Thermal Modeling and Performance Analysis of U-Tube Evacuated Solar Collector using CO_2

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Abstract: This study deals with the thermal modeling of U-tube evacuated solar collectors to investigate the heat transfer and performance characteristics. The U-tube evacuated solar collectors are integrated into solar-assisted new generation experimental organic Rankine cycle working with environmentally friendly supercritical CO_2. For the modeling, one-dimensional heat transfer analysis is applied to the U-tube collectors for determining the heat transfer characteristics of the collectors as well as the CO_2 exit temperature from the collectors for the steady heat transfer process. Additionally, heat loss coefficient and overall heat transfer coefficients are determined for the experimental U-tube collectors. The obtained results are also compared with the results of the experimental study.

Keywords: Solar energy, carbon dioxide, Rankine cycle; U-tube evacuated solar collector.

1. INTRODUCTION

For the last decades, the utilization of renewable energy resources got more attention due to projected fossil fuel depletion in reserves and environmental issues such as global warming and ozone depletion. In such a manner, because of its abandonment of availability and environmental concerns, solar energy is expected to be one of the most reliable alternative energy sources [1]. Naturally, solar thermal technology, an environmentally benign and efficient technology, is becoming increasingly important and promising, such as the continuous research into water heaters utilizing evacuated solar collectors (ESC), which can supply hot water below 90 °C. Compared with the low-temperature technologies, the high-temperature solar thermal applications such as the concentrated solar thermal system for power generation is relatively mature [2]. However, the generation of power from concentrated solar power plants and comparable medium-temperature heat-sources continues to face obstacles including high investment costs and the utilization of flammable, toxic, or high Global Warming Potential (GWP) working-fluids. Additionally, efficiencies in solar plants are currently limited by the maximum allowable temperatures of heat-transfer-fluids used in solar collector fields. Also, solar power plant efficiencies are presently restricted by the highest permissible heat transfer fluid temperatures [3]. Considering such restrictions, solar resources are not cost-competitive at present because of the high capital expenditures. However, the potential exists for reducing the costs by improving the performance of solar thermal power systems [4].

As a low-cost alternative, several researchers have shifted their interests to develop high efficiency solar thermal energy systems for thermodynamic power cycles. Among these researches, low-temperature solar power/heat thermodynamic cycle systems have been the most promising ones by their high efficiency of heat collection [5]. In recent years, low-temperature solar technologies have been developed, and the processes are more energy and cost-efficient than in previous systems. Thermodynamic cycles for combined power and heat system have a great potential to become competitive with fossil fuels based power systems. In order to reduce the cost and improve the thermodynamic performance of low-temperature solar power cycles, it is essential to use integrated cycle approaches and to employ new and innovative ideas [6]. For achieving these goals, an experimental solar-assisted thermodynamic cycle was proposed by Zhang *et al*. [7] using supercritical CO_2. In the system, solar energy and CO_2 are used to form a cogeneration system of heat and power with environmental preservation. The representative diagram of the experimental CO_2-based organic Rankine cycle (ORC) is shown in Figure **1**. In the system, U-tube ESCs are used to heat CO_2 passing through the collectors, and the temperature of supercritical CO_2 reaches above 200 °C [8].

CO_2 has been identified as a promising working fluid with many advantages, such as its environmental compatibility and the anomalous behavior of certain

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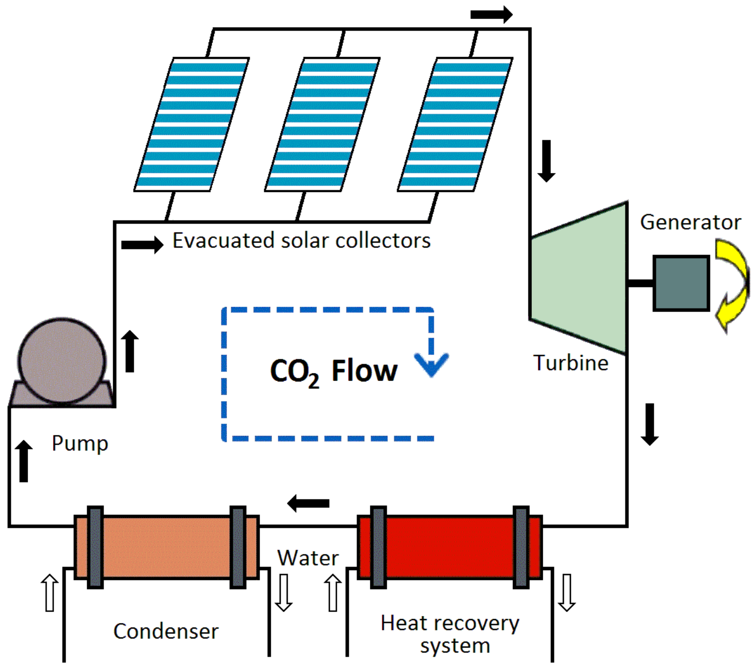

