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Simulation Modelling and Techno-Economics of Supercritical Carbon Dioxide Recompression Closed Brayton Cycle

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Abstract

In recent years, there has been global interest in meeting targets relating to energy affordability and security while taking into account greenhouse gas emissions. This has heightened major interest in potential investigations into the use of supercritical carbon dioxide (sCO_2) power cycles. Climate change mitigation is the ultimate driver for this increased interest; other relevant issues include the potential for high cycle efficiency and a circular economy. In this study, a 25 MW_e recompression closed Brayton cycle (RCBC) has been assessed, and sCO_2 has been proposed as the working fluid for the power plant. The methodology used in this research work comprises thermodynamic and techno-economic analysis for the prospective commercialization of this sCO_2 power cycle. An evaluated estimation of capital expenditure, operational expenditure, and cost of electricity has been considered in this study. The ASPEN Plus simulation results have been compared with theoretical and mathematical calculations to assess the performance of the compressors, turbine, and heat exchangers. The results thus reveal that the cycle efficiency for this prospective sCO_2 recompression closed Brayton cycle increases (39% - 53.6%) as the temperature progressively increases from 550˚C to 900˚C. Data from the Aspen simulation model was used to aid the cost function calculations to estimate the total capital investment cost of the plant. Also, the techno-economic results have shown less cost for purchasing equipment due to fewer components being required for the cycle configuration as compared to the conventional steam power plant.

Keywords

Supercritical Carbon Dioxide (sCO_2), Closed Brayton Cycle, Techno-Economics, Simulation, Capital Expenditure, Gas Turbine,

Thermodynamic, Equipment Cost, Optimization and Sensitivity

1. Introduction

In recent years, there has been a fast-growing global demand for energy, which has led to rapid industrial research and technological advancements to meet the international demand for energy and power [1]. However, the use of fossil fuel reserves is on the rise, which could be accountable for ecological complications, such as global warming and air pollution. Global warming caused by greenhouse gas emissions is certainly a human-induced effect due predominantly to burning fossil fuels. As a result, the UK is devoted to decreasing greenhouse gas emissions by 80% by 2050 concerning the levels of 1990 [2] [3]. Fired power plants are the largest source of CO_2 emissions, as reported in the Global Energy CO_2 Report 2018 [1].

However, fossil energy is projected to remain one of the most important sources of energy shortly [4]. Therefore, innovative power generation solutions must be considered. CO_2 can be used as a derived source of energy when captured and recycled. Carbon capture and storage (CCS) is one area that will create a vital contribution towards reducing CO_2 emissions from power plants to meet the international CO_2 emission targets [4]. Hence, there is a need for developmental research into favorable alternatives such as supercritical carbon dioxide (sCO_2) or hydrogen power plants.

Because of their small size and excellent thermal efficiency, supercritical carbon dioxide power plants are becoming a viable option for producing energy efficiently [2]. However, despite their promise, several theoretical shortcomings and difficulties have been identified in the literature that prevent their broad use and optimization. These gaps cover, among other things, problems with thermodynamics, heat exchanger design, cycle integration, material selection, and environmental impacts [5]. This work employed a leveraging tool, such as ASPEN Plus, to model the thermodynamic parameters of sCO_2 across a range of temperatures and pressures.

Theoretical gaps in the literature for sCO_2 power plants are significant, but they can be unheeded by intensive study and simulation efforts. Along with these advancements, this work has designed efficient components, optimized a 25 MWe sCO_2 cycle, evaluated the environmental and economic implications, and provided a thorough techno-economics and sensitivity analysis for feasible commercialization. This comprehensive approach is crucial for advancing the development and deployment of sCO_2 power plants as a feasible and sustainable power generation technology. sCO_2 cycles are promising alternatives that can lower CO_2 emissions and be a substitute for conventional power plants.

sCO_2 power cycles will provide less costly, reliable, efficient, clean energy and a perspective for full carbon capture to meet the power generational demand without the effect of seasonal variations [5]. For instance, unlike other renewable energy sources such as solar or wind energy, which are affected by seasonal variations, sCO_2 power plants can provide a stable supply of power to meet peak consumption needs. For example, seasonality affects India's wind generation, it either peaks for the period of the monsoon season or declines during winter in Europe as a result of prolonged poor weather conditions like foggy weather with no sunshine or windy months during winter in Europe [6]. Increased investment in CO_2 power cycles will produce work and energy that can compensate for production power losses during periodic disparity and decarbonize the power industry [6].

On this note, it is equally important to recognize RCBC power cycles for industrial transformation with sCO_2 technology for immense power development. The uncertainties in the energy industry, such as energy supply, demand, commercial interest, and government policies, cannot be overlooked. However, establishing readily available data on the techno-economics of sCO_2 power plants will provide an enabling environment for the commercialization of sCO_2 power plants. The ultimate goal for commercializing sCO_2 power plants will be to improve the reduction in CO_2 emissions and also to provide additional power supply to the national grid when deployed [7].

