A Feasibility Study of Carbon-dioxide Based Rankine Cycle Powered by the Linear Fresnel Reflector Solar Concentrator System

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ABSTRACT

Theoretical analysis of a linear Fresnel reflector solar concentrator powered Rankine thermodynamic cycle utilizing supercritical CO_2 as a working fluid is presented. The system model consists of a linear Fresnel reflector solar concentrator with trapezoidal cavity absorber, a power generating turbine, a heat recovery system and a feed pump. The effects of the principal parameters of the supercritical CO_2 on the performance of the system are investigated numerically by means of MATLAB simulation program under the assumed design conditions. It is shown that the key performance parameters, such as concentrator area, concentrated power reached to the absorber, CO_2 flow rate have significant effects on the thermal performance of the supercritical CO_2 in the trapezoidal cavity absorber. Analytical simulations show that the proposed system may have 0.3-0.38 kW power generation and 2.0-2.14 kW heat output for the various mass flow rates of the CO_2. The results recommend the potential of this new system for applications to electricity power and heat power generation.

Keywords: Linear Fresnel reflector solar concentrator, supercritical carbon dioxide, Rankine cycle, power generation and heat output.

Nomenclature

A_a Solar concentrator area [m^2]
A_{HE} Heat exchanging area of the condenser [m^2]
A_p Absorber plate area [m^2]
CP Total concentrated power on the tubular absorber [W]
c Specific heat of the CO_2 [kJ/kg °C]
D_e Distance between the absorber surface and transparent cover [mm]
d_o Outer diameter of tubular absorber [m]
d_i Inner diameter of tubular absorber [m]

F'	Concentrator efficiency factor
f	Height of tubular absorber from the reflector frame [m]
$h_{p\text{-}c}$	Heat loss coefficient from the absorber surface [W/m^2-K]
h_1	Specific enthalpy at turbine inlet [kJ/kg]
h_2	Specific enthalpy at turbine outlet [kJ/kg]
h_3	Specific enthalpy at condenser outlet [kJ/kg]
h_4	Specific enthalpy at pump outlet [kJ/kg]
I_b	Direct component of solar flux [kW/m^2]
k_a	Thermal conductivity of air [W/m^2-K]
L	Length of the tubular absorber [m]
m	mass flow rate of the CO_2 fluid [kg/s]
m_w	mass flow rate of the water in the condenser [kg/s]
N	Number of tubes in the cavity
Nu	Nusselt number
n	Reflector position number on either side of the central reflector of the concentrator, n for the central reflector being zero.
P	Pitch between the tubes [m]
P_1	Pump outlet pressure [kN/m^2]
P_2	Pump inlet pressure [kN/m^2]
Q	Total heat gain [W]
Q_R	Heat rejected in the condenser [W]
Ra	Rayleigh number
q_u	Useful heat gain [W]
R_n	Distance of the left edge of the nth reflector on the right side of the concentrator from the concentrator axis [m]
S	Absorbed flux [W/m^2]
s_n	Distance between the nth and (n - 1)th reflector [m]
T_a	Ambient temperature [^{o}C]
T_f	Fluid inlet temperature in the control volume [^{o}C]
T_p	Absorber plate temperature [^{o}C]
T_{fi}	Inlet temperature of fluid in the absorber tube for one-dimensional analysis [^{o}C]
T_{fo}	Outlet temperature of fluid in the absorber tube for one-dimensional analysis [^{o}C]
T_1	Absorber outlet or turbine inlet temperature of fluid [^{o}C]
T_2	Turbine outlet temperature of fluid [^{o}C]
T_3	Condenser outlet temperature of fluid [^{o}C]
T_4	Pump outlet or absorber inlet temperature of fluid [^{o}C]
T_5	Inlet water temperature in the condenser [^{o}C]
U_l	Overall heat loss coefficient [W/m^2 ^{o}K]
W_T	Turbine work [kW]
W_p	Pump work [kW]

w Width of the reflector [m]
z constant used to find Nusselt number

Greek Symbols

θ_n Tilt of the nth reflector (degrees)
ε Half of the angular subtense of the sun at any point on the earth
ρ Density of working fluid (kg/m^3)
η_{heat} Heat recovery efficiency (%)
η_p Efficiency of the pump (%)
η_t Efficiency of the turbine (%)
η_{th} Thermal efficiency (%)

INTRODUCTION

In recent years, accelerated consumption of fossil fuels has caused many serious environmental problems such as global warming, ozone layer destruction and atmospheric pollution. New energy conversion technologies are required to utilize energy resources suitable for power generation without causing environmental pollution. Low-grade heat sources are considered as candidates for the new energy sources. Solar heat, waste heat and geothermal energy are typical examples for low-grade heat sources with their available temperatures ranging between 60 and 200°C. The use of such low-grade heat sources as an alternative energy source generating electricity has long been investigated using power turbine cycles [1]. Chen et al. [2] examined the performance of the CO_2 trans-critical power cycle utilizing energy from low-grade waste heat in comparison to an Organic Rankine cycle (ORC) using R123 as working fluid. They found that when utilizing the low grade heat source with equal mean thermodynamic heat rejection temperature, the carbon dioxide trans-critical power cycle had a slightly higher power output than the ORC. A thermodynamic cycle powered by solar energy for both power and heat generation using supercritical carbon dioxide as a working fluid on theoretical aspect were investigated by Zhang et al. [3-4], and they examined the effects of various design conditions and climate conditions on the performances of this CO_2-based Rankine cycle. They also set up an experimental system to validate the feasibility of this supercritical carbon dioxide cycle [5-8]. Cayer et al. [9] expressed a detailed analysis of a carbon dioxide trans-critical power cycle using an industrial low-grade stream of process gases as its heat source.