Figure 1: Representative diagram of the experimental CO_2-based organic Rankine cycle.

thermo-physical properties [9]. It has low critical properties with a critical pressure of 7.38 MPa and a critical temperature of 31.1 °C. Among other advantages, CO_2 is abundant, inexpensive, non-toxic and non-combustible. Higher efficiencies are possible using supercritical CO_2 because it can withstand high temperatures without degradation, and requires less compression work close to the critical point [10]. Additionally, it has a strong potentiality to be used as an alternative refrigerant as it has very low GWP and neutral effect on the depleting ozone layer [11].

ESC is a device usually utilized to supply heat to various applications such as power generation, water heating, refrigeration, etc. This collector can reach temperatures above 200 ° C, thanks to their combining effects of highly selective surface coating and vacuum insulation. Since they have high heat extraction capability, ESCs are more attractive when compared to flat collectors. In addition, they are cost-effective, most reliable, and have a reasonably long lifetime [1]. The thermal analysis of evacuated tube solar collectors (ESC) has been widely investigated for the last decades. Some researchers analyzed ESC using air as working fluid [12-13] while some of them investigated the ESC performance using water as working fluid [14-16]. Naik *et al*. [1] introduced mathematical modeling of a U type ESC for predicting the outlet temperature and net heat gain using aqueous lithium chloride solution, water, and air as working fluids. Naik and Muthukumar [17] presented three-dimensional thermal modeling of U-type ETSC using finite element method for estimating the performance of the collector. Li *et al*. [18] theoretically investigated concentrating ESC in terms of optical and thermal behaviors using heat transfer fluid as a working fluid. Islam and Sumathy [11] carried out an analytical study for solar-assisted heat pump integrated water heating system where ETSC was used for heat energy demand using CO_2 as working fluid. For the analysis, they have also presented a one-dimensional heat transfer analysis of the ESC. Aboulmagd *et al*. [19] studied on a new mathematical model to analyze the ETSC in terms of heat transfer using MATLAB software. Yadav and Saikhedkar [20] developed a simulation model in order to investigate the performance of ETSC. Shafieian *et al*. [21] studied on a mathematical model for calculating the optimum number of glass tubes of the heat pipe solar collector. Moslemi and Keshtkar [22], investigated the thermal performance of an ETSC for the best thermal efficiency. The efficiency of the collector was examined for different parameters. A thermal network was conducted for the ETSC. Yang *et al*. [23] presented a dynamic mathematical model for the evacuated tube glass collector manifold header for water heating. For the model validation, they constructed an experimental test rig.

The scope of this study is to model and investigate the thermal performance of the U-tube ESC, which is integrated into a new generation experimental power generation system working with supercritical CO_2. For