Indeed, sCO_2 Brayton cycle technology has been reported to be more economically sustainable and efficient compared to Rankine cycles. A sCO_2 Brayton cycle is less complex and more compact due to decreased components, resulting in a reduction of the overall plant size, and improved maintenance, operational, and installation capital costs [8].

sCO_2 is a fluid state of carbon dioxide above the critical pressure and temperature. sCO_2 as a working fluid is chemically stable, reliable, low-cost, non-flammable, and readily available [9]. Additionally, one benefit that makes the application of sCO_2 commendable in a sCO_2 Brayton cycle is the rapid transformation in its thermophysical properties above its supercritical point. The density of sCO_2 is similar to that of its liquid but with low friction, which significantly reduces the work of the compressor [9]. Other key benefits of sCO_2 RCBC include the following:

- Increased Efficiency: Carbon Capture and Storage (CCS) technology used sCO_2 in power plants to increase overall efficiency by up to 50%, making them a much more cost-effective and reliable energy source.
- Reduced Carbon Emissions: sCO_2 power plants capture and store carbon dioxide, leading to a significant reduction in the amount of greenhouse gases released into the atmosphere.
- Improved Reliability: sCO_2 plants are much more reliable than other forms of power generation due to their ability to store energy for long periods without the need for frequent refueling.
- Reduced Fuel Consumption: The use of existing fuel sources, such as natural gas and coal, is much more efficient in sCO_2 power plants, leading to a decrease in fuel consumption and cost.
- Increased Flexibility: sCO_2 plants are much easier to scale up or down

depending on energy demand, giving utilities the flexibility to adjust their energy production to match changes in energy demand.

- Lower Maintenance Costs: Maintenance costs for sCO_2 plants are much lower than other forms of power generation due to their simplified design and fewer moving parts.

2. Economic Analysis

Because of the increased need for energy consumption, the need for power would keep rising globally [10]. This is likely to stimulate the economy in the power sector, which could result in increased prices for fuels and energy [10]. On the other hand, the fight against global warming will result in a significant decrease in energy costs and CO_2 emissions as a result of the application of CO_2 emission-reduction technologies and policies, such as the shutdown of coal plants and the capture of CO_2 from power cycles [10].

According to the International Energy Agency (IEA), renewable energy routes, economic activity and population growth are the two main factors that affect GDP. According to ETP projections, the global GDP growth for energy and fuel between 2017 and 2060 is projected to experience a significant increase [10]. However, accelerating climate change may have a variety of effects on the economy's potential growth [10]. These variables were not included in the ETP analysis, and as economic growth is anticipated to experience non-marginal changes as a result of advances, GDP growth is not likely to be equal to population growth in light of the effects of the climate.

The Russia-Ukraine War has had a significant impact on the global energy sector. The conflict has disrupted Europe's energy supply, particularly concerning natural gas. Ukraine is a major transit route for Russian natural gas exports to Europe, and the conflict has caused numerous interruptions in gas deliveries. This has led to a rise in energy prices, as well as a decrease in supply security in Europe. Additionally, the conflict has led to instability in the region as a whole, making it more difficult for energy companies to do business. Finally, the conflict has also contributed to a rise in geopolitical tensions in the region, further limiting the potential for energy projects. These and many more are factors that could escalate the uncertainties in the energy sector.

Thermodynamic Assumptions

The following assumptions and expectations for an ideal cycle simplify the analysis of complex gas cycles, such as closed Brayton cycles, to allow for an optimum operating condition with analysis to be evaluated [11].

- It is assumed that the working fluid CO_2, will continuously circulate in a closed-loop cycle and behaves as an ideal gas.
- Isentropic compressor and turbine: it is assumed that the compressor and turbine components of the closed Brayton cycle can be presumed to be isentropic, which implies that they are insulated and there is no energy lost to the atmosphere.

- Constant pressure heat addition or rejection: This assumption states that the heat addition and rejection stages occur at a constant pressure in the turbine.
- It is assumed that there will be an adequate amount of heat from carbon dioxide heat recovery processes to meet the operating steady-state and pressure of sCO_2 power cycles.
- It is assumed the cycle is operated under steady-state conditions.

3. The Aspen Simulation Environment

The study model was developed using ASPEN Plus process simulation software (ASPEN plus v11). Figure 1 and Figure 2 illustrate a schematic diagram and the simulation environment of the study model. The Peng-Robinson equation solver was selected to model the working fluid CO_2 properties. The inlet stream into the main compressor was selected as the feed stream with a specified temperature of 35˚C a pressure of 73 bars, mass flow rate of 350 kg/s. These input parameters are slightly greater than the critical pressure and temperature of CO_2. It is assumed the selected values will avoid or minimize condensation occurrence in the cycle. Both the isentropic and mechanical efficiencies were assumed to be 0.89 and 0.99, respectively, to provide high compressibility and density. The pressure ratio in the two compressors is 3.0 and 2.79, respectively, whereas that of the turbine is 3. The turbine has an inlet temperature of 700˚C and a pressure of 250 bar with an isentropic efficiency of 0.9 and mechanical efficiency of 0.99. On the other hand, the cooler operates at 32˚C [7]. Table 1 is a summary of the parametric values used in the simulation environment.

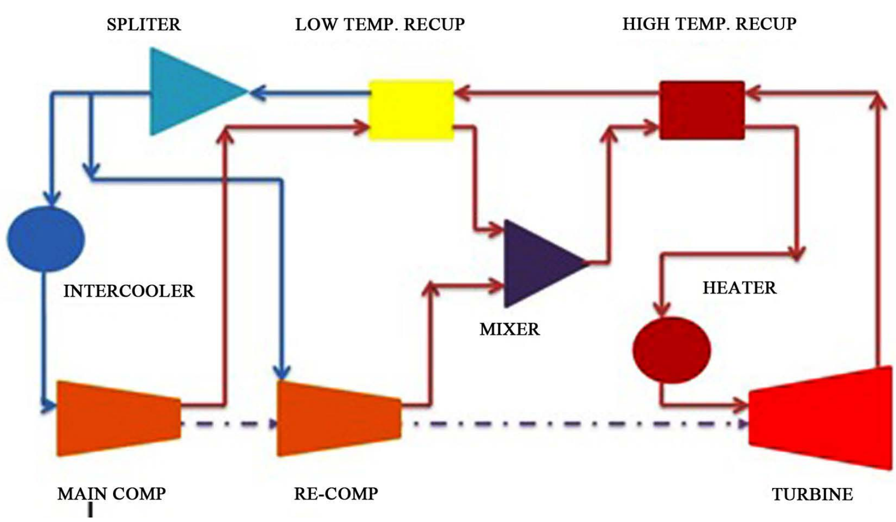

Figure 1. Schematic diagram of the study model.

