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Theoretical Investigation of Solar Energy Driven Combined Power and Refrigeration Cycle

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Abstract; *This investigation is done for energy and exergy analysis of a combined power refrigeration cycle using oil as the heat transfer medium. This cycle is an integration of Rankine cycle for power production and ejector refrigeration cycle for cold production. The effects of parameters like; steam temperature, and the evaporator temperature of ejector have been observed on first and second law performance. The first law efficiency of solar driven combined cycle is found to be is 20% while second law efficiency is 11%.*
Keywords: *Solar energy, ejector, first law efficiency, second law efficiency*.

Nomenclature

A_h	aperture area of heliostat [m^{-2}]
q	solar radiation received per unit area [Wm^{-2}]
$\dot{E}$	exergy rate [kJ s^{-1}]
$\dot{Q}$	energy rate [kJ s^{-1}]
$\Delta\dot{E}$	exergy change [kJ s^{-1}]
RC	Rankine cycle
ERC	ejector refrigeration cycle
μ	entrainment ratio
η	efficiency[%]

Subscript

E	evaporator
C	condenser
CR	central receiver
D	destruction
HRVG	heat recovery vapor generator
EJE	ejector
d	diffuser
n	nozzle
m	mixing chamber
pf	primary flow
sf	secondary flow
n1	inlet of nozzle
n2	outlet of nozzle
1, 2, 3……….	state points in Fig.1

I. Introduction

Low temperature heat sources driven combined power and cooling thermodynamic cycle is proposed by Goswami (2000) has been under intense investigation. This cycle is the combination of Rankine power and vapor absorption refrigeration cycle operates on NH_3/H_2O mixture. On the other hand ejector refrigeration cycle has the advantages of simplicity in construction, low capital cost, high reliability, silent operation and very low maintenance cost. The refrigerants applied to the ejector refrigeration cycles, has almost zero ozone depletion potential and moderate global warming potential. Very few investigations are reported in the literature on thermodynamic evaluation of ejector refrigeration cycles. Sun and Eames (1996) described a simulation model for ejector refrigeration system working with R123 as an alternative for R11. Their results showed that R123 is a suitable replacement for R11 in space cooling applications.

Description of the proposed cycle: This cycle consists of six major components: heliostat field, central receiver, HRSG, turbine, evaporator, condenser, ejector, and pump and the schematic diagram is shown in Fig.1. The working fluid (4), which is water, is pumped to the HRSG (1) capable of producing the higher pressure of interest. Leaving the pump (1), the working fluid passes through a vertical stacked plate heat recovery steam generator (HRSG) where it generates steam from the heat source of 450^0C which oil that generate superheat steam at the temperature of 400^0C The superheated steam (2) passes through the turbine where electricity is produced with the help of generator. The bleed steam (5) from the turbine is mixed with the steam coming from the evaporator (7) and then goes to the top of the condenser (6) through a spray nozzle across the cooling element inside. The working fluid steam which is also working as refrigerant for cooling goes to the HRSG via pump (4). Water is filled in the evaporator (flash chamber) from where it is evaporated to the ejector (7) and cooling of the remaining water takes place in the flash chamber. The makeup water is entering at (8) as shown in figure 1