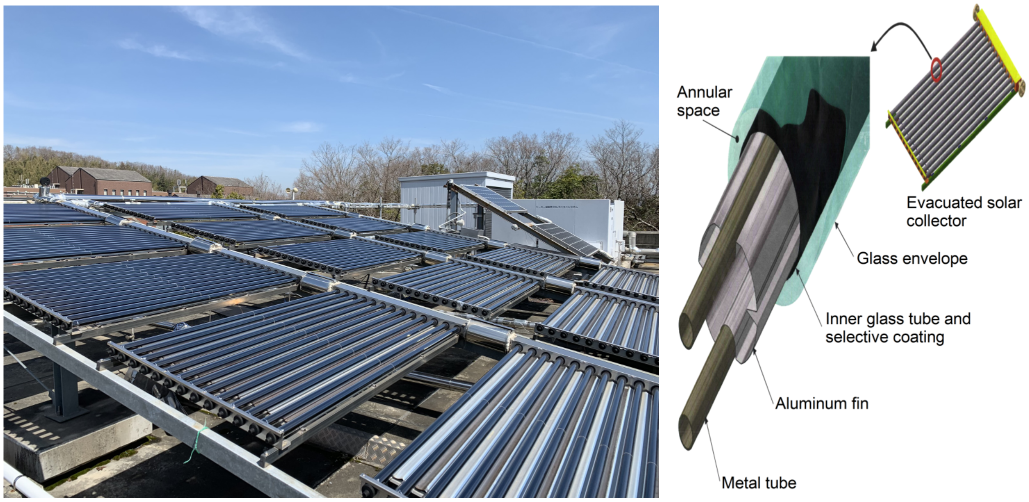

Figure 2: The picture of solar collectors and cross-section of the evacuated tube.

this aim, one-dimensional heat transfer analysis is applied to U-tube ESC in order to obtain thermal characteristics of the collectors for steady heat transfer processes. Additionally, heat loss coefficient and overall heat transfer coefficients are determined for the experimental U-tube ESC. The analysis is carried out to predict the outlet temperature of CO_2 at the exit of the solar collector and are compared with the experimental results.

2. EVACUATED TUBE SOLAR COLLECTORS

The experimental solar-assisted CO_2 based ORC was designed, constructed, and tested in 2004. It was installed on the roof of the Energy Conversion Research Center in Doshisha University Kyoto, Japan [6]. The first manufactured prototype was consisting of only U-tube ESC, a feed pump, heat recovery system (HRS), and an expansion valve to simulate the turbine operation. Later on, a turbine which was specially designed for the experimental setup has been mounted on the system [8]. The U-tube ESC is the heart of the power generation system. Its characteristics play an important role in the successful operation of such systems. It is developed and provided by Showa Denko. K.K. as a commercial product (Figure **2**). The collector consists of a glass envelope over a glass tube coated with a selective solar absorber coating. The inner glass tube is evacuated to maintain a vacuum environment and coated by an absorber coating which has a high solar absorbance 0.927 and a low emissivity 0.193. The absorbed heat is conducted through the inner glass tube wall and then absorbed by the CO_2 in a metal U-tube inserted in the inner tube with a fin. The thickness is of the fin is 0.2 mm, and it is connecting the outlet arm of the U-tube to the inner glass tube. Each U-tube is 3.6 m long and 0.005 m internal diameter. It has an efficient area of about 9.6 m^2. 15 units of collectors can be connected in parallel or in series by adjusting the valves. Each collector unit is made of 13 collector tubes. The collectors can stand temperature up to 250 °C [8, 24].

It can be seen from the figure that, solar radiation passes through both the glass tubes to be absorbed by the selective surface. The surface heats up, and the heat is transferred by conduction to the metal tube and then by convection to the CO_2. The thermophysical properties and geometry of the U-tube ESC are given in Table 1 [4].

Table 1: Properties of the U-Tube ESC

Properties of glass	
Thermal conductivity, λ_g(W/mK)	1.25
Solar transmittance, τ_g	0.90
Solar absorbtance, α_g	0.05
Solar reflectance, ρ_g	0.05
Thermal emittance, ε_c, ε_r	0.83
Properties of absorber coating	
Solar transmittance, τ_f	0.04
Solar absorbtance, α_f	0.927
Solar reflectance, ρ_f	0.033
Geometry	
Length of the tubes, L (m)	1.7
Mean radius of the glass envelope, r_c (m)	0.018
Mean radius of the inner glass tube, r_r (m)	0.013
Mean radius of the metal tube, r_d (m)	0.03

