

Thermal-Hydraulic Performance of a Solar Flat Plate Collector Using CO₂ under Superheated and Supercritical Regimes

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How To Cite: Ashraf, W.; Ramgopal, M.; Reddy, M.V. Thermal-Hydraulic Performance of a Solar Flat Plate Collector Using CO₂ under Superheated and Supercritical Regimes. *Thermal Science and Applications* **2026**, *1*(3), 296–314. <https://doi.org/10.53941/tsa.2026.100018>

Received: 4 February 2026

Revised: 28 July 2026

Accepted: 21 August 2026

Published: 31 August 2026

Abstract: This study presents a thermal-hydraulic assessment of an indirect solar water heating system employing carbon dioxide (CO₂) as the heat-transfer fluid in a flat-plate collector. The work compares superheated and supercritical operating regimes under identical geometric, flow, and standardized outdoor conditions. A distributed one-dimensional MATLAB model is formulated by coupling the Hottel-Whillier energy balance with real-gas CO₂ thermophysical properties obtained by integrating REFPROP. The collector tube is discretized axially to capture the strong variations in specific heat, density, viscosity, thermal conductivity, Nusselt number, pressure drop, and local heat transfer coefficient near the critical and pseudo-critical regions. The model is evaluated for inlet temperatures of 300–340 K and operating pressures of 50–100 bar under blue-sky, hazy-sky, and grey-sky Standard Reporting Conditions of ISO 9806:2017. The analysis shows that operating near 80 bar provides the most favorable thermal-hydraulic outcomes. The heat transfer improves due to enhanced real-gas properties, while hydraulic loss and pumping power decrease because of the higher density and lower flow velocity of pressurized CO₂. Within the investigated range, supercritical operation improves collector efficiency by approximately 15–20% compared with superheated operation at comparable boundary conditions. The results provide design guidance for CO₂-based solar thermal collectors and highlight that high-pressure material selection, sealing, safety, and techno-economic assessment are necessary before practical deployment.

Keywords: indirect solar water heating system; flat plate collector; thermophysical properties of CO₂; superheated and supercritical-CO₂; thermal hydraulic model

1. Introduction

The global demand for energy, particularly thermal and electrical energy, is projected to rise substantially as population growth and industrial development continue. Current energy generation technologies that rely on conventional energy sources are increasingly inadequate to meet this escalating demand in a sustainable manner. For several decades, fossil fuels such as coal, crude oil, gasoline, and petroleum products have dominated global energy production. The extensive consumption of these non-renewable and environmentally polluting sources, however, has raised significant social and environmental problems. Large quantities of carbon dioxide are released into the atmosphere when coal and petroleum are burned, hereby intensifying climate change and environmental degradation. In addition, emissions of harmful sulphur and nitrogen oxides contribute to air pollution, which is bad for human health as well for ecological systems [1]. These challenges highlight the urgent need to switch to clean, renewable, and environmentally benign energy sources. Consequently, technologies that rely on renewable



energy resources, such as wind, solar, and hydropower are expanding rapidly due to their sustainability and low environmental impact. Among these resources, solar energy is one of the most widely utilized and promising options. Effective utilization of solar energy requires efficient solar collectors which can convert incident solar radiation into useful thermal energy [2].

The energy ecosystem in India has undergone a significant transformation in recent years, driven by the advent of solar power. Millions of residents in rural India have benefited from solar-powered applications that meet their daily energy needs, including cooking, hot water, and electrical power. Solar energy has considerable untapped potential in India. The incident solar energy on the Indian landmass is approximately 5000 trillion kilowatt-hours (kWh) annually, or 4–7 kWh/m² each day [3]. Capturing solar energy efficiently offers enormous development opportunities in India. When it comes to energy supply reliability, solar energy is the safest option due to its abundance. A small fraction of the solar energy incident upon the country's land area is theoretically sufficient to meet its energy demand. Solar energy can be converted into electricity using photovoltaic systems and stored in batteries, or used thermally by heating a circulating fluid in flat-plate solar collectors (FPCs). The FPCs offer the best combination of low cost, ease of installation, and operational efficiency [4]. These collectors collect both direct and diffuse solar irradiation, unlike concentrated collectors. FPCs are now prevalent in residences, schools, hospitals, medical institutions, dairies, agricultural and industrial areas, and other settings, primarily for hot water production [5].

With its year-round availability, using solar energy for water heating aligns with India's goal of utilizing renewable resources to meet its growing energy needs. The motivation is to decrease reliance on traditional energy sources, minimize environmental impacts by reducing greenhouse gas emissions, and enhance energy availability in both urban and rural areas. The crucial factors to consider are the cost-effectiveness and availability of solar water heating systems, particularly in remote regions. The Indian government's policies and incentives to encourage the use of solar energy are driving research efforts to improve the efficiency of solar water heaters, while simultaneously minimizing their price [6,7].

Apart from the Indian energy scenario, in Europe, the solar thermal systems may supply 20–30% of total heat demand to efficient buildings, and domestic hot water systems can cover 60–70% of yearly energy requirements. The European Solar Thermal Technology Platform (ESTTP) states that the Active Solar Building standard anticipates that by 2030, all building energy needs for heating and cooling will be satisfied primarily by solar-based technologies. Furthermore, by 2030, at least 50% of renovated buildings must use solar thermal systems to meet their heating and cooling requirements, according to the Active Solar Renovation standard [8,9].

The working principle of a solar flat plate collector (FPC) is presented in Figure 1a. As illustrated, a typical FPC comprises an absorber plate with integrated riser tubes, a transparent glazing cover, thermal insulation layers, and an external casing, as schematically depicted in Figure 1b. The FPC is enclosed in a metal case with a protective glass top (glazing). The top side of the FPC features absorber tubes attached to the metallic absorber plate. The sides and back of the FPC are provided with thick insulation [10].

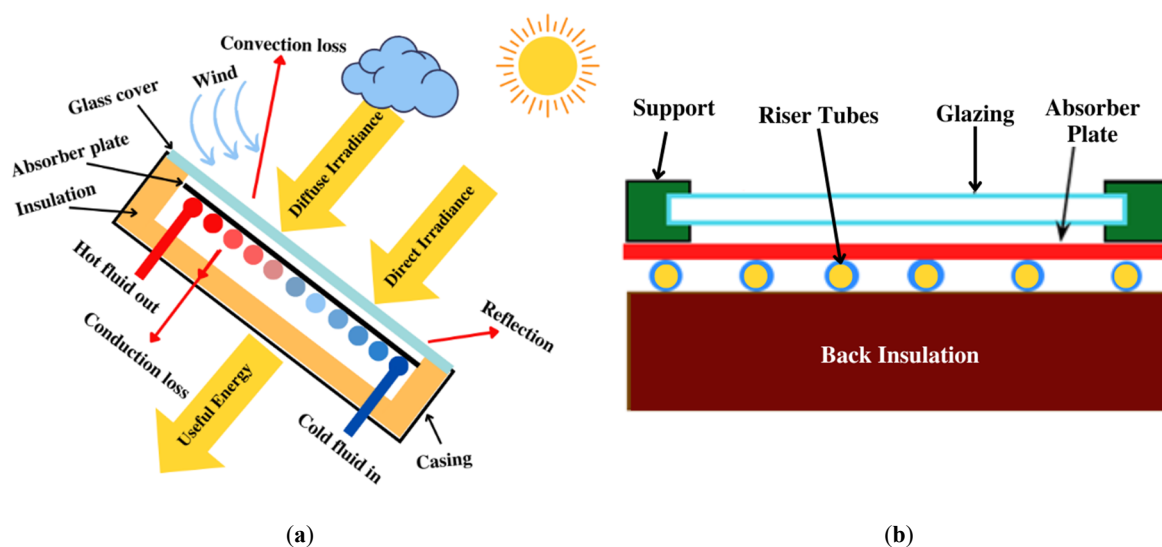


Figure 1. (a): Working principle of an FPC (b): cross-sectional view of an FPC.

2. Literature Review

The enhancement of solar flat-plate collector performance has been the subject of extensive research and technological development for a considerable period. Improving efficiency, increasing energy production, and addressing problems related to solar thermal technology are key objectives of the study in this field. Currently, solar flat plate collectors are widely used for absorbing and transferring solar energy, which is not only free but also beneficial to the planet and a renewable source of energy [8]. Solar flat plate collectors, on the other hand, remain costly, have a low efficiency level, and are bulky. Consequently, as a result of the increasing need for enhanced energy concentration in thermal systems, it is now crucial to enhance the effectiveness of solar collectors, which can be achieved through various methods [9,10].