Table 1. Values of simulation condition.

COMPONENT	TEMPERATURE (˚C)	PRESSURE (bar)
Turbine	700	250
Main Compressor	32	76

Continued

Re-compressor	100	125
LTR	186	87
HTR	400	87
Heater	700	250
Intercooler	32	-

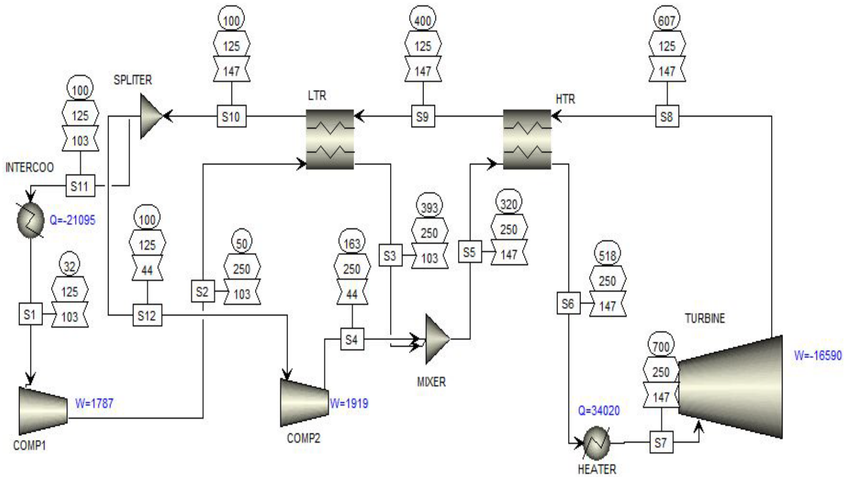

Figure 2. Simulation environment study.

3.1. Cost Evaluation of the Components

This study is largely focused on providing improved data on the estimation of sCO_2 components, to create a shift in the attention from cycle efficiency to techno-economics as the main driver toward commercialization of sCO_2 power generation. Hence the cycle was modelled based on the thermodynamic functionality of the individual components. The equations below defined how to determine the performance of the cycle, where the work of the compressor (W), re-compressor, turbine, and heater was defined by their thermodynamic correlations. $\dot{m}$ is the mass flow rate.

$$\text{Turbine} \quad W_T = \dot{m}_{CO_2}\left(h_7 - h_8\right) \tag{1}$$

$$\text{Main Compressor} \quad WM_C = \dot{m}_{CO_2} X\left(h_2 - h_1\right) \tag{2}$$

$$\text{Re-compressor} \quad WR_C = \dot{m}_{CO_2}\left(1 - X\right)\left(h_4 - h_{12}\right) \tag{3}$$

$$\text{Heater} \quad Q_H = \dot{m}_{CO_2}\left(h_7 - h_6\right) \tag{4}$$

The thermal efficiency (η_{th}) of the cycle, in other words, the performance indicator of the cycle, is given by Equation (5). The cycle thermal efficiency was calculated from the turbine power, subtracting the compression power and dividing by the total heat duty input, multiplied by a hundred percent.

$$\eta_{th} = \left(\frac{W_{T - WM_C - WR_C}}{Q_H}\right) \times 100\% \tag{5}$$

Imagine that the flow merges equation for mass conservation is ideal, which is mixed with the main flow fraction of recompression at constant pressure conditions. Then, the split fraction going into the main compressor can be expressed by Equation (6) [12].

$$X = \frac{m\dot{M}_c}{m\dot{M}_c + m\dot{R}_c} \quad (6)$$

But the constant flow fraction at constant pressure conditions for the main compression and recompression is given by Equation (7) where (1 − X) is the recompression fraction due to mass conservation.

$$(1 - X)h_4 + Xh_3 = h_5 \quad (7)$$

Equations (8) and (9), on the other hand, can be expressed as the energy conservation equations in LTR and HTR, with the assumption that both recuperators have been insulated and can exchange heat only between the hot and cold streams [12].

$$(h_9 - h_{10}) = X(h_3 - h_2) \quad (8)$$

$$(h_8 - h_9) = X(h_6 - h_5) \quad (9)$$

From Frank Incropera, in designing or estimating the performance of a heat exchanger, it is important to correlate the overall heat transfer coefficient with the heat transfer across the total surface area and the temperature variation. This can be expressed as in Equation (10), for HTR and LTR, respectively, where $\dot{Q}$ is the heat transfer, UA is the overall heat transfer and ΔTl_m is the log mean temperature difference [13].

$$\dot{Q} = \text{UA}\Delta Tl_m \quad (10)$$

$$\Delta Tl_m = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \quad (11)$$

where ΔT_1 and ΔT_2 refers to the endpoint temperature variation for counter flow heat exchangers which can be defined as in Equations (12) and (13).

$$\Delta T_1 = T_{hi} - T_{co} \quad (12)$$

$$\Delta T_2 = T_{ho} - T_{ci} \quad (13)$$

where T_{hi} refers to the inlet temperature of the hot fluid, T_{co} Refers to the inlet temperature of the cold fluid, T_{ho} Refers to the outlet temperature of the hot fluid, T_{ci} Refers to the outlet temperature of the cold fluid [13].

In computing the cost of the various components used in the current study, cost functions proposed by [3] and [14] were adapted for the studied model.