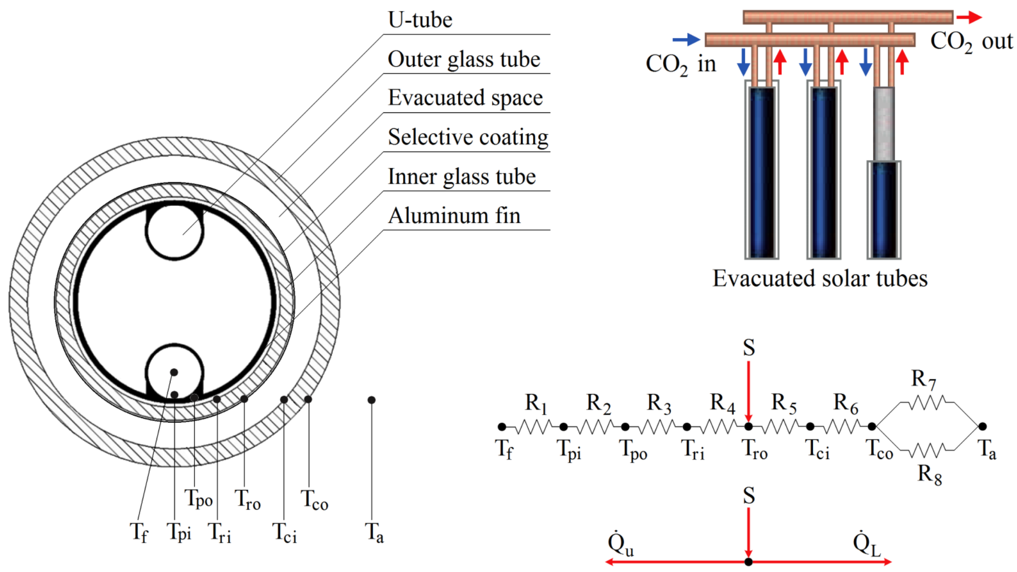

Figure 3: The thermal network of the evacuated solar U-Tube (adapted from [11] and [18]).

3. THERMAL MODELLING OF U-TUBE ESC

Evacuated collectors consist of a U-type pipe inside a vacuum-sealed tube. These tube collectors have shown that the combination of a selective surface and an effective convection suppressor can result in a good performance at high temperatures. The vacuum envelope reduces convection and conduction losses so that the collectors can operate at higher temperatures than flat plate collectors. This gives evacuated collectors an advantage over flat types in terms of daylong performance [25].

For the thermal modeling of the U-tube ESC, which is integrated into the power generation system, one-dimensional heat transfer analysis is carried out. Figure **3** shows the cross-sectional of the U-Tube ESC with the thermal network. The U-tube ESC is heated by the solar radiation across the evacuated space and effective incident solar irradiance is then absorbed by the selective coating. The big amount of absorbed solar energy is conducted aluminum fin to U-tube and then transfer to the working fluid.

In order to determine the radiation heat transfer coefficient between the outer glass tube and the sky, the equation given by Kalogirou [25] is used:

$$h_{r,co-sky} = \varepsilon_c\, \sigma\, (T_{co} + T_{sky})(T_{co}^2 + T_{sky}^2) \tag{1}$$

where σ is Stefan–Boltzmann constant, ε_c is the emittance of the glass cover, T is the temperature, and subscript co is outer glass cover. The sky temperature can be determined from the equation given below [26]:

$$T_{sky} = 0.0552 T_a^{1.5} \tag{2}$$

In the above equation, T_a is the environment temperature, and all the temperatures are in K. Convection heat transfer coefficient between the outer glass tube, and the environment is given below.

$$h_{c,co-a} = \frac{Nu_a\, k_a}{D_{co}} \tag{3}$$

where k_a is the thermal conductivity of air and D_{co} is the outer glass diameter. In the above equation Nusselt number can be determined by;

$$Nu_{air} = 0.4 + 0.54\, Re^{0.52} \quad \text{for } 0.1 < Re < 1000 \tag{4a}$$

$$Nu_{air} = 0.3\, Re^{0.6} \quad \text{for } 1000 < Re < 50000 \tag{4b}$$

The conduction heat transfer coefficient through the outer glass tube can be expressed as [27]:

$$h_{d,co-ci} = \frac{k_c}{r_{ci} \ln\left(\frac{r_{co}}{r_{ci}}\right)} \tag{5}$$

In Equation (5), subscript ci is inner glass cover. Since the place between inner and outer glass tubes is evacuated, the convection heat transfer is assumed to be zero. Thus, heat transfer occurs only by radiation. The radiation heat transfer coefficient between the absorber surface and the outer glass tube is defined as [25]:

$$h_{r,ro-ci} = \frac{\sigma}{\frac{1}{\varepsilon_r} + \frac{r_{ro}}{r_{ci}}\left(\frac{1}{\varepsilon_c} - 1\right)} (T_{ro} + T_{ci})(T_{ro}^2 + T_{ci}^2) \tag{6}$$

where ε_r is the emissivity of the selective absorbing coating, subscripts i and o stand for inner and outer glass tubes. For the inner tube, the conduction heat transfer through the absorber wall can be expressed as;

$$h_{d,ro-ri} = \frac{k_r}{r_{ri} \ln\left(\frac{r_{ro}}{r_{ri}}\right)} \tag{7}$$