The adoption of working fluids with superior heat transfer properties is a promising method to improve efficiency. Alternate fluids other than water are also needed to prevent freezing of the heat transfer fluid in the collector. Propylene glycol, ethylene glycol, glycol-water mixtures, silicones, and different nanofluids are some alternatives to water as heat transfer fluid [11,12]. In recent years, researchers have conducted studies on the use of nanofluids in solar flat-plate collectors; Alfellag et al. tested eco-friendly MWCNT and hexagonal boron nitride (hBN) hybrid nanofluids in a flat-plate solar collector. At 0.1 wt% and 1.2 L min⁻¹, thermal and exergy efficiencies improved by 18.33% and 19.7%, respectively. The collector area decreased by 18.2%, although viscosity and pressure drop increased [13]. Apart from the various nanofluids as a possible HTF [14], researchers have shown immense interest in the use of CO₂ as a potential alternative heat transfer fluid in solar FPC due to its unusual features and environmental benefits [15–20]. CO₂ is a natural refrigerant with a Global Warming Potential (GWP) of 1, which is significantly lower than that of many conventional synthetic fluids. In addition, CO₂ is non-toxic, non-flammable, readily available, and environmentally benign when used in closed-loop energy systems. It is compatible for use with standard lubricants and metals that are typically used in the manufacturing of machines, it is readily available, and it is very inexpensive [21]. It can absorb and transport heat efficiently within the collector due to its excellent heat transfer properties. CO₂ also has a very low normal boiling point (−78.5 °C), which makes it useful to operate across a large range of temperatures [22]. Unlike water, it does not boil when the temperature exceeds 100 °C, say due to pump failure in a forced circulation type FPC. The use of CO₂ as a heat transfer fluid in both thermo-syphon and forced circulation systems can make the system compact.

Due to its superior thermo-physical characteristics, CO₂ has been suggested for use in a variety of heat transfer applications, including heat pumps [23], next-generation nuclear reactors, extraction of chemicals, cryogenic refrigeration systems [24], electronic cooling systems, and solar thermal applications [15–21]. The natural circulation loops that are based on carbon dioxide are significantly more compact in comparison to other conventional heat transfer fluids [22].

In order to operate in cold environments, Kiran Kumar et al. [21] conducted research on a heat exchanger based on a natural circulation loop with CO₂. Yamaguchi et al. [22] conducted studies on an s-CO₂ solar collector. In the context of indirect solar heating systems, Yohanis et al. [25,26] suggested the usage of CO₂ as collector fluid and reported favorable outcomes from the investigations. In the course of performing research on the estimation of the heat transfer coefficient for in-tube heating for turbulent forced convection of s-CO₂, it is common practice to make use of the correlations that were proposed by Pitla et al. [24]. Even though there have been several studies on FPC using solar energy [7–10], these systems may be substantially improved if s-CO₂ were used as a heat transfer fluid in place of conventional heat transfer fluids. A review of the literature shows that there have not been many studies on s-CO₂-based solar water heaters. Thus, the aim of this work is to apply a simple mathematical model for assessing how well a solar FPC performs with the use of CO₂ as a working fluid inside a closed loop of an indirect solar water heating system. Its performance is evaluated under different pressures and temperatures under the three sky models as per ISO 9806:2017 standard testing protocols [27]. A simple schematic diagram for indirect solar water heating system based upon a CO₂ loop is presented in Figure 2, and the methodology followed in the present work is shown in Figure 3.

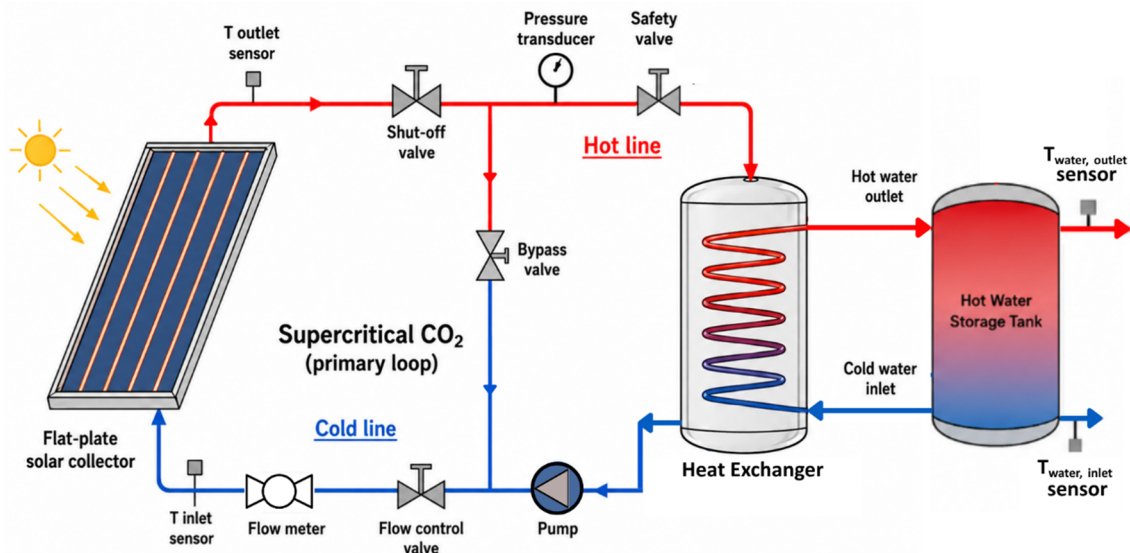


Figure 2. CO₂-based indirect solar water heating system [10].

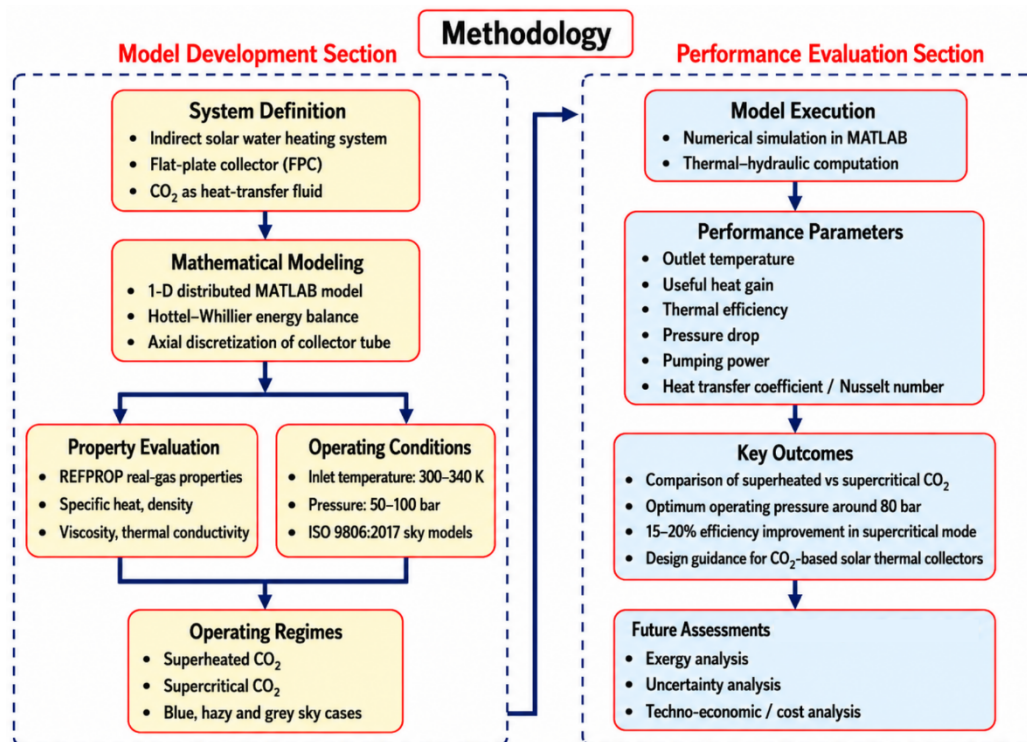


Figure 3. Methodology followed in the present work.

Research Gaps and Objectives

Solar FPC performance enhancement has been attempted through selective absorber coatings, improved insulation, altered tube geometry, nanofluids, phase-change materials, and alternative working fluids. CO₂-based systems have attracted attention because compact loops can be developed when the fluid operates at high density. For supercritical CO₂, several studies in power cycles and compact heat exchangers show that heat transfer and pressure drop are strongly coupled to pseudo-critical behavior. Reviews on supercritical CO₂ heat transfer report that conventional correlations may lose accuracy near the critical region, especially when buoyancy, acceleration, and strong property gradients are present. These findings justify the use of CO₂-specific correlations and local property evaluation in the present study.