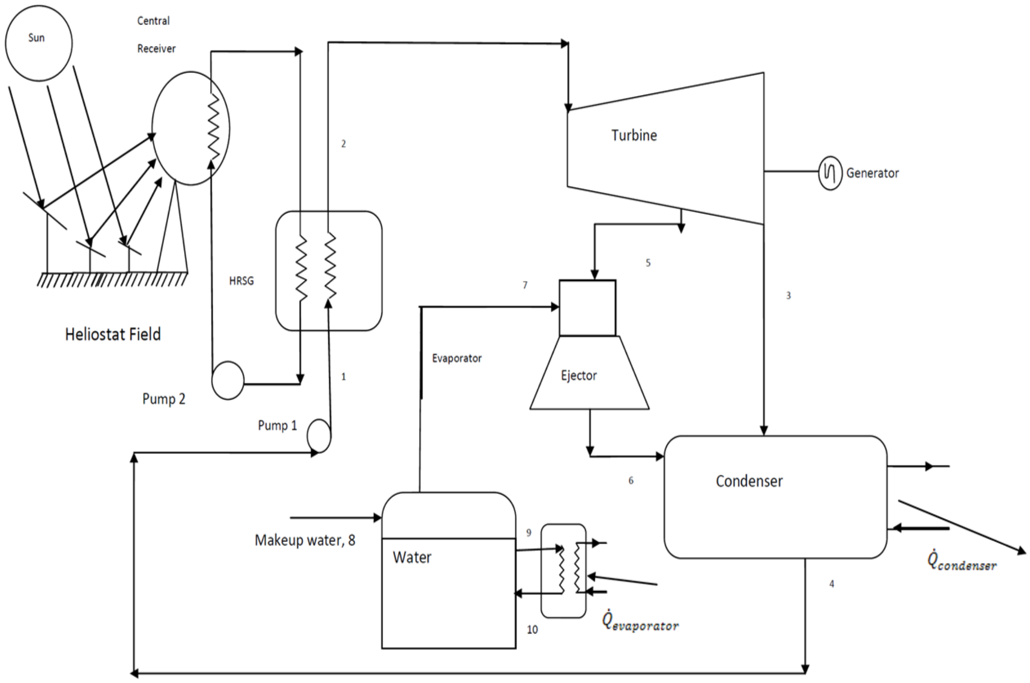

Figure 1. Schematic Diagram of the proposed solar thermal driven combined power and cooling system

The following assumptions have been made for the analysis of the proposed cycle;

(1) The components of the cycle are in steady state, and pressure drop in pipes and heat losses to the environment in the HRVG, turbine, condensers and evaporators are neglected.
(2) The flow through the throttle valve is isenthalpic.
(3) The condenser outlet state is saturated liquid.
(4) The evaporator outlet state is saturated vapor.
(5) Only physical exergies are used for the solar heat source and vapor flows.
(6) Kinetic, potential and chemical exergies of the substances are neglected.

For the analysis, the specifications of the combined RC, ERC,cycle are given in Table 1

Table 1: Main parameters considered for the analysis

Environment Temperature(^{0}C)	15
Turbine inlet pressure range (MPa)	4
Hot oil outlet temperature (^{0}C)	450(^{0}C)
Steam temperature(^{0}C)	400
Solar radiation received per unit area(Wm^{-2})	850
Apparent Sun temperature(K)	4500
Heliostat aperture area(m^2)	3000

Turbine back pressure range (MPa)	0.0074
Turbine isentropic efficiency (%)	85
Condenser temperature (^{0}C)	40
Hot oil mass flow rate (kg s^{-1})	8.0
Ejector evaporator temperature range (^{0}C)	4-10
Pump isentropic efficiency (%)	70
Nozzle efficiency (%)	90
Mixing chamber efficiency (%)	85
Diffuser efficiency (%)	85
First law efficiency of heliostat field (%)	75
First law efficiency of central receiver (%)	90
Second law efficiency of heliostat field (%)	75
Second law efficiency of central receiver (%)	30

Thermodynamic analysis: The basic principle of the model was introduced by Keenan et al. (1950) based on gas dynamics and formulated by Huang et al. (1999) and Ouzzane et al.(2003). The formulation and assumption of entrainment ratio is based on mass, momentum and energy equations which is recently developed by Dai et al.(2009) and may be reported as

$$\mu = \sqrt{\eta_n \eta_m \eta_d (h_{pf,n1} - h_{pf,n2,s'})/(h_{mf,d,s'} - h_{mf,m})} - 1 \tag{1}$$

The efficiencies of nozzle, mixing chamber, and diffuser are reported in Table 1 and the required enthalpy values at various state points of the ejector cycle for a given refrigerant are taken from REFPROP 6.01(1998). The ejector mainly consists of three section that is nozzle, mixing and diffuser section. In the nozzle section, the energy conservation equation for the adiabatic and steady primary flow is given as:

$$\dot{m}_{pf} h_{pf,n2} + \frac{\dot{m}_{pf} u^2_{pf,n2}}{2} = \dot{m}_{pf} h_{pf,n1} + \frac{\dot{m}_{pf} u^2_{pf,n1}}{2} \tag{2}$$

The nozzle efficiency may be defined as:

$$\eta_n = \frac{h_{pf,n1} - h_{pf,n2}}{h_{pf,n1} - h_{pf,n2,s'}} \tag{3}$$

In the mixing section, the momentum conservation equation is given as:

$$\dot{m}_{pf} u_{pf,n2} + \dot{m}_{sf} u_{sf,n2} = (\dot{m}_{pf} + \dot{m}_{sf}) u_{mf,m,s'} \tag{4}$$

In the diffuser section, the energy equation is given as:

$$\frac{1}{2}\left(u^2_{mf,m} - u^2_{mf,d,s'}\right) = h_{mf,d,s'} - h_{mf,m} \tag{5}$$