The conduction heat transfer through the aluminum fin is given by Incropera *et al*. [27]:

$$h_{d,ri-b} = \frac{k_b}{\delta_b} \tag{8}$$

where δ_b is the thickness of the U-tube (δ_b=1 mm). To determine the convective heat transfer from the U-tube to the CO_2, the modified correlation of Krasnoshchekov and Protopopov is used [28]. This correlation is evaluated by Li *et al*. [18] for CO_2 in heating mode at supercritical pressures:

$$Nu_b = 0.023 Re_b^{0.8} Pr_b^{0.4} \left(\frac{\rho_w}{\rho_b}\right)^{0.3} \left(\frac{\overline{C_p}}{C_{p,b}}\right)^{n} \tag{9}$$

In Equation (9), the superscript n can be calculated from;

$$n = 0.4 \quad \text{for } T_b < T_w < T_{pc} \text{ and } T_w > T_b > 1.2T_{pc} \tag{10a}$$

$$n = 0.4 + 0.2\left(\frac{T_w}{T_{pc}} - 1\right) \quad \text{for } T_b < T_{pc} < T_w \tag{10b}$$

$$n = 0.4 + 0.2\left(\frac{T_w}{T_{pc}} - 1\right)\left(1 - 5\left(\frac{T_b}{T_{pc}} - 1\right)\right)$$
$$\text{for } T_{pc} < T_b < 1.2T_{pc} \text{ and } T_b < T_w \tag{10c}$$

In the above equations, the subscript b and w denote bulk and wall conditions while pc denotes "pseudo-critical". The temperatures are in K and $\overline{C_p}$ is the average specific heat capacity and determined by the ratio of enthalpy and temperature differences:

$$\overline{C_p} = \frac{h_w - h_b}{T_w - T_b} \tag{11}$$

The pseudo-critical temperature of CO_2 as a function of gas pressure, P_{gc}, was obtained using the curve-fitting method by [29];

$$T_{pc} = a + b2P_{gc} + cP_{gc}^2 + dP_{gc}^{2.5} + eP_{gc}^3 \tag{12}$$

$a = -1.226 \times 10^2, b = 6.124 \times 10^{-2},$

$c = -1.657 \times 10^{-5}, d = 1.7739 \times 10^{-7},$

$e = -5.6 \times 10^{-10}$

It should be noted that in the above equation, P_{gc} is in kPa. The overall heat loss coefficient of the evacuated solar collector can be determined by:

$$U_L = \left[\frac{r_{co}}{h_{r,ro-ci}\, r_{ro}} + \frac{r_{co}}{h_{d,ci-co} r_{ci}} + \frac{1}{h_{c,co-a} + h_{r,co-sky}}\right]^{-1} \tag{13}$$

Also, from Figure **9**, the heat loss of the evacuated U-tube tube can be expressed as;

$$U_L(T_{ro} - T_a) = h_{c,co-a}(T_{co} - T_a) + h_{r,co-sky}(T_{co} - T_{sky}) \tag{14}$$

Absorber temperature, T_{ro}, can be easily calculated iteratively by using Equation (14). The overall heat transfer coefficient from the environment to the fluid inside the U-tube is calculated by;

$$U_0 = \left[\frac{1}{U_L} + \frac{r_{co}}{h_{d,ro-ri}\, r_{ro}} + \frac{r_{co}}{2h_{d,pipe}\, r_{ro}} + \frac{r_{co}}{h_{fi}\, r_{pipe}}\right]^{-1} \tag{15}$$

After determining heat transfer coefficients, the useful collected solar energy has to be determined:

$$\dot{Q}_u = F_R A[S - U_L(T_{in} - T_a)] \tag{16}$$

where S is the solar irradiance, F_R is the heat removal factor, A is the collector area, and T_{in} is the inlet CO_2 temperature. The heat removal factor F_R is described by [25]:

$$F_R = \frac{\dot{m}C_p}{AU_L}\left[1 - \exp\left(\frac{-AU_L F'}{\dot{m}C_p}\right)\right] \tag{17}$$

where F' is the collector efficiency factor and defined as:

$$F' = \frac{U_0}{U_L} \tag{18}$$

For determining the CO_2 temperature at the collector exit, the useful solar energy collected by evacuated solar tubes can also be written as:

$$\dot{Q}_u = \dot{m}c_p(T_{out} - T_{in}) \tag{19}$$

4. RESULTS AND DISCUSSION

The performance of the collector depends not only on the function of the U-tube ESC but also on the flow and heat transfer characteristics of CO_2 inside the tube. In the supercritical region, the thermophysical properties of CO_2 are very complicated [5]. The variation of the specific heat capacity, C_P of CO_2 with temperature is shown in Figure **4**. As seen from the figure, specific heat characteristics of CO_2 varies changes significantly with the temperature near the critical point. Because of this distinctive thermophysical properties, it is thought that near the supercritical point, CO_2 has better heat transfer characteristics than the CO_2 in the liquid and gas phase.

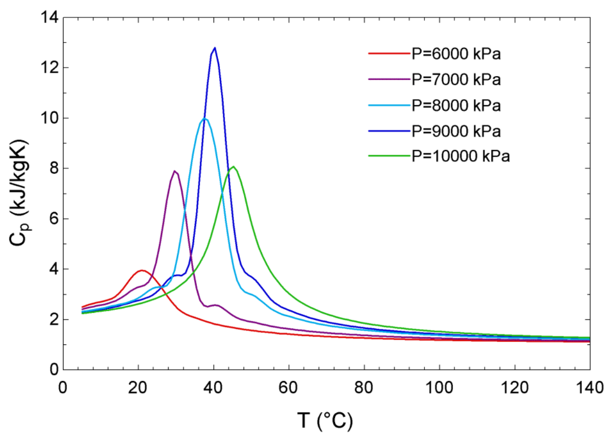

Figure 4: Variation of C_P with temperature for various pressure values.

Using the mathematical model described in the previous section, the outlet temperature of CO_2 at the exit of the U-tube ESC calculated. In Figure **5**, the calculated temperature of CO_2 and the experimental outlet temperature is compared for a typical day in March between 9:00-14:00. It can be seen from the figure that the calculated and the experimental CO_2 temperatures at the exit of ESC are very close to each other. According to the results, the maximum error is estimated to be 2.5 % while the average error is calculated to be 1.98 %.

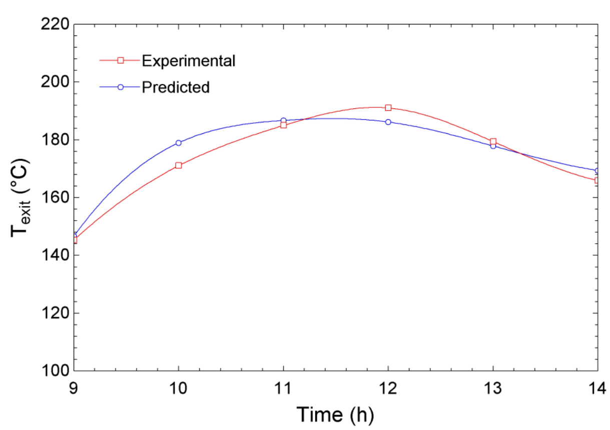

Figure 5: Experimental and predicted CO_2 temperature at the exit of U-tube ESC.

The CO_2 mass flow rate is also an important parameter for the exit temperature. Figure **6** shows the variation of CO_2 exit temperature with mass flow rate. The calculation was made for noontime when the solar radiation was high, that was 875 W/m^2. As expected, with the increase of mass flow rate, the exit temperature of the CO_2 decreases. In the actual case, the mass flow rate of CO_2 was 0.007 kg/s.

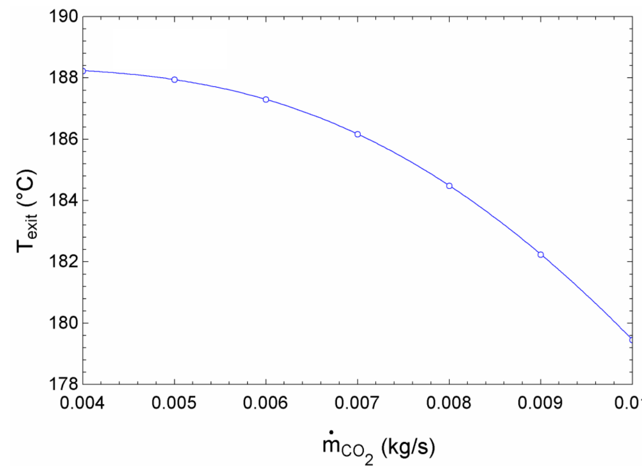

Figure 6: Variation of CO_2 temperature at the exit of U-tube ESC with the mass flow rate.

