Thermal and Geometrical Assessment of Parabolic Trough Collector Mounted Double Evacuated Novel Receiver Tube System

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Research Article

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Abstract

This paper particularly aims to highlight the necessity of optimal geometric design considerations of a parabolic trough collector (PTC) mounted novel receiver tube in view of efficient operation and high-end performance. Many investigations, analysis, and validation have been done in this regard as solar energy based PTC now a commercially mature technology acknowledges a variety of role in the form of power generation and other thermal applications. This article identifies the optimal rim angle corresponding to its tube size as required for high exergetic gains. Almost six receiver tubes, distinct in terms of dimensions and number of covers are compared for their best results to be mounted on adequate geometry with different rim angle (40°, 80°, and 120°). A significant variation of flow rate (i.e. 16 to 216 *litre/hr*) and inlet fluid temperature (i.e. 323 *K*, 423 *K*, 523 *K*, 623 *K*, and 723 *K*) has been extensively detailed about high energy and exergy retrieval from the system. The study reports that all the favorable results are found with the receiver tube having a diameter of 0.027 m and a double envelope, compared to other design considerations. Results show that as the flow rate increases energy efficiency also increases up to some extent along with increasing receiver tube temperature. The highest energy and exergy efficiency as reported to be 79.4% and 47% respectively with 80° being the optimal rim angle for a 5.7 *m* wide parabolic aperture.

Keywords: Energy efficiency, Fluid inlet temperature, Heat transfer fluid, Loss coefficient, Parabolic trough collector, Rim angle.

Nomenclature and Symbols

A_a	Aperture area (PTC)
D_{c1i}	Inner diameter of cover 1
D_{c1o}	Outer diameter of cover 1
D_{c2i}	Inner diameter of cover 2
D_{c2o}	Outer diameter of cover 2
D_r	Receiver diameter
D_{ri}	Receiver inner diameter
F_R	Heat removal factor (collector)
h_c	Heat transfer coefficient (convection)
K_c	Conductivity (glass cover)
L	Receiver length
Q_{loss}	Energy loss
Q_u	Useful energy gain
S	Irradiance (solar)
T_a	Temperature (ambient)
T_{ci}	Temperature (inner cover)
T_{co}	Temperature (outer cover)
T_{fi}	Temperature (inlet fluid)
T_{fo}	Temperature (outlet fluid)
T_{sky}	Temperature (sky)
T_{sun}	Temperature (sun)
T_r	Temperature (receiver surface)
U_L	Heat Loss coefficient

Symbols

ε_r & ε_c	Emissivity of receiver and cover
F'	Collector efficiency factor
η	System efficiency
σ	Stefan-Boltzmann constant

Abriviations

DERT   Double Evacuated Receiver Tubes

FVM   Finite Volume Method

HTF   Heat Transfer Fluid

MCRT   Monte-Carlo Ray Trace

PTC   Parabolic Trough Collector

SERT   Single Evacuated Receiver Tubes

1. Introduction

1.1 Background and current status: Solar energy is one of the potential alternatives which is substantially fulfilling global energy requirements with adequate demand side management (Zaharil & Hasanuzzaman 2020). Solar photovoltaic modules on one hand directly convert the light into electricity while thermal conversions are based on well fabricated concentrating and non-concentrating solar collectors. As non-concentrating thermal collectors are ideal for low-grade thermal applications in particular while operating as a decentralised system while concentrating solar collectors are suitable for medium to high-temperature applications essentially meant for power generation. A variety of attempts have been made in the form of constructional design and appropriate mount to encase the radiation for maximum and optimal end usage. These anticipated concentrating solar collector systems are appreciated for their wide range applications as per specific topography and availability (Fernandez-Garcia *et al.* 2018). As per their shape and concentrating potential we have solar power tower systems, linear Fresnal reflectors, Scheffler dish, parabolic dish, and troughs, all having their specified working principle which offers a wide range of domestic and commercial applications. Parabolic trough collector (PTC) is one among the most widely used, a mature and efficient thermal conversion technology which uses its concavity to concentrate the available irradiance on to the linear focal mount. PTC being a reliable system has become the most opted technology for steam generation in factories and industries due to timely advancements and modifications.

PTC systems are capable of working in a wide range of 150 to 500 $^{\circ}$C depending upon the

advancements used in the system (Manikandan *et al.* 2019). The working range of the system temperature can be maximized by using modifications like improved glass cover, proper evacuation, and by using a hybrid system i.e. using PTC in combination with other power generation technologies like coal or diesel-based thermal power plant. The working range of the system can also be maximized by using advanced heat transfer fluids (HTFs) which may be either molten salts or metal suspended nano-fluids (Bellos & Tzivanidis 2019). In order to enhance the performance of the system, many researchers investigated PTC in different aspects: like improvising the design of the PTC or receiver tube, developing coatings for the receiver tube for better absorptivity and providing appropriate empirical methods to calculate the thermal losses and optimizing overall performance. Some researchers studied the effect of varying the mass flow rate in the receiver tube, some tried to vary the rim angle for the PTC system, while some have studied the effect of different HTFs in the system. This section gives the literature survey where researchers tried to enhance the system performance of the system by investigating one or more aspects stated above. In a study, Bhowmik & Mullick (1985) studied the performance of a PTC system having tubular absorber tube. They proposed an empirical approach to predict the absorber's heat loss factor, an important factor to understand the performance of system. The working temperature range was taken between 60 to $220°C$ for the system. A minor error of approximately 5% was reported in the results. Similarly, Mullick & Nanda (1989) calculated the heat loss factor with the help of empirical approach for a PTC receiver tube covered by a concentric glass cover. Qiu *et al.* (2017) combined the Monte Carlo ray Tracing (MCRT) and Finite Volume Method (FVM) to study a PTC system with s-CO_2 as HTF. Temperature difference on the circumference of the absorber tube was found to be 18 to 60 *K* while the optical efficiency has been reported to be 84.1%. Similarly, Odeh *et al.* (1998) used direct steam generation over an indirect steam generation in a PTC system and the system performance was calculated with the help of an equation. Agagna *et al.* (2018) predicted the accuracy and performance of the system by comparing their PTC model with an existing system while the results reported about 0.5% of uncertainty in the predicted performance where results were reported to be reliable. Behar *et al.* (2015) developed a PTC model to predict the performance of the system and compared the results to EES model made by National Renewable Energy Laboratory

(NREL). Results reported that uncertainty of 0.64% which was 1.11% in case of EES model.

Similarly, Lamrani *et al.* (2018) investigated a PTC system and compared it to existing system under transient environmental conditions. This study was carried out considering the effect of the mass flow rate of HTF and length of the absorber tube over the thermal performance of the system. The absorber tube used for the study was 0.0115 *m* in diameter, aperture width was 5 *m* while the focal length was reported to be 1.84 *m*. The system was reported to have a thermal efficiency of 76 % approximately. Okonkwo *et al.* (2018) studied the effect of nano-fluid made from olive leaf extract and barley husk as HTF in a PTC system. EES model shows that used nano-fluid is corrosion resistant and economic to produce when compared to other conventional HTFs. The results show a 0.073% improvement in thermal efficiency when water/BH-SiO_2 was used while a 0.077% enhancement has been reported when water/OLE-TiO_2 was used as HTF. Zaharil & Hasanuzzaman (2020) investigated a PTC system in MATLAB environment, taking six different HTFs for its energetic and exergetic comparison varying inlet temperature from 300 to 900 K. Results reported that out of six HTFs, liquid sodium was found to be better in terms of exergetic gains at 700 *K*. Shahdost *et al.* (2019) investigated technical as well as an economic assessment of a PTC system for fluid pre-heating in a gas refinery. The study model was prepared in TRNSYS which reports that using a PTC system can help in reducing conventional fuel consumption by 23%, annually. This reduction in fuel consumption is equivalent to reducing the production of harmful 3555 *tons* of CO_2. Thus helping in minimizing the greenhouse effect. Similarly, a PTC system was investigated in MATLAB/Simulink environment by Castellanos *et al.* (2020) to evaluate system performance when coupled with a hydraulic accumulator. It has been reported that the hydraulic accumulator is capable to provide a one-hour backup when solar irradiation decreases up to 14%.

Vouros *et al.* (2020) investigated different optical and numerical models in a PTC system in terms of performance factors. Parameters considered for the study are the incidence-angle modifier (IAM) and thermal losses. Results reported that the incidence-angle modifier has a huge impact on optical performance correlation. Small thermal losses do not cause much variation in the correlation because its magnitude is much smaller than

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the absorbed solar radiation. Bellos & Tzivanidis (2020) studied a LS-2 type PTC and tried to enhance the thermal performance by using a reflector shield (secondary reflector) and longitudinal fins. Study involved HTF (Syltherm 800) inlet temperature 350 to 650 *K* and inlet flow rates from 50 to 200 *litre/min*. Results reported enhancement in thermal performance in case of fins throughout the operation while the use of secondary reflector has been reported to be beneficial over high temperatures only. The combined use of fins and secondary reflector reported an enhancement of around 2.4%. Maatallah & Ammar (2020) studied a PTC system with dual reflector and proposed a 3-D model for the same. The study revealed that the system achieved maximum performance when the rim angle was 68°. The study also reported a drop of 7% optical efficiency but on the other hand reported an improved solar flux distribution i.e. solar flux distribution gradient was reduced by 86%, while the average local concentration has been reported as 25 *kW/m²* approximately. Similarly, Li *et al.* (2020) prepared an optical-thermal model for a PTC system insulated with solar transparent aerogel coating. It has been reported that optical efficiency has been reduced by 0.16% but receiver efficiency is enhanced by 0.012-75%.

The literature shows that a number of researchers have tried to enhance the output performance of the system. It is found that many researchers have tried to work on the design parameters to achieve better performance which has given a direction to this study.

1.2 Research gap and significance of the proposed work:

It has been found that there is not significant representation and generalization with regard to geometrical design or constructional constraints of adequate concavity of the parabolic surface in view of availability. Similarly, a lack of analytical interpretation for rating the adequate tube size corresponding to particular concavity has been addressed by the researchers. **Table 1** shows different considerations taken by various researchers for their investigations and also highlights some major gaps.

Reddy & Ananthsornaraj (2020) investigated a PTC system where the trough length was taken as 4.6 *m*, trough width 5.7 *m*, focal length 1.7 *m* while 80.3° rim angle. Literature shows that a lot of researchers have studied receiver tubes (different in diameter) used over a similar sized parabolic troughs (say 12.18 *m* in length and 5.7 *m* wide) which can be problematic for anyone to understand which tube will aid in optimum output. For example, Allouhi *et al.* (2018); Chandra *et al.* (2017) & Khakrah *et al.* (2018)

investigated PTC system, almost similar in size and shape while the used receiver tube is different in each case. All these researchers proposed their results but the gap prevailing here is that what would be the suitable diameter of the receiver tube for a particular PTC system. Many researchers have taken different rim angles, like Mwesigye & Meyer (2017) investigated on a PTC system taking 80° rim angle while Khakrah *et al.* (2018) studied similar (similar in terms of dimensions) PTC system but with 90° rim angle. Similarly, many other researchers have also used different rim angles for their study which leads to confusion in choosing a suitable rim angle for a particular PTC system.

Six different receiver tubes have been considered for present study to understand the significance of variation in measurements and dimensions of the receiver tubes. All the designs vary from each other in terms of receiver diameter or the number of glass envelopes. The vacuum is also provided in-between the receiver tube and the glass cover(s) in order to have high performance. This study also depicts the effect of varying the rim angle i.e. this study has been carried out for three different rim angles (*viz.* 40°, $80^{o,}$ and 120°). Thus the different parameters considered for this study include tube size, number of envelops, rim angle, inlet fluid temperature and inlet fluid flow rate. So a detailed analytical study has been presented in this communication.

Table 1: Research gaps and solution

S. No.	Reference	Study detail	Research gap	Solution in present article
1	(Okonkwo *et al.* 2018)	A bio-matter based HTF (olive leaf extract and barley husk based HTF) has been studied for a PTC system through EES model. Results show an improvement in performance by 0.07% approximately.	No role of design parameters has been stated in the study.	Design parameters are discussed and optimized in detail for a novel receiver tube system
2	(Zaharil & Hasanuzzaman 2020)	Six different HTFs (Pressurized water, therminol VP-1, Syltherm 800, Solar 16 salt, Hitec XL, and liquid sodium) for a PTC system with varying inlet temperature. The study revealed that the liquid sodium provides better exergy output at 700 *K*.	The thermal stability of the HTF for high temperatures at different mass flow rates has not been discussed in the study.	The importance of the thermal stability of the HTF is explained in the present study.
3	(Vouros *et al.* 2020)	Different optical and numerical models for a PTC system have been discussed and the results have shown that the incidence-angle modifier has a huge impact on the correlations to predict the system performance.	This study gives a brief knowledge about the IAM models but does not explain how to select an appropriate model for PTC systems that differ in terms of dimensions.	The present study discusses the effect of variation in PTC geometry (rim angles).
4	(Bellos & Tzivanidis 2020)	A combination of a secondary reflector and longitudinal fins has been investigated in a PTC with Syltherm 800 as a HTF. The study has reported an enhancement of 2.4% in overall performance but the use of secondary reflector	Although fins enhance the heat transfer between the absorber and the fluid but no discussion about the exergy loss associated with the turbulence has been discussed.	Results explain how the increased fluid flow rate beyond turbulent flow

		has led to lower efficiency at low-temperature values.	Also, no detailed discussion about the design parameters for a PTC system is provided.	causes variation in energy and exergy efficiencies.
5	(Zima *et al.* 2020)	A two-axis sun tracking PTC system with two U-tubes installed in a collector has been studied with the help of appropriate energy balance differential equations. The study shows an agreement between the developed model and the ANSYS Fluent results.	No discussion about how does the absorber behavior changes with varying fluid flow rates.	A range of inlet fluid temperature and flow rate variation for different tube size has been compared

2. Design configuration of single and double evacuated receiver tube systems

A conventional PTC used for medium high-temperature applications is generally cylindrical. It typically consists of a lustrous trough for reflection and linear metallic receiver to intercept the reflected radiations and other support structures. PTC systems can be either sun-tracking or fixed in a particular direction. A Sun-tracking system (single-axis or double-axis tracking) can be used depending upon the economic availability and application. The receiver tube is a selective coated metallic tube, covered with glass cover (generally borosilicate) and is evacuated for maximal thermal gains. The receiver- tube carries the heat transfer fluid as per the required application and is an essential component for PTC driven thermal systems.

In this communication, six different absorber tubes used for a PTC system are compared with each other analytically. The difference in absorber tubes is in terms of their tubular size and number of concentric covers. Results predicting the performance of the system are plotted with the help of various graphs, which optimizes an absorber tube out of all the studied designs. A stainless steel receiver tube is being considered for this investigation with several different designs as illustrated in **Table 2**. The receiver tubes are different in terms of the number of evacuated layers and diametric sizes.