The present solar concentrator is an in-line focus linear concentrator with a modular design concept. It uses a bank of parallel mirror rows, each row 1m long, to focus solar radiation onto a long absorber positioned above the mirrors (Figure 1). The absorber consists of a number of closely-packed pipes mounted at the top of a downward-facing trapezoidal cavity. The cavity is insulated above and covered with a glass window on the bottom. The interest on LFRSC started around 43 years ago, when Francia [10] studied an LFR installed in Genoa (Italy) in 1963. In 2000, Mills and Morrison [11] developed the Australian compact linear Fresnel reflector (CLFR) concept which uses water as a working fluid. Studies and installation of LFRSC were conducted by Solarmundo group of Belgium and Germany [12]. More recently, the new AUSRA, U.S. Company installed LFRs in California [13]. The cavity absorber can reduce the heat loss because it is covered and insulated from surroundings. Young shuai et al. [14] suggested use of an upside-down cavity receiver in view of directional attributes of focal flux.

Until now, none of the published studies on the supercritical CO_2 power cycle focused on the parameter optimization and thermodynamic analysis to convert solar energy to useful work by using linear Fresnel reflector solar concentrator (LFRSC). So, the aim of this study is to conduct the parameter analysis for the supercritical CO_2 power cycle to harness solar energy by using LFRSC. In this work, a new type of environmentally friendly system called an Organic Rankine Cycle (ORC) powered by LFRSC, which uses carbon dioxide as a working fluid is theoretically analyzed to utilize the low temperature heat source. The working fluid is heated at the cavity absorber by using a solar reflector and is converted into high pressure vapor. The vapor is changed to low pressure vapor when it is passed through the turbine. The thermal energy of the high pressure vapor is converted into mechanical energy. The resultant mechanical energy is converted to electricity by using the generator. The low pressure vapor coming out of the turbine is passed to the heat exchanger. In the heat exchanger, the water is supplied to gain the heat which is rejected by the carbon dioxide. Again the low pressure and low temperature vapor is pumped back to cavity absorber. Thus the ORC is utilized to generate both electrical output and heat output. Therefore, in this paper, from the theoretical stand point, a feasibility study of CO_2-based Rankine cycle powered by LFRSC, in which both renewable energy and ecologically safe fluid are used to form a cogeneration system of heat and power with environmental preservation, is presented.

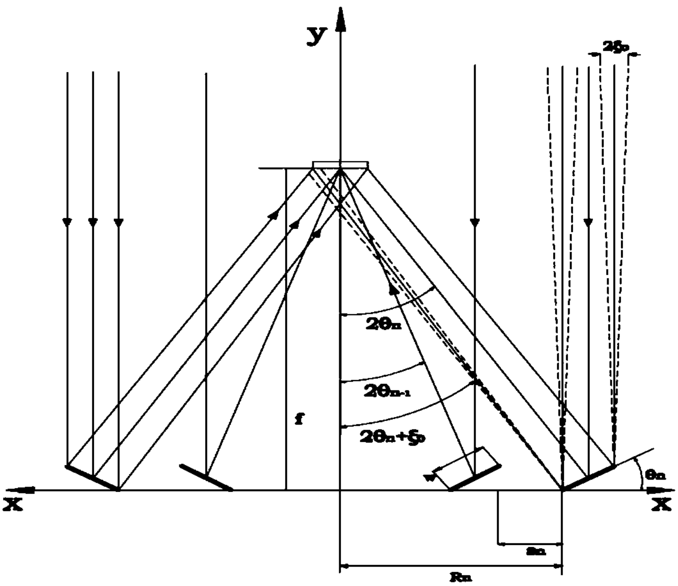


Figure 1: Schematic view of a linear Fresnel reflector solar concentrator system

SYSTEM DESCRIPTION

Linear Fresnel reflector solar concentrator with cavity absorber

The proposed LFRSC (Figure 1) system consists of 40 reflecting glass mirrors on each side of the trapezoidal cavity absorber [15], each having 1 m length and 40 mm width, fixed onto a pipe frame. Each mirror is assumed to be fixed separately on the frame and they could be tilted individually at any angle (θ_n), to adjust the angle of reflected light towards the focus. The distance between the mirror elements is called as shift (s_n) and the distance from the center of the concentrator plane to the edge of the n^{th} mirror is R_n. The reflecting unit was kept over the supporting structure which is mounted on self aligning ball bearings to facilitate tracking. The mirror frame is assumed with a telescopic pipe arrangement to keep the whole reflecting unit at the required slope so that sun rays could fall perpendicular to the absorber plane [16]. The cross-sectional view of the trapezoidal cavity absorber with round pipe covered with glass is shown in Figure 2. Absorber tube is assumed to be made of a set of six copper round tubes of 15.8 mm outside diameter, 12.7 mm inside diameter and 1m length. All the tubes are assumed to be brazed together without gap and placed above the copper absorber plate of width 96 mm. Sidewalls of the cavity were provided with 5 mm thick ceramic tile plates. The cavity is assumed to have the depth of 45 mm and the side plate has an inclination of 50° angle. Glass wool insulation (thickness of 35 mm) was provided at the upper top and sides of the absorber tube to reduce heat loss (17). A plane glass (4 mm thick and 196 mm width) was provided at the bottom to allow reflected solar radiation.