Figure **7** shows the variation of CO_2 exit temperature with solar radiation for different ambient temperatures. It is clear from the figure that the CO_2 temperature increases linearly with the increase of solar radiation. However, the increment ratio of the exit temperature for higher ambient temperatures is smaller than the lower ambition temperatures. This is because of higher heat losses at higher ambient temperature due to the larger radiation heat transfer coefficient.

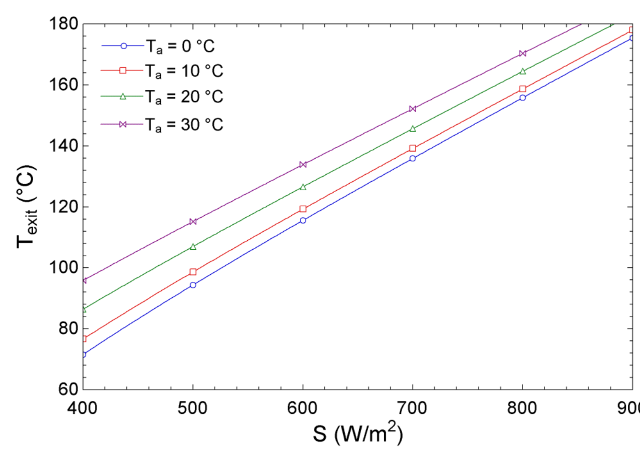

Figure 7: Variation of CO_2 temperature at the exit of U-tube ESC with solar radiation.

The variation of collector efficiency with solar radiation is given in Figure **8**. Contrary to the effect of solar radiation on the CO_2 exit temperature, collector efficiency decreases with the increasing solar radiation. This is because, the temperature difference between the solar collector surface and the surrounding air becomes greater for higher radiation levels, and then the thermal loss of the solar collector to the ambient also increases, which may also contribute to the occurrence of this phenomenon.

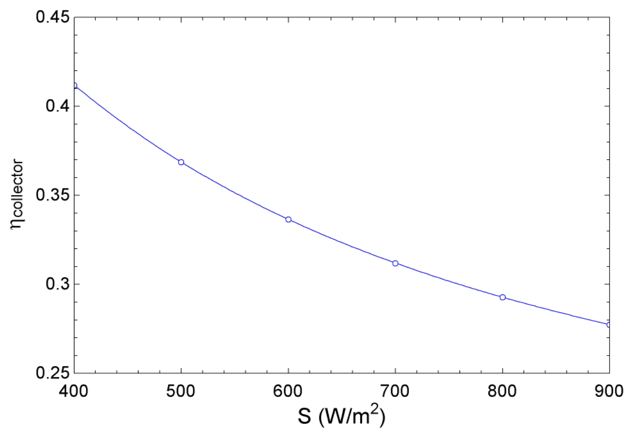

Figure 8: Variation collector efficiency with solar radiation.

The CO_2 temperature variation while passing throughout the evacuated solar collector is given in Figure **9**. The results were obtained using the mathematical modeling and solar radiation data for March. According to the results, the CO_2 temperature increment is higher in the first collectors. After the 8^{th} collector, the increment ratio decreases, and the temperature of CO_2 nearly remains constant for the final collectors. This is mainly because of the temperature difference between the solar collector and the CO_2 since at the end of collector arrays, the CO_2 temperature approaches collector temperature. The second reason may be the increasing heat loses at high temperatures.

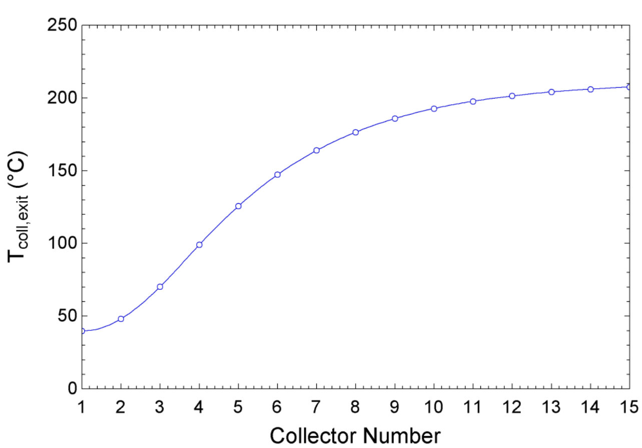

Figure 9: Variation of CO_2 temperature at the exit each collector.