Table 2: Different tube size with single and double glass configurations

S. No.	The outer diameter of the stainless steel receiver tube (m)	The outer diameter of borosilicate single glass cover (m)	The outer diameter of borosilicate double glass cover (m)	Total number of Glass Covers over the receiver tube
1.	0.07	0.115	-	01
2.	0.027	0.06	-	01
3.	0.016	0.027	-	01
4.	0.07	0.093	0.115	02
5.	0.032	0.047	0.06	02
6.	0.011	0.019	0.027	02

Fig. 1-2 represent a single evacuated receiver tube (SERT) with a corresponding thermal circuit as required for evaluating the loss coefficient. It shows the mode of heat transfer inside the receiver tube along with various temperature sites i.e. receiver temperature 'T_R', inside cover temperature 'T_{Ci}',

outside cover temperature 'T_{Co}', ambient temperature 'T_a', and sky temperature 'T_{sky}'. Fig. 1 (a and b) of SERT shows different dimensions and labels *viz.* receiver inside diameter (D_{Ri}), receiver outside diameter (D_R), inside cover diameter (D_{Ci}), and outside cover diameter (D_{Co}). The central section '*a*' represents the fluid flow area inside the metallic tube, '*b*' is the metallic tube, '*c*' is the evacuated region and '*d*' is the borosilicate glass cover. **Fig. 2** shows the thermal circuit for SERT which is labeled to explain temperature sites and the mode of heat transfer from the tube to the environment. All three modes i.e. radiation, conduction, and convection occur during operation.
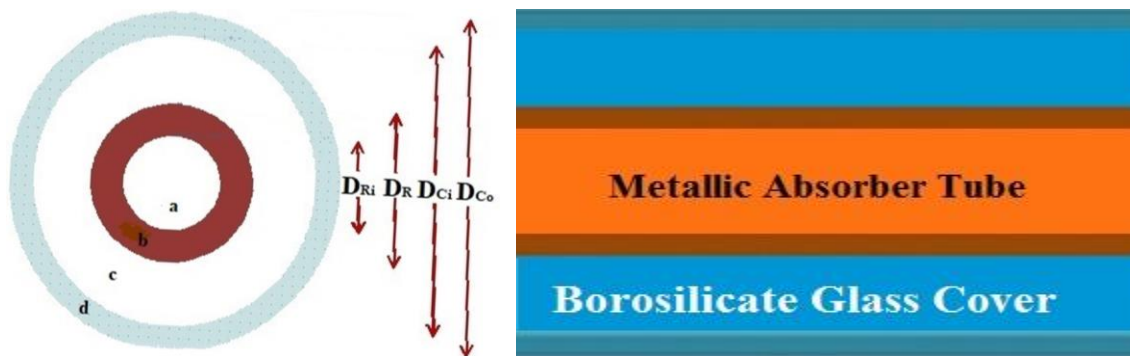
Fig. 1 a) Single evacuated receiver tube (SERT)   b) Sectional view
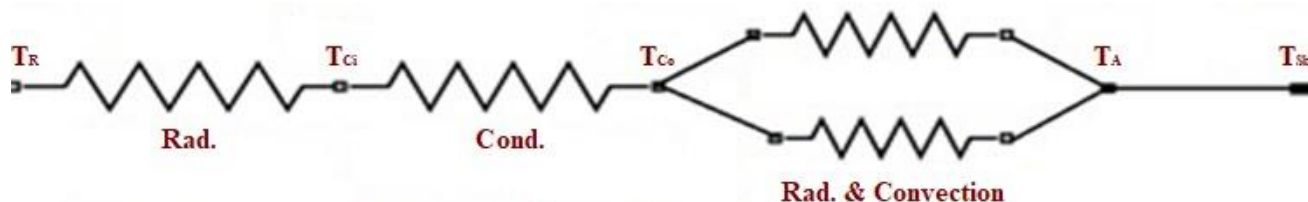
Fig. 2 Thermal Circuit Diagram (SERT)

Similarly, **Fig. 3** of a two-layered double evacuated receiver tube (DERT) shows two glass covers with receiver tube diameter being (D_r), inside diameter of the first cover (D_{c1i}), first cover outside diameter (D_{c1o}), second cover inside diameter ($Dc2i$) and second cover outside diameter ($Dc2o$). Section '*a*' is the fluid flow region while '*b*' and '*c*' represent evacuated regions, inside the dual covers over the receiver throughout the length of the tube. Some other labels in **Fig. 3** are temperature variables viz. receiver temperature 'Tr', first cover inside and outside temperature as '$Tc1i$ and $Tc1o$', second cover inside and outside temperature as'$Tc2i$ and $Tc2o$', ambient temperature outside the tube 'Ta' and sky temperature '$Tsky$'. The thermal circuit diagram for DERT is well illustrated in **Fig. 4** which clearly shows different modes of heat transfer. It also shows various temperature variables that

predict heat loss factor, energy gain/loss, and other parameters related to solar collectors (Duffie & Beckman 2013). These equations compare and contrast the overall performance of different receiver tubes for a PTC system having fixed aperture, trough length, and rim angle which are 5.7 *m*, 12.27 *m*, and 80°, respectively. This analytical investigation depicts the results for dimensions and other parameters based on literature study i.e. standard reference size, length, width and diameter have been taken as parametric measurements. There are still some standard parameters that need to be assumed and these assumption may vary as per the nature of the material, climatic conditions, and required application. Based on assumptions given in **Table 3**, the analytical study of all the considered receiver tubes has been carried out and the results are presented in graphical form which shows a significant performance difference for all the receiver tubes.

Table 3: Assumed parameters

Parameter	Assumed Value
Receiver emissivity (ε_r)	0.31
Emissivity of cover (ε_c)	0.88
Stefan-Boltzmann constant (σ)	5.67E-08
Stainless steel receiver tube conductivity (k) [W/m^2 ^{o}C]	19
Ambient temperature (T_a) [K]	293
Transmittance-Absorptance product ($\tau\alpha$)	0.88
Glass cover conductivity (k) [W/m^2 ^{o}C]	0.0226
Therminol VP-I heat transfer coefficient [W/m^2 ^{o}C]	300
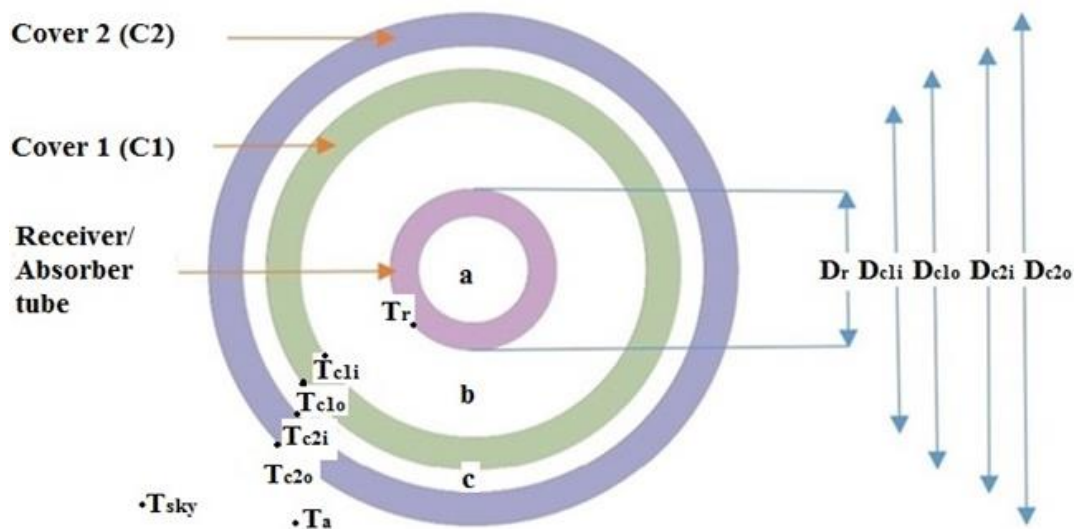
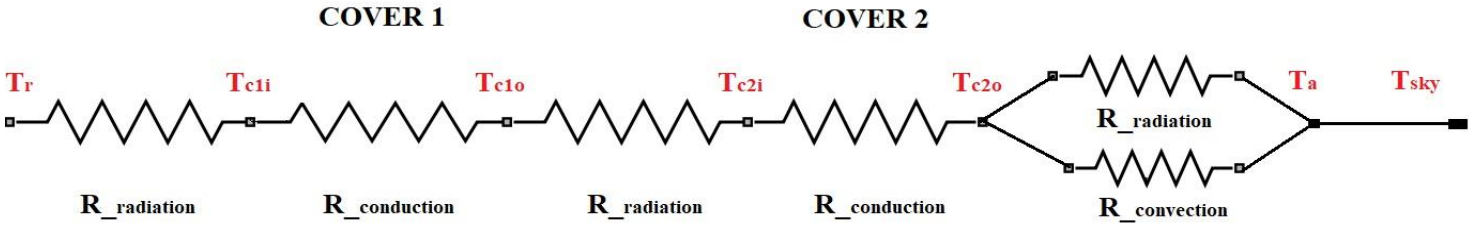**Fig. 3** Double evacuated receiver tube (DERT)

Fig. 4 Thermal circuit diagram (DERT).

3. Governing Equations and performance parameters

This section accounts for the fundamental equations to compare the performance parameters of the proposed receiver tube design configurations to be mounted on preferable sized PTC. The incident radiation over the receiver tube is trapped inside the glass cover which results in heating of the fluid aided with an adequate vacuum. The glass cover is evacuated from inside resulting in high thermal gains as dissipative losses are minimized. The receiver/absorber tube is the most essential part of a PTC system through which heat transfer fluid (HTF) flows and absorbs the heat from the incident radiations. The proposed configuration of the receiver tubes can be modeled as per the thermal circuit illustrated in fig. 2 and 4 with the corresponding number of covers used. The figures 1 and 3 represent the design configurations of single evacuated (SERT) and double evacuated receiver tube (DERT).

3.1. Energy analysis: Since modeling of optical parameters enables us to utilize the maximum of incoming radiations and solar concentration; it is essential to take care of energy balance and different losses across the receiver tube from the inner core to the sky. So, mathematical equations are used to model the receiver tube so that various losses, gains, and other parameters can be predicted. Receiver tube modeling is important as after a certain limit of high heat absorption for a longer period, the receiver tube itself may become emissive and it can lead to certain losses in the system. This modeling study will help in optimizing receiver tubes and various parameters associated with them like tube diameter, cover diameter, and the number of covers. This modeling results in predicting overall loss coefficient (U_L), collector efficiency factor (F'), collector heat removal factor (F_R), useful energy gain (Q_u), and system efficiency (η) which collectively decide the performance of the receiver tube and ultimately the PTC system. This empirical approach

is carried out by calculating thermal losses (Q_{loss}) of the receiver tube by considering the length and diameter of the tube, the temperature at various points of receiver tube/cover, emissivity, and modes of heat transfer. So, the thermal losses of SERT are given in eq^n. (1) as:

$$Q_{loss} = \frac{(T_r - T_{ci}) \times (\pi D_r L\sigma) \times (T_r^2 + T_{ci}^2) \times (T_r + T_{ci})}{\frac{1}{\varepsilon_r} + \frac{(1 - \varepsilon_c)}{\varepsilon_c} \times \left(\frac{D_r}{D_{ci}}\right)} = \frac{(T_{ci} - T_{co}) \times 2\pi K_c L}{\ln\left(\frac{D_{co}}{D_{ci}}\right)}$$

$$= (T_{co} - T_{sky})$$

$$\times \left(\pi D_{co} L h_c + \varepsilon_{co} \pi D_{co} L\sigma (T_{co}^2 + T_{sky}^2)\right.$$

$$\left. \times (T_{co} + T_{sky})\right) \dots \dots \dots \dots \dots \dots \dots eq^n(1)$$

For DERT, due to an additional layer of a glass cover over the receiver tube, the thermal loss equation changes. So, eq^n. (2) is used to calculate the losses in case of DERT as:

$$Q_{loss} = \frac{(T_r - T_{c1i}) \times (\pi D_r L\sigma) \times (T_r^2 + T_{c1i}^2) \times (T_r + T_{c1i})}{\frac{1}{\varepsilon_r} + \frac{(1 - \varepsilon_{c1i})}{\varepsilon_{c1i}} \times \left(\frac{D_r}{D_{c1i}}\right)} = \frac{(T_{c1i} - T_{c1o}) \times 2\pi K_{c1} L}{\ln\left(\frac{D_{c1o}}{D_{c1i}}\right)}$$

$$= \frac{(T_{c1o} - T_{c2i}) \times (\pi D_{c1o} L\sigma) \times (T_{c1o}^2 + T_{c2i}^2) \times (T_{c1o} + T_{c2i})}{\frac{1}{\varepsilon_{c1o}} + \frac{(1 - \varepsilon_{c2i})}{\varepsilon_{c2i}} \times \left(\frac{D_{c1o}}{D_{c2i}}\right)}$$

$$= \frac{(T_{c2i} - T_{c2o}) \times 2\pi K_{c2} L}{\ln\left(\frac{D_{c2o}}{D_{c2i}}\right)}$$

$$= (T_{c2o} - T_{sky})$$

$$\times \left(\pi D_{c2o} L h_c + \varepsilon_{c2o} \pi D_{c2o} L\sigma (T_{c2o}^2 + T_{sky}^2) \times (T_{c2o} + T_{sky})\right) \dots \dots eq^n(2)$$

Some other important equations for the PTC system modeling are for calculating heat loss factor (U_L) given in eq^n. (3), collector efficiency factor (F') given in eq^n. (4), collector heat removal factor (F_R) given in eq^n. (5) and (6), useful energy gain (Q_u) given in eq^n. (7) and efficiency is given in eq^n. (8). The heat loss factor can be calculated using thermal losses, the receiver area, and the temperature difference between the receiver and the ambient.

$$U_L = \frac{Q_{loss}}{\pi D_r L \times (T_r - T_a)} \dots \dots \dots \dots \dots eq^n(3)$$

This U_L value is used to form the equation of collector efficiency factor (F′) as:

$$F' = \frac{1/U_L}{\frac{1}{U_L} + \frac{D_o}{h_{fi}D_i} + \left(\frac{D_o}{2k}\ln\frac{D_o}{D_i}\right)} \quad \dots\dots\dots\dots\dots eq^n(4)$$

Parameters involved in calculating the "collector efficiency factor" are the loss coefficient of the receiver, diameter of the tube, and all the envelopes along with the thermal conductivity of the tube. Similarly, collector efficiency factor is used in calculating the "collector heat removal factor (F_R) " as:

$$F_R = \frac{\dot{m}C_p}{A_rU_L}\left[1 - \exp\left(-\frac{A_rU_LF'}{\dot{m}C_p}\right)\right] \dots\dots\dots\dots\dots eq^n(5)$$

The collector heat removal factor can also be found by another relation, stated below as:

$$F_R = \frac{\dot{m}C_p(T_{fo} - T_{fi})}{A_a\left[S - U_L\frac{A_r}{A_a}(T_{fo} - T_{fi})\right]} \dots\dots\dots\dots\dots eq^n(6)$$

Now to find "useful energy gain" (Q_u), the value of the collector heat removal factor is required. So, 'Q_u' can now be evaluated with an equation stated as:

$$Q_u = F_rA_a\left[S(\tau\alpha) - \frac{A_r}{A_a}U_L(T_{fi} - T_a)\right]\dots\dots\dots\dots\dots eq^n(7)$$