Table 2 illustrates constant values applicable to the cost functions. C_1 and C_2 used in the cost function for the compressor and turbine are correction factors that will permit better cost estimation. These constant values are consistently used in the literature on sCO_2 components cost estimates.

Values for the intercooler, recuperator, and flue gas heat exchanger as proposed by [3] were multiplied by a thousand in this present study analysis to be able to compensate for any lost in the overall heat transfer (UA) value. It is assumed that these values will minimize the doubts in the cost estimation to about plus or minus 30%, subject to the maximum material temperature and pressure required.

Table 2. Summary of constant values. [3]

Constant	Value	Unit	Application
A_1	266.3	$/(kW$_{th}$/K)	Turbine
A_2	39.5	$/(kW$_{th}$/K)	Compressor
A_3	1700	$/(kW$_{th}$/K)	Intercooler (H_2O)
A_4	2500	$/(kW$_{th}$/K)	LTR and HTR
A_5	5000	$/(kW$_{th}$/K)	Heater (Flue gas sCO_2)
A_6	1000	$/kW$_e$	Generator
C_1	1.051	-	Turbine and compressor
C_2	1.207	-	Turbine

The turbine was modeled using a general guideline method in ASPEN plus simulation with the isentropic efficiency and pressure ratio specification. These coefficients were assumed to increase the overall cycle efficiency. Equation (14) was used for the turbine cost estimation, and the formula was adapted from [3].

$$\text{Gas Turbine} = \frac{C_1 \times A_1 m\left(\frac{kg}{s}\right)\ln\left(PR\right)\times\left(1+\exp^{\left(0.036\times\left(T_{\max}-54.4\times C_2\right)\right)}\right)}{0.92-\eta_T} \quad (14)$$

where $\dot{m}$ is the mass flow rate (kg/s), P_R is the pressure ratio, η_T is the isentropic efficiency of the turbine and $T_{\max}$ is the maximum temperature of the turbine.

Compressor: the compression unit has been designed using reheating, intercooling, and split recompression to account for high compressibility. Equation (15) was used for the cost estimation [3].

$$\text{Compressor} = \frac{\mathrm{C}_1 \times \mathrm{A}_2 \mathrm{m}\left(\frac{\mathrm{kg}}{\mathrm{s}}\right)\times \mathrm{PR}\ln\left(\mathrm{PR}\right)}{0.9-\eta_\mathrm{C}} \quad (15)$$

Heat exchanger: The printed circuit heat exchanger was used for this model because they are effective in operation, less expensive, robust, and very compact, and can also withstand high pressure and temperature up to 60 MPa (600 bars) and 900˚C [15].

Because the closed Brayton cycle depends on a great sum of internal heat recuperation to increase the cycle efficiency, the model used two recuperators, the LTR and HTR. However, following an investigation of the literature, the cost of the heat exchanger can contribute to about 30% or more of the entire cycle cost [16]. Hence, there was the need to compute the cost of the heat exchanger by linking that to its thermodynamic measured parameters because the performance of a heat exchanger is dependent on the overall heat transfer coefficient and the log mean temperature difference. Equation (16) was used for the cost estimation as proposed by [3].

$$\text{Recuperator} \quad R_{HE} = A_4 \times \text{UA} \tag{16}$$

Heater: this acts as the central heating point of the working fluid, transferring the hot sCO_2 from the primary heating source to the turbine by further raising the sCO_2 pressure and temperature. The main parameter which defines its thermodynamic condition is the overall heat transfer coefficient. However, the overall heat transfer coefficient not only describes the thermal condition but also defines the size and capital cost [3].

$$\text{Heater} \quad C_{HE} = A_5 \times \text{UA} \tag{17}$$

Intercooler: this serves as an intermediate cooling device between the LTR and the main compressor, its ultimate purpose is to cool sCO_2 exiting the LTR and entering the inlet of the main compressor. The magnitude of heat rejection will determine the thermodynamic operating point of the main compressor feed. Although a preview from the literature has indicated that the intercooler and heaters thus have the same cost function, this present study analyses the intercooler cost separately from the heaters using the cost function in Equation (18).

$$\text{Intercooler} \quad I_{HE} = A_3 \times \text{UA} \tag{18}$$

Generator: the cost estimation for the generator in this model is an inclusive cost of the following: electrical equipment and materials, piping, oil seals and bearings, gearing systems, oil lubricating system, balance of plant (BOP), instrumentation and controls. The cost was estimated using the net power produced by the constant introduced in Equation (19) as proposed by [3].

$$\text{Generator} = A_6 \times \text{Net power} \tag{19}$$

3.2. Capital Expenditure (CAPEX)

To accomplish the main goal of conducting a thermo-economics analysis, there was the need to investigate the capital expenditure of the study model. Nonetheless, taking into consideration some onsite and offsite factors, but in order to obtain the optimal output of an electricity generating facility to meet the system load at the lowest possible cost subject to transmission and operational constraints. An annual capital cost was predicted based on 20-year period assumption with a 10% discount rate annually, assuming the plant will run for 8000 hours annually. Table 3 indicates techno-economic assumptions applied to the present study as proposed by [3].

Table 3. Indicates the economic assumption. [3]

Description	Value	Unit
Lifetime (n)	20	yrs.
Discount rate (i)	10	%
Annual operating hours (Oh)	8000	Hrs.
Maintenance factor (MF)	0.06	-
Fixed O&M	65	$/kW-yr.
Variables O&M	3.5	$/kWh

Other important concern parts of a cost analysis of power cycles are costs associated with procuring or maintaining the life cycle of the plant, such as fixed and variable operation and maintenance costs and levelized cost of electricity. The capital recovery factor was calculated using Equation (20). [3]

Capital Recovery Factor (CRF).