The research gap addressed here is not the first use of CO₂ in solar collectors. Instead, the gap is the lack of a consistent, regime-to-regime thermal-hydraulic comparison for a solar FPC, in which superheated and supercritical CO₂ are evaluated under identical geometry, mass flow rate, inlet temperature, pressure range, and standardized ISO 9806:2017 sky models. The study also quantifies the hydraulic consequences of increasing

pressure and provides a design-oriented explanation for the optimal value near 80 bar. A precise contribution of the present study is shown through the Table 1.

Table 1. Contribution of the present study as per various aspects.

Aspect	Previous Emphasis	Contribution of the Present Study
Working fluid modeling	Constant-property or limited-property variation models are common for conventional fluids.	Real-gas CO ₂ properties are updated locally using REFPROP across 50–100 bar and 300–340 K.
Operating regime	Superheated and supercritical conditions are often studied separately.	A unified comparison is carried out across subcritical/superheated and supercritical regimes.
Outdoor condition	Single or idealized irradiance conditions are often used.	Blue, hazy, and grey sky models are compared using ISO 9806:2017 SRC.
Hydraulic analysis	Thermal efficiency is frequently emphasized.	Pressure drop and pumping power are evaluated in addition to thermal efficiency.
Thermodynamic interpretation	Exergy is often omitted in low-temperature collector comparisons.	A supplementary exergy formulation is added to interpret useful energy quality and irreversibility.
Practical feasibility	High-pressure implications may be under-discussed.	Material strength, safety, cost, and limitations are explicitly discussed.

Based upon the literature survey and the research gaps the following objectives are considered for the present study:

- (1) To develop a distributed mathematical model for a CO₂-based indirect SWHS using a solar FPC.
- (2) To compare superheated and supercritical CO₂ under identical collector and environmental conditions.
- (3) To evaluate the effects of pressure, inlet temperature, and sky condition on outlet temperature, useful heat gain, efficiency, Reynolds number, pressure drop, and pumping power.
- (4) To explain the physical mechanisms responsible for the thermal-hydraulic advantage of supercritical operation.
- (5) To select the suitable material for design of such high-pressure operating system, and feasibility discussion to strengthen scientific rigor.

3. Mathematical Model

3.1. Design Parameters of Solar Flat Plate Collector

In the present study, a solar water heating system is considered equipped with an FPC to collect solar energy. The specifications of the solar FPC used in developing the mathematical model are shown in Table 2.

Table 2. Specifications of the solar flat-plate collector [10].

S.N.	Variables	Value
1.	Effective absorber plate surface area	1.7 m ²
2.	Inclination angle of the flat-plate collector	32.2°
3.	inner diameter of riser tubes	8 mm
4.	outer diameter of riser tubes	9 mm
5.	Thermal conductivity of the absorber plate	240 W/mK
6.	Thermal conductivity of rear insulation	0.04 W/mK
7.	Total number of riser tubes	8
8.	Thickness of rear-side insulation layer	0.05 m
9.	Thickness of top-side insulation layer	25 mm
10.	Working fluid used for heat transfer	Supercritical and superheated CO ₂

3.2. Thermophysical Properties of CO₂

In this study, the thermophysical properties of CO₂ were obtained using the REFPROP database [28]. The variations in key properties—namely specific heat capacity, kinematic viscosity, thermal conductivity, and density—with respect to operating pressure and temperature are presented in Figure 4a–d. A pronounced non-linear behavior of these properties is observed, particularly in the vicinity of the critical point, where supercritical CO₂ exhibits sharp variations in response to small changes in thermodynamic conditions. Compared to water over the same temperature range, CO₂ shows significantly higher sensitivity of its thermophysical properties, which

can substantially influence heat transfer and flow characteristics in the system [7]. Hence the thermo-physical properties of CO₂ cannot be assumed to be constant while assessing the collector performance. Capturing this abrupt variation requires a suitable discretization of the collector tubes, allowing for the accurate evaluation of local properties.

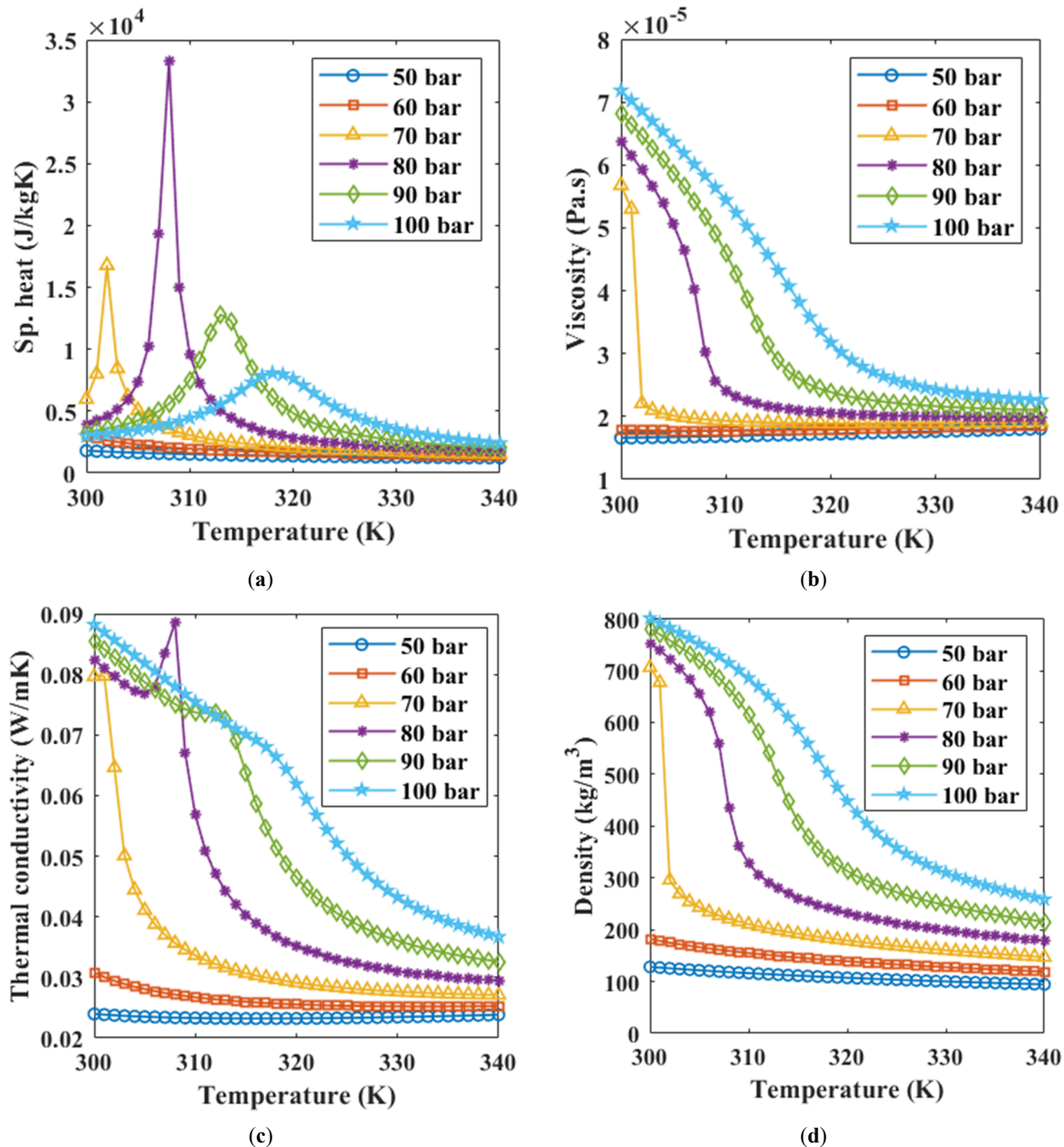


Figure 4. (a) Variation of specific heat. (b) Variation of viscosity. (c) Variation of thermal conductivity. (d) Variation of density.

3.3. Selection of Operating Ranges

The inlet temperature range of 300–340 K is selected because it matches with the low-temperature solar water-heating range and includes the CO₂ critical temperature of 304.13 K. This allows comparison of behavior below, at, and above the critical temperature while keeping the collector within the operating domain of a conventional FPC. The pressure range of 50–100 bar is selected to cover both superheated/subcritical operation below the critical pressure and supercritical operation above the critical pressure of CO₂. Pressures near 80 bar are of particular interest because the pseudo-critical temperature lies close to the collector operating temperature range, enabling favorable heat transfer enhancement without exceeding excessively high pressure.

