of thermodynamics, using such concepts as exergy and exergy destruction in order to evaluate the efficiency with which the available energy is consumed. Exergetic analysis provides the tool for a clear distinction between energy losses to the environment and internal irreversibilities in the processes [1].

In view of the growing worldwide emphasis on the optimization of the energy conversion processes [2], second law based exergy analysis of thermal power plants has become important for the researchers. The first law is basically concerned with the enthalpy balance while the second law makes an exergy assessment, which allows one to quantify the loss of efficiency in a process due to loss of quality of the energy. This analysis can indicate the where the process can be improved and, therefore, what areas should receive attention. So it is clear that improved utilisation of energy can come with the reduction of availability destruction within a device or process or with the decrease of irreversibility. Through availability analysis, those sites where destructions or losses occur can be identified and necessary improvements can be made on those areas [2].

Since the use of natural gas as the fuel in combustion chamber will produce CO_2 / CO as the pollutants, solar energy option is a very clean source of energy. Solar energy is a very large and inexhaustible source of energy. The power from the sun intercepted by the earth is approximately $1.8X10^{11}$ MW, which is many thousands of times larger than the present consumption rate on the earth of all commercial energy sources. Thus, in principle, solar energy could supply all the present and future energy needs of the world on a continuing basis. This makes it one of the most promising of the unconventional energy sources. Two main factors are in the favour of utilization of solar energy. First unlike fossil fuels and nuclear power, it is environmentally clean source of energy. Second it is free and available in adequate quantities in almost all parts of the world where people live [3].

Solar Thermal Power Plants (STPP) with optical concentration technologies are important candidates for providing a major share of the clean and renewable energy needed in the future, even though they still suffer from lack of dissemination and confidence among citizens, scientists and decision makers. Although the solar radiation is a high quality energy source because of the high temperature and exergy at its source, its power density at the earth's surface makes it difficult to extract work and achieve reasonable temperatures in common working fluids [4-5].However recent developments enable use of air as working fluid in solar thermal plant. Using concentrators and volumetric receivers, air can be heated to a temperature level as high as 1000^{O}C and expanded in a gas turbine [6]. This also enables integration of bottoming steam cycle with the gas turbine plant and thus leading to solar power combined cycle [7].

NOMENCLATURE

W	[MW]	Power output
T	[K]	Temperature
HRSG	[-]	Heat Recovery Steam Generator
GT	[-]	Gas Turbine
X	[MW]	Exergy flow rate
p	[bar]	Absolute pressure
Q	[MW]	Heat rate
h	[kJ/kg]	Enthalpy of steam
C	[kJ/kg-K]	Specific heat
m	[kg/s]	Mass flow rate
S	[kJ/kg-K]	Entropy
R	[kJ/kg-K]	Characteristic gas constant

Special characters

η_c	[%]	Isentropic efficiency of compression
η_{hfo}	[%]	Heliostat field optic efficiency
η_{rac}	[%]	Receiver and air circuit efficiency
$(\eta_e)_{GT}$	[%]	Isentropic efficiency of expansion of gas turrbine
η_{ex}	[%]	Exergetic efficiency
$(\eta_e)_{ST}$	[%]	Isentropic efficiency of expansion of steam turrbine

Subscripts

0	Dead state
a	Air
th	Thermal
comp	Compression
exp	Expansion
p	At constant pressure
v	At constant volume
W	Water
E,i	Evaporator inlet
E,o	Evaporator outlet