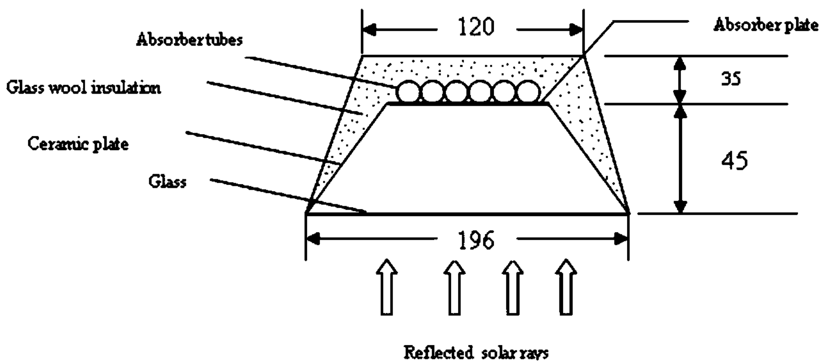


Figure 2: Cross-sectional detail of trapezoidal cavity absorber

CO_2-based Rankine cycle powered by LFRSC

Figure 3 shows a schematic diagram of the CO_2-based Rankine cycle. A LFRSC is used to heat CO_2 contained in the cavity absorber. The heating in the concentrator makes supercritical CO_2 high temperature state (Figure 3, state 1). The high temperature supercritical CO_2 drives the turbine of the Rankine system and power output can be available from the turbine generator. The lower pressure carbon dioxide, which is expelled from turbine, is cooled in the heat recovery system. At the outlet of turbine, supercritical CO_2 still has a higher temperature (Figure 3, state 2), which can be utilized to provide heat source for boiling water, which can be achieved in the heat recovery system. The heat recovery system is actually a heat exchanger. After leaving the heat recovery system (Figure 3, state 3), CO_2 is pumped by the feed pump, back into the higher pressure condition (Figure 3, state 4), and then the cycle recommences.

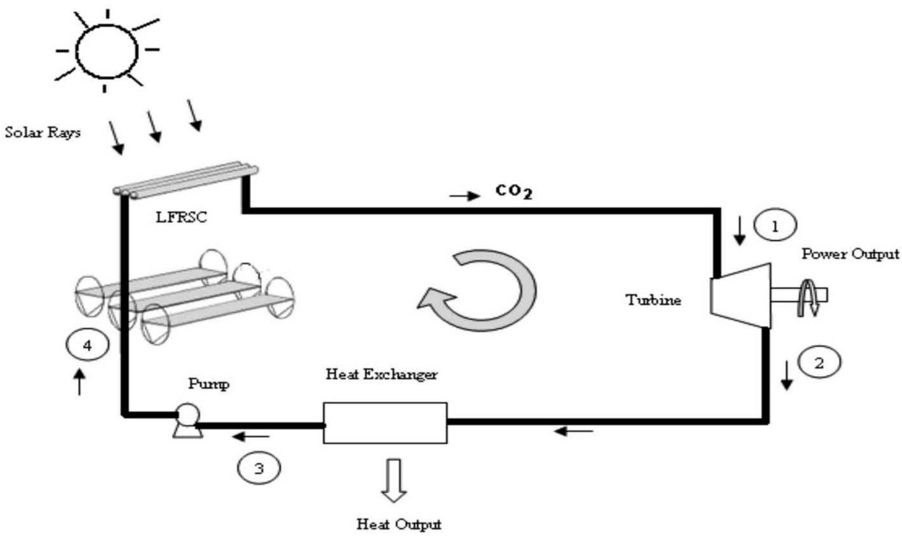


Figure 3: A CO_2-based Rankine cycle powered by LFRSC for power generation and heat recovery

The LFRSC is the heart of the CO_2-based Rankine cycle system. Its characteristics play an important role in the successful operation of such systems. The CO_2 temperature at the outlet of the solar concentrator is an important performance parameter in achieving a good heat collection and power generation from the Rankine cycle. A higher temperature of working fluid in the cycle will make it easy to collect heat from the cycle, and a higher pressure of working fluid is helpful to drive turbine in producing electric power. In this paper, as a first step of understanding the performance of the LFRSC in CO_2-based Rankine cycle, efforts were made to measure the cycle temperatures, pressures, enthalpy, turbine output and heat recovery etc,. for various mass flow rates of CO_2. It can be used to study the feasibility of the LFRSC system powered Rankine cycle using carbon dioxide.

THERMODYNAMIC ANALYSIS OF THE RANKINE CYCLE

The use of CO_2 as a working fluid for a solar energy-powered Rankine cycle provides a new field of interest for process modeling and simulation. As the first step towards a fundamental understanding and estimation of the performance and characteristics of the system, a mathematical model that simulates the system behavior of the Rankine cycle considering a steady state was constructed. The thermodynamic properties [3], illustrated in Figure 4, indicate critical data and phase envelopes of CO_2 in pressure–enthalpy coordinates. This heat transfer process in the solar concentrator is above the critical point resulting in a transcritical cycle, i.e. with a subcritical low-side and a supercritical high-side pressure. In these proposed transcritical cycle conditions, developments of temperature and pressure of CO_2 were sometimes close to the critical point of CO_2. With the available thermodynamic properties data, the components of a system based on the proposed cycle have been modeled.