3.4. Mathematical Formulation

To make the analysis simple, some assumptions have been considered and the same are presented in Table 3. Additionally, it is assumed that the heat losses from the surfaces of the collector are to the identical surroundings. Ambient temperature, inlet fluid temperature, and wind velocity were accounted for in defining the collector parameters and operating conditions. These parameters include the glass surface, material of the absorber plate, riser tube dimensions, position, etc., in accordance with the testing apparatus's dimensions, and the same has been mentioned in Table 2. Factors affecting the performance were analyzed based on pilot trials on an existing collector set-up that served as the basis for the models.

Table 3. Model assumptions and justification.

Assumption	Justification
Quasi-steady operation	Environmental variables vary slowly over the calculation interval; this is standard in FPC performance modeling.
One-dimensional flow	The riser tube length is much larger than the tube diameter; axial variation dominates the collector-scale response.
Uniform flow distribution	Identical riser tubes connected to common headers are assumed to receive equal mass flow under symmetric conditions.
Element-wise constant properties	CO ₂ properties are updated from element to element; within a sufficiently small element, property variation is negligible.
Single-phase operation	Selected states avoid two-phase flow; this simplifies the model and is consistent with superheated/supercritical operation.
Negligible header losses	This is a simplifying assumption.
Uniform irradiance	The ISO standard conditions are used as uniform boundary conditions over the collector aperture.

The useful heat output from an element of the collector tube is given by [29]:

$$Q_u = \dot{m}C_p(T_{f,out} - T_{f,in}) \quad (1)$$

Here, the rate of mass flow, heat capacity of the working fluid, exit temperature of the fluid, and fluid inlet temperature are denoted by $\dot{m}$, C_p , $T_{f,out}$, and $T_{f,in}$, respectively. The following equations are used to account for thermal and optical losses [30].

$$Q_u = F_r A_p \{S - U_l(T_{f,in} - T_a)\} \quad (2)$$

where F_r denotes the heat removal factor, $T_{f,in}$ & T_a are the collector inlet and ambient temperatures respectively, and U_l is overall heat-loss coefficient. Equation (2), is the well-known Hottel-Whillier-Bliss equation that eliminates the unknown mean plate temperature and is the principal steady-state collector performance relation.

$$S = \eta_o I_t \quad (3)$$

where S represents the rate of solar energy which is absorbed by the collector per unit surface area (W/m²), and η_o represents the optical efficiency of the collector.

The heat removal factor is given by [29]:

$$F_r = \frac{\dot{m}}{A_p U_l} \left[1 - \exp \left(-\frac{F' U_l A_p}{\dot{m} C_p} \right) \right] \quad (4)$$

The overall heat-loss coefficient can be calculated from the equation:

$$U_L = U_t + U_b + U_e \quad (5)$$

where U_t , U_b and U_e denote the top, bottom and edge loss coefficients, respectively. Each coefficient is expressed per unit collector area.

$$U_b = k_b / L_b \quad (6)$$

where k_b and L_b are the thermal conductivity and thickness of the back insulation. Equation (8) assumes one-dimensional conduction and negligible external film resistance.

$$U_e = k_e A_e / (L_e A_c) \quad (7)$$

where k_e and L_e are the conductivity and effective conduction length of the edge insulation, and A_e is the effective edge-loss area.

Here, U_t is the convective heat loss coefficient for the top surface given by [30]:

$$U_t = \left[\frac{1}{\frac{C}{T_p} \left[\frac{T_p - T_a}{1+f} \right]^e} + \frac{1}{h_w} \right]^{-1} + \left[\frac{\sigma(T_p + T_a)(T_p^2 + T_a^2)}{\frac{1}{\varepsilon_p + 0.00591 h_w} + \frac{1+f+0.133\varepsilon_p}{\varepsilon_g} - 1} \right] \quad (8)$$

Here, the temperature of absorber plate, ambient air, and the top of the collector's surface are represented by T_p , T_a , and T_c , respectively.

The rate of total heat lost from the top surface is evaluated as:

$$\dot{Q}_{t,l} = U_t A_p (T_p - T_a) \quad (9)$$

To evaluate the thermal efficiency of the FPC, the following formula has been employed [29]:

$$\eta_{th} = \frac{\dot{m} c_p [(T_{f,in} - T_a) - \frac{S}{U_t} (\exp(-\frac{F' U_t A_p}{\dot{m} c_p}) - 1)]}{A_p I_T} \quad (10)$$

To evaluate the Nusselt number for CO₂, the following correlation suggested by Pitla et al. [24] has been employed,

$$Nu = Nu_b \left(\frac{\mu_b}{\mu_w} \right)^{0.11} \left(\frac{k_w}{k_b} \right)^{0.33} \left(\frac{h_w - h_b}{c_{pb}(T_w - T_b)} \right)^{0.35} = \frac{\alpha_t D_i}{k_b} \quad (11)$$

where Nu_b represents the Nusselt number evaluated at the bulk temperature of the working fluid [24]

$$Nu_b = \frac{(f/8) Re Pr}{12.7 \sqrt{f/8} (Pr^{2/3} - 1) + 1.07} \quad (12)$$

The Reynolds number is given by:

$$Re = \frac{\rho V_{avg} D_h}{\mu A_f} \quad (13)$$

where Re is the Reynolds number, ρ is the fluid density, V_{avg} is the mean velocity, μ is the dynamic viscosity, D_h is the hydraulic diameter, and A_f is the total flow area associated with the specified mass-flow rate. For a circular tube, D_h equals the internal diameter.

$$Pr = \mu c_p / k \quad (14)$$

where Pr is the Prandtl number and k is the fluid thermal conductivity. All thermophysical properties should be evaluated at a consistent bulk or film temperature.

The pressure drop across an i th element of the tube is given by Darcy-Weisbach equation [16]:

$$\Delta P_i = p_f^i - p_f^{i+1} = \frac{G_f^2}{2\rho_f} \left[f \frac{\Delta L}{d} + \left(\frac{\rho_f^i}{\rho_f^{i+1}} - \frac{\rho_f^{i+1}}{\rho_f^i} \right) \right] \quad (15)$$

where, f represents the friction factor, and ρ_f is the fluid density. Assuming fully developed flow conditions, the friction factor for both the laminar and turbulent regimes can be evaluated as:

$$f = 64 / Re \quad (16)$$

$$f = [0.79 \ln(Re) - 1.64]^{-2} \quad (17)$$

where Re is the Reynolds number calculated as per Equation (10).

For superheated or supercritical CO₂, density and viscosity may vary appreciably along the collector. The absorber tube should therefore be divided into discrete elements, and the local thermophysical properties should be evaluated at the temperature and pressure of each element. Therefore, the required pump power is then approximated by:

$$\dot{W}_p = \frac{\dot{m}}{\eta_p} \sum_{i=1}^N \frac{\Delta P_i}{\rho_i} \quad (18)$$

Substituting the pressure drop across the collector tubes, the element-wise pump power becomes

$$\dot{W}_{p,i} = f_i \frac{\Delta L_i}{D_i} \frac{\dot{m}^3}{2\rho_i^2 A_c^2 \eta_p} \quad (19)$$

where W_p is the pumping power (W) and η_p is the pump efficiency. As pressure increases, the density of CO_2 increases strongly; consequently, the volumetric flow rate and velocity decrease. This reduces the frictional pressure drop and, more importantly, lowers the hydraulic power required to maintain the same mass flow rate. Therefore, although the absolute operating pressure is higher in the supercritical region, the flow resistance and pumping power may decrease due to the dense-fluid behavior of CO_2 .

3.5. Numerical Procedure and Grid Independence

The model is solved iteratively in MATLAB R2023a. The inlet pressure, inlet temperature, mass flow rate, collector geometry, and sky condition are specified. For each element, local CO_2 properties are obtained from REFPROP v10.0 at the local bulk temperature and pressure. The local heat transfer coefficient, useful heat gain, outlet temperature increment, pressure drop, and pumping power are then updated. The procedure continues until the collector outlet is reached. A grid independence test is performed by varying the number of elements from 10 to 50. The variation in outlet temperature and useful heat gain becomes negligible beyond $N = 35$; therefore, $N = 35$ is adopted for the final simulations [31,32].