PLANT CONFIGURATION AND COMPONENT DESCRIPTION

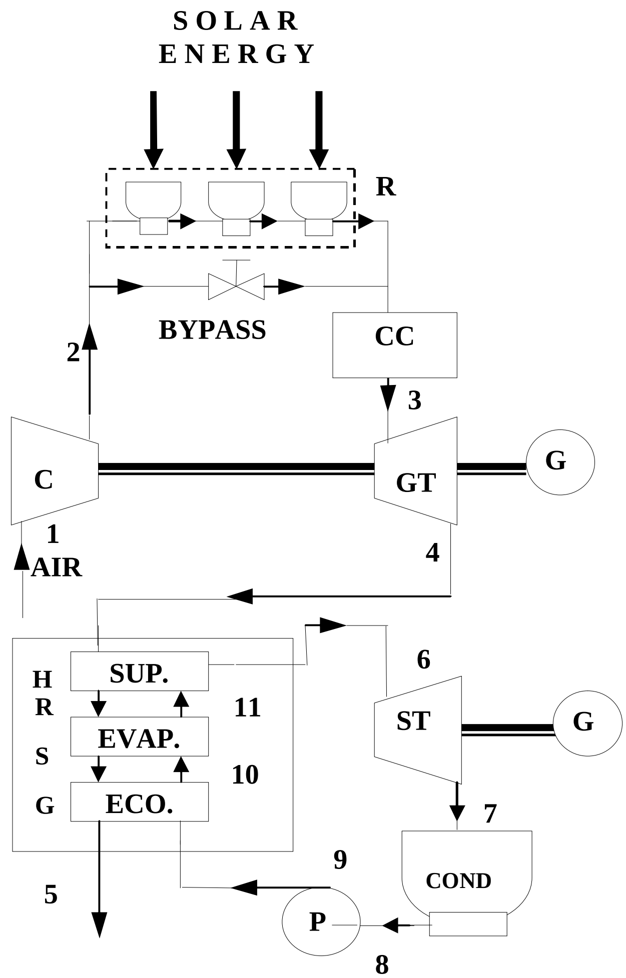

Fig.1: Schematic of Solar Powered Gas Turbine Plant

Figure 1 shows the schematic of a solar powered gas turbine combined cycle considered in the present study. Ambient air (m_a=15 Kg/s) at point 1 (300K and 1.01325 bar pressure) enters the compressor (**C**) and leaves at point 2. Compressed air then enters the solar receiver (**R**) which raises the air temperature in three stages to a final temperature of 1000^{0}C. Heated air enters the gas turbine (**GT**) at point 3 and expands to point 4. A bypass is provided for the solar receiver leading directly to the combustion chamber (**CC**) of the gas turbine. When proper solar insolation is not there due to bad weather or at off-sunshine hours, the combustion chamber is switched on and supplementary gas firing is accomplished to maintain the required GT inlet temperature. Air is heated until 1000^0 C [6] (1273 K) is achieved. Heated air at 1000^0 C enters into the gas turbine (**GT**) at point 3 and expands in the turbine. This exhaust from GT is passed through a HRSG to produce superheated steam which in turn will be utilized to run a steam turbine in the bottoming cycle. In the HRSG, air is first used to raise the temperature of the steam in the superheater (**SUP**) from point 11 to the point 6. Then the air is used to evaporate the water from point 10 to 11 in the evaporator (**EVAP**) and finally, it is utilized to heat the feed water in the economiser (**ECO**) from point 9 to 10. Then the air is left to the atmosphere at minimum 120^0 C (393 K) at point 5. The superheheated steam at point 6 (90 bar pressure and 350^0 C) enters into the steam turbine (**ST**), and after doing work in the steam turbine, steam is exhausted in the condenser (**COND**) at point 7 at 0.075 bar pressure. After being condensed in the condenser, water is pumped from point 8 to point 9 i.e. at the boiler pressure by a feed pump (**P**). The system as well as the programming 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. $T_{10} = T_{11}$) does not fall below 15^0 C for better heat transfer.