This is expressed as a ratio of a constant annuity of the current value receiving that annuity for a given period, where n is the annuity. [3]

$$\mathrm{CRF} = i \times \frac{(1+i)^n}{(1+i)^n - 1} \tag{20}$$

Cost Rate (CR) $/S

The method used to calculate the cost rate of the present study in Equation (21), was based on the evaluation of the capacity recovery factor, which corresponds to the sum of the equipment purchase cost divided by the operating hours of the plant. The total cost rate was defined from the cost rate and variable operation and maintenance (O&M) using Equation (22) as proposed by [3]. However, the fuel cost rate (FR) expressed in Equation (22) has been assumed to be zero since the cycle is assumed to integrate into a pre-existing combustion cycle.

$$\mathrm{CR} = \frac{\mathrm{CRF} \times \sum \text{purchased equipment cost} + \text{Fixed O \& M}}{\text{OPerating Hours} \times 3600} \tag{21}$$

Total Cost Rate (TR) $/S

The total cost rate is the sum of the cost rate, fuel cost, and variable operation and maintenance.

$$\mathrm{TR} = \mathrm{CR} + \mathrm{FR} + \text{Variable O \& M} \tag{22}$$

Cost of Electricity (COE) $/kWh

The COE was determined through Equation (23), as suggested by [3]. This equation was used to estimate the revenue expected by running the generator per net kWh for the first annual operation of the plant.

$$\mathrm{COE} = \frac{\mathrm{TR} \times 3600 \times 10^5}{\text{Net Power}} \tag{23}$$

Capacity Factor (CF)

The capacity factor of the plant has been expressed as a ratio of the plant's actual output over a period of time to its potential output, assuming the plant could be operated at a maximum capacity [16]. The capacity factor was calculated using Equation (24); this is to allow estimation and comparison based on the impact of capital and variable operating hours.

$$\mathrm{CF} = \frac{\text{Energy generated}}{8760 \times \text{Net power}} \tag{24}$$

Levelized Cost of Electricity (LCOE) $/kWh

This is an economic assessment that determines the cost of electricity generated over the lifespan of the plant. In other words, this also permits the comparison of diverse technologies with different scales of operation, investment, and operating periods so that policymakers can make informed decisions [17]. Equation (25) has been used to calculate the LCOE for the present study as proposed by [3].

$$\text{LCOE} = \frac{\text{CRF} \times \text{MF} \times \sum \text{PEC} + \text{FR} + \text{Variable O \& M}}{8760 \times \text{Capacity factor}} \tag{25}$$

Operation and Maintenance Expenditure (OPEX) ($/kW-yr)

Operation and maintenance costs are the expenditures related to charges connected to operating and maintaining the plant over its expected lifespan. This cost is made up of fixed operation and maintenance, which is expressed over the annual hour, while variable operation and maintenance is expressed over the operating hours, as indicated in Equation (26).

$$\text{OPEX} = \frac{\text{fixed O \& M}}{8760} + \frac{\text{variable O \& M}}{\text{Oh}} \tag{26}$$

Profit Measure Analysis

Following the cost analysis, profitability measures were conducted to assess the prospects of the sCO_2 power cycle's investments.

Return on Investment (ROI)

Return on investment was introduced as a performance measure to evaluate the amount of return on sCO_2 power cycle cost relative to the total investment expenditure. ROI was calculated based on the annual earnings over the total capital investment, as shown in Equation (27) proposed by [18].

$$\text{ROI} = \frac{\text{Annual earnings}}{\text{Total capital investment}} \tag{27}$$

Net Present Value (NPV)

The net present value was expressed using Equation (28), as proposed by [19]. This is defined as the sum of the CAPEX and OPEX by the discount rate to the lifetime of the cycle.

$$\text{NPV} = \text{CAPEX} + \sum_{k=0}^{n} \frac{\textbf{OPEX}}{(1+r)^n} \tag{28}$$

Pay Back Period (PBP)

The duration to recover the cost of investment was calculated using Equation (29) as proposed by [19]. Where SR is the sales revenue. But sales revenue is calculated using Equation (30).

$$\text{PBP} = \frac{\textbf{Total Capital cost}}{\textbf{SR} - (\textbf{Operation cost}) - (\textbf{Tax})} \tag{29}$$

3.3. Challenges of SCO_2 Brayton Cycle Power Plants

The supercritical carbon dioxide (sCO_2) Brayton Cycle Power Plants offer promising advantages over traditional power cycles, including higher efficiency and reduced environmental impact. However, their widespread adoption is hindered by several operational challenges. Here are some of the main challenges:

- **High-Temperature Materials**

Material Selection: Operating at high temperatures and pressures necessitates materials that can withstand these conditions without degrading. The availability of suitable materials is limited and often expensive [20].

Corrosion and Oxidation: sCO_2 can be corrosive, especially at high temperatures, which can lead to material degradation. Developing materials or coatings resistant to these effects is a significant challenge [21].

- **Turbomachinery Design**

Compressor and Turbine Efficiency: Designing a turbomachinery that can efficiently handle the unique properties of sCO_2 is complex. Traditional designs often need significant modifications [22].

Sealing and Leakage: Ensuring proper sealing is crucial because sCO_2 can easily leak through traditional seals, leading to efficiency losses and safety hazards.

- **Heat Exchanger Performance**

Compact Heat Exchangers: The high density and thermal capacity of sCO_2 necessitate compact heat exchanger designs, which require advanced manufacturing techniques and materials to handle high pressure and thermal stresses [23].

Fouling and Maintenance: Heat exchangers can suffer from fouling, reducing efficiency and increasing maintenance requirements. Cleaning and maintenance can be challenging due to the compact design [24].

- **Control and Integration**

Dynamic Response: sCO_2 systems require precise control systems to manage rapid changes in pressure and temperature, ensuring safe and efficient operation [25].

Integration with Other Systems: Integrating sCO_2 cycles with existing energy systems (like solar, nuclear, or fossil fuel) requires sophisticated control strategies to manage different energy inputs and outputs.

- **Startup and Shutdown Procedures**

Thermal Stresses: The thermal stresses during startup and shutdown can be significant, leading to material fatigue and failure. Proper procedures and systems must be developed to minimize these stresses.

Transient Behavior: Understanding and controlling the system's transient behavior during startup and shutdown is essential to prevent damage and inefficiencies.

4. Results

The model was operated at different turbine inlet temperatures as shown in **Table 4**, to observe the effect of temperature variation on cycle efficiency. This study's results support evidence from previous observations by [8] [11] as the temperature increases.

Table 4. Cycle efficiencies at varying temperatures.

Turbine Inlet Temperature (˚C)	Cycle Efficiency (%)
550	38.9
700	46.9
750	48.5
800	50.3
900	53.6

Figure 3 shows the cycle efficiency and the temperature of the heater concerning the inlet temperature of the turbine. The efficiency of the cycle increases progressively as the inlet temperature of the turbine increases from the heater. From **Figure 3**, when the inlet temperature of the turbine is 550˚C, the cycle efficiency is approximately 39%, as the temperature further increases to 700˚C, the efficiency is raised to 46.9%. The maximum efficiency of the cycle corresponded to the highest operating inlet temperature of the turbine at 900˚C and 53.6%, respectively. However, this was expected, as suggested by [11].

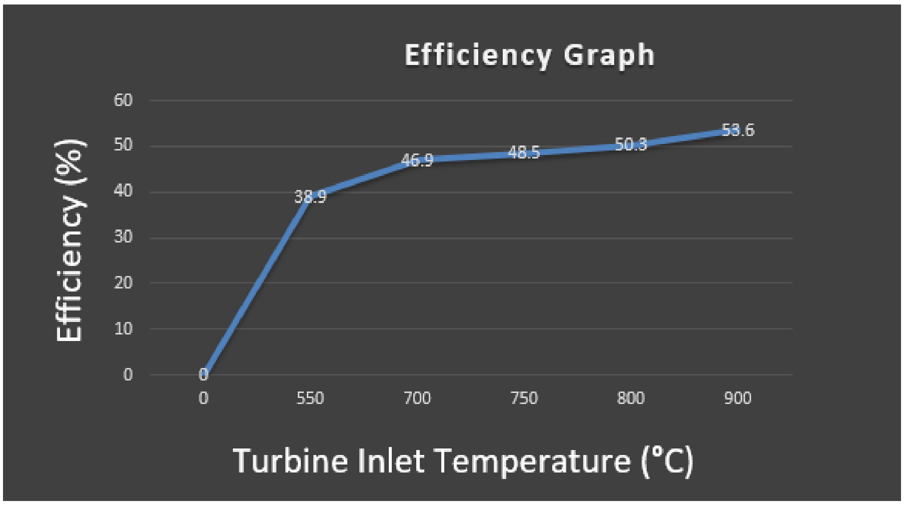

Figure 3. Efficiency TIT graph.

Table 5 shows a summary of the CAPEX for this study analysis. The total CAPEX is the total of direct and indirect costs. The direct cost represents the equipment purchase cost and offsite cost, which amounted to $48,406,028 and $27,107,376 respectively. This accounted for a total direct cost of $75,513,404. On the other hand, the indirect cost which represents other outlay costs, amounted to $57,119,112. This present research work for the sCO_2 cycle layout used for the investment cost estimation represents a total CAPEX of $132,632,515. PEC amounted to $ 48,406,028, which translates to about 36% of the total capital investment cost. The CAPEX for this study analysis was compared with the sCO_2 pilot power plant price quoted by Marion [26] to be $119 million.

Table 5. Summary of the capital expenditure.

	CAPITAL INVESTMENT COST CALCULATION FOR A 25 MW sCO_2 CYCLE			
Type of cost	Components	Detail	Quantity	Sub-Total
Direct Cost (DC)	Turbine	15,374,052	1	$15,374,052
	Compressors	2,490,215	2	$4,980,431
	Recuperators	1,020,000	2	$2,040,000

Continued

	Heater	214,635	1	$214,635
	Intercooler	88,910	1	$88,910
	Generator	25,708,000	plus auxi	$25,708,000
	Total Equipment Purchase Cost (PEC)			$48,406,028
	Offsite cost			
	Land & Civil works	21%(PEC)	Area	$10,165,266
	Service facilities 35% (PEC)			$16,942,110
	Total Direct cost	(DC)		$75,513,404
Indirect Cost (IC)	Engineering &Supervision	40% (PEC)		$19,362,411
	Contingency 30% (DC)			$22,654,021
	Construction cost & contractor profit 20% (DC)			$15,102,681
	Total Indirect cost			$57,119,112
TOTAL CAPITAL INVERSTMENT				$132,632,515
	Overnight Cost			$5,895

Table 6 indicates results for other cost evaluations, which can be used to compare alternative technologies with different scales of operating periods. LCOE and OPEX in this current study were found to be 44 and 0.007 $/kWh, respectively, while COE is 6.5 $/kWh. This result was expected as compared to [27] which quoted 44.8 $/MWh for LCOE. The current study results, however, follow previous research from [28] that suggests that the LCOE is likely to be 40.2 $/kWh by 2023-2024 for 100 - 300 MW.