4. Results and Discussion

To validate the proposed mathematical model, experiments were conducted using a solar flat-plate collector (FPC) simulator with water employed as the working fluid. A numerical algorithm was developed in MATLAB to solve the governing equations, and the predicted results were systematically compared with the experimental data to assess the accuracy and reliability of the model. Studies conducted have shown a close match between the results obtained from analytical analysis and experimental investigations. The uncertainty analysis was done while validating the model associated with water as a heat transfer fluid. It was calculated that approximately 2.304% uncertainty in thermal efficiency along with 0.565% in temperature measurement [10].

The present study focuses on CO_2 as a heat transfer fluid in the collector operated under superheated and supercritical pressures. Here, CO_2 at $T > 304.13 \text{ K}$ and $P > 73.77 \text{ bar}$, is considered under supercritical state and while as vapor-phase CO_2 below critical pressure and temperatures above saturation for that pressure is superheated CO_2 . Unlike water, which is treated as an incompressible fluid with little or no effect of pressure on the performance, for the same temperature range, CO_2 can operate under different pressures yielding varying performance. Depending upon the pressure, the fluid can be in the superheated region or in the supercritical region. Hence, it is essential to study the effect of the operating pressure on collector performance. When CO_2 is in a superheated state, its pressure is much lower than that of the supercritical state, which has practical advantages, provided the efficiency is high. With this objective, the study has been done in two steps: firstly, overall performance analysis of the FPC, and secondly, elemental analysis of the various performance parameters of the FPC by considering a single riser tube along its length.

4.1. Performance Analysis of FPC under Superheated and Supercritical Zones

Collector output can be reported consistently, independent of testing technique or collection technology, according to the newly incorporated Standard Reporting Conditions (SRC), which are a part of ISO 9806:2017 [27]. The SRC defines three distinct outdoor instances, namely “Blue Sky”, “Hazy Sky”, and “Grey Sky”, which are given in Table 4. Each of these has its own set of required limits of radiation exposure. In this analysis, the performance of solar FPC with a CO_2 operating pressure of 50–100 bar under the three sky models is studied through the fluid flow and heat transfer equations.

Table 4. ISO 9806:2017 Standard Reporting Conditions [27].

S.N.	Variable	Blue Sky	Hazy Sky	Grey Sky
1.	I_b	850 W/m^2	440 W/m^2	0 W/m^2
2.	I_d	150 W/m^2	260 W/m^2	400 W/m^2
3.	I_t	1000 W/m^2	700 W/m^2	400 W/m^2
4.	v_w		1.3 m/s	
5.	T_a		20 °C	

The results obtained from this analysis are presented in Figures 5–9. Figure 5 illustrates the relationship between the Reynolds number and operating pressure, which indicates a reduction in the Reynolds number with a rise in operating pressure. Under the operating pressure range (50–100 bar), it is found to operate in turbulent flow

regimes only. The overall pressure loss in the riser tubes falls substantially at increased operating pressure, as seen in Figure 6, which results in reduced pump power consumption.

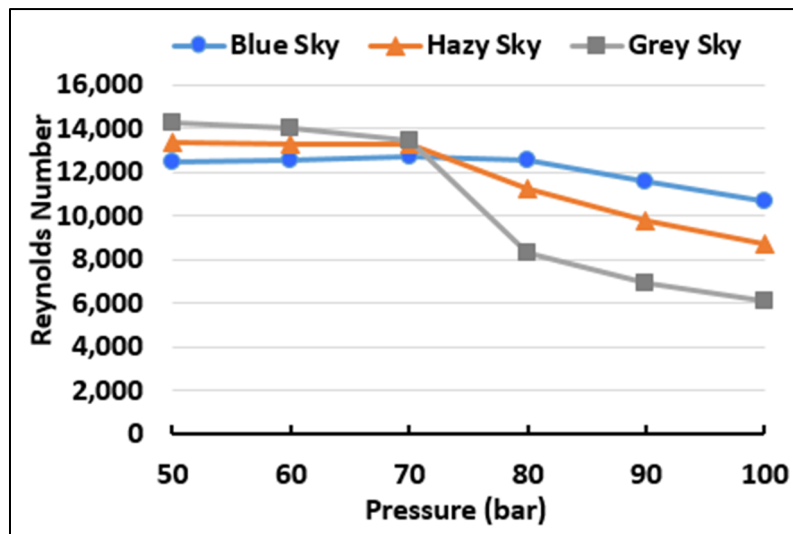


Figure 5. Variation of Reynolds number.

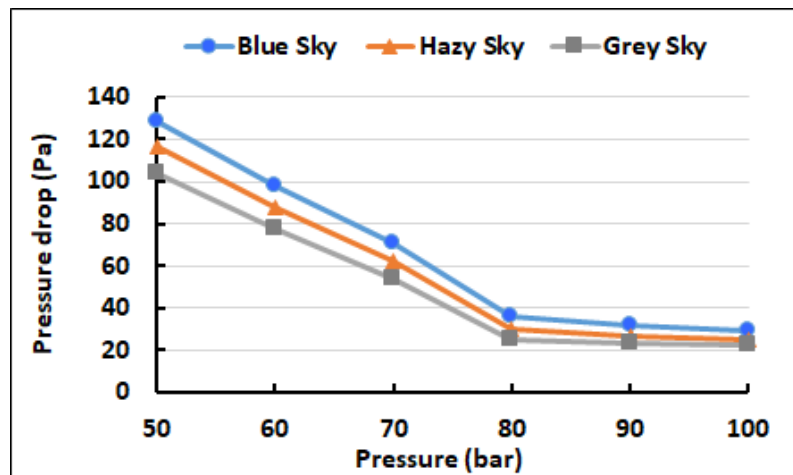


Figure 6. Pressure drop variation.

It is observed that under supercritical conditions (pressures above 72 bar) the heat transfer coefficient increases significantly as compared to superheated conditions (pressures below 72 bar). A high heat transfer coefficient (h) significantly enhances thermal efficiency by accelerating the rate of heat exchange between a surface and a fluid and reducing the mean plate temperature. This improvement allows systems to minimize energy losses and reduce the need for excessive heating or cooling inputs. This is advantageous for a solar collector, as it ensures a higher rate of heat transfer. Under identical mass flow rate and tube diameter conditions, the outlet temperature achieved with supercritical CO_2 is higher than that obtained with water, indicating the superior thermal performance of s-CO_2 as a heat transfer fluid.

A higher heat gain is also observed with higher solar radiation and the supercritical state of the working fluid, shown in Figure 7. This implies that a greater proportion of the solar energy that is captured is efficiently transmitted to the working fluid and indicates that the collector is successful in efficiently capturing and transferring solar energy, minimizing losses, and maximizing the overall efficiency of the system, as shown in Figure 8. When the pressure is between 50 and 80 bar, the changes in efficiency across different sky conditions are roughly 1–2%. Blue sky conditions attain the highest efficiency at approximately 80 bar, surpassing hazy sky conditions by approximately 2–3% and grey sky conditions by approximately 4–6%, respectively. There is a slight decrease in efficiency that is noticed beyond 80 bar, despite the fact that blue sky conditions continue to have a three to five percent advantage over grey sky situations throughout the upper pressure range. As the operating pressure increases, the density and specific heat of the fluid increase, which results in more absorption of heat and ultimately leads to an improved thermal performance of the collector. A fluid with high density and high specific

heat absorbs and stores significantly more thermal energy than low specific heat fluids. This property is crucial in applications requiring efficient heat transfer, or thermal energy storage.

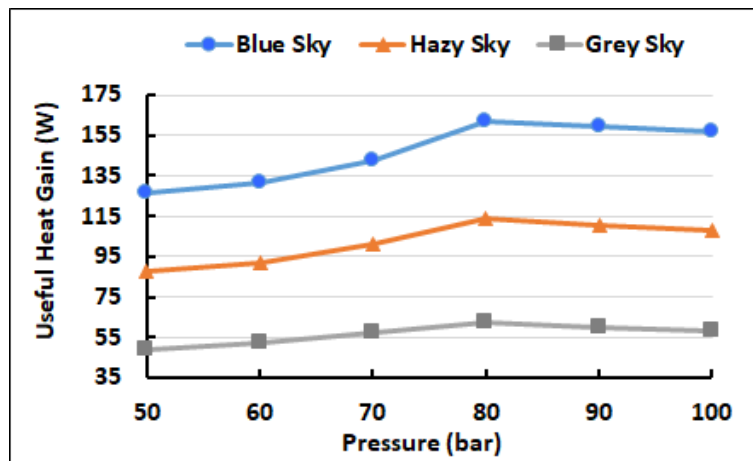


Figure 7. Variation of useful heat gain per tube with pressure.

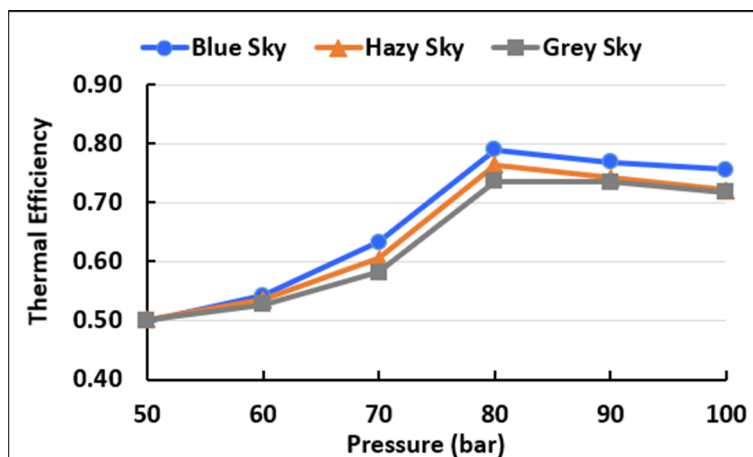


Figure 8. Thermal efficiency variation with pressure.

In the context of solar collector, the outlet temperature is the most desired output, therefore the examination for the outlet temperature is of interest. Results show that the obtained outlet temperature is higher with higher solar radiations and supercritical conditions but it is lower under superheated conditions. The thermal efficiency depends on the useful heat gain relative to the solar input, whereas the outlet temperature depends on heat capacity and mass flow. Near the pseudo-critical region, CO₂ can absorb more heat with a smaller temperature rise because its effective heat capacity is high. The observed variations are shown in Figure 9.

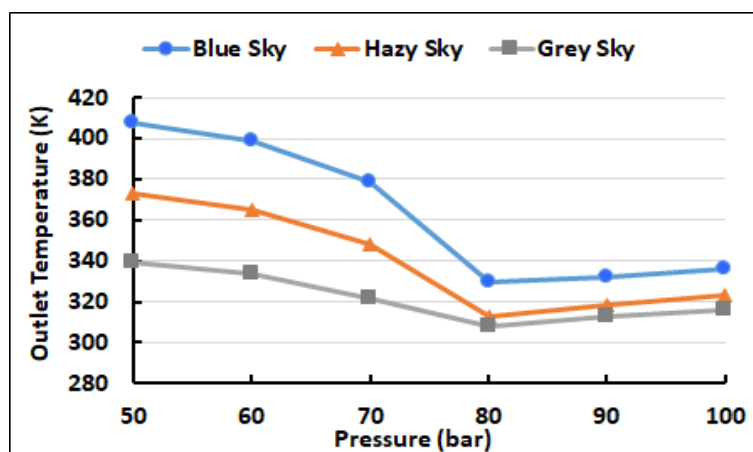


Figure 9. Outlet temperature variation with pressure.

Under the hazy and grey sky conditions, outlet temperatures are about 8–12% and 15–18% lower than the blue-sky and the lower pressure conditions. This also means the thermal efficiency decreases. As pressure gets closer to the ideal range, the difference in temperature drops to about 4–8%. This shows that pressure has a more dominating effect than atmospheric conditions when performance is at its best. When the pressure goes above 80 bar, the efficiency goes down a little and the output temperature goes up a little. This means that additive losses are forming that make it difficult to utilize heat efficiently. Apart from this the lower outlet temperature seen during overcast conditions is also due to less solar radiation and more air scattering. This means that less thermal energy is available to the system and, in turn, less heat is absorbed by the working fluid.

4.2. Performance Comparison for Single Riser Tube under the Superheated and Supercritical Zone of CO₂

It is observed from the thermodynamic characteristics of the CO₂ that the variations are very large with operating pressure and temperature. So, it is necessary to analyze the performance of the system in a stepwise manner to capture the property variations. Hence, the comparison of the performance of the solar FPC is made for the same outdoor conditions of the blue-sky model under superheated and supercritical zone of CO₂. The results obtained from the analysis are presented in Figures 10–13.

Figures 10 and 11 show a monotonic change in Nusselt and Prandtl numbers at 60 bar. This is because the properties of superheated CO₂ vary smoothly with temperature. At 80 bar, both parameters vary non-monotonically and reach a maximum near 0.5 m. This occurs when the fluid approaches the pseudo-critical region, where its properties change sharply. The higher average Nusselt number at 80 bar increases the heat-transfer coefficient, as shown in Figure 12. As a result, supercritical CO₂ provides better heat transfer and higher collector efficiency.

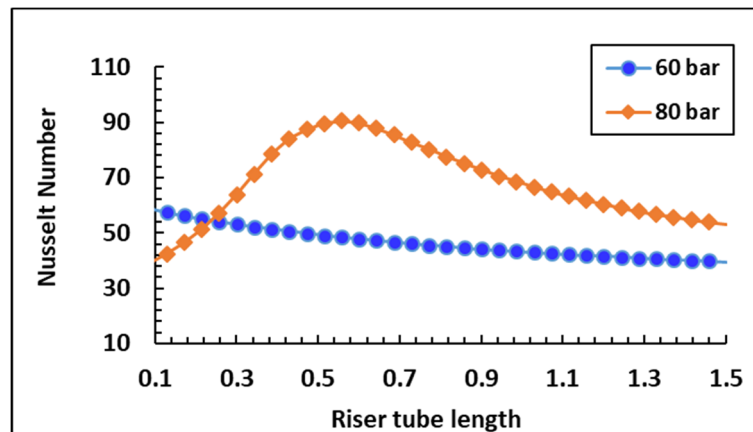


Figure 10. Nusselt number variation with tube length.

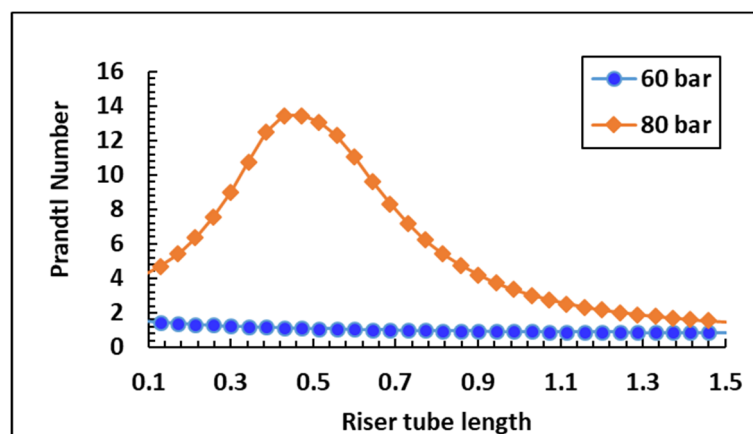


Figure 11. Prandtl number variation with tube length.

In superheated conditions, the low density leads to high fluid velocity, and despite a low friction factor and low heat transfer coefficient, the pressure drop remains high due to the strong influence of the momentum term (ρV^2) in the pressure drop equation. Therefore, a higher cumulative pressure drop is also seen under superheated conditions, as shown in Figure 13.

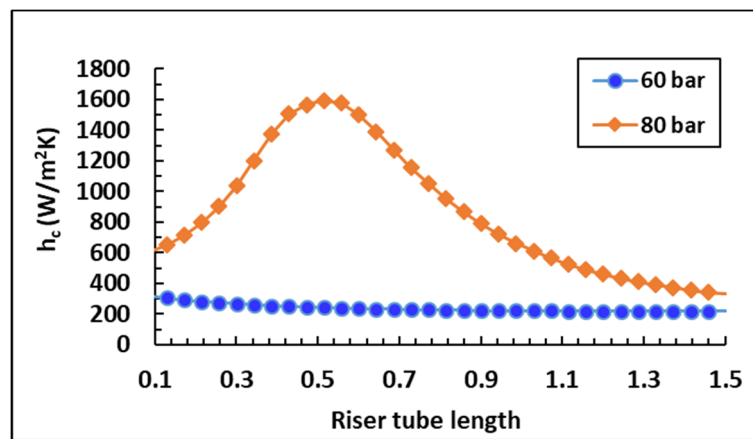


Figure 12. Convective heat transfer coefficient.

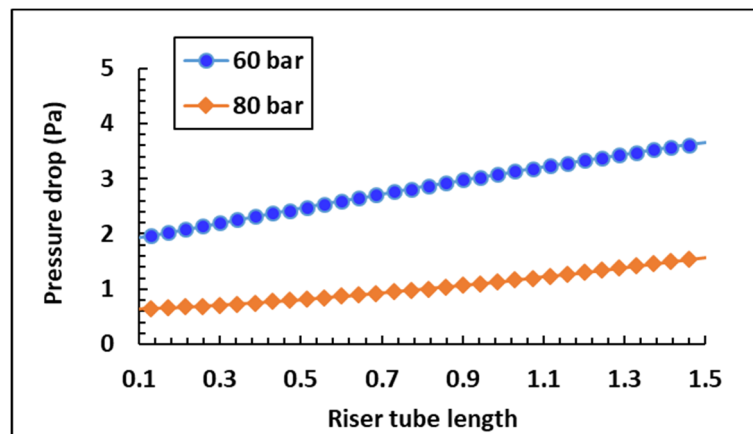


Figure 13. Pressure drop variation with tube length.

The thermal performance of the solar flat plate collector at various ambient temperatures is shown in Figures 14 and 15 for superheated and supercritical conditions, respectively. For superheated conditions, the ambient temperature has a negative impact on performance due to increased plate temperatures, resulting in higher heat losses. However, for a supercritical condition, there is an optimum ambient temperature at which the efficiency reaches a peak value.

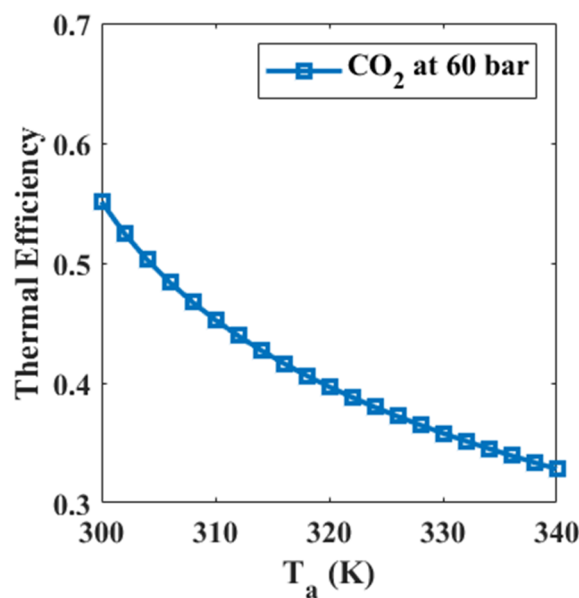


Figure 14. Thermal efficiency under superheated CO_2 with ambient temperature.

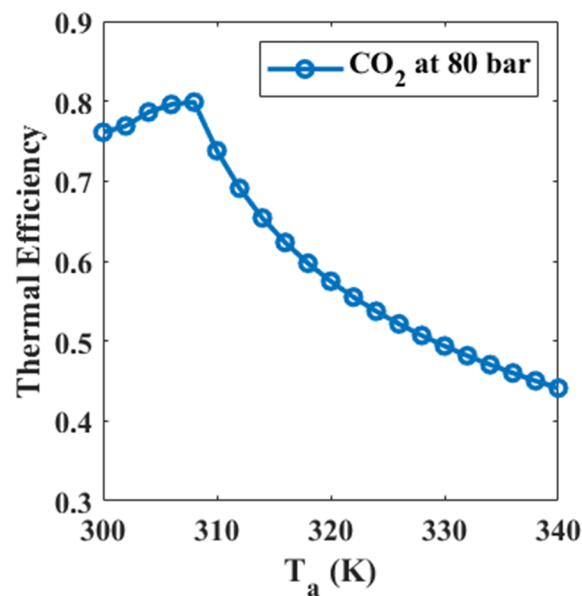


Figure 15. Thermal efficiency under supercritical CO₂ with ambient temperature.

From these Figures, it is clear that, depending on the ambient temperature, the theoretical efficiency under supercritical conditions can be as high as 80%, whereas the maximum thermal efficiency under superheated conditions is around 55%. Near the critical point, supercritical CO₂ exhibits exceptionally high specific heat and enhanced convective heat transfer characteristics, resulting in improved heat removal from the absorber surface and reduced thermal losses. Consequently, higher collector efficiencies can be achieved compared with conventional working fluids. Similar results have been reported by Sarkar (2013) [17], who demonstrated significant energetic performance enhancement of flat plate solar collectors using supercritical CO₂ compared to water and conventional fluids.

5. Novelty, Practical Feasibility and Implementation Challenges

5.1. Novelty

This research offers innovative perspectives on the utilization of carbon dioxide (CO₂) as a working fluid in solar flat plate collectors operating at superheated and supercritical conditions. Earlier research on flat plate collectors has mostly employed conventional fluids, such as water, oils, and water-ethylene glycol blends [12,14], which often exhibit problems with thermal stability, viscosity, and environmental consequences when used at very low or very high temperatures. On the other hand, CO₂ exhibits favourable thermophysical characteristics near its critical point, including high density, low viscosity, and improved heat transmission [17]. Moreover, CO₂ possesses non-toxic and non-flammable properties, along with a negligible risk of ozone depletion, making it a sustainable substitute for synthetic refrigerants [19].

This study also examines several sky conditions: blue, hazy, and grey—illustrating practical atmospheric fluctuations that directly influence the collector's efficiency. This approach enhances the practical relevance of the research, as several contemporary studies assumed only optimal or unobstructed sky irradiance conditions [9]. The novelty of the present study lies in transforming a classical flat-plate collector framework into a real-gas, distributed, thermal-hydraulic model for CO₂ across the superheated and supercritical regimes. The model does not claim that CO₂ or s-CO₂ collectors are new in general. Instead, it provides a direct and consistent comparison of the two thermodynamic regimes under ISO 9806:2017 sky models and identifies the pressure range where heat transfer enhancement and pressure drop reduction are simultaneously favorable.

Operation at 80–100 bar is technically feasible with properly selected tubes, headers, heat exchangers, seals, pressure relief devices, and proof-tested components. Stainless steel or other compatible high-pressure tubing may be required. The design must comply with applicable pressure vessel and piping standards, including over-pressure protection, leak detection, safe charging procedures, and an appropriate factor of safety. These requirements increase capital cost relative to water or glycol systems. Therefore, the technology is more likely to be attractive in compact, high-performance, cold-climate, or industrial solar thermal applications where freezing resistance, compactness, and enhanced heat transfer are valuable.

A preliminary feasibility comparison is summarized below. The Table 5 is qualitative because detailed cost data depend strongly on local materials, pressure rating, manufacturing scale, and maintenance practice [32].

Table 5. Comparison of heat transfer fluids [32].

Fluids	Advantages	Limitations
Water	Low cost, high heat capacity, simple hardware	Freezing, boiling risk, corrosion or scaling, not very compact
Glycol-water	Freeze protection, mature technology	Higher viscosity, degradation at high temperature and pumping penalty
Nanofluids	Potential heat transfer enhancement	Stability, agglomeration, cost, maintenance uncertainty
Supercritical CO ₂	High density, low viscosity, strong heat-transfer enhancement near pseudo-critical region, compact loop	High-pressure components, sealing, safety, techno-economic uncertainty

5.2. Practical Feasibility at 100 Bar

Operation of CO₂ at pressures up to 100 bar requires explicit consideration of tube strength, pressure containment, sealing, and safety protection. A mechanical feasibility analysis has been done using standard pressure-vessel relations and representative material data [33].

For a cylindrical collector tube subjected to internal pressure, the circumferential or hoop stress is estimated using the thin-walled pressure vessel relation, also known as Barlow's relation, when the wall thickness is small relative to the tube diameter ($t/D < 0.1$) [34].

$$\sigma_h = \frac{Pd}{2t} \quad (20)$$

where σ_h is the hoop stress (Pa), P is the internal operating pressure (Pa), d is the tube diameter (m), and t is the tube wall thickness (m). Rearranging this expression gives the required wall thickness:

$$t = \frac{Pd}{2\sigma_{allow}} \quad (21)$$

where σ_{allow} is the allowable stress of the selected tube material. For design purposes, the wall thickness is multiplied by a safety factor N_s

$$t_{design} = N_s t \quad (22)$$

These relations are standard in pressure vessel design and are consistent with ASME Boiler and Pressure Vessel Code Section VIII and classical mechanical design texts such as Shigley's Mechanical Engineering Design [34].