When all the required factors for performance assessment of a PTC are available, system efficiency (η) is found out that too analytically. In this investigation, the system efficiency is calculated using the useful energy gain (Q_u), solar radiation, and aperture area. The efficiency of the PTC system considered for this investigation in mathematical form is given as:

$$\eta = \frac{Q_u}{S \times A_a} \dots\dots\dots\dots\dots eq^n(8)$$

3.2 Exergy analysis

Exergy analysis of a system is very useful as it enables us to look for system inefficiencies i.e. it helps us monitor the degradation of energy quality from time to time. So, exergy

efficiency is important because it gives us more clarity about the nature of dissipative effects and its remission measure as compared to energy analysis. This paper shows how the exergy efficiency of a typical PTC system can vary if receiver tube design and size is varied on different rim angles. Exergetic efficiency is defined as the ratio of exergy gain to the exergy of input solar radiation (Allouhi *et al.* 2018); (Chandra *et al.* 2017); (Khakrah *et al.* 2018). It is represented in the form of an equation below:

$$\eta_{ex} = \frac{\dot{m} \int_{T_i}^{T_o} C_p(T)dT - T_a \int_{T_i}^{T_o} \frac{C_f(T)}{T}dT}{W_a L G_{bt} \left[1 - \frac{4}{3}\left(\frac{T_a}{T_{sun}}\right) + \frac{1}{3}\left(\frac{T_a}{T_{sun}}\right)^4\right]} \dots\dots\dots\dots\dots eq^n(9)$$

Exergy efficiency of the system in equation (9) can be simplified as:

$$\eta_{ex} = \frac{\dot{m}C_p \left[(T_{fo} - T_{fi}) - T_a \times \ln\left(\frac{T_{fo}}{T_{fi}}\right)\right]}{I \times A_a \times (0.93)} \dots\dots\dots\dots\dots eq^n(10)$$

4. Heat transfer fluid

HTF is a thermal fluid used to store the thermal energy coming from the sun in the form of radiations which are made incident over the receiver tube. The heating effect of radiations heats-up the receiver tube and the HTF flowing through the tube absorbs that energy which increases the HTF temperature to a higher extent due to high concentration. This heated HTF is passed through water to make high-temperature steam out of it. This steam is further used for various applications as per requirement whether it's for electricity generation or space heating. A lot of synthetic oils and molten salts used as HTFs are easily available in the market as per requirement and working temperature. For this particular study, Therminol VP1 has been taken as a HTF. The reason behind this being the high working temperature range of Therminol VP1 and this study is carried out for a temperature range between 400 to 600 *K*. Therminol VP1 being suitable for a working temperature range up to 700 *K* approx.

Allouhi *et al.* (2018) and Bellos *et al.* (2016) have presented equations to calculate required properties like density, specific heat capacity, thermal conductivity, and dynamic viscosity of the HTF, which are as under:

The density of the HTF (ρ_{HTF}) is calculated in Kg/m^3 with the relation:

$$\rho_{HTF} = -2.379 \times 10^{-6} T^3 + 0.002737 \times T^2 - 1.871\,T + 1439 \ldots\ldots\ldots\ldots\text{eq}^\text{n}\,(11)$$

Specific heat capacity of the HTF (C_p) is calculated in J/Kg. K with the following equation as:

$$C_p = 8.877 \times 10^{-6} T^3 + 0.01234 \times T^2 + 8.28\,T - 50.85 \ldots\ldots\ldots\ldots\text{eq}^\text{n}\,(12)$$

Thermal conductivity (W/m K) of the HTF is calculated as:

$$\lambda_{htf} = 1.062 \times 10^{-11}\,T^3 - 1.937 \times 10^{-7}\,T^2 + 2.035 \times 10^{-5}\,T + 0.1464 \ldots\ldots\ldots\text{eq}^\text{n}\,(13)$$

Similarly, **Dynamic Viscosity (Pa.s)** can be calculated as:

$$\mu_{htf} = 30.24 \exp(-0.03133\,T) + 0.008808 \exp(-0.006729\,T) \ldots\ldots\ldots\text{eq}^\text{n}\,(14)$$

In the above equations 11 to 14, the variable 'T' is the temperature of the fluid during its use in the PTC system. These equations are used in calculating the system performance throughout the operation as all these factors are required for system modeling.

5. Geometrical consideration of parabolic trough

Parabolic trough surface design directly influences the performance and the working range of a PTC system. Designing a parabolic trough involves various assumptions and measurements. So, to optimize system performance, it is necessary to model the adequate concavity of the trough through some novel interpretative and empirical approaches. Hence parameters like receiver tube size (Dr), focal (f), reflector rim angle (φ_r) and reflector aperture (A) are required. Some of the factors are to be assumed while others have to be found out by using empirical equations. Rim angle and receiver diameter are given in eq$^\text{n}$. (15) and eq$^\text{n}$. (16)

$$\varphi_r = \tan^{-1}\left[\frac{8(f/a)}{16(f/a) - 1}\right] \ldots\ldots\ldots\ldots\ldots\text{eq}^\text{n}\,(15)$$

Here, f/a is the focal length to aperture ratio.
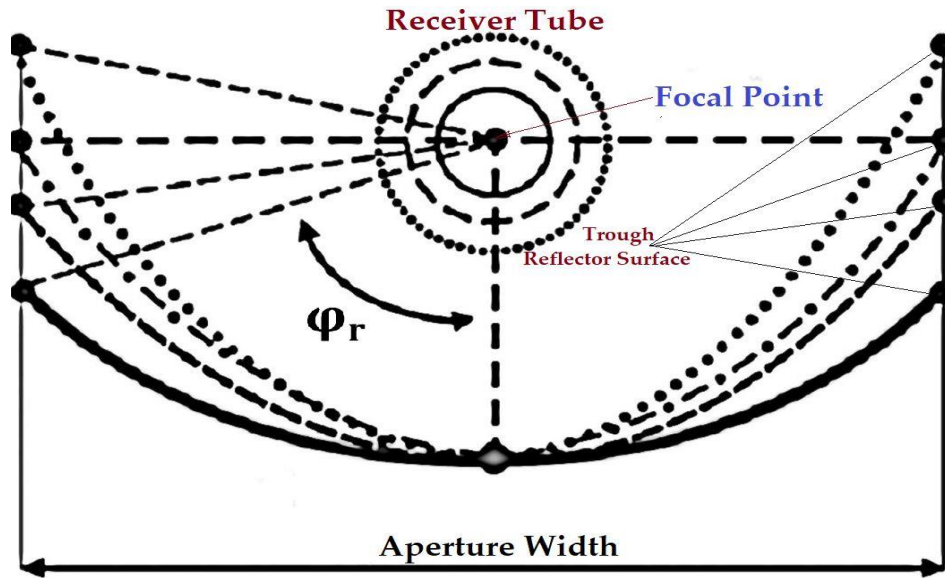
Fig. 5 Rim Angle (Hoseinzadeh *et al.* 2018).

Once rim angle for the PTC is calculated, diameter of receiver tube can be calculated as:

$$D = \frac{a \sin 0.267}{\sin \varphi_r} \dots \dots \dots \dots \dots \dots \dots \dots \dots \text{eq}^n \ (16)$$

Different rim angles (i.e. 40°, 80°, and 120°) have been checked for the optimal geometric configuration of the system corresponding to all the receiver tube designs as detailed in table 2. The aperture width and the length of the PTC collector considered for this study have been selected as a standard LS-3 type design because many researchers have investigated PTC systems having almost similar dimensions as stated in this article. Researchers like (Allouhi *et al.* 2018; Khakrah *et al.* 2018; Allouhi *et al.* 2018) (Khakrah *et al.* 2018) (Bellos *et al.* 2017) & (Reddy *et al.* 2012) have studied PTC system with similar dimensions in terms of aperture width and length.

6. Results validation

This study is carried out using a numerical cum iterative program made in the MS Excel 2013 sheet, the equations and assumptions were put into the program, and iterations were made to find out various results. A reference study, carried out by Bellos *et al.* (2017) and the present study results were found relatable and within permissible variation of 10 %, which validates the reliability of this study. All the dimensions and standard measurements of the reference system as well as the present system are given in **Table 4**. The comparison stats of reference system to the present system parameters are shown below in **Fig. 6**.
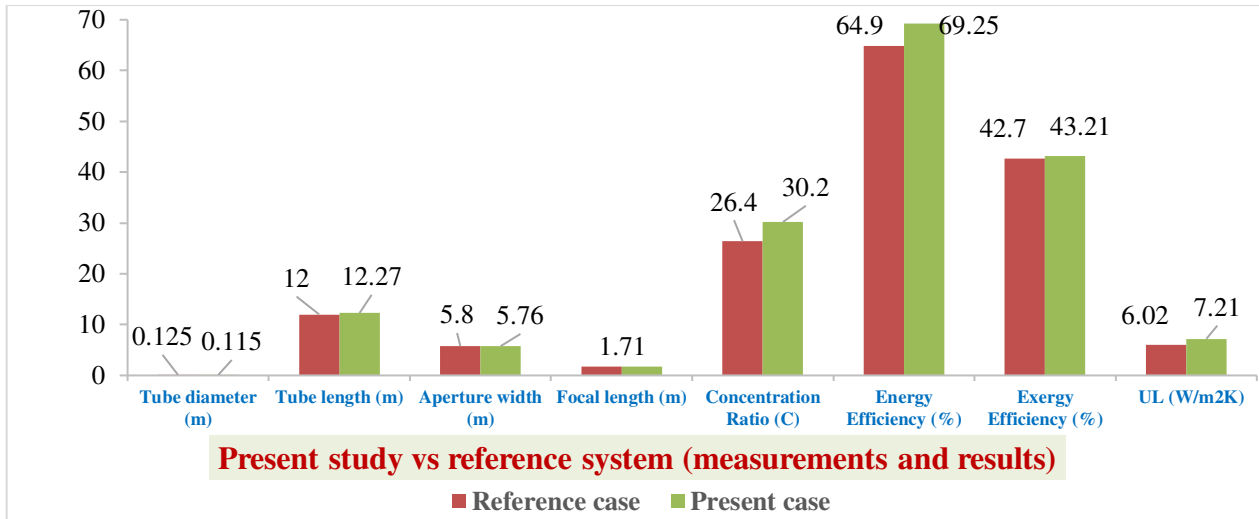
Fig. 6 Validation of the present study with reference [Bellos *et al.* (2017)] system.

The results of the present study are presented in **Table 4-7** and illustrated in **Fig 7-14** for optimal configuration against standard size. The tubes considered for this study are all indifferent from each other in terms of size, number of covers and vacuum gap over the metallic tube, so it is obvious that the results will vary but this variation will provide us with a suitable size/dimension of a receiver tube with most efficient results. For this study, as stated earlier a fully developed laminar flow of Therminol VP1 was forced inside the receiver with varying flow rates (16 to 216 *litre/hr*) and varying inlet temperature (323 *K*, 373 *K, 423 K, 523 K, 623 K*, and 723 *K*) to account for energy and exergy gains at different rim angles (40°, 80°*,* and 120°). The flow rate variation is considered to study the effect of heating inside the tube on both laminar-turbulent regimes for all receiver sizes.

6.1. Effect of heat transfer fluid:

Heat transfer fluids (HTFs) are thermal fluids, which are used to maximize energy encapsulation or storage. HTFs can be paraffin, molten salts, or any other metal suspended nano-fluids. A lot of different HTFs have been investigated and studied by the researchers while a few of them are presented here. Barbosa *et al.* (2020) investigated a PTC system with an aim to make a cost effective system using thermo-syphon regime instead of pump powered flow in the receiver tube. The fluids used for this study were water and thermal oil. The results reported that thermal oil as HTF has maximum energy gain. Similarly, Said *et al.* (2020) used coating made from a composition of grapheme oxide and cobalt oxide ($rGO-Co_3O_4$) hybrid nano-fluid with water to investigate the energetic behaviour of the

system. The study revealed that addition of graphene and cobalt based nanoparticles in water as HTF enhances its thermo-physical properties. Another example where, Martinez-Merino *et al.* (2020) studied the potential of two-dimensional Tungsten-Diselenide (2D-WSe$_2$) based HTF in PTC, a concentrating solar power (CSP) system. It is evident that, an increase in thermal conductivity and heat transfer thus enhancement in energy efficiency has been reported in the results. In a similar study, Rafiei *et al.* (2020) studied three different HTFs *viz.* water, thermal oil and air. It is evident that for high temperature applications, heat transfer fluid plays an important role in improving the system performance. Results reported that the thermal performance of the system was found to be better in case when thermal oil as compared to water. It is also reported that the losses are reduced in case HTF is air, while water was found least efficient of the three HTFs. **Table 4** illustrates the significance of different heat transfer fluids with Therminol VP1 being used for the present study yielding highest output with better stability until 723 *K* at optimal size mounted on 80° concave surfaced PTC (Reddy & Ananthsornaraj 2020). It is highly stable and doesn't change its physical properties with continuous and repeated temperature fluctuations.

Table 4: Role of heat transfer fluid in energy and exergy output of the PTC system.

S. No.	Reference	HTF used	PTC parameters	Energy efficiency (%)	Exergy efficiency (%)
1	(Reddy *et al.* 2012)	Therminol VP1	Aperture : 5.76 m; absorber tube length : 12.27 m Rim angle : NA; Concentration ratio (C): 25 Focal length : 1.71 m; Vacuum thickness : 0.0195 m	66.8	39.1
2	(Yilmaz & Soylemez 2014)	Syltherm 800	Aperture : 1.14 m; Absorber tube length : 3 m Rim angle : 77°; Concentration ratio (C): 9.37 Focal length : 0.36 m; Vacuum thickness : NA	69.4	---
3	(Bellos *et al.* 2017)	CO_2	Aperture : 5.8 m; Absorber tube length : 12 m Rim angle : 77°; Concentration ratio (C): 26 Focal length : 1.71 m; Vacuum thickness : 0.025 m	78	45.9
4	(Khakrah *et al.* 2018)	Al_2O_3/Synthetic oil	Aperture : 3.4 m; Absorber tube length : 12.18 m Rim angle : 90°; Concentration ratio (C): 15.46 Focal length : 0.88 m; Vacuum thickness : 0.02 m	68	38
5	(Zheng *et al.* 2019)	Synthetic oil	Aperture : 2.5 m; Absorber tube length : 50 m Rim angle : NA; Concentration ratio (C): 7.8 Focal length : 0.85 m; Vacuum thickness : 0.027 m	70	---
6	(Reddy & Ananthsornaraj 2020)	Therminol VP1	Aperture : 5.77 m; Absorber tube length : 4.06 m Rim angle : 80.3; Concentration ratio (C): 26.3 Focal length : 1.71 m; Vacuum thickness : 0.0245 m	70	---
7	Present study (optimum case)	Therminol VP1	Aperture : 5.76 m; Absorber tube length : 12.18 m Rim angle : 80; Concentration ratio (C): 43.9	79.4	45.9

			Focal length : 1.71 m; Vacuum thickness : 0.0025 m		

6.2. Effect of fluid inlet temperature

Fluid inlet temperature is an important parameter to be considered while assessing output of a PTC system. Inlet fluid temperature can result in variation in output performance of the system. This study reports the effect of fluid inlet temperature and shows how the inlet fluid temperature plays an important role in output performance of the system. This study includes analysis at different fluid inlet temperatures (323 K, 373 K, 423 K, 523 K, 623 K, and 723 K) for the receiver tube. And the fluid inside the receiver tube has to be circulated in the system and undergo cycles of heating and cooling. **Fig. 11** shows that the efficiency increases steadily with an increase in inlet fluid temperature up to 523 K and drops afterward as per high irreversible thermal losses. Similarly, varying fluid inlet temperature also affects the exergy of the system. The exergy curves show that as the fluid inlet temperature is increased the curve goes to rise and then steadily falls down. This study shows that the drop in energy and exergy efficiency is to be encountered beyond a temperature value suitable for a particular system. This can be well understood by **Table 5,** which compares a trend between results found from the optimum absorber tube (**0.027 m**) and the reference size (**0.115 m**) receiver tube. It clearly shows that the trend for energy & exergy efficiency is different in case of a **0.027 m** diameter tube and different in a **0.115 m** diameter tube.