Table 6. LCOE and OPEX Results.

Cost	Amount
Cost Rate ($/s)	0.20
Total Cost Rate ($/s)	3.7
Cost of Electricity($/kWh)	6.5
Levelized Cost of Electricity ($/kWh)	44
Operation and Maintenance Cost ($/kWh)	0.007

Sensitivity Analysis

Figure 4 shows a sensitivity study conducted to analyze the effect of engineering and supervision costs on the CAPEX. However, the results from the graph in Figure 4 reveal that when the percentage of engineering and supervision costs relative to the PEC increases, the sub-total cost for engineering and supervision also increases the CAPEX. At 10%, an amount of $4 million is added to the CAPEX as compared to $19 million at 40%.

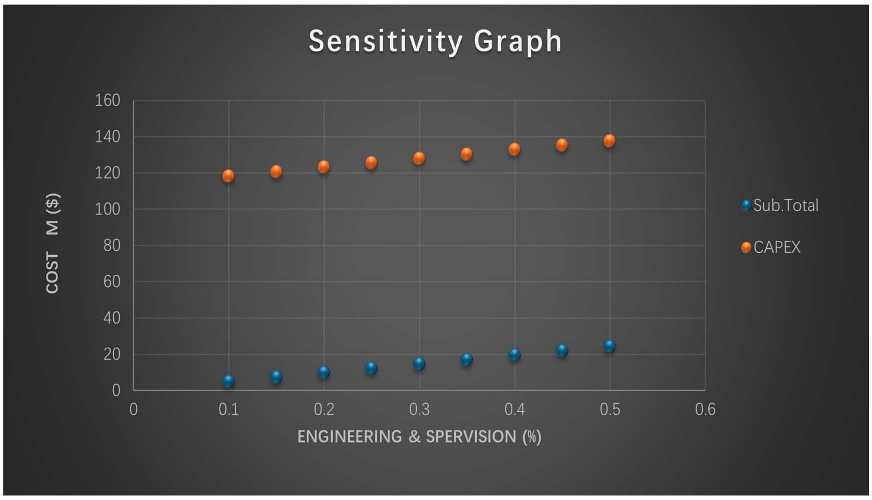

Figure 4. Sensitivity to engineering and construction costs.

Figure 5 shows a sensitivity evaluation conducted on the contingency cost. From the graph, it can be seen that when there is an increase in the percentage of contingency cost, the total indirect cost increases proportional to the sub-total of contingency cost. In this present study, contingency cost was estimated at 30% of the total direct cost. This corresponds to $22 million from the graph as indicated earlier in **Table 5**. However, when the contingency cost is reduced to 10%, the sub-total cost for contingency becomes $7 million.

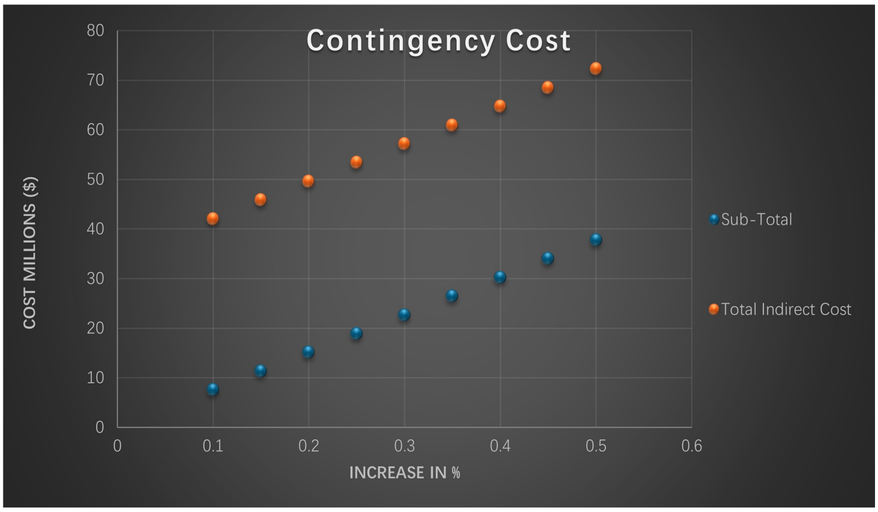

Figure 5. Sensitivity to contingency costs.

A sensitivity analysis was also carried out for land and civil works, as shown in **Figure 6**. Results, however, reveal that incremental changes in land and civil works prices place marginal increments on the total direct cost. However, the CAPEX records a significant positive value due to an increase in the direct cost. In this study results, the total direct cost contributes to about 70% of the CAPEX.

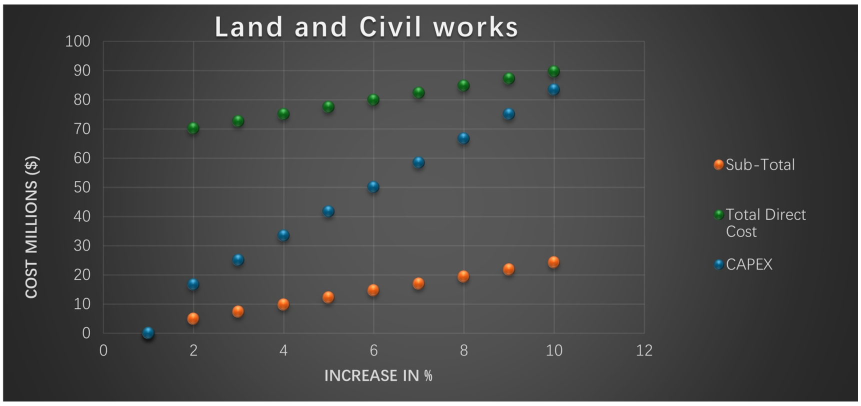

Figure 6. Sensitivity to land and civil works costs.

