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Exergy and Parametric Analyses of a Solar Driven Ejector Cooling and Power Cycle Operating with Butane as a Refrigerant

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Abstract—*The proposed cycle is a combination of ejector refrigeration cycle and organic Rankine cycle to produce cooling and power simultaneously by using butane as a refrigerant and single source of solar heat. The exergy destruction rate in each component of the proposed cycle is determined to identify the components having largest exergy destruction. The effects of key thermodynamic parameters like; turbine inlet pressure, turbine back pressure, ejector evaporator temperature and condenser temperature on exergy destruction rate in each components of the cycle are determined. It is observed that only 6.68% of the input exergy is available as exergetic refrigeration and power output and remaining is destroyed due to irreversibilities in the various components of cooling and power cycle. The energy efficiency is found to be 11.02%. The maximum exergy destruction takes place in central receiver followed by heliostat, heat recovery vapour generator, condenser and ejector. It is found that the total exergy destruction in the combined cycle decreases with increase in turbine inlet pressure. The increasing turbine back pressure makes the exergy destruction increasing in the ejector, condenser and evaporator and decreasing in the turbine. The total exergy destruction in the cycle increases with increase in the turbine back pressure. The variation of evaporator temperature has little effect on the exergy destruction for each component of the cycle. With the variation of condenser temperature influences exergy destruction in the condenser drastically. The exergy destruction in HRVG, ejector and evaporator decreases while in condenser increases with increase in condenser temperature. Due to combined effect, the total exergy destruction in the cycle increases with increase in the condenser temperature.*

Keywords: *ejector refrigeration system, exergy, R600, performance, exergy destruction rate.*

1. INTRODUCTION

To overcome the twin menace of electricity consumption and environmental degradation; like ozone depletion and global warming while using vapour compression refrigeration systems, an imperative need is felt to search out the solar thermal driven refrigeration and air conditioning systems, which utilize the refrigerants having lesser ozone depletion potential and global warming potential.

In view of this, various investigations have been carried out on solar heat and non CFCs operated refrigeration systems like ejector and vapour absorption cooling. Al-Khalidy [1] analyzed the theoretical and experimental performance of a solar-driven ejector refrigeration system. Five refrigerants (R717, R12, R11, R113, and R114) were compared. It was found that R113 was more suitable then any other refrigerant. Huang et al. [2] have developed a solar ejector cooling system using R141b as the refrigerant. Subsequently, more environmental friendly refrigerants were suggested e.g. the hydrocarbon refrigerants by Pridasawas and Lundqvist [3]. Several simulation models and experimental studies are found in the literature for example by Dorantes et al. [4] and Sokolov and Hershgal [5]. Cizungu et al. [6] studied systems with several environmentally friendly working fluids such as R123, R134a, R152a and ammonia. Sun [7] studied 11 refrigerants including water, halocarbon compounds, a cyclic organic compound and an azeotrope. Dai et al. [8] carried out exergy analysis, parametric analysis and optimization for a novel combined power and ejector refrigeration cycle.

In the present study, exergy method along with energy method is applied to investigate the effect of various operating parameters on exergy destruction rates in each component of the proposed ejector cooling and power cycle.

2. SYSTEM DESCRIPTION

The proposed cogeneration cycle combines the Rankine cycle (RC) and the ejector refrigeration cycle (ERC) as shown in Fig. 1. Solar energy falls on the heliostat field and reflected on the aperture area of central receiver which is located at the top of the tower. The concentrated rays which falls on to the central receiver results in higher temperature of the central receiver, is used to heat the oil (Duratherm600). The oil flows through the pipes which transfer the thermal energy from central receiver to the HRVG (1-2).

Superheated refrigerant vapour (4) is expanded in a turbine to generate power. The turbine exhaust (5) passes through converging diverging supersonic nozzle of ejector. The very high velocity refrigerant vapour at the exit of the nozzle creates a very high vacuum at the inlet of the mixing chamber and extract secondary vapour (11) from the evaporator of ERC into the mixing chamber and this causes cooling effect at evaporator-1(E1) of ERC. The primary vapour (5) and secondary vapour (11) are mixed in the mixing chamber. The mixed stream (6) is condensed in the condenser-1(C1). The saturated liquid (7) is divided in to two parts (8, 9), one part (9) is passed through throttling valve-1 (TV1) where pressure is reduced to evaporator (E-1) pressure (10) and feed to evaporator-1 (E1), and second part (8) is pumped by pump-1(P1) to the HRVG of RC cycle. The stream (2) coming out from HRVG enters in to the central receiver.

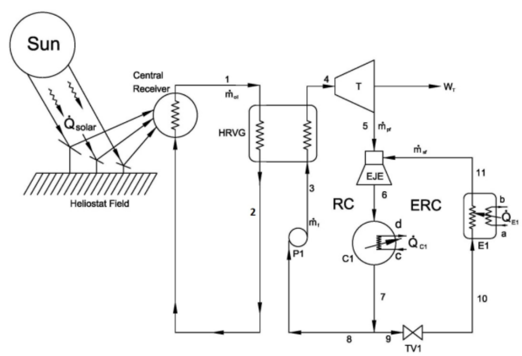

Fig. 1: Schematic diagram of solar assisted ejector cooling and power cycle

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

Table 1: Main parameters considered for the analysis

Parameters	R600
Environment temperature (0C)	15
Environment pressure (MPa)	0.10135
Turbine inlet pressure range (MPa)	1.0-2.5
Hot oil outlet temperature from CR (0C)	170
Solar radiation received per unit area (Wm-2)	850
Apparent Sun temperature (K)	4500
Heliostat aperture area (m2)	3000
Turbine back pressure range (MPa)	0.35-0.65
Turbine isentropic efficiency (%)	85
Ejector evaporator temperature range (K)	264-272
Condenser temperature range (0C)	20-30
Pump isentropic efficiency (%)	70
HRVG efficiency (%)	100
Pinch point temperature difference (0C)	10.0
Approach point temperature difference (0C)	5.0
Nozzle efficiency (%)	90
Mixing chamber efficiency (%)	85
Diffuser efficiency (%)	85
Energy efficiency of heliostat field (%)	75
Energy efficiency of central receiver (%)	90
Exergy efficiency of heliostat field (%)	75
Exergy efficiency of central receiver (%)	30

3. THERMODYNAMIC ANALYSIS

Exergy is defined as the maximum amount of work which can be produced by a system when it comes to equilibrium with a reference environment, which may be defined mathematically as:

$$\dot{E} = \dot{m}[(h - h_0) - T_0(s - s_0)] \tag{1}$$

The entropy generation over a control volume is given by Bejan [9]

$$\dot{S}_{gen} = \frac{dS}{dt} - \sum_{i=o}^{n} \frac{\dot{Q}_i}{T_i} - \sum_{in} \dot{m}s + \sum_{out} \dot{m}s \geq 0 \tag{2}$$

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

$$\dot{E}_D = T_0 \dot{S}_{gen} \tag{3}$$

3.1. Energy Efficiency (η_{energy})

The energy efficiency of the proposed cycle for simultaneous production of cooling and power is given by

$$\eta_{energy} = \frac{\dot{Q}_{E1} + \dot{W}_{NET}}{\dot{Q}_{Solar}} \tag{4}$$

where, $\dot{Q}_{E1}$, $\dot{W}_{NET}$ and $\dot{Q}_{solar}$ are the refrigeration output of ERC, net power output of Rankine cycle and solar heat input respectively which can be calculated as:

For Turbine (T):

$$\dot{W}_T = \dot{m}_f (h_4 - h_5) \tag{5}$$

For Pump (P1):

$$\dot{W}_{P1} = \dot{m}_f (h_3 - h_8) \tag{6}$$

Net Work done:

$$\dot{W}_{NET} = \dot{W}_T - \dot{W}_{P1} \tag{7}$$

$$\dot{Q}_{E1} = \dot{m}_E (h_a - h_b) = \dot{m}_{sf} (h_{11} - h_{10}) \tag{8}$$

3.2. Exergy Efficiency (η_{exergy})

The exergy efficiency for simultaneous production of cooling and power cycle is given by

$$\eta_{exergy} = \frac{\Delta \dot{E}_{E1} + \dot{W}_{NET}}{\dot{E}_{Solar}} \quad (9)$$

where, $\dot{E}_{Solar}$ is incoming exergy associated with solar radiation falling on heliostat, $\Delta\dot{E}_{E1}$ and $\dot{W}_{NET}$ are the change in exergy at ejector evaporator of ERC and exergy associated with net power output of RC respectively.

$$\dot{E}_{Solar} = \dot{Q}_{Solar}\left(1-\frac{T_0}{T_S}\right) \quad (10)$$

where, T_s = Apparent Sun temperature (K)=4500K

$$\Delta\dot{E}_{E1} = \dot{m}_{sf}[(h_{10}-h_{11})-T_0(s_{10}-s_{11})] \quad (11)$$

The basic equations of exergy destruction rate in the components of RC and ERC are written as follows:

For HRVG

$$\dot{E}_{D,HRVG} = T_0[\dot{m}_{oil}(s_2-s_1)+\dot{m}_{pf}(s_4-s_3)] \quad (12)$$

For Turbine (T)

$$\dot{E}_{D,T} = \dot{m}_{pf}[(h_{s4}-h_{s5})-T_0(s_4-s_5)]-\dot{W}_T \quad (13)$$

For Pump (P)

$$\dot{E}_{D,P} = T_0[\dot{m}_{pf}(s_3-s_8)] \quad (14)$$

For Ejector (EJE)

$$\dot{E}_{D,EJE} = T_0[(\dot{m}_{pf}+\dot{m}_{sf})s_6-\dot{m}_{pf}(s_5)-\dot{m}_{sf}(s_{11})] \quad (15)$$

For Condenser (C)

$$\dot{E}_{D,C} = T_0(\dot{m}_{pf}+\dot{m}_{sf})(s_7-s_6) \quad (16)$$

For Throttle Valve (TV)

$$\dot{E}_{D,TV} = T_0\,\dot{m}_{sf}(s_{10}-s_9) \quad (17)$$

For Evaporator

$$\dot{E}_{D,E} = T_0[\dot{m}_{sf}(s_{11}-s_{10})+\dot{m}_w(s_b-s_a)] \quad (18)$$

4. RESULTS AND DISCUSSION

In this paper, parametric analysis and exergy analysis are carried out to find out the effect of variation of turbine inlet pressure, turbine back pressure, condenser temperature and evaporator temperature on the performance and exergy destruction rates in each component of the solar assisted cogeneration cycle for simultaneous production of power and cooling. The thermodynamic properties of refrigerant R600 were calculated by NIST Standard Reference Database 23 [10].

Table 2 shows that out of 100% solar heat energy supplied to the system, around 11.02% is available as useful refrigeration output in ERC and net power output in RC cycle and the rest of the energy which includes the heat rejected at the condenser of ERC is lost to the environment.

Table 2: Percentage (%) of Sun's energy distribution in the cycle

Energy input/output	Amount (kW)	%
Energy input from Sun into system	2550	100
Turbine work	160.5	6.29
Pump work	10.72	0.42
Net power output	149.8	5.87
Refrigeration output of ERC	131.3	5.15
Energy efficiency		11.02

Table 3: Percentage (%) of Sun's exergy distribution in the cycle

Exergy inputs/outputs and destructions	Amount (kW)	%
Exergy input	2387	100
Exergy output of ERC	9.81	0.41
Exergy of net power output	149.8	6.27
Exergy efficiency		6.68
Exergy destruction in HRVG	127.4	5.33
Exergy destruction in condenser	118.6	4.96
Exergy destruction in ejector	79.64	3.33
Exergy destruction in turbine	25.06	1.05
Exergy destruction in throttle valve	6.44	0.27
Exergy destruction in pump	7.87	0.33
Exergy destruction in evaporator	12.17	0.51
Exergy destruction in Heliostat field	596.75	25
Exergy destruction in central receiver	1253.17	52.5

Table 3 indicates that out of 100% solar exergy, around 0.41% is produced as exergy output of refrigeration in ERC and around 6.27% is produced as exergy output (net work output) of RC cycle. The rest of the exergy is destroyed due to irreversibilities in various components of the cogeneration cycle. In this context, it is indicated that majority of the irreversibility occurred in central receiver (52.5%), heliostat field (25%), HRVG (5.33%), condenser (4.96%), ejector of ERC (3.33%). The results indicate that the exergy efficiency (6.68%) is very less as compared the energy efficiency (11.02%). The component like central receiver, Heliostat field, HRVG, condenser and ejector of ERC where maximum exergy destroyed needs special attention from second law point of view. A reduction in the irreversibility in these components will improve the overall performance of the cycle.

Fig. 2 indicate that by increasing turbine inlet pressure results in sharp reduction of exergy destruction in the HRVG and condenser due to the reduction in the heat transfer temperature difference. The total exergy destruction in the combined cycle decreases with increase in turbine inlet pressure.

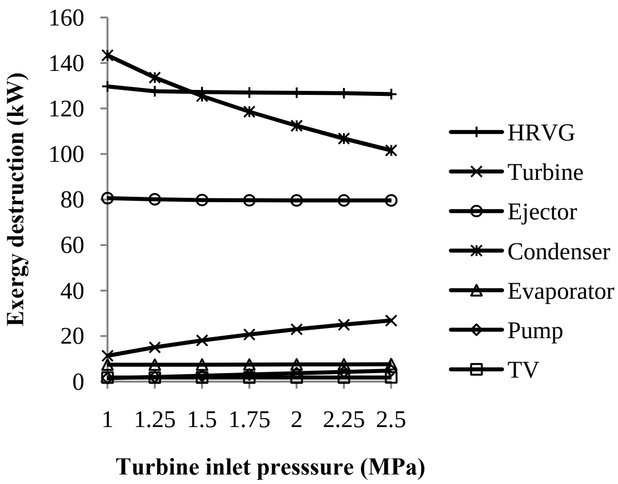

Fig. 2: Effect of turbine inlet pressure on exergy destruction in each component of the combined cycle

The increasing turbine back pressure makes the exergy destruction increasing in the ejector, condenser and evaporator and decreasing in the turbine as shown in Fig. 3. The total exergy destruction in the cycle increases with increase in the turbine back pressure.

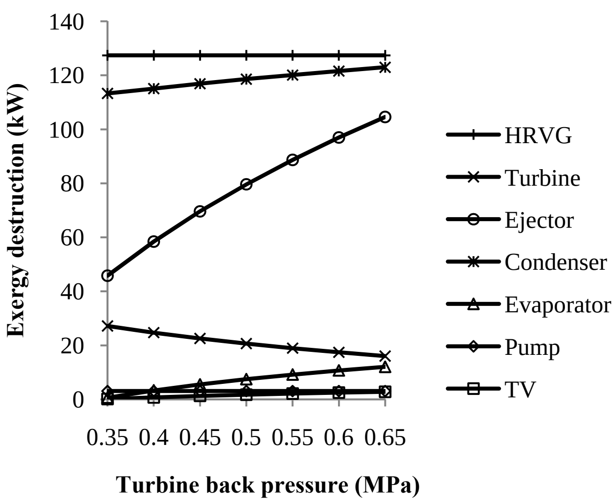

Fig. 3: Effect of turbine back pressure on exergy destruction in each component of the combined cycle

The variation of evaporator temperature has little effect on the exergy destruction for each component of the cycle as indicated in the Fig. 4. Total exergy destruction decreases marginally due to decrease in the exergy destruction in ejector and evaporator with increase in evaporator temperature.