6.3. Effect of fluid flow rate in laminar to turbulent regimes

It is clear that in **Fig. 7 (a & b)**, **Fig. 8 (a & b)**, and **Fig. 9 (a & b)** the system shows an increase in efficiency with increasing flow rate. The efficiency curve for 0.115 m (reference size) and 0.027 m (optimal) diameter tubes gradually becomes flat at a flow rate of 170 and 140 *litres/hr* respectively. The reason for this drop is "reduced heat transfer between fluid and the tube due to two envelopes over large-sized tubes i.e. the flow becomes more turbulent and less heat is absorbed. **Fig. 7 to 9** show a variation in exergy efficiency with varying flow rates. A continuous drop in exergy efficiency is noticed with increasing flow rate while maximum exergy efficiency is noticed in the case of 0.027 m diameter double-covered tube with all the fluid inlet temperatures. **Table 6** shows a comparison between efficiencies of the reference absorber tube and the studied tube at varying flow rates. It shows that when the fluid flow is laminar, the energy and exergy are showing a symmetrical curve but when fluid flow becomes turbulent, the response of the

energy efficiency as well as exergy efficiency suddenly flatten and exergy efficiency goes down to minimum. Thus, it is evident that the fluid flow rate plays a significant role in portraying the performance parameters of the system. So, it is valid to say that if the fluid inlet flow rate is not kept steady for a particular system it can result in performance losses and if it is kept below or above the desired value, it can cause energy efficiency drop and exergy to go to the bottom line.

6.4. Effect of tube size

This section explains how the size of the receiver tube can affect the energy and exergy efficiency of the system. Many researchers mentioned earlier in the literature study, used different sized receiver tubes and reported different energy and exergy efficiencies. **Table 4** also cites some literature which shows how different lengths and size of the system affect the performance of a PTC system. This study has proved that the receiver size is one of the most important parameter when it comes to maximum gains in terms of efficiency. **Figure 10 (a)** shows the effect of tube size on the energy efficiency. It is clearly visible that decreasing tube size results into more energy efficiency while increasing tube size will result in reduction in energy efficiency. It means, every PTC system has a particular tube size and if the used receiver tube is larger than the preferable tube size, efficiency will be low. **Table 7** shows the case where, the PTC system has an aperture width of **5.76 *m*** and trough length of **12.18 *m***. It shows that, different tube sizes affect the efficiency of the system, i.e. if the diameter is increasing, efficiency shows a constant drop. Similarly, **Fig. 10 (b)** shows the behaviour of exergy efficiency with different tube size. So, energy as well as exergy are maximum with smaller diameter receiver tubes and minimum for larger diameter tubes when used for a same aperture width and trough length PTC system.

As this study investigates the effect of double evacuated receiver tubes, it is visible in **Fig. 10 (a & b)** which shows performance of different receiver tubes. **Fig. 10 (a & b)** shows that 0.027 *m* diameter absorber tube with double evacuated cover shows better results among all other designs. The trend lines in **Fig. 12** show an interesting variation of energy efficiency of the system as per varying exergy and fluid flow rate compared to the optimal size receiver tube and the reference size receiver tube.

6.5. Effect of rim angle variation and significances

This section gives detailed results of the present investigation in terms of significance of the rim angle on the PTC system. Rim angle is responsible for appropriate concentration of the solar radiations on the receiver tube so it is an important parameter which has to be deduced as per the system. The literature stated in this communication makes it clear that different PTC systems have different rim angle i.e. a common rim angle cannot be used for any type if PTC system. Rim angle is supposed to vary with the varying aperture area i.e. every aperture and trough size has an optimal concavity for which the system performs at the high-end output. In this study, the given graphs show how a single cover receiver tube and double cover receiver tube differ in performance when the rim angle is varied. Graphs shown in **Fig. 13** and **14** represent results which clearly shows that for the investigated PTC system, 80° rim angle (φ_r) possess the better energy as well as exergy efficiency. The average highest energy efficiency value is reported in the case of DERT, approximately 79.4 % with corresponding exergy efficiency being 47%. **Fig. 13 (a & b)** and **14** show a visual comparison of energy efficiency to exergy efficiency along with varying rim angle for all six receiver tube size at an average flow rate of 100 *litre/hr* and inlet temp (323 – 373 *K*).

Table 5: Efficiency variation in optimum and standard absorber tube with varying inlet fluid temperature.

S. No	Inlet fluid temperature (K)	Energy Efficiency of optimum (0.027 m diameter) tube (%)	Energy efficiency of standard (0.115 m diameter) tube (%)	Exergy Efficiency of optimum (0.027 m diameter) tube (%)	The exergy efficiency of standard (0.115 m diameter) tube (%)
1	323	77.08	69.94	35.2	31.3
2	373	78.2	70.8	38.7	34.5
3	423	78.95	71.5	41.5	37.2
4	523	79.4	72.3	45.75	40.7
5	623	78.9	72	45.9	40.75
6	723	78.1	71	45.91	40.79

(Note: In the present study, 0.027 m diameter double cover absorber tube is found optimum absorber tube while 0.115 m diameter absorber tube is considered as the standard tube).

S. No.	Flow rate (litre/hr)	Energy Efficiency of optimum absorber tube (%)	The energy efficiency of standard absorber tube (%)	Exergy Efficiency of optimum absorber tube (%)	The exergy efficiency of standard absorber tube (%)	Flow Type for optimal size
1	16	72	63.35	51.2	31.48	Laminar
2	41	74	64.85	48.56	29.51	Laminar
3	66	76	66.32	46.1	27.35	Laminar
4	91	78	67.65	42.85	25.1	Laminar
5	116	80	68.9	40.2	23.43	Transition flow
6	141	82	70.1	38.1	21.4	Transition flow
7	166	84	71.2	35.35	18.85	Turbulent

| 8 | 191 | 84.5 | 71.95 | 32.2 | 14.63 | Turbulent |
| 9 | 216 | 84.5 | 72.7 | 28.6 | 11.2 | Turbulent |

Table 6: Variation of energy and exergy efficiency compared between optimal and standard sized tube in laminar and turbulent flow regimes (at 50-100°C inlet)

Table 7: Efficiencies of various tube diameters

S. No	Tube size (m)	No. of Covers	Energy Efficiency (%)	Exergy Efficiency (%)
1.	0.115	02	77.2	44.3
2	0.06	02	78.05	44.9
3	0.027	02	79.4	46.9
4	0.115	01	75.01	41.5
5	0.06	01	75.9	42.14
6	0.027	01	76.4	43.17
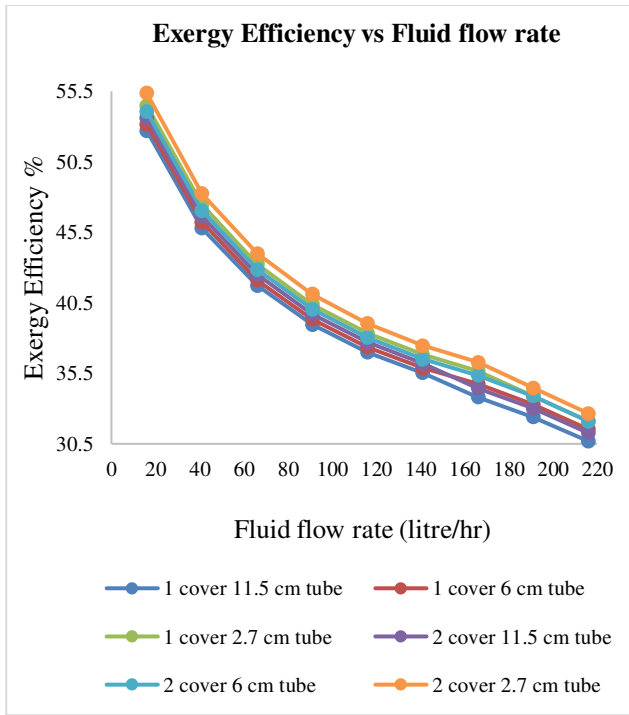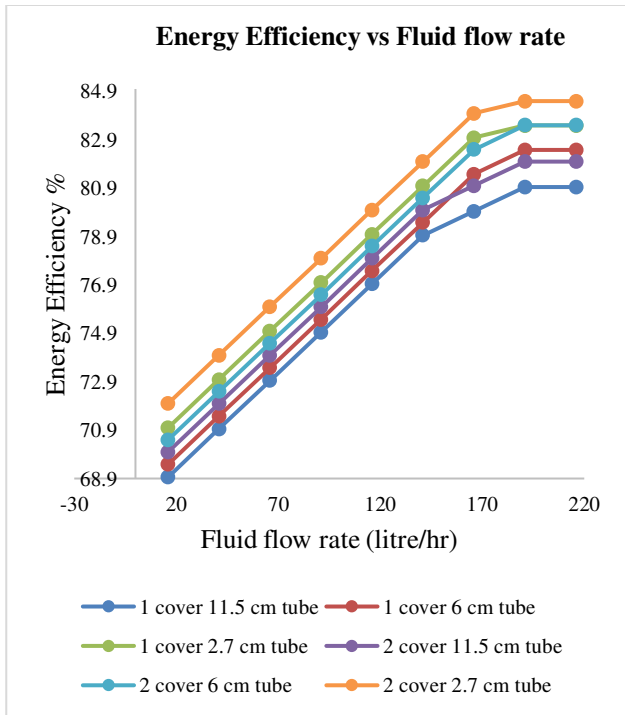
Fig. 7 a) Exergy Efficiency vs Fluid Flow Rate   b) Energy Efficiency vs Fluid Flow Rate