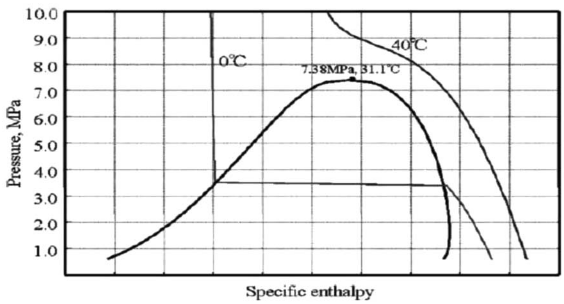


Figure 4: Pressure-enthalpy diagram of carbon dioxide

System simulation

The thermodynamic state conditions of the proposed combined cycle (Figure 3) are evaluated under the processes associated with the flow and heat transfer in the solar concentrator and expansion processes in the turbine and so on with the following assumptions:

1. In the cavity absorber, the working fluid (supercritical CO_2) is distributed uniformly in all heat removal tubes.
2. In the cavity absorber, thermal resistance along the absorber tubes is neglected. The temperatures of the absorber tubes are considered the same.
3. The solar concentrator is placed facing south and exposed to solar radiation throughout the day; it is also kept reasonably free from dust.
4. At point 4 (Figure 3, at the pump outlet), the pressure is 9.0 MPa.
5. The pressure drop in the turbine has a constant value of 4.5 MPa.
6. The tubes are assumed to be connected by header and footer pipes and the length of the tubes (six numbers) is considered as 1m with single pass of the working fluid.

7. The total amount of CO_2 of about 6.0 kg was assumed to be charged into the CO_2 loop of the theoretical analysis.

Although the assumptions and simplifications limit the usefulness of this analysis, the results show the potential of using the supercritical CO_2 as the working fluid in the Rankine cycle powered by LFRSC.

Concentrated Power at the Cavity Absorber Tubes

In the Fresnel concentrator-tubular collector system, different reflector elements, reflect energy to different portions of the absorber plate and the concentrated flux from all the reflectors distributes large on the absorber plate. The heating of CO_2 and its temperature rise solely depends on the concentrated power reached to the absorber plate. So, it is more meaningful to determine the total concentrated power reaching the absorber plate and then, the obtained value is used for further heat transfer analysis. Solar power (P_n) reached to the absorber which is contributed from the n^{th} reflector is given by [15],

$$P_n = \rho\, I_b\, w\, \mathrm{Cos}\, \theta_n\, L \tag{1}$$

where, ρ is the reflectivity of the reflector and I_b represents the intensity of beam radiation which are assumed as 0.98 and 800 W/m^2 respectively in the present study. Thus, the total concentrated power on the absorber due to the contributions from all the reflector elements is given by [15],

$$CP = (2\sum_{n=1}^{N} P_n) \tag{2}$$

Mathematical model

In this section, a mathematical model that describes the processes in each component of the system is presented.

Theoretical analysis of the LFRSC system (4-1)

The generalized thermal analysis of a concentrating solar collector is similar to that of a flat plate collector. It is necessary to derive appropriate expression for the loss coefficient U_l, considering the heat loss between the absorber plate and the transparent bottom glass cover (depth of the cavity absorber) and neglecting the side plate loss. For the estimation of heat loss coefficient, standard heat transfer relations for glazed surfaces have been used for the present work.

The LFRSC can be imagined as a broken-up parabolic trough reflector and thermal analysis was carried out as similar to the parabolic trough reflector [15]. The absorber plate is assumed to be made of thin copper sheet with thickness (∂) of 0.15 mm, while the tubes are assumed to be soldered on the top of the absorber plate. The solar thermal performance of a multi tubular absorber is assumed as employing selective surface coated absorber tube whose absorbance (α) is assumed as 0.96 [16] has an outer diameter d_o and inner diameter d_i. The working fluid CO_2, which is to be heated in the absorber has a mass flow rate m, specific heat c, an inlet temperature T_4, an outlet temperature T_1 and ambient temperature is T_a (assumed as 35°C). The cover which is placed below the absorber has been made of a material which is highly transparent to incoming reflected solar radiation and at the same time, opaque to long wavelength re-radiation emitted by the absorber plate. Glass with low ferric oxide content satisfies these requirements.

The final one-dimensional analysis has been performed along the direction of fluid flow with the objective of determining the variation of fluid temperature. This analysis will help in linking the useful heat gain rate with the fluid inlet temperature. Consider the control volume, an elementary length dy of one tube (Figure 5). Apply the first law of thermodynamics (18),

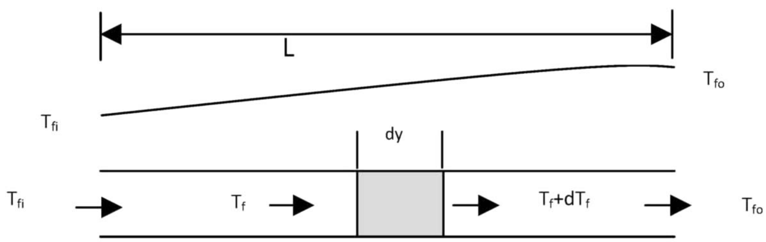


Figure 5: Variation of fluid temperature in flow direction

Rate of change of enthalpy of the fluid flowing through the control volume is equal to rate of heat transfer to fluid inside the control volume.

Thus,

$$\left(\frac{m}{N}\right)cdT_f = \frac{1}{N}\,dq_u = PF'\left[S - U_l\left(T_f - T_a\right)\right]\,dy \tag{3}$$

$$\frac{dT_f}{dy} = \frac{PFU_l}{(m/N)c}\left[\left(\frac{S}{U_l} + T_a\right) - T_f\right] \tag{4}$$

Integrating and using the inlet condition y=0, $T_f = T_{fi}$, we obtain the temperature distribution,

$$\frac{\left(\frac{S}{U_l} + T_a\right) - T_f}{\left(\frac{S}{U_l} + T_a\right) - T_{fi}} = \exp\left\{ - \frac{F'U_l y L_2}{mc} \right\} \tag{5}$$

The fluid outlet temperature T_{fo} is obtained by substituting T_f=T_{fo} and y = L_1 in the above equation and after simplification for whole length of the tube, the useful heat gain rate for the collector is,

$$Q = F_R [\mathcal{P} \quad - U_l A_p (T_4 - T_a)] \tag{6}$$

$$= m \ (T_1 - T_4) \tag{7}$$

Also, to determine the overall heat loss coefficient from the absorber plate, the natural convection coefficient for the enclosed space between the absorber plate and the first cover is calculated by using the following correlation [19],

$$Nu = 0.27 \, (Ra)^{0.25} \tag{8}$$

$$h_{p-c} = \frac{Nuk_a}{D_e} \tag{9}$$

Nu and Ra are the Nusselt and Rayleigh numbers, respectively. The characteristics dimension D_e is the spacing between the absorber plate and the cover, while the properties are evaluated at the arithmetic mean of the surfaces temperatures. Eq. (6) & (7) are very convenient expression for calculating the useful energy gain because the inlet fluid temperature is usually a assumed quantity.