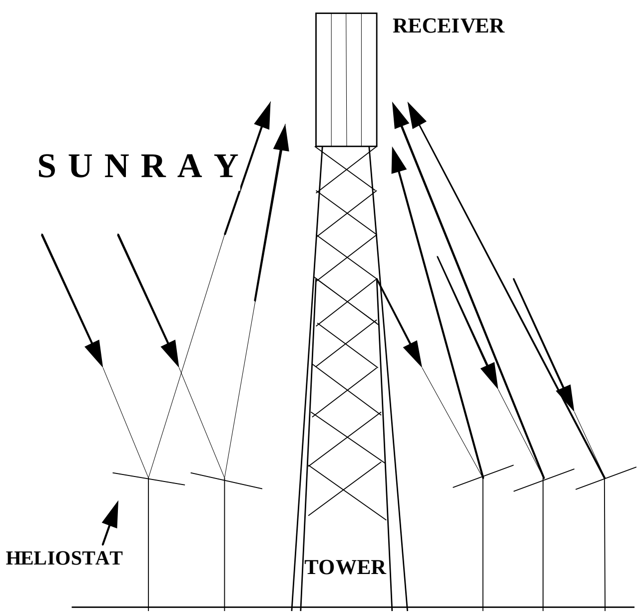

Fig. 2: Schematic of Central Receiver System

METHODOLOGY ADOPTED IN PRESENT STUDY

Gas turbine cycle:

Since temperature of air at the entry to the compressor and pressure ratio (r) is known, temperature after isentropic compression is given by the following equation considering polytropic index of compression (k1) as 1.4

$$T_{2'} = T_1 \left(\frac{p_2}{p_1} \right)^{\frac{k1-1}{k1}} = T_1 (r)^{\frac{k1-1}{k1}} \quad (1)$$

But after actual compression in the compressor, temperature is higher than the temperature after isentropic compression due to irreversibility. Considering the isentropic efficiency of compression (η_c) equal to 0.8, temperature after actual compression is

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

Considering the mass of air (m_a) as 15 Kg/s, workdone during compression is given by the following equation

$$W_{comp} = m_a C_{pa}(T_2 - T_1) \quad (3)$$

For the gas turbine cycle, heating of the working fluid i.e. air is considered by the solar insolation. Solar radiation is tracked by the large field heliostats and then it is concentrated to the receiver. Considering heliostat field optic efficiency (η_{hfo}) and receiver and air circuit efficiency (η_{rac}) as 0.729 and 0.740 respectively, solar insolation required for heating is given by

$$Q = \frac{m_a C_{pa}\left(T_3 - T_2\right)}{\eta_{hfo}\eta_{rac}} \quad (4)$$

Gas turbine inlet temperature (T_3) has been taken as 1273 K [6]. Considering polytropic index of expansion (k2) as 1.4 and the same pressure ratio, temperature after isentropic expansion is given by

$$T_{4'} = T_3 \left(\frac{p_4}{p_3} \right)^{\frac{k2-1}{k2}} = T_3 \left(\frac{1}{r} \right)^{\frac{k2-1}{k2}} \quad (5)$$

Again temperature after actual expansion process in the gas turbine is higher than the temperature after isentropic expansion due to irreversibility. Considering the isentropic efficiency of expansion ($\eta_{e,GT}$) as 0.85, the actual temperature of air after expansion in gas turbine is given by

$$T_4 = T_3 - \eta_{e,GT}\left(T_3 - T_{4'}\right) \quad (6)$$

Gas turbine expansion work (W_{exp}) is given by the following equation

$$W_{\exp} = m_a C_{pa}(T_3 - T_4) \quad (7)$$

Net workdone in topping gas turbine cycle can be found out by deducting the compressor work from the gas turbine expansion work and it is

$$(W_{net})_{GT} = W_{exp} - W_{comp} \qquad (8)$$

Thermal efficiency of gas turbine cycle is the ratio of net workdone in the gas turbine cycle and the required solar insolation.

$$(\eta_{th})_{GT} = \frac{(W_{net})_{GT}}{Q} \qquad (9)$$

HRSG Plant:

Approximate mass flow rate of water (m_w) in the bottoming steam turbine plant can be found out from the overall energy balance in the heat recovery steam generator (HRSG) and the enthalpy (h) at different point can be found out from the steam table. Approximate mass flow rate of water in HRSG can be calculated from the following equation:

$$m_w\left(h_6 - h_9\right) = m_a C_{pa}\left(T_4 - T_5\right)$$
$$or, m_w = \frac{m_a C_{pa}\left(T_4 - T_5\right)}{\left(h_6 - h_9\right)} \qquad (10)$$

Initially, T_5 is taken as 120^0 C. Later on T_5 is corrected.