(At 50 ^{o}C fluid inlet temperature)
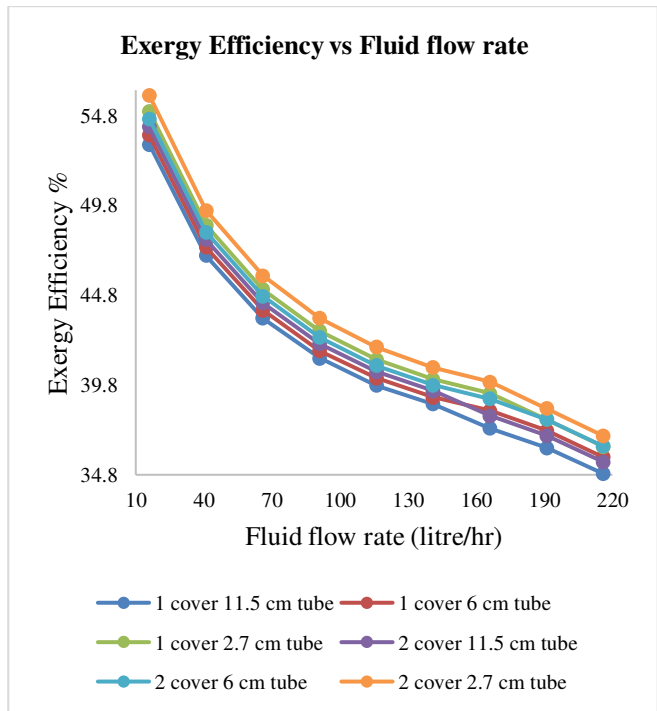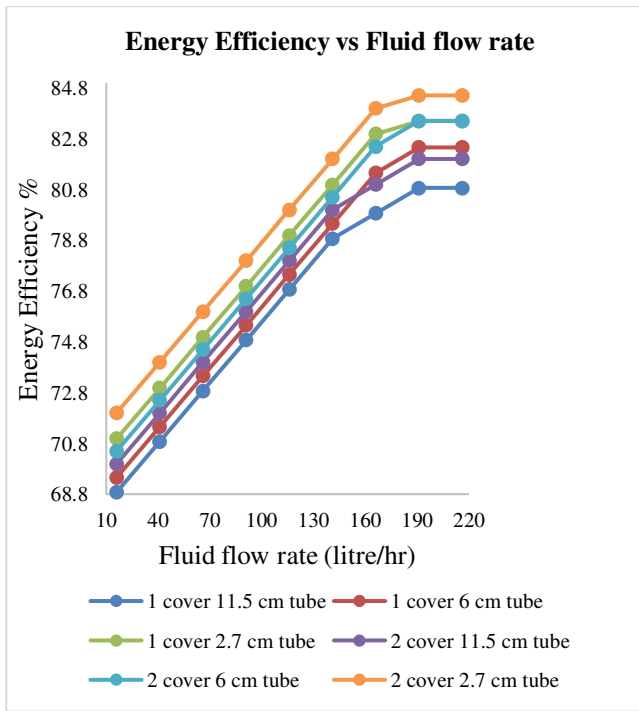
Fig. 8 a) Energy Efficiency vs Fluid Flow Rate   b) Exergy Efficiency vs Fluid Flow Rate (At 100 ^oC fluid inlet temperature).
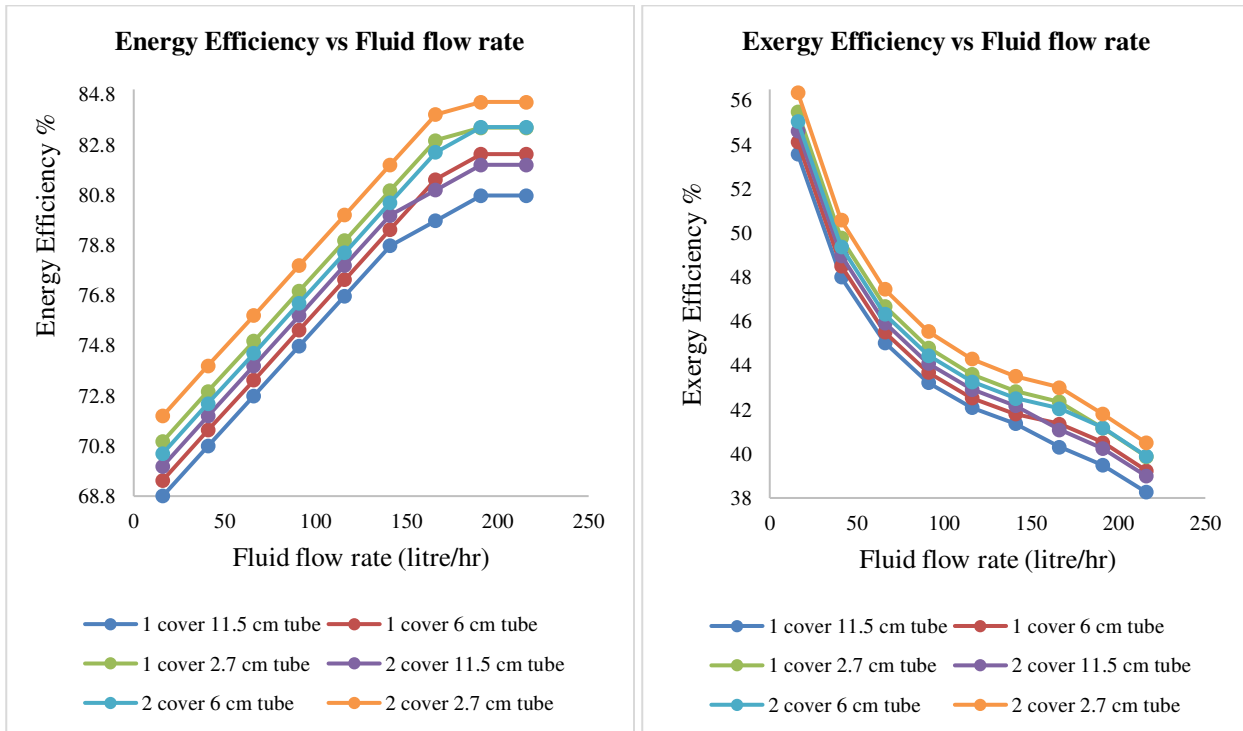
Fig. 9 a) Energy Efficiency vs Fluid Flow Rate   b) Exergy Efficiency vs Fluid Flow Rate (At 150 ^oC fluid inlet temperature).
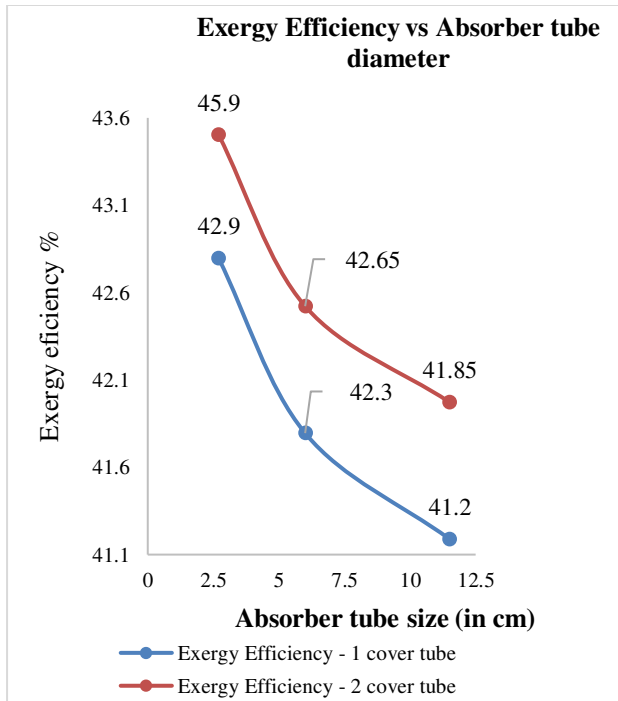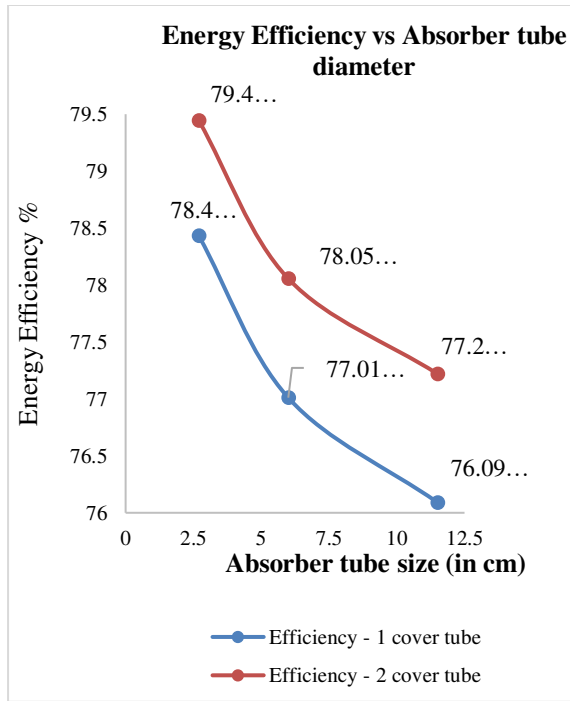
Fig. 10 a) Energy Efficiency vs tube diameter   b) Exergy Efficiency vs tube diameter
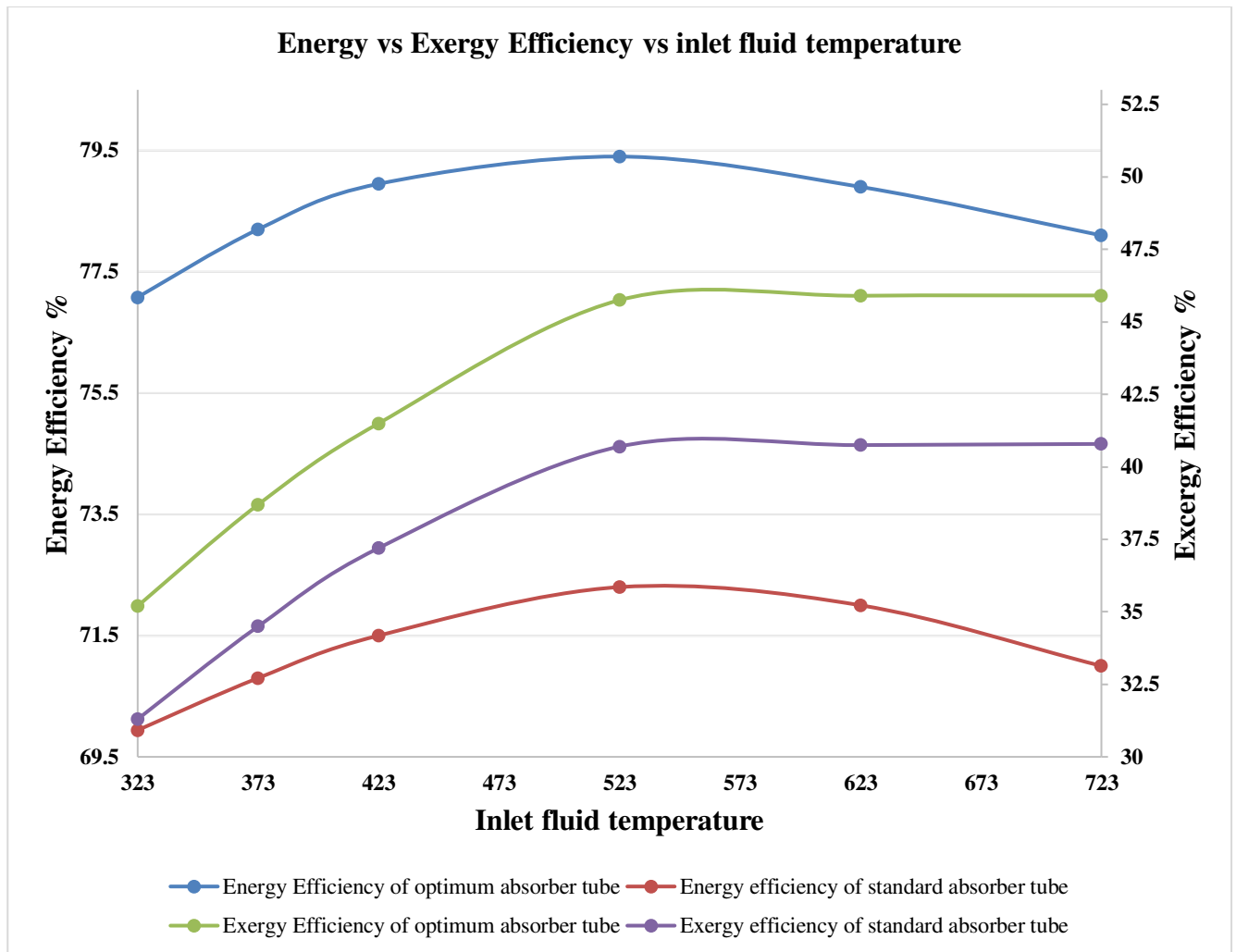
Fig. 11 Inlet fluid temperature vs energy and exergy of system
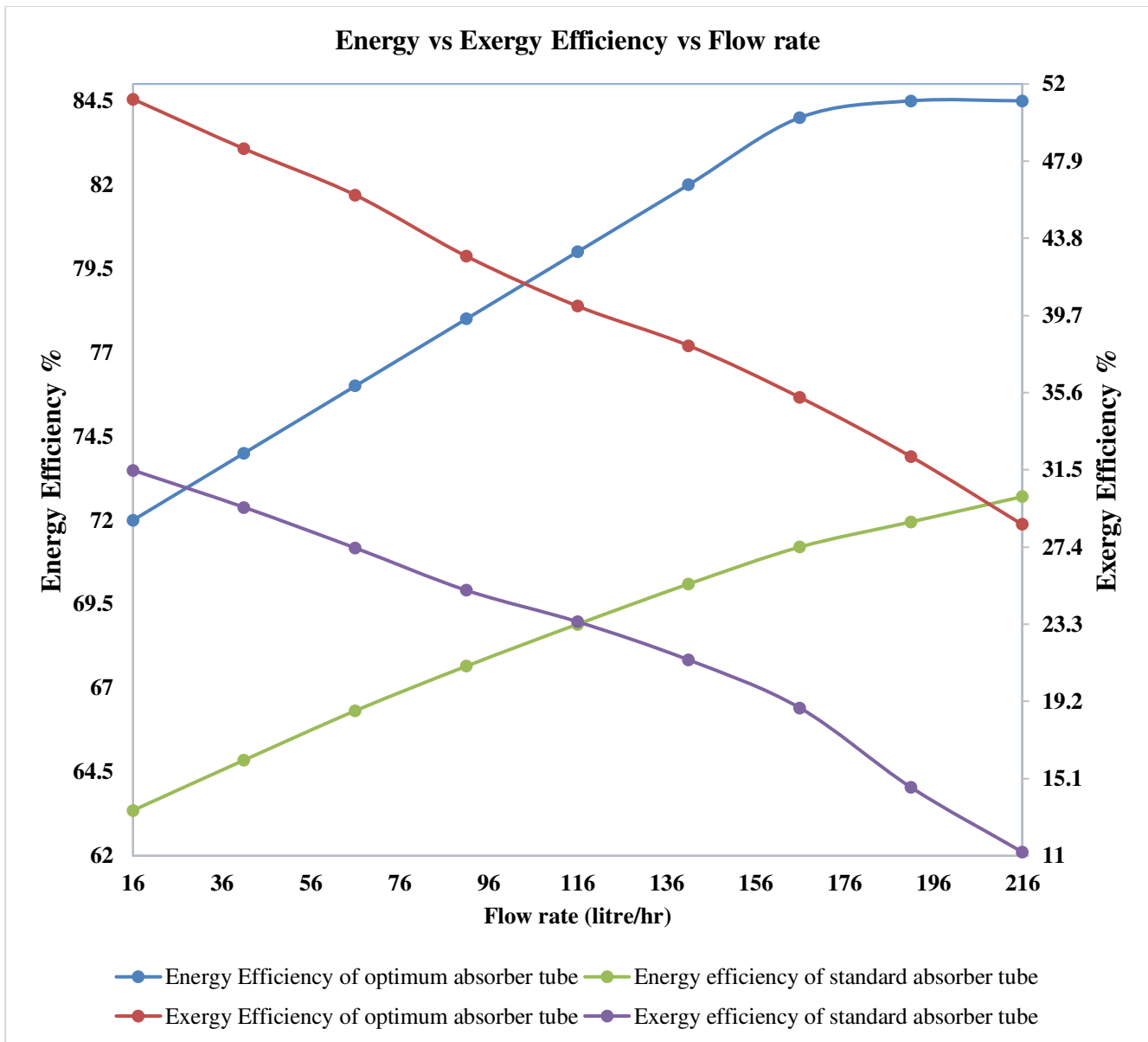
Fig. 12 Energy vs Exergy efficiencies vs Inlet flow rate
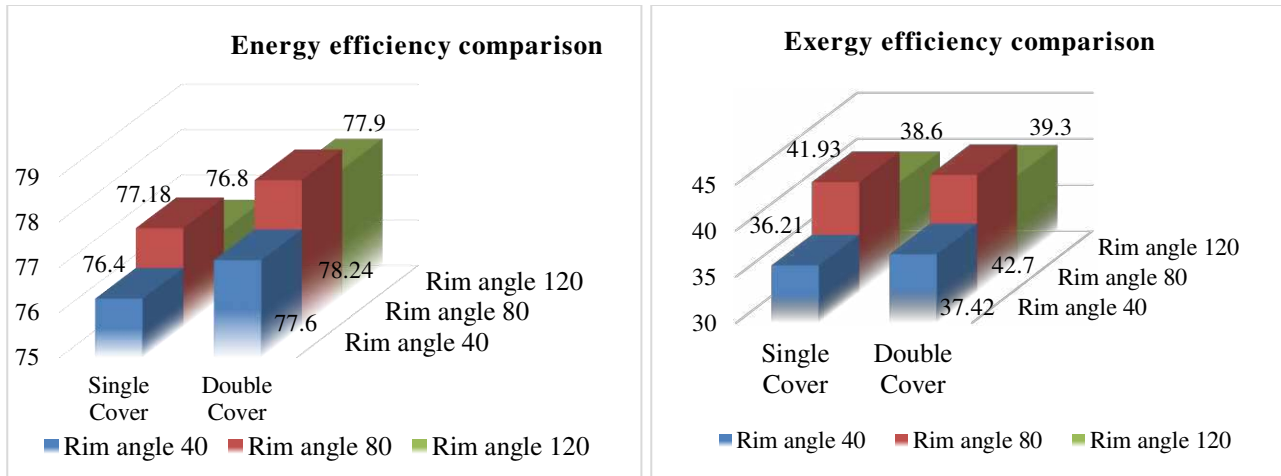
Fig. 13 (a) Energy efficiency vs Rim angle (b) Exergy efficiency vs Rim angle
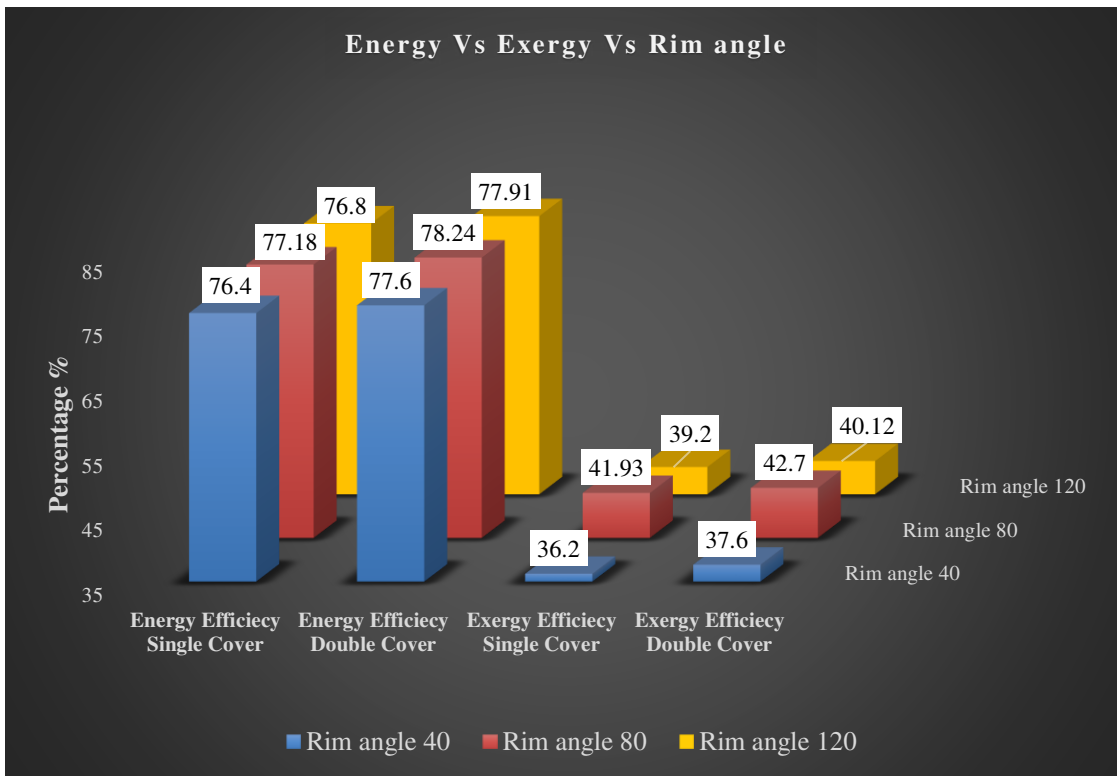
Fig. 14 Energy efficiency vs exergy efficiency vs rim angle