Expansion in turbine (1-2)

The superheated or saturated vapor of the working fluid passes through the turbine to generate mechanical power. After the vapor expands, it is depressurized by the turbine blades. The vapor comes out of the turbine at lower pressure P_2 and at low temperature T_2. Then, the turbine output is given by,

$$W_T = \eta_t \, m \, (h_1 - h_2) \tag{10}$$

where, η_t is the turbine efficiency (assumed as 0.9)

Process in the condenser (2-3)

The condenser is intended to recover heat from the CO_2-based Rankine cycle section, and at the same time to cool CO_2 to a temperature low enough to change into a liquid state to form a complete Rankine cycle. In this simulation, CO_2/water heat exchanger was considered, to heat low temperature water. In the CO_2/water heat exchanger, the heat capacity of CO_2 is calculated based on the average temperature of the CO_2-side of the heat exchanger. The outlet temperature for CO_2 loop and water loop in the CO_2/water heat exchanger and heat-exchanging quantity are calculated based on the heat transfer equations and computations of heat balance [19]. The vapor of the working fluid goes through a constant pressure phase change process in the condenser into a state of liquid, rejecting the heat into the environment or the condenser coolant. The pressure of the working fluid within the condenser is equal to the Rankine cycle lower pressure, P_2, and the temperature is equal to the saturation temperature of the pressure, P_2. The condenser load, Q_R, which is the rate of heat rejection from condensing working fluid, can be calculated from the following basic equation,

$$Q_R = m\,(h_2 - h_3) \tag{11}$$

Process in the pump (3-4)

The circulation pump is the driving mechanism of the proposed system. The working fluid (saturated liquid) leaving the condenser at low pressure P_2 regains high pressure here to P_1. The working fluid is pumped back into the evaporator. The circulation pump W_p is calculated by the following equation [20],

$$W_p = \frac{(P_1 - P_2)m}{\rho \eta_p} \tag{12}$$

where ρ and η_p (assumed as 0.9) denote the density of working fluid (saturated condition) and the adiabatic efficiency of the circulation pump, respectively. The specific enthalpy of the working fluid at the circulation pump outlet, h_4, is

$$h_4 = h_3 + \frac{W_p}{m} \tag{13}$$

where, h_3 is the specific enthalpy of the working fluid at the circulation pump inlet.

Thermal efficiency and heat recovery efficiency

The following efficiencies are defined to describe the cycle performance, thermal efficiency (η_{th}) of the Rankine cycle and heat recovery efficiency (η_{heat}) (5),

$$\eta_{th} = \frac{W_t - W_p}{m(h_1 - h_4)} \tag{14}$$

$$\eta_{heat} = \frac{m(h_2 - h_3) - m(h_4 - h_3)}{m(h_1 - h_4)} \tag{15}$$

RESULTS AND DISCUSSION

The mathematical relations presented in the thermodynamic analysis section are employed to determine the performance of the proposed system. Also, in this study, the objective is to analyze the main design parameters that influence the system performance and also to optimize the design of such a CO_2-based Rankine system. The various governing parameters for the cycle are the amount of concentrated power reached to the absorber, the solar concentrator area, heat exchanger area of condenser (A_{HE}), inlet water temperature in the condenser (T_5) and water flow rate in the condenser (m_w). In order to study the effects of the parameters on the cycle performance, numerical calculations are carried out for a wide range of parameters by using MATLAB simulation program. For all simulations, the three values of CO_2 flow rate (0.01, 0.0125, 0.015 kg/s) are assumed, which are chosen to be same with the value taken in the experiment [5-7].

Simulation results for the various concentrated power

The concentrated power reached to the absorber depends on the solar intensity falling on the concentrator area. Figure 6 shows the variation of concentrated power with respect to solar intensity at optimum values of design parameters for the concentrator area of 4.2 m^2 by using the Eqns. (1) & (2).

By the geometrical calculations, the collector area varies from 0.7728 to 4.02 m^2 and concentrated power varies from 686 to 2474 watts for the total number of 20, 40, 60, 80 rows of mirrors in the concentrator plane. Table 1 shows the simulated results for the various concentrated power