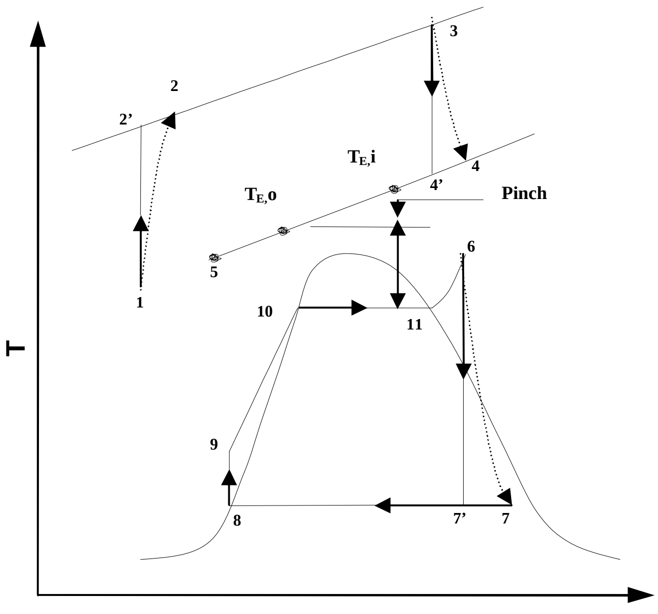

Fig. 3: T-S diagram for the combined cycle

In the superheater, steam is superheated by the hot exhaust air from the gas turbine. Evaporator inlet temperature of air ($T_{E,}$i) can be calculated from the energy balance equation in the superheater and is given by

$$m_w\left(h_6 - h_{11}\right) = m_a C_{pa}\left(T_4 - T_{E,i}\right)$$
$$or, T_{E,i} = T_4 - \frac{m_w\left(h_6 - h_{11}\right)}{m_a C_{pa}} \qquad (11)$$

In the evaporator also, water is evaporated by taking the heat energy from air. Outlet temperature of air ($T_{E,}$o) from the evaporator can be calculated from the energy balance equation in evaporator:

$$m_w\left(h_{11} - h_{10}\right) = m_a C_{pa}\left(T_{E,i} - T_{E,o}\right)$$
$$or, T_{E,o} = T_{E,i} - \frac{m_w\left(h_{11} - h_{10}\right)}{m_a C_{pa}} \qquad (12)$$

Based on equation (10), first m_w is calculated and subsequently from equations (11) and (12), $T_{E,i}$ and $T_{E,O}$ have been found out. For better heat transfer, $T_{E,O}$ should be 15^0C more than the saturation temperature of water. If the calculated $T_{E,O}$ does not satisfy this condition, m_w is reduced by 0.05 Kg/s and $T_{E,O}$ is recalculated. The process is repeated until $T_{E,O}$ satisfies the above condition and finally corrected T_5 is calculated from the following energy balance equation in economizer:

$$m_w\left(h_{10} - h_9\right) = m_a C_{pa}\left(T_{E,o} - T_5\right)$$
$$or, T_5 = T_{E,o} - \frac{m_w\left(h_{10} - h_9\right)}{m_a C_{pa}} \qquad (13)$$