7. Conclusion

This communication gives a detailed comparison of the various thermal and geometrical profiles of a novel receiver tube mounted parabolic trough collector system in view of optimal receiver tube size,

rim angle, fluid flow rate, and inlet fluid temperature while using Therminol VP1 as a heat transfer fluid. The aforementioned results can be summarized as:

1. The energy efficiency of the system with 0.027 m diameter tube was found to increase with increasing inlet fluid temperature up to 523 K followed by a gradual decline afterward attributed to finite temperature gradient and irreversible conversion losses. Similarly, energetic variation was found to grow almost constant.

2. Energy efficiency increases linearly with increasing flow rate until the fluid flow is transformed into a turbulent regime. Similarly, exergy efficiency reduces gradually with flow marked by a significant reduction in turbulent transitions. Due to different diameters of the receiver tubes, all the transition flow rates are different i.e. 120, 150, and 200 *litre/hr* for 0.027 m, 0.06 m, and 0.115 m diameter receiver tube, respectively.

3. The illustration in Fig.10 is quite useful to rate the flow configuration of the both optimal and standard sized tube. The intersection of energy and exergy curves at 100 *litre/hr* for 0.027 m and 120 *litre/hr* for 0.115 m sized receiver was found to be reasonable as per the fully developed laminar profile. This flow rate identifies appreciable energy and exergetic gains from the system, thus avoiding both extremities.

4. A two-layered DERT system having a diameter 0.027 m was found to be best suited for the most commonly used parabolic trough with aperture and length being 5.7 and 12.27 m respectively mounted on optimized geometry of 80°.

5. The highest energy and exergy efficiency reported from the proposed hydrodynamic, thermal, and geometric consideration of a novel receiver tube mounted on a typical PTC system were 79.4 and 45.9 % respectively.

Challenges and future research directions

This study has addressed the key issues encountered while designing a PTC system for maximum and constant output but still has some challenges and future directions which need to be worked upon are listed below.

i.   The receiver tube can be studied for different composite materials.

ii.   Cost analysis of the system can be a topic of research.

iii.   Thermal losses due to emissive nature of the receiver tubes can be worked upon by using different inserts inside the receiver tube.

Conflict of Interest Statement: The author(s) declare(s) that there is no conflict of interest.

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Figures
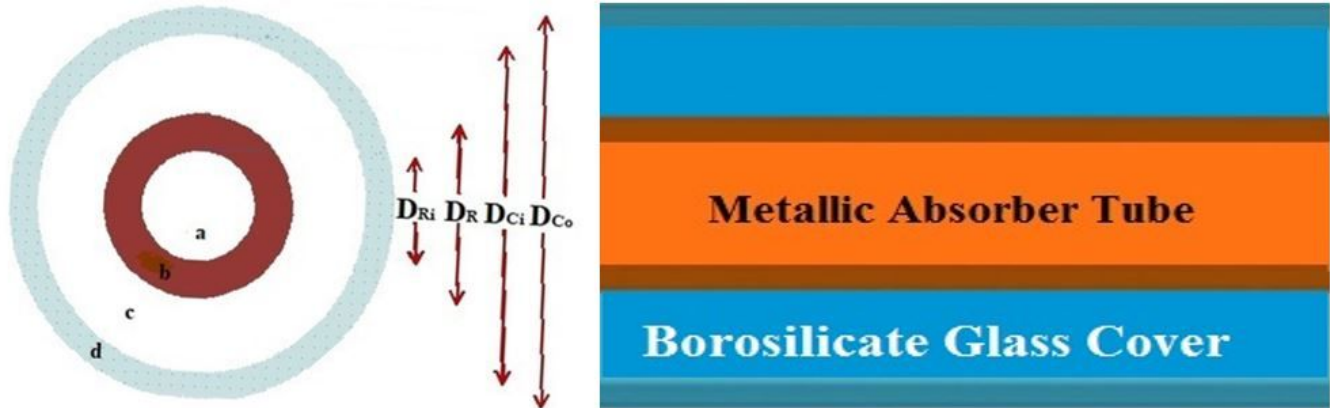
Figure 1

a) Single evacuated receiver tube (SERT) b) Sectional view
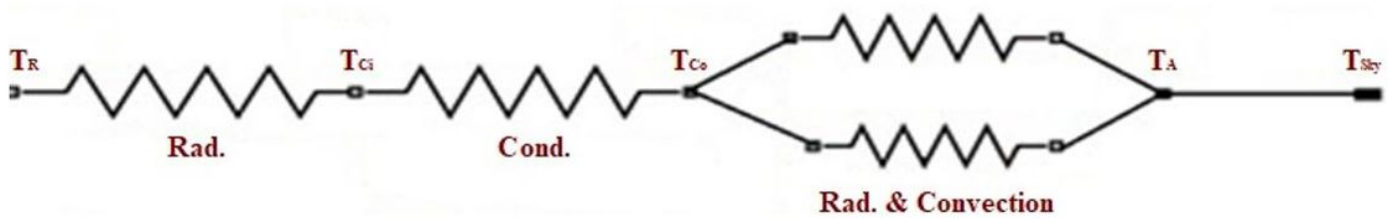
Figure 2

Thermal Circuit Diagram (SERT)
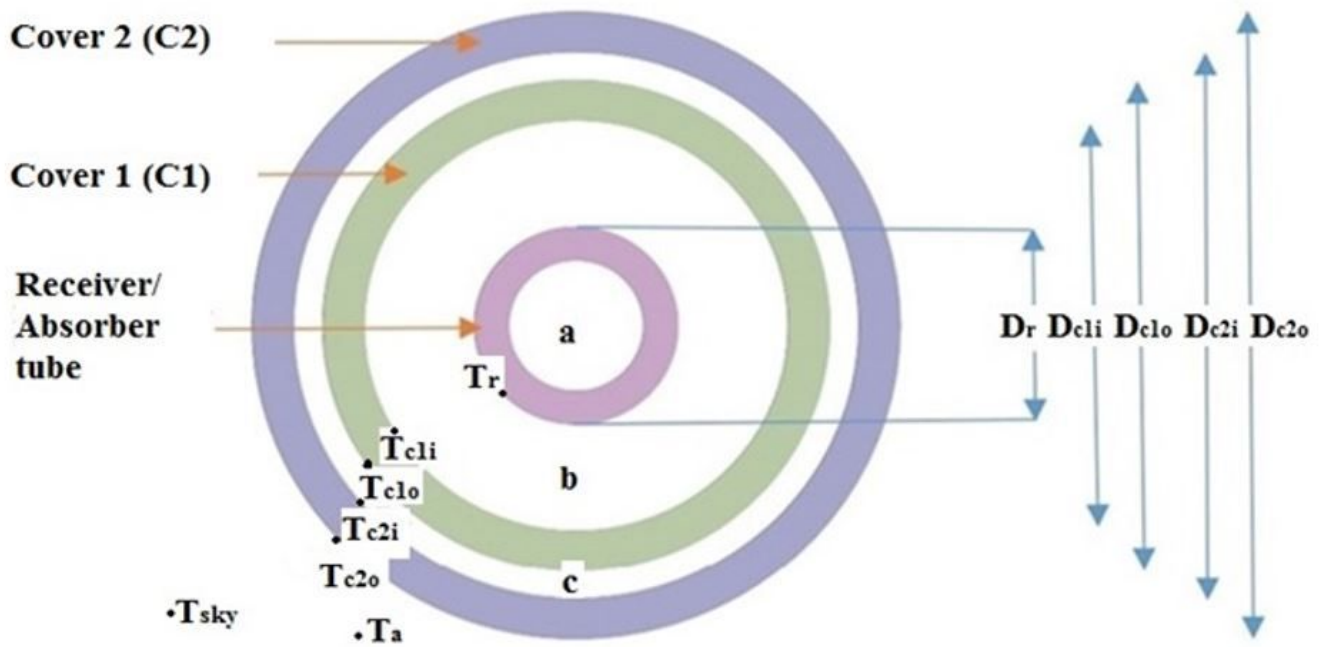
Figure 3

Double evacuated receiver tube (DERT)
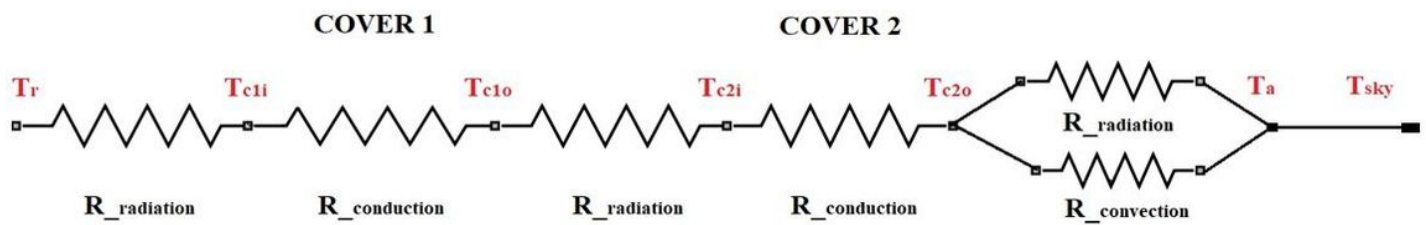
Figure 4

Thermal circuit diagram (DERT).
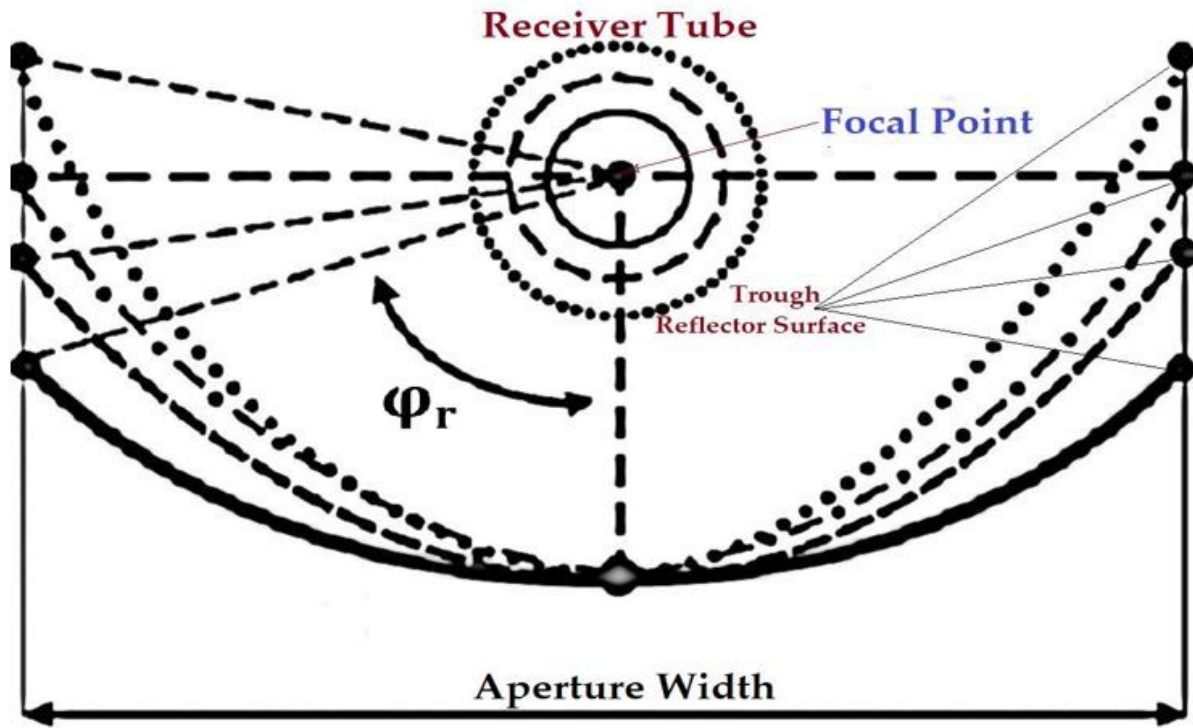
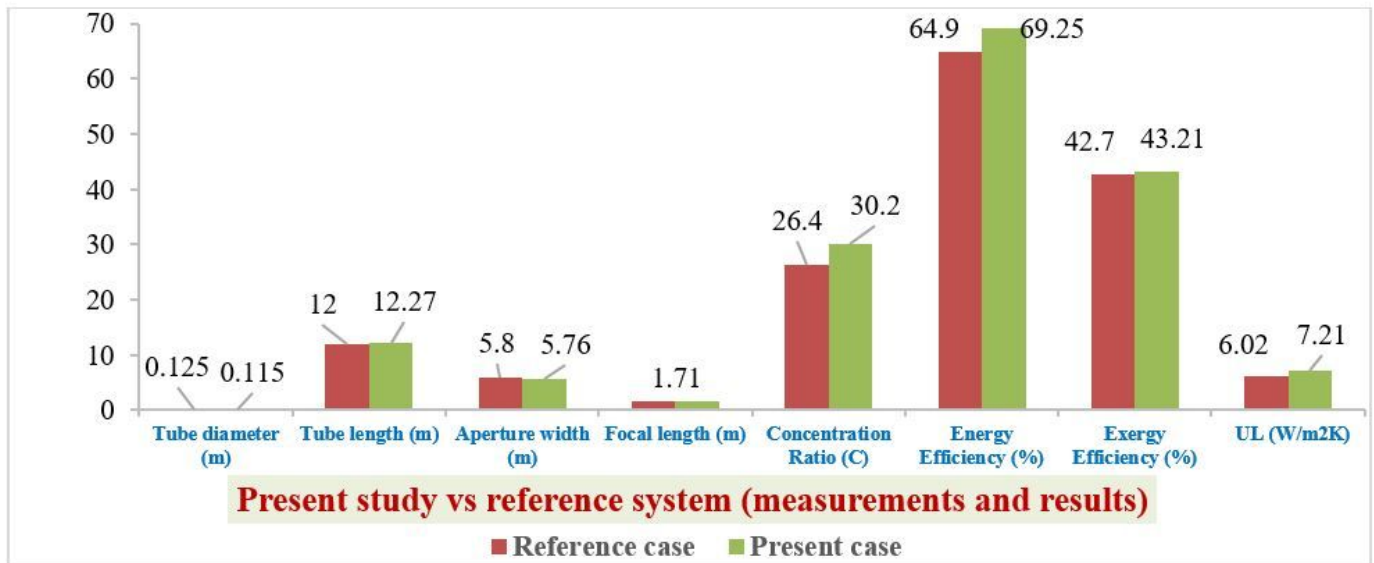**Figure 5**

Rim Angle (Hoseinzadeh et al. 2018).