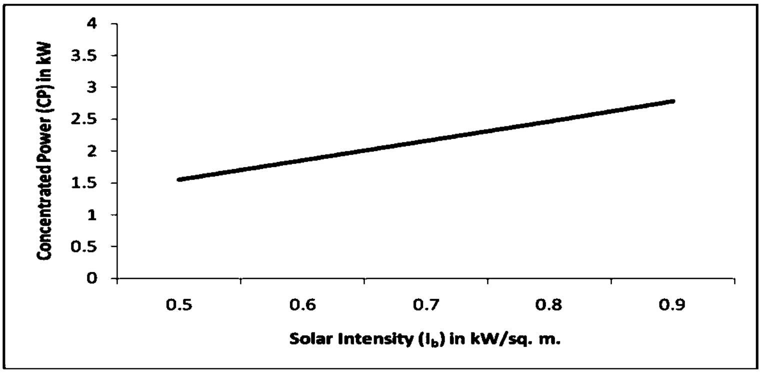


Figure 6: Solar Intensity Vs Concentrated power

by varying the number of reflectors in the LFRSC system for the CO_2 mass flow rate of 0.01 kg/s with the heat exchanging area A_{HE} of 0.1 m^2, the inlet water temperature of 30^{0}C and water flow rate of 0.17 kg/s (5). It can be seen that there is an obvious influence with the concentrated power on the cycle performance. The temperature of supercritical CO_2 at the outlet of the solar collector (T_1) is reached about 89^{o}C for 80 rows of reflectors at mass flow rate of 0.01 kg/s. For the total collector area of 4.02 m^2, the temperature achieved is 89^{o}C. It can also be seen that there is about a temperature difference of 25-35^{o}C between the inlet and outlet of the turbine. At the turbine outlet, the CO_2 temperature T_2 is about 40-20^{o}C (app.) for various concentrator area, which can easily be used as useful heat resource. If the system is used for the large collector area, the turbine output as well as heat recovery can be increased marginally. Based on the data, it can be seen that the CO_2 temperature at the outlet of the low-temperature heat recovery system (T_3) is enough to produce hot water for domestic use.

It is seen from Table 1 that the useful outputs of the thermodynamic cycle vary obviously with the concentrated power for the mass flow rate of 0.01 kg/s including the turbine power and the heat power outputs achieved in the heat recovery systems. The two outputs increase with the concentrated power. It can be clearly seen that the CO_2-based Rankine cycles work in the trans-critical region at the high-pressure side of 9.0 MPa and the low pressure side of 6.5 MPa. Under the supercritical state, CO_2 is heated by the absorber surface in the LFRSC system. It should be mentioned here that the data in Table 1 can be optimized by using variable pressures.

Table 1:
Simulated results of the Rankine cycle for the various concentrated power

Mass flow rate, m (kg/s)	Total number of reflectors	20	40	60	80
	Concentrated power (Watts)	686	1290	1868	2474
0.01	CO_2 temperature, T_1 (°C)	53.4	65.3	76.6	89
	CO_2 temperature, T_2 (°C)	20.5	30.4	40.8	46
	Turbine output (W)	200.3	236.8	270.6	300
	Heat output (W)	473.7	1032.2	1567.4	2136
	Thermal Efficiency (%)	7.5	8.1	9	9.6
	Heat recovery efficiency (%)	27	37	48.5	59

The turbine power output and heat recovery increase with the concentrated power and with mass flow rates. It can be explained that although the turbine output and heat output increase with concentrated power, the heat quantity absorber into CO_2 in the absorber area also increases. Furthermore, the increasing amplitude of the heat quantity is larger than that of power output. It is found that the thermal efficiency and heat recovery efficiency varies from 7.5% to 10% and 27% to 60% for the various concentrator area and considered three mass flow rates.

Effect of CO_2 flow rate on cycle temperatures and energy output for the solar concentrator area of $4.02m^2$

The flow rate of CO_2 in the cycle is very important from the performance point of view. This question is closely related to the optimal flow rate of CO_2 in the cycle. The effects of CO_2 flow rate over the range of 0.01–0.015 kg/s were investigated with the efficient collector area taken as 4.02 m^2. The variations of the cycle temperatures with CO_2 flow rate are shown in Figure 7. It can be seen from Figure 7 that increasing the CO_2 flow rate can decrease the Rankine cycle temperatures slightly. The outlet temperature of the fluid which is coming out of the absorber tubes decreases from about 89.0 to 73.0°C in the range of the present study. Further, it can be seen that the slope of declination is very less up to 0.0125 kg/s and large between 0.0125 and 0.015 kg/s, although the solar radiation and concentrator area are the same. Figure 8 shows the effect of CO_2 flow rate on the cycle output and power efficiency, it is obvious that the outputs, including turbine power output which increase with the CO_2 flow rate and slight decrease in the heat power output with respect to the CO_2 flow rate. Thus, the CO_2 flow rate in the Rankine cycle loop (i.e. the amount of CO_2 charged in the loop) should be raised as much as possible under the condition of a safe operation in order to increase the useful outputs and cycle efficiencies. Also, from the graph, it is inferred that, both for heat and power output the variation in the slope is large from 0.01 to 0.0125 kg/s after that the slope is very less.

In the condenser, the heat exchanging area, the inlet water temperature and water flow rate are also the governing parameters for the Rankine system. The effect of the heat exchanging area in the perfor-