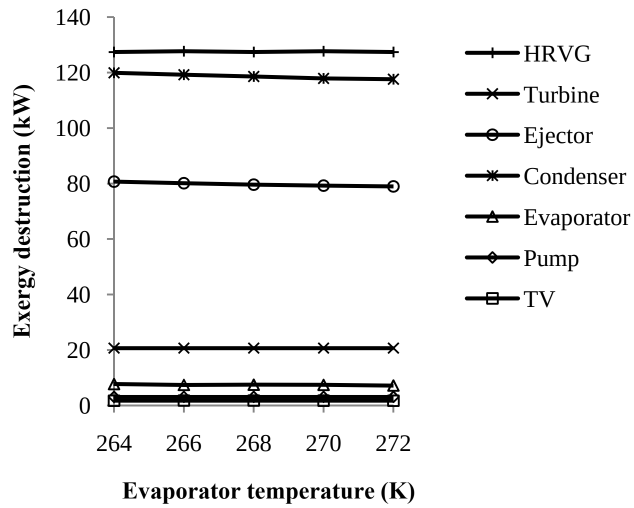

Fig. 4 Effect of evaporator temperature on exergy destruction in each component of the combined cycle

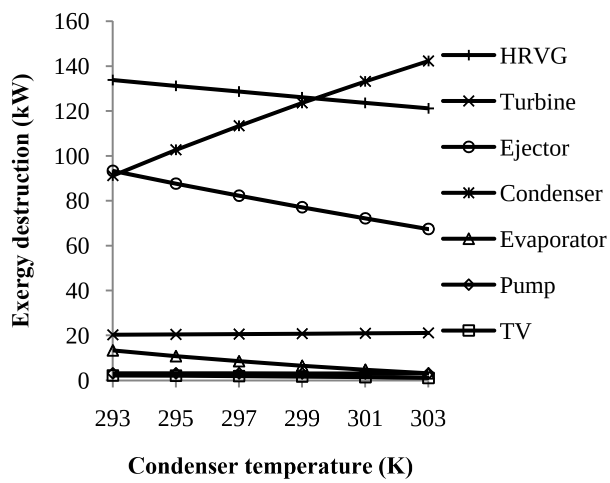

Fig. 5: Effect of condenser temperature on exergy destruction in each component of the combined cycle

Fig. 5 clearly indicates that the variation of condenser temperature influences exergy destruction in the condenser drastically because the heat transfer temperature difference in the condenser increases with increasing condenser temperature. The exergy destruction in HRVG, ejector and evaporator decreases while in condenser increases with increase in condenser temperature. Due to combined effect, the total exergy destruction in the cycle increases with increase in the condenser temperature.

5. CONCLUSION

From the discussion above, it can be concluded that the amounts of exergy loss in the central receiver, heliostat field, HRVG, ejector and condenser account for large percentage through exergy analysis. Therefore, it is significant to employ methods for reducing exergy losses of these components, such as increasing the area of heat transfer and the coefficient of heat transfer in the HRVG and condenser, optimization design and manufacture in the ejector and turbine. Thus the performance for this combined cycle could be improved greatly. The effect of variation of various key thermodynamic parameters on exergy destruction rate are summarized below:

- The total exergy destruction in the combined cycle decreases with increase in turbine inlet pressure.
- The total exergy destruction in the cycle increases with increase in the turbine back pressure.
- Total exergy destruction decreases marginally due to decrease in the exergy destruction in ejector and evaporator with increase in evaporator temperature.
- The total exergy destruction in the cycle increases with increase in the condenser temperature.

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