Considering the isentropic efficiency of steam turbine ($\eta_{e,ST}$) as 0.85, enthalpy of steam at the exit of steam turbine (h_7) can be found out from the following equation

$$\eta_{e,ST} = \frac{h_6 - h_7}{h_6 - h_{7'}}$$
$$OR, h_7 = h_6 - \eta_{e,ST}\left(h_6 - h_{7'}\right) \qquad (14)$$

Workdone in steam turbine is calculated from the following equation

$$W_{ST} = m_w\left(h_6 - h_7\right) \qquad (15)$$

Overall thermal efficiency of the combined plant is ratio of total work obtained from the gas turbine as well as the steam turbine and the required solar insolation. It is given by

$$(\eta_{th})_{overall} = \frac{(W_{net})_{GT} + W_{ST}}{Q} \qquad (16)$$

Exergy Analysis:

For a flow stream, the total exergy (X) is the sum of its component exergies, vis: physical exergy (X_{ph}), chemical exergy (X_{ch}), kinetic exergy (X_{kin}) and potential exergy (X_{pot}).

$$X = X_{ph} + X_{ch} + X_{kin} + X_{pot} \qquad (17)$$

The physical exergy is the maximum theoretical useful work obtainable as the system passes from its initial state to the restricted dead state specified by the assumed ambient condition described by p_0=1.01325 bar and T_0=300K. The chemical exergy is the maximum theoretical useful work obtainable as the system passes from the restricted dead state to the dead state where it is in complete equilibrium with the reference environment. Neglecting the kinetic and potential exergies and heat transfer to and from the control volume, the exergy input (X_{in}) to equipment of the plant is the sum of exergies of the streams entering into it and the work inputs (W_{in}), if any [8]

$$X_{in} = \sum (X_i)_{in} + \sum (W_i)_{in} \tag{18}$$

where, suffix 'i' indicates the stream of flow or work and X is the stream Exergy (X), expressed as:

$$\begin{aligned} X &= m_a (h - h_0) - m_a T_0 (S - S_0) \\ &= m_a C_{pa} (T - T_0) - m_a T_0 \left[C_{va} \ln\left(\frac{T}{T_0}\right) - R_a \ln\left(\frac{p X T_0}{p_o X T}\right)\right] \end{aligned} \tag{19}$$

Similarly, exergy coming out of any equipment (X_{out}) is described by :

$$X_{out} = \sum (X_i)_{out} + \sum (W_i)_{out} \tag{20}$$

Following this definition exergetic efficiency (η_{ex}) can be expressed as

$$\eta_{ex} = \frac{X_{out}}{X_{in}} \tag{21}$$

As per Exergetics by Goran Wall Exergy associated with solar radiation [9]

$$X_{solar} = 0.933 \times Q \tag{22}$$

Exergetic efficiency of solar powered GT plant only, without considering HRSG, is given by:

$$(\eta_{ex})_{GT} = \frac{(W_{net})_{GT} + X_4}{X_{solar}} \tag{23}$$

Overall exergetic efficiency of the combined cycle plant is:

$$(\eta_{ex})_{overall} = \frac{(W_{net})_{GT} + W_{ST} + X_5}{X_{solar}} \tag{24}$$

Exergy Loss for any component in the cycle is given by:

$$X_{Loss} = X_{in} - X_{out} \tag{25}$$

RESULTS AND DISCUSSIONS

The base case performance of the solar powered gas turbine plant is shown in **Table 1**. At the topping cycle (GT) pressure ratio of 4, the plant delivers a net power of 6.142 MW while both the GT and the ST share the load almost equally. To generate this power, the plant is required to absorb solar power of about 24.418 MW which is equivalent to about 34883 sq. m of collector surface exposed to an insolation level of 700 W/m^2.