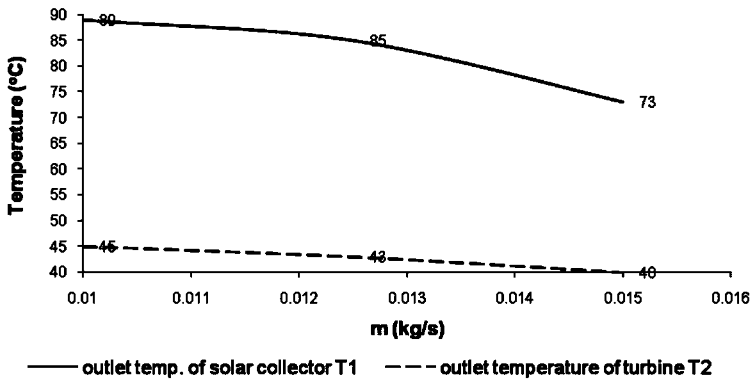


Figure 7: Effect of CO_2 flow rate on cycle performance

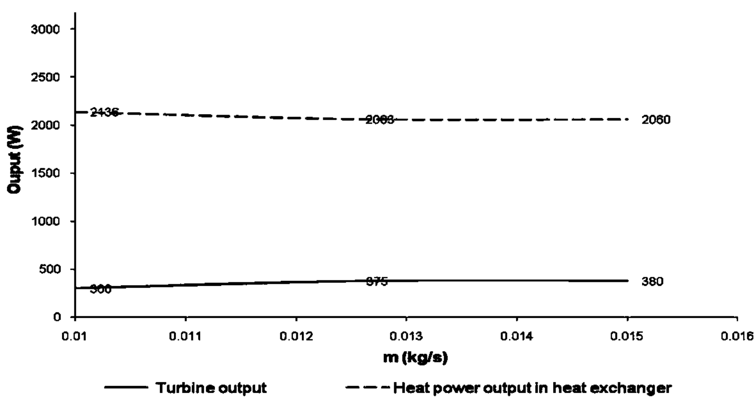


Figure 8: Effect of CO_2 flow rate on the useful cycle outputs

mance is studied in this paper and is shown in Figs. 9 (i) and (ii) at the solar radiation of 800 W/m^2, the concentrator area of 4.02 m^2, the inlet water temperature of T_5 = 20°C and the water flow rate of m_w = 0.013 kg/s for the CO_2 mass flow rate of 0.01 kg/s. It is seen that the cycle temperature decreases with increase in area A_{HE}. The area increase means the more heat quantity is recovered from the CO_2, which leads to the decreases of the CO_2 temperatures in the cycle. The decreases of the CO_2 temperatures result in a drop of the heat and power output shown in Figure 9(ii).

Expected P-h Diagram of the Proposed System

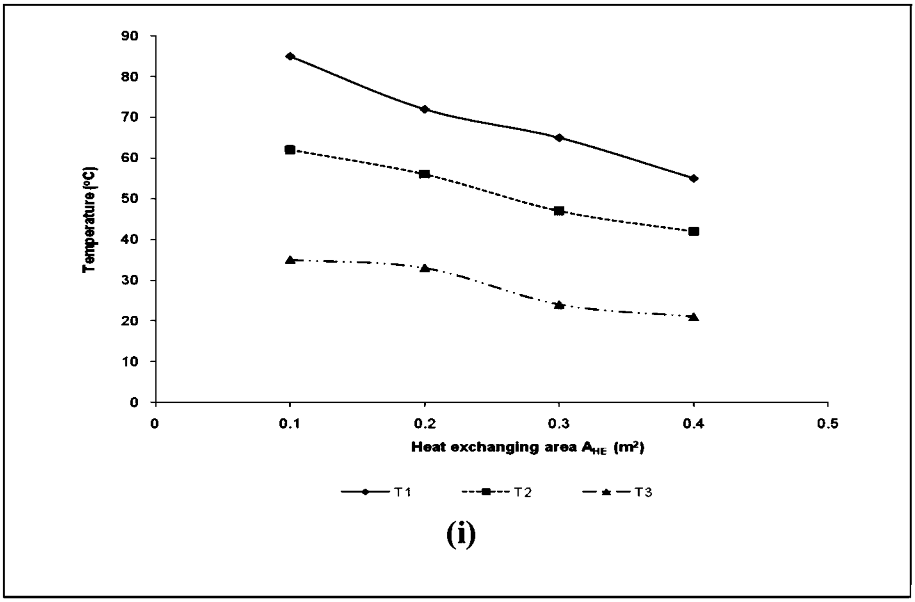
Figure 10 shows the expected CO_2-based trans-critical Rankine cycle by P-h diagram, in which thermodynamic and transport properties of CO_2 are calculated based on the heat transfer analysis. As compared with Figure 4, it can be clearly seen that these working cycles are used for simultaneous heating and heat recovery in the temperature from about 40°C to 89°C for the heating process in the concentrator and about 20°C to 45°C for heat recovery. Based on the above results, the proposed system has a potential to be used for the energy loads of commercial requirements, hotels, schools and hospitals that can be benefited from an independent source of power supply and heat supply. The turbine and hot water system are used to supply energy for the user and for the small buildings, only hot water supply or turbine and hot water system may be better.

VERFICATION AND VALIDATION OF SIMULATION MODEL

Efficiency Modeling

The formulation of the numerical model in this study is expected to have a capability of reasonably dealing with the CO_2-based Rankine cycle. Therefore, for validation of the analytical model, the various assumptions are taken from the existing literatures [5-7]. The theoretical and experimental study on evacuated tube solar collector using supercritical CO_2 was conducted by various authors. The experiment was conducted under a typical summer condition in Osaka area [7], and the power generation and heat output efficiency were measured at 0.25 and 0.65 respectively. The predicted values for the present LFRSC system, are η_{th} = 10% and η_{heat} = 60% respectively. A comparison of these efficien-

Effect of heat exchanging area (A_{HE})