The diffuser efficiency is given as:

$$\eta_d = \frac{h_{mf,d,s'} - h_{mf,m}}{h_{mf,d} - h_{mf,m}} \tag{6}$$

Exergy analysis determines the system performance based on exergy, which is defined as the maximum possible reversible work obtainable in bringing the state of the system to equilibrium with that of the environment. In the absence of magnetic, surface tension effect, nuclear, electrical, and considering the system is at rest relative to the environment, the total exergy associated with the work obtainable by bringing a stream of matter from its initial state to a state that is in thermal and mechanical equilibrium with the environment. Mathematically,

$$\dot{E} = \dot{m}[(h - h_0) - T_0(s - s_0)]$$
(7)

According to Gouy-Stodola theorem, the exergy destruction and entropy generation are related as

$$\dot{E}_D = T_0 \dot{S}_{gen} \quad (8)$$

First law Efficiency (η_I): It can be defined as the ratio of the desired effect to the thermal energy of solar input $\dot{Q}_{solar}$.

The first law efficiency of the triple effect cooling cycle is given by

$$\eta_I = \frac{\dot{Q}_{E1} + \dot{W}}{\dot{Q}_{Solar}} \quad (9)$$

The basic equation obtained from the law of conservation of energy in the components of RC, and ERC are written as follows:

For Heliostat: A part of thermal energy received by heliostat is delivered to the central receiver and rest is lost to the environment

$$\dot{Q}_{Solar} = A_h q$$

Where, A_h and q are the aperture area and solar radiation per unit area

$$\dot{Q}_{Solar} = \dot{Q}_{CR} + \dot{Q}_{lost,heliostat} \quad (11)$$

$$\eta_{I,heliostat} = \frac{\dot{Q}_{CR}}{\dot{Q}_{Solar}} \quad (12)$$

For Central Receiver (CR): A part of thermal energy received by central receiver is absorbed by oil and rest is lost to the environment

$$\dot{Q}_{CR} = \dot{Q}_{oil} + \dot{Q}_{lost,CR} = \dot{m}_{oil}(h_1 - h_{12}) + \dot{Q}_{lost,CR} \quad (13)$$

So, $\eta_{I,CR} = \frac{\dot{Q}_{oil}}{\dot{Q}_{CR}}$ (14)

Second law Efficiency (η_{II}) : The amount of exergy supplied in the product to the amount of exergy associated with the fuel is more accurate measure of the thermodynamic performance of the system which is defined as the ratio of exergy contained in the product to the exergy associated with the solar energy input and the second law efficiency of triple effect refrigeration cycle may be reported as

$$\eta_{II} = \frac{\Delta\dot{E}_{E1} + \dot{W}}{\dot{E}_{Solar}} \quad (15)$$

Where, $\dot{E}_{Solar}$ is incoming exergy associate with solar radiation falling on heliostat, $\Delta\dot{E}_{E1}$ is the change in exergy at ejector evaporator of ERC, $\dot{W}$ is the work output.

$$\dot{E}_{Solar} = \dot{Q}_{Solar}\left(1 - \frac{T_0}{T_S}\right) \quad (16)$$

Where, T_s the apparent sun temperature = 4500K
For Heliostat: A part of exergy received by heliostat is delivered to the central receiver and rest is lost to the environment (irreversibility),

$$\dot{E}_{Solar} = \dot{E}_{CR} + \dot{E}_{lost,heliostat} \tag{17}$$

So, $$\eta_{II,heliostat} = \frac{\dot{E}_{CR}}{\dot{E}_{Solar}} \tag{18}$$

For Central Receiver (CR): A part of exergy received by central receiver is absorbed by oil and rest is lost to the environment

$$\dot{E}_{CR} = \dot{E}_{oil} + \dot{E}_{lost,CR} \tag{19}$$

So, $$\eta_{II,CR} = \frac{\dot{E}_{oil}}{\dot{E}_{CR}} \tag{20}$$

II. Results and discussion

Energy and exergy analysis has been done to find out the effect of some influenced parameters on the performance of the solar driven combined power and refrigeration system. Following parameters have been chosen in the typical range of its operation; Motive steam pressure 4-7 MPa, evaporator temperature of ERC(4^0C -10^0C). The parameter under consideration is varied over a given typical range while values of other parameters are kept constant at the level of base case values.
Fig.2 shows the variation of the first and second law efficiency of combined cycle with the change in evaporator temperature.