Figure 6

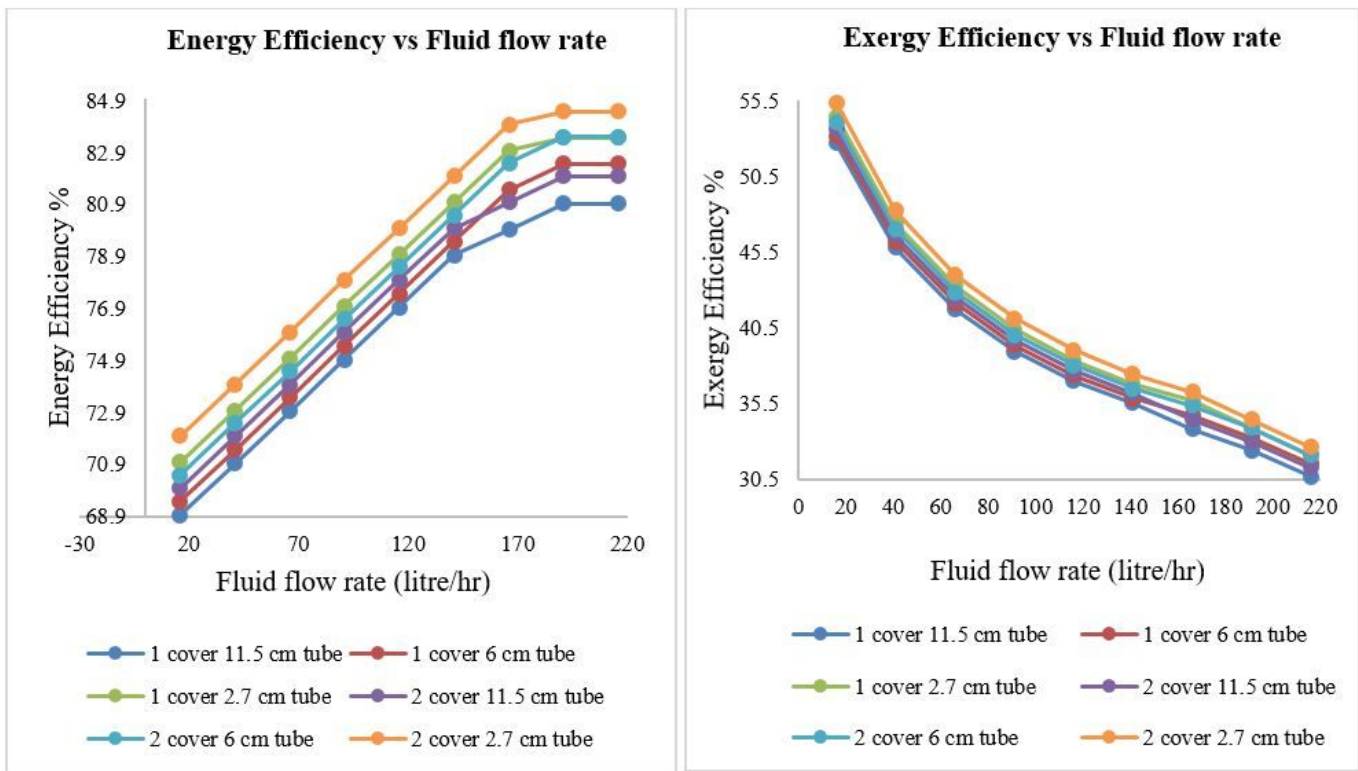Validation of the present study with reference [Bellos et al. (2017)] system.

Figure 7

a) Exergy Efficiency vs Fluid Flow Rate b) Energy Efficiency vs Fluid Flow Rate (At 50 oC fluid inlet temperature)
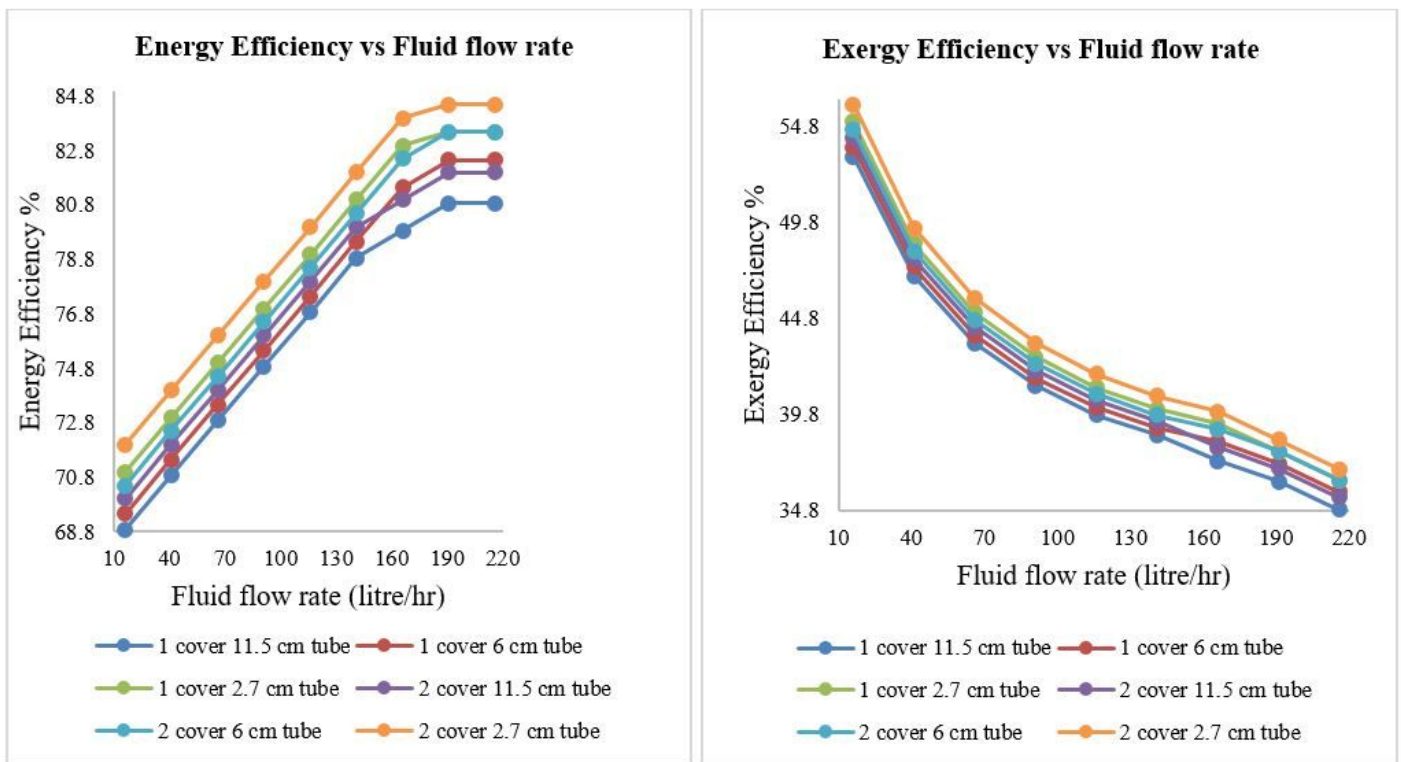
Figure 8

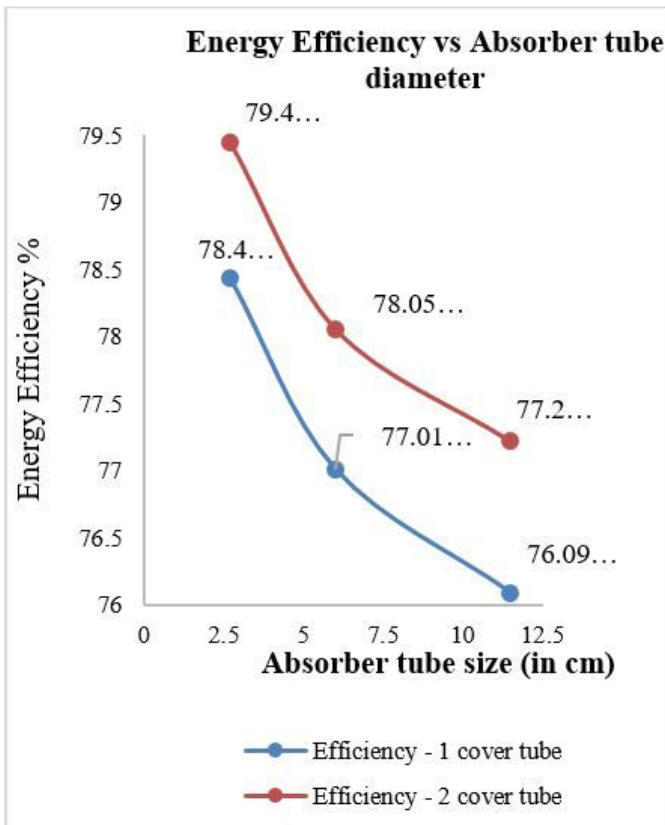a) Energy Efficiency vs Fluid Flow Rate b) Exergy Efficiency vs Fluid Flow Rate (At 100 oC fluid inlet temperature).
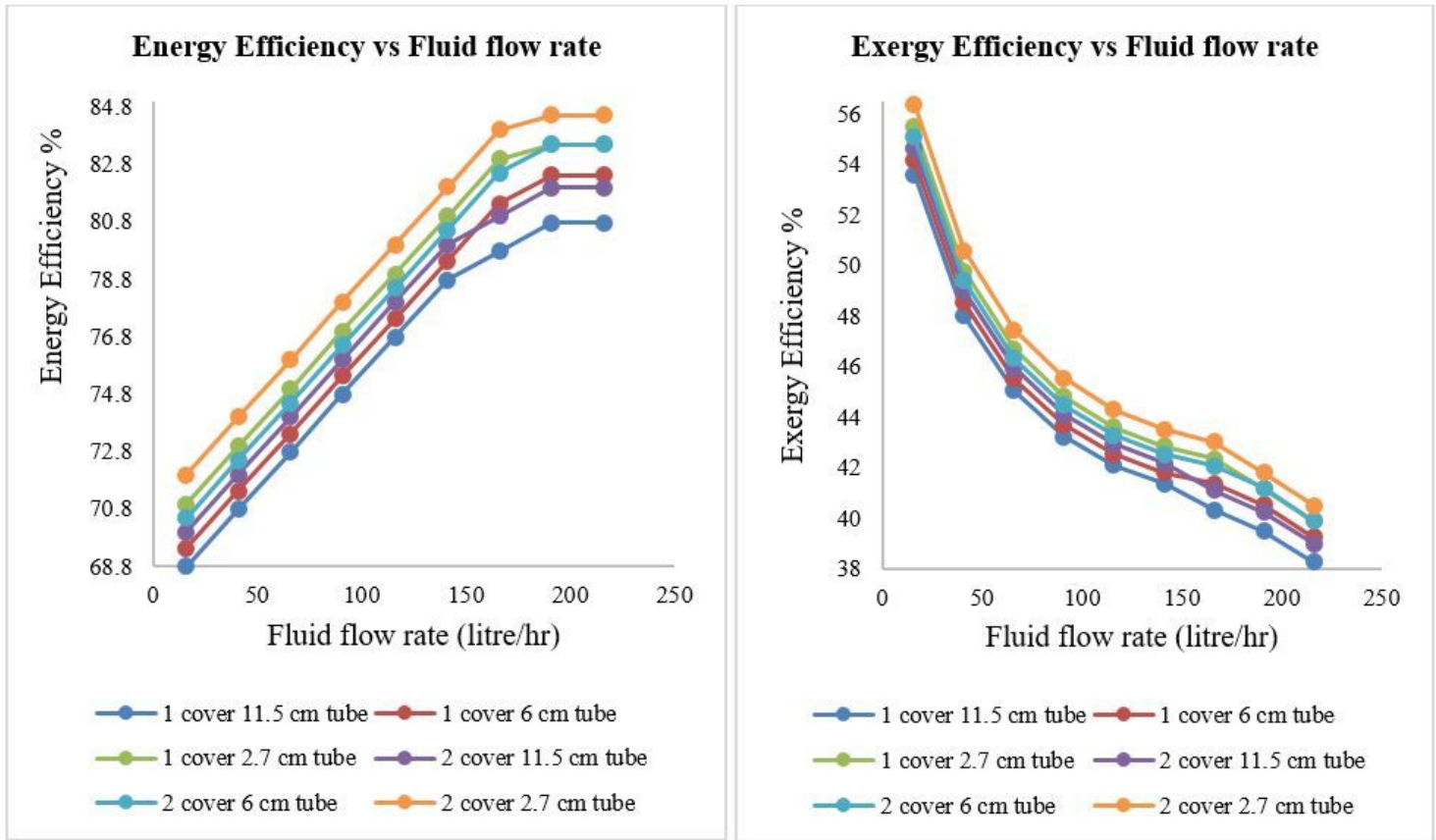
Figure 9