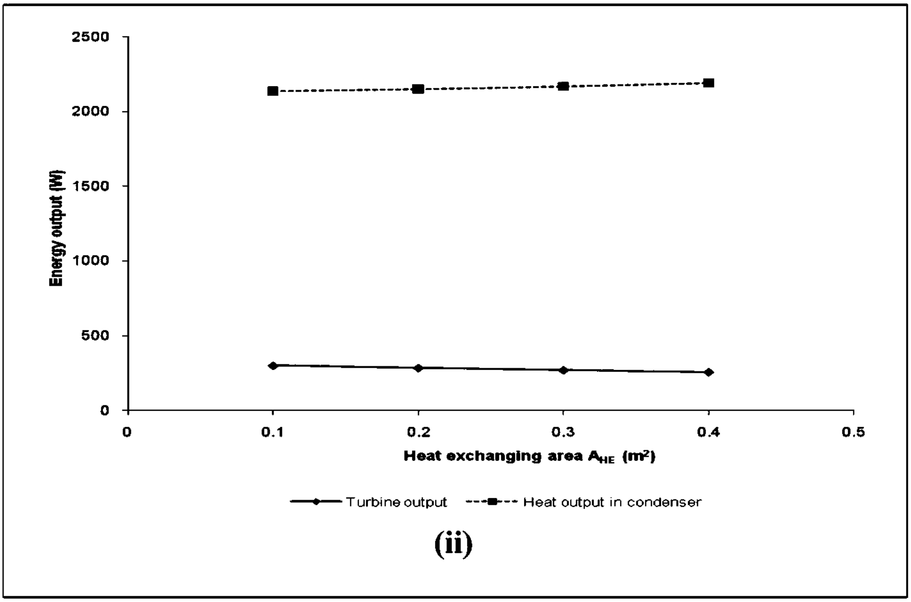


Figure 9: (i) Effect of heat exchanging area of A_{HE} on the CO_2 temperatures in the cycle; (ii) Effect of heat exchanging area of A_{HE} on the useful cycle outputs

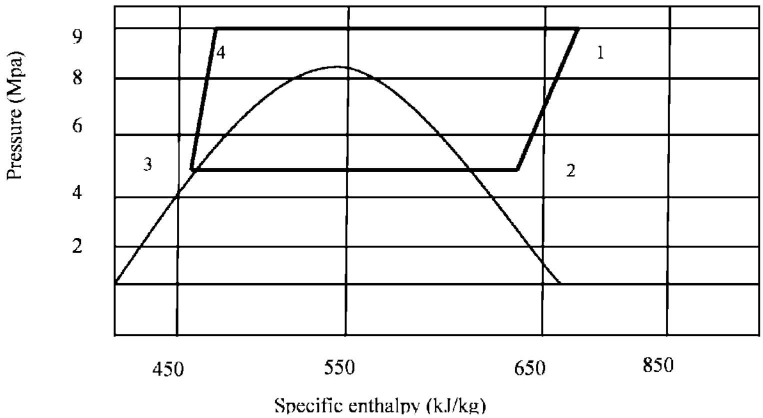


Figure 10: Expected P-h diagram of Rankine cycle powered by LFRSC

cies between the evacuated tube system and LFRSC system shows that, the present system can also be used for the purpose. The power output of the evacuated tube system is higher than the calculated value of the present system. The reason given to this phenomenon may be the considered system, collector area, CO_2 flow rate circulation is little higher that the value used in the simulation and evacuation maintained in the tube of the system. But to a certain extent, the agreement is found and therefore the present LFRSC model can be used with confidence.

Effect of CO_2 flow rate on cycle temperatures and heat exchaning area

Increasing CO_2 flow rate can enhance the turbine power output even-though the temperature output is decreased. It is inferred that the temperature increase is nearly about 4% to 15% but the mass flow rate increase is 50%. Thus the increase in mass flow rate increases the turbine output from 300W to 380 W, as can be seen from Figure 8. The simulated turbine output with increase in mass flow rate and decrease in temperature is similar as compared to trend presented in the existing literature [20]. The heat power output is slightly in increasing order with respect to increase in area A_{HE}. Also, it is inferred that, thermal efficiency output decrease with increase in the heat exchanging area, due to the decrease of the turbine output. The simulated results with increase in heat exchanging area and decrease in temperature is similar as compared to trend presented in the existing literature [5].

CONCLUSION

A feasibility study was conducted to analyze the performance of solar energy powered Rankine cycle using supercritical carbon dioxide and the possibility of using LFRSC system as a heat source. The system provides electrical output and heat power output. The obtained results show supercritical CO_2 can effectively collect heat in the LFRSC system and CO_2 temperature at the outlet of the trapezoidal cavity absorber can reach about 73°C to 89°C, which help to achieve heat recovery of 2.0 to 2.14 kW and power generation of 0.30 to 0.380 kW for the increase in mass flow rate of 0.01 to 0.015 kg/s. Choosing a heat sink with temperature low enough in the heat exchanger is important for high efficiency operation. The thermodynamic analysis based on the measured data show that the CO_2-based Rankine cycle can achieve heat collection and electricity generation with a reasonable thermal efficiency of 7.5% to 10% and heat recovery efficiency of approximately 30% to 60% for various concentrator areas and for mass flow rate of 0.01 to 0.015 kg/s. The results reveal that the turbine output is increased with concentrator area and mass flow rate but decreased with the increase of the heat exchanging area. So, the present study shows the potential of the application of the CO_2-based Rankine cycle powered by LFRSC. In addition, the objective of the paper is only to give a feasibility study of using LFRSC for supercritical fluid and it is recognized that a more detailed transient simulation is needed to further estimate and understand the cycle performance and establish the practical usefulness of this cycle.

FOR FUTURE RESEARCH

Analytical modeling of the trapezoidal absorber by deriving the equation for cavity system will be required for further comparison. Theoretical modeling accuracy will be sought for the trapezoidal cavity absorber by deriving still better overall heat loss coefficient correlations. The results show that the proposed cycle has great potential to achieve a 'true' green energy generation, relieving energy pressure in the world and greatly reducing CO_2 emission. But further investigation is needed in the future to study the nature of the supercritical CO_2 flow and heat transfer in the collector. Also, it is recognized that future studies, especially transient studies and experimental studies, are needed to establish the practical usefulness of this cycle.

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