The hoop stress relation becomes less accurate when the wall is not very thin relative to the internal radius. The maximum circumferential stress at the inner surface of a thick cylindrical tube is therefore verified using Lamé's equation [35]:

$$\sigma_{\theta,i} = P_i \left(\frac{r_o^2 + r_i^2}{r_o^2 - r_i^2} \right) \quad (23)$$

By setting $\sigma_{\theta,i} = S_T$ and using $r_o = r_i + t_L$, the pressure-required wall thickness is:

$$t_L = r_i \left[\sqrt{\frac{S_T + P_i}{S_T - P_i}} - 1 \right] \quad (24)$$

The Lamé result is adopted for the minimum pressure thickness because it is equal to or greater than the Barlow estimate for the materials considered. By considering the manufacturing tolerance, the nominal thickness after including a fractional negative wall tolerance δ is:

$$t_{nom} = \frac{t_L}{1 - \delta} \quad (25)$$

For the present calculation:

$$\delta = 0.10, \quad t_{nom} = \frac{t_L}{0.90}$$

5.3. Material Selection for High-Pressure CO₂ Collector Loops

Material selection, as referred to Table 6, should consider allowable stress, corrosion resistance, thermal compatibility, manufacturability, and cost. Stainless steels are generally preferable for high-pressure CO₂ applications because they combine mechanical strength with corrosion resistance and compatibility with pressure-rated fittings [34]. Copper has high thermal conductivity, but its lower allowable stress may require greater wall thickness for safe operation at 80–100 bar.

Table 6. Calculated tube thickness for different standard materials [35].

Material	Barlow Thickness, t_B (mm)	Lame Thickness, t_L (mm)	Nominal Thickness with 10% Tolerance, (mm)
SS304L/SS316L	0.236	0.245	0.272
Carbon steel	0.303	0.318	0.354
Soft-annealed copper	0.981	1.188	1.320
Aluminium 6061-T6	0.337	0.356	0.396

The operation of CO₂ at pressures up to 100 bar is technically feasible when pressure-rated tubing and appropriate safety systems are adopted [35–37]. The detailed comparison of tube material and corresponding thickness design calculations are shown in Table 7 and Figure 16 respectively. Nevertheless, the use of pressure relief devices, leak detection, high-pressure fittings, and compliance with pressure-vessel standards is essential. Therefore, while supercritical CO₂ provides improved thermal-hydraulic performance, its practical implementation requires careful mechanical design and safety assessment [37].

Table 7. Comparison of materials for high-pressure CO₂ collector tubing [35–37].

Material	Typical Allowable Stress (MPa)	Suitability	Remarks
Soft-annealed copper	70–110	High	High thermal conductivity, but lower allowable stress; needs thicker wall for high-pressure operation
SS304/SS316L stainless steel	120–140	High	Common pressure-rated tubing material; good strength and manufacturability, but lower thermal conductivity than copper
Carbon steel	140–160	High	Mechanically strong and economical, but corrosion protection is required
Aluminium 6061-T6	60–90	Medium-high	Better corrosion resistance than copper; strength lower than stainless steel but costly than copper

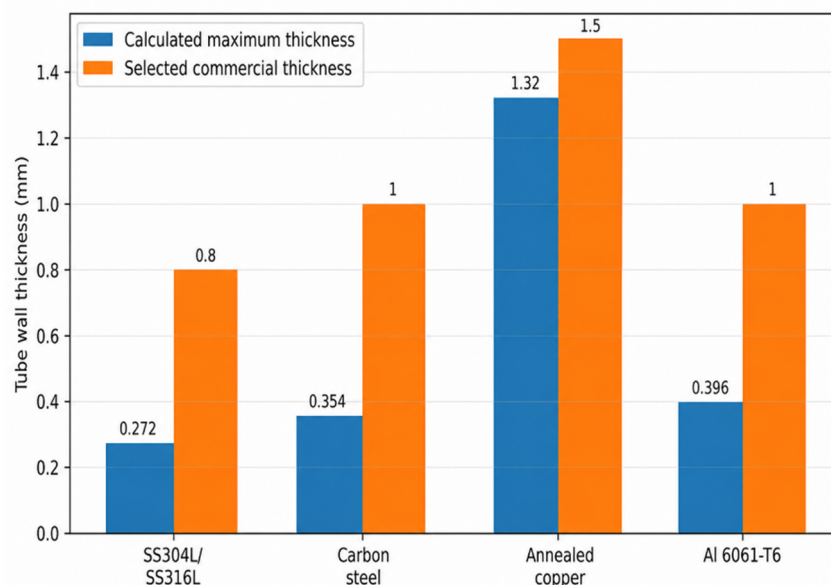


Figure 16. Calculated and commercially selected tube-wall thicknesses for 100-bar CO₂ loop.

6. Conclusions

A simple mathematical model is used to evaluate the solar FPC performance under CO₂ conditions in accordance with ISO 9806:2017. As the thermo-physical properties of CO₂ are distinctly different from those of water, the fluid property variations with operating temperature and pressure are properly considered in the present study. Based on the model, the performance of the FPC is compared under superheated and supercritical conditions by varying the pressure from subcritical to supercritical. The results showed that the FPC with s-CO₂ (80 bar) provides superior thermal performance compared to superheated CO₂ (60 bar) under similar outdoor conditions. The maximum efficiency enhancement for the investigated ranges by employing s-CO₂ compared to superheated CO₂ is about 20%. The maximum outlet temperature obtained at 80 bar pressure is 380 K, which is about 14% higher than the highest outlet temperature obtained by operating FPC with CO₂ under 60 bar superheated conditions. The results also showed that the pressure drop is much lower with s-CO₂, which implies lower pump power consumption. Hence, s-CO₂ is suggested for applications in indirect solar water heating systems, provided the high operating pressures are taken care of through proper design of the tubing and heat exchangers.

The s-CO₂ shows promising thermal-hydraulic behaviour within the investigated range, but practical deployment requires high-pressure design, safety assessment, long-term reliability testing, and techno-economic analysis.

Author Contributions

W.A.: conceptualization, methodology, software, data curation, investigation, visualization, writing—original draft preparation; M.R.: supervision, investigation, visualization, writing—reviewing and editing; M.V.R.: supervision, investigation, visualization, writing—reviewing and editing. All authors have read and agreed to the published version of the manuscript.

Funding

This research received no external funding.

Institutional Review Board Statement

Not applicable.

Informed Consent Statement

Not applicable.

Data Availability Statement

All data used is presented in the paper.

Acknowledgments

The authors acknowledge the School of Energy Science and Engineering, Indian Institute of Technology Kharagpur, West Bengal-721302, for providing access to the MATLAB R2023a and REFPROP v10.0 software and other laboratory resources to that enabled the successful completion of this study.

Conflicts of Interest

The authors declare no conflict of interest.

Use of AI and AI-Assisted Technologies

During the preparation of the manuscript, the authors used Grammarly (Grammarly Inc.) for English correction, proofreading and editing. After using this tool, the authors reviewed and edited the content as needed and take full responsibility for the content of the published article.

Nomenclature

Symbol	Description	Unit
A_p	Absorber plate area	m ²
C_p	Specific heat at constant pressure	J kg ⁻¹ K ⁻¹
D_i	Inner diameter of riser tube	m
D_o	Outer diameter of riser tube	m

E_d	Exergy destruction rate	W
F_r	Collector heat removal factor	-
f	Darcy friction factor	-
G	Solar irradiance incident on collector plane	W m^{-2}
h	Convective heat transfer coefficient	$\text{W m}^{-2} \text{K}^{-1}$
I_b	Beam irradiance	W m^{-2}
I_d	Diffuse irradiance	W m^{-2}
I_t	Total irradiance	W m^{-2}
k	Thermal conductivity	$\text{W m}^{-1} \text{K}^{-1}$
L	Tube length	m
$\dot{m}$	Mass flow rate	kg s^{-1}
N	Number of discretized elements	-
Nu	Nusselt number	-
P	Operating pressure	bar or Pa
Pr	Prandtl number	-
Q_u	Useful heat gain	W
Re	Reynolds number	-
S	Absorbed solar radiation per unit area	W m^{-2}
T_a	Ambient temperature	K
T_c	Glass cover temperature	K
T_f	Fluid bulk temperature	K
T_i	Collector inlet temperature	K
T_o	Collector outlet temperature	K
T_p	Absorber plate temperature	K
U_L	Overall heat loss coefficient	$\text{W m}^{-2} \text{K}^{-1}$
v	Mean fluid velocity	ms^{-1}
W_p	Pumping power	W
ΔP	Pressure drop	Pa
η_{th}	Thermal efficiency	-
μ	Dynamic viscosity	Pa.s
ρ	Density	kg m^{-3}
τ_a	Transmittance-absorptance product	-

Abbreviation

FPC	Flat plate collector
HTF	Heat transfer fluid
SRC	Standard reporting conditions
s-CO ₂	Supercritical carbon dioxide
SWHS	Solar water heating system

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