Table 1: Base case performance of the solar powered gas turbine combined plant

Parameter	Unit	Value
Air mass flow rate for GT	kg/s	15
Compressor pressure ratio	-	4
Gas Turbine inlet	K	1273
Net power from GT Block	MW	3.342
Total incident solar radiation	MW	24.418
Exergetic efficiency of GT	%	35.533
Water mass flow for HRSG	kg/s	3.024
Steam turbine output	MW	2.8
Net plant output (GT+ST)	MW	6.142
Overall Thermal Efficiency	%	25.152
Overall ExergeticEfficiency	%	27.761
Overall heat rate of combined cycle plant	kJ/kWh	14314

It is interesting to see the variation of plant performance with a varying compressor pressure ratio while GT inlet temperature is held constant at a value of 1000^{O}C .

Figure 4 shows the variation of work done with pressure ratio. For the combined cycle, work output is maximum at pressure ratio 4, work output being 6.14 MW. With the increase of pressure ratio, mass flow rate of water is decreasing and so steam turbine work output is decreasing. Therefore, combined cycle work output takes a downward trend with increase of pressure ratio.

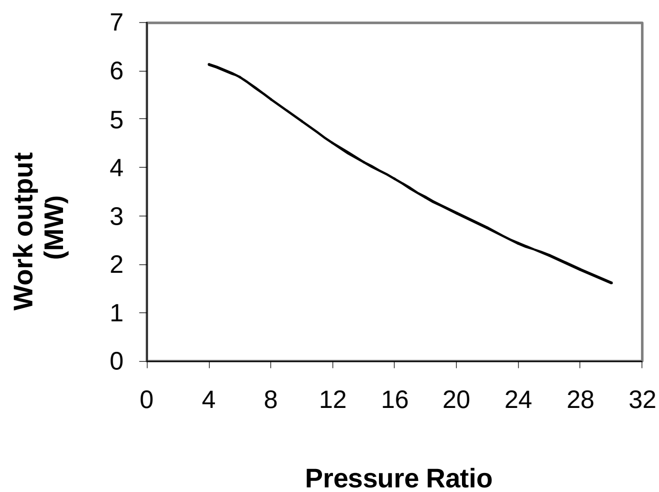

Fig. 4: Variation of work output with pressure ratio

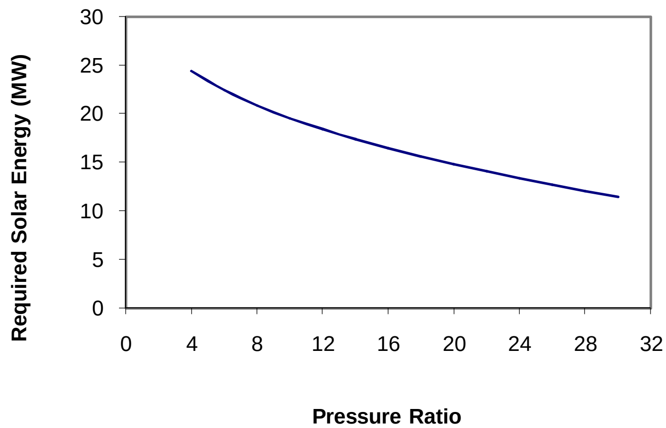

Fig. 5: Variation of required solar energy with pressure ratio

Figure 5 shows the variation of solar insolation required with pressure ratio. With increase in pressure ratio, temperature after compression increases. Since gas turbine inlet temperature is fixed, the required solar energy goes on decreasing with the increase of pressure ratio.

Figure 6 shows the variation of mass flow rate of water in the HRSG with pressure ratio. For the combined cycle, since the minimum pinch point temperature difference is set at minimum 15^{O} C and temperature after gas turbine expansion is decreasing with pressure ratio, the evaporation rate for the HRSG is decreasing with increase in pressure ratio.

Figure 7 shows the variation of overall thermal efficiency with pressure ratio. Overall thermal efficiency of the combined cycle initially increases 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 both the GT work and the ST work are reduced significantly at higher pressure ratio.