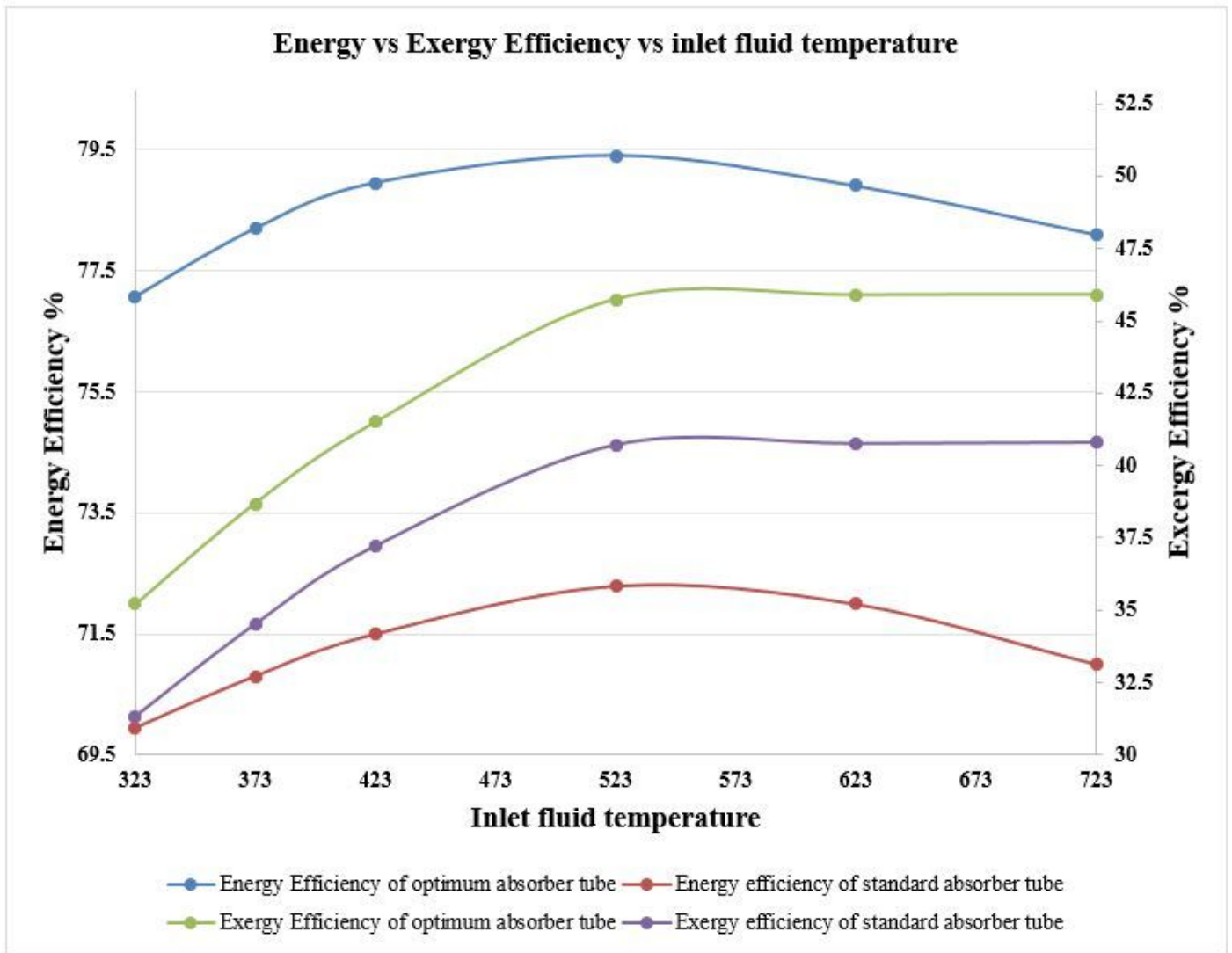a) Energy Efficiency vs Fluid Flow Rate b) Exergy Efficiency vs Fluid Flow Rate (At 150 oC fluid inlet temperature).
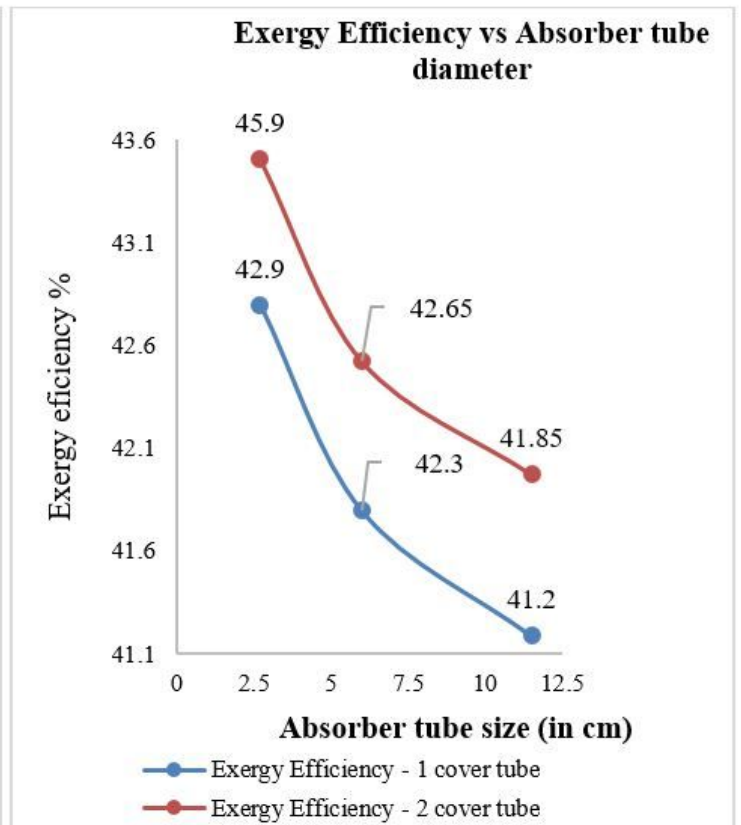
Figure 10

a) Energy Efficiency vs tube diameter b) Exergy Efficiency vs tube diameter

Figure 11
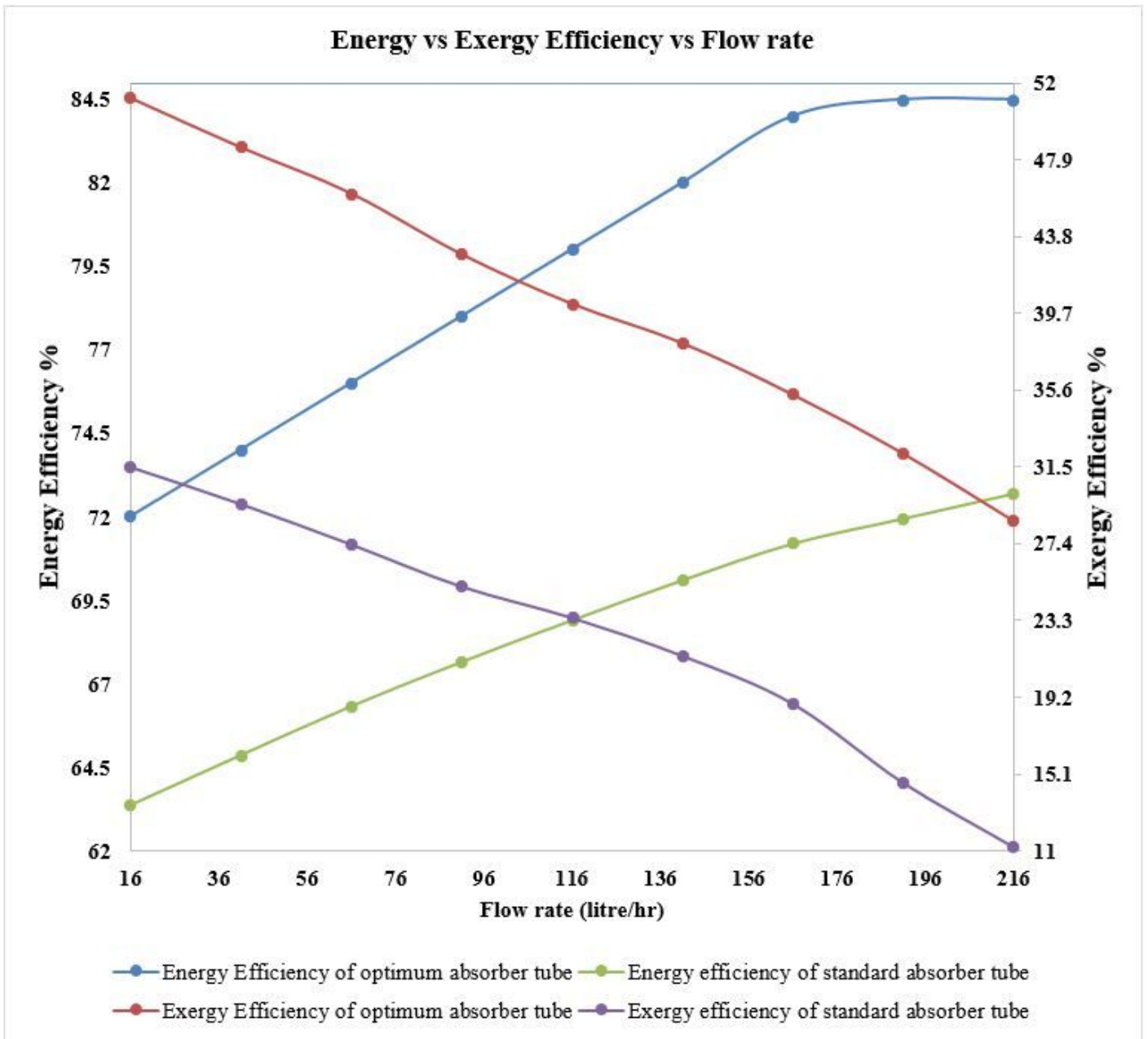
Inlet fluid temperature vs energy and exergy of system

Figure 12
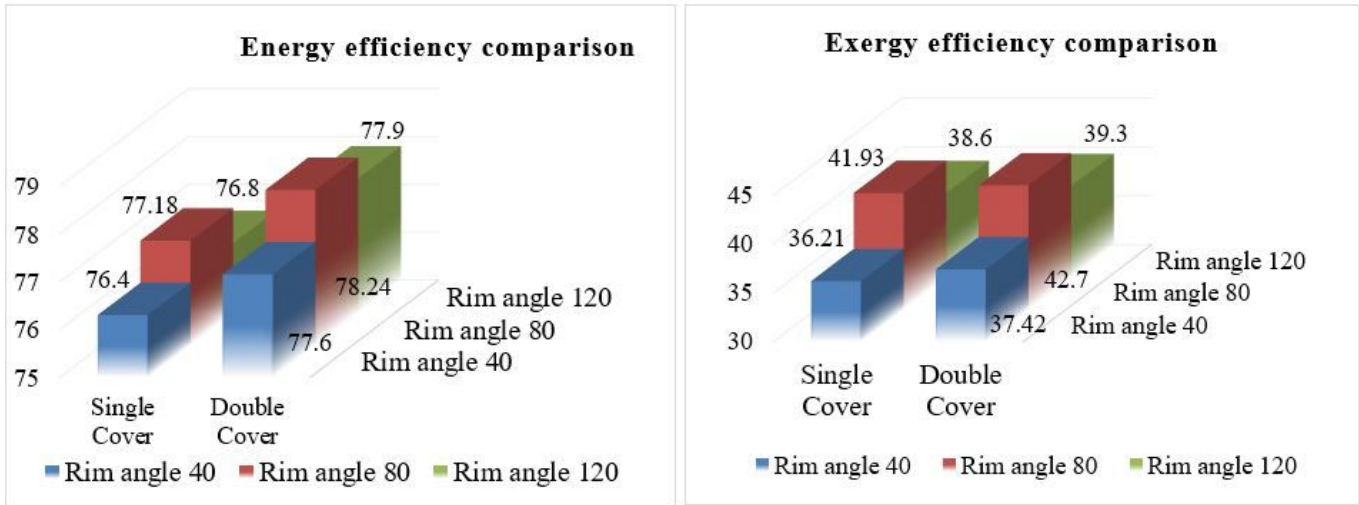
Energy vs Exergy efficiencies vs Inlet flow rate

Figure 13

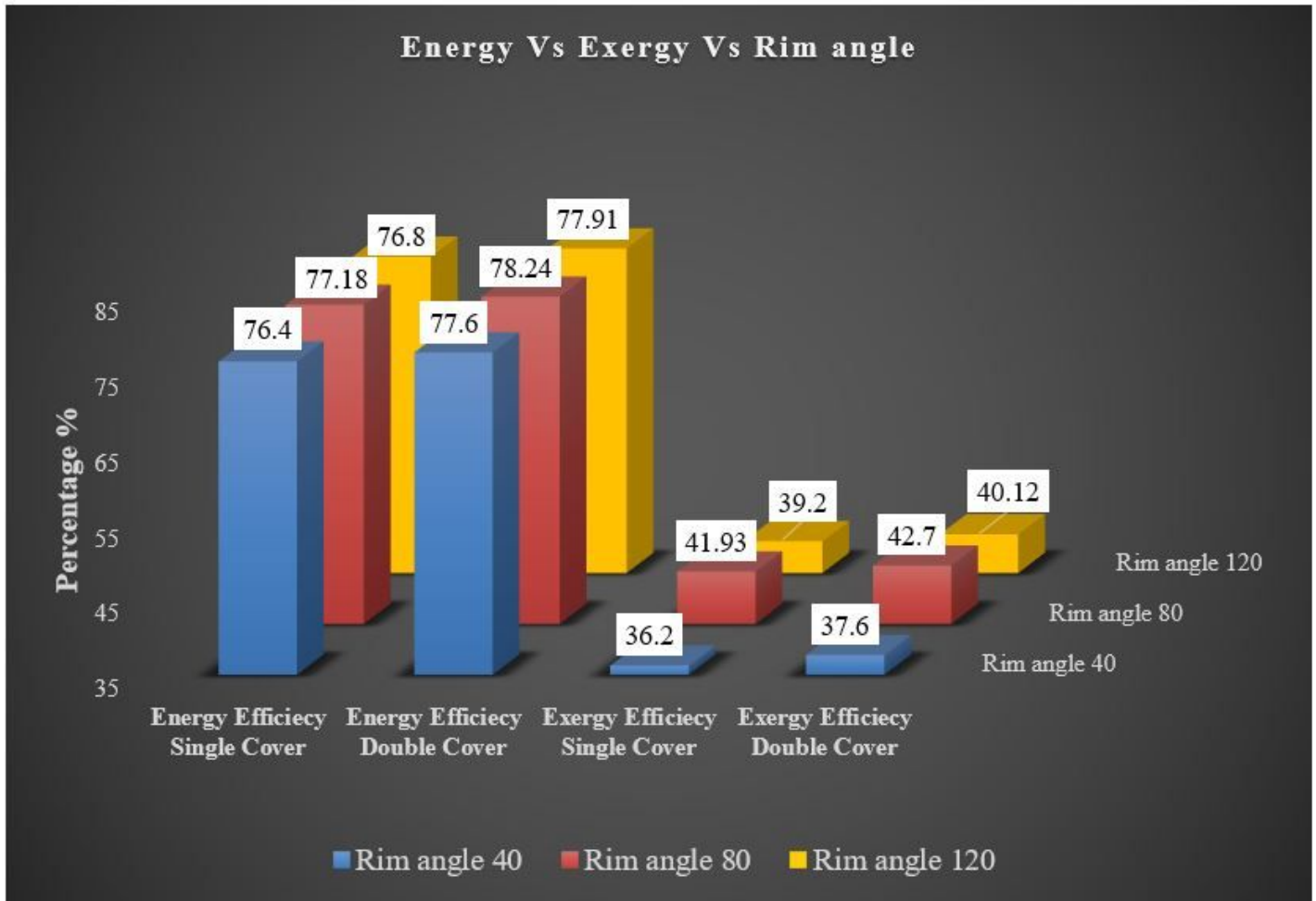(a) Energy efficiency vs Rim angle (b) Exergy efficiency vs Rim angle

Figure 14

Energy efficiency vs exergy efficiency vs rim angle

Supplementary Files

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- PTCgraphicalabstract3D.jpg