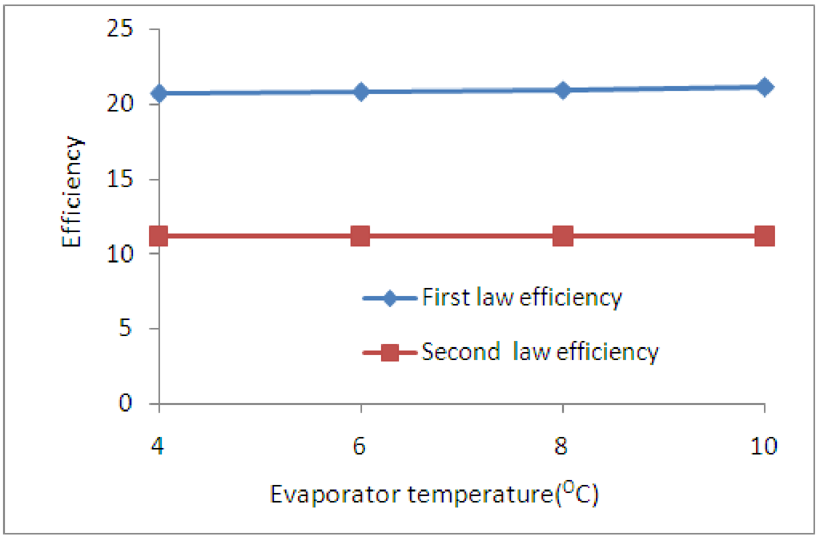

Figure 2: Variation of first and second law efficiency with the change in Evaporator temperature

And Fig.3 shows the variation of the first and second law efficiency of combined cycle with the change in motive steam pressure

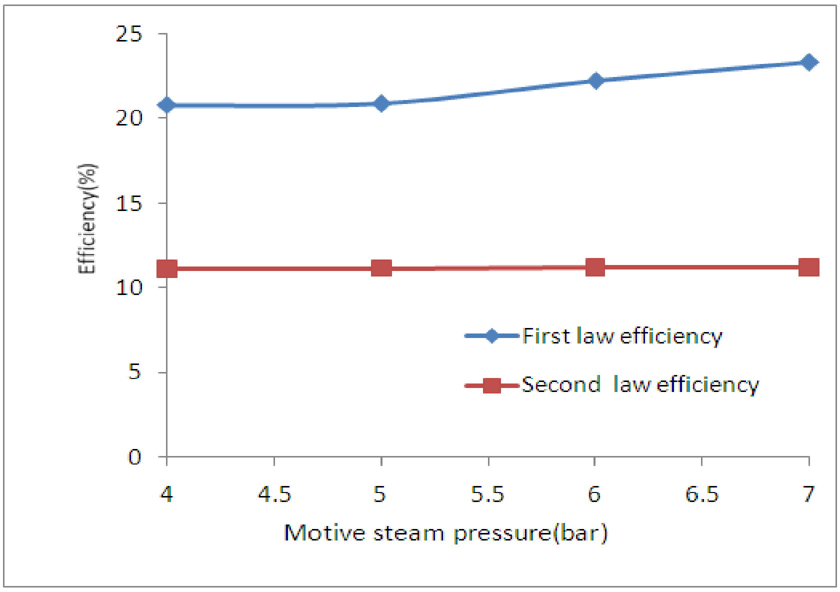

Figure 3: Variation of first and second law efficiency with the change in motive steam pressure

It is observed that first and second law efficiency increases with the increase in evaporator temperature and motive steam pressure. This is due to the fact that the refrigeration capacity of the cooling increases with the increase in evaporator temperature. The power produced by the turbine remains same so total effect of this is to increase the first and second law efficiency.

III. Conclusion

New solar driven combined power and cooling cycle is proposed for the production of cooling in the range of 4^0C to 10^0C. Energy and exergy methods are employed which enable us to develop a systematic approach that can be used to identify the sites of the real destructions/losses of valuable energy in thermal devices. The effect of design parameters were observed on energy and exergy performance of the proposed cycle. The conclusions of the present analysis can be summarized as follows:

- Energy efficiency of the cycle changes from about 20.5% to 21% with the change in evaporator temperature
- Exergy efficiency of the cycle changes from about 11.13% to 11.14% with the change in evaporator temperature
- Energy efficiency of the cycle changes from about 20.79% to 23.33% with the change in motive steam pressure
- Exergy efficiency of the cycle changes from about 11.14% to 11.22% with the change in motive steam pressure

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