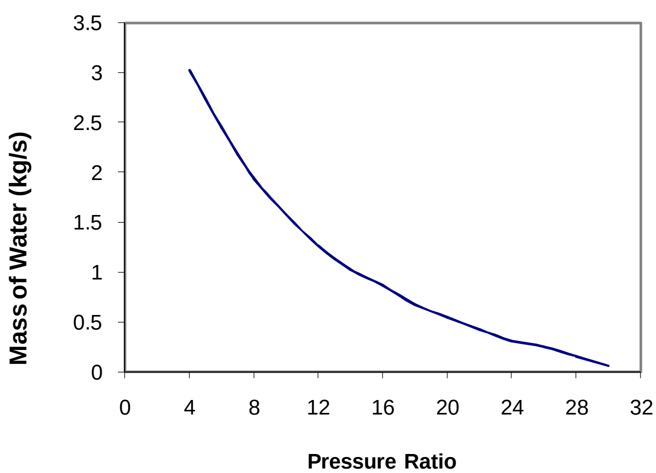

Fig.6: Variation of mass flow rate of water with pressure ratio

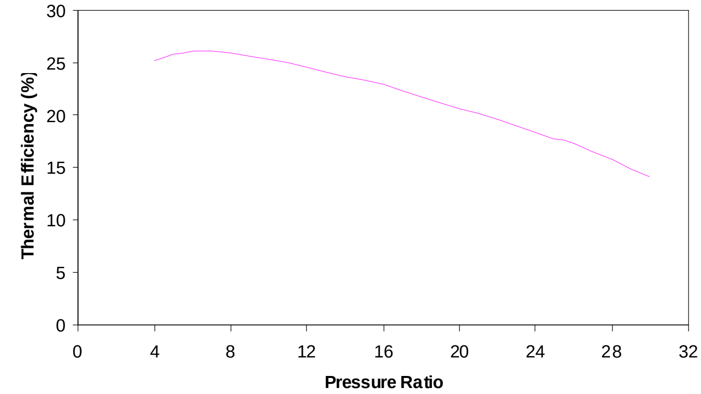

Fig. 7: Variation of overall thermal efficiency with pressure ratio

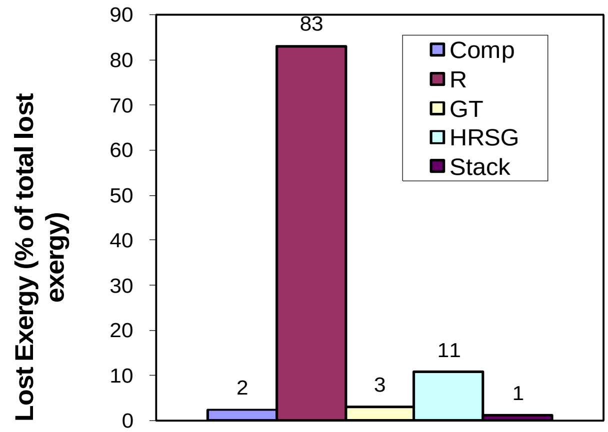

Fig. 8: Lost exergy of different components at base case pressure ratio 4

Figure 8 shows the lost exergy for different components (compressor, solar receiver, gas turbine, HRSG) and also the stack loss for the base case configuration (at pressure ratio of 4). It is evident from the figure that considerable amount of exergy loss is occurring for solar receiver and for HRSG while the losses in the other components are insignificant.

Variation of exergy loss of solar receiver is shown in **Fig. 9**. 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.

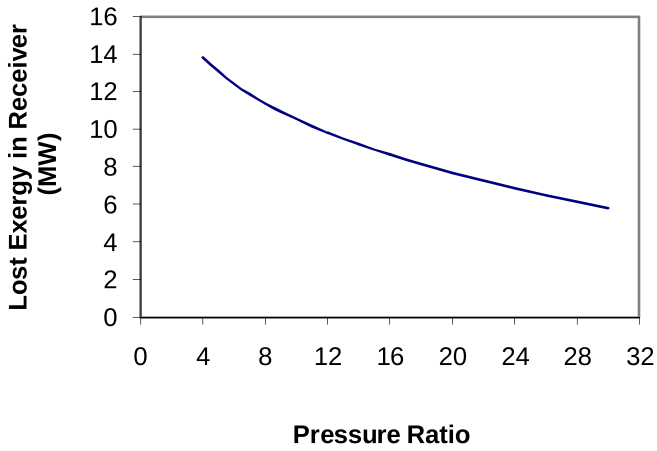

Fig.9: Variation of lost exergy in receiver

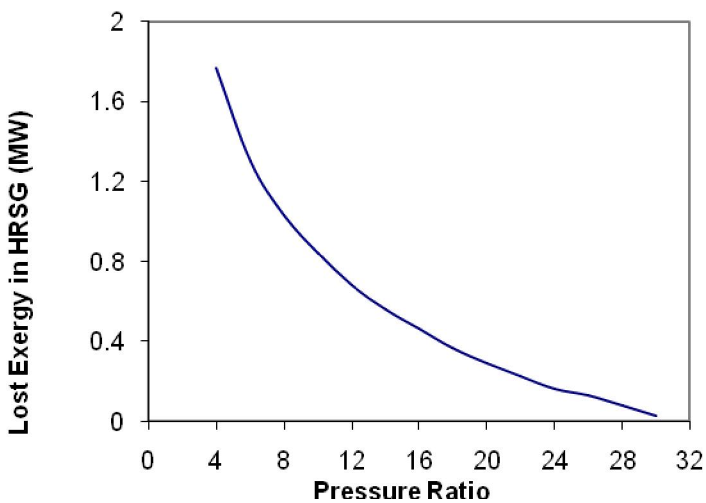

Fig. 10: Variation of lost exergy in HRSG

The variation of exergy loss in HRSG is shown in **Fig. 10**. With the increase of pressure ratio, evaporation rate of the HRSG plant decreases as the minimum pinch point temperature difference is set at 15^0 C. Therefore temperature of stack is increasing with the increase of pressure ratio increasing the exergy associated with air at exit from HRSG and so, exergy loss in the HRSG is decreasing with the increase of pressure ratio.

Figure 11 shows the variation of overall exergetic efficiency of the combined cycle with pressure ratio. It shows that exergetic efficiency of the combined cycle initially increases but then decreases with pressure ratio.

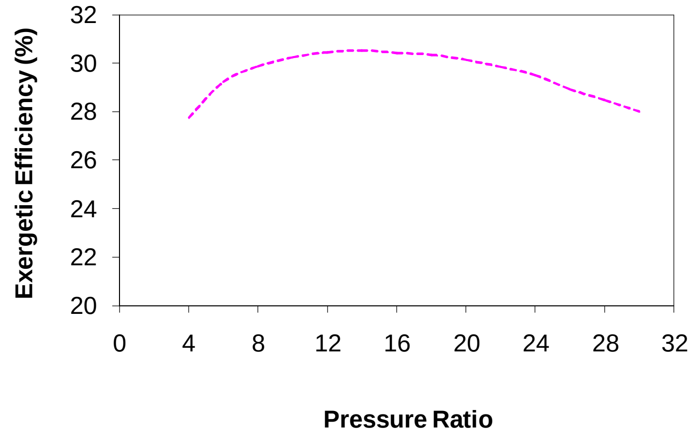

Fig. 11: Variation of overall exergetic efficiency with pressure ratio

CONCLUSION

From the energetic and exergetic analysis, it is found that for the combined cycle the maximum overall thermal efficiency is at pressure ratio 6 (value being 26.16%), and at this pressure ratio, exergetic efficiency is 29.24%. The maximum exergetic efficiency for the combined cycle is 30.54% occurring at pressure ratio 14. Therefore the combined cycle can be designed to operate at pressure ratio of 6 to yield maximum overall thermal efficiency and close to maximum overall exergetic efficiency.

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