Calculated results from the profitability measures were used for these sensitivity analyses. Key variables such as the discount rate in time, the CAPEX, and OPEX were the main determinants in this study evaluation. From the preliminary cost estimation, the annual cash flow is estimated to be \$88 million with a projected NPV of \$19 million for the estimated period and with an estimated payback period of 9 years. Figure 7 shows sensitivity results on Net Present Value (NPV). From the graph, it can be seen that the NPV reduces as the period increases; this is an indication that the investment will yield profitable income as time increases. The sensitivity result indicates that profitability with the NPV depends much on the assumptions made and the projected lifetime period.

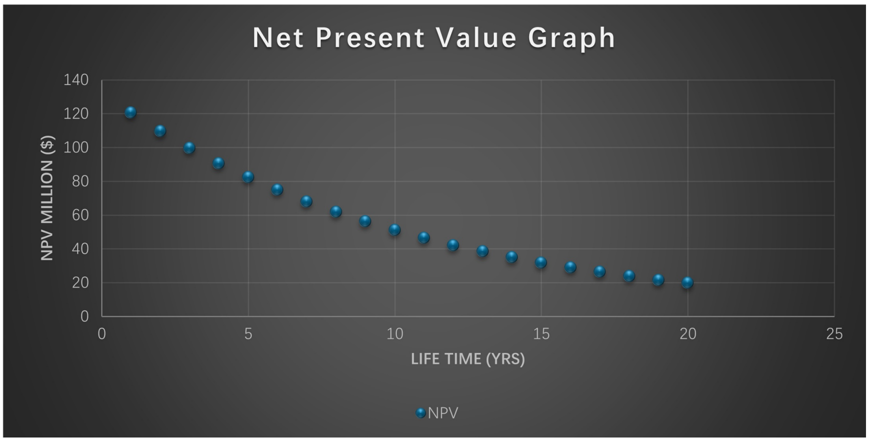

Figure 7. Net present value graph.

5. Discussion of Results

The sCO_2 power cycle configuration seems possible with compact components. However, achieving high cycle efficiency depends on the range of turbine inlet temperature and the pinch point temperature between the LTR and HTR. One interesting finding that emerged from the simulations was that increasing the operating temperature of the turbine increases the cycle efficiency. However, a much-improved cycle efficiency of about 60% plus can be obtained when the recuperators can operate with an approach temperature difference between the hot and cold fluid without crossover.

Another important finding in the thermodynamic simulation was that optimizing the cycle compression ratio improves the cycle efficiency. This is because the cycle can extract more work from the high-temperature gas expansion. More of the heat input is converted into work rather than being lost to the environment. The second law of thermodynamics tells us that the efficiency of converting heat into work is greater when heat is added at a higher temperature [11]. Hence, by increasing the turbine's operating temperature, you raise the average temperature at which heat is added to the cycle. This improves the conversion efficiency because high-temperature heat addition is more effective in producing work. An increased cycle efficiency of 58% - 60.4% was recorded at 900˚C - 1000˚C, respectively. This finding, when compared to results published by [8] [11] [29], is encouraging.

The capital expenditure for the study model includes anticipated cost estimates for the equipment, materials, and offsite costs such as land and civil works, facilities, engineering and supervision, construction, and contingency. The contingency cost estimate was included in this study analysis to account for costs that are not yet known and might have been omitted due to a lack of a complete project description. These cost units were expressed in percentages relative to the PEC based on the assumption of maturity of sCO_2 cycles.

For technologies that are not yet fully matured for commercial deployment (*i.e.*, have a low technology readiness level: TRL), the cost estimate offers a high degree of uncertainty [30]. The economic results in this work's analysis yielded an overall CAPEX of $132,632,515 with an overnight cost of $5,895. This result is higher compared to [26], therefore, it is possible to suggest that the difference shown in the results might have been caused by variations in input parameters from the thermodynamic properties database used for the components calculations. The LCOE, COE, and O & M were calculated to allow for comparison with other alternative technologies. Results for these costs look different compared to [27]. The difference is due to the CAPEX and capacity factor values used; these values serve as the main drivers for LCOE, and COE cost estimate.

6. Conclusions

This research work outlines an approach used to determine a thorough techno-economic analysis for a 25MW sCO_2 closed Brayton cycle. The analysis was carried out based on clearly defined objectives used to achieve the aim of this present study. An investigation was conducted covering literature for viable potential sCO_2 power cycles to aid in the study's analysis. Following the investigation, a thermodynamic simulation was conducted in ASPEN Plus to evaluate the performance of the components in a sCO_2 thermodynamic environment.

Parametric values such as flow rate, temperature, and pressure ratio were obtained from the simulation model and used for theoretical calculations, followed by preliminary sizing of the heat exchangers UA and ΔTl_m values (52.321 kW/K and 859 K). The UA value was the main cost driver for the heat exchangers, whereas the flow rate and pressure ratio were the cost determinants for the turbine and compressors. The cost of the generator, on the other hand, depended on the amount of power (25708 kW) generated from the cycle. The components performance evaluation was carried out based on cost function calculations.

Before the component cost function, one key finding was that an increase in temperature from (550˚C - 900˚C) does not affect the cost of the turbine. However, the flow rate and pressure ratio have an impact on increasing and decreasing the cost of both the turbine and compressors. The overall capital investment for the cycle configuration was calculated based on direct and indirect cost analysis. Techno-economic results showed that sCO_2 power cycles have low capital investment costs compared to conventional steam cycles and coal plants.

Conflicts of Interest

The author declares no conflicts of interest regarding the publication of this paper.

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