Figure **10** shows the variation of the overall heat loss coefficient with the temperature difference between receiver temperature (T_{ro}) and ambient temperature (T_a). During the experimental measurements, the outside temperature varied between 29-33 °C. From the figure, it is clear that the overall heat loss coefficient increases with the temperature difference. These predicted results obtained using mathematical modeling have a good agreement with the experimental measurements.

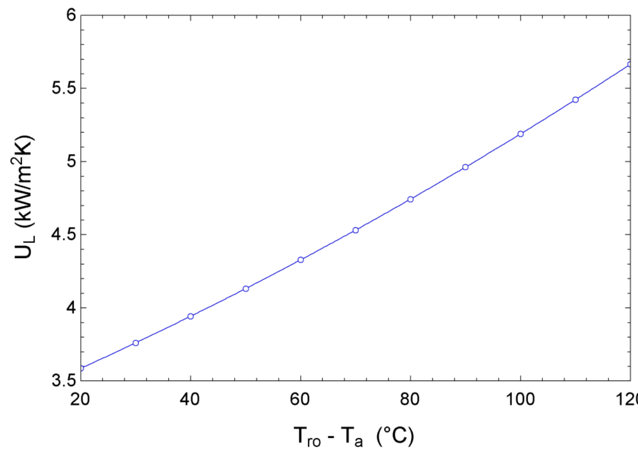

Figure 10: Variation of the overall heat loss coefficient with (T_{ro}- T_a).

5. CONCLUSIONS

In this study, thermal modeling of the U-tube ESC was carried out for performance investigation. The investigated U-tube ESC was integrated into a CO_2 based ORC which was designed by Yamaguchi and his research group [6] for eco-friendly power generation purposes. The analyses were made using a one-dimensional heat transfer model for steady-state heat transfer processes. Heat loss coefficient and overall heat transfer coefficients were determined for the experimental U-tube ESC. Also, the outlet temperature of CO_2 at the exit of the solar collectors was predicted and compared with the experimental results. From the analysis, the followings were concluded:

1. The predicated CO_2 temperature at the exit of ESC was found to have a good agreement with the experimental. The maximum error was estimated to be 2.5 % while the average error was calculated as 1.98 %.

2. The mass flow rate of CO_2 was found to have an important effect on CO_2 exit temperature. So an optimization procedure is necessary for the optimum mass flow rate according to the operating conditions.

3. The thermal efficiency of collectors was decreased with the increase of solar radiation due to the greater temperature difference between the solar collector surface and the surrounding air at higher radiation levels.

4. The CO_2 temperature increment was found to be higher at the exit of preliminary collectors than the subsequent ones. Also, the temperature of CO_2 nearly remained constant for the last two collectors. From this result, it is clear that longer collector lengths do not mean higher collector efficiencies.

5. The overall heat loss coefficient highly depended to the temperature difference between receiver temperature (T_{ro}) and ambient temperature (T_a). According to the results, with the temperature difference from 20-120 °C, the U_L varied between 3.58-5.66 kW/m^{2}K. This result is also the reason for the lower collector efficiencies at higher solar radiation levels.

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NOMENCLATURE

A - collector area, m^2

Cp = Specific heat, kJ/kgK

D = diameter, m

F' = collector efficiency factor

F_R = heat removal factor

h = heat transfer coefficient, W/m^{2}K

k = thermal conductivity, W/mK

L = length, m

$\dot{m}$ = mass flow rate, kg/s

Nu = Nusselt number

$\dot{Q}_u$ = useful collected solar energy, W

r = radius, m

Re = Reynolds number

S = solar irradiance, W/m^2

T = temperature

U_L = overall heat loss coefficient, W/m^{2}K

U_0 = overall heat transfer coefficient, W/m^{2}K

Greek Letters

τ = Solar transmittance

α = Solar absorbtance

ρ = Solar reflectance

ε = Thermal emittance

σ = Stefan–Boltzmann constant

δ = thickness

Subscripts

a = ambient

c = glass cover

d = metal tube

f = absorber coating

i = inner

o = outer

pc = pseudo-critical

r = absorber coating

w = wall

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