THERMO-ECONOMIC OPTIMIZATION OF HYBRID COMBINED POWER CYCLES USING HELIOSTAT FIELD COLLECTOR

by

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Abstract

Electricity has an essential role in our daily life. However, with the ever increasing cost of fossil fuels and natural gas, power generation with higher efficiency and lower capital cost is in high demand. Nowadays, global warming and climate change have become vital issues prompting investigations into increasing the share of renewable sources of energy implementation in power generation. Solar energy is arguably the most favorable solution for a greener power generation technology. With solar technology’s current level of maturity, solar energy cannot provide a significant contribution to the world’s energy demand due to intermittency and storage issues. A possible solution to the aforementioned difficulties is power plant hybridization. In particular, concentrated solar power technologies are displaying significant potential for electricity production. The United Arab Emirates’ hot, sunny climate is an indication of the great potential it possesses for hybrid and solar only power plant implementation. In this research work, the feasibility of a 50 MWe hybrid (solar and natural gas) combined cycle power plant with a topping gas turbine cycle and four different bottoming cycles are assessed. Power plant hybridization is accomplished by employing a solar tower collector (Heliostat field collector). Three rather unconventional bottoming cycle configurations have been chosen including gas turbine (air bottoming cycle), water injected gas turbine (humid air bottoming cycle), and the Maisotsenko cycle (Maisotsenko bottoming cycle). These three configurations along with the conventional combined cycle power plant (steam bottoming cycle) are optimized by conducting thermo-economic and transient analyses in MATLAB to identify the most economically justified plant configuration for the United Arab Emirates. Additionally, two different heliostat field layouts are taken into consideration including the radial-staggered and spiral layouts. Moreover, thermo-economic evaluation is accomplished by utilizing five different economic approaches, i.e. net present value, payback period, life cycle saving, Knopf objective function, and levelized cost of electricity.

Search Terms: Air bottoming cycle, steam bottoming cycle, Maisotsenko bottoming cycle, humid air bottoming cycle, hybrid, heliostat field collector, thermodynamic analysis, economic analysis, thermo-economic optimization
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<tbody>
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<td>$A$</td>
<td>Surface area [m$^2$]</td>
</tr>
<tr>
<td>$a_1 \cdots a_n$</td>
<td>Dimensionless NASA polynomial curve fit coefficients</td>
</tr>
<tr>
<td>$a, b$</td>
<td>Control variables for spiral pattern</td>
</tr>
<tr>
<td>$B$</td>
<td>Solar radiation coefficient</td>
</tr>
<tr>
<td>$C$</td>
<td>Unit cost [US$]</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Specific heat capacity [kJ/kgK]</td>
</tr>
<tr>
<td>$D$</td>
<td>Hydraulic diameter [m]</td>
</tr>
<tr>
<td>$DM$</td>
<td>Heliostat characteristic diameter [m]</td>
</tr>
<tr>
<td>$d_R$</td>
<td>Distance between the heliostat and receiver [m]</td>
</tr>
<tr>
<td>$d_{sep}$</td>
<td>Additional separation distance between adjacent heliostats [m]</td>
</tr>
<tr>
<td>$E$</td>
<td>Thermal energy [kWh], time error coefficient, effectiveness</td>
</tr>
<tr>
<td>$f$</td>
<td>Factor, focal distance [m], friction factor</td>
</tr>
<tr>
<td>$g_{on}$</td>
<td>Rate of extraterrestrial radiation on a normal surface to the radiation for the $n^{th}$ day of the year [W/m$^2$]</td>
</tr>
<tr>
<td>$g_{sc}$</td>
<td>Solar constant [W/m$^2$]</td>
</tr>
<tr>
<td>$h$</td>
<td>Specific Enthalpy [kJ/kg], height [m], convective heat transfer coefficient [W/m$^2$K]</td>
</tr>
<tr>
<td>$HR$</td>
<td>Receiver height [m]</td>
</tr>
<tr>
<td>$H_t$</td>
<td>Image dimension in the tangential plane</td>
</tr>
<tr>
<td>$I$</td>
<td>Solar heat fluxing [kW/m$^2$]</td>
</tr>
<tr>
<td>$i$</td>
<td>Loan interest rate</td>
</tr>
<tr>
<td>$k$</td>
<td>Thermal conductivity [W/mK]</td>
</tr>
<tr>
<td>$L$</td>
<td>Meridian</td>
</tr>
<tr>
<td>$LH$</td>
<td>Heliostat height [m]</td>
</tr>
<tr>
<td>$LHV$</td>
<td>Fuel Lower heating value [kJ/kg]</td>
</tr>
<tr>
<td>$LMTD$</td>
<td>Log mean temperature difference [K]</td>
</tr>
<tr>
<td>$LW$</td>
<td>Heliostat width [m]</td>
</tr>
<tr>
<td>$Lx', Ly'$</td>
<td>Analytical integration limits</td>
</tr>
</tbody>
</table>
\( M \) Molecular weight [kg/kmol]
\( \dot{M} \) mass flux [kg/m\(^2\)s]
\( \dot{m} \) Mass flow rate [kg/s]
\( mf \) Mass fraction
\( N \) Quantity
\( n \) Number of a day in the year, mirror surface number of stripes
\( \dot{n} \) Number of moles [kmol]
\( Nhel \) Number of heliostats within a row
\( Nrow \) Number of rows of heliostats for a specific zone
\( Nu \) Nusselt number
\( P \) Pressure [kPa]
\( Pr \) Prandtl number
\( \dot{Q} \) Rate of thermal energy [kW]
\( R \) Gas constant [kJ/kgK], radial distance [m]
\( r \) Pressure ratio, radius
\( r_i \) Polar radius of the \( i \)th element of the spiral pattern [m]
\( Re \) Reynolds number
\( r_{ins} \) Annual insurance rate
\( RR \) Receiver radius [m]
\( S \) Life cycle saving [MU$]$
\( s^0 \) Temperature dependent specific entropy [kJ/kgK]
\( sal \) Salary [US$]
\( T \) Temperature [K]
\( t \) Time [hour]
\( U \) Overall heat transfer coefficient [W/m\(^2\)K]
\( u \) Velocity [m/s]
\( u, v \) Overlapping rectangle dimensions
\( V \) Volume [m\(^3\)]
\( \dot{V} \) Volumetric flow rate [m\(^3\)/s]
\( \vec{V} \)  
Vector

\( n \)  
Specific volume \( [m^3/kg] \)

\( W \)  
Generated electricity \( [kWh] \)

\( \dot{W} \)  
Power \( [kWe] \)

\( w \)  
Specific work \( [kJ/kg] \)

\( W_s \)  
Image dimension in the sagittal plane

\( x \)  
Mole fraction, x axis coordinates

\( x', y' \)  
Integration coordinates \([m]\)

\( y \)  
y axis coordinates

\( Z \)  
Capital investment \([US\$]\)

\( z \)  
z axis coordinates

**Greek letters**

\( \alpha_s \)  
Solar altitude angle \([rad]\)

\( \alpha, \beta \)  
Thermo-economic coefficients

\( \beta \)  
Surface slope \([rad]\)

\( \gamma \)  
Solar azimuth angle \([rad]\)

\( \gamma_s \)  
Solar azimuth angle \([rad]\)

\( \delta \)  
Declination angle \([rad]\)

\( \Delta az \)  
Azimuth angular spacing

\( \Delta P \)  
Pressure drop \([kPa]\)

\( \Delta R \)  
Radial increment \([m]\)

\( \Delta T \)  
Temperature difference \([K]\)

\( \varepsilon \)  
Heat transfer effectiveness, emissivity

\( \varepsilon_T \)  
Tower unit vector elevation angle \([rad]\)

\( \eta \)  
Efficiency

\( \theta \)  
Incident angle \([rad]\)

\( \theta_H \)  
Angular position of the heliostat in the field from north \([rad]\)

\( \theta_{hour} \)  
Hour angle \([rad]\)

\( \theta_i \)  
Polar angle of the \( i^{th} \) element of the spiral pattern \([rad]\)
\( \theta_{\text{lat}} \)  
Latitude [rad]

\( \theta_{T} \)  
Tower elevation angle [rad]

\( \theta_{z} \)  
Zenith angle [rad]

\( \rho \)  
Mirror reflectivity, density [kg/m\(^3\)], mirror density

\( \sigma \)  
Standard deviation, Stefan Boltzmann constant [W/m\(^2\)K\(^4\)]

\( \tau \)  
Attenuation transmission coefficient

\( \phi \)  
Maintenance and installation factor

\( \omega \)  
Solar radiation incident angle [rad]

**Abbreviation**

ABC  
Air bottoming cycle

AHX  
Air heat exchanger

ASDH  
Air saturator degree of humidification

BCAH  
Bottoming cycle air humidification

CCC  
Conventional combined cycle

CEPCI  
Chemical engineering plant cost index

CRF  
Capital recovery factor

CSP  
Concentrated solar power

DNI  
Direct normal radiation

DOSH  
Degree of superheating

DSG  
Direct steam generation

GTIT  
Gas turbine inlet temperature

HABC  
Humid air bottoming cycle

HFC  
Heliostat field collector

HRSG  
Heat recovery steam generator

LCOE  
Levelized cost of electricity

LCOEN  
Levelized cost of energy

LFR  
Linear Fresnel reflector

MBC  
Maisotsenko bottoming cycle

MGTC  
Maisotsenko gas turbine cycle

NPV  
Net present value
<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>NREL</td>
<td>National Renewable Energy Laboratory</td>
</tr>
<tr>
<td>PDC</td>
<td>Parabolic dish collector</td>
</tr>
<tr>
<td>PTC</td>
<td>Parabolic trough collector</td>
</tr>
<tr>
<td>PV</td>
<td>Photo-voltaic</td>
</tr>
<tr>
<td>SBC</td>
<td>Steam bottoming cycle</td>
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<tr>
<td>SEGS</td>
<td>Solar electric generating system</td>
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**Subscript**

<table>
<thead>
<tr>
<th>Subscript</th>
<th>Description</th>
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<tr>
<td>1y</td>
<td>First year</td>
</tr>
<tr>
<td>a</td>
<td>Air</td>
</tr>
<tr>
<td>ABC</td>
<td>Airbottoming cycle</td>
</tr>
<tr>
<td>AC</td>
<td>Air cooler</td>
</tr>
<tr>
<td>ad</td>
<td>Additional</td>
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<tr>
<td>aer</td>
<td>Aerosol</td>
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<tr>
<td>AHX</td>
<td>Air heat exchanger</td>
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<tr>
<td>amb</td>
<td>ambient</td>
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<tr>
<td>ann</td>
<td>annual</td>
</tr>
<tr>
<td>approach</td>
<td>Approach</td>
</tr>
<tr>
<td>AS</td>
<td>Air saturator</td>
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<tr>
<td>ast</td>
<td>Astigmatic effect</td>
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<td>at</td>
<td>Attenuation</td>
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<td>Auxiliary</td>
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<td>Average</td>
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<tr>
<td>B</td>
<td>Bottoming</td>
</tr>
<tr>
<td>b</td>
<td>Blocking</td>
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<tr>
<td>bpipe</td>
<td>Branching pipeline</td>
</tr>
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<td>bq</td>
<td>Beam quality</td>
</tr>
<tr>
<td>brs</td>
<td>Branching station</td>
</tr>
<tr>
<td>c</td>
<td>Compressor, cold</td>
</tr>
<tr>
<td>cc</td>
<td>Combustion chamber</td>
</tr>
<tr>
<td>cell</td>
<td>Cell</td>
</tr>
</tbody>
</table>
*cg*  Common gases
*Civil*  Civil engineering
*cld*  Cloud
*CO₂*  CO₂
*con*  Condenser, construction
*cont*  Contingency
*cos*  Cosine
*cs*  Contract service
*cw*  Compressor washing
*da*  Dry air
*Dec*  Decommissioning
*dew*  Dew point
*e*  projection
*ec*  Economizer
*ele*  Electrical
*eqp*  Equipment
*ev*  Evaporator
*ex*  Exhaust
*ext*  Exterior
*f*  Fuel, field
*fan*  fan
*G*  Generator
*g*  Gas
*gk*  Ground keeping
*Gt*  Gas turbine
*GW*  Greenwich
*H*  Higher heat capacity, heliostat
*h*  Hot
*HABC*  Humid air bottoming cycle
*hel*  heliostat
helf Heliostat field
hour Hour of the day
HRSG Heat recovery steam generator
i In, \( i^{th} \)
if Indirect factor
ins Installation
int Internal
inv Investment
L Lower heat capacity, lower section of the air saturator
lab Labor
land Land
loc Local
M Mechanical
m Mixture, midpoint, maximum
mai Maintenance
MBC Maisotsenko bottoming cycle
min Minimum
mirror Mirror
mw Mirror washing
n Normal
net Net
NG Natural gas
NGS Natural gas substation
o Out
opr Operator
opt Optical, operating
outer Outer
oz Ozone
p Pump, pressure
pb Payback period
<table>
<thead>
<tr>
<th>Acronym</th>
<th>Description</th>
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<tr>
<td>pinch</td>
<td>Pinch</td>
</tr>
<tr>
<td>piping</td>
<td>Piping</td>
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<td>prs</td>
<td>Pressure reduction station</td>
</tr>
<tr>
<td>rec</td>
<td>Receiver</td>
</tr>
<tr>
<td>red</td>
<td>Reduction</td>
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<tr>
<td>ref</td>
<td>Reference</td>
</tr>
<tr>
<td>rev</td>
<td>Revenue</td>
</tr>
<tr>
<td>s</td>
<td>Isentropic, steam, sun</td>
</tr>
<tr>
<td>s&amp;b</td>
<td>Shading and blocking</td>
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<td>sat</td>
<td>Saturated</td>
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<td>sav</td>
<td>Saving</td>
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<td>SBC</td>
<td>Steam bottoming cycle</td>
</tr>
<tr>
<td>sct</td>
<td>Scattering</td>
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<tr>
<td>sh</td>
<td>Superheated, superheater</td>
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<td>sol</td>
<td>Solar</td>
</tr>
<tr>
<td>sp</td>
<td>Spillage</td>
</tr>
<tr>
<td>sre</td>
<td>Surface error</td>
</tr>
<tr>
<td>St</td>
<td>Steam turbine</td>
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<tr>
<td>STcon</td>
<td>Steam condenser</td>
</tr>
<tr>
<td>std</td>
<td>Standard</td>
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<td>stf</td>
<td>Staff</td>
</tr>
<tr>
<td>sun</td>
<td>Sun-shape</td>
</tr>
<tr>
<td>T</td>
<td>Tower, topping</td>
</tr>
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<td>t</td>
<td>Turbine</td>
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<tr>
<td>tec</td>
<td>Technician</td>
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<td>tot</td>
<td>Total</td>
</tr>
<tr>
<td>tow</td>
<td>Tower</td>
</tr>
<tr>
<td>tre</td>
<td>Tracking error</td>
</tr>
<tr>
<td>U</td>
<td>Upper section of the air saturator</td>
</tr>
<tr>
<td>uw</td>
<td>unweighted</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
</tr>
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<td>--------</td>
<td>-------------------------------------</td>
</tr>
<tr>
<td>w</td>
<td>Water, weighted</td>
</tr>
<tr>
<td>wbt</td>
<td>Wet bulb temperature</td>
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<tr>
<td>wire</td>
<td>Wire</td>
</tr>
<tr>
<td>wv</td>
<td>Water vapor</td>
</tr>
<tr>
<td>wtr</td>
<td>Water treatment</td>
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<tr>
<td>x</td>
<td>x axis</td>
</tr>
<tr>
<td>y</td>
<td>y axis</td>
</tr>
<tr>
<td>z</td>
<td>z axis</td>
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Chapter 1: Introduction

1.1. Motivation

Electricity has become an inseparable element in our daily life. However, with the ever increasing cost of fossil fuels and natural gas, power generation with higher efficiencies and lower capital cost is in high demand. Additionally, multiple reports point to the exceptional rate of increase in the world’s energy demand. A 35% growth in energy demand from 2010 to 2035 is predicted [1]. Another important factor in fossil fuel power generation is their considerable contribution to greenhouse gas emissions. It is reported that fossil fuel and natural gas power plants are responsible for 80% of electricity generation worldwide [2]. Nowadays, global warming and climate change have become vital issues prompting investigation into increasing the share of power generation implementing renewable sources of energy. Solar energy is arguably the most favorable solution for greener power generation.

Concentrated solar power (CSP) and photovoltaic (PV) are two leading solar technologies in the power generation industry. PV collectors are capable of directly converting solar radiation into electricity. On the other hand, CSP collectors convert solar radiation into thermal energy. Therefore, a heat engine must be employed to utilize the acquired thermal energy to generate electricity. CSP technology can be integrated with different types of power generation cycles including but not limited to a steam turbine cycle, gas turbine cycle, and combined cycles. With the current available solar technology maturity, solar energy cannot provide a significant contribution to the world’s energy demand due to intermittency and storage issues [3, 4]. A possible solution to the aforementioned difficulties is power plant hybridization.

There are three recommended approaches for power plant hybridization including the solarized gas turbine, the hybrid combined cycle, and the solar reforming system [5]. In the first two categories, solar energy is utilized along with a supplementary heat source to operate the plant. In other words, a portion of the necessary thermal input for power generation is provided by a renewable energy source such as CSP technology. In solar reforming, solar energy is employed to convert the fuel, mostly natural gas, into syngas. Afterward, the produced syngas is employed for power generation, bearing in mind that syngas has higher heating value. Hybridization can be considered a temporary solution for increasing the renewable energy share of
contribution in power generation. Hybrid power plants are capable of generating electricity with higher efficiency compared to solar only power plants and they are more economically justified. Additionally, storage difficulties associated with solar only power plants are alleviated by the auxiliary combustion chamber utilization during nights and low insolation periods.

In a simple gas turbine cycle, substantial waste heat is available in the exhaust gases which can be recovered and further exploited. One alternative is to employ the available waste heat in the gas turbine exhaust gases as a source of process heat [6]. There are several different industries that rely on process heat including oil production and refining, steel making, food processing, and textile industries [6]. Steam is commonly used as the heat transfer fluid in these industries. Another option is to devise a bottoming cycle with significantly lower operating temperature to generate additional power and enhance the plant’s overall efficiency. The most popular and widely used bottoming cycle is the steam (Rankine) turbine cycle. It is a well-known fact that the conventional combined cycle (CCC), i.e. topping gas turbine and bottoming steam turbine cycles, is the most thermodynamically efficient combined plant configuration [7]. Nonetheless, CCC power plants are not the most economically justified configuration for small-scale power plants less than 50 MWe [8]. For capacities less than 50 MWe, the complication and high expenses due to the heat recovery steam generator (HRSG) and steam turbine argue in favor of seeking alternatives [7].

An alternative to CCC configurations is to employ another gas turbine cycle for heat recovery purposes. This combined cycle configuration, which is referred to as the air bottoming cycle (ABC), was patented by W. Farrell in 1988 [9]. ABC has several advantages over CCC power plants such as shorter installation time, shorter start up time, lower capital investment, lower operating and maintenance cost, more compact size, and simpler operation [10-12]. Additionally, ABC’s low water consumption enables it to be implemented in regions with water shortage problems [13]. Bearing in mind that regions with high solar radiation are commonly suffering from water scarcity [3], ABC can be an interesting option for hybrid power plant installation. In particular, we are interested in a study completed by Ghazikhani et al. [14] in which steam and water injection in the bottoming cycle air were proposed and investigated. Water/steam injection in the ABC configurations improves the bottoming cycle heat recovery and
thermal efficiency. It was reported that steam and water injection enhanced the plant’s thermal efficiency from 49.83% to 52.43% and 54.63%, respectively. It is important to note that the results presented by Ghazikhani et al. [14] indicated that water injection is a more effective approach in ABC power plants.

Furthermore, the Maisotsenko gas turbine cycle (MGTC) is a recently proposed humid air turbine cycle [15-17]. In principle, MGTC is an evaporative gas turbine cycle which utilizes water addition for reduction in NO\textsubscript{x} formation and power augmentation [18, 19]. Alsharif et al. [20] conducted energy and exergy analyses for MGTC power plant configuration. In a study by the authors [21], a detailed analysis of MGTC power plant configuration with a comprehensive air saturator model was carried out. The results indicated that MGTC can be competitive with humid air turbine cycles due to its high thermal efficiency. Consequently, it was decided to combine the proposed approaches presented by Ghazikhani et al. [14] and MGTC configuration to present the Maisotsenko bottoming cycle (MBC) [22].

The United Arab Emirates’ (UAE) hot, sunny climate is an indication of the great potential it possesses for hybrid and solar only power plant implementation. A number of investigations have been conducted on the implementation of CSP technology in the UAE. In 2010, a joint study by The Petroleum Institute and the University of Maryland estimated the average direct normal radiation (DNI) for Abu Dhabi to be 400 W/m\textsuperscript{2} during a year. The reported value is a confirmation of the UAE’s considerably high potential for solar thermal power plant integration [23]. Moreover, from March to May and September to November, the monthly averaged DNI was reported to be at its highest while August and July recorded the lowest values [23]. Thus, studies on implementation of CSP technology in the UAE are highly regarded as a step toward a cleaner and more sustainable future.

1.2. Problem Statement

While hybridization of a power plant cannot be considered as a long term solution, it is certainly a temporary solution to the difficulties associated with renewable energy integrations such as intermittency and storage [3]. Hybridization will increase power plant efficiency (compared with solar only plants) and enable the plant to operate for 24 hours a day. Moreover, CSP has a low environmental impact regarding
construction materials as compared to the other renewable energy technologies [24]. Hybrid solar power plants can be competitive with conventional fossil fuel power plants in high insolation areas. The United Arab Emirates’ (UAE) hot, sunny climate is an indication of the great potential it possesses for hybrid or solar only power plant implementation. However, CCC power plants might not be the most cost effective configurations for small scale power generation (less than 50MWe) [8]. Therefore, ABC configuration is selected for the analysis of this research work. Additionally, it was decided to introduce water injection in the air bottoming cycle based on the suggestion by Ghazikhani et al. [14] to improve ABC’s thermal and economic performances. Similarly, MBC configuration can present water injection in the bottoming cycle air stream. Consequently, MBC can also be an alternative for power plant configurations.

By considering all these issues, it is of high interest to investigate the feasibility of a 50 MWe hybrid (solar and natural gases) power plant in Abu Dhabi. Power plant hybridization is accomplished by employing a solar tower collector (Heliostat field collector). Three rather unconventional bottoming cycle configurations have been chosen including gas turbine (air bottoming cycle), water injected gas turbine (humid air bottoming cycle), and the Maisotsenko cycle (Maisotsenko bottoming cycle). These three configurations along with the conventional combined cycle power plant (steam bottoming cycle) are optimized by conducting thermo-economic and transient analyses in MATLAB to identify the most economically justified plant configuration for the United Arab Emirates. Additionally, two different heliostat field layouts are taken into consideration including radial-staggered and spiral layouts. Moreover, thermo-economic evaluation is accomplished by utilizing five different economic approaches, i.e. net present value, payback period, life cycle saving, Knopf objective function, and levelized cost of electricity.
Chapter 2: Background and Literature Review

In this section, an overview of the basic concepts and information required to fully comprehend the analysis and result of this research work are presented. Moreover, an extensive literature survey is presented to better understand the significant contributions of this research work.

2.1. Solar Radiation

This part of the report concentrates on different aspects of the sun and solar radiation in regard to CSP technology. Additionally, important solar definitions are presented for further understanding of the discussed solar technologies.

2.1.1. Sun-earth relationships.

There are different sources of energy available for human beings; however, 99.9% of the energy available on earth is provided by the sun [25]. The sun is made of concentrated gases in a spherical shape with a diameter of $1.39 \times 10^9 \text{m}$ [26]. It is located on average $1.5 \times 10^{11} \text{m}$ away from earth. However, because the earth’s polar axes are tilted at an angle of 23.45 degrees [27], the earth’s eccentric orbit causes a variation in its distance from the sun by 1.7% [26]. Furthermore, the sun’s surface temperature is 5777 K [26]. The amount of available solar radiation in outer space does not experience a significant variation. Figure 1 sums up all the important information regarding the sun-earth relationship. Solar constant $G_{sc}$ is the amount of instantaneous energy received per unit area on a surface perpendicular to the direction of the solar radiation which is located at an average distance between sun and earth.

2.1.2 Extraterrestrial solar radiation.

There are two main reasons behind the reported variation in extraterrestrial radiation (amount of radiation that would be received by the earth in the absence of atmosphere). First, it is reported that sun radiation experiences an insignificant alteration of less than $\pm 1.5\%$. Second, alternation of the sun-earth distance results in a fluctuation in extraterrestrial radiation for $\pm 3.3\%$ [26]. Figure 2 demonstrates the variation in extraterrestrial radiation ($G_{on}$) during a year. $G_{on}$ is the amount of extraterrestrial radiation on a normal surface to the radiation for the nth day of the year and can be calculated by the following equations [26]:
\[ G_{on} = \begin{cases} G_{sc} \left( 1 + 0.033 \cos \frac{360n}{365} \right) \\
G_{sc}(1.000110 + 0.034221 \cos B + 0.00128 \sin B + 0.000719 \cos 2B + 0.000077 \sin 2B) \end{cases} \quad (2.1) \]

where \( n \) is the number of days within a year which can be from 1 to 365 and \( B \) is calculated by [26]:

\[ B = (n - 1) \frac{360}{365} \quad (2.2) \]

Figure 1: Important information regarding sun-earth relationship [6]
2.1.3. Atmospheric solar radiation.

Solar radiation entering the atmosphere may be scattered by the atmosphere into many different directions. Part of the scattered radiation may reach the earth’s surface while the remaining might return to space. Therefore, total solar radiation reaching the earth’s surface either comes directly from the sun or is diffused in the atmosphere before reaching the earth’s surface. It is important to distinguish between these solar radiations because CSP technologies can only benefit from direct radiation. Beam radiation or direct radiation is the solar radiation collected from the sun directly without any disruption by the atmosphere. On the other hand, diffuse radiation is referred to as the portion of solar radiation received from the sun whose direction has been changed by the atmosphere. The sum of these two radiations on a surface is labeled as total solar radiation (global solar radiation). Typically, the horizontally oriented surface is employed for the total solar radiation measurement which leads to global horizontal radiation assessment. Another important indicator in solar energy is the direct normal
radiation \((DNI)\) that is the normal component of the beam radiation and it is calculated by [25]:

\[
DNI = G_{on} \cos(\theta_x) \tau_{sct} \tau_{wv} \tau_{oz} \tau_{cg} \tau_{aer} \tau_{cld}
\]  

(2.3)

where \(\theta_x\) is the angle between solar radiation and the normal on the horizontal surface, \(\tau_{sct}\) is the attenuation transmission coefficient for scattering, \(\tau_{wv}, \tau_{oz}, \tau_{cg}, \tau_{aer}, \tau_{cld}\) are the attenuation transmission coefficients for water vapor, ozone, common gases (\(O_2\) and \(CO_2\)), aerosol, and clouds, respectively.

### 2.1.4. Important solar definition.

Other important definitions that must be provided before proceeding to more advanced solar integrated technologies are explained in the following section. Solar time is the time based on the position of the sun in the sky. For example, solar noon is the time that the sun is at its highest point in the sky. Solar time is important as it is constantly utilized in all the presented solar angle relationships. Solar time can be calculated based on the observer’s location (longitude) and the location that the local standard time is based on. Four minutes is taken for each degree deviation from the reference meridian [26]. Solar time is calculated by [26]:

\[
t_{sol} = t_{std} + 4(L_{std} - L_{loc}) + E
\]  

(2.4)

where \(t_{sol}\) and \(t_{std}\) are the solar and standard time, respectively, \(L_{std}\) and \(L_{loc}\) are the standard meridian for the local time zone and the meridian of the observer, and \(E\) is the equation of time that can be calculated by [26]:

\[
E = 229.2(0.000075 + 0.001868 \cos B - 0.032077 \sin B - 0.014615 \cos 2B - 0.04089 \sin 2B)
\]  

(2.5)

And \(L_{std}\) is computed by [26]:

\[
L_{std} = 15(t_{std} - t_{GW})
\]  

(2.6)

where \(t_{GW}\) is Greenwich Mean Time.

Figure 3 illustrates some of the major sun-angle relationships. These definitions are required to study the sun and its beam radiation on a surface:
• Latitude ($\theta_{lat}$) is an angular location with respect to the equator. It is positive for
the northern hemisphere and negative for southern. Its value varies
between $-90^\circ \leq \phi \leq 90^\circ$.

• Declination ($\delta$) is the angular location of the sun at solar noon compared to the
equator plane. Its value varies from $-23.45^\circ \leq \delta \leq 23.45^\circ$ (north is positive).

• Slope ($\beta$) is the slope of the surface receiving the solar beam with respect to the
horizon and ranges from $0^\circ \leq \beta \leq 180^\circ$.

• Surface azimuth angle ($\gamma$) is the angle between the projection of the normal of the
surface on a horizontal plane and the local meridian while south is taken to be zero.
It ranges from $-180^\circ \leq \gamma \leq 180^\circ$ (west as positive).

• Hour angle ($\theta_{\text{hour}}$) is the angular location of the sun based on the earth’s rotation
around its axis for $15^\circ$ per hour (morning is positive).

• Angle of incidence ($\omega$) is defined as the angle between the beam radiation on a
plane and the normal of that plane.

Following are other angle definitions related to the situation of the sun in the sky:

• Zenith angle ($\theta_z$) is the angle between the line of the beam radiation and the normal
to a horizontal plane.

• Solar altitude angle ($\alpha_s$) is the angle between the line of the beam radiation and the
horizontal. This angle is the complement of the zenith angle.

• Solar azimuth angle ($\gamma_s$) is the deviation of the projection of the beam radiation on
a normal plane from south. East is considered to be negative.

The declination angle can be approximated by [26]:

$$\delta = 23.45 \sin \left(360 \frac{284 + n}{365}\right) \quad (2.7)$$

Moreover, the following equations enable us to determine any unknown angle
[26]:

$$\cos \theta = \sin \delta \sin \phi \cos \beta - \sin \delta \cos \phi \sin \beta \cos \gamma + \cos \delta \cos \phi \cos \beta \cos \omega$$
$$+ \cos \delta \sin \phi \sin \beta \cos \omega + \cos \delta \sin \beta \sin \gamma \sin \omega \quad (2.8)$$
\[ \cos \theta = \cos \theta_z \cos \beta + \sin \theta_z \sin \beta \cos (\gamma_s - \gamma) \] (2.9)

Figure 3: Important angles and definitions in sun-angle relationships [26]

2.1.5. Selection of a proper site location

The best indicator in selecting a potentially superior location for solar thermal integration is the location’s amount of annual DNI. Figure 4 demonstrates a map of the world with the amount of DNI integrated over a year, noting that 2000 kWh/m² is the minimum annual DNI required for solar thermal implementation, whereas 2500 kWh/m² of annual DNI is an indication of the competitiveness of solar thermal power plants with fossil fuel power generation plants [28].
Figure 4: Direct normal irradiation averages annual sum [28]

The above figure is an indication of the potential which the Middle East has in integrating solar thermal power plants for future energy production. Moreover, the land factor has to be considered as the other main element in selecting a proper location. Ground structure, slope, underground water level, and sand shifting are the other factors that can disqualify a location for solar thermal integration. Appropriate regions for implementation of solar thermal power technologies are depicted in Figure 5. The results present the Middle East, Australia, and northern Africa as possible candidates for CSP integration. In this research work, Abu Dhabi, located in the Middle East, was selected for solar thermal power plant integration.
2.2. Abu Dhabi

Abu Dhabi was selected as the plant location due to its high potential for solar thermal power plant implementation. Therefore, it is important to present substantial information regarding its climate and solar radiation to better understand its capability for solar thermal power generation. In this section, Abu Dhabi’s dry bulb temperature, relative humidity, and rate of DNI received annually are presented.

2.2.1. Air temperature.

Table 1 represents the average dry bulb temperature of Abu Dhabi during the day for each month [29]. It is clearly shown that the average temperature reaches its maximum during noon whilst August recorded the highest average temperature. Air temperature has a significant effect on the gas cycle thermal performance; therefore, it must be taken into account for designing a combined power plant employing a topping gas turbine cycle. Poullikkas [8] reported that an increase in inlet air temperature can have a severe impact on the plant’s performance. Consequently, an inlet air cooling technology can be utilized to improve plant performance. Though, designing an inlet air cooling system is beyond the scope of this research work, the authors proposed an innovative inlet air cooling technology integrating Maisotsenko cooler and desiccant cooling systems in [30].

2.2.2. Relative humidity.

Relative humidity is another important factor that affects cycle performance significantly. The effect of relative humidity on cycle performance was studied by Poullikkas [8], and the result indicated that humid climates diminish the plant’s thermal performance. Furthermore, it was pointed out that the reduction of power output, however insignificant and negligible, is taken into account nowadays because of an increase in the size of the gas turbine and introduction of steam or water injection in gas turbines. Nevertheless, other studies regarding inlet air humidity ratio growth indicate an improvement for gas turbine power output and efficiency. Probably, the main reason behind the difference between the acquired results can be the size of the gas turbine that has been considered for investigation. However, it can be confirmed that humid climates diminish the performance of humid air turbine cycles [8]. Note Abu Dhabi’s hourly relative humidity for a complete year presented in Table 2. These data
have been taken from the National Renewable Energy Laboratory (NREL) database
[29].
Table 1: Abu Dhabi temperature during a day [29]
Tim
e of
day
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
21
22
23

JAN
Temp.
(℃)
15.651
15.212
14.771
14.154
13.803
13.654
13.267
14.106
16.187
18.277
20.632
21.806
22.58
22.945
23.116
22.661
22.058
20.783
19.516
18.751
17.983
17.403
16.7
16.1

FEB
Temp.
(℃)
16.928
16.314
15.867
15.735
15.035
14.464
14.267
15.407
17.825
20.342
22.789
24.546
25.760
26.196
25.957
25.7
24.596
23.253
21.960
20.917
19.878
18.960
18.146
17.664

MAR
Temp.
(℃)
19.771
19.393
18.977
18.787
18.487
18.141
17.890
19.167
21.145
23.103
25.106
26.396
26.958
27.125
26.851
26.590
25.858
24.774
23.367
22.403
21.983
21.074
20.425
20.054

APR
Temp.
(℃)
23.336
22.836
22.51
21.976
21.83
21.34
21.71
24.2
26.803
29.03
30.836
31.763
32.7
32.046
31.466
30.886
29.636
28.573
27.313
26.456
25.716
25.273
24.626
23.99

MAY
Temp.
(℃)
26.509
26.067
25.696
25.077
24.919
24.687
25.477
29.106
32.458
34.632
36.471
37.471
37.335
37.054
36.364
35.909
35.048
33.629
31.732
30.416
29.635
28.696
27.916
27.212

JUN
Temp.
(℃)
28.956
28.203
27.843
27.303
27.143
26.803
28.02
30.89
33.743
35.846
37.943
38.846
39.216
39.13
38.966
37.906
37.023
35.6
33.856
32.44
31.576
30.753
30.016
29.493

JUL
Temp.
(℃)
30.754
30.274
29.729
29.348
29.274
28.945
29.319
31.880
34.351
36.780
38.829
40.158
40.693
40.854
40.651
39.567
38.235
36.951
35.812
34.516
33.509
32.748
32.138
31.432

AUG
Temp.
(℃)
31.564
31.248
30.290
29.832
29.564
29.148
29.222
31.661
34.654
36.767
39.129
40.641
40.874
41.354
40.790
39.941
38.571
37.290
36.112
34.896
33.9
33.306
32.480
31.980

SEP
Temp.
(℃)
29.49
28.903
28.45
27.896
27.63
27.303
27.53
29.736
31.91
33.803
36.52
38.196
38.816
39.243
38.6
37.583
36
34.55
33.516
32.486
31.76
31.006
30.426
29.86

OCT
Temp.
(℃)
25.535
25.006
24.293
23.693
23.277
22.909
23.132
25.616
28.767
30.758
33.151
34.741
34.683
34.764
34.283
33.158
32.129
30.764
29.535
28.371
27.803
27.129
26.412
25.806

NOV
Temp.
(℃)
21.88
21.193
20.67
20.11
19.57
19.443
19.136
20.853
23.05
25.88
27.873
29.03
29.796
29.89
29.53
29.086
27.933
26.726
25.513
24.753
23.833
23.5
22.816
22.403

DEC
Temp.
(℃)
17.735
17.212
16.725
16.561
16.103
15.767
15.548
16.174
18.241
20.709
23.119
24.616
25.045
25.203
25.054
24.671
23.922
22.603
21.377
20.535
19.893
19.212
18.5
17.887

Table 2: Abu Dhabi relative humidity during a day [29]
Time
of
day
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
21
22
23

JAN
(%)

FEB
(%)

MAR
(%)

APR
(%)

MAY
(%)

JUN
(%)

JUL
(%)

AUG
(%)

SEP
(%)

OCT
(%)

NOV
(%)

DEC
(%)

81
82.12
81.80
83.35
83.54
84.03
84.51
82.80
77
70.70
61.41
54.70
50.61
47.80
47.16
49.51
53.54
61.32
68.87
72.80
76.19
77.77
78.32
79.74

79.03
80.14
80.92
80.42
81.10
82.32
81.57
76.03
69.75
61.96
51.64
44.25
40.85
38.21
40.57
42.78
46.78
53.32
61.32
65.39
70.07
73.78
76.35
77.46

77.87
79.77
80.12
80.51
81.58
82.87
83.45
78.58
70.58
65.32
56.29
51.96
51.35
51.38
51.096
53.25
54.51
61.29
67.64
72
73.32
75.41
77.80
78.06

69.1
69.2
70.66
73.03
72.36
72.4
72.4
61.2
51.96
44.56
35.3
32.3
30.56
33.13
35
38.83
43.16
47.2
52.23
56.93
60.9
64.16
66.03
67.8

65.93
66.35
65.67
67.80
67.35
67.96
64.90
52.06
40.74
32.41
26.838
24.61
26.32
28.93
30.12
32
34.64
41.06
50.87
57.74
59.90
64.48
64.12
65.22

69.06
71.03
70.03
70.96
69.3
68.43
64.06
55.2
47.43
40.6
32.3
30.06
27.53
29.1
30.6
34.7
37.36
43.76
51.1
58.33
61.8
64.43
67.16
68.4

70.80
71.16
72.70
72.58
71.54
72.19
70.58
62.51
51.38
41.87
35.38
32.45
30.74
30.22
31.29
35.38
40.74
46.22
51.51
58.58
63.25
66.29
68.35
70.06

67.96
67.48
69.70
69.58
68
68.38
68.22
58.96
45.96
38.70
31.774
28.22
28.32
29.22
32.77
36.29
40.74
47.58
52.32
59.19
62.90
65.41
67.77
68.32

79.86
81.76
82.93
84.73
82.8
82.46
80.63
68.6
56.83
48.93
37.63
32.63
30.8
31.4
35.06
39.73
47.36
55.63
61.36
66.56
69.63
73.86
76.2
77.83

72.80
74.70
75.25
76.58
75.70
74.77
74.16
64.38
54.06
47.90
38.32
32.45
34.32
36.25
38.64
42.70
48.16
55.58
61.77
66.93
68.35
70.67
71.48
71.80

83.43
84.73
85.3
86.23
86.96
86.16
85.36
79.1
71.93
63.33
56.6
51.3
48.03
47.86
48.9
50.3
55.1
63.5
71.53
74.3
77.83
79.4
81.06
82.633

76.45
77.64
78.51
77.93
77.83
79.19
80.29
76.03
68.67
60.77
52.67
48.06
47.06
46.06
47.38
49.38
52.48
60.06
65.12
69.19
70.51
72.25
74.45
75.67

2.2.3. Direct normal radiation.
The most important factor that must be taken into consideration for CSP
analysis and solar thermal power generation is the rate of DNI available within the
38


selected site’s location. The hourly averaged rate of DNI available for Abu Dhabi is tabulated in Error! Not a valid bookmark self-reference.. It can be seen from these results that the rate of DNI supplied is maximum during noon.

2.2.5. Water scarcity.

Water scarcity is another issue that has to be addressed in designing a power plant. Water treatment facilities are responsible for a massive portion of power plant capital and operating cost. In particular, UAE’s annual amount of rainfall is significantly lower than the country’s annual water consumption. Therefore, power plants with lower specific water consumption are a priority.

Table 3: Abu Dhabi DNI during a day [29]

<table>
<thead>
<tr>
<th>Time of day</th>
<th>JAN DNI (W/m²)</th>
<th>FEB DNI (W/m²)</th>
<th>MAR DNI (W/m²)</th>
<th>APR DNI (W/m²)</th>
<th>MAY DNI (W/m²)</th>
<th>JUN DNI (W/m²)</th>
<th>JUL DNI (W/m²)</th>
<th>AUG DNI (W/m²)</th>
<th>SEP DNI (W/m²)</th>
<th>OCT DNI (W/m²)</th>
<th>NOV DNI (W/m²)</th>
<th>DEC DNI (W/m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
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<tr>
<td>6</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>19.6</td>
<td>70.7</td>
<td>66.1</td>
<td>23.5</td>
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<td>0</td>
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</tr>
<tr>
<td>7</td>
<td>96.0</td>
<td>141.3</td>
<td>182.3</td>
<td>277.2</td>
<td>332.4</td>
<td>323.5</td>
<td>237.9</td>
<td>222.8</td>
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<td>197.0</td>
<td>211.3</td>
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<td>8</td>
<td>234.5</td>
<td>416.6</td>
<td>371.1</td>
<td>475.6</td>
<td>572.6</td>
<td>576.6</td>
<td>471.2</td>
<td>477.0</td>
<td>507.9</td>
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<td>485.4</td>
<td>367.6</td>
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<td>543.5</td>
<td>639.9</td>
<td>502.0</td>
<td>596.7</td>
<td>688.2</td>
<td>703.0</td>
<td>611.1</td>
<td>650.6</td>
<td>687.4</td>
<td>728.0</td>
<td>679.9</td>
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<td>726.9</td>
<td>552.5</td>
<td>629.2</td>
<td>771.1</td>
<td>793.7</td>
<td>710.4</td>
<td>749.4</td>
<td>794.1</td>
<td>819.6</td>
<td>754.1</td>
<td>657.6</td>
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<td>577.8</td>
<td>661.7</td>
<td>813.6</td>
<td>854.6</td>
<td>807.0</td>
<td>816.6</td>
<td>856.8</td>
<td>848.4</td>
<td>751.8</td>
<td>670.7</td>
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<td>722.0</td>
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<td>585.6</td>
<td>648.9</td>
<td>836.5</td>
<td>859.6</td>
<td>794.6</td>
<td>830.4</td>
<td>867.5</td>
<td>833.6</td>
<td>735.8</td>
<td>660.0</td>
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<td>709.5</td>
<td>809.1</td>
<td>580.8</td>
<td>648.3</td>
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<td>855.1</td>
<td>791.3</td>
<td>825.6</td>
<td>843.6</td>
<td>814.6</td>
<td>735.8</td>
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<td>583.6</td>
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<td>802.0</td>
<td>816.2</td>
<td>735.8</td>
<td>777.8</td>
<td>779.1</td>
<td>761.4</td>
<td>694.1</td>
<td>617.7</td>
</tr>
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<td>605.5</td>
<td>700.2</td>
<td>561.5</td>
<td>546.8</td>
<td>714.8</td>
<td>728.8</td>
<td>644.6</td>
<td>686.0</td>
<td>686.8</td>
<td>634.8</td>
<td>562.2</td>
<td>531.1</td>
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<td>583.1</td>
<td>470.3</td>
<td>452.1</td>
<td>574.7</td>
<td>582.5</td>
<td>508.7</td>
<td>506.2</td>
<td>417.6</td>
<td>365.5</td>
<td>308.0</td>
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<td>253.8</td>
<td>234.1</td>
<td>217.7</td>
<td>282.4</td>
<td>288.2</td>
<td>232.9</td>
<td>201.3</td>
<td>101.5</td>
<td>10.8</td>
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<td>2.8</td>
<td>0</td>
<td>0</td>
<td>0</td>
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<td>0</td>
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2.2.4. Electricity consumption.

Abu Dhabi has the highest electricity demand in the UAE [23]. It is reported that Abu Dhabi’s peak load demand in August 2009 was 6255 MWe whilst 5516 MWe was recorded in August 2008 [23]. It can be noted that electricity consumption is increasing in Abu Dhabi which can be broadened to reflect an increase in electricity demand within the country. Consequently, more power plants and higher electricity production are in high demand and is the subject of extensive research. Figure 6 illustrates the growth in electricity consumption in Abu Dhabi which clearly demonstrates the rising demand for electricity in the coming years. ADWEC forecasts
the electricity consumption peak for 2015 will reach 11200 MWe compared to 6,885 MWe in 2010 [23]. In addition, Figure 7 depicts the electricity capacity and peak load demand for Abu Dhabi. It should be noted that the economic crisis in 2008 did not prevent continued growth in electricity demand in the UAE, and the country’s reserves of the natural gas were not sufficient to generate the required electricity demand in recent years [23]. Therefore, UAE officials are investigating various solutions to reduce fossil fuel and natural gas consumption in electricity production. In consequence, the solar thermal power plant is a promising solution to the aforementioned concerns.

2.2.6. Carbon dioxide emission.

CO₂ emission is increasing in the UAE. It was reported that per capita CO₂ emissions rose from 26.435 ton/pop to 32.186 ton/pop in a period of only three years [4]. Figure 8 illustrates the growth in the UAE’s CO₂ emission. In addition, the distribution of CO₂ emission from fuel consumption by different sectors of industry in the UAE is depicted in Figure 9. It is noteworthy that 39% of the CO₂ emission is associated with electricity and heat production. This amount can be significantly reduced by introducing renewable energy for electricity and heat applications.

2.2.7. Concentrated solar power projects in the UAE.

There are several studies focusing on solar thermal and CSP technologies integrated within the UAE. Shams 1 is the first solar thermal power plant utilizing CSP technologies in the UAE. Shams 1 designed total power output is 120 MWe which regards as the largest single unit CSP integrated solar thermal power plant in the world [31]. This power plant located at Madinat Zayed utilizes 768 solar collector assemblies (258,048 mirrors) to heat the oil inside the tubes [32]. The temperature of the oil inside the tubes varies from 293°C to 393°C. The heated oil is employed in the steam generator to deliver the required thermal input for the plant’s operation [23]. Additionally, Shams 1 is the first CSP integrated solar thermal power plant implementing dry cooling to reduce the amount of water consumed within the cooling tower [31]. Furthermore, it employs a supplementary heater to superheat steam temperature from 380°C to 540°C and improve the plant overall efficiency. The supplementary heater operates by utilizing fossil fuels accounted for approximately 45% of power generated.
Figure 6: Growth in electricity consumption in Abu Dhabi [23]

Figure 7: electricity capacity and peak load demand for the emirate of Abu Dhabi [23]
Beam-Down Solar Tower is another ongoing project in the UAE which is investigating a new design for solar tower or heliostat field collectors [23]. This project involves The Masdar Institute of Science and Technology, Tokyo Institute of Science and Technology, and Japan Cosmo Oil. A pilot plant with a capacity of 100 kWe has already been built near Masdar City [23]. The proposed configuration employs a set of secondary mirrors redirecting reflected solar radiation toward the air receiver which
is located on the ground [32]. The pilot plant employs 33 mirrors on the ground with two axis tracking systems [32].

2.3. Concentrated Solar Power Technology

In this section, a thorough explanation of different CSP technologies available in the market is presented. Furthermore, a comparison between CSP and other technologies which implemented solar energy to produce electricity is provided. Finally, different types of CSP collectors along with their advantages, significance, and innovative designs are explained.

Worldwide investigation of affordable renewable and sustainable energy resources has become a societal challenge for humanity with the constant reduction of fossil fuel sources [33]. There are several studies available in literature investigating different aspects of renewable energy integration [34-38]. For instance, techno-economic assessment of a ground source heat pump system in eastern Turkey is accomplished by Esen et al. [39] whereas experimental analysis for a solar assisted ground source heat pump system is presented by Esen et al. [40]. Moreover, techno-economic evaluation of ground-coupled and air-coupled heat pump systems are carried out by Esen et al. [41]. Additionally, comparative and experimental evaluations for different renewable energy sources employment in greenhouse heating are accomplished by Esen and Yuksel [42]. Furthermore, economic analysis of hydrogen production for municipal fuel cell buses is conducted by Hatch et al. [43]. Additionally, a new observational wind atlas is presented by utilizing a proposed methodology by Doubrawa et al. [44]. Finally, a new concept of wave energy conversion capable of trapping ocean wave energy into a basin is introduced by Saadat et al. [45].

Nonetheless, it should be noted that the amount of energy provided by the sun in just one hour is greater than the annually required energy for the entire planet [46]. Therefore, solar energy has great potential to be the answer for a cleaner future. For instance, compressed air energy storage is proposed for bulk storage of electric energy in solar and wind systems by Safaei and Keith [47]. Furthermore, Sandler et al. [48] utilize solar thermal energy to generate steam for thermal enhanced oil recovery. Moreover, penetration of solar power without utilizing any type of storage is investigated by Stodola and Modi [49]. In another study, modeling and optimization of a new hybrid solar thermoelectric system utilizing a thermosyphon is presented by
Miljkovic and Wang [50]. Furthermore, cost-effective design of ringwall storage hybrid power plants is presented by Weibel and Madlener [51]; whilst, Tri-generation based hybrid power plants with energy storage are investigated by Pazheri [52].

There are two main technologies employing solar energy to generate electricity: CSP and PV collectors. PV directly converts solar radiation into electricity by semiconductors and photoelectric effects while CSP captures the thermal energy within solar radiation by implementing mirrors and collectors that direct solar radiation toward the receiver. In a study by the authors [53], photovoltaic/thermal collectors are integrated with desiccant based air conditioning systems to be implemented in residential and commercial buildings for air conditioning and power generation. Concentrating solar radiation heats up the fluid in the receiver (mostly synthetic oil or steam). Heated fluid is expanded in an appropriate type of turbine to generate mechanical work. The produced mechanical work is converted into electricity in the generator. While the concept is easy, CSP has been studied for decades in order to achieve higher efficiency and lower capital cost. There is another solar technology which is under investigation called non-concentrating solar power. This technology, which is currently in the demonstration phase, consists of solar updraft tower and solar pond. These technologies are not mature enough at the moment and more studies must be conducted in order to commercialize them.

The main advantage of CSP technology over other solar technologies is its capability to be partially implemented in a conventional power plant, bearing in mind that solar technologies are economically unfavorable. Furthermore, solar energy, in particular, suffers from intermittency and storage issues [3]. Therefore, power generation cannot fully rely on solar energy. A possible solution to the aforementioned difficulties is power plant hybridization. Beretta et al. [5] proposed three approaches of power plant hybridization comprising hybrid combined cycles, solar reforming systems, and solarized gas turbine cycles. Part of the required thermal input is provided by CSP technology in hybrid combined cycles and solarized gas turbine cycles, while solar energy is utilized to convert natural gas into syngas in solar reforming systems. However, hybridization can only be considered as a temporary solution in order to reduce the CO$_2$ emission associated with power generation. Additionally, hybrid power plants can constantly operate during low insulation periods without any storage.
difficulty. Finally, hybrid power plants are a more economically attractive option with the CSP technology’s current level of maturity.

CSP systems can be categorized based on their types of collectors. There are four major CSP collectors available in the industry that are presented in Figure 10:

i. Parabolic Trough Collectors (PTC)
ii. Linear Fresnel Reflectors (LFR)
iii. Heliostat Field Collector (HFC)
iv. Parabolic Dish Collectors (PDC)

Table 4 gives a brief description and specifications of the four CSP technologies.

Table 4: Description and specifications of the four CSP technologies [46]

<table>
<thead>
<tr>
<th>Collector type</th>
<th>Relative thermodynamic efficiency</th>
<th>Operating temp. range (°C)</th>
<th>Relative cost</th>
<th>Concentration ratio (1000 W/m²)</th>
<th>Technology maturity</th>
<th>Tracking</th>
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<tbody>
<tr>
<td>PTC</td>
<td>Low</td>
<td>50-400</td>
<td>Low</td>
<td>15-45</td>
<td>Very mature</td>
<td>One axis</td>
</tr>
<tr>
<td>LFR</td>
<td>Low</td>
<td>50-300</td>
<td>Very low</td>
<td>10-40</td>
<td>Mature</td>
<td>One axis</td>
</tr>
<tr>
<td>HFC</td>
<td>High</td>
<td>300-2000</td>
<td>High</td>
<td>150-1500</td>
<td>Most recent</td>
<td>Two axis</td>
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<tr>
<td>PDC</td>
<td>high</td>
<td>150-1500</td>
<td>Very high</td>
<td>100-1000</td>
<td>Recent</td>
<td>Two axis</td>
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Figure 10: Four major types of collectors available in the industry [46]
2.3.1. Parabolic trough collector.

The parabolic trough collector (PTC) is the most mature CSP collector technology which is widely used in solar thermal power plants around the world. For instance, Solar Electric Generating Systems (SEGS) is a solar only thermal power plant utilizing PTC technology to generate electricity [46]. SEGS solar thermal power plant, which is located in a California desert with the capacity of 354 MW, is the largest PTC power plant in the world. The PTC technology is manufactured by bending a sheet of reflective material, mostly aluminum or silvered acrylic, in the shape of a parabola. The collector focuses the solar radiation on a linear receiver. The operating temperature of the PTC starts from 50°C and goes to temperatures as high as 400°C. This temperature range is appropriate for the implementation in Rankine (steam turbine) cycles which leads to a relatively moderate efficiency and thermal performance. However, a higher operating temperature is necessary for integration of a CSP collector within gas turbine power plants.

The heat transfer fluid flowing through the receiver conveys the thermal energy acquired from solar radiation to operate a heat engine. Commonly, water or synthetic oil are employed as the heat transfer fluid in PTC collectors. However, oil is a more appropriate heat transfer fluid due to its low volatility and high boiling temperature [46]. There are two alternatives for PTC heat transfer fluid selection. The first approach is to utilize water to directly generate steam within the receiver. This method is referred to as direct steam generation (DSG). However, molten salt or synthetic oil can be used in the receiver to generate steam in the steam generator which is, in principle, a heat exchanger with oil as the hot fluid and water as the cold fluid. Nonetheless, Sunter and Carey presented a comparative analysis for different working fluids which can be utilized in PTC collectors [54]. Finally, a tracking device must be devised to accurately follow the path of the sun during daylight in order to further improve the collector and, accordingly, the power plant’s overall efficiency. Both east-west and north-south tracking orientation have been studied, and it is reported that east-west orientation has better performance [55].

2.3.2. Heliostat field collector.

The heliostat field collector (HFC), which is also referred to as the solar tower collector, is the most recent CSP technology emerging into the industry. This relatively expensive configuration has been implemented in quite a few power plants in the USA
and Spain. In particular, HFCs are employed in the 10 MW Solar One and Solar Two projects in California. In Spain, 11 MW PS10 and 20 MW PS20 are two power plants that selected HFC technology for their operation [46]. Basically, HFC technology features a heliostat field with a tower situated within the center of the field. The heliostat field contains several flat mirrors that direct the solar irradiation toward the top of the tower where the central receiver is located. A two axis tracking mechanism is required to permanently focus the solar radiation at the receiver’s surface. The surface area of each heliostat ranges from 50 to 150m² [46]. The implementation of lightly concave mirrors will increase heliostat field efficiency; however, manufacturing cost rises, too. HFCs have considerable potential to be utilized for mid load solar power plants (> 50 MWe) [56].

Furthermore, the central tower height can be varied from 75 to 150m [46]. Increasing the height of the tower will improve the heliostat field optical efficiency; therefore, a greater rate of thermal energy is delivered at the receiver’s surface. It should be noted that the HFC technology concentrates solar radiation on a single receiver resulting in lower heat loss and easier operation [46]. Because of their high capital cost, HFCs are usually associated with large scale power plants (greater than 10 MW) to benefit from the economy of scale [46]. Moreover, the high concentration ratio which leads to a high operating temperature enables solar tower collectors to be integrated within high temperature thermodynamic cycles such as gas turbine cycles. There are several control variables for designing a heliostat field such as the tower height, heliostat field area, receiver’s dimensions, and heliostat surface area. Consequently, based on the importance of the heliostat field performance and the available budget, the optimum design criteria must be chosen.

2.3.3 Linear Fresnel reflectors.

The linear Fresnel reflectors’ (LFR) principle of operation and arrangement is similar to PTC. Basically, the reflectors feature long arrays of flat mirrors that concentrate solar radiation toward a linear receiver. Moreover, the linear receiver is suspended on a tower with a height of 10 to 15 m along the collector arrays. It should be noted that a single axis tracking device is implemented to consistently direct sun radiation at the receiver’s surface. In addition, the mirrors are flat and elastic, resulting in a cheaper configuration than the PTC technology. The foremost deficiency of the LFR collector is its blocking factor that can be solved by either increasing the space
between the mirrors or the tower height. We should take into consideration that both solutions lead to a higher capital investment [46]. LFR technology is employed in PE1, a 1.4 MW solar power plant constructed by Novatec Biosel in Germany [46].

2.3.4. Parabolic dish collector.

The parabolic dish collector (PDC) focuses sun radiation at a single point. Consequently, a high concentration ratio can be achieved which leads to a notably high operating temperature. The collectors are circularly concaved reflectors in the shape of a parabola. A two axis tracking mechanism focuses solar radiation at the focal point of the collector where the receiver is located. PDC is mostly associated with a Stirling engine mounted at the focal point of each collector to reduce heat loss due to the heat transfer fluid transportation. However, gas turbines can also be coupled with the PDC in the form of Brayton or combined cycles. It is reported that PDC performance is better with the Stirling engine for temperatures lower than 950°C whereas gas turbines will perform better at higher temperatures [57]. It should be noted that the PDC collector is the most expensive CSP technology available for solar thermal power plants [46].

2.3.5. CSP technology selection.

By providing a brief description for different CSP technologies available within the market, it is time to select the most appropriate technology for implementation in this study based on their advantages and disadvantages. The main difference between the aforementioned CSP collectors lies in their approaches of concentrating solar irradiation. PTC and LFR technologies concentrate solar beams linearly, and a single axis tracking device enables them to follow the sun accurately. For two dimensional concentrators, i.e. PTC and LFR technologies, a maximum concentration ratio can be calculated by implementation of the second law of thermodynamics to the radiative heat exchange between the sun and the receiver as follows [26]:

\[
C_{max} = \frac{1}{\sin \theta_s}
\]

(2.10)

where \(\theta_s\) is the half angle subtended by the sun and its value is 0.27°; the maximum concentration ratio is 212.

However, HFC and PDC collectors are single point concentrators focusing sun radiation at a single point receiver. A two axis tracking mechanism is necessary to track the sun’s position. These concentrators are categorized a three dimensional
concentrators, and the maximum concentration ratio for a three dimensional concentrator is obtained by [26]:

$$C_{max} = \frac{1}{(\sin \theta_s)^2}$$  \hspace{1cm} (2.11)

With $\theta_s = 0.27^\circ$, a maximum concentration of 45,000 can be achieved in ideal circumstances. Even though these values can only be achieved in theory, a comparison between them clearly indicates the superiority of three dimensional concentrators over two dimensional for high temperature operations. Bearing in mind that this study’s main objective is to implement a CSP collector within gas turbine cycles, it is believed that HFC technology provides the best fit due to its high operating temperature range and low capital investment cost compared with PDC.

2.4. Basic Thermodynamic Cycles

In this section a basic configuration of thermodynamic cycles that will be studied in this research work is provided. The most commonly used thermodynamic cycles in the industry such as Rankine and Brayton cycle are thoroughly explained. Furthermore, two relatively new and recently proposed thermodynamic cycles, i.e. evaporative gas turbine and Maisotsenko gas turbine cycles, are introduced.

2.4.1. Rankine cycle.

The Rankine cycle basic configuration is presented in Figure 11. The cycle configuration consists of a steam turbine, a pump, a condenser, and a boiler. Saturated liquid water is pumped adiabatically to the boiler (1-2) where water is heated up and steam is generated in an isobaric process (2-3). Depending on the requirements, steam can be either saturated or superheated. Afterward, steam is expanded adiabatically in the turbine to produce mechanical work which is converted to electricity in the generator (3-4). Expanded steam is condensed to saturated liquid water in the condenser (4-1) to complete the closed cycle.
2.4.2. Brayton cycle.

The Brayton cycle basic configuration is presented in Figure 12. The cycle configuration consists of a gas turbine, an air compressor, and a combustion chamber. Ambient air is drawn into the compressor (1) in which it is compressed adiabatically (1-2). After compression, the air goes through the combustion chamber where heat is added to the air fuel mixture in an isobaric process (2-3). In the last step, the flue gases departing the combustion chamber are expanded adiabatically in the gas turbine to produce mechanical work (3-4), that is converted into electricity by the generator.
2.4.3 Evaporative gas turbine cycle.

Water or steam can be injected into gas turbine cycles for power augmentation purposes. The aforementioned water/steam addition can be performed during compression which is referred to as the wet compression process. In this approach, water cools down the air in the compressor such that the compression process is nearly isothermal.

Moreover, water may be injected after the compressor or in the combustion chamber. In principle, steam performs additional work in the turbine without increasing required work in the compressor [58]. Water injection has been implemented in industrial gas turbines since the 1960s which increases the mass flow rate and the specific heat of the gas expanding in the turbine [8]. In addition, it will reduce the amount of NO\textsubscript{x} formation in the combustion chamber. Additionally, humid air has a higher specific heat capacity and requires more thermal energy to be heated up [18].

By performing water/steam injection during expansion within the turbine, the turbine’s blades are more effectively cooled down than employing air for gas turbine blade cooling [8]. Basically, the amount of water injection depends on the NO\textsubscript{x} formation requirement; however, gas turbines are able to take up to 5% of the air mass flow rate passing through the compressor for the water/steam injection [8]. The main drawback of steam/water injected gas turbine cycles is the rise in the plant’s water consumption. In particular, water consumption can be a concern for regions with water limitations [58]. Additionally, steam injection in gas turbines complicates the water
condensation from the turbine exhaust gases under low vapor partial pressure. Therefore, condenser integration is not economically justified. On the other hand, steam/water injection prior to the combustion chamber leads to a higher vapor partial pressure; consequently, steam can be easily condensed by a natural heat sink [58]. In this method, water consumption can be reduced as the recovered water can be reused after being mechanically cleaned [58].

The basic configuration of an evaporative gas turbine cycle is presented in Figure 13. Ambient air is drawn into the compressor (1) where it is compressed adiabatically (1-2). After the compression, water will be injected into the air in the evaporator (2-3). In other words, the evaporator will cool down and humidify the incoming air stream. Afterward, humid air goes through the recuperator where it is preheated (3-4). Next, humid air enters the combustion chamber and is heated up in an isobaric process (4-5). Afterward, flue gases from the combustion chamber are expanded adiabatically in the gas turbine to produce mechanical work (5-6), which is converted into electricity in the generator. The exhaust gases from the turbine enter a heat exchanger (6-7) which is basically a recuperator. In the recuperator, a portion of the available waste heat in the exhaust gases are recovered by the humid air stream. Next, exhaust gases pass through the condenser to recycle the water for plant operation (7-8).

Figure 13: Evaporative gas turbine cycle configuration
2.4.4. Maisotsenko gas turbine cycle.

Idalex Inc. in cooperation with an R&D firm based in Arvada, Colorado, developed a new power generation cycle for gas turbines named Maisotsenko gas turbine cycle (MGTC) [15-17]. Air has been employed as the working fluid; the process combines evaporative cooling with thermodynamic processes of heat exchange to cool down the fluid to its dew point temperature [19].

In principle, MGTC is a humid air cycle and the addition of water results in reduction of NOx formation in the combustion chamber while power is augmented. In practice, the humidification constraint for the compressed air is the amount of heat released from cooling the flue gases to their dew point temperatures. This issue is automatically controlled in the MGTC configuration by employing a high pressure shell and tube air saturator [19]. Unfortunately, humid air cycles’ main obstacle in commercializing is their limitation in the air humidification process such that for further air humidification of the compressed air extra boilers are required. Therefore, additional equipment capital costs such as the saturation tower, boiler, and several heat exchangers will make humid air cycles economically unattractive [18].

MGTC developers claim that their developed cycle configuration does not suffer from part load due to its ability of adding moisture to the air [18]. The exergetic efficiency of the MGTC can reach to 53% which is an indication of its promising potential for future power plant generations [20]. The innovative part of the MGTC that separates it from other humid air turbine cycles is the implementation of the shell and tube air saturator. This saturator is illustrated in Figure 14.

The principle of the operation for the air saturator is to employ waste heat in the turbine exhaust gas to increase air wet bulb temperature. Compressed air divides into three streams in the lower tube air saturator. These streams are cooled down sensibly to their dew point temperatures by indirect evaporation of the water. Two of the streams come together while the third stream goes backward across the lower tube. The third stream humidity increases as it travels through the lower tube while it cools down the incoming air in the tubes. The combined stream enters the upper tube where it is heated up by exhaust gas from the turbine. Moreover, its moisture contents increase as it travels across the upper tube. At a specific point, the two moist air streams join each other and the hot humid air exits the air saturator. One of the major benefits of the air saturator is...
its ability to control moisture content of the air. By manipulating the amount of water in the saturator, the air humidity ratio and temperature can easily be regulated.

Figure 14: Air Saturator for Maisotsenko gas turbine cycle [18]

Figure 15 displays a basic configuration of the MGTC. Air is drawn into the compressor (1) where air is compressed in an adiabatic process (1-2). After compression, air enters the air saturator for heating and humidification. In principle, the air saturator utilizes the waste heat of the gas turbine flue gas to increase the air wet bulb temperature. Air is divided into three streams when it enters the lower section of the air saturator. All three air streams are cooled down sensibly to the air saturator inlet air dew point temperature by indirect evaporation of water (2-3). Two of the air streams are mixed together before arriving at the upper section of the air saturator inlet. The other air stream is fed back to the air saturator lower section while it is heated up and humidified to its maximum capacity (saturated air) (3-4). Cooled dry air (state 3) is directed to the upper section of the air saturator where it recovers a portion of the waste heat available in the gas turbine exhaust. Air is heated up and humidified in the upper section of the air saturator (3-5) before it is mixed with the other air stream directed from the lower section at a predetermined location (4-5-6). Humid air entering the combustion chamber is heated up in an isobaric process (6-7). After the combustion chamber, exhaust gas is expanded in an adiabatic process (7-8). Afterward, exhaust gas enters the air saturator upper section to be cooled down to the wet bulb temperature of
the incoming air (state 3) (8-9). One of the significant aspects of the air saturator is its ability to control the state of air entering the combustion chamber. By varying the upper section of the air saturator water injection rate, the humid air stream at state 6 can be either saturated or superheated.

![ Figure 15: Maisotsenko gas turbine cycle configuration](image)

### 2.5. Integration of Concentrated Solar Power Technologies with Different Thermodynamic Cycles

In this section, previous studies conducted on different power generation cycles and the integration of CSP technology within these power plants configuration are presented in detail.

#### 2.5.1. Rankine cycle.

In this section, a review of the previous studies conducted on the integration of CSP with Rankine cycle is presented. In 1975, the first study on the hybridization of the steam cycle was conducted by Zoschak and Wu [59] where solar energy was considered as a direct thermal input for an 800 MW power plant with the integration of HFC technology. Seven different methods of heat absorption were studied including air preheating, feed water heating, steam superheating, water evaporating, combined evaporation and superheating, steam reheating, and combined air preheating and feed water heating. A computer modeling of a hybrid power plant using HFC as the solar collector with the integration of thermal energy storage was developed by Griffith and
Brandt [60]. It was found that the hybrid power plant capital cost must not exceed 2.5 times the capital cost of a fossil fuel power plant to be economically competitive. In a study by Pai [61], integration of CSP collectors with a 210 MWe coal fired power plant was accomplished and 24.5% fuel saving was reported during high insolation periods. It was reported by You and Hu [62] that integration of CSP with a reheat regenerative Rankine cycle is a proper scheme for medium temperature hybrid power plants. A model was developed by Yan et al. [63] to evaluate the economics of CSP integration with different plant size. It was concluded that large scale power plants are more appropriate for hybridization with the same level of solar share.

2.5.2. Brayton cycle.

In this section, a review of previous studies on the integration of CSP collectors with gas turbine cycle is presented. A hybrid gas turbine power plant with pressurized volumetric air receiver was examined by Buck et al. [64] and it was confirmed that hybrid gas turbine configurations have high potential to become competitive in the near future. A hybrid gas turbine power plant with HFC technology implementation was investigated by Fisher et al. [65]. The study looked at two configurations, i.e. conventional fossil fuel power plants and hybrid power configuration with solar energy as a supplementary energy reserve for the operation. The results confirmed that the latter configuration is theoretically proven for future power plant generations. Volumetric pressurized receivers enable air to reach temperatures as high as 1000°C which enhances the hybrid gas turbine power plant capability to reach higher efficiency [66]. This phenomenon was addressed by Heller et al. [67] by testing the first prototype of a solar gas turbine. Furthermore, hybrid power plant configurations with different capacity were investigated by Schwarzbözl et al. [68]. Their results point to the economic superiority of the hybrid gas turbine for medium scale power plants (10-15 MWe) while the hybrid combined cycle is suitable for power stations with capacities greater than 15 MWe.

A thermo-economic analysis of a hybrid solar gas turbine power plant with HFC technology was conducted Spelling et al. [69]. TRNSYS was employed to simulate the power plant transient analysis. Three different configurations, i.e. hybrid gas turbine, hybrid gas turbine with thermal energy storage, and hybrid combined cycle were optimized by accomplishing a multi-objective optimization to minimize the levelized cost of electricity and specific CO₂ emission. It was concluded that hybridization leads
to a significant reduction in the plant’s CO₂ emission. At annual solar share below 20% combined cycle had the lowest cost; however, only the hybrid configuration with thermal energy storage is capable of achieving solar share of 85-90% with a notably high levelized cost of electricity.

2.5.3. Conventional combined cycle.

In this section, previous investigations conducted on integration of CSP technologies with conventional combined cycle (CCC) are given. Hybridization of combined cycle is a relatively new concept and proposed by Luz solar international in the 1990s [66]. It was suggested to integrate PTC collectors with CCC power plants [70]. This new concept was first studied by Allani [71] by integrating low temperature PTC technology in the bottoming Rankine cycle. Solar preheating in the topping Brayton cycle was studied by Bohn et al. [72]. The study indicates that higher conversion efficiency can be achieved; even though, the solar share in electricity production is limited by integrating PTC collectors in the topping gas turbine cycles. Employment of an HFC technology in the topping gas turbine cycle was studied by Price et al. [73]. An economic comparison between solar-only and hybrid power plants for a combined cycle configuration with HFC implementation was performed by Kolb [74]. The acquired result is an indication of the economic superiority of the hybrid power plants in the current situation. Kribus et al. [75] presented the economic and performance advantages of integrating a high temperature HFC collector over other solar thermal concepts. Moreover, this configuration might be economically competitive with non-hybrid power plant configuration. A thermodynamic optimization model for a hybridized combine cycle power plant was developed by Kane et al. [76]. The authors concluded that hybrid power plants are competitive with non-hybrid configurations. Technical and economic aspects of PTC and HFC integration in a combined cycle in Egypt was conducted by Horn et al. [77]. They concluded that integration of the air tower (HFC) is economically and environmentally beneficial for hybrid power plants’ implementation in the Egypt. Addition of thermal energy storage to a hybrid combined cycle power plant with HFC implementation was studied by Bonadies et al. [78]. It is reported that the operating period of the investigated power plant can reach to 17 hours per day.
2.5.4. Unconventional cycle configurations.

In this section, a number of investigations on different aspects of CSP integration are presented. The following investigations do not necessarily belong to any of the previously discussed configurations. Technical and economic effects of the integration of CSP technologies on power generation cost in Cyprus was investigated by Poullikkas et al. [79]. It was concluded that a 50 MWe solar only power plant with PTC and 19 hours of thermal storage capability provides the lowest cost for future references. Integration of CSP with combined Rankine/Kalina cycle was suggested by Mittelman and Epstein [80]. The plant has a capacity of 50 MWe and implements PTC technology for its operation. It was concluded that the power block has a lower installation cost due to its smaller condensing system. Moreover, it can produce electricity during low insolation periods. Furthermore, a 4-11% reduction in electricity production cost was predicted. Integration of CSP within a triple cycle power plant was proposed by Kribus [81]. This power cycle is composed of magneto-hydro-dynamic as the topping cycle and an intermediate Brayton cycle with a Rankine bottoming cycle. Taking into consideration that HFC was selected as the solar collector because of the required high operating temperature for the magneto-hydro-dynamic (2000-2500°C), the result points to the promising potential of the triple power block. However, high operating temperature is a significant deterrent in commercialization of the proposed configuration. Additionally, the plant is economically unfavorable.

Three different plant configurations including CCC, ABC, and hybrid ABC with parabolic trough integration in the bottoming cycle were studied by Khaldi [12]. Energy efficiency of the hybrid ABC was reported to be 31.87% whereas simple ABC efficiency was 43.63%. CCC configuration achieved the highest efficiency of 46.78%. However, it was more interesting to conduct economic and environmental analyses of the three investigated configurations to better illustrate the advantages of the hybrid ABC. A water free hybrid ABC was studied by Sandoz et al. [82] where multi-objective optimization of the plant was performed to achieve the best combination of economic and environmental performances. A Heliostat field collector was considered for air preheating in the topping cycle. ABC’s performance was evaluated against a hybrid simple gas turbine cycle. Results indicate an increase in cycle efficiency by ABC implementation. Nevertheless, it could have been more appealing if the authors had
presented a comparative analysis between hybrid ABC and CCC power plants for small-scale power generation.

2.6. Proposed Cycles

In a simple gas turbine cycle, substantial waste heat is available in the exhaust gases which can be recovered and further exploited. One alternative is to employ the available waste heat in the gas turbine exhaust gases as a source of process heat [6]. There are several different industries that rely on process heat including oil production and refining, steel making, food processing, and textile industries [6]. Steam is commonly used as the heat transfer fluid in these industries. Another option is to devise a bottoming cycle with significantly lower operating temperature to generate additional power and enhance the plant’s overall efficiency. It is important to note that the most important factor in selecting an appropriate bottoming cycle is its operating temperature in order to be capable of achieving satisfactory waste heat recovery effectiveness. Accordingly, power generation cycles’ operating temperatures are depicted in Figure 16. The most popular and widely used bottoming cycle is the steam (Rankine) turbine cycle. It is a well-known fact that CCC, i.e. topping gas turbine and bottoming steam turbine cycles, is the most thermodynamically efficient combined plant configuration [7]. Nonetheless, CCC power plants are not the most economically justified configuration for small-scale power plants less than 50 MWe [8]. For capacities less than 50 MWe, the complication and high expenses due to the HRSG and steam turbine argue in favor of seeking alternatives [7].

An alternative to CCC configurations is to employ another gas turbine cycle for heat recovery purposes. This combined cycle configuration, which is referred to as the air bottoming cycle (ABC), was patented by W. Farrell in 1988 [9]. ABC has several advantages over CCC power plants such as shorter installation time, shorter start up time, lower capital investment, lower operating and maintenance cost, more compact size, and simpler operation [10-12]. Additionally, ABC’s low water consumption enables it to be implemented in regions with water shortage problems [13]. Keeping in mind that regions with high solar radiation are commonly suffering from water scarcity [3], ABC can be an interesting option for hybrid power plant installation.
There are several studies investigating different aspects of ABC power plants. Korobitsyn [13] investigated the heat and power cogeneration ABC for industrial applications including milk, whey powder, bakery, and glass industries. Efficiency enhancement of 18% due to the ABC integration was reported by Hirs et al. [83]. The intercooling impact on ABC efficiency was studied by Czaja et al. [11]. Economic analysis of ABC accomplished by Chmielniak et al. [10] indicates the economic advantages of ABC over CCC power plants. Furthermore, the cost analysis for a 22 MWe ABC power plant was conducted by Bolland et al. [84] who found that ABC is an economical alternative. A thermo-economic analysis of ABC power plants with and without an intercooler in the bottoming cycle was carried out by Saghaafifar and Poullikkas [7]. The results point to the economic advantages of intercooler integration in the ABC bottoming cycle. In another study by Saghaafifar and Poullikkas [85], a comparative analysis between different approaches of power augmentation in ABC configurations was accomplished. Steam injection in the combustion chamber was reported to be the most economically justified approach. There have been several other studies on ABC configurations [14, 86-89]. In particular, we are interested in a study by Ghazikhani et al. [14] in which steam and water injection in the bottoming cycle air were proposed and investigated. Water/steam injection in the ABC configurations improved the bottoming cycle heat recovery and thermal efficiency. It was reported that steam and water injection enhanced the plant’s thermal efficiency from 49.83% to 52.43% and 54.63%, respectively. It is important to note that the results presented by

Figure 16: different power generation cycles’ operating temperature [13, 90]

60
Ghazikhani et al. [14] indicated that water injection is a more effective approach in ABC power plants.

Furthermore, the Maisotsenko gas turbine cycle (MGTC) is a recently proposed humid air turbine cycle [15-17]. In principle, MGTC is an evaporative gas turbine cycle which utilizes water addition for reduction in NO\textsubscript{x} formation and power augmentation [18, 19]. Alsharif et al. [20] conducted energy and exergy analyses for MGTC power plant configuration. In a study by the authors [21], a detailed analysis of MGTC power plant configuration with a comprehensive air saturator model was carried out. Mainly, the results indicated that MGTC can be competitive with humid air turbine cycles due to its high thermal efficiency. Consequently, it was decided to combine the proposed approaches presented by Ghazikhani et al. [14] and MGTC configuration to present the Maisotsenko bottoming cycle (MBC) [22]. Furthermore, in a study by the authors [22], it was determined that the MBC power plant configuration is thermodynamically superior over ABC power plants. Integration of an MBC in an already existing simple gas turbine cycle improved the cycle thermal efficiency to 44.9% whereas an ABC integration enhanced the plant’s thermal efficiency to 43.3%. Furthermore, the difference in the presented thermal efficiencies can lead to more than 2600 tons of natural gas fuel savings per year.

To sum up, it was decided to investigate the hybridization of different combined cycle configurations with capacities less than 50 MWe. Power plant hybridization was accomplished by employing a solar tower collector (Heliostat field collector). Three rather unconventional bottoming cycle configurations were chosen including gas turbine (air bottoming cycle), water injected gas turbine (humid air bottoming cycle), and the Maisotsenko cycle (Maisotsenko bottoming cycle). These three configurations along with the conventional combined cycle power plant (steam bottoming cycle) were optimized by conducting thermo-economic and transient analyses in MATLAB to identify the most economically justified plant configuration for the United Arab Emirates.

2.6.1. Steam bottoming cycle.

The steam bottoming cycle (SBC) or CCC consists of a gas turbine topping cycle and a steam turbine bottoming cycle. The goal of this study is to determine the most economically justified power plant configuration for a 50 MWe hybrid power
plant in Abu Dhabi. All other proposed cycles’ performances will be compared to SBC as the reference configuration.

Figure 17 illustrates the SBC configuration with integration of HFC technology in the topping gas turbine cycle. The conventional combined cycle consists of topping gas turbine and bottoming steam turbine cycles. Ambient air is drawn into the topping cycle compressor (1) where it is compressed adiabatically (1-2). After compression, air enters the central tower where it is preheated with the available solar flux at the receiver (2-3). Next, air enters the combustion chamber where the air fuel mixture is further heated in an isobaric process (3-4). In the last step, the flue gases departing the combustion chamber are expanded adiabatically in the gas turbine to produce mechanical work (4-5) which is converted into electricity by the generator. Exhaust gases from the turbine go through the heat recovery steam generator (HRSG) where they will be cooled down by generating steam in the bottoming cycle (5-6). Afterward, cooled exhaust gasses are released to the atmosphere. In the bottoming cycle, saturated liquid water is pumped adiabatically to the HRSG (7-8) where water is heated up and steam is generated in an isobaric process (8-9). Depending on the requirements, steam can be either saturated or superheated. Afterward, steam is expanded adiabatically in the steam turbine (9-10). Expanded steam is condensed to the saturated liquid water in the condenser (10-7) to complete the bottoming cycle.
Figure 17: Conventional combined cycle configuration with heliostat field collector integration within the topping gas turbine cycle

2.6.2. Air bottoming cycle.

The air bottoming cycle (ABC) is one of the proposed thermodynamic cycles which will be studied to determine its performance for a hybrid 50 MWe power plant in Abu Dhabi. ABC configuration consists of two gas turbine cycles as the topping and
bottoming cycles. The major advantage of the ABC over SBC is the absence of HRSG, condenser, and water treatment facilities in the cycle. Consequently, water consumption will be diminished to an extremely low value. This configuration will be suitable for deserts where water availability is an issue. Moreover, low capital, operation and maintenance cost, short construction and startup time, compact size, and low pollution are the other benefits of integrating ABC as opposed to SBC [10, 12]. ABC was patented by W. Farrell of the General Electric Company in 1988 [9]. Wicks developed the theory of ABC to a realistic concept by implementing the ideal fuel-burning engine model in 1991 [91].

Figure 18 displays the hybrid ABC cycle configuration with integration of HFC technology for air preheating in the topping gas turbine cycle. The air bottoming cycle consists of a topping gas turbine and a bottoming air turbine cycle. Ambient air is drawn into the topping cycle compressor (1) where it is compressed adiabatically (1-2). After compression, air enters the central tower where it is preheated with the available solar flux at the receiver (2-3). Next, air goes through the combustion chamber and heat is added to the air fuel mixture in an isobaric process (3-4). In the next stage, flue gases departing the combustion chamber are expanded adiabatically in the gas turbine for power generation. Exhaust gases from the topping gas turbine go through the air heat exchanger where a portion of the available waste heat is recovered by the bottoming cycle air flow (5-6). In the bottoming cycle, ambient air is drawn into the bottoming cycle compressor (7) and it is compressed adiabatically (7-8). Afterward, compressed air goes through the air heat exchanger where it is heated by waste heat available in the topping cycle turbine’s exhaust gases (8-9). Heated air is expanded in the bottoming cycle air turbine producing additional power and enhancing the plant’s overall efficiency (9-10).

2.6.3. Humid air bottoming cycle.

In order to increase cycle efficiency, an evaporator can be employed in the bottoming cycle in order to further cool down the topping cycle exhaust gases. In other words, the evaporator will act as an aftercooler which cools down the compressed air before it enters the air heat exchanger.
Figure 18: Air bottoming cycle configuration with heliostat field collector integration within the topping gas turbine cycle

Figure 19 displays the humid air bottoming cycle (HABC) configuration with integration of HFC as the solar collector in the topping cycle. Ambient air is drawn into the topping cycle compressor (1) where it is compressed adiabatically (1-2). After compression, air enters the HFC and is preheated with solar energy in the receiver (2-3). Next, air goes through the combustion chamber and heat is added to the air fuel mixture in an isobaric process (3-4). In the last step, flue gases departing the combustion chamber are expanded adiabatically in the gas turbine to produce mechanical work (4-5). Exhaust gases from the turbine go through the AHX where they will be cooled down by the bottoming cycle air flow (5-6). In the HABC, ambient air is drawn into the
bottoming cycle compressor (7) and is compressed adiabatically (8-9). After compression, it goes to the evaporator where water is added to the air (8-9). As a result, the compressed air is cooled down and humidified simultaneously. Humid air goes through the AHX and recovers a portion of waste heat available in the topping cycle turbine’s exhaust gas (9-10). Hot and humid air is expanded in the bottoming cycle turbine to produce extra electricity which results in an increase in the total cycle efficiency with respect to a simple gas turbine cycle (10-11).

![Diagram of Humid air bottoming cycle configuration with heliostat field collector integration within the topping gas turbine cycle](image)

Figure 19: Humid air bottoming cycle configuration with heliostat field collector integration within the topping gas turbine cycle

### 2.6.4 Maisotsenko bottoming cycle.

Another possibility for the bottoming cycle is the integration of MGTC as the bottoming cycle. The main difference between the Maisotsenko bottoming cycle (MBC) and HABC is in their waste heat recovery approaches. MBC utilizes an air saturator which replaces both evaporator and AHX within the HABC configuration.
This reduction in the number of pieces of equipment may result in a reduction of the capital investment cost and improve the plant’s thermal performance.

Figure 20 demonstrates the MBC configuration with integration of HFC as the solar collector in the topping cycle. Ambient air is drawn into the topping cycle compressor (1) where it is compressed adiabatically (1-2). After compression, air enters the central tower where it is preheated with the available solar flux at the receiver (2-3). Next, air goes through the combustion chamber and heat is added to the air fuel mixture in an isobaric process (3-4). In the last step, flue gases departing the combustion chamber are expanded adiabatically in the gas turbine to produce mechanical work (4-5) which is converted into electricity in the generator. Exhaust gases from the turbine go through an air saturator where they will be cooled down by the bottoming cycle air flow (5-6). Moreover, it will humidify the bottoming cycle air flow. In the bottoming cycle, ambient air is compressed adiabatically (7-8). Next, the compressed air enters the air saturator where it is heated and humidified by the exhaust gases from the topping cycle gas turbine (8-12).

In principle, the air saturator utilizes the waste heat available in the topping cycle gas turbine exhaust gases to increase the air wet bulb temperature. In this basic operation, the compressed air is divided into three streams in the lower section of the air saturator. These streams are cooled down sensibly to their dew point temperatures by indirect evaporation of water (8-9). Two of these streams mix together while the third stream is fed backward to the lower section of the air saturator (9-10). The third stream's humidity increases as it travels through the lower section of the air saturator while it cools down the incoming air streams. The combined stream enters the upper section of the air saturator where it is heated by the exhaust gases supplied from the topping cycle turbine (9-11). Moreover, its moisture content increases as it travels across the upper section of the air saturator. At a specific point, the humid air stream at state 10 is mixed with the other humid air stream at state 11 before leaving the air saturator (12). In this operation, one of the major benefits of the air saturator is its ability to control the moisture content of the supplied air to the second air turbine (expander). By controlling the required amount of water in the air saturator, the air humidity ratio and its temperature can be adjusted. Then, the humid air is expanded adiabatically in the turbine to produce mechanical work (12-13), which is converted into electricity in the second generator.
Figure 20: Maisotsenko bottoming cycle configuration with heliostat field collector integration within the topping gas turbine cycle
Chapter 3: Mathematical Formulation

In this section, detailed thermo-economic modeling developed for the analysis of the proposed technologies is presented. First the thermodynamic modeling is discussed which is followed by the presentation of the economic model. The developed formulation is implemented in a MATLAB code for the simulation requirements of this research work.

3.1. Thermodynamic Model

Air and exhaust gases heat capacities and entropies are evaluated based on NASA polynomial curve fits [92-95] as follows:

\[ C_p = R [a_1 + a_2 T + a_3 T^2 + a_4 T^3 + a_5 T^4] \]  
\[ s^0 = R \left[ a_1 \ln(T) + a_2 T + \frac{a_3}{2} T^2 + \frac{a_4}{3} T^3 + \frac{a_5}{4} T^4 + a_7 \right] \]

where \( C_p \) is the specific heat capacity in kJ/kgK, \( R \) is the gas constant in kJ/kgK, \( T \) is temperature in K, \( s^0 \) is the temperature dependent specific entropy in kJ/kgK, and \( a_1 \cdots a_n \) are dimensionless NASA polynomial curve fit coefficients. Developed models are implemented in MATLAB for simulation requirement of this research work. All thermodynamic assumptions, constraints, and design parameters employed in this research work are tabulated in Table 5.

An ideal gas mixture model is selected for moist air thermodynamic property assessment based on Dalton's Partial Pressure Law [96]. For humidified gas turbine analysis, an accurate humid air thermo-physical properties evaluation is required in order to develop reliable mathematical models. In particular, high pressure, low temperature, and high humidity level deviate air-water mixture from the ideal gas model. Nonetheless, several studies on humidified gas turbine cycles relied on the ideal gas mixture model to provide sufficient accuracy for their analyses [85, 97-101]. Comparison between ideal gas and real gas models indicate that the real gas model predicts higher saturation humidity for a specific temperature [102]. It is important to note that the real gas model of water is more volatile as compared with its ideal gas model. Nevertheless, the overall cycle efficiency is not affected considerably by different air-water mixture models [102]. Thermodynamic properties of moist air and humid exhaust gases can be calculated by:
\[ c_{p,m} = \sum_{i} mf_i c_{p,i} \]  \hspace{1cm} (3.3)

\[ s_{m}^0 = \sum_{i} mf_i s_{i}^0 \]  \hspace{1cm} (3.4)

where \( c_{p,m} \) is the mixture specific heat capacity in kJ/kgK, \( s_{m}^0 \) is the mixture temperature dependent specific entropy in kJ/kgK, and \( mf_i \) is the mass fraction of \( i \)th component in ideal gas mixture.

Table 5: Thermodynamic assumption, constraints, and design parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Assigned value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel lower heating value (kJ/kg)</td>
<td>50142</td>
</tr>
<tr>
<td>Reference temperature (K)</td>
<td>298</td>
</tr>
<tr>
<td>Combustion chamber efficiency</td>
<td>98%</td>
</tr>
<tr>
<td>Combustion chamber pressure drop</td>
<td>4%</td>
</tr>
<tr>
<td>Net power output (MWe)</td>
<td>50</td>
</tr>
<tr>
<td>Compressor isentropic efficiency</td>
<td>85%</td>
</tr>
<tr>
<td>Compressor mechanical efficiency</td>
<td>99%</td>
</tr>
<tr>
<td>Turbine isentropic efficiency</td>
<td>85%</td>
</tr>
<tr>
<td>Turbine mechanical efficiency</td>
<td>99%</td>
</tr>
<tr>
<td>Generator electrical efficiency</td>
<td>99%</td>
</tr>
<tr>
<td>Generator mechanical efficiency</td>
<td>98%</td>
</tr>
<tr>
<td>Heat exchanger pressure drop</td>
<td>4%</td>
</tr>
<tr>
<td>Heat exchanger pinch temperature (K)</td>
<td>9</td>
</tr>
<tr>
<td>Pump Isentropic efficiency</td>
<td>90%</td>
</tr>
<tr>
<td>Solar Tower maximum outlet temperature (K)</td>
<td>1223</td>
</tr>
<tr>
<td>HRSG pinch temperature (K)</td>
<td>9</td>
</tr>
<tr>
<td>HRSG approach temperature (K)</td>
<td>9</td>
</tr>
<tr>
<td>HRSG pressure drop (both sides)</td>
<td>4%</td>
</tr>
<tr>
<td>Condenser pressure (kPa)</td>
<td>10</td>
</tr>
<tr>
<td>Air saturator pressure drop (both sides)</td>
<td>4%</td>
</tr>
<tr>
<td>Input water temperature (K)</td>
<td>298</td>
</tr>
<tr>
<td>Evaporator pressure drop</td>
<td>4%</td>
</tr>
</tbody>
</table>

3.1.1. Compressor.

Proposed configurations include two compressors for their topping and bottoming cycles. Both compressors are identical with air as their working fluid. Their pressure ratios depend on the required topping and bottoming cycle compression ratios. The compressor’s outlet temperature and specific work input are calculated as follows:

\[ s_{c,o}^0 = s_{c,i}^0 + R \ln r_c \]  \hspace{1cm} (3.5)

\[ s_{c,o}^0 \rightarrow T_{sc,o} \]  \hspace{1cm} (3.6)
\[ T_{c,o} = T_{c,i} + \frac{T_{sc,o} - T_{c,i}}{\eta_c} \] (3.7)

\[ w_c = \frac{C_{p,avg}(T_{c,o} - T_{c,i})}{\eta_M} \] (3.8)

where \( s_{c,i}^0 \) and \( s_{c,o}^0 \) are the corresponding entropies of the compressor inlet and isentropic outlet in kJ/kg.K, \( T_{c,i} \) is the air temperature at compressor inlet in K, \( T_{sc,o} \) and \( T_{c,o} \) are isentropic and actual air temperature at the compressor outlet in K, \( C_{p,avg} \) is the specific heat capacity for average temperatures across the compressor in kJ/kg.K, \( r_c \) is the compressor pressure ratio, \( \eta_M \) is the turbomachinery mechanical efficiency, \( \eta_c \) is the compressor isentropic efficiency and \( w_c \) is the compressor required specific work input in kJ/kg.

**3.1.2. Turbine.**

Each configuration consists of two turbines with different working fluids and pressure ratios including gas, air, and humid air turbines. Consequently, full regard must be given to different working fluid’s specific properties calculation. A turbine’s outlet temperature and specific work output are determined by:

\[ s_{t,o}^0 = s_{t,i}^0 + R \ln r_t \] (3.9)

\[ s_{t,o}^0 \rightarrow T_{st,o} \] (3.10)

\[ T_{t,o} = T_{t,i} - \eta_t (T_{t,i} - T_{st,o}) \] (3.11)

\[ w_t = \frac{C_{p,avg}(T_{t,i} - T_{t,o})}{\eta_M} \] (3.12)

where \( s_{t,i}^0 \) and \( s_{t,o}^0 \) are corresponding entropies of turbine inlet and isentropic outlet in kJ/kg.K, \( T_{t,i} \) is the temperature of working fluid at turbine inlet in K, \( T_{st,o} \) and \( T_{t,o} \) are isentropic and actual air temperature at turbine outlet in K, \( r_t \) is the turbine pressure ratio, \( \eta_t \) is the turbine’s isentropic efficiency and \( w_t \) is the turbine work output in kJ/kg.

**3.1.3. Combustion chamber.**

Methane is taken as the primary fuel for combustion purposes in the combustion chamber. Complete combustion is considered for the analysis of this research with
relevant specific heat capacities and entropies of exhaust gases calculated by following molar contributions:

\[ CH_4 + 2(O_2 + 3.76N_2) \rightarrow CO_2 + 2H_2O + 7.52N_2 \]  \hspace{1cm} (3.13)

The required amount of fuel addition for achieving the predetermined GTIT and exhaust gases mass flow rate is calculated by:

\[ m_f = \frac{\dot{m}_a c_{pg,avg} (T_{o,cc} - T_{ref}) - \dot{m}_a c_{pa,avg} (T_{i,cc} - T_{ref})}{\eta_{cc} LHV - c_{pg,avg} (T_{o,cc} - T_{ref})} \]  \hspace{1cm} (3.14)

\[ \dot{m}_g = \dot{m}_a + \dot{m}_f \]  \hspace{1cm} (3.15)

where \( \dot{m}_f \), \( \dot{m}_g \), and \( \dot{m}_a \) are fuel, exhaust gases, and air mass flow rates in kg/s, \( T_{i,cc} \) and \( T_{o,cc} \) are temperatures at combustion chamber inlet and outlet in K, \( c_{pa,avg} \) and \( c_{pg,avg} \) are average specific heat capacities of air and gas, \( \eta_{cc} \) is the combustion chamber efficiency, \( LHV \) is the fuel lower heating value in kJ/kg, and \( T_{ref} \) is reference temperature state associated with the fuel \( LHV \) in K.

**3.1.4. Air heat exchanger.**

ABC and HABC configurations consist of an air to air heat exchanger which is responsible for heat recovery from the topping cycle gas turbine exhaust gases for the bottoming cycle air and humid air heating requirement. The most important factors in heat exchanger design are high effectiveness and low pressure drop. On the other hand, low heat transfer coefficient, high temperature gases, significant pressure difference, and low density of gases involved in this heat exchanger complicate its design procedure [82].

Pinch point analysis is selected for modeling purposes of the air heat exchanger in this research work. Pinch temperature difference is located where the minimum temperature difference between the hot and cold fluids is achieved. For a counter flow heat exchanger, the location of the pinch temperature difference depends on the fluids’ mass flow rates as well as their specific heat capacities. If a hot fluid’s heat capacity \( (\dot{m}c_p) \) is greater than the cold fluid’s heat capacity, pinch temperature difference occurs at the hot stream entry and cold stream exit. Otherwise, pinch temperature difference’s location shifts to the other end of the heat exchanger. In the case of equal heat capacities,
pinch temperature difference can be attained on both ends of the heat exchanger. Based on the presented explanation of pinch point analysis, the air heat exchanger is modeled as follows:

\[ T_{o,L} = T_{i,H} \pm \Delta T_{\text{pinch}} \]  
\[ T_{o,H} = T_{i,H} - \frac{\dot{m}c_{p,L}(T_{o,L} - T_{i,L})}{\dot{m}c_{p,H}} \]

where \( T_{o,L} \) and \( T_{o,H} \) are the temperatures at air heat exchanger inlet and outlet associated with the fluid having lower heat capacity in K, \( T_{i,H} \), and \( T_{o,H} \) are the temperatures at air heat exchanger inlet and outlet associated with having higher heat capacity in K, \( \dot{m}c_{p,H} \) and \( \dot{m}c_{p,L} \) are higher heat capacity and lower heat capacity for fluids involved in the air heat exchanger respectively in kJ/K, and \( \Delta T_{\text{pinch}} \) is the pinch temperature difference for the air heat exchanger in K.

### 3.1.5. Heat recovery steam generator.

The integrated HRSG within the SBC configuration consists of three separated sections of economizer, evaporator, and superheater. In SBC, steam must be superheated to enhance the bottoming cycle efficiency. Based on the condition of the steam departing the HRSG, its outlet temperature is either selected to be a predetermined superheated temperature or its saturation temperature based on design pressure. Economizer outlet air and steam temperatures are calculated by:

\[ T_{m,g} = T_{ev,s} + \Delta T_{\text{pinch}} \]  
\[ T_{m,s} = T_{ev,g} - \Delta T_{\text{approach}} \]

where \( T_{m,g} \) and \( T_{m,s} \) are the exhaust gases and steam temperatures at the economizer outlet (the evaporator inlet) in K, \( \Delta T_{\text{pinch}} \) and \( \Delta T_{\text{approach}} \) are the HRSG pinch and approach temperature differences in K, and \( T_{ev,s} \) and \( T_{ev,g} \) are steam and exhaust gases temperatures at the HRSG outlet in K. In the case of superheated steam, steam temperature leaving the HRSG and air temperature entering the HRSG replace \( T_{ev,s} \) and \( T_{ev,g} \) in the mentioned equations, respectively. The amount of generated steam (saturated or superheated) is determined by the following:
\[ T_{ec,g} = T_{m,g} - \frac{\dot{m}_s (h_{m,s} - h_{ec,s})}{\dot{m}_g c_{pg}} \]  
(3.20)

\[ \dot{m}_s = \frac{\dot{m}_g c_{pg} (T_{ev,g} - T_{m,g})}{(h_{ev,s} - h_{m,s})} \]  
(3.21)

where \( \dot{m}_g \) is the generated steam mass flow rate in kg/s, \( T_{ev,g} \) and \( T_{ec,g} \) are the exhaust gases’ temperature arriving at and departing from the HRSG in K, \( h_{ev,s}, h_{m,s}, \) and \( h_{ec,s} \) are the generated steam specific enthalpy at the HRSG evaporator, midpoint (between the economizer and the evaporator), and economizer in kJ/kg, respectively. Keeping in mind the case of superheated steam, all related temperatures and specific enthalpies of evaporator are replaced by their corresponding superheated values. Afterward, exhaust gas and steam temperatures at the evaporator outlet can be determined.

### 3.1.6. Pump.

Including specific work input of the pump does not have a considerable impact on the plant’s efficiency; nevertheless, its inclusion boosts the accuracy of the acquired model. The pump specific work input is calculated by:

\[ w_p = \frac{v_{i,p} (P_{o,p} - P_{i,p})}{\eta_p} \]  
(3.22)

where \( P_{i,p} \) and \( P_{o,p} \) are the water pressures at inlet and outlet of the pump in kPa, \( v_{i,p} \) is the specific volume of water arriving at the pump inlet in m\(^3\)/kg, \( \eta_p \) is the pump isentropic efficiency, and \( w_p \) is the pump specific work input in kJ/kg.

### 3.1.7. Condenser.

The condenser is another component of the SBC configuration. The condenser’s formulation is accomplished by taking into account that saturated liquid water is achieved at its outlet. Consequently, water temperature at the condenser outlet is determined based on its pressure from the saturated steam table such as:

\[ P_{con} \xrightarrow{saturated \ liquid} T_{con} \]  
(3.23)

where \( P_{con} \) and \( T_{con} \) are the water pressure within the condenser in kPa and water temperature at the condenser outlet in K, respectively.
3.1.8. Air saturator.

To increase the accuracy of the suggested air saturator model, exclusive mathematical formulations are developed for upper and lower sections of the air saturator. The developed model was published in two peer-reviewed journal papers [21, 22] and its correctness was evaluated by other scholars. The model was developed based on the assumption of fully water evaporation in the air saturator caused by the compressed air. Thus, one may argue that rather than water evaporation with an excess of water which is considered as a significant feature of evaporative gas turbine cycles, MGTC is reduced to a water injection cycle with full water evaporation. Nonetheless, our assumption does not affect either the operation of MGTC nor the air saturator performance. To obtain superheated air at the air saturator outlet, the amount of water injected in the upper section of the air saturator must be controlled [19]. Therefore, the air saturator upper section must be formulated as a fully evaporated water injection heat and mass exchanger. On the other hand, the main task of the lower section of the air saturator is to cool down the compressed air. Consequently, the air outlet must be fully saturated at state 10 with excess water evaporation. For simplicity, it is assumed that the amount of water added in the lower section of the air saturator is exactly equal to the required amount of water to obtain fully saturated air at state 10. Then, air entering the lower section of the air saturator is cooled down sensibly to the air saturator inlet air dew point temperature [19]. Air is then used as the working fluid in which it combines evaporative cooling with the thermodynamic process of exchanging heat in order to cool down the fluid to its dew point temperature [19]. Consequently, the temperature of the air at state 9 can be determined by:

\[ T_9 = T_8 - E_{dew}[T_8 - T_{dew,8}] \]

(3.24)

where \( E_{dew} \) is the dew point effectiveness of the air saturator. Based on the results presented in Wicker [18], \( E_{dew} \) is chosen to be 0.8 in this research work. \( T_{dew,8} \) is the dew point temperature of the air at state 8 in K. All temperature indices are based on the indices presented in Figure 20. Therefore, \( T_8 \) and \( T_9 \) are air temperatures at states 8 and 9 in K, respectively.

Since air is saturated at state 10, it is necessary to calculate the mole fraction of water in humid air mixture by:
\[
\dot{n}_{wi} = \frac{\dot{m}_{wi} + \dot{m}_{winL}}{M_w} \\
\dot{n}_{da} = \frac{\dot{m}_{da}}{M_{da}} \\
x_{wi} = \frac{\dot{n}_{wi}}{\dot{n}_{da}}
\]

where \(\dot{m}_{wi}\) and \(\dot{m}_{winL}\) are the mass flow rates of water at state \(i\) and the amount of water addition in the lower section of the air saturator in kg/s, \(\dot{m}_{da}\) is the mass flow rate of dry air at state \(i\) in kg/s, \(M_w\) and \(M_{da}\) are the molecular weight of the water and humid air respectively in kg/kmol, \(\dot{n}_{wi}\) and \(\dot{n}_{da}\) are the number of moles for water and humid air at state \(i\) in kmol, and \(x_{wi}\) is the mole fraction of steam at state \(i\).

Consequently, the saturation pressure and temperature can be calculated by:

\[
P_{sat_i} = x_{wi}P_i
\]

\[
P_{sat_i} \rightarrow \text{Steam Table} \rightarrow T_{sat_i}
\]

where \(P_{sat_i}\) and \(P_i\) are the partial pressure of water and total pressure of the mixture at state \(i\) in kPa. By assessing saturation pressure, the temperature of the mixture at state 10 is determined from the saturated steam table.

Energy balance equation for the lower section of the air saturator will be:

\[
(\dot{m}_{w8}h_{w8} + \dot{m}_{da8}h_{da8}) + \dot{m}_{winL}h_{win} = (\dot{m}_{w9}h_{w9} + \dot{m}_{da9}h_{da9}) + (\dot{m}_{w10}h_{w10} + \dot{m}_{da10}h_{da10})
\]

where \(h_{win}, h_{wi}\) and \(h_{da}\) are the specific enthalpy of inlet water, water, and dry air at state \(i\) in kJ/kg, respectively.

The amount of water injection in the lower section of the air saturator has to satisfy Equation 3.30. Consequently, an initial guess is assumed for \(\dot{m}_{winL}\). Afterward, saturated air temperature at state 10 is determined and Equation 3.30 is employed to find a new value for \(\dot{m}_{winL}\). These procedures are repeated until the maximum amount of water injection in the lower section of the air saturator is determined.
An identical approach is employed for the analysis of the upper section of the air saturator. The exhaust gas from the topping cycle turbine is cooled down sensibly to the wet bulb temperature of the air at state 9 [24].

\[ T_6 = T_4 - E_{wbt} [T_4 - T_{wbt9}] \]  

(3.31)

where \( T_{wbt9} \) is the wet bulb temperature of the air at state 9, and \( E_{wbt} \) is the wet bulb temperature effectiveness of the air saturator. Its value \( (E_{wbt}) \) is selected to be 0.9 for this research work. To prevent corrosive acid condensation in the air saturator upper section, 373 K was considered to be the minimum allowable temperature for exhaust gases to be cooled down in the air saturator. However, the gas turbine mainly utilizes natural gas which is a Sulphur-free fuel and its exhaust gases can be cooled down to 338 K [89].

In order to determine the maximum amount of water addition in the upper section of the air saturator, air has to be saturated at state 12. Therefore, similar procedures which are utilized for the lower section of the air saturator's analysis including Equations 3.25-3.29 are implemented.

The energy balance for the upper section of the air saturator is:

\[
(m_{w9} h_{w9} + m_{da9} h_{da9}) + (m_{w10} h_{w10} + m_{da10} h_{da10}) + m_{win\text{m}} h_{win} \\
+ (m_g h_g5 + m_{w5} h_{w5}) = (m_g h_{g6} + m_{w6} h_{w6}) + (m_{w12} h_{w12} + m_{da12} h_{da12})
\]

(3.32)

Where \( h_{gi} \) is the exhaust gas specific enthalpy at state \( i \) in kJ/kg, and \( m_{winm} \) is the maximum mass flow rate of water injected in the upper section of the air saturator in kg/s.

Like the lower section of the air saturator, an iterative procedure is necessary to find the maximum water addition. Manipulating the amount of water injection in the upper section of the air saturator enables designers to adjust the air saturator outlet air temperature and humidity ratio to their desired values. In the case of the MBC configuration, air has to be superheated at the air saturator’s outlet to be able to produce mechanical work in the turbine without any water condensation. Hence, a new variable is introduced as follows:
where \( ASDH \) is the degree of humidification for the upper section of the air saturator, \( \dot{m}_{w_{inU}} \) is the desired amount of water addition in the upper section of the air saturator in kg/s. Thus, the energy balance will be:

\[
\left( \dot{m}_w h_w + \dot{m}_{da} h_{da} \right) + \left( \dot{m}_{w_{10}} h_{w_{10}} + \dot{m}_{da_{10}} h_{da_{10}} \right) + \dot{m}_{w_{inU}} h_{w_{in}} \\
+ (\dot{m}_g h_s + \dot{m}_{w_s} h_{w_s}) = (\dot{m}_w h_{w_{12}} + \dot{m}_{da_{12}} h_{da_{12}})
\]

(3.34)

Selecting a proper value for ASDH allows designers to determine the temperature of humid air departing the air saturator from Equation 3.34. Nevertheless, water addition reduction in the air saturator degrades its capability to fully recover the waste heat available in the topping turbine exhaust gas. This can be verified by calculating the air saturator’s outlet air temperature in Equation 3.34. If the calculated value is greater than the exhaust gas inlet temperature, the air saturator cannot cool down the exhaust gases to the mentioned temperature and the actual exhaust gases’ temperature dumped into the atmosphere is greater than the calculated value. In order to deal with this problem, the maximum achievable air temperature at the air saturator outlet is considered to be:

\[
T_{12} = T_5 - 20
\]

(3.35)

If air temperature at state 12 surpasses the mentioned limit, Equations 3.34 and 3.35 are employed to find \( T_{12} \) and \( T_6 \), respectively.

3.1.9. Evaporator.

An evaporator is devised in the HABC configuration which is responsible for the bottoming cycle air stream humidification. In principle, the compressed air is cooled down and humidified in the evaporator, simultaneously. Therefore, a rather similar formulation to the air saturator model is utilized for the evaporator. Initially, maximum rate of water injection is calculated by assuming that air leaving the evaporator is fully saturated. As a result, mass and energy balance equations are developed to calculate the maximum amount of water injected. Since air is assumed to be saturated at the
evaporator outlet, it is necessary to calculate the mole fraction of water in humid air mixture by:

\[
\dot{n}_{w, ev, o} = \frac{\dot{m}_{w, ev, o} + \dot{m}_{w, in}}{M_w}
\]  

(3.36)

\[
\dot{n}_{d_a, ev, o} = \frac{\dot{m}_{d_a, ev, o}}{M_{d_a}}
\]  

(3.37)

\[
x_{w, ev, o} = \frac{\dot{n}_{w, ev, o}}{\dot{n}_{d_a, ev, o}}
\]  

(3.38)

where \(\dot{m}_{w, ev, o}\) and \(\dot{m}_{w, in}\) are the mass flow rates of water at the evaporator outlet and the amount of water addition in the evaporator in kg/s, \(\dot{m}_{d_a, ev, o}\) is the mass flow rate of dry air at the evaporator outlet in kg/s, \(M_w\) and \(M_{d_a}\) are the molecular weight of the water and humid air respectively in kg/kmol, \(\dot{n}_{w, ev, o}\) and \(\dot{n}_{ev, o}\) are the number of mole for water and humid air at the evaporator outlet in kmol, and \(x_{w, ev, o}\) is the mole fraction of steam at the evaporator outlet.

Consequently, the saturation pressure and temperature can be calculated by:

\[
P_{sat, ev, o} = x_{w, ev, o} P_{ev, o}
\]  

(3.39)

\[
P_{sat, ev, o} \xrightarrow{Steam Table} T_{sat, ev, o}
\]  

(3.40)

where \(P_{sat, ev, o}\) and \(P_{ev, o}\) are the partial pressure of water and total pressure of the mixture at the evaporator outlet in kPa. By assessing the saturation pressure, the temperature of the mixture at the evaporator outlet is determined from the saturated steam table. Consequently, the energy balance equation at the evaporator is:

\[
(\dot{m}_{w, ev, i} h_{w, ev, i} + \dot{m}_{d_a, ev} h_{d_a, ev, i}) + \dot{m}_{w, in} h_{w, in} = (\dot{m}_{w, ev, o} h_{w, ev, o} + \dot{m}_{d_a, ev} h_{d_a, ev, o})
\]  

(3.41)

where \(\dot{m}_{w, ev, i}\) and \(\dot{m}_{w, ev, o}\) are the water mass flow rates at the evaporator inlet and outlet in kg/s, \(h_{w, ev, i}\) and \(h_{w, ev, o}\) are the water specific enthalpy at the evaporator inlet and outlet in kJ/kg, \(\dot{m}_{d_a, ev}\) is the evaporator dry air mass flow rate in kg/s, \(h_{d_a, ev, i}\) and \(h_{d_a, ev, o}\) are the dry air specific enthalpy at the evaporator inlet and outlet in kJ/kg, and \(h_{w, in}\) is the injected water specific enthalpy in kJ/kg. Consequently, a new design variable which
controls the amount of water injected in the bottoming cycle air is introduced. The bottoming cycle air humidification (BCAH) factor which is the ratio of the maximum possible water addition mass flow rate over the desired amount of water addition is calculated as follows:

\[ BCAH = \frac{\dot{m}_{w_{in}}}{\dot{m}_{w_{inm}}} \]  

(3.42)

where \( BCAH \) is the bottoming cycle air humidification, \( \dot{m}_{w_{in}} \) and \( \dot{m}_{w_{inm}} \) are the desired and maximum amount of water addition in the bottoming cycle air in kg/s.

3.1.10. Heliostat field.

Heliostat field collectors are one of the most promising CSP technologies; nonetheless, their mathematical formulation is arguably the most complicated among all the solar collectors available within the market. Complexity in design and optimization of the heliostat field prompted extensive research in developing different codes since 1970s including UHC-RCELL [103-105], DELSOL3 (winDEDLSOL) [106], HFLCAL [107, 108], MIRVAL [109], solTRACE, Fiat Lux, SCT-HGM [110], and HFLD [111, 112]. Garcia et al. [113] categorized the available heliostat field codes into two major clusters. The main criteria for the classification was the accuracy in the codes’ calculation of the reflected power over the receiver for each heliostat. For example, MIRVAL, solTRACE, and Fiat Lux’s main objective is to give a detailed description of the reflected power from the heliostat field without providing any thermodynamic or optimization options. On the contrary, UHC-RCELL, DELSOL, and HFLCAL accuracy in calculating the solar flux on the receiver surface is lower, resulting in faster approximation of the heliostat field performance for optimization purposes.

Instantaneous optical efficiency of a heliostat is determined based on Sandia nomenclature such that [114, 115]:

\[ \eta_{opt,f} = \rho f_{\cos}(x, y, t)f_{at}(x, y)f_{sp}(x, y, t)f_{s&b}(x, y, t) \]  

(3.43)

where \( \eta_{opt,f} \) is the optical efficiency of a single heliostat at a specific time and location, \( \rho \) is the actual mirror reflectivity, \( f_{\cos} \) is the cosine of the incidence angle between the normal vector and the vector toward the sun, \( f_{at}, f_{sp} \) and \( f_{s&b} \) are attenuation, spillage, and shading and blocking factors. As can be seen, calculation of instantaneous
efficiency depends on time and location of the investigated heliostat. Consequently, annual field efficiency calculation is significantly complex and time consuming. Furthermore, optimization of the field is even more complex based on all the required calculations for each heliostat throughout a whole year, considering that commercial heliostat fields consist of thousands of heliostats. Some simplifying assumptions must be taken into account for analyzing the most time consuming factor, i.e. the shading and blocking factor [116].

A new optimization method was proposed by Collado [115] and Collado and Guallar [56, 116], named Campo, which systemizes the optimization procedure. This method reduces the complexity and time consumption associated with the optimization of heliostat fields. In heliostat field design, heliostat density decreases as its radial distance from the tower increases. In other words, areas closer to the tower must have higher mirror densities since they have greater optical efficiencies. Unfortunately, as the distance between two adjacent heliostat abates (greater density), the shading and blocking factor decreases as a higher portion of the mirror will be covered or blocked by other heliostats. This issue reduces the overall optical efficiency of the field. In consequence, it is necessary to place heliostats within the medium range of the tower with relatively lower spillage and attenuation efficiencies. In general, there will be a trade-off between the shading and blocking factors and the other optical efficiency factors. As the density of the field reduces, the number of heliostats located within the medium to long ranges of the solar tower increases which results in a better shading and blocking factor and at the same time, worsens the other factors. This trade-off is the main objective of the optimization process: to find a balance between the number of heliostats within a single row and radial dispersion of rows of mirrors from the tower.

Regardless of the optimization method, a preliminary field layout is the initial step to begin the optimization process. It is required to generate thousands of coordinates for mirrors within the field in a fast and efficient manner. Most of the introduced codes’ field layouts are based on radially staggered positions of the heliostats in the field. The radial-staggered configuration is the most popular and commonly used field layout which has been widely utilized in heliostat field projects. Figure 21 depicts the radial-staggered field layout. There are other different heliostat field layouts proposed by Noone et al. [117]. The author argued that the transition between the high and low density areas within the radial-staggered field is not
continuous. Thus, a new heuristic is presented based on the spiral patterns of phyllotaxis discs. This layout was studied by Besarati and Goswami [118] for designing and optimization of a 50 MWth heliostat field for Dagget, California. The maximum yearly insolation weighted efficiency is estimated to be 0.683 by utilizing 594 heliostats. In another study by Ramos and Ramos [119] a systematic procedure for optimization of different aspects of the heliostat field collectors including field layout, heliostats’ position, tower height, and receiver’s dimensions are presented. The major advantage of the proposed optimization procedure is its explicit definitions of different design variables which enable the optimized layout to come in different shapes. Siala and Elayeb [120] proposed a graphical formulation for a no-blocking heliostat field. In their radially staggered layout, the distance between two adjacent rows of heliostats is allocated such that there will be no blocking of any mirrors within the field.

![Radial-staggered heliostat field layout](image)

**Figure 21: Radial-staggered heliostat field layout [116]**

The densest possible heliostat field layout is fed into the Campo code [56, 115, 116] as its initial field layout to begin the optimization procedure. In principle, Campo starts its optimization with the densest field layout and continues expanding the layout until the efficiency enhancement due to the layout expansion approaches zero. This optimization method follows a certain path by increasing the radial distance between adjacent rows of heliostats within each zone to maximize the annual optical efficiency.
The main advantage of Campo code is its clarity in optimization, as the users are fully aware of the steps that must be taken during optimization. Additionally, the optimization process will be less time consuming as it only needs to follow a certain path rather than wandering around the feasible region searching for the optimum layout.

Initially, the characteristic diameter of each heliostat is defined as follows [116]:

$$DM = \sqrt{LW^2 + LH^2 + dsep}$$  \hspace{1cm} (3.44)

where $DM$ is the characteristic diameter of the heliostat in m, $LW$ and $LH$ are the width and height of the heliostat in m, and $dsep$ is any additional separation distance between adjacent heliostats. The minimum radial increment within the first zone of the field is determined by [116]:

$$\Delta R_{min} = DM\cos30 - \left( R_1 - \sqrt{R_1^2 - \frac{DM^2}{4}} \right)$$  \hspace{1cm} (3.45)

where $\Delta R_{min}$ is the minimum radial increment within the first zone of the field in m, and $R_1$ is the radial distance from the tower to the first row of heliostats in m. Due to the considerably greater value of $R_1$ than $DM$, the expression within the parentheses is almost zero and can be neglected. Finally, the azimuth angular spacing for the first zone (constant within the first zone) of the heliostat field can be determined by [116]:

$$\Delta az_1 = 2\sin \left( \frac{DM}{2R_1} \right)$$  \hspace{1cm} (3.46)

where $\Delta az_1$ is the azimuth angular spacing for the first zone in rad.

Based on the number of heliostats in each row of the first zone, radial distance from the tower to the first row of the heliostats ($R_1$) can be determined as follows [116]:

$$R_1 = N_{hel1} \left( \frac{DM}{2\pi} \right)$$  \hspace{1cm} (3.47)

where $N_{hel1}$ is the number of heliostats within each row of the first zone.

Due to the radially staggered configuration, the distance between two adjacent mirrors (length in meters) depends on their radial distance from the tower. Consequently, as the rows get further away from the tower, the space between their heliostats increases. Eventually, spaces between the mirrors become greater than $DM$.
making it possible to place extra mirrors between two adjoining mirrors in the same row. A zone is considered to be completed as the possibility of the addition of extra mirrors within its final row is achieved. Consequently, the azimuth angular spacing for the $i^{th}$ zone can be determined by [116]:

$$\Delta az_i = \left(\frac{\Delta az_1}{2^{i-1}}\right)$$

(3.48)

where $\Delta az_i$ is the azimuth angular spacing for the $i^{th}$ zone in rad. The radial distance from the tower to the first row of the $i^{th}$ zone and number of heliostats for each row within the $i^{th}$ zone are calculated such that [116]:

$$R_i = 2^{i-1}\left(\frac{DM}{\Delta az_1}\right)$$

(3.49)

$$N_{hel_i} = \left(\frac{2\pi}{\Delta az_i}\right)$$

(3.50)

where $R_i$ is the radius of the first row for the $i^{th}$ zone in m, $N_{hel_i}$ is the number of heliostats in each row within the $i^{th}$ zone. The number of rows for the $i^{th}$ zone is determined by [116]:

$$N_{row_i} = \left(\frac{R_{i+1} - R_i}{\Delta R_{min}}\right)$$

(3.51)

where $N_{row_i}$ is the number of rows of heliostats for $i^{th}$ zone of the field. Now, we are able to generate the densest field layout to begin the expansion and optimization of the heliostat field.

To set up the densest configuration, several initial design parameters are required. All initial assumptions, constraints, and design variables to set up and analyze the optical efficiency of the field are tabulated in Table 6. The densest field layout configuration as the initial input fed into the optimization process is depicted in Figure 22. Each zone is distinguished with a different color. The first row consists of 20 heliostats. As seen, field density reduces within each zone by moving outward away from the center of the field (tower). The first zone contains 60 mirrors spread equally in 3 rows. The second zone contains 280 mirrors distributed equally within 7 rows. Finally, the third zone contains 1120 mirrors distributed equally within 14 rows. In total, this initial design field layout contains 24 rows of heliostats and 1460 mirrors.
The Campo densest field layout generation is significantly dependent on the number of heliostats within its first row. For instance, increasing the number of heliostats within the first row from 20 to 25 increases the number of heliostats within the field from 1460 to 2350. Consequently, the number of heliostats in the first row is an important design variable, and extra care must be taken in choosing an appropriate value.

Figure 22: Densest field layout configuration for heliostat field design

Optimization of the heliostat field is achieved by means of expanding the layout to improve shading and blocking factors while other optical efficiency factors are degraded. Optimum configuration is obtained as the positive and negative impacts of layout expansion are approximately equal. In such a case, further alteration in field layout will only result in decreasing the layout optical efficiency. Expansion of the field
must follow a systematic procedure similar to the process of generating the densest configuration. Based on an initial estimation of the blocking factor, local radial increment between two staggered rows is calculated for each heliostat within the field. Therefore, the local radial increment for each heliostat is verified by [116]:

\[
\Delta R_{i,j} = \left[ \frac{\cos(\omega_{i,j})}{\cos(\varepsilon_{T,i,j})} \left( 1 - \frac{(1 - f_{b,ref})(LW/LH)}{2(LW/LH) - \left( \sqrt{1 + \left( \frac{LW}{LH} \right)^2 + \frac{dsepa}{LH}} \right)} \right] LH \quad (3.52)
\]

where \(\Delta R_{i,j}\) is the local radial increment between the \(j^{th}\) heliostat in the \(i^{th}\) row and its preceding row in m, \(\omega_{i,j}\) is the incident angle of the sunrays onto the \(j^{th}\) heliostat in the \(i^{th}\) row surface in rad, \(\varepsilon_{T,i,j}\) is the tower unit vector elevation angle origins from the zenith, and \(f_{b,ref}\) is the reference blocking factor that can be selected in order to expand the field layout during the optimization. It is important to indicate that the reference blocking factor is a rough estimation of the shading and blocking factor and cannot be used in analyzing the annual optical efficiency [116]. For the shading and blocking factor, accurate analysis presented by Sassi [121] is considered.

Table 6: Initial assumptions, constraints, and design variables for heliostat field design [25, 56, 115-117, 122]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Assigned value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tower optical height (m)</td>
<td>120</td>
</tr>
<tr>
<td>Receiver radius (m)</td>
<td>4</td>
</tr>
<tr>
<td>Receiver height (m)</td>
<td>9</td>
</tr>
<tr>
<td>Receiver inner radius (m)</td>
<td>1</td>
</tr>
<tr>
<td>Receiver outer radius (m)</td>
<td>2</td>
</tr>
<tr>
<td>Receiver nominal outlet temperature (K)</td>
<td>1223</td>
</tr>
<tr>
<td>Heliostat height (m)</td>
<td>12.30</td>
</tr>
<tr>
<td>Heliostat width (m)</td>
<td>9.75</td>
</tr>
<tr>
<td>Heliostat vertical elevation from the ground (m)</td>
<td>5</td>
</tr>
<tr>
<td>Standard deviation of surface error (mrad)</td>
<td>0.94</td>
</tr>
<tr>
<td>Standard deviation of tracking error (mrad)</td>
<td>0.63</td>
</tr>
<tr>
<td>Standard deviation of sun shape (mrad)</td>
<td>2.51</td>
</tr>
<tr>
<td>Effective reflectivity</td>
<td>0.836</td>
</tr>
<tr>
<td>Field number of zones</td>
<td>3</td>
</tr>
<tr>
<td>Number of heliostat in the first row</td>
<td>20</td>
</tr>
<tr>
<td>Number of heliostat in the field</td>
<td>1460</td>
</tr>
<tr>
<td>Extra separation distance dsepa (m)</td>
<td>0</td>
</tr>
<tr>
<td>Latitude location (°N)</td>
<td>24.47</td>
</tr>
<tr>
<td>Field Layout</td>
<td>Radial-staggered/Spiral</td>
</tr>
<tr>
<td>Field number of cells</td>
<td>100</td>
</tr>
<tr>
<td>Field life cycle (years)</td>
<td>25</td>
</tr>
<tr>
<td>Loan interest rate</td>
<td>0.07</td>
</tr>
<tr>
<td>Insurance rate</td>
<td>0.01</td>
</tr>
<tr>
<td>Typical Meteorological Year (TMY)</td>
<td>Abu Dhabi [123]</td>
</tr>
</tbody>
</table>
It is important to note that \( \cos(\varepsilon_{T_{i,j}}) \) and \( \cos(\omega_{i,j}) \) depend on the location of the heliostat with respect to the tower. Consequently, the local radial increment can differ from one heliostat to the other within the same row. In addition, \( \cos(\omega_{i,j}) \) also depends on the selected time for optimization. Collado recommends spring equinox-solar noon as the most appropriate time for the analysis; however, he argues that a time-averaged distribution of \( \cos(\omega_{i,j}) \) is a more accurate alternative [115]. For this study, it was decided to choose a daily time-averaged (during sunlight hours) distributed \( \cos(\omega_{i,j}) \) for spring equinox rather than employing an annual time-averaged distributed \( \cos(\omega_{i,j}) \) to accelerate the optimization process. The reason behind the selected approach is further explained in the following sections. Furthermore, one may notice that calculating \( \Delta R_{i,j} \) is an implicit problem since \( \cos(\varepsilon_{T_{i,j}}) \) and \( \cos(\omega_{i,j}) \) are dependent on the location of the heliostat which can only be verified after calculating \( \Delta R_{i,j} \). Thus, an iterative process with an initial guess of \( \Delta R_{min} \) is needed to be executed in order to determine the actual value of \( \Delta R_{i,j} \).

Calculating \( \Delta R_{i,j} \) enables designers to systematically expand the field layout by adjusting the value of the reference blocking factor as follows:

\[
\Delta R_{i,j} = \max(\Delta R_{i,j}, \Delta R_{min})
\]  

(3.53)

In order to understand the impact the reference blocking factor has on the field layout, its effect is studied with the results depicted in Figure 23. These layouts are generated by considering spring equinox solar noon as the design instant. The dashed black circle represents the densest field layout configuration depicted in Figure 22. Obviously, increasing the reference blocking factor leads to a more expanded field layout. Nevertheless, expansion is different from one zone to the other as the most considerable expansion is achieved within the third zone. Moreover, the field layout is more extended in the north than the south as the heliostat density is greater in the southern semicircle of the field. Additionally, the generated field layout with blocking factor of 0.5 is identical to the densest field layout, whereas the presented layout for blocking factor of 0.75 results in a minor expansion in the northern section of the field.
Consequently, low blocking factors have no effect on the field layout southern semicircle.

![Figure 23: Field layout configuration for different reference blocking factors (spring equinox-solar noon), a) Field layout for reference blocking factor of 0.5, b) Field layout for reference blocking factor of 0.75, c) Field layout for reference blocking factor of 0.95, d) Field layout for reference blocking factor of 1.0](image)

As previously discussed, $\cos(\omega_{i,j})$ is a function of time. Consequently, selected time of the optimization influences the expanded layout. Furthermore, selecting an appropriate design period is necessary. Initially, the effects of varying the optimization design day are depicted in Figure 24. It is important to note that analysis is performed at solar noon with reference blocking factor of 0.95. One may conclude that variation in day does not have as significant an effect on the expanded field layout as the blocking factor. As long as the analyses are executed at solar noon, it can be noted that the southern semicircle of the field has higher mirror density. Nevertheless, the difference
between the mirror density of the northern and southern semicircles are minimized during summer and maximized during winter.

Figure 24: Field layout configuration for different selected design days at solar noon ($f_{b,ref} = 0.95$), a) Field layout for 21\textsuperscript{st} of March (spring equinox), b) Field layout for 30\textsuperscript{th} of May, c) Field layout for 7\textsuperscript{th} of September, d) Field layout for 16\textsuperscript{th} of December

Expanded field layouts for different selected design times of the day at the spring equinox with reference blocking factor of 0.95 are illustrated in Figure 25. One can clearly see the significant impact the selected time of the day has on the field layout. Note that all the mentioned hours are solar hours rather than standard local hours. During morning, the expanded layout has a higher mirror density within the eastern semicircle of the field. Considering that the black dashed circle represents the densest possible field layout, almost zero expansion is required within the eastern quarter of the field. As the selected hour approaches solar noon, mirror concentration moves from east to south of the tower. During afternoon, higher mirror density is located in the west.
side of the tower. It can be concluded that selecting a proper hour for optimization is more crucial than the selected day for the analysis.

Figure 25: Field layout configuration for different selected design times of the day at spring equinox ($f_{\text{b,ref}} = 0.95$), a) Field layout at solar hour of 8:00, b) Field layout at solar hour of 10:00, c) Field layout at solar noon (solar hour of 12:00), d) Field layout at solar hour of 14:00

Collado’s [116] suggestion of having a more accurate time-averaged distributed $\cos(\omega_{i,j})$ is intriguing. Nonetheless, it only increases the calculation load during optimization. Considering that the process of field expansion will be more precise by implementing a time-averaged distributed $\cos(\omega_{i,j})$, the field optical efficiency is accurately analyzed without any significant assumption and generalization. Therefore, the time-averaged distributed $\cos(\omega_{i,j})$ will not change the optimum field layout and only affects the expansion process. It might accelerate or decelerate the optimization process. As a result, it was decided to choose a daily time-averaged (during sunlight hours) distributed $\cos(\omega_{i,j})$ for the spring equinox rather
than employing an annual time-averaged distributed $\cos(\omega_{i,j})$ to accelerate the optimization process. Field layout configuration for the daily time-averaged and instantaneous cosine factor calculations at spring equinox with reference blocking factor of 0.95 are depicted in Figure 26. Note that the time-average field layout expansion within the northern hemisphere of the field is lower than the field layout attained with instantaneous cosine factor calculation. It can be concluded that the expansion process has lower dependency on the variation of the reference blocking factor when the daily time-averaged approach is employed. Therefore, it is believed that the proposed modification reduces the calculation load associated with the optimization of the field.

![Field layout configuration for time-averaged and instantaneous cosine factor calculation at spring equinox ($f_{b,ref} = 0.95$), a) Field layout with cosine factor calculation at noon, b) Field layout with time average cosine factor calculation](image)

Figure 26: Field layout configuration for time-averaged and instantaneous cosine factor calculation at spring equinox ($f_{b,ref} = 0.95$), a) Field layout with cosine factor calculation at noon, b) Field layout with time average cosine factor calculation

There is another promising heliostat field layout proposed by Noone et al. [117]. The authors argue that the transition between the high and low density areas within the field is not continuous in the radial-staggered layout. Thus, a new heuristic is presented based on the spiral patterns of the phyllotaxis discs [124]. It is reported that the new pattern replacing radial-staggered configuration will improve the field optical efficiency and reduce the land area [118]. The proposed field layout has the advantage of continuous density function unlike the radial-staggered configuration. Nonetheless, this problem is not an issue for the proposed method of layout generation by Collado [115] since a systematic formulation based on a reference blocking factor is
implemented for expansion. To generate the field with a new pattern of phyllotaxis in the form of florets on the head of a sunflower, the following formulas are employed [117, 118]:

$$\theta_i = 2\pi i \left(\frac{1 + \sqrt{5}}{2}\right)^{-2}$$

$$r_i = a i^b$$

where $\theta_i$ is the polar angle of the $i^{th}$ element of the spiral pattern origin at north in rad, $r_i$ is the polar radius of the $i^{th}$ element of the spiral pattern in m, $a$ and $b$ are the control variables in designing the field layout as their variation can be utilized within the optimization process. For the optimization of a field design based on the introduced spiral pattern, $a$ and $b$ are varied in the ranges of $(2,8)$ and $(0.5,0.7)$, respectively [117]. The new field layout’s annual optical efficiency is also calculated to determine the optimum field layout configuration for a specific location.

Another important constraint that is taken into account while optimizing the heliostat field is to make sure that the distances between the centers of every two adjacent heliostats in the field are greater than $DM$. Therefore, the developed MATLAB code calculates every new heliostat center distance from its neighboring heliostats’ centers to ensure that there is no overlapping between any two heliostats within the field. To better understand the effect of the aforementioned constraint, Figure 27 is presented. It is important to mention that with low values of $a$ and $b$, the spiral layout will be significantly condensed with several heliostats overlapping each other. On the other hand, with high values of $a$ and $b$, the field layout will be overexpanded. Consequently, the developed MATLAB code only considers the heliostats that do not overlap with other heliostats’ surfaces. By this method, the plant expansion can be properly controlled by $a$ and $b$ values.
Figure 27: Spiral field layout ($a = 2, b = 0.5$), a) The constraint flow chart b) Field layout without constraint consideration, c) Field layout with constraint consideration

Since spiral field layouts have two design variables, the proposed method by Ramos and Ramos [119] is selected for its optimization. An algorithm based on Powell’s method [125] is suggested. The optimization begins with an initial approximation of the optimum values of the design variables. Afterwards, a line search
is performed to identify the optimum value for the first design variable. The same procedure is repeated to calculate all design variables’ optimum values completing the first iteration. Multiple iterations are performed until the variation in all the optimum values of the design variables reaches satisfactory accuracy [119]. Ramos and Ramos [119] argued that exploring the design variables one by one main advantage over approaching the optimum value along a direct approach is that the objective function does not need to be fully evaluated for each step. Moreover, calculating the objective function gradient is extremely complicated for heliostat field design, whereas re-evaluating the objective function is considerably simpler [119].

A preliminary field layout using spiral patterns of phyllotaxis discs is presented in Figure 28. The number of heliostats within the field is intentionally selected to be similar to the aforementioned radial-staggered configuration for proper comparison (1460 heliostat mirrors). Considering hypothetical circles dividing the field into three zones like the radial-staggered configuration, the first zone contains 64 mirrors whereas the radial-staggered configuration first zone contains 60 mirrors. Second zones of the spiral and radial-staggered layouts include 172 and 280 mirrors, respectively. The third zone of the spiral layout consist of 1224 mirrors, whereas the number of mirrors within the third zone of the radial-staggered layout is 1120. It is important to mention that the comparison is mainly focused on the difference between the spiral arbitrary layout and the densest radial-staggered configuration. Therefore, there is no surprise that the densest radial-staggered configuration has higher mirror density. Nevertheless, comparing the first zone number of mirrors and keeping in mind that closer mirrors to the tower have higher optical efficiencies, it is fascinating that the spiral layout manages to place a greater number of mirrors within a specific distance of the tower (first zone) as compared with the densest radial-staggered configuration. Consequently, a spiral field layout has the potential to achieve higher optical efficiency compared with the radial-staggered configuration. Moreover, no two heliostat centers within the spiral layout share the same azimuth angle which is another advantage over the radial staggered layout as it might improve the blocking factor significantly [117].
The most influential optical factor on heliostats’ performance is cosine factor [118]. In principle, cosine factor is the cosine of the angle between the incident solar beam radiation and the normal vector of the surface of the heliostat. Consequently, its value is a strong function of time (position of sun) and location of the heliostat in the field. Based on the vectors and angles illustrated in Figure 29, the normal vector from the sun to the surface of the heliostat and from the surface to the receiver are determined by:

\[
\vec{v}_T = \begin{bmatrix}
- \sin \theta_H \cos \theta_T \\
- \cos \theta_H \cos \theta_T \\
\sin \theta_T
\end{bmatrix}
\] (3.56)
\[ \vec{V}_s = \begin{bmatrix} -\sin \theta_z \sin \gamma_s \\ -\sin \theta_z \cos \gamma_s \\ \cos \theta_z \end{bmatrix} \] (3.57)

where \( \theta_H \) is the angular position of the heliostat in the field from north (east positive) in rad, \( \theta_T \) is tower elevation angle in rad, \( \theta_z \) and \( \gamma_s \) are sun zenith and azimuth angles in rad, \( \vec{V}_T \) is the unit vector from the heliostat surface to the receiver, and \( \vec{V}_s \) is the vector from heliostat surface to the sun.

\[ \vec{n} = \frac{\vec{V}_s + \vec{V}_T}{\| \vec{V}_s + \vec{V}_T \|} \] (3.58)

\[ f_{\cos} = \cos \omega = \vec{V}_s \cdot \vec{n} \] (3.59)

where \( \vec{n} \) is the normal vector of the surface of the heliostat, and \( \cos \omega \) is the cosine of the angle between the heliostat normal and incident solar beam radiation. Position angles of the sun and heliostats are determined based on the solar relationships defined and presented by Duffie and Beckman [26] as follow:

Figure 29: Geometric parameters defining the heliostat field [122]
\[\theta_T = \tan^{-1}\left[\frac{h_T - h_H}{r_H}\right]\] (3.60)

\[\delta = 23.45 \sin\left(360 \times \frac{284 + n}{365}\right)\] (3.61)

\[\theta_{\text{hour}} = \frac{\pi}{12} \left[\text{hour} - 12\right]\] (3.62)

\[\theta_z = \cos \theta_{\text{lat}} \cos \delta \cos \theta_{\text{hour}} + \sin \theta_{\text{lat}} \sin \delta\] (3.63)

\[\gamma_S = \text{sign}(\theta_{\text{hour}}) \left|\cos^{-1}\left(\frac{\cos \theta_z \sin \theta_{\text{lat}} - \sin \delta}{\sin \theta_z \cos \theta_{\text{lat}}}\right)\right|\] (3.64)

where \(h_T\) and \(h_H\) are the tower and heliostat heights in m, \(r_H\) is the radial distance between the tower and the heliostat in m, \(\delta\) is the declination angle in rad, \(n\) is the \(n^{th}\) day of the year, \(\theta_{\text{hour}}\) is the sun hour angle in rad, \(\text{hour}\) is an indication of the time of the day, and \(\theta_{\text{lat}}\) is the latitude of the plant location in rad.

Another important optical factor that impacts the field efficiency which must be studied is the atmospheric attenuation factor. The reflected beam from the heliostat surface might be scattered. Based on the distance of the heliostat from the receiver, the percentage of the losses due to the reflected beam scattering is increased as follows [118]:

\[d_R = \sqrt{r_H^2 + (h_T - h_H)^2}\] (3.65)

\[f_{at} = \begin{cases} 0.99321 - 0.000176d_R + 1.97 \times 10^{-8}d_R^2 & d_R \leq 1000 \\ e^{-0.0001106d_R} & d_R > 1000 \end{cases}\] (3.66)

where \(d_R\) is the distance between the point of reflection from the heliostat surface to the point of absorption on the receiver in m.

Another important and cumbersome optical factor to evaluate is the spillage factor. In some studies, spillage factor is referred to as the intercept factor. Understandably, a portion of the reflected sun image might not intercept the receiver. Different reasons can cause the intercept inefficiency including tracking error, shape of sun, and mirror surface non-uniformity [126]. There are two major approaches in analyzing this efficiency factor. The first method, which is ray tracing or the Monte
Carlo ray tracing method, is more accurate but time consuming. The second method is the analytical integration of the reflected image on the receiver surface over the receiver domain (convolution method). The first method, Monte Carlo ray tracing, is mainly employed when an accurate performance analysis of the field is required. On the other hand, analytical integration can be used as an acceptable approximation of the intercept factor for the optimization of the field layout. There are several codes adopting the Monte Carlo ray tracing method such as SolTRACE, MIRVAL, and TONATIUH [113]. Other heliostat field codes employ the convolution method including UHC, DELSOL, HFLCAL, and FIAT LUX [113].

There are two popular analytical integration methods available in the literature, i.e. the UNIZAR model developed by Universidad de Zaragoza [127, 128] and the HFLCAL model developed by the German Aerospace Center (DLR) [108]. Based on the study conducted by Collado [129], both methods are appropriate tools for estimating the spillage factor whilst HFLCAL is the simpler and more accurate method. Therefore, it was decided to adopt the HFLCAL method in this research work to estimate the intercept factor of the heliostat field. HFLCAL considers that all the heliostats are well-canted concentrating facets of spherical curvatures [56]. A circular normal distribution represents the flux density in the HFLCAL model. Therefore, the intercepted fraction of energy on the receiver surface is estimated as follows [56]:

\[
    f_{sp} = \frac{1}{2\pi \sigma_{tot}^2} \int_{x'} \int_{y'} \exp\left(-\frac{x'^2 + y'^2}{2\sigma_{tot}^2}\right) dy' dx'
\]  

(3.67)

where \(\sigma_{tot}\) is the total standard deviation measured on the image plane, and \(x'\) and \(y'\) are the integration coordinates in m. Integration limits are calculated based on the dimensions of the receiver given in Table 6. The cylindrical receiver will be seen as a rectangle with the following dimensions [56]:

\[
    Lx' = 2RR
\]  

(3.68)

\[
    Ly' = HR \cos \theta_T
\]  

(3.69)

where \(Lx'\) and \(Ly'\) are the analytical integration limits, \(RR\) and \(HR\) are the receiver actual radias and height in m.
The total standard deviation is calculated based on the convolution of four Gaussian error functions, i.e. sun-shape error, beam quality error, astigmatic effect, and tracking error. Thus, the total standard deviation is determined by [56]:

$$\sigma_{\text{tot}} = \sqrt{d_R \left( \sigma_{\text{sun}}^2 + \sigma_{\text{bq}}^2 + \sigma_{\text{ast}}^2 + \sigma_{\text{tre}}^2 \right)}$$  \hspace{1cm} (3.70)

where $\sigma_{\text{sun}}$ is the standard deviation associated with sun-shape error in rad, $\sigma_{\text{bq}}$ is the standard deviation associated with beam quality in rad, $\sigma_{\text{ast}}$ is the standard deviation associated with astigmatic effect in rad, and $\sigma_{\text{tre}}$ is the standard deviation associated with tracking error in rad. Standard deviation due to the sun-shape is taken from [56] as 2.51 mrad.

The beam quality standard deviation is associated with the slope error due to the non-uniformity within the shape of the curvature of the mirror and waviness and roughness of the reflecting surface. Its value can be calculated by [56]:

$$\sigma_{\text{bq}}^2 = (2\sigma_{\text{sre}})^2$$  \hspace{1cm} (3.71)

where $\sigma_{\text{sre}}$ is the standard deviation associated with surface error and its value is taken from [56] to be 0.94 mrad. Additionally, standard deviation associated with tracking error is considered to be 0.63 mrad [56]. Finally, astigmatic standard deviation is determined by [56]:

$$\sigma_{\text{ast}} = \frac{\sqrt{0.5(H_t^2 + W_s^2)}}{4d_R}$$  \hspace{1cm} (3.72)

$$H_t = \left(\sqrt{LW \times LH}\right)\left|\frac{d_R}{f} - \cos \omega \right|$$  \hspace{1cm} (3.73)

$$W_s = \left(\sqrt{LW \times LH}\right)\left|\frac{d_R}{f} \cos \omega - 1 \right|$$  \hspace{1cm} (3.74)

where $H_t$ and $W_s$ are the image dimensions in the tangential and sagittal planes at a distance $d_R$ from the mirror, respectively, and $f$ is the focal distance which is equal to $d_R$ when the heliostat is focused on its slant range as usual [56].

The last optical efficiency factor required to complete the analysis of the heliostat field collectors is the shading and blocking factor. Shading is mainly the result of the obstruction of the incoming solar radiation by neighboring heliostats whereas blocking occurs as the reflected beams from the heliostat surface are partially
obstructed by another heliostat before reaching the receiver. Shading and blocking factor’s calculation is the most time consuming among all discussed optical efficiencies since its value is a strong function of the positions of the sun, heliostat, and neighboring heliostats. In consequence, the shading and blocking factor is required to be re-evaluated with variations in heliostat positions and time. Nonetheless, the computational time could be significantly shortened by identifying neighboring heliostats with a high potential of shading and blocking. In radial-staggered field layout, only three specific neighboring heliostats could block the analyzed heliostat. As depicted in Figure 30, the analyzed heliostat is illustrated as a black circle and possible blocking heliostats are shown in blue and yellow circles. Therefore, only shouldering heliostats (blue circles) which are the ones located in the front row of the analyzed heliostat field and the nose heliostat (yellow circle) which is the closest heliostat in the front of the analyzed heliostat with the same azimuth angle are possible candidates for blocking effect. In general, two shoulders and one nose heliostat are identified for each heliostat and blocking detailed analysis is performed.

On the other hand, identifying possible shading heliostats is a more complicated task since the potential candidates of shading heliostats vary with the position of the sun and time of the day. Collado and Guallar [116] recommend dividing the field into seven unequal sectors based on trial and error. Each sector’s possible shading heliostats are identified. Consequently, a maximum of three shading heliostats along with three identified blocking heliostats are examined for shading the analyzed heliostat. Besarati and Goswami [118] proposed a novel model for identifying possible shadowing and blocking heliostats. An imaginary circle with its center at the analyzed heliostat and radius of $2.5DM$ is considered. Afterward, projection of the vector from the heliostat to the sun on the horizontal plane is drawn. A line perpendicular to the projected vector is considered, dividing the circle into two semicircles. The heliostats located within the semicircle closer to the sun are the potential candidates for shading. Out of those heliostats located within the selected semicircle, those with lower perpendicular distance from the projected vector have a higher potential of shading the analyzed heliostat.
Nevertheless, it is decided to choose the method presented by Noone et al. [117] for the analysis of this research work. Similar to the proposed method by Besarati and Goswami [118], a projection of the vector connecting the center of the analyzed heliostat to the sun on the horizontal plane is considered. Any heliostat whose center distance from the projected vector is less than $D_M$ could be a potential candidate for shading the analyzed heliostat. In addition, in this research work we have calculated the distance between the centers of the shading candidate and the analyzed heliostat. Based on the analyses by Besarati and Goswami [118], we decided to further analyze the shading effect of the heliostats that satisfy the following two conditions:

1. Its center distance from the projected sun vector is less than $D_M$

2. Its center distance from the analyzed heliostat center is less than $2.5D_M$

Unlike the radial-staggered field layout, spiral layout blocking heliostats cannot be identified without prior knowledge of the neighboring heliostats’ positions.
Therefore, the same procedure presented for identifying the possible shading heliostats in the radial-staggered layout is adopted for spotting the possible blocking and shadowing heliostats for spiral field layout shading and blocking factor calculation. The implementation of the abovementioned constraints is depicted in Figure 31. It should be noted that the red circle represents the analyzed heliostat whereas the green circles are the potential blocking candidates. Furthermore, it is important to mention that even the slightest connection between a heliostat and the rectangle or circle representing the two constraints implies that the mentioned heliostat satisfies that specific constraint.

As an example, the analyzed heliostat in Figure 31 has three potential blocking candidates. Clearly, one of the candidates is not viable as it is not located between the tower and the heliostat. Consequently, one can conclude that the utilized model can be further improved by dividing the circle into two semicircles as proposed by Besarati and Goswami [118]. Furthermore, it should be noted that the above said model will result in four potential candidates. Nonetheless, we still believe that the aforementioned model is more efficient in areas with higher mirror density; though, the calculation load associated with our model can be further reduced by considering the approach suggested by Besarati and Goswami [118] in which the circle is divided into two semicircles. Nevertheless, we are satisfied with the accuracy and calculation load of our utilized model and decided against the implementation of the aforementioned approach.

To calculate the shading and blocking factor, the formulation presented by Sassi [121] is adopted. The mirror surface is divided into several narrow vertical bands or stripes. Previously identified shading and blocking heliostats’ surfaces are projected on the plane of the analyzed heliostat. Every stripe’s height associated with every possible shading and blocking candidates are collected, and the maximum height for each stripe is elected for determining the percentage of the mirror area free of blocking and shadowing. The shading and blocking factor is simply determined by dividing the free area of the mirror over the overall area of the mirror. An important assumption which considerably reduces the computational time of the shading and blocking factor analysis is to assume that the blocking and shadowing candidates and the analyzed heliostat are located in parallel planes [130]. This simplification is less favorable than the actual situation. Nonetheless, it implies that only the centers of the possible blocking and shadowing heliostats are needed to be projected on to the analyzed heliostat [130].
Figure 31: Potential blocking heliostat candidates (red circle: analyzed heliostat, brown circle: heliostat satisfying only one of the constraints, yellow circle: heliostat satisfying only one of the constraints, green circle: potential blocking candidates)

By projecting the center of an arbitrary neighboring heliostat onto the plane of the analyzed heliostat in a coordinate system which lies in the plane of the analyzed heliostat, its coordinates are determined as follows [121]:

\[
x_e = \frac{1}{\phi} \left\{ \left[ \frac{V_{nx}}{\Omega} (V_x V_{ny} - V_y V_{nx}) - V_{ny} \right] X_0 + \left[ \frac{V_{ny}}{\Omega} (V_x V_{ny} - V_y V_{nx}) + V_{nx} \right] Y_0 \right.
\]
\[
+ \left[ \frac{V_{nz}}{\Omega} (V_x V_{ny} - V_y V_{nx}) \right] Z_0 \} \tag{3.75}
\]

\[
y_e = \frac{1}{\Omega \phi} \left\{ -V_{nx} X_0 - V_{ny} Y_0 + (V_x V_{nx} + V_y V_{ny}) Z_0 \right\} \tag{3.76}
\]

\[
z_e = 0 \tag{3.77}
\]

\[
\Omega = V_x V_{nx} + V_y V_{ny} + V_z V_{nz} \tag{3.78}
\]
\[ \phi = \sqrt{V_{nx}^2 + V_{ny}^2} \]  

(3.79)

where \( V_x, V_y \) and \( V_z \) can be either the components of the vector from the heliostat to the sun or the tower depending on calculating the possibility of blocking or shadowing factors. \( V_{nx}, V_{ny} \) and \( V_{nz} \) are the components of the normal vector of the surface of the analyzed heliostat, \( X_0, Y_0 \) and \( Z_0 \) are the coordinates of the neighboring heliostat center in a coordinate system where the analyzed heliostat center is \([0,0,0]\), and \( x_e, y_e \) and \( z_e \) are the coordinates of the projection within the analyzed heliostat plane. Overlapping can only occur if the following conditions are satisfied [121]:

\[
\frac{1}{\Omega} \left\{ -V_z V_{nx} X_0 - V_z V_{ny} Y_0 + (V_x V_{nx} + V_y V_{ny}) Z_0 \right\} \leq Z_0 
\]  

(3.80)

\[ |x_e| \leq LW \]  

(3.81)

\[ |y_e| \leq LH \]  

(3.82)

The significant contribution of the mathematical formulation proposed by Sassi [121] is its direct approach in treating the possible overlapping without any simplification. Possible overlapping is categorized into four different types as depicted in Figure 32:

\[ a: x_e < 0 \& y_e > 0 \]  

(3.83)

\[ b: x_e > 0 \& y_e > 0 \]  

(3.84)

\[ c: x_e < 0 \& y_e < 0 \]  

(3.85)

\[ d: x_e > 0 \& y_e < 0 \]  

(3.86)
Figure 32: Different types of overlapping by neighboring heliostats [121]

Dimensions of the overlapping rectangle are calculated by [121]:

\[ u = LW - |x_e| \]  \hspace{1cm} (3.87)
\[ v = LH - |y_e| \]  \hspace{1cm} (3.88)

where \( u \) and \( v \) are the overlapping rectangle dimensions. Finally, the stripes’ heights are determined such that [121]:

\[ F(i) = v \] \hspace{1cm} (3.89)

where \( i \) is the index of the vertical band and is computed by [121]:

\[ a: i = \left[ 1, n \left( \frac{u}{LW} \right) \right] \]  \hspace{1cm} (3.90)
\[ b: i = \left[ n \left( \frac{LW - u}{LW} \right), n \right] \]  \hspace{1cm} (3.91)
\[ c: i = \left[ n + 1, n + 1 + n \left( \frac{u}{LW} \right) \right] \]  \hspace{1cm} (3.92)
\[ d: i = \left[ n + 1 + n \left( \frac{LW - u}{LW} \right), 2n \right] \]  \hspace{1cm} (3.93)

where \( n \) is the number of stripes the mirror surface is divided into. For our analysis, we decided to divide the mirror surface into 20 stripes.

Finally, the fraction of the mirror free of overlapping is [121]:

\[ f_{s&b} = \frac{\sum_{i=1}^{2n} \max(F(i))}{nLH} \] \hspace{1cm} (3.94)
It should be noted that a band on the heliostat surface is only considered to be affected by overlapping as long as its full width is covered by the projection. Previously, instantaneous field efficiency was thoroughly explained and the required mathematical formulations derived. Annual heliostat field thermal performance can be evaluated against two different scales. Initially, the yearly insolation weighted efficiency is expressed by [118]:

\[
\eta_{\text{ann},w} = \frac{\sum_{i=1}^{365} \int_{\text{sunrise}}^{\text{sunset}} DNI(t) \eta_{\text{opt},f}(t) \, dt}{\sum_{i=1}^{365} \int_{\text{sunrise}}^{\text{sunset}} DNI(t) \, dt}
\]  

(3.95)

where \(\eta_{\text{ann},w}\) is the annual weighted field efficiency, and \(DNI(t)\) is the instantaneous direct normal radiation in kW/m\(^2\).

Another approach for the field layout thermal assessment is the unweighted annual efficiency which is computed such that [118]:

\[
\eta_{\text{ann},uw} = \frac{\sum_{i=1}^{365} \int_{\text{sunrise}}^{\text{sunset}} \eta_{\text{opt},f}(t) \, dt}{\sum_{i=1}^{365} \int_{\text{sunrise}}^{\text{sunset}} dt}
\]  

(3.96)

where \(\eta_{\text{ann},uw}\) is the field annual unweighted efficiency.

Typical meteorological year (TMY) of time instants sampled of direct normal radiation for Abu Dhabi as an example is utilized for analyses of this research work [123]. To reduce the computational load of the optimization process, it was decided to take into account only the direct solar insolation of the 21\(^{st}\) of each month. In conclusion, each instantaneous optical efficiency factor (cosine, attenuation, spillage, shading and blocking) for every single heliostat within the field throughout a sunny day hours must be calculated to determine the monthly performance of the field. After calculating the monthly averaged optical efficiency throughout the year, the annual performance of the field can be calculated. This method will reduce the number of instances required for calculating the field efficiency considerably (approximately by 1/30) which will speed up the optimization process for good.

3.1.11. Central tower.

A solar receiver is located on top of the central tower within the heliostat field. Note that the power cycle is situated on the ground, implying that the compressed air has to be piped up and down the tower. Based on the studies conducted on hybrid solar
gas turbine cycles employing heliostat field collectors [25, 122], a concentric tube ducting arrangement is considered for the analysis of this research work. This type of piping arrangement minimizes the sources of heat loss to the atmosphere as the cold compressed air passes through the outer annulus while the hot air stream goes through the inner annulus.

The concentric piping arrangement for the central tower of the heliostat field collector design is depicted in Figure 33. Another significant advantage of this piping arrangement is its ability of cooling the hot inner pipe with the cold compressed air passing through the outer annulus [122]. Moreover, a stationary air-gap between the hot and cold air streams is considered based on Spelling’s recommendation. This stagnant air minimizes the heat transfer between the air streams without integrating any expensive high temperature insulation within the piping system of the central tower.

![Figure 33: Concentric piping arrangement for the central tower of the heliostat field collector design [122]](image)

It is necessary to determine the overall heat transfer coefficient between the hot and cold air streams in the solar tower to calculate the air temperature entering the receiver and the combustion chamber. There is no heat loss to the atmosphere as the outer annulus surface is considered to be perfectly insulated. Furthermore, hot and cold air streams are completely separated (no mixing). Spelling [122] proposed modeling the piping arrangement as a counter flow heat exchanger.
Total heat transfer surface area is calculated by:

\[ A = 2\pi r_{int} h_T \]  

(3.97)

where \( A \) is the total heat transfer surface area in m\(^2\), \( r_{int} \) is the inner annulus internal radius in m, and \( h_T \) is the central tower height in m. Based on Figure 33, the overall heat transfer coefficient is computed by:

\[
\frac{1}{U} = \frac{1}{h_h} + \frac{r_{int}}{h_c r_{ext}} + \sum_{n=1}^{N} \frac{r_{int}}{k_n} \ln \frac{r_n}{r_{n-1}}
\]  

(3.98)

where \( U \) is the overall heat transfer coefficient in W/m\(^2\)K, \( h_h \) and \( h_c \) are the hot and cold air streams’ convective heat transfer coefficient W/m\(^2\)K, \( r_{ext} \) is the inner annulus external radius in m, \( r_n \) is the radius of the \( n^{th} \) layer of insulation in m, \( k_n \) is the thermal conductivity of the \( n^{th} \) layer of insulation in W/mK, and \( N \) is the number of insulation layers.

Petukhov’s correlation for the Nusselt number in turbulent flow with 6% accuracy is chosen for computing the convective heat transfer coefficient as follows [131]:

\[
Nu_D = \frac{(\frac{f}{8}) Re_D Pr}{1.07 + 12.7 \sqrt{(\frac{f}{8})(Pr^2 - 1)}}
\]  

(3.99)

where \( Nu_D, Re_D \) and \( Pr \) are the Nusselt, Reynolds, and Prandtl numbers, and \( f \) is the friction factor, bearing in mind that the presented equation is only valid within the ranges of \( 0.5 < Pr < 200 \) and \( 10^4 < Re_D < 5 \times 10^6 \). Finally, the convective heat transfer coefficient is determined by:

\[
Re_D = \frac{\rho u D}{\mu}
\]  

(3.100)

\[
Pr = \frac{\mu C_p}{k}
\]  

(3.101)

\[
f = \frac{1}{(1.82 \log(Re_D) - 1.64)^2}
\]  

(3.102)
\[ h = \frac{N_u D k}{D} \]  

(3.103)

where \( \rho \) is the fluid density in kg/m\(^3\), \( u \) is the air velocity in the pipes in m/s, \( D \) is the pipe (hydraulic) diameter in m, and \( \mu \) is the fluid dynamic viscosity in Pa.s. It should be noted that the air velocity is assumed to be 10 m/s. Determining the overall heat transfer coefficient enables us to estimate the outlet air temperature by the Effectiveness-NTU technique as follows [132]:

\[ NTU = \frac{UA}{m_a c_p} \]  

(3.104)

\[ \varepsilon = \frac{NTU}{1 + NTU} \]  

(3.105)

\[ Q = \varepsilon m_a c_p (T_{o,rec} - T_{i,T}) \]  

(3.106)

where \( \varepsilon \) is the heat transfer effectiveness, and \( T_{o,rec} \) and \( T_{i,T} \) are the receiver outlet and tower inlet (compressor outlet) temperatures in K. Taking into consideration that calculating the heat loss between hot and cold air streams is an implicit problem, as the receiver outlet temperature depends on the air temperature entering the receiver, an iterative procedure must be implemented. The pressure drop across the central tower piping system (each side) is determined by [133]:

\[ \Delta P_T = \frac{f h_T \rho u^2}{D} \]  

(3.107)

where \( \Delta P_T \) is the turbulent pressure drop within the smooth tubes across each side (ascending or descending) of the central tower piping system.

**3.1.12 Receiver.**

The most challenging issue in commercializing hybrid solar gas turbine power plants is the design and operation of a high-temperature solar air receiver which is located at the top of the central tower [122]. Gas turbine cycles are operating at high temperatures. Therefore, the air receiver of hybrid and solar-only gas turbines must withstand high temperatures and radiant heat flux to achieve satisfactory efficiency and justify their utilization additional cost. Nevertheless, Spelling claims that the solar air receiver is the least standardized component of the power plant since most of the available alternatives are more of a prototype rather than a final design [122].
One of the foremost challenges in designing a solar air receiver is the air’s poor heat transfer characteristic even at high pressures. Considering that compressed air is utilized as the heat transfer fluid in the solar air heater within the solar tower, one of the more popular methods for improving the solar air heater is the implementation of volumetric material such as porous ceramic foams [122]. Another key factor in designing the solar air receiver is to seal the receiver, as the pressurized air must be separated from the ambient air. Consequently, the receiver surface must be covered with a window of quartz glass which increases the optical losses and reduces the convective losses from the receiver surface. Additionally, the air pressure drop within the receiver must be kept at a minimum to improve the hybrid cycle efficiency [122].

One of the most exciting and promising solar pressurized volumetric air receivers is the concept employed for the SOLGATE project. This receiver’s absorber material is silicon-silicon carbide while its aperture is covered by an elliptic dome quartz glass window [122]. The SOLGATE pressurized air receiver concept is depicted in Figure 34. This receiver can operate at high temperatures of about 960°C under peak flux of 800 kW/m² [122]. The thermal efficiency and pressure losses of 70% and 40 mbar are reported which demonstrate the potential this receiver has for direct integration within the hybrid gas turbine configurations [122]. It is important to note that the material operating temperature limit for the uncooled metal piping is 950°C which restricts the integrated solar receiver and tower piping system (solar subsystem) maximum allowable temperature. Consequently, maximum operating temperature in the solar subsystems is 950°C.

The pressurized volumetric air receiver is considered for this research work based on the concept developed as part of the SOLGATE project. Radiation losses are the only viable type of heat losses at high temperature in developing the mathematical formulation of the air receiver. Furthermore, the receiver is presumed as a grey-body with known absorptivity value. The overall air receiver efficiency is determined by [25]:

\[
\eta_{\text{rec}} = \eta_{\text{opt,rec}} - \frac{\sigma \varepsilon_{\text{rec}}}{I_{\text{rec}}} \left( \left( \frac{T_{i,\text{rec}} + T_{o,\text{rec}}}{2} \right)^4 - T_{\text{amb}}^4 \right)
\]  

(3.108)

where \( \eta_{\text{rec}} \) is the air receiver efficiency, \( \eta_{\text{opt,rec}} \) is the receiver optical efficiency, \( \varepsilon_{\text{rec}} \) is the emissivity of the volumetric absorber with its value assumed to be 0.85, \( \sigma \) is the
Stefan Boltzmann constant which is $5.674 \times 10^{-8} \text{ W/m}^2\text{K}^4$, $I_{\text{rec}}$ is the solar heat fluxing at the receiver in kW/m$^2$, $T_{i,\text{rec}}, T_{o,\text{rec}}$ and $T_{\text{amb}}$ are the receiver’s inlet, outlet and ambient air temperature in K.

Considering that high pressure air requires thicker glass resulting in lower optical efficiency, receiver optical efficiency is determined by [122]:

$$\log(\eta_{\text{opt,rec}}) = \frac{P_{i,\text{rec}}}{P_{\text{ref}}} \log(\eta_{\text{opt,rec}}^{\text{ref}})$$  \hspace{1cm} (3.109)

where $P_{i,\text{rec}}$ and $P_{\text{ref}}$ are the receiver inlet pressure and SOLGATE receiver reference pressure in kPa, and $\eta_{\text{opt,rec}}^{\text{ref}}$ is the SOLGATE receiver reference optical efficiency. Corresponding values for reference pressure and optical efficiency are considered to be 650 kPa and 87%, respectively [67, 134].

![Figure 34: SOLGATE pressurized volumetric air receiver concept [134]](image)

Pressure drop within the receiver is another significant factor in analysis of the pressurized volumetric air receiver. Air pressure drop within the receiver is estimated by [122]:

$$\Delta P_{\text{rec}} = \Delta P_{\text{rec}}^{\text{ref}} \left( \frac{\dot{M}_{a}^{\text{ref}}}{\dot{M}_a} \right) \left( \frac{P_{\text{ref}}}{P_{i,\text{rec}}} \right) \left[ \frac{T_{i,\text{rec}} + T_{o,\text{rec}}}{2T_{\text{ref}}} \right]$$  \hspace{1cm} (3.110)

where $\Delta P_{\text{rec}}$ is the pressure drop in the receiver in kPa, $\Delta P_{\text{rec}}^{\text{ref}}$ is the reference value of pressure drop in the volumetric receiver in kPa, $\dot{M}_a$ and $\dot{M}_a^{\text{ref}}$ are the actual and reference amounts of mass flux in the receiver in kg/m$^2$s, and $T_{\text{ref}}$ is the receiver reference temperature in K. Reference values for the pressure drop, mass flux, and
temperature are taken from Spelling [122, 134] to be 4 kPa, 1.063 kg/m².s, and 700°C, respectively.

### 3.1.13. Plant performance.

In order to connect topping and bottoming cycle thermodynamic analysis, a new parameter is defined as the ratio of the intake air mass flow rate at the bottoming cycle compressor over the intake air mass flow rate at the topping cycle compressor. This parameter will be referred to as mass flow rate ratio (MFRR) and is determined by:

\[
MFRR = \frac{\dot{m}_{aB}}{\dot{m}_{aT}}
\]  
(3.111)

where \(\dot{m}_{aT}\) and \(\dot{m}_{aB}\) are the topping and bottoming cycles mass flow rates in kg/s.

To properly analyze the effect of superheated steam temperature on the SBC power plant’s performance, a new control variable is outlined as the HRSG degree of superheating (DOSH) such that steam outlet temperature leaving the HRSG is determined by:

\[
DOSH = \frac{T_{s,sh} - T_{s,sat}}{(T_{g,ex} - \Delta T_{pinch,sh}) - T_{s,sat}}
\]  
(3.112)

where \(T_{s,sh}\) and \(T_{s,sat}\) are the superheated and saturated steam temperatures in K, \(T_{g,ex}\) is the topping cycle turbine exhaust gases temperature in K, \(\Delta T_{pinch,sh}\) is the superheater pinch temperature in K, and DOSH is the HRSG degree of superheating.

Net power output and efficiency of the plant can be calculated:

\[
W_{net,ABC} = \left[ \eta_{M,G}(\dot{m}_g w_{tT} + \dot{m}_{aB} w_{TB}) - \frac{\left(\dot{m}_{aT} w_{CT} + \dot{m}_{aB} w_{CB}\right)}{\eta_{M,G}} \right] \eta_{ele,G}
\]  
(3.113)

\[
W_{net,SBC} = \left[ \eta_{M,G}(\dot{m}_g w_{tT} + \dot{m}_s w_{TB}) - \frac{\left(\dot{m}_{aT} w_{CT} + \dot{m}_s w_p\right)}{\eta_{M,G}} \right] \eta_{ele,G}
\]  
(3.114)

\[
W_{net,HABC} = \left[ \eta_{M,G}(\dot{m}_g w_{tT} + (\dot{m}_{aB} + \dot{m}_w) w_{TB}) - \frac{\left(\dot{m}_{aT} w_{CT} + \dot{m}_{aB} w_{CB} + \dot{m}_w w_p\right)}{\eta_{M,G}} \right] \eta_{ele,G}
\]  
(3.115)
\[ W_{\text{net, MBC}} = \eta_{M,G} \left( m_g w_{tT} + (m_{aB} + m_w) w_{tB} \right) \]
\[ - \left( \frac{m_{aT} w_{cT} + m_{aB} w_{cB} + m_w w_p}{\eta_{M,G}} \right) \eta_{\text{ele,G}} \]  

(3.116)

\[ \eta = \frac{W_{\text{net}}}{m_f LHV + \dot{Q}_{\text{sol}}} \]  

(3.117)

where \( w_{tT} \) and \( w_{cT} \) are the topping cycle specific works of turbine and compressor respectively in kJ/kg, similarly, \( w_{tB} \) and \( w_{cB} \) are the bottoming cycle specific works of turbine and compressor respectively in kJ/kg, \( \eta_{\text{ele,G}} \) and \( \eta_{M,G} \) are the generator electrical and mechanical efficiencies, \( \dot{Q}_{\text{sol}} \) is the rate of solar thermal input in kW, \( W_{\text{net}} \) is the total net power output in kW, and \( \eta \) is the plant overall efficiency.

3.2. Economic Model

Economic analysis provides a better understanding of the best system configuration, noting that the most efficient power configuration is not necessarily the most cost effective candidate. Thus, cost assessment and economic analysis were implemented. In this section, detailed mathematical formulation for economic analysis of every component of different hybrid combined cycles is presented. Bearing in mind that the economic models are derived from different sources with different publication dates, the chemical engineering plant cost index (CEPCI) is implemented to enhance the accuracy of our economic model. Each component capital investment cost is multiplied by a correction factor which is determined by:

\[ f_{\text{CEPCI}} = \frac{CEPCI_{\text{ref}}}{CEPCI_{2015}} \]  

(3.118)

where \( CEPCI_{\text{ref}} \) and \( CEPCI_{2015} \) are the reference and 2015 CEPCI values, and \( f_{\text{CEPCI}} \) is the cost correction factor.

3.2.1. Compressor.

The economic model for the air compressors implemented in the gas turbine topping cycle and waste heat recovery bottoming cycle is [135, 136] (reference year 1994):

\[ Z_c = \left( \frac{39.5m_{a,c} r_c}{0.9 - \eta_c} \right) \ln(r_c) \]  

(3.119)
where \( m_{a,c} \) is the compressor air mass flow rate in kg/s, and \( Z_c \) is the compressor capital investment cost in US$.

### 3.2.2. Turbine.

The cost of the gas turbines in the proposed configurations is estimated using the cost function by Knopf [135, 136] as follows (reference year 1994):

\[
Z_{Gt} = \left( \frac{266.3 m_t}{0.92 - \eta_t} \right) \ln \left( \frac{1}{\eta_t} \right) \left[ 1 + \exp \left( 0.036 T_{t,i} - 54.4 \right) \right]
\]

(3.120)

where \( m_t \) is the fluid (exhaust gas, air, humid air) mass flow rate passing through the turbine in kg/s, and \( Z_{Gt} \) is the gas turbine capital investment cost in US$.

For the steam turbine, its required capital investment cost is estimated employing the cost model presented by Frangopoulos [137] such that (reference year 1991):

\[
Z_{St} = 150 \left( m_s w_{St} \right) \left( 1 + \exp \left( 0.096 (T_{t,i} - 866) \right) \right) \left( \frac{50000}{m_s w_{St}} \right)^{0.67}
\]

(3.121)

where \( Z_{St} \) is the steam turbine capital investment cost in US$.

### 3.2.3. Combustion chamber.

In order to study the economical aspect of the combustion chamber, its capital investment cost is calculated by [135, 136] (reference year 1994):

\[
Z_{cc} = \left( \frac{25.6 m_{aT}}{0.995 \frac{P_{o,cc}}{P_{i,cc}}} \right) \left[ 1 + exp \left( 0.018 T_{o,cc} - 26.4 \right) \right]
\]

(3.122)

where \( P_{i,cc} \) and \( P_{o,cc} \) are the air pressure at the combustion chamber entry and exhaust gases pressure at the combustion chamber exit in kPa, and \( Z_{cc} \) is the combustion chamber capital investment cost in US$.

### 3.2.4. Air heat exchanger.

Likewise, the cost function for an air to air heat exchanger is taken from Turton [138] such that (reference year 1998):

\[
Z_{AHX} = 1.53 + 1.27 \left( \frac{T_{h,i}}{623} \right)^{2.4} (10)^{3.8528 + 0.42421 \log \left( \frac{\dot{Q}_{AHX}}{0.018 LMTD_{AHX}} \right)}
\]

(3.123)
where $\dot{Q}_{AHX}$ is the rate of heat exchange between the hot and cold fluids within the heat exchanger in kW, $LMTD_{AHX}$ is the log mean temperature difference in the heat exchanger in K, $T_{h,i}$ is the hot fluid inlet temperature in K, and $Z_{AHX}$ is the heat exchanger capital investment cost in US$. We must take into account that the presented cost model is associated with the capital investment cost estimation of a plate heat exchanger.

### 3.2.5. Heat recovery steam generator.

Cost function for the HRSG is divided into three main sections of economizer, evaporator, and superheater as follows [139] (reference year 1991):

$$
Z_{HRSG} = 3650 f_p \left[ f_{g,ec} f_{s,ec} \left( \frac{\dot{Q}_{ec}}{LMTD_{ec}} \right)^{0.8} + f_{g,ev} f_{s,ev} \left( \frac{\dot{Q}_{ev}}{LMTD_{ev}} \right)^{0.8} 
+ f_{g,sh} f_{s,sh} \left( \frac{\dot{Q}_{sh}}{LMTD_{sh}} \right)^{0.8} \right] + 11820 f_p \dot{m}_s + 658 \dot{m}_g^{1.2}
$$

(3.124)

where $\dot{Q}_{ec}$, $\dot{Q}_{ev}$, and $\dot{Q}_{sh}$ are the rate of heat exchanges between the exhaust gases and steam (water) for the economizer, evaporator and superheater in kW, respectively, $LMTD_{ec}$, $LMTD_{ev}$, $LMTD_{sh}$ are the log mean temperature differences in the economizer, evaporator, and superheater in K, respectively, $Z_{HRSG}$ is the HRSG capital investment cost in US$, $F_p$ is the heat exchanger pressure cost correction factor, $f_{g,ec}$, $f_{g,ev}$, $f_{g,sh}$ are the gas side temperature cost correction factor for the economizer, evaporator, and superheater, and $f_{s,ec}$, $f_{s,ev}$, $f_{s,sh}$ are the steam side temperature cost correction factor for the economizer, evaporator, and superheater, respectively. The aforementioned cost correction factors are determined by [137, 139-141]:

$$
f_p = \frac{0.0971 P_s}{3000} + 0.9029
$$

(3.125)

$$
f_g = 1 + \exp \left( \frac{T_{o,g} - 990}{500} \right)
$$

(3.126)

$$
f_s = 1 + \exp \left( \frac{T_{o,s} - 830}{500} \right)
$$

(3.127)

where $P_s$ is the steam pressure in kPa, $T_{o,g}$ is the outlet gas temperature in K, and $T_{o,s}$ is the steam (water) outlet temperature in K.
3.2.6. Pump.

Pump cost function is obtained from the model presented by Frangopoulos [137] as follows (reference year 1991):

\[
Z_p = 442\left[\dot{m}_w w_p\right]^{0.71} 1.41 \left[1 + \left(\frac{1 - 0.8}{1 - \eta_p}\right)\right] \tag{3.128}
\]

where \(Z_p\) is the pump capital investment cost in US$.

3.2.7. Condenser.

Capital investment cost for the condenser is divided into four main components including the condenser, air cooler, pump, and fans [139]:

\[
Z_{con} = Z_{STcon} + Z_{AC} + Z_p + Z_{fan} \tag{3.129}
\]

where \(Z_{con}\) is the condenser total capital investment cost in US$, \(Z_{STcon}\) is the steam condenser investment capital cost in US$, \(Z_{AC}\) is the air cooler capital investment cost in US$, \(Z_p\) and \(Z_{fan}\) are the pump and fans’ capital investment costs in US$. Steam condenser capital investment cost is evaluated by the model presented by Frangopoulos [137] as follows (reference year 1991):

\[
Z_{STcon} = 248\left(\frac{\dot{Q}_{con}}{2.2 \text{ LMTD}_{con}}\right) + 659\left(\frac{\dot{Q}_{con}}{4.185 \times 11.5}\right) \tag{3.130}
\]

where \(\dot{Q}_{con}\) is the rate of heat transfer in the condenser in kW, and \(\text{LMTD}_{con}\) is the log mean temperature difference in the condenser in K. Furthermore, capital investment cost for the air cooler is evaluated by the model presented by Spelling [122] (reference year 1998):

\[
Z_{AC} = 1.53 + 1.27(3)(10)^{3.6418+0.4053\log\left(\frac{\dot{Q}_{AC}}{0.3 \text{ LMTD}_{AC}}\right)} \tag{3.131}
\]

where \(\dot{Q}_{AC}\) is the rate of heat exchange between the hot and cold fluids within the air cooler in kW, and \(\text{LMTD}_{AC}\) is the log mean temperature difference in the air cooler in K. Furthermore, the required fans’ capital investment cost is determined by Boehm [117] (reference year 1987):

\[
Z_{fan} = 1500N_{\text{fan}} \left(\frac{\dot{V}_{\text{fan}}}{10}\right)^{0.36} \tag{3.132}
\]
where $\dot{V}_{fan}$ is the fan volumetric air flow rate in $m^3/s$, and $N_{fan}$ is the number of fans required for the air cooler operation. It should be noted that the cost of the pump is already presented in the previous section.

### 3.2.8. Air saturator.

At the time of writing, there is no actual information available regarding the possible capital investment cost for an air saturator. A preliminary economic analysis was performed by the author in [22] which is selected for economic investigation of the air saturator in this study. The air saturator’s upper and lower sections are considered as two separate heat exchangers for economic assessments. An economic model for a gas to gas heat exchanger with overall heat transfer coefficient of 0.018 kW/m²K was selected [135]. Consequently, the cost function for a shell and tube heat exchanger adopted from Knopf [135] is employed to calculate the cost of the air saturator as follows (reference year 1994):

$$ Z_{AS} = 2290 \left( \frac{\dot{Q}_{AS,L}}{0.018 \ LMTD_{AS,L}} \right)^{0.6} + 2290 \left( \frac{\dot{Q}_{AS,U}}{0.018 \ LMTD_{AS,U}} \right)^{0.6} $$  \hspace{1cm} (3.133)

where $\dot{Q}_{AS,L}$ and $\dot{Q}_{AS,U}$ are the lower and upper sections of the air saturator rate of heat exchange in kW, $LMTD_{AS,L}$ and $LMTD_{AS,U}$ are the log mean temperature difference for the lower and upper sections of the air saturator in K, and $Z_{AS}$ is the air saturator capital investment cost in US$.

### 3.2.9. Evaporator.

The evaporator economic model is taken from the model presented by Desideri and Di Maria [142] as follows (reference year 1999):

$$ Z_{ev} = 23.5 \dot{Q}_{ev} $$  \hspace{1cm} (3.134)

Where $\dot{Q}_{ev}$ is the evaporator rate of heat exchange in kW, and $Z_{ev}$ is the evaporator capital investment cost in US$.

### 3.2.10. Heliostat field.

The heliostat field initial investment consists of three major factors, i.e. land cost, mirror cos, and wiring cost for the heliostat control. The economic model of the heliostat field is adopted from the model available in the DELSOL3 simulation program.
accomplished by Kistler [106]. The land and heliostat costs are estimated by [122] (reference year 1986):

\[ Z_{\text{land}} = 0.62(1.5A_{\text{land}} + 1.8 \times 10^5) \]  

(3.135)

\[ Z_{\text{mirror}} = 126A_{\text{hel}}N_{\text{hel}} \]  

(3.136)

where \( A_{\text{land}} \) is the land area required for the heliostat field in \( m^2 \), \( A_{\text{hel}} \) is the surface area of each heliostat mirror in \( m^2 \), \( N_{\text{hel}} \) is the total number of heliostats in the field, \( Z_{\text{land}} \), and \( Z_{\text{mirror}} \) are the capital investment costs of the land and mirrors in US$, respectively. Cost of the wiring is more complicated since two types of wiring are employed within the heliostat field. Initially, the field is divided into several cells, each containing a number of heliostats. All cells located far from the central tower need more primary cables whereas more expanded cells require longer secondary cables [25].

Taking into account that the economic model adopted from the DELSOL3 user manual where the cell wise method is utilized for optimizing the heliostat field performance, nonetheless, a different method is considered for heliostat field optimization in this study. Thus, the field must be divided into several cells. For this study, we considered dividing the field into 100 cells, noting that all the cells within the heliostat field have equal area whereas their heliostat densities are reduced by getting further away from the central tower. Heliostat density for each cell is calculated by [25]:

\[ \rho_{\text{cell},i} = \frac{A_{\text{hel}}N_{\text{hel,cell},i}}{A_{\text{cell},i}} \]  

(3.137)

where \( A_{\text{cell}} \) is the cell area in \( m^2 \), \( N_{\text{hel,cell},i} \) is the total number of heliostats within a cell, and \( \rho_{\text{cell},i} \) is the cell density, remembering that reducing the cell area leads to a greater number of cells within the field and longer computational time. The wiring cost is calculated by [106] (reference year 1986):

\[ Z_{\text{wire}} = \sum_{i=1}^{N_{\text{cell}}} N_{\text{hel,cell},i} \left[ 0.031r_{\text{cell},i} + 24 \sqrt{\frac{A_{\text{hel}}}{\rho_{\text{cell},i}}} \right] \]  

(3.138)

where \( N_{\text{cell}} \) is the total number of cells within the field (100 for this study), \( r_{\text{cell},i} \) is the radial distance between the center of the cell and the central tower in m, and \( Z_{\text{wire}} \) is the capital investment cost for wiring between the heliostats in US$. Finally, the total
capital investment cost for the heliostat field is calculated by [106] (reference year 1986):

\[
Z_{\text{helf}} = Z_{\text{land}} + Z_{\text{mirror}} + Z_{\text{wire}}
\]

(3.139)

where \(Z_{\text{helf}}\) is the heliostat field capital investment cost in US$.

### 3.2.11. Central tower.

The central tower total capital investment consists of the tower and piping costs. Based on the study accomplished by Sterns Roger Engineering [143], it is concluded that steel towers are the most economical alternative for tower heights less than 120m while concrete towers are more cost-effective for central tower heights greater than 120m. Thus, the cost function for the central tower is [106] (reference year 1986):

\[
Z_{\text{tow}} = \begin{cases} 
1.09025 \times 10^6 \exp(0.00879h_T) & h_T < 120 \\
0.78232 \times 10^6 \exp(0.01130h_T) & h_T \geq 120 
\end{cases}
\]

(3.140)

where \(Z_{\text{tow}}\) is the capital investment cost of the tower in US$. The cost function for the tower piping is [25, 122, 144] (reference year 2010):

\[
Z_{\text{piping}} = \left[3600 \frac{r_{\text{outer}}}{1.31} + 420 \frac{r_{\text{int}}}{0.87}\right]h_T + 90000 \frac{r_{\text{int}}}{0.87}
\]

(3.141)

where \(r_{\text{outer}}\) and \(r_{\text{int}}\) are the inner and outer diameters of the concentric piping in m as depicted in Figure 33, and \(Z_{\text{piping}}\) is the central tower piping capital investment cost in US$. Total capital investment cost for the central tower is estimated by:

\[
Z_T = Z_{\text{tow}} + Z_{\text{piping}}
\]

(3.142)

where \(Z_T\) is the total capital investment cost for the central tower in US$.

### 3.2.12. Receiver.

The receiver economic model is adopted from the work done by Spelling [122]. It is stated that the solar receiver economic analysis is subject to significant uncertainty because of the limited information available concerning high temperature air solar receivers’ purchasing cost [122]. Moreover, Spelling predicts that the cost of the solar receiver will experience a considerable abatement in the near future as mass production begins [122, 145]. Based on the receiver cost, the economic model for the receiver which is presented by Schwarzbözl et al. [68] is (reference year 1996):
\[ Z_{\text{rec}} = A_{\text{rec}}[79T_{o,\text{rec}} - 42000] \]  

(3.143)

where \( T_{o,\text{rec}} \) is the receiver nominal outlet temperature in K, \( A_{\text{rec}} \) is the receiver surface area in m\(^2\), and \( Z_{\text{rec}} \) is the receiver capital investment cost in US$. Note that the minimum cost of the receiver is set to be 23,500 US$/m\(^2\). Therefore, in case the calculated cost of the receiver from the aforementioned equation is less than the assigned lower boundary, the cost of the receiver as determined by [68] is (reference year 1996):

\[ Z_{\text{rec}} = 23500A_{\text{rec}} \]  

(3.144)

3.2.13. Generator.

The economic model for the generator is adopted from the cost function presented by Frangopoulos [137] as follows (reference year 1991):

\[ Z_G = 4 \times 10^6 \left( \frac{\dot{W}_{\text{net}}}{160} \right)^{0.7} + Z_{G,\text{aux}} \]  

(3.145)

where \( \dot{W}_{\text{net}} \) is the cycle electrical power output in MWe, \( Z_G \) is the generator capital investment cost in US$, and \( Z_{G,\text{aux}} \) is the cost of auxiliary electrical equipment, control systems, and transformers in US$ which is estimated by [137] (reference year 1991):

\[ Z_{G,\text{aux}} = 10^7 \left( \frac{\dot{W}_{\text{net}}}{120} \right)^{0.65} \]  

(3.146)


The turbine economic model presented in previous sections does not include the auxiliary equipment cost such as fuel pumps and lubrication systems. The gas turbine auxiliary equipment cost is estimated by the cost function presented by Frangopoulos [137] as follows (reference year 1991):

\[ Z_{Gt,\text{aux}} = 4 \times 10^6 \left( \frac{\dot{W}_{\text{net}}}{160} \right)^{0.7} \]  

(3.147)

where \( Z_{Gt,\text{aux}} \) is the gas turbine auxiliaries’ capital investment cost in US$. The steam turbine auxiliary equipment cost is estimated by [122] (reference year 1991):
\[ Z_{St,aux} = 10^7 \left( \frac{W_{net}}{75} \right)^{0.7} \]  

(3.148)

where \( Z_{St,aux} \) is the steam turbine auxiliaries’ capital investment cost in US$.

### 3.2.15. Water treatment facility.

The required investment for water treatment facilities is attained from a report by the National Renewable Energy Laboratory [122, 146] as follows (reference year 2010):

\[ Z_{wtr} = 2.03 \times 10^6 \left( \frac{W_{net}}{110} \right)^{0.8} \]  

(3.149)

where \( Z_{wtr} \) is the steam turbine cycle water treatment facilities’ capital investment in US$.

### 3.2.16. Civil engineer.

The capital investment needed for a new site infrastructure and buildings for SBC configuration is estimated based on the work by Pelster [139] (reference year 1995):

\[ Z_{Civil,SBC} = 66.9 \times 10^6 \left( \frac{W_{net,SBC}}{144} \right)^{0.8} \]  

(3.150)

where \( Z_{Civil,SBC} \) is the capital investment cost for the civil engineering of the SBC configuration in US$. For the other proposed configurations, there is no available study in the literature concerning the required investment for civil engineering of the plant. For a simple gas turbine, Pelster [139] has provided a cost function to estimate the civil engineering capital investment. It was decided to consider the other investigated configurations as two gas turbine cycles to estimate the required investment for the site infrastructure and buildings such that [139] (reference year 1995):

\[ Z_{Civil} = 7.6 \times 10^6 \left( \frac{W_{net,T}}{52.8} \right)^{0.8} + 7.6 \times 10^6 \left( \frac{W_{net,B}}{52.8} \right)^{0.8} \]  

(3.151)

where \( W_{net,T} \) and \( W_{net,B} \) are the topping and bottoming cycles’ net electrical output in MWe, and \( Z_{Civil} \) is the civil engineering capital investment cost for ABC, HABC, and MBC configurations.
3.2.17. Equipment installation.

Peters and Timmerhaus [147] stated that the installation cost can be estimated as 20% of the capital investment cost of the equipment. Consequently, the installation cost is:

$$Z_{ins} = 0.2 \sum Z_{eqp}$$ (3.152)

where $Z_{eqp}$ is the equipment initial purchasing cost mentioned in the aforementioned sections in US$, and $Z_{ins}$ is the installation capital investment cost in US$.

3.2.18. Natural gas substation.

Natural gas is required to be delivered to the power plant by a new branch of the pipeline which connects the plant to an existing high pressure gas network [122]. Moreover, delivered natural gas pressure must be reduced to the combustion chamber pressure implying that a pressure reduction station is necessary for the power plant. Consequently, the total investment required for the natural gas delivery to the combustion chamber is [139]:

$$Z_{NGS} = Z_{brs} + Z_{prs} + Z_{bpipe}$$ (3.153)

where $Z_{brs}$, $Z_{bpipe}$, and $Z_{prs}$ are the capital investment costs for the branching station, branching pipeline, and pressure reduction substation in US$, and $Z_{NGS}$ is the total capital investment cost for the natural gas branching in US$. Pipeline cost represents an insignificant proportion of the total cost and is not considered in the economic analysis of this study [25, 122, 139]. The station branching cost is estimated by [139] (reference year 1995):

$$Z_{brs} = 219000 \left( \frac{m_f}{14.4} \right)^{0.7} + 221000$$ (3.154)

Pressure reduction station required investment, which comprises a single stage pressure reduction unit, a measuring station, and a “pig” arrival station, is given by Pelster [139] (reference year 1998):

$$Z_{prs} = 1.27 \times 10^6 \left( \frac{m_f}{14.4} \right)^{0.7}$$ (3.155)
3.2.19. Project engineering and contingencies.

There are other indirect factors which contribute in power plant economic analysis during planning and construction stages. These indirect cost factors can be estimated by [122, 148, 149]:

\[ Z_{if} = 0.05 \left( \sum Z_{eqp} + Z_{ins} + Z_{NGS} + Z_{Civil} \right) \]  \hspace{1cm} (3.156)

where \( Z_{if} \) is the capital investment cost for the abovementioned indirect factors such as planning and management of the construction and permitting in US$. Occurrence of unpredicted technical and regularity problems within the construction process might result in additional investment. Additional expenses associated with the abovementioned factors can be calculated by [122, 148, 149]:

\[ Z_{cont} = 0.1 \left( \sum Z_{eqp} + Z_{ins} + Z_{NGS} + Z_{Civil} \right) \]  \hspace{1cm} (3.157)

Where \( Z_{cont} \) is additional funding required for unforeseen technical and regulatory problems in US$.

3.2.20. Decommissioning.

Power plants will be dismantled at the end of their lifetime to return the site to its condition prior to the power plant installation. Spelling stated that natural gas and solar power plants do not cause any permanent damage to the local environment and decommissioning costs can be calculated by [122]:

\[ Z_{Dec} = 0.05 \left( \sum Z_{eqp} + Z_{ins} + Z_{NGS} + Z_{Civil} \right) \]  \hspace{1cm} (3.158)

where \( Z_{Dec} \) is the capital investment cost for decommissioning in US$.

3.2.21. Operating cost.

Operating cost of the plant is affected by three factors as follows:

\[ Z_{opt} = Z_f + Z_w + Z_{CO_2} \]  \hspace{1cm} (3.159)

where \( Z_w \) and \( Z_f \) are water and fuel consumption capital investment cost in US$, \( Z_{CO_2} \) is the cost penalty for CO\(_2\) emission in US$, and \( Z_{opt} \) is the operation cost in US$. The fuel consumption cost is calculated based on the presented natural gas cost per unit of energy as follows [150]:

123
\[ Z_f = 2.53 \times 10^{-6} (\dot{m}_f \text{ LHV}) \quad (3.160) \]

It should be noted that the water consumption cost required a more detailed calculation. One of the main sources of water consumption which is common between all four proposed configurations is the water utilized for mirror washing. Based on the results presented in the US Department of Energy report [151] and Spelling’s investigation [122], annual water consumption associated with mirror washing is determined by:

\[ V_{w,mw} = 0.05A_{hel}N_{hel} \quad (3.161) \]

where \( V_{w,mw} \) is the annual water consumption for heliostat field mirror washing in \( m^3 \).

Another source of water consumption in power plants is water usage for compressor washing. Spelling recommends daily compressor washing prior to start up considering that the UAE’s atmosphere contains a considerable amount of dust. The annual water consumption for compressor washing is estimated by [152]:

\[ V_{w,cw} = 365[0.09 + 0.0005\dot{m}_{a,c}] \quad (3.162) \]

where \( V_{w,cw} \) is the annual water consumption for compressor washing in \( m^3 \).

The aforementioned sources of water consumption are common between all four plant configurations under investigation. Nevertheless, HABC and MBC’s amounts of water consumption are considerably greater than ABC’s water consumption since it is assumed that a new feed of water is constantly provided for the operation of the system. Therefore, the amount of water consumed in the air saturator of the MBC and evaporator of HABC configurations is added to the other sources of water consumption. For the SBC configuration, additional water consumption employed for steam-drum blowdown must be considered. Steam blowdown water consumption can represent up to 12% of the steam mass flow rate in the bottoming cycle [153] while Spelling has considered 3% as an appropriate value [122, 154].

Additionally, it must be taken into account that water consumption can be divided into two categories based on the quality of the required water. Water consumed for mirror and compressor washing as well as the water employed for the condenser operation does not necessarily need to be processed. On the other hand, water consumption associated with the blowdown steam and water injection in HABC and
MBC configurations need to have high quality. Additionally, water employed in the proposed configurations can be categorized into water consumption and water circulation. Water circulation refers to the water which is consumed in a closed loop cycle such as the steam turbine or an open loop cycle with a condenser where water is recycled and reintroduced into the cycle. However, addition of a condenser (even an air cooled condenser) to HABC and MBC configurations is not economically justified considering the small amount of water required for their operations. Moreover, a water treatment facility will be required to process the recycled water in the condenser for reusing. Therefore, it is decided not to integrate a condenser in MBC and HABC configurations.

Gallo [99] classifies water consumption in power plants into three types based on their quality. The first type, as-received water, is the water used in the power plant without any processing except for coarse filtration [99]. This type of water is considered to be used for mirror and compressor washing as well as the condenser operation. Nonetheless, it is claimed by Turchi [146] that mirror efficiency can be ameliorated by utilizing water with high quality for mirror washing to achieve spotless reflectors. The second type, industrial water, only requires simplified processes of biological growth control, PH correction, as well as coarse filtrations [99]. However, industrial water quality is not sufficient for steam and gas turbine operation. The third type, polished demineralized water, is the high pressure boiler quality water which needs a high level of demineralization [99]. This type of water is necessary whenever steam is needed to be expanded in a turbine [155]. Consequently, water consumption associated with HABC and MBC air saturation as well as the blowdown water consumption in the SBC configurations and mirror and compressor washing are polished demineralized water. Nonetheless, a water treatment facility is only considered for the SBC configuration which implies that the demineralized water consumption unit cost is lower for the SBC configuration than other configurations.

The cost of water is 1 US$/m^3 for SBC power plants located in deserts [156]. For HABC, ABC, and MBC configurations, a water cost of 1.2 US$/m^3 is considered [122]. Reference year for all the presented water unit costs is 2006. Proper action must be taken into consideration to update the water consumption cost. Nevertheless, it is reported by Spelling [122] and Sandoz [25] that water consumption has an insignificant effect on the plant’s economic analysis.
Finally, the UAE’s officials’ concerns regarding the rate and effects of CO\textsubscript{2} emission is discussed. Additionally, the power generation contribution in CO\textsubscript{2} emission for Abu Dhabi is presented in detail. One of the more popular approaches of carbon dioxide emission abatement is to assign a cost penalty for CO\textsubscript{2} emission. There is no data available indicating the carbon dioxide emission tax for the UAE. Therefore, the CO\textsubscript{2} emission cost penalty of 40 US$/ton CO\textsubscript{2} presented by Sandoz [25] is considered for economic analysis of this study. The amount of CO\textsubscript{2} emission is determined based on the thermodynamic analysis presented for the combustion chamber by:

\[
\dot{m}_{CO_2} = \frac{44}{16} \dot{m}_f \tag{3.163}
\]

### 3.2.22 Maintenance cost.

It is common to estimate the annual repair and maintenance cost as a percentage of the initial investment. Therefore, the same approach is taken for this study for maintenance cost approximation. Civil engineering demands the lowest repair and maintenance and 1% of its initial capital investment is selected for the maintenance cost estimation [25]. Turbomachinery components’ maintenance costs are estimated as 2% of their initial investment costs. For the heliostat field mirrors, maintenance costs are considered to be 3% of their initial investments since this type of technology is at the intermediate development stage. Receiver and central tower maintenance costs are considered to be greater than the other components due to immaturity in their technology. Thus, receiver and central tower maintenance costs are 4% of their initial purchasing cost. Moreover, four sets of services are needed for the operation of the plant including field control system, ground keeping, mirror washing, and water treatment (for SBC only) [122]. The control system contract cost is reported to be 100,000 US$/year [122] (reference year 2010). Other costs of contract services are calculated as follows [122] (reference year 2010):

\[
Z_{cs,\text{gk}} = 10^5 \left( \frac{A_{hel}N_{hel}}{854000} \right)^{0.5} \tag{3.164}
\]

\[
Z_{cs,\text{mw}} = 3.5 \times 10^5 \left( \frac{A_{hel}N_{hel}}{854000} \right) \tag{3.165}
\]
\[ Z_{cs,wtr} = 1.45 \times 10^5 \left( \frac{\dot{W}_{net,SBC}}{110} \right) \] (3.166)

where \( Z_{cs,gk}, Z_{cs,mw}, Z_{cs,wtr} \) are the contract services capital investment for ground keeping, mirror washing, and water treatment in US$, respectively.

### 3.2.23. Labor cost.

To improve the accuracy of our economic model, necessary salaries for power plant employees must be taken into consideration. The number of employees and their corresponding salaries are taken from the study conducted by Spelling [122] which are tabulated in Table 7.

**Table 7: Labor rates and requirements [122]**

<table>
<thead>
<tr>
<th>Staff</th>
<th>Salary US$/year</th>
<th>Required Employees</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plant Manager</td>
<td>95000</td>
<td>0.5</td>
</tr>
<tr>
<td>Plant Engineer</td>
<td>92000</td>
<td>1</td>
</tr>
<tr>
<td>Maintenance supervisor</td>
<td>48000</td>
<td>2</td>
</tr>
<tr>
<td>Power block technician</td>
<td>40000</td>
<td>6</td>
</tr>
<tr>
<td>Solar field technician</td>
<td>40000</td>
<td>( N_{tec} )</td>
</tr>
<tr>
<td>Operation manager</td>
<td>84000</td>
<td>2</td>
</tr>
<tr>
<td>Control room operator</td>
<td>40000</td>
<td>( N_{opr} )</td>
</tr>
</tbody>
</table>

Number of solar field technicians is determined by [122]:

\[ N_{tec} = 1 + 3 \left( \frac{A_{het}N_{het}}{100000} \right) \] (3.167)

Additionally, the number of control room operators is calculated by [122]:

\[ N_{opr} = 3 + 2 \left( \frac{A_{het}N_{het}}{100000} \right) \] (3.168)

Therefore, total labor cost is calculated such that [122]:

\[ Z_{lab} = 1.5 \sum_{staff} sal_{stf} N_{stf} \] (3.169)

where \( sal_{stf} \) is the annual salary of each plant staff in US$/year, \( N_{stf} \) is the necessary number of employees tabulated in Table 7, and \( Z_{lab} \) is the labor annual cost in US$/year.

All the required investment for plant construction as well as the necessary operating, maintenance, and decommissioning expenses were discussed in depth in previous sections providing necessary information to evaluate the plant’s performance economically. There are several plausible methods for economic evaluation of a power plant available in the literature. Each method is an indicator of a specific aspect of the plant’s economic performance. It is important to bear in mind that constructing a power plant requires a significant amount of investment which is commonly provided as loans. Consequently, there will be no income from the investment during the construction period. After the plant is fully constructed and equipped for operation, there will be a positive cash flow due to the generated electricity sale to the grid. Nonetheless, there will be a negative cash flow associated with annual operation and maintenance costs of the plant. At the end of the plant’s life time, decommissioning cost comes into the equation in the form of a negative cash flow whereas no positive cash flow exists.

One of the methods which is considered for economic analysis of this study is the commonly utilized and popular net present value (NPV) approach. This method evaluates the potential of an investment by determining the investment worth growth during the plant’s life time. Consequently, an investment is only advisable as long as its net present value is positive. Negative net present value implies that the required investment is greater than the positive cash flow during the plant’s life cycle. The net present value is determined by [122]:

\[
NPV = - \sum_{t=0}^{N_{\text{con}}-1} \frac{Z_{\text{inv}}}{N_{\text{con}}(1+i)^t} + \sum_{t=N_{\text{con}}}^{N_{\text{con}}+N_{\text{opt}}-1} \frac{C_{\text{ele}}W_{\text{net}} - (Z_{\text{opt}} + Z_{\text{mai}} + Z_{\text{lab}} + r_{\text{ins}}Z_{\text{inv}})}{(1+i)^t} \sum_{t=N_{\text{con}}+N_{\text{opt}}}^{N_{\text{con}}+N_{\text{opt}}+N_{\text{dec}}-1} \frac{Z_{\text{dec}}}{N_{\text{dec}}(1+i)^t}
\]

(3.170)

where \(NPV\) is the net present value, \(N_{\text{con}}\) is the number of years it takes to construct the power plant, \(i\) is the loan interest rate, \(N_{\text{opt}}\) is the number of years in which the plant is operational (plant life time cycle), \(C_{\text{ele}}\) is the local electricity sale price in US$/kWh, \(W_{\text{net}}\) is the annual electricity generated in kWh, \(Z_{\text{mai}}\) is the annual maintenance cost in
US$, $r_{\text{ins}}$ is the annual insurance rate which is pertaining to the cost of insuring the plant’s components, $N_{\text{dec}}$ is the number of years it takes to dismantle the plant and return it to the original condition, and $Z_{\text{inv}}$ is the initial investment required for constructing the plant in US$ which is calculated by:

\[
Z_{\text{inv},ABC} = Z_{c,T} + Z_{c,B} + Z_{GT,T} + Z_{GT,B} + Z_{cc} + Z_{AHX} + Z_{\text{hel}f} + Z_{T} + Z_{\text{rec}} \\
+ Z_{G,T} + Z_{G,B} + Z_{G,aux,T} + Z_{G,aux,B} + Z_{GT,aux,T} + Z_{GT,aux,B} + Z_{\text{Civil}} + Z_{\text{ins}} + Z_{NGS} + Z_{if} + Z_{\text{cont}}
\] (3.171)

\[
Z_{\text{inv},HABC} = Z_{c,T} + Z_{c,B} + Z_{GT,T} + Z_{GT,B} + Z_{cc} + Z_{ev} + Z_{AHX} + Z_{p} \\
+ Z_{\text{hel}f} + Z_{T} + Z_{\text{rec}} + Z_{G,T} + Z_{G,B} + Z_{G,aux,T} + Z_{G,aux,B} + Z_{GT,aux,T} + Z_{\text{Civil}} + Z_{\text{ins}} + Z_{NGS} + Z_{if} \\
+ Z_{\text{cont}}
\] (3.172)

\[
Z_{\text{inv},MBC} = Z_{c,T} + Z_{c,B} + Z_{GT,T} + Z_{GT,B} + Z_{cc} + Z_{AS} + Z_{\text{hel}f} + Z_{p} + Z_{T} \\
+ Z_{\text{rec}} + Z_{G,T} + Z_{G,B} + Z_{G,aux,T} + Z_{G,aux,B} + Z_{GT,aux,T} + Z_{\text{Civil}} + Z_{\text{ins}} + Z_{NGS} + Z_{if} + Z_{\text{cont}}
\] (3.173)

\[
Z_{\text{inv},SBC} = Z_{c,T} + Z_{p} + Z_{GT,T} + Z_{ST,B} + Z_{cc} + Z_{HRSG} + Z_{\text{con}} + Z_{\text{hel}f} + Z_{T} \\
+ Z_{\text{rec}} + Z_{G,T} + Z_{G,B} + Z_{G,aux,T} + Z_{G,aux,B} + Z_{GT,aux,T} + Z_{\text{Civil},SBC} + Z_{\text{ins}} + Z_{NGS} + Z_{if} \\
+ Z_{\text{cont}}
\] (3.174)

where $Z_{\text{inv},ABC}$, $Z_{\text{inv},HABC}$, $Z_{\text{inv},MBC}$, and $Z_{\text{inv},SBC}$ are the total initial investment costs for ABC, HABC, MBC and SBC configurations in US$, respectively.

Taking into account that solar thermal power plants, i.e. hybrid or solar-only configurations, are not economically competitive with fossil fuel power plants. Therefore, the net present value of the plant is expected to become negative with the current value of the electricity sale price in the market. Consequently, it was decided to consider another economic analysis method to further evaluate plant performance. A rather simple but informative economic indicator is payback time. For payback period calculation, the required additional investment is calculated and divided by the first year saving [27]. Thus, the payback time for the addition of the heliostat field to an already existing plant is investigated. Extra necessary investments associated with solar components of the plant are calculated. First year saving due to fuel consumption
reduction is determined. We take into account that the payback period can provide an approximate ranking of different investments which are comparable in terms of duration and functionality. Payback time is calculated as follows [27]:

\[ N_{pb} = \frac{Z_{sol,inv}}{Z_{1y,sav}} \]  \hspace{1cm} (3.175)

where \( N_{pb} \) is the payback period time, \( Z_{sol,inv} \) is the additional investment required for the solar component of the plant in US$, and \( Z_{1y,sav} \) is the first year saving due to the hybridization of the plant. Additional solar investment is determined as follows:

\[ Z_{sol,inv} = Z_{helf} + Z_{T} + Z_{rec} + [Z_{civit} + Z_{ins} + Z_{if} + Z_{cont}]_{sol} \]  \hspace{1cm} (3.176)

Bearing in mind that additional capital investment associated with the integration of the solar components for civil engineering, equipment installation, project engineering, and contingencies must be calculated. Hybridization of the plant does not require any additional funding in natural gas branching since fuel consumption is reduced. However, smaller natural gas branching is not considered since the hybridization of an already existing plant is under investigation. First year saving is estimated by:

\[ Z_{1y,sav} = C_{f,NG}(m_{f,sav} LHV) + C_{CO_{2}}m_{CO_{2},red} - C_{w}V_{w,mw} - Z_{cs,w} - Z_{cs,g} - Z_{inv,sol} \]  \hspace{1cm} (3.177)

where \( C_{NG} \) is the unit cost of natural gas in US$/kJ, \( C_{w} \) is the unit cost of water in US$/m$^3$, \( C_{CO_{2}} \) is the cost penalty for CO\(_2\) emission in US$/tonneCO_{2}$, \( m_{f,sav} \) is the annual mass of fuel saving in kg/year, and \( m_{CO_{2},red} \) is the annual mass of CO\(_2\) emission reduction in kg/year.

Another economic method considered for evaluation of this study is the model presented by Knopf [135]. The author presented an objective function to improve the thermodynamic optimization of cogeneration power plants. His proposed objective function is selected to further analyze the plant’s thermo-economic performance and enhance the optimization process. This method calculates the total cost rate including fuel consumption, equipment capital investment, and maintenance in US$/s. Additionally, this method will convert capital investment cost into an annual basis by
employing the capital recovery factor method, making it more convenient to comprehend and follow. Knopf’s recommended objective function is modified to better represent the situation in this research work as follows [135]:

\[
\dot{C}_{total} = C_{NG}(m_f LHV) + C_{CO_2}(m_{CO_2}) + C_w(m_w) \\
+ \sum Z_{eqp}(CRF)_{eqp}\phi_{eqp} + \left[ Z_{Dec} + Z_{cont} + Z_{lf} \right] CRF \tag{3.178}
\]

where \(\dot{C}_{total}\) is the total cost rate in US$/s, \(N_{eqp}\) is the number of pieces of equipment in the plant, \(Z_{eqp}\) is each equipment piece’s capital investment cost in US$, \((CRF)_{eqp}\) is each component capital recovery factor, \(\phi_{eqp}\) is each component maintenance and installation factor, and \(N\) is the plant hours of operation annually. The capital recovery factor is calculated by [135]:

\[
CRF = \frac{i[1 + i]^{N_{opt}}}{[1 + i]^{N_{opt}} - 1} \tag{3.179}
\]

In this method, fuel and equipment cost are being brought to their corresponding dollar per second annual basis with the capital recovery factor [135].

An economic model which presents a similar methodology to the payback period method is life cycle saving. Like the payback period method, an already existing non-hybrid plant operating with natural gas only is considered. This method is utilized to investigate the economic advantages of hybridization on an already operating plant. Thus, additional investment required to hybridize the plant is studied. Additionally, this assessment approach is employed to investigate the feasibility of a bottoming cycle for an already existing gas turbine power plant. Consequently, the reference plant configuration is a simple gas turbine cycle. It is of high interest to investigate the most economically justified waste heat recovery bottoming cycle for simple gas turbine power plants with different capacities. Hence, additional investment associated with the bottoming cycle incorporation or plant hybridization is calculated. Life cycle saving is determined by [27]:

\[
S = \frac{Z_{ann, rev}}{CRF} - (Z_{inv, ad}) \tag{3.180}
\]
where $S$ is the life cycle saving in US$, $Z_{\text{ann,rev}}$ is the profit or loss associated with the variation in the annual operating and maintenance costs in US$/year, and $Z_{\text{inv,ad}}$ is the additional capital investment cost for the bottoming cycle integration or plant hybridization in US$.

Finally, the most popular economic indicator for power plants’ economic assessment is the levelized cost of electricity (LCOE). Understandably, lower LCOE implies that the investment is more cost-effective. In principle, LCOE is the cost of electricity sale which makes the net present value of the investment equal to zero. If the actual cost of the electricity sale to the grid is lower than the calculated LCOE, the investment is not profitable and net present value will be negative. Therefore, there is an internal connection between the net present value and LCOE. However, LCOE is a more convenient economic indicator to understand and interpret. LCOE is determined by [122]:

$$LCOE = \frac{\alpha Z_{\text{inv}} + \beta Z_{\text{Dec}} + Z_{\text{opt}} + Z_{\text{mat}} + Z_{\text{lab}}}{W_{\text{net}}}$$

(3.181)

where $LCOE$ is the levelized cost of electricity in US$/kWh, and coefficients $\alpha$ and $\beta$ are calculated as follows [122]:

$$\alpha = \left[ \frac{(1+i)^{N_{\text{con}}}-1}{N_{\text{con}}i} \right] \left[ \frac{i(1+i)^{N_{\text{opt}}}}{[1+i]^{N_{\text{opt}}}-1} \right] + r_{\text{ins}}$$

(3.182)

$$\beta = \left[ \frac{(1+i)^{N_{\text{dec}}}-1}{N_{\text{dec}}i[1+i]^{N_{\text{dec}}-1}} \right] \left[ \frac{i}{[1+i]^{N_{\text{opt}}}-1} \right]$$

(3.183)

Each economic indicator provides specific information regarding the economics of the power plant. Net present value and levelized cost of electricity present the economic performance of the plant throughout construction, operation, and decommissioning. Knopf’s [135] method presents a simplified annual economic performance of the plant in a dollar per second basis and simplifies the optimization process. The payback period method offers general yet trustworthy information regarding the economic benefits and demerits of power plant hybridization (solar tower integration). While life cycle saving indicates the feasibility of power plant hybridization. We must take into account that the real rate of interest is considered to be 7% whereas the annual insurance rate is 1% [122]. Construction, operation, and
decommissioning times are selected to be 2, 25, and 2 years, respectively [122]. Furthermore, the generated electricity average sale price to the grid is considered to be 0.07 US$/kWh [30, 156]. Moreover, the plant is considered to be operational for 8760 hours a year [7, 135].

To solely focus on the heliostat field thermo-economic performance, the levelized cost of energy (LCOEN) is selected. The levelized cost of thermal energy supplied to the receiver could be a proper indicator of the heliostat field thermo-economic performance which is calculated such that:

\[
LCOEN = \frac{\left[ i \left[ 1 + \frac{r_{ins}}{1 + i} \right]^{N_{opr}} \right] \left[ Z_{helf} + Z_T + Z_{rec} \right] + Z_{opr,sol} + Z_{mai,sol}}{E_{net}}
\]

where, \( E_{net} \) is the annual thermal energy delivered at the receiver’s surface in kWh, and \( LCOEN \) is the levelized cost of energy in US$/kWh.

3.3. Model Validation

In this section, two main components of the developed thermo-economic models are evaluated. In other words, these models’ levels of accuracy are tested to accredit the acquired results presented in this research work. Initially, the gas turbine model is validated prior to assessing the precision of the shading and blocking factor model for the heliostat field analysis.

3.3.1. Gas turbine model.

In order to assess the accuracy of the gas turbine model developed in the previous sections, the acquired results are compared with two Siemens gas turbine plants. The aforementioned gas turbine plants’ specifications along with the developed model results are given in Table 8 and Table 9. The analyses are conducted at ISO conditions with ambient air temperature of 15 °C and pressure of 1.013 bar.

Initially, a simple gas turbine cycle with cogeneration of heat and power is analyzed by utilizing the SGT-700 data within the developed MATLAB model. The results indicate that the model is capable of acquiring a relatively accurate set of results to carry out the analysis for the main parts of this research work. Power output is extremely accurate with only 0.61% relative error. Moreover, efficiency’s relative
inaccuracy is acceptable with only 1.54%. Even a 6% error in steam mass flow rate calculation is considered to be a good result. In the second stage of the gas turbine model evaluation, a SGT-800 gas turbine cycle is taken into consideration. Bearing in mind that the SGT 800 gas turbine’s capacity is greater, the model reliability for larger scale power plants are evaluated. The results are sufficiently accurate with all the relative errors in the range of 1% to 3%.

Table 8: Results acquired by the developed model and presented in SGT-700 catalogue [157]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Siemens GT-700</th>
<th>Model Result</th>
<th>Relative Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor pressure ratio</td>
<td>18.6</td>
<td>18.6</td>
<td>-</td>
</tr>
<tr>
<td>Exhaust gas temperature</td>
<td>528 °C</td>
<td>528 °C</td>
<td>-</td>
</tr>
<tr>
<td>Exhaust gas mass flow rate</td>
<td>95 kg/s</td>
<td>95 kg/s</td>
<td>-</td>
</tr>
<tr>
<td>Pinch temperature difference</td>
<td>8 k</td>
<td>8</td>
<td>-</td>
</tr>
<tr>
<td>Approach temperature difference</td>
<td>5 k</td>
<td>5</td>
<td>-</td>
</tr>
<tr>
<td>Steam Pressure</td>
<td>40 bar</td>
<td>40 bar</td>
<td>-</td>
</tr>
<tr>
<td>Water Temperature</td>
<td>250 °C</td>
<td>250 °C</td>
<td>-</td>
</tr>
<tr>
<td>Power output</td>
<td>31.21 MWe</td>
<td>31.40 MWe</td>
<td>+ 0.61 %</td>
</tr>
<tr>
<td>Efficiency</td>
<td>36.4%</td>
<td>35.84%</td>
<td>- 1.54 %</td>
</tr>
<tr>
<td>Steam mass flow rate</td>
<td>16</td>
<td>17.06</td>
<td>+ 6.63 %</td>
</tr>
</tbody>
</table>

Table 9: Results acquired by the developed model and presented in SGT-800 catalogue [158]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Siemens GT-800</th>
<th>Model Result</th>
<th>Relative Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor pressure ratio</td>
<td>21.1</td>
<td>21.1</td>
<td>-</td>
</tr>
<tr>
<td>Exhaust gas temperature</td>
<td>553 °C</td>
<td>553 °C</td>
<td>-</td>
</tr>
<tr>
<td>Exhaust gas mass flow rate</td>
<td>134.2 kg/s</td>
<td>134.2 kg/s</td>
<td>-</td>
</tr>
<tr>
<td>Fuel mass flow rate</td>
<td>2.67 kg/s</td>
<td>2.69 kg/s</td>
<td>+ 0.75 %</td>
</tr>
<tr>
<td>Power output</td>
<td>50.5 MWe</td>
<td>49.95 MWe</td>
<td>- 1.09%</td>
</tr>
<tr>
<td>Efficiency</td>
<td>38.3%</td>
<td>37.17 %</td>
<td>- 2.95 %</td>
</tr>
</tbody>
</table>

The main reason behind the errors can be traced back to the implementation of the NASA polynomial for specific heat capacity and specific entropy of air and exhaust gases. Additionally, this research work only considers a simple model for exhaust gases. We need to bear in mind that CH₄ is assumed to be the only source of energy in the combustion chamber while the natural gas employed in real gas turbine plants contains multiple components. For enhancing the accuracy of the model, real gas properties of air and exhaust gas and a more complex fuel can be taken into consideration. However, the complexity that the aforementioned solutions bring to the calculation does not justify the enhancement in the model’s accuracy. Moreover, its complexity may restrain the analysis of the proposed configurations. Taking into
account that hybrid power plants are considerably complicated problems, it was decided to carry out the analysis with the aforementioned conditions.

3.3.2. Heliostat field model.

The shading and blocking factor is the most complicated and error-prone optical efficiency factor. Consequently, its accuracy must be closely examined. The developed shading and blocking factor model is validated by evaluating its result with the shading and blocking factor results presented by Collado and Guallar [116]. For comparison purposes, the same field layout (densest field layout) with 35 heliostats placed in the first row is generated. The analysis is carried out for PSA (solar plant in Almeria, Spain) with latitude of 37.1° on the 11th of December (Day: 345) at solar hour of 9:00 in the morning. Shadowing and blocking results for four different random heliostats within the third zone of the field are considered. Results presented by Collado and Guallar [116] are depicted in Figure 35 while shading and blocking factor results achieved by our developed code in MATLAB are illustrated in Figure 36. Additionally, the field shading and blocking factors achieved by the developed model and available in the literature [116] are presented as heat maps in Figure 37.

Comparing Collado and Guallar’s results [116] with our suggested model results, one can clearly notice the strong agreement between these two models of shading and blocking factors. Our model evaluates more possible shading candidates which in turn improves its accuracy as compared with the aformentioned model [116]. On the other hand, considering a higher number of potential shading candidates increases the instances of expensive shading and blocking factor evaluation and optimization computational time in our model. Comparing the shading and blocking heliostats’ projections on the 56th heliostat of the 10th row in the third zone of the field shows that only one shading candidate affected the analyzed heliostat surface in both results. Additional shading candidates considered in our model do not have any effect on the analyzed heliostat shading and blocking factor calculations. Likewise, three other heliostats are analyzed and the projection of their potentially shading and blocking heliostats are illustrated. In none of the investigated cases, additional heliostats which are considered in our model affect the shading and blocking factor of the analyzed heliostat. Nonetheless, it can be said that our developed model is more accurate since a higher number of shading and blocking candidates are considered in the developed model.
Furthermore, Figure 37 is a validation of the accuracy of our developed model. Heat maps are approximately identical considering that there is only a minor dissimilarity between their colorbars. Moreover, shading and blocking factors associated with the first, second, and third zones of the field are in strong agreement with Collado and Guallar’s results [116] by two significant figures. One can note that our model results in lower shading and blocking factors in all cases. This can be traced back to the higher number of shading heliostat candidates considered in our developed model. Furthermore, accuracy of the shading and blocking factor implies that the field layout generation, heliostats, and sun coordinates and unit vectors, as well as the cosine efficiency factor provide satisfactory precision. In general, it can be concluded that our developed model in MATLAB can be trusted for further analysis of the heliostat field design in the UAE and any solar fields elsewhere.
Figure 35: Shading and blocking factor analysis presented by Collado and Guallar [116]
Figure 36: Shading and blocking factor analysis accomplished by the developed model, a) Shading and blocking heliostats projection (zone =3, row=10, Heliostat=56), b) Shading and blocking heliostats projection (zone =3, row=6, Heliostat=71), c) Shading and blocking heliostats projection (zone =3, row=20, Heliostat=74), d) Shading and blocking heliostats projection (zone =3, row=6, Heliostat=115), e) Densest field layout for PSA
Figure 37: Field shading and blocking factor evaluation, a) current developed model, b) model developed by Collado and Guallar [116]
Chapter 4: Results and Discussion

In this section, all the acquired results for the proposed configurations are presented and discussed. Initially, the two presented heliostat field layouts are thoroughly investigated. In the next section, a comparative analysis between the thermo-economic performances of all the proposed power plant configurations are presented. The thermo-economic analyses are carried out for non-hybrid power plants without an integration of a solar collector to investigate the most economically justified combined cycle configuration and waste heat recovery bottoming cycle. In the next sections, each hybrid power plant configuration is comprehensively evaluated and optimized. The analyses begin with ABC power plants and continue for SBC, HABC, and MBC configurations, respectively.

4.1. Heliostat Field Design

In this section, three different objectives including field annual weighted efficiency, annual unweighted efficiency, and levelized cost of energy (LCOEN) are selected. It is of high interest to investigate the optimization objective impact on the optimized field layout. Optimization is performed for both introduced field layouts, i.e. radial-staggered and spiral, to accomplish a comparative analysis between the field layouts.

4.1.1. Weighted efficiency.

Initially annual weighted efficiency is considered for the optimization of the field. The radial-staggered field layout is optimized by varying the reference blocking factor which controls the local radial increment within the field. By increasing the reference blocking factor, the field layout is expanded from the most condensed configuration to the optimum set up. Results concerning the effects of the reference blocking factor on the radial-staggered layout LCOEN, weighted and unweighted efficiencies are depicted in Figure 38. At the densest configuration, weighted efficiency is found to be 57.72%. By expanding the field layout, the weighted efficiency value rises to its peak value of 58.61% with a reference blocking factor of 0.93.
There are two control variables for spiral layout field expansion. While the aforementioned line search optimization method is utilized to assess the optimum field layout, it is decided to provide more information on how the field weighted efficiency varies against changing control variables. It is worth to indicate that an increase in each control variable results in a more expanded field layout. Thus, the optimum field layout is achieved when both control variables contribute to expanding the field layout. The goal of the optimization is to find a balance between all control’s variables contributed in the process of the field expansion. Spiral field layout annual weighted efficiency changes with varying values of control variables is shown in Figure 39. Results indicate that there are three regions which possibly contain the optimum weighted efficiency. Hence, each region is presented in a more detailed manner for better understanding of the optimum weighted efficiency and its corresponding combination of control variables.
Figure 39: Weighted efficiency optimization of the spiral field layout by control variables alterations, a) $a = [2,8]$, $b = [0.5,0.7]$, b) $a = [3,4]$, $b = [0.63,0.67]$, c) $a = [5,6]$, $b = [0.58,0.62]$, d) $a = [7,8]$, $b = [0.53,0.58]$.

Radial-staggered optimum field layout is presented in Figure 40. It can be noticed that the northern semicircle of the field is more expanded as compared with the southern semicircle. In this Figure, the dashed line represents the densest field layout. Likewise, expansion is greater in the eastern section of the field. Understandably, heliostats located in the first and second zones have greater annual efficiency. Moreover, the northern section of the field displays much better annual efficiency. Optimum spiral field layout is depicted in Figure 41. In this layout, the minimum distance between the tower and the first row of mirrors is considered and kept constant for both field layouts’ generations. Additionally, selected values for the control variables result in an incomplete field layout for regions close to the tower. Adding more heliostats within these regions will cause heliostats’ overlapping. Finally, heliostats located approximately 100m away from the tower do not experience overlapping with the assigned values of the control variables. Similar to radial-staggered layout, northern mirrors have the highest annual efficiencies.
Figure 40: Radial-staggered optimized field layout with weighted efficiency as the objective (reference blocking factor = 0.93)
Figure 41: Spiral optimized field layout with weighted efficiency as the objective \((a = 3.22, b = 0.67)\)

Results concerning the optimum field configurations for spiral and radial-staggered layouts are presented in Table 10. The radial-staggered field design presents better performance with annual weighted efficiency of 58.61%. Whereas, the spiral field maximum achievable annual weighted efficiency is 58.38%. The weighted efficiency is considered the best indicator of the field thermal performance since it has a direct relationship with the annual thermal energy delivered to the central receiver. Annual thermal energy supplied by the radial-staggered field is about 197,660 MWh while the spiral field can provide 196,880 MWh. On an annual basis, the radial-staggered layout can deliver additional thermal energy of 780 MWh/year as compared with the spiral layout which can be translated into about 89 kW of additional rate of heat flux at the receiver surface on average. In addition, initial capital investment required for the spiral layout design is slightly lower. Nonetheless, the radial-staggered field displays superior thermo-economic performance by attaining a lower value of
LCOEN. The difference between the layouts’ LCOEN values might seem insignificant in current format; however, this rather insignificant difference between the fields’ LCOEN can result in a considerable amount of saving for the field life span of 25 years. Additional interesting information which can be seen in the results presented in Table 10 is the optical efficiency factors. At the optimum field layout, the spiral layout illustrates better performance in shading and blocking factors but poorer results for all three other optical efficiency factors, i.e. spillage, cosine, and attenuation. It is important to note that the spiral layout achieves a better shading and blocking factor by occupying smaller land since the land capital investment cost will be less.

Table 10: Optimized field layout specifications with weighted efficiency as the optimization objective

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Spiral layout</th>
<th>Radial-staggered layout</th>
</tr>
</thead>
<tbody>
<tr>
<td>Optimization control variables</td>
<td>𝑎 = 3.22, 𝑏 = 0.67</td>
<td>𝑓_{b, ref} = 0.93</td>
</tr>
<tr>
<td>Weighted efficiency (%)</td>
<td>58.38</td>
<td>58.61</td>
</tr>
<tr>
<td>Unweighted efficiency (%)</td>
<td>54.78</td>
<td>54.90</td>
</tr>
<tr>
<td>LCOEN (US$/MWh)</td>
<td>34.08</td>
<td>34.01</td>
</tr>
<tr>
<td>Annual energy delivered at the receiver surface (MWh)</td>
<td>196880</td>
<td>197660</td>
</tr>
<tr>
<td>Capital investment cost (MUS$)</td>
<td>52.49</td>
<td>52.57</td>
</tr>
<tr>
<td>Land capital investment cost (MUS$)</td>
<td>1.536</td>
<td>1.615</td>
</tr>
<tr>
<td>Wiring capital investment cost (MUS$)</td>
<td>1.388</td>
<td>1.396</td>
</tr>
<tr>
<td>Annual operating cost (MUS$/year)</td>
<td>0.012</td>
<td>0.012</td>
</tr>
<tr>
<td>Annual maintenance cost (MUS$/year)</td>
<td>1.67</td>
<td>1.67</td>
</tr>
<tr>
<td>Annual field average shading &amp; blocking factor (%)</td>
<td>93.55</td>
<td>92.92</td>
</tr>
<tr>
<td>Annual field average cosine factor (%)</td>
<td>76.48</td>
<td>76.98</td>
</tr>
<tr>
<td>Annual field average attenuation factor (%)</td>
<td>92.53</td>
<td>92.73</td>
</tr>
<tr>
<td>Annual field average spillage factor (%)</td>
<td>93.80</td>
<td>94.04</td>
</tr>
</tbody>
</table>

4.1.2. Unweighted efficiency.

Similar analyses are carried out by selecting the field annual unweighted efficiency as the objective of the optimization. The analyses provide a better insight on the influence of the optimization objective over the optimum field layout and its thermo-economic performance. Based on the results depicted in Figure 38, radial-staggered field layout maximum unweighted efficiency is achieved by assigning the reference blocking factor to be 0.95. Results concerning the values of the unweighted efficiency for the spiral field layout within the search domain are shown in Figure 42. In general, the unweighted efficiency behavior is significantly similar to the weighted efficiency; however, the value of the unweighted efficiency is generally smaller. Similarly, there are three regions within the search domain which might contain the optimum unweighted efficiency. Consequently, these regions are illustrated with more
details. The optimum value for the design variables which provide the maximum unweighted efficiency are \( a = 3.32, b = 0.67 \).

The optimum field layout for the radial-staggered configuration is presented in Figure 43. Like the field layout presented in the previous section, the optimum field layout for unweighted efficiency is more expanded within its north and east regions. However, the optimum layout depicted in Figure 43 is a more expanded version of the layout presented in Figure 40. In addition, the spiral field optimum layout is presented in Figure 44. Once more, the optimum field layout for the weighted and unweighted efficiencies for spiral configuration are approximately identical. Nonetheless, the layout obtained for the optimum unweighted efficiency is a bit more extended.

Figure 42: Unweighted efficiency optimization of the spiral field layout by control variables variations, a) \( a=[2,8], b=[0.5,0.7] \), b) \( a=[3,4], b=[0.63,0.67] \), c) \( a=[5,6], b=[0.58,0.62] \), d) \( a=[7,8], b=[0.53,0.58] \)
Results concerning the optimum field configurations for spiral and radial-staggered layouts are presented in Table 11. Results indicate that the radial-staggered optimum unweighted efficiency is about 54.92% while the spiral layout maximum attainable unweighted efficiency is about 54.79%. In general, the radial-staggered layout performs better than the spiral layout in the weighted and unweighted efficiencies as well as the LCOEN. Moreover, the radial-staggered optimum field layout supplies a greater amount of energy to the receiver. On the other hand, the spiral layout presents better shading and blocking factors and requires lower capital investment as compared with the radial-staggered layout. Comparing the performance of the optimum field layouts for weighted and unweighted efficiencies, one can notice that selecting weighted efficiency as the main optimization objective function results in greater annual energy supplied and lower initial capital investment cost for both radial-staggered and spiral configurations. It can be perceived that optimizing heliostat fields
with unweighted efficiency as the main objective results in over expanding the field with degrading its performance.

Figure 44: Spiral optimized field layout with unweighted efficiency as the objective

(a = 3.32, b = 0.67)

Table 11: Optimized field layout specifications with unweighted efficiency as the optimization objective

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Spiral layout</th>
<th>Radial-staggered layout</th>
</tr>
</thead>
<tbody>
<tr>
<td>Optimization control variables</td>
<td>$a = 3.32, b = 0.67$</td>
<td>$f_{b,ref} = 0.95$</td>
</tr>
<tr>
<td>Weighted efficiency (%)</td>
<td>58.31</td>
<td>58.60</td>
</tr>
<tr>
<td>Unweighted efficiency (%)</td>
<td>54.79</td>
<td>54.92</td>
</tr>
<tr>
<td>LCOEN (US$/MWh)</td>
<td>34.19</td>
<td>34.08</td>
</tr>
<tr>
<td>Annual energy delivered at the receiver surface (MWh)</td>
<td>196630</td>
<td>197600</td>
</tr>
<tr>
<td>Capital investment cost (MUS$)</td>
<td>52.58</td>
<td>52.67</td>
</tr>
<tr>
<td>Land capital investment cost (MUS$)</td>
<td>1.601</td>
<td>1.684</td>
</tr>
<tr>
<td>Wiring capital investment cost (MUS$)</td>
<td>1.420</td>
<td>1.425</td>
</tr>
<tr>
<td>Annual operating cost (MUS$/year)</td>
<td>0.012</td>
<td>0.012</td>
</tr>
<tr>
<td>Annual maintenance cost (MUS$/year)</td>
<td>1.67</td>
<td>1.68</td>
</tr>
<tr>
<td>Annual field average shading &amp; blocking factor (%)</td>
<td>94.03</td>
<td>93.25</td>
</tr>
<tr>
<td>Annual field average cosine factor (%)</td>
<td>76.37</td>
<td>76.92</td>
</tr>
<tr>
<td>Annual field average attenuation factor (%)</td>
<td>92.44</td>
<td>92.68</td>
</tr>
<tr>
<td>Annual field average spillage factor (%)</td>
<td>93.49</td>
<td>93.83</td>
</tr>
</tbody>
</table>
4.1.3. Levelized cost of energy.

The third possible alternative that can be considered as an optimization objective is the field LCOEN. LCOEN combines both thermal and economic performances of the solar field. Consequently, LCOEN can be considered as a more comprehensive and reliable indicator of the field general proficiency. Radial-staggered field layout optimum LCOEN is obtained at a lower value of reference blocking factor compared with the other two selected objectives. Therefore, the layout is denser and more compact. The optimum field layout for the radial-staggered design is shown in Figure 46. It is worthwhile to note that the expansion in the southern region of the tower is insignificant. This result indicates that the thermal improvement achieved by expanding the southern region of tower is not economically justifiable. For the spiral field layout, its corresponding values of LCOEN within the feasible region are depicted in Figure 45. Once again, the same three regions are possibly comprising the optimum values of the design variables. Thus, the optimum field layout which is depicted in Figure 47 is achieved by assigning design variables $a$ and $b$ to be 3.88 and 0.64, respectively. Additionally, the field layout is found to be more condensed compared with the abovementioned field configurations.

Results concerning the optimum layouts for the spiral and radial-staggered field configurations are listed in Table 12. The most cost effective field layouts are almost identical in terms of thermal performance. It is important to note that the weighted efficiency and the amount of energy supplied by the field are approximately equal for both spiral and radial-staggered layouts. Nevertheless, the main difference between their LCOEN values originates from their initial capital investments dissimilarity. Additionally, it can be deduced from the results that the optimum fields presented in this section provide a lower amount of energy compared with the optimum layout achieved for the weighted efficiency optimization. Therefore, a proper optimization objective must be selected with care to achieve the desired performance from the solar field. Selecting weighted efficiency as the optimization objective results in augmenting the amount of energy provided by the solar field. On the other hand, LCOEN optimization illustrates that field with over expansion for maximizing the supplied energy might not be economically justifiable. Optimizing the field layout with the unweighted efficiency generates an overexpanded field which only increases the capital investment cost and degrades the field thermal proficiency.
Figure 45: LCOEN optimization of the spiral field layout by control variables alterations, a) $a = [2,8], b = [0.5,0.7]$, b) $a = [7,8], b = [0.53,0.58]$, c) $a = [5,6], b = [0.58,0.62]$, d) $a = [3,4], b = [0.63,0.68]$

Table 12: Optimized field layout specifications with LCOEN as the optimization objective

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Spiral layout</th>
<th>Radial-staggered layout</th>
</tr>
</thead>
<tbody>
<tr>
<td>Optimization control variables</td>
<td>$a = 3.88, b = 0.64$</td>
<td>$f_{b,ref} = 0.82$</td>
</tr>
<tr>
<td>Weighted efficiency (%)</td>
<td>58.31</td>
<td>58.35</td>
</tr>
<tr>
<td>Unweighted efficiency (%)</td>
<td>54.60</td>
<td>54.41</td>
</tr>
<tr>
<td>LCOEN (US$/MWh)</td>
<td>34.03</td>
<td>33.88</td>
</tr>
<tr>
<td>Annual energy delivered at the receiver surface (MWh)</td>
<td>196660</td>
<td>196780</td>
</tr>
<tr>
<td>Capital investment cost (MUSS)</td>
<td>52.33</td>
<td>52.14</td>
</tr>
<tr>
<td>Land capital investment cost (MUSS)</td>
<td>1.430</td>
<td>1.317</td>
</tr>
<tr>
<td>Wiring capital investment cost (MUSS)</td>
<td>1.335</td>
<td>1.259</td>
</tr>
<tr>
<td>Annual operating cost (MUSS/year)</td>
<td>0.012</td>
<td>0.012</td>
</tr>
<tr>
<td>Annual maintenance cost (MUSS/year)</td>
<td>1.67</td>
<td>1.66</td>
</tr>
<tr>
<td>Annual field average shading &amp; blocking factor (%)</td>
<td>92.69</td>
<td>90.91</td>
</tr>
<tr>
<td>Annual field average cosine factor (%)</td>
<td>76.62</td>
<td>77.24</td>
</tr>
<tr>
<td>Annual field average attenuation factor (%)</td>
<td>92.63</td>
<td>92.95</td>
</tr>
<tr>
<td>Annual field average spillage factor (%)</td>
<td>94.19</td>
<td>94.89</td>
</tr>
</tbody>
</table>
Figure 46: Radial-staggered optimized field layout with LCOEN as the objective (reference blocking factor = 0.82)
4.1.4. Sensitivity analysis.

There are additional factors (design parameters) that could influence a heliostat field performance including tower height, mirror surface area, receiver’s dimensions, and number of mirrors in the field. In the previous section, these variables were kept constant to perform a proper comparative analysis between two popular alternatives for heliostat field layouts designs. Nevertheless, it is of high interest to investigate the impact each of these additional design parameter variables might have on the field performance both thermodynamically and economically. In this section, the radial-staggered layout is the only one considered since variation in the aforementioned parameters have rather similar effects on the field performance regardless of the field layout design. It is recommended to mention that the optimum field configuration might change by varying the central tower height, mirror surface area, and number of mirrors. Therefore, optimization is carried out for every investigated design parameter. For
example, maximum weighted efficiency achievable by central tower height of 100m is calculated and compared with the maximum possible weighted efficiency for a field with central tower height of 150m. In other words, the field layout is adapted with respect to the assigned design variables’ values.

Results concerning the effect of mirrors surface area on the field LCOEN and weighted efficiency are depicted in Figure 48. In order to minimize the computational efforts, only five different values of heliostat surface area are considered. It is worthwhile to mention that the field layout optimization is conducted twice for both weighted efficiency and LCOEN optimizations. Consequently, ten optimized field layouts are obtained for the presented results. Understandably, bigger heliostats will degrade the plant optical efficiency, in particular through the shading and blocking factor. Hence, the plant weighted efficiency is constantly dropping by increasing the size of the mirrors. Nevertheless, a greater amount of thermal energy is provided by the field. In general, larger mirrors require greater initial investment, maintenance, and operating costs as compared with small mirrors. Therefore, there exists an optimum condition where the additional energy supplied by the field justifies the extra financial requirements. Based on the results presented in Figure 48, the most cost effective mirror size lies in the neighborhood of 125 m².

Figure 48: Effects of the mirrors surface area on the field LCOEN and weighted efficiency

In the model presented by Collado and Guallar [116], the number of heliostats integrated in the field is calculated by selecting a proper value for the first row number
of heliostats. Consequently, the number of heliostats in the first row variation, in principle, represents the change in the number of heliostats within the solar field. Results concerning the impact of increasing the number of heliostats in the field are shown in Figure 49. As expected, increasing the number of heliostats only reduces the field annual weighted efficiency. Extra needed mirrors can be realized by having more mirrors in areas far away from the central tower with a significantly lower optical efficiency or increasing the field density and degrading the shading and blocking factor. Nonetheless, lower weighted efficiency does not necessarily mean that less energy is provided by the field. Moreover, a higher number of heliostats requires greater financial investment. Like mirror sizing, this tradeoff implies that an optimum condition may exist in which the positive and negative impacts of extra heliostat addition neutralize each other. Results indicate that having approximately 1460 (20 mirrors in the first row) is the most economical alternative with LCOEN of 33.88 US$/MWh, whereas the minimum LCOEN obtainable with 2350 mirrors in the field (25 in the first row) is about 34.34 US$/MWh.

![Figure 49: Effects of the number of heliostats on the field LCOEN and weighted efficiency](image)

The impact of the central tower height on the field LCOEN and weighted efficiency are investigated and the results presented in Figure 50. It is noted that taller central towers improve the field optical efficiency. Consequently, the weighted efficiency is continuously ameliorating by increasing the height of the central tower. It is desirable to mention that the optical tower height is the vertical distance between the heliostat’s surface and the receiver while the tower actual height is the vertical elevation
of the receiver from the ground. Improvement in the field annual weighted efficiency is an indication of the enhancement in the field thermal performance. Nonetheless, the field initial investment cost increases as well. The minimum LCOEN of 33.88 US$/MWh is achieved by having a 125m central tower.

![Figure 50: Effects of central tower height on the field LCOEN and weighted efficiency](image)

Results concerning the effect of the receiver’s dimensions on the field LCOEN and weighted efficiency are depicted in Figure 51. Variation in the receiver’s surface area could be achieved by altering the receiver’s height and radius. It is better to mention that increasing the receiver’s radius and height might have different impacts on the field performance. To establish a general understanding of the receiver’s dimensions’ influence on the field thermo-economic performance, we decided to vary the receiver’s height and radius simultaneously with step size of 1m. Increasing the receiver’s dimensions only improves the spillage efficiency factor and does not have any other effects on the other optical factors. Consequently, a larger receiver’s surface area provides a bigger target for the mirrors to focus the reflected solar radiation which enhances the annual weighted efficiency. It is important to remember that the solar receiver economic analysis is subject to significant uncertainty [122]. Thermo-economic results indicate that the local optimum receiver’s height and radius are 9m and 4m, respectively. Minimum LCOEN attained with the aforementioned receiver’s dimensions is 33.88 US$/MWh.

It is better to mention that the presented thermo-economic analyses are obtained by considering Abu Dhabi’s weather data as an example for detailed analysis. Thus, the
results might not be extendable to other site locations. Moreover, the sensitivity analyses mainly provide an understanding of design variables’ impacts on the field thermo-economic performance rather than an optimum operating conditions. In other words, the aforementioned optimum values are more of a local minimum and rough approximation of the optimum operating conditions.

4.1.5. Concluding remarks.

In this section, two different field layouts, including radial-staggered and spiral layouts, are selected for optimization. Three different objectives are considered for the field layout optimization. Additionally, a comparative analysis is conducted between the studied field layouts’ thermo-economic performance. Finally, the effects of the design parameters such as the central tower height, number of heliostats in the field, size of the mirrors and the receiver’s dimensions on the field thermal and economical capabilities are investigated.

In general, the optimization method, proposed by Collado and Guallar [116], systemizes the optimization procedure of the radial-staggered field layouts and improves their optimum performance by calculating the local radial increment between the rows of mirrors. Therefore, the radial-staggered layout displays better performance than the spiral layout in all considered objectives. Noone et al. [117] argued that the transition between the high and low density areas of the field is not continuous in the radial-staggered layout. Nevertheless, the proposed optimization method by Collado
and Guallar [116] solves this problem to some extent by locally calculating the radial increment between the rows of mirrors. Radial-staggered optimum weighted efficiency is 58.61% while the maximum achievable weighted efficiency for the spiral layout is 58.38%. Moreover, optimum LCOEN for the radial-staggered and spiral layouts are 33.88 US$/MWh and 34.03 US$/MWh, respectively. Additionally, field optimization with weighted efficiency as the objective results in maximizing the amount of energy provided by the field. On the other hand, LCOEN optimization shows that field overexpansion for maximizing the supplied thermal energy might not be economically justified. Optimizing the field layout with the annual unweighted efficiency is not advised as it results in field overexpansion and thermo-economic deterioration.

Sensitivity analysis provides a comprehensive understanding of the optimum field design variables. The most cost effective mirror size lies in the neighborhood of 125 m² whereas the optimum tower height is 125m. Moreover, the optimum number of heliostats within the field is 1460. Finally, the optimum receiver’s radius and height are found to be 4m and 9m, respectively. The minimum LCOEN achieved with the aforementioned optimum values is about 33.85 US$/MWh.

4.2. Non-Hybrid Plants Thermo-Economic Analyses

Three different scenarios are considered to fully evaluate the aforementioned combined cycle configurations’ thermo-economic performances for small scale power generation. Initially, the plant total operating cost method proposed by Knopf [135] is employed to investigate the economics of a new combined cycle power plant installment. In the second stage of the analysis, an already existing gas turbine cycle is taken into consideration. Waste heat recovery capability and thermo-economic performance of different bottoming cycles are investigated utilizing the life cycle saving approach. Finally, the LCOE method is applied to investigate the selected combined cycles’ thermo-economic performance for a hot and humid climate such as in the UAE. Consequently, transient analysis of ABC, SBC, HABC, and MBC configurations must be accomplished to specify their optimum LCOE.

4.2.1. New plant installment economic analysis.

In this section, economic analysis of a 50 MWe combined cycle power plant is conducted. The abovementioned plant’s combined configurations including SBC (CCC), ABC, HABC, and MBC are considered and their optimum operating costs are
determined to identify the most cost effective alternative for small scale power plants. It should be noted that the optimization is accomplished by taking the plant operating cost as the objective. Furthermore, each configuration is optimized with different design variables. The design variables and their corresponding search domains are listed in Table 13. It should be noted that MBC and HABC power plant optimizations are carried out with five design variables, i.e. TCPR, BCPR, MFRR, ASDH (BCAH for HABC configuration), and GTIT. The optimizations of the ABC power plant configuration consist of four design variables comprising TCPR, BCPR, MFRR, and GTIT. In addition, for the economic optimization of the SBC power plant, four design variables are selected including steam bottoming cycle pressure ratio (BCPR), DOSH (determines the bottoming cycle steam mass flow rate), GTIT, and TCPR. Bearing in mind that the plant’s economic performance can be compared since an identical plant capacity is presumed for all four investigated configurations.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Assigned search domain</th>
<th>Assigned configurations</th>
</tr>
</thead>
<tbody>
<tr>
<td>Topping cycle pressure ratio (TCPR)</td>
<td>10.0 — 25.0</td>
<td>ABC, SBC, HABC, MBC</td>
</tr>
<tr>
<td>Bottoming cycle pressure ratio (BCPR)</td>
<td>2.0 — 8.0</td>
<td>ABC, HABC, MBC</td>
</tr>
<tr>
<td>Bottoming cycle pressure ratio (Pump)</td>
<td>500 — 1000</td>
<td>SBC</td>
</tr>
<tr>
<td>Gas turbine inlet temperature (GTIT)</td>
<td>1223 K — 1673 K</td>
<td>ABC, SBC, HABC, MBC</td>
</tr>
<tr>
<td>Air saturator degree of humidification (ASDH)</td>
<td>0.00 — 1.00</td>
<td>ABC, SBC, HABC, MBC</td>
</tr>
<tr>
<td>Degree of superheating (DOSH)</td>
<td>0.10 — 1.00</td>
<td>ABC, HABC, MBC</td>
</tr>
<tr>
<td>Bottoming cycle air humidification (BCAH)</td>
<td>0.10 — 1.00</td>
<td>HABC</td>
</tr>
<tr>
<td>Mass flow rate ratio (MFRR)</td>
<td>0.50 — 1.50</td>
<td>ABC, HABC, MBC</td>
</tr>
</tbody>
</table>

Power plant analysis and optimization present a complicated problem. Taking into account that the developed mathematical models are nonlinear, non-differentiable and discontinuous, traditional gradient based optimization methods cannot be utilized. Consequently, a different optimization method must be employed for the analysis of this research work. The genetic algorithm optimization approach is the most appropriate method for the problem on our hands since it only needs the objective function value. As a result, genetic algorithm methods can effortlessly handle nonlinear, non-differentiable and discontinuous problems [159]. To carry out the genetic algorithm, the MATLAB optimization toolbox provides an efficient built-in function [160]. Nonetheless, the main weakness of the genetic algorithms is the uncertainty in their capability to identify the optimum condition. In other words, obtaining the global optimum is not guaranteed by genetic algorithms. Thus, a hybrid optimization approach
is taken into consideration. In hybrid optimizations, two distinctive optimization methods are carried out in a successive manner.

For the second stage of the hybrid optimization, the recommended method by Ramos and Ramos [119] which adopts the Powell [125] algorithm is selected. The algorithm implements a line search approach to the optimization of power plants. Nonetheless, its conversion is strongly dependent on the selected starting point. Consequently, the optimum values of design variables provided by the MATLAB genetic algorithm are utilized to initiate this line searching optimization method. The procedure is initiated by optimizing each design variable separately while the other design variables are kept constant. This process is repeated to find the optimum value for each design variable. At this stage, the first iteration of the optimization is completed. Next, several iterations are carried out to attain the optimum values of all the considered design variables with adequate precision.

Results concerning the economic optimization of different combined cycle configurations are tabulated in Table 14. Initially, the 50 MWe SBC power plant optimum configuration is discussed in details. Four design variables are selected for economic optimization of the SBC power plant comprising TCPR, BCPR, GTIT, and DOSH. The optimum TCPR value is determined to be 13.9 while the most economical GTIT value is 1511 K. It should be noted that high GTIT values drastically increase the topping gas turbine cost and result in a costly power plant configuration. Moreover, it is understood that increasing the HRSG steam pressure is more cost effective as the obtained optimum value is the considered search domain’s upper limit. Another important design variable is the temperature of steam leaving the HRSG superheater. We note that increasing the generated steam temperature results in a lower steam mass flow rate in the bottoming cycle. Nonetheless, high inlet steam temperature leads to a substantial rise in the steam turbine capital investment cost. Consequently, the acquired optimum value for DOSH is 0.67.

Moreover, the thermodynamic and economic performance of the SBC power plant are of high interest. We must take into consideration that the analyses are carried out at ISO conditions. The reported overall thermal efficiency is 48.41% whereas the net specific work output, which is calculated per one kilogram of air delivered at the topping cycle compressor, is 606.1 kJ/kg. Furthermore, the required fuel mass flow rate
is 2.060 kg/s while the air and steam mass flow rates are 82.5 kg/s and 17.7 kg/s, respectively. In addition, the exhaust gases’ temperature rejected to the atmosphere is 390.0 K. Assuming that the exhaust gases can be cooled down to a minimum of 338 K, the waste heat recovery effectiveness is 93.13%. Furthermore, the bottoming cycle share of power generation is 38.8% implying that 19.4 MWe of electricity, out of the net 50 MWe, is generated by the bottoming cycle. The bottoming cycle efficiency, which is calculated as the ratio of the bottoming cycle power output over the recovered waste heat from the topping cycle, is 33.29%. Moving on to the economic analysis, topping and bottoming cycle equipment costs are 17.2 MUS$ and 26.6 MUS$, respectively. It is worth mentioning that the bottoming steam cycle requires more capital investment due to its HRSG, cooling tower, condenser, and water treatment facility. The SBC power plant total capital investment cost is estimated to be 111.3 MUS$. Finally, the plant total operating cost (optimization objective) is 0.8093 US$/s.

Table 14: Economic optimization of different combined cycle configurations

<table>
<thead>
<tr>
<th>Parameter</th>
<th>SBC</th>
<th>ABC</th>
<th>HABC</th>
<th>MBC</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Optimization result</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>TCPR</td>
<td>13.9</td>
<td>16.0</td>
<td>14.7</td>
<td>14.0</td>
</tr>
<tr>
<td>BCPR</td>
<td>1000</td>
<td>3.9</td>
<td>5.4</td>
<td>5.9</td>
</tr>
<tr>
<td>GTIT (K)</td>
<td>1511</td>
<td>1516</td>
<td>1519</td>
<td>1522</td>
</tr>
<tr>
<td>MFRR</td>
<td>-</td>
<td>1.36</td>
<td>1.22</td>
<td>1.08</td>
</tr>
<tr>
<td>ASDH</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.00</td>
</tr>
<tr>
<td>DOSH</td>
<td>0.67</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>BCAH</td>
<td>-</td>
<td>-</td>
<td>0.70</td>
<td>-</td>
</tr>
<tr>
<td><strong>Thermodynamic result</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Power plant capacity (MWe)</td>
<td>50.0</td>
<td>50.0</td>
<td>50.0</td>
<td>50.0</td>
</tr>
<tr>
<td>Overall thermal efficiency</td>
<td>48.41%</td>
<td>38.68%</td>
<td>39.52%</td>
<td>40.10%</td>
</tr>
<tr>
<td>Net specific work output (kJ/kg)</td>
<td>606.1</td>
<td>474.7</td>
<td>494.7</td>
<td>508.3</td>
</tr>
<tr>
<td>Topping cycle air mass flow rate (kg/s)</td>
<td>82.5</td>
<td>105.3</td>
<td>101.1</td>
<td>98.4</td>
</tr>
<tr>
<td>Bottoming cycle air mass flow rate (kg/s)</td>
<td>-</td>
<td>141.3</td>
<td>123.0</td>
<td>106.0</td>
</tr>
<tr>
<td>Bottoming cycle steam mass flow rate (kg/s)</td>
<td>17.67</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Injected steam/water mass flow rate (kg/s)</td>
<td>-</td>
<td>-</td>
<td>4.89</td>
<td>7.33</td>
</tr>
<tr>
<td>Fuel mass flow rate (kg/s)</td>
<td>2.060</td>
<td>2.578</td>
<td>2.523</td>
<td>2.487</td>
</tr>
<tr>
<td>Rejected exhaust gases temperature (K)</td>
<td>390.0</td>
<td>458.1</td>
<td>402.0</td>
<td>390.6</td>
</tr>
<tr>
<td>Available waste heat in the exhaust gases (MWth)</td>
<td>62.6</td>
<td>76.9</td>
<td>76.1</td>
<td>75.5</td>
</tr>
<tr>
<td>Bottoming cycle overall waste heat recovery (MWth)</td>
<td>58.3</td>
<td>62.4</td>
<td>68.7</td>
<td>69.6</td>
</tr>
<tr>
<td>Waste heat recovery effectiveness</td>
<td>93.13%</td>
<td>81.14%</td>
<td>90.28%</td>
<td>92.18%</td>
</tr>
<tr>
<td>Bottoming cycle net generated electricity (MWe)</td>
<td>19.4</td>
<td>10.6</td>
<td>12.0</td>
<td>12.9</td>
</tr>
<tr>
<td>Bottoming cycle share of electricity generation</td>
<td>38.82%</td>
<td>21.18%</td>
<td>24.09%</td>
<td>25.84%</td>
</tr>
<tr>
<td>Bottoming cycle thermal efficiency</td>
<td>33.29%</td>
<td>16.99%</td>
<td>17.53%</td>
<td>18.58%</td>
</tr>
<tr>
<td><strong>Economic result</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Plant total operating cost (US$/s)</td>
<td>0.8093</td>
<td>0.7970</td>
<td>0.7901</td>
<td>0.7799</td>
</tr>
<tr>
<td>Topping cycle equipment cost (MUS$)</td>
<td>17.169</td>
<td>22.463</td>
<td>21.263</td>
<td>20.364</td>
</tr>
<tr>
<td>Total capital investment cost (MUS$)</td>
<td>111.3</td>
<td>64.683</td>
<td>64.467</td>
<td>62.757</td>
</tr>
</tbody>
</table>

Furthermore, the economic optimization is carried out for a 50 MWe ABC power plant. The ABC optimum topping cycle has a greater pressure ratio compared...
with the SBC power plant, though both power plant configurations’ optimum GTIT values are approximately equal. In addition, the waste heat recovery is maximized by having equal cold and hot fluids’ heat capacities in the air heat exchanger. Consequently, the optimum MFRR value is found to maximize the waste heat recovery and minimize the bottoming cycle’s capital investment cost, simultaneously. Furthermore, increasing the BCPR value has contradictory effects on the ABC plant’s thermodynamic performance. On one hand, increasing the BCPR value improves the bottoming cycle efficiency. On the other hand, high BCPR values degrade the waste heat recovery capability of the bottoming cycle by delivering hotter air into the air heat exchanger.

Compared to the SBC power plant thermodynamic performance, the ABC configuration is considerably inferior. Plant overall thermal efficiency and net specific work output are 38.68% and 474.7 kJ/kg, respectively. Thus, more topping cycle air and fuel mass flow rates are required in the ABC power plant. The SBC and ABC power plants’ fuel mass flow rates are 2.060 kg/s and 2.578 kg/s, respectively. Therefore, having an SBC power plant instead of an ABC power plant can save up to 16,300 tons of natural gas annually. Noting that CH₄ is assumed for the analysis of this research work, annual CO₂ emission is reduced by about 45,000 tons. Furthermore, the rejected exhaust gases’ temperature in the ABC power plant is notably greater. In other words, air bottoming cycle waste heat’s recovery ability is not as efficient as the steam bottoming cycle due to its higher operating temperature. Furthermore, the bottoming cycle in the ABC configuration is only responsible for 10.6 MWe (21.2%) of the total 50 MWe power output whereas the bottoming cycle in the SBC power plant’s share of contribution is about 38.8%. Additionally, ABC’s bottoming cycle efficiency is only 15.35%. Therefore, only 15.35% of the recovered waste heat is converted into electricity and the rest is wasted. In general, there is not even a single thermodynamic indicator in which the ABC power plant is superior compared to the SBC power plant.

Nevertheless, ABC’s economic analysis indicates its great potential for small scale power generation. Though more capital investment is required for the ABC’s topping cycle due to the greater air mass flow rate and pressure ratio, the bottoming cycle’s simplicity and inexpensiveness improves the ABC’s economic performance significantly. It is worth mentioning that the total capital investment for installment of
An ABC power plant is 47.5 MUS$ less than the necessary investment for an SBC power plant installment. In conclusion, the ABC power plant’s total operating cost is 0.7970 US$/s whereas the SBC total capital investment is 0.8093 US$/s. As a result, the ABC configuration is a more economical alternative for a 50 MWe power plant. Additionally, it should be mentioned that constructing an ABC power plant can save up to 9.7 MUS$ throughout the twenty-five years of plant operation.

Another power plant configuration investigated in this research work is the HABC. In other words, water injection in the ABC’s bottoming cycle air stream is studied thermodynamically and economically. Water injection in the bottoming cycle reduces the optimum TCPR value to 14.7. Moreover, the lower bottoming cycle air mass flow rate is required for heat recovery due to the injected water and humid air greater specific heat capacity. In addition, water injection abates the air temperature entering the air heat exchanger. As a result, greater BCPR values can be utilized without degrading the bottoming cycle heat recovery capacity.

Overall, the ABC power plant thermodynamic performance can be considerably enhanced by the proposed water injection in the bottoming cycle. As can be seen from the acquired results, there is a significant enhancement in the plant’s thermal efficiency and net specific work output. Additionally, the plant’s nominal air mass flow rate is lessened, resulting in a more compact plant. The main advantage of the HABC configuration is its capability to further recover the waste heat available in the topping cycle exhaust gases by cooling and humidifying the bottoming cycle air stream in the devised evaporator. Consequently, the exhaust gases can be further cooled down to a notably lower temperature. In fact, the main impact of water injection in the bottoming cycle is evident on bottoming cycle performance. With water injection, the bottoming cycle net power output and share of power generation are enhanced to 12.0 MWe and 24.09%, respectively. While the bottoming cycle is capable of recovering a greater portion of the waste heat, it is also capable of converting a greater portion of the recovered heat into electricity as the bottoming cycle efficiency is improved to 17.53%.

Furthermore, water injection in the ABC power plant’s bottoming cycle improves its economic performance. In other words, the aforementioned enhancement in the plant’s waste heat recovery ability results in a reduction in the required initial investment. Results indicate that the humid bottoming cycle requires more investment.
Nevertheless, both the topping cycle equipment cost and plant total capital investment are decreased by 1.20 MUS$ and 0.22 MUS$, respectively. Therefore, the plant’s total operating cost is decreased by 0.9% to 0.7901 US$/s. It should be noted that water injection in ABC power plants can save up to 5.4 MUS$ throughout their twenty-five years of operation.

The last combined cycle configuration studied in this research work is the MBC. It is noteworthy that MBC and ABC power plants are somewhat identical except for their approaches in waste heat recovery. The air heat exchanger, which is devised for waste heat recovery in ABC power plants, is replaced by a duplex heat exchanger (air saturator) utilized in Maisotsenko gas turbine cycles. One of the advantages of the air saturator is its capability to cool down the air departing the bottoming cycle compressor to improve the bottoming cycle heat recovery. Thus, the bottoming cycle compressor and turbine can be designed with greater pressure ratios. Additionally, a lower bottoming cycle air mass flow rate is required due to the higher humid air specific heat capacity. Moreover, the optimum ASDH value is zero, implying that water addition is only beneficial for air cooling purposes in the lower section of the air saturator. In the upper section of the air saturator, no water addition is advised to fully maximize the bottoming cycle air temperature departing the air saturator.

The MBC power plant thermodynamic results indicate its superiority over ABC and HABC configurations. Compared with the ABC power plant, MBC’s overall thermal efficiency and net specific work output are augmented by 1.42% point and 7.1%, respectively. It should be noted that MBC’s efficiency and net specific work output are second only to SBC power plants. Furthermore, a greater amount of water is added to the bottoming cycle air in the MBC configuration compared to HABC power plants. It is important to bear in mind that one of the main advantages of the MBC configuration over HABC is its ability to control the injected water mass flow rate and air temperature. Consequently, the degree of air humidification and superheating can be easily regulated by the amount of water provided in the upper and lower sections of the air saturator. The increase in the plant’s thermal efficiency suggests that the fuel mass flow rate is reduced by 3.5% from 2.578 kg/s to 2.487 kg/s. Hence, replacing the air heat exchanger in ABC power plants with the air saturator will save up to 2870 tonne of CH$_4$ and prevent about 7900 tonne of CO$_2$ from being emitted to the atmosphere,
annually. Moreover, the MBC configuration employs a more capable waste heat recovery approach as the topping cycle exhaust gases are cooled down to a significantly lower temperature with a more effective waste heat recovery capacity. The bottoming cycle share of power generation is improved to 25.8%. Finally, bottoming cycle thermal efficiency is enhanced by 1.59% points due to the employment of the air saturator instead of a conventional heat recovery heat exchanger.

Additionally, economic performance of the MBC power plant must be evaluated. Results indicate that the MBC power plant requires the lowest initial investment among all four studied combined cycle configurations. Though the bottoming cycle equipment cost of the MBC power plant is greater than the ABC, its superior thermal efficiency and net specific work output lead to a cheaper topping cycle. Consequently, MBC displays the best economic performance with the lowest total operating cost of 0.7799 US$/s. Therefore, one can conclude that replacing the conventional heat recovery heat exchanger with an air saturator can save about 13.5 MUS$ during twenty-five years of operation.

Power plant efficiency enhancement is mainly accompanied by an increase in its initial capital investment. In other words, more investment is required to improve a plant’s thermal efficiency. To further evaluate the abovementioned trade-off between the plant’s total operating cost and thermal efficiency, a multi-objective optimization is carried out. The optimization objectives are to maximize the plant thermal efficiency and minimize its total operating cost, simultaneously. MATLAB provides a strong multi-objective genetic algorithm built-in function which is employed in this research work. The multi-objective genetic algorithm approach implements the Pareto dominance concept to identify sets of optimum operating conditions. The Pareto dominance concept is achieved when a set of solutions cannot be improved in any objective without degrading at least one other objective [85].

The Pareto optimal fronts concerning SBC, ABC, HABC, and MBC configurations are depicted in Figure 52. Results indicate that SBC power plants can operate with a significantly higher thermal efficiency at an identical total operating cost. Nonetheless, the minimum attainable total operating cost is lower for ABC, HABC and MBC power plants; thus, these configurations are more economical. Furthermore, results clearly show the advantages of the air saturator integration in ABC power plants.
both economically and thermodynamically. By comparing the presented results for MBC and ABC plant configurations, one may conclude that MBC is capable of achieving a higher thermal efficiency at an identical total operating cost. Furthermore, the presented Pareto fronts for ABC and HABC configurations depict the effectiveness of water injection in the ABC bottoming cycle. It should be noted that the presented result is a confirmation that water injection in the bottoming cycle air improves ABC’s thermal and economical performance. Nonetheless, the positive effect of the air saturator integration is more evident than the evaporator incorporation in the HABC configuration. In general, SBC power plants can operate with the highest thermal efficiency whereas an optimized MBC power plant configuration can be the most economical alternative followed by the HABC configuration.

Figure 52: Pareto optimal fronts for proposed power plant combined configurations

4.2.2. Heat recovery bottoming cycle economic analysis.

In this section, an already existing gas turbine power plant is taken into consideration. Implementation of different bottoming cycles, i.e. air bottoming cycle (ABC), steam bottoming cycle (SBC), Maisotsenko bottoming cycle (MBC), and humid air bottoming cycles (HABC), are investigated. Bottoming cycles are integrated to recover a portion of the waste heat available in the simple gas turbine cycle exhaust and generate extra electricity to enhance the plant’s overall thermal efficiency. For the thermo-economic analysis, the aforementioned life cycle savings method is considered by assessing the additional capital investment, maintenance, and operation expenses of the bottoming cycles against the additional electricity production revenue on an annual
basis. Consequently, the proposed investment in the form of a bottoming cycle integration life cycle revenue or loss is determined. One should note that the variations in ambient air conditions, i.e. temperature and relative humidity, are not taken into account in this section and the assessment is carried out at ISO standards. To better understand the effect of the simple gas turbine cycle capacity on its most profitable waste heat recovery bottoming cycle option, two already existing simple gas turbine cycles with different capacities are evaluated. The main characteristics of the considered gas turbine plants are presented in Table 15, bearing in mind that a similar approach to the optimization method utilized in the previous section is considered to perform a thermo-economic optimization for the aforementioned scenarios. The optimization objective is to maximize the plant’s life cycle saving. Moreover, the selected design variables and their corresponding search domains are listed in Table 13.

Table 15: Reference gas turbine power plants main characteristics

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Case 1</th>
<th>Case 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure ratio</td>
<td>14</td>
<td>14</td>
</tr>
<tr>
<td>Air mass flow rate (kg/s)</td>
<td>115.5</td>
<td>28.9</td>
</tr>
<tr>
<td>Turbine inlet temperature (K)</td>
<td>1500</td>
<td>1500</td>
</tr>
<tr>
<td>Net specific work output (kJ/kg)</td>
<td>373.7</td>
<td>373.7</td>
</tr>
<tr>
<td>Net power output (MWe)</td>
<td>43.161</td>
<td>10.790</td>
</tr>
<tr>
<td>Fuel mass flow rate (kg/s)</td>
<td>2.8386</td>
<td>0.7096</td>
</tr>
<tr>
<td>Overall thermal efficiency</td>
<td>30.32%</td>
<td>30.32%</td>
</tr>
<tr>
<td>Turbomachinery equipment cost (MUS$)</td>
<td>21.997</td>
<td>7.436</td>
</tr>
<tr>
<td>Total capital cost (MUS$)</td>
<td>42.793</td>
<td>14.662</td>
</tr>
</tbody>
</table>

Initially, thermo-economic optimization results for different bottoming cycle implementations in the 43.2 MWe simple gas turbine power plant are tabulated in Table 16. Integration of the SBC expanded the plant nominal power output at ISO standard by 25.4 MWe. One may notice that SBC incorporation provided the maximum plant capacity expansion whereas MBC, HABC, and ABC additional generated powers are 13.6 MWe, 13.1 MWe, and 11.3 MWe, respectively. Accordingly, plant thermal efficiency enhancement due to the addition of the SBC is the greatest followed by MBC, HABC, and ABC, sequentially. Nevertheless, the maximum amount of heat recovered by the bottoming cycle is associated with the HABC configuration. It should be noted that integration of a bottoming cycle degrades the topping cycle performance, marginally. The aforementioned degradation in the topping cycle performance is due to the extra pressure drop in the air heat exchanger or HRSG. One should note that the topping cycle net power output reduction can be determined by calculating the
difference between the plant capacity expansion and bottoming cycle net power output. Consequently, topping cycle net power output reductions for all the proposed bottoming cycles are about 1.0 MWe.

Table 16: Thermo-Economic optimization of different bottoming cycle integrations for case I

<table>
<thead>
<tr>
<th>Parameter</th>
<th>SBC</th>
<th>ABC</th>
<th>HABC</th>
<th>MBC</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Optimization result</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>BCPR</td>
<td>1000</td>
<td>4.2</td>
<td>5.8</td>
<td>6.2</td>
</tr>
<tr>
<td>MFRR</td>
<td>-</td>
<td>1.34</td>
<td>1.19</td>
<td>1.06</td>
</tr>
<tr>
<td>ASDH</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.00</td>
</tr>
<tr>
<td>DOSH</td>
<td>0.66</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>BCAH</td>
<td>-</td>
<td>-</td>
<td>0.86</td>
<td>-</td>
</tr>
<tr>
<td><strong>Thermodynamic result</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Plant capacity expansion (MWe)</td>
<td>25.428</td>
<td>11.292</td>
<td>13.050</td>
<td>13.599</td>
</tr>
<tr>
<td>Overall thermal efficiency</td>
<td>48.19%</td>
<td>38.26%</td>
<td>39.49%</td>
<td>39.88%</td>
</tr>
<tr>
<td>Net specific work output (kJ/kg)</td>
<td>593.8</td>
<td>471.5</td>
<td>486.7</td>
<td>491.4</td>
</tr>
<tr>
<td>Bottoming cycle steam mass flow rate (kg/s)</td>
<td>24.28</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Injected steam/water mass flow rate (kg/s)</td>
<td>-</td>
<td>-</td>
<td>7.08</td>
<td>8.77</td>
</tr>
<tr>
<td>Bottoming cycle overall waste heat recovery (MWth)</td>
<td>79.700</td>
<td>68.817</td>
<td>80.061</td>
<td>79.322</td>
</tr>
<tr>
<td>Waste heat recovery effectiveness</td>
<td>92.42%</td>
<td>79.80%</td>
<td>92.84%</td>
<td>91.98%</td>
</tr>
<tr>
<td>Bottoming cycle net power output (MWe)</td>
<td>26.460</td>
<td>12.324</td>
<td>14.081</td>
<td>14.630</td>
</tr>
<tr>
<td>Bottoming cycle thermal efficiency</td>
<td>33.20%</td>
<td>17.91%</td>
<td>17.59%</td>
<td>18.44%</td>
</tr>
<tr>
<td><strong>Economic result</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Life cycle saving (MUS$)</td>
<td>70.726</td>
<td>50.648</td>
<td>56.286</td>
<td>60.702</td>
</tr>
<tr>
<td>Payback periods (years)</td>
<td>6.56</td>
<td>3.85</td>
<td>3.84</td>
<td>3.55</td>
</tr>
<tr>
<td>Bottoming cycle equipment cost (MUS$)</td>
<td>34.746</td>
<td>15.199</td>
<td>17.173</td>
<td>16.119</td>
</tr>
<tr>
<td>Total annual profit (MUS$)</td>
<td>13.884</td>
<td>6.486</td>
<td>7.208</td>
<td>7.489</td>
</tr>
<tr>
<td>Additional capital investment cost (MUS$)</td>
<td>91.071</td>
<td>24.942</td>
<td>27.714</td>
<td>26.573</td>
</tr>
</tbody>
</table>

Economic analysis of the integration of a bottoming cycle for a simple gas turbine power plant is accomplished by utilizing two approaches, i.e. life cycle savings and payback period. Each economic approach provides a specific insight on different bottoming cycles’ performance for waste heat recovery. As expected, SBC equipment cost and additional capital investment are considerably greater than the other three investigated bottoming cycles, noting that SBC’s additional capital investment is almost four times greater than the other configurations. Among the other three configurations, ABC and HABC have the lowest and highest additional capital investment, respectively. Nonetheless, annual investment profit, which is the difference between the additional generated electricity sale revenue and extra operating and maintenance cost, is the greatest for SBC with 13.9 MUS$/year. Moreover, MBC, HABC, and ABC integrations result in notably inferior annual profit with 7.5 MUS$/year, 7.2 MUS$/year, and 6.5 MUS$/year, respectively. Finally, the number of years it takes to reimburse the additional capital investment for the bottoming cycle integration is the shortest for MBC implementation with only 3.55 years. In addition,
ABC and HABC configurations’ payback periods are 3.85 years and 3.84 years. Nevertheless, the SBC incorporation payback period is the longest with 6.56 years. Results concerning the plant life cycle saving is telling a different story. As the plant will be operational for 25 years, life cycle savings provide a more accurate economic assessment. Note that the payback period is significantly shorter than 25 years; thus, the payback period approach does not provide a complete picture. The maximum amount of life cycle savings is reported for SBC integration with 70.7 MUS$. Additionally, MBC, HABC, and ABC incorporation in a simple gas turbine power plant leads to 60.7 MUS$, 56.3 MUS$, and 50.6 MUS$ savings during the plant’s 25 years of operation, respectively.

To investigate the significance of the reference gas turbine power plant capacity, a smaller gas turbine power plant with capacity of 10.8 MWe is considered to assess the feasibility of a bottoming cycle integration. Thermo-economic optimization is accomplished for different bottoming cycle integrations comprising SBC, ABC, HABC, and MBC. The optimization results for the investigated bottoming cycles are listed in Table 17 are relatively similar to the results listed in Table 16, though, in a considerably lower scale. SBC integration leads to the greatest plant capacity expansion, overall thermal efficiency, and net specific work output. It should be noted that the available waste heat in the exhaust gases is identical for all the investigated bottoming cycles. Accordingly, MBC provides the most effective heat recovery capability. Nonetheless, the SBC configuration is capable of converting a greater portion of the recovered waste heat into electricity with the bottoming cycle thermal efficiency of 33.34%.

Economic performances of the bottoming cycle integration are seemed to be of higher interest for the presented results. Understandably, additional investment required for the bottoming cycle incorporation reduces as the reference gas turbine power plant capacity abates. In other words, bottoming cycle capacity and the required additional capital investment are proportional to the reference plant capacity. Nevertheless, the additional capital investment necessary for SBC integration in a 10 MWe plant is greater than the required investment for the incorporation of ABC, HABC, and MBC in a 43 MWe plant. Moreover, MBC integration leads to the lowest extra investment whereas SBC requires the highest investment. In general, economic analyses indicate
that the integrating of a bottoming cycle is a more cost effective option for relatively large gas turbine power plants. Noting that the payback periods for all investigated plants are extended for the case II power plant compared with case I. Furthermore, the payback period for the MBC addition is the shortest with only 5.22 years while SBC requires 8.94 years to reimburse its initial investment. It should be noted that the greatest amount of life cycle savings for the case I power plant belongs to SBC. Nonetheless, MBC replaces SBC with the highest life cycle savings of 11.492 MUS$ for the case II plant. Consequently, it is more profitable to integrate an MBC in small scale power plants (about 10 MWe) followed by HABC, SBC, and ABC, sequentially. Though, it should be noted that the payback period analysis provides a different ranking of the bottoming cycles. Based on the payback period approach, MBC is the most cost effective bottoming cycle followed by HABC, ABC, and SBC, respectively. Thus, it is safe to say that MBC integration is the most cost effective alternative regardless of the utilized economic assessment approaches.

Table 17: Thermo-Economic optimization of different bottoming cycle integrations for case II

<table>
<thead>
<tr>
<th>Parameter</th>
<th>SBC</th>
<th>ABC</th>
<th>HABC</th>
<th>MBC</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Optimization result</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>BCPR</td>
<td>1000.0</td>
<td>4.0</td>
<td>5.4</td>
<td>5.9</td>
</tr>
<tr>
<td>MFRR</td>
<td>-</td>
<td>1.40</td>
<td>1.24</td>
<td>1.08</td>
</tr>
<tr>
<td>ASDH</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.00</td>
</tr>
<tr>
<td>DOSH</td>
<td>0.70</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>BCAH</td>
<td>-</td>
<td>-</td>
<td>0.63</td>
<td>-</td>
</tr>
<tr>
<td><strong>Thermodynamic result</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Plant capacity expansion (MWe)</td>
<td>6.355</td>
<td>2.732</td>
<td>3.041</td>
<td>3.334</td>
</tr>
<tr>
<td>Overall thermal efficiency</td>
<td>48.18%</td>
<td>38.00%</td>
<td>38.87%</td>
<td>39.69%</td>
</tr>
<tr>
<td>Net specific work output (kJ/kg)</td>
<td>593.8</td>
<td>468.3</td>
<td>479.0</td>
<td>489.2</td>
</tr>
<tr>
<td>Bottoming cycle steam mass flow rate (kg/s)</td>
<td>5.99</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Injected steam/water mass flow rate (kg/s)</td>
<td>-</td>
<td>-</td>
<td>1.27</td>
<td>2.16</td>
</tr>
<tr>
<td>Bottoming cycle overall waste heat recovery (MWth)</td>
<td>19.836</td>
<td>17.521</td>
<td>19.084</td>
<td>19.883</td>
</tr>
<tr>
<td>Waste heat recovery effectiveness</td>
<td>92.01%</td>
<td>81.27%</td>
<td>88.52%</td>
<td>92.22%</td>
</tr>
<tr>
<td>Bottoming cycle net power output (MWe)</td>
<td>6.612</td>
<td>2.990</td>
<td>3.299</td>
<td>3.592</td>
</tr>
<tr>
<td>Bottoming cycle thermal efficiency</td>
<td>33.34%</td>
<td>17.06%</td>
<td>17.29%</td>
<td>18.07%</td>
</tr>
<tr>
<td><strong>Economic result</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Life cycle saving (MUS$)</td>
<td>9.058</td>
<td>7.943</td>
<td>9.456</td>
<td>11.492</td>
</tr>
<tr>
<td>Payback periods (years)</td>
<td>8.94</td>
<td>6.38</td>
<td>5.89</td>
<td>5.22</td>
</tr>
<tr>
<td>Bottoming cycle equipment cost (MUS$)</td>
<td>11.300</td>
<td>6.201</td>
<td>6.159</td>
<td>5.807</td>
</tr>
<tr>
<td>Total annual profit (MUS$)</td>
<td>3.338</td>
<td>1.507</td>
<td>1.641</td>
<td>1.787</td>
</tr>
</tbody>
</table>

4.2.3. Transient analysis and levelized cost of electricity optimization.

In this section, the LCOE approach is considered to fully evaluate the proposed combined cycle configurations. In order to conduct the LCOE analysis, transient analysis must be carried out. Transient analysis provides an interesting set of information since gas turbine performances are strongly dependent on the ambient air
temperature [30]. Additionally, humid air turbine cycles, which include HABC and MBC configurations presented in this research work, are affected by the variation in the ambient air relative humidity [8]. Therefore, transient analysis can give us valuable information, in particular, for a hot and humid climate such as in the UAE.

In order to carry out the transient analysis, UAE weather data are utilized to appropriately assess the considered configurations’ performance. In the developed model, steady state analyses are conducted based on the assigned design variables presented in Table 13. For ABC, HABC, and MBC power plants, steady state model calculates the nominal topping and bottoming cycle air mass flow rates at ISO condition. Consequently, transient analysis can be performed by calculating the air mass flow rates based on the ambient air temperature at that specific moment. Additionally, it is considered that the amount of water injected in HABC and MBC power plants is constantly adjusted, based on the ambient air temperature and relative humidity, to keep the air condition leaving the saturator (BCAH and ASDH) constant, taking into consideration that the injected water mass flow rate is relatively insignificant and the aforementioned adjustment can be easily accomplished similar to the variation in the combustion chamber’s fuel mass flow rate. Furthermore, the temperature of water injected in MBC and HABC power plants is equal to ambient air temperature. Similarly, transient analysis is carried out for the SBC power plant to determine the topping cycle air mass flow rate and bottoming cycle steam mass flow rate at ISO conditions.

The optimization results for the investigated combined cycle power plant utilizing LCOE as the objective are presented in Table 18. The reported optimum design variables for different configurations are relatively similar to the previously presented results. Therefore, one can conclude that the selected economic approach and objective for the thermo-economic optimization has an insignificant impact on the plant’s optimum operating conditions. The most notable difference between the presented results is the BCAH optimum value. Noting that transient analysis is conducted in this section by taking into account the effects of ambient air conditions (temperature and relative humidity), it can be perceived that the presented results for LCOE optimization are more accurate and trustworthy. Expectedly, the SBC power plant can generate a greater amount of electricity throughout a complete year than other considered configurations. Therefore, the SBC configuration has the highest annual overall efficiency of 48.20%. It is important to note that MBC has the second highest thermal
efficiency followed by HABC and ABC configurations. Similarly, SBC plant annual fuel consumption and specific CO\(_2\) emission are considerably lower than the other configurations. By utilizing the SBC power plant configuration instead of an MBC, HABC, or ABC power plant, annual fuel consumption is abated by 13,500 tonne, 14,300 tonne, or 16,200 tonne, respectively. Furthermore, the SBC power plant bottoming cycle has the biggest share of power generation with 41.10%, whereas, MBC configurations’ bottoming cycle share of power generation, which is the second best, is only 27.83%.

Table 18: Economic optimization of different 50 MWe combined cycle configurations

<table>
<thead>
<tr>
<th>Parameter</th>
<th>SBC</th>
<th>ABC</th>
<th>HABC</th>
<th>MBC</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Optimization result</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>TCPR</td>
<td>12.6</td>
<td>14.4</td>
<td>12.8</td>
<td>11.8</td>
</tr>
<tr>
<td>BCPR</td>
<td>1000</td>
<td>3.8</td>
<td>5.6</td>
<td>6.2</td>
</tr>
<tr>
<td>GTIT (K)</td>
<td>1506</td>
<td>1512</td>
<td>1510</td>
<td>1523</td>
</tr>
<tr>
<td>MFRR</td>
<td>-</td>
<td>1.23</td>
<td>1.12</td>
<td>1.04</td>
</tr>
<tr>
<td>ASDH</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.00</td>
</tr>
<tr>
<td>DOSH</td>
<td>0.63</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>BCAH</td>
<td>-</td>
<td>-</td>
<td>0.97</td>
<td>-</td>
</tr>
<tr>
<td><strong>Thermodynamic result</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Annual electricity output (GWhe)</td>
<td>410.4</td>
<td>399.2</td>
<td>401.9</td>
<td>404.1</td>
</tr>
<tr>
<td>Annual overall energy efficiency</td>
<td>48.20%</td>
<td>37.06%</td>
<td>38.25%</td>
<td>38.86%</td>
</tr>
<tr>
<td>Specific CO(_2) emission (kgCO(_2)/MWhe)</td>
<td>409.6</td>
<td>532.8</td>
<td>516.1</td>
<td>508.0</td>
</tr>
<tr>
<td>Specific water consumption (ltrH(_2)O/MWhe)</td>
<td>42.33</td>
<td>0.27</td>
<td>539.68</td>
<td>593.33</td>
</tr>
<tr>
<td>Annual fuel consumption (tonne)</td>
<td>61131</td>
<td>77349</td>
<td>75425</td>
<td>74650</td>
</tr>
<tr>
<td>Bottoming cycle share of electricity generation</td>
<td>41.10%</td>
<td>21.18%</td>
<td>25.46%</td>
<td>27.83%</td>
</tr>
<tr>
<td><strong>Economic result</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Annual operating cost (MUS$)</td>
<td>14.499</td>
<td>18.321</td>
<td>18.165</td>
<td>18.013</td>
</tr>
<tr>
<td>Bottoming cycle cost (MUS$)</td>
<td>27.418</td>
<td>13.307</td>
<td>15.496</td>
<td>14.496</td>
</tr>
<tr>
<td>Total capital cost (MUS$)</td>
<td>111.07</td>
<td>62.855</td>
<td>63.572</td>
<td>61.455</td>
</tr>
<tr>
<td>Net present value (MUS$)</td>
<td>5.038</td>
<td>15.847</td>
<td>18.653</td>
<td>24.637</td>
</tr>
<tr>
<td>LCOE (US$/MWh)</td>
<td>68.88</td>
<td>66.36</td>
<td>65.75</td>
<td>64.41</td>
</tr>
</tbody>
</table>

It is a well-known fact that SBC configurations are thermodynamically superior over the other proposed configurations. Thus, it is more interesting to compare the presented results for the other three configurations to comprehend the effect of water injection and air saturator integration in ABC configurations. The presented results indicate that water injection can improve an ABC power plant’s annual overall thermal efficiency and bottoming cycle share of generation by 1.19% points and 4.28% points, respectively. Furthermore, the plant’s annual fuel consumption dropped by 2.49%. In addition, the rate of CO\(_2\) emitted for generating 1 MWh of electricity is reduced by 16.7 kgCO\(_2\); whereas, 539.4 liters of additional water is utilized. Likewise, a comparative analysis between MBC and ABC power plants can be conducted. Employing an air saturator instead of a simple heat recovery heat exchanger enhances the plant’s thermal
efficiency and bottoming cycle share of power generation by 1.80% points and 6.65%, respectively. Accordingly, the plant’s annual fuel consumption is reduced by 3.49%. Additionally, the plant’s specific CO₂ emission is decreased by 24.8 kgCO₂, while specific water consumption is increased by 593.1 ltrH₂O.

Economic analysis is provided to further study the advantages and disadvantages of different waste heat recovery bottoming cycles. It is important to note that the SBC power plant configuration has the lowest operating cost due to its lower annual fuel consumption. Nevertheless, initial capital investment and bottoming cycle’s equipment cost tell a different story. Initial capital investment for the SBC power configuration is almost twice the required capital investment for the other configurations, noting that the MBC configuration requires the minimum initial investment. On the other hand, ABC’s bottoming cycle equipment cost is the least among the investigated plants. Finally, LCOE and net present value analyses indicate that MBC is the most cost effective power plant configuration followed by HABC, ABC, and SBC configurations. Consequently, one can conclude that water injection in ABC power plants can improve the plant’s LCOE by 0.9% which leads to 2.8 MUS$ of profit throughout twenty-five years of plant operation (based on the net present value results). Similarly, integration of an air saturator instead of a simple heat exchanger reduces the plant’s LCOE by 2.9%. Accordingly, 8.8 MUS$ can be saved by utilizing the air saturator for heat recovery purposes.

4.2.4. Concluding remarks.

In this section, four waste heat recovery bottoming cycles were selected to be incorporated within simple gas turbine power plants. In other words, thermo-economic analysis and optimization were carried out for four combined cycle configurations including BC, ABC, MBC, and HABC. Primarily, the economic evaluation approach proposed by Knopf [135] was employed to identify the most economically justified 50 MWe combined cycle power plant. Afterward, a life cost saving analysis was conducted to investigate the most thermo-economically efficient bottoming cycle for implementation in an already existing simple gas turbine power plant. Furthermore, the LCOE method was applied to study the proposed combined cycle configurations thermal and economical performances in hot and humid climates.
In general, SBC power plants can operate with the highest thermal efficiency whereas an optimized MBC power plant configuration can be the most economical alternative followed by the HABC configuration. Utilizing Knopf’s [135] assessment approach, MBC displays the best economic performance with the lowest total operating cost of 0.7799 US$/s followed by HABC, ABC, and SBC power plant configurations with total operating costs of 0.7901 US$/s, 0.7970 US$/s, and 0.8093 US$/s, respectively. On the other hand, the life cycle saving analysis for the integration of a waste heat recovery bottoming cycle within a 43.2 MWe gas turbine power plant indicate that SBC incorporation leads to the maximum revenue. Whereas, similar analysis for a 10.8 MWe gas turbine presents MBC implementation as the most cost effective alternative. Finally, transient analysis shows that MBC is the most cost effective power plant configuration for hot and humid climates with LCOE of 64.41 US$/MWh followed by HABC, ABC, and SBC with reported LCOE values of 65.75 US$/MWh, 66.36 US$/MWh, and 68.88 US$/MWh, respectively.

4.3. Hybrid Steam Bottoming Cycle Thermo-Economic Optimization

In this section a hybrid SBC power plant is comprehensively investigated. Initially, the thermo-economic procedure which is utilized for the optimization of the power plant is presented. Afterward, the thermo-economic optimization results are given and discussed. Finally, hybridization of an already existing SBC power plant is fully evaluated. The results presented in this section are published in [161].

4.3.1. Thermo-economic optimization procedure.

A rather complicated procedure must be carried out for thermo-economic optimization of hybrid power plants. Therefore, a simplified flow chart is provided in Figure 53 depicting the necessary steps for thermo-economic optimization of the hybrid SBC power plant. Initially, the heliostat field design and optimization must be conducted. In the first step, the aforementioned heliostat field design constraints, listed in Table 6, and meteorological data [123] are fed to the heliostat field optimizer. The heliostat field optimizer utilizes the optimization method presented by Collado [115] and Collado and Guallar [56, 116] to maximize the field annual thermal performance. Afterwards, coordinates of each mirror associated with the optimum field layout and an array of the ranked mirrors based on their annual thermal performances are obtained.
Simultaneously, initial design variables for the power plant are considered for the steady state analysis. Steady state analysis is a necessary step within the thermo-economic analysis since it provides several pieces of important information such as the required solar heating, tower piping dimensions, air and steam mass flow rates at ISO conditions, and power plant components’ initial investment. Required solar heating and attained information from the heliostat optimizer are employed to determine the heliostat field sizing. With heliostat field sizing, number and coordinates of the heliostats required to obtain a specific thermal output at a specific time of the year are determined. Afterward, heliostat field transient analysis is performed to evaluate its performance for a complete year. Additionally, the initial investment required for the heliostat field components is calculated. In the next step, acquired information regarding the heliostat field’s annual performance and power plant steady state analysis are used to carry out the plant transient analysis. Power plant transient analysis determines annual electricity generation, annual fuel and water consumption, and annual CO₂ emission. Consequently, operating cost and plant LCOE can be calculated.
The abovementioned complexity in hybrid power plant optimization prevents the utilization of traditional gradient based optimization methods. Non-linearity, discontinuity, and differentiability are the most challenging factors in hybrid power plant optimization. As a result, the genetic algorithm is the most appropriate technique for the optimization of these types of problems. The genetic algorithm has several considerable advantages for energy system optimizations. In the genetic algorithm, only the objective function value is required. In addition, objective function continuity and differentiability are not necessary [159]. MATLAB provides a strong optimization toolbox containing different built-in optimizers. One of the optimization approaches presented by MATLAB is the genetic algorithm [160]. Therefore, the hybrid power plant model (steady state and transient analysis of the heliostat field and power block) is evaluated by black box approach [122]. In other word, the thermo-economic evaluator and MATLAB optimizer only exchange information regarding the design variables and objective function values.

The thermo-economic optimization objective is the plant LCOE while four design variables are considered including gas and steam turbine cycles pressure ratios, topping cycle turbine inlet temperature, and steam degree of superheating. Therefore, the MATLAB genetic algorithm provides a set of values for the selected design variables to the thermo-economic model. Afterward, the thermo-economic model determines the plant LCOE and feeds it back to the MATLAB optimizer. To improve the accuracy of the optimization and make sure that the genetic algorithm has provided the global optimum not the local optimum, it was decided to utilize the optimum point presented by the genetic algorithm in another MATLAB optimizer in the form of hybrid optimization (two successive approaches of optimization). A MATLAB code was written utilizing the line search approach presented by Ramos and Ramos [119]. The recommended method is developed based on an algorithm by Powell [125]. The optimization must begin with an initial approximation of the optimum point. Hence, the optimum point attained by the MATLAB built-in functions is employed as the starting point for the developed line search optimization. Initially, the optimum value for the first design variable is determined. Similar steps were taken to calculate all design variables’ optimum values and completing the first iteration. Several iterations were carried out until acceptable precision in the optimum values of all the design variables were achieved.
Based on the results presented for different heliostat field layouts, it was decided to consider the heliostat field layout with maximum weighted efficiency for analyzing the hybrid configurations. The optimum radial-staggered heliostat field layout is depicted in Figure 40. The field weighted efficiency is 58.61% with 197.7 GWh of thermal energy delivered annually at the receiver surface. Moreover, the heliostat field unweighted efficiency and the required capital investment are 54.90% and 52.57 MUS$, respectively. In addition, the field annual shading and blocking, spillage, cosine, and attenuation factors are 92.92%, 94.04%, 76.98%, and 92.73%, respectively. However, this is not the final layout of the heliostat field which will be utilized for the SBC plant hybridization. It is common to begin the analysis of the heliostat field collectors with a field containing a considerably greater number of mirrors. Afterward, the heliostats (mirrors) within the field are sorted by their annual thermal performance. Within the plant hybridization procedure, a subset of the mirrors displaying the best annual thermal performances are selected based on the required plant solar share. Typically, spring equinox noon is selected as the design time for the field. Starting from the most efficient mirror, the power provided by each heliostat is added in sequence to reach the required power for the plant operation.

4.3.2. Thermo-economic optimization results.

Thermo-economic optimization for construction of a new hybrid SBC power plant in Abu Dhabi was conducted. The optimization was carried out with the plant’s LCOE minimization as its objective. The previously discussed multi-stage procedures for thermo-economic analysis and optimization of a hybrid power plant was utilized to acquire the minimum LCOE for the plant’s operation in Abu Dhabi. The optimization design variables with their respective assigned search domains are listed in Table 19.

Table 19: Design variables and their respective search domains

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Assigned search domain</th>
</tr>
</thead>
<tbody>
<tr>
<td>Topping cycle pressure ratio (TCPR)</td>
<td>10.0 — 25.0</td>
</tr>
<tr>
<td>Bottoming cycle pressure ratio (BCPR)</td>
<td>500 — 1000</td>
</tr>
<tr>
<td>Gas turbine inlet temperature (GTIT)</td>
<td>1223 K — 1673 K</td>
</tr>
<tr>
<td>Degree of humidification (DOsh)</td>
<td>0.10 — 1.00</td>
</tr>
</tbody>
</table>

The thermo-economic optimization results for the hybrid SBC power plant in Abu Dhabi, UAE, are tabulated in Table 20. Initially, the selected design variables optimum values are discussed. One should note that increasing the TCPR value improves the topping cycle efficiency and degrades the available waste heat for the
bottoming cycle recovery. Additionally, gas turbine cycles with greater pressure ratios require significantly higher capital investment. Due to the contradictory impacts of TCPR on plant performance, a moderate pressure ratio is reported to provide the most cost effective plant. For the bottoming cycle pressure ratio (BCPR) analysis, one may conclude that increasing the HRSG steam pressure constantly enhances the plant’s economic performance as the assigned upper limit is determined to be the optimum value.

Table 20: Hybrid ABC power plant thermo-economic optimization results

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Results</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Optimization result</strong></td>
<td></td>
</tr>
<tr>
<td>TCPR</td>
<td>12.7</td>
</tr>
<tr>
<td>BCPR</td>
<td>1000</td>
</tr>
<tr>
<td>GTIT (K)</td>
<td>1519</td>
</tr>
<tr>
<td>DOSH</td>
<td>0.63</td>
</tr>
<tr>
<td><strong>Power plant thermodynamic result</strong></td>
<td></td>
</tr>
<tr>
<td>Annual electricity output (GWhe)</td>
<td>419.27</td>
</tr>
<tr>
<td>Annual overall thermal efficiency</td>
<td>47.16%</td>
</tr>
<tr>
<td>Specific CO₂ emission (kgCO₂/MWhe)</td>
<td>371.9</td>
</tr>
<tr>
<td>Specific water consumption (ltrH₂O/MWhe)</td>
<td>51.67</td>
</tr>
<tr>
<td>Annual solar share</td>
<td>8.87%</td>
</tr>
<tr>
<td>Annual fuel consumption (tonne)</td>
<td>56700</td>
</tr>
<tr>
<td>Bottoming cycle share of electricity generation</td>
<td>41.22%</td>
</tr>
<tr>
<td>Capacity factor</td>
<td>1.0</td>
</tr>
<tr>
<td><strong>Heliostat field result</strong></td>
<td></td>
</tr>
<tr>
<td>Designed heliostat field thermal output at spring equinox noon (MWth)</td>
<td>53.867</td>
</tr>
<tr>
<td>Collector’s annual efficiency</td>
<td>75.86%</td>
</tr>
<tr>
<td>Solar multiple</td>
<td>0.824</td>
</tr>
<tr>
<td>Annual thermal energy delivered by the heliostat field (GWh)</td>
<td>99.275</td>
</tr>
<tr>
<td>Heliostat field averaged cosine factor</td>
<td>83.21%</td>
</tr>
<tr>
<td>Heliostat field averaged attenuation factor</td>
<td>95.23%</td>
</tr>
<tr>
<td>Heliostat field averaged spillage factor</td>
<td>97.68%</td>
</tr>
<tr>
<td>Heliostat field averaged shading and blocking factor</td>
<td>93.40%</td>
</tr>
<tr>
<td>Heliostat field weighted efficiency</td>
<td>64.25%</td>
</tr>
<tr>
<td>Heliostat field averaged unweighted efficiency</td>
<td>60.67%</td>
</tr>
<tr>
<td>Heliostat field’s number of mirrors</td>
<td>669</td>
</tr>
<tr>
<td><strong>Economic result</strong></td>
<td></td>
</tr>
<tr>
<td>Annual operating cost (MUS$)</td>
<td>13.455</td>
</tr>
<tr>
<td>Solar Equipment cost (MUS$)</td>
<td>28.940</td>
</tr>
<tr>
<td>Bottoming cycle cost (MUS$)</td>
<td>28.034</td>
</tr>
<tr>
<td>Total capital cost (MUS$)</td>
<td>152.37</td>
</tr>
<tr>
<td>Net present value (MUS$)</td>
<td>-34.918</td>
</tr>
<tr>
<td>LCOE (US$/MWh)</td>
<td>77.663</td>
</tr>
</tbody>
</table>

The other two design variables are related to the topping and bottoming cycle turbine inlet temperatures, taking into consideration that turbine capital investment rises substantially for significantly high inlet temperatures. Consequently, a gas turbine inlet temperature (GTIT) of 1519 K is the optimum gas turbine inlet temperature leading to the lowest LCOE. Finally, the optimum value for the level of steam superheating in the HRSG is calculated to be 0.63. It is understood that a high level of superheating leads
to considerable growth in steam turbine capital investment. Furthermore, additional investment is necessary for the HRSG’s superheater. Nevertheless, reducing the DOSH value leads to a greater bottoming cycle as the design steam mass flow rate increases. Nonetheless, a high level of superheating is not advised, economically.

Furthermore, optimum hybrid SBC power plant thermodynamic results are presented in Table 20. It must be taken into account that the analysis is carried out by presuming that the plant is operating at full load for 8760 hours of the year (complete year). As a result, the plant capacity factor is equal to one. Obviously, decreasing the plant’s annual operating hours at full load and including occasional shut downs in our analysis increases the calculated LCOE. On the contrary, the plant annual fuel consumption and solar share will be improved. Nevertheless, the variation in the plant’s operating condition (full load or part load) is beyond the scope of this research work. It is assumed that the plant’s additional generated electricity will be sold to the grid to improve its economic capability. Furthermore, it should be noted that the gas turbine cycles net power outputs are strongly dependent on the ambient air temperature. Consequently, the plant’s net power output is fluctuating during the year. It is reported that the hybrid SBC power plant is capable of generating 419.27 GWh of electricity. Therefore, the plant is capable of producing 47.9 MWe which is slightly short of the plant’s designed capacity of 50 MWe due to the hot Abu Dhabi climate.

In addition, 41.22% of the annually generated power (172.84 GWh) is provided by the steam bottoming cycle. Moreover, the plant is capable of converting about 47.2% of the presented thermal energy by the heliostat field and natural gas into electricity as the reported annual thermal efficiency is 47.02%. Additionally, the hybrid power plant’s specific CO$_2$ emission is 371.9 kgCO$_2$/MWh. Consequently, about 371.9 kg of CO$_2$ enters the atmosphere for generating 1 MWh of electricity. Similarly, the plant’s specific water consumption is 51.67 ltrH$_2$O/MWhe. As previously discussed, the main suppliers of thermal energy are the heliostat field and natural gas. The required thermal input for the plant’s annual operation is provided by 56700 tons of natural gas whereas the remaining 8.87% is supplied by the heliostat field. Thus, the plant’s annual solar share is 8.87%.

The final heliostat field layout utilized for the hybridization of the optimum SBC power plant is depicted in Error! Reference source not found. Bearing in mind
that the initial heliostat field utilizes 1430 mirrors, a subset of 669 mirrors displaying
the best thermal output are selected for the final layout. The presented subset of mirrors
is capable of providing 53.867 MWth of thermal output during spring equinox noon.
The designed heliostat field output is determined by calculating the required thermal
input to increase the air temperature leaving the topping cycle compressor to the
assigned nominal receiver temperature of 1223 K. Nonetheless, in our initial analysis,
thermal losses in the receiver and tower are not considered. Consequently, the nominal
receiver outlet temperature of 1223 K is not obtained during the spring equinox. The
ratio of thermal input provided by the field over thermal energy required to raise the air
temperature leaving the receiver to 1223K is the field solar multiple.

Figure 54: Heliostat field layout for the optimum hybrid SBC power plant

Note that solar multiple values greater than one are not recommended for hybrid
plants that do not utilize thermal energy storage, since a portion of the solar energy
provided by the field will be wasted due to the central tower and receiver’s operating
temperature limits. The solar multiple for the analyzed hybrid SBC power plant is
0.824. Furthermore, heliostats situated in the northern sections of the field have higher
annual thermal efficiency. Hence, greater number of mirrors are located in the northern
section.

The net solar energy delivered by the heliostat field in a complete year is 99.275
GWh. The considered field is utilizing 669 mirrors only, whereas the optimum field
layout depicted in Figure 40 can supply up to 197.66 GWh of thermal energy
throughout a complete year by implementing 1460 mirrors. Moreover, about 75.31
GWh of energy delivered by the heliostat field is effectively used in the collector as the
remaining 23.96 GWh of solar energy is wasted due to heat losses in the receiver and
central tower piping. Thus, the collector’s overall thermal efficiency, which includes
the central tower and receiver, is estimated to be 75.86%. In addition, utilizing the most
thermally efficient heliostats enhances the field optical efficiencies including cosine,
spillage, shading and blocking, and attenuation. Furthermore, the field weighted and
unweighted efficiencies are improved to 64.25% and 60.67%, respectively.

Air temperatures departing the heliostat field central receiver during a day for a
complete year are depicted in Figure 55. By assigning a solar multiple value of less than
one for the analysis, the receiver outlet air temperature will not reach the nominal outlet
temperature of 1223 K during spring equinox noon. Nevertheless, the presented results
indicate that the nominal outlet temperature of 1223 K is never obtained through the
year. A maximum receiver outlet temperature of 1156.1 K was attained on September
at 12:00 PM. Moreover, it should be noted that the heliostat field is constantly capable
of delivering hot air streams with temperatures greater than 1000 K during high
insolation periods. On the contrary, the central receiver is incapable of providing hot
air streams during noon within low insolation months including January, February, and
December.

The equipment capital investment cost breakdown for the optimum SBC power
plant is presented in Figure 56. One may notice that the solar components’ capital
investment, i.e. central tower, receiver, and heliostat field, are responsible for 27% of
the initial investment. Solar components’ capital investment cost is estimated to be
28.940 MUS$. Nonetheless, the highest share of investment is associated with the
construction of new infrastructures and civil engineering with 37%. Furthermore, it should be noted that the reported total capital investment in Table 20 includes the necessary costs of contingency, indirect factors, and installation while the cost breakdown depicted in Figure 56 only considers the equipment capital investment. The optimum SBC power plant’s initial investment is determined to be 152.37 MUS$ whereas the annual operating cost is 13.455 MUS$/year.

Figure 55: Central receiver’s outlet temperature throughout the day for a complete year, a) winter, b) spring, c) summer, d) fall

Figure 56: Hybrid SBC power plant equipment’s initial investment breakdown

Net present value and LCOE approaches are implemented for a feasibility assessment of a hybrid SBC power plant’s construction in the UAE. Furthermore, the thermo-economic optimization is carried out to minimize the plant LCOE. Accordingly,
the minimum acquired LCOE for the hybrid SBC power plant is reported to be 77.663 US$/MWh. Consequently, one may predict that the net present value for this investment will be negative as the electricity sale price in the UAE is about 0.07 US$/kWh (70 US$/MWh). We should bear in mind that LCOE determines the electricity sale price in which the plant’s net present value is zero. The plant’s net present value is -34.918 MUS$ implying that 34.918 MUS$ of the initial investment is not compensated during the plant’s 25 years of operation. Nonetheless, environmental gains associated with hybrid power plant implementation are the main incentive for electricity generation authorities to initiate more hybrid power plant constructions around the world. In particular, the UAE’s significant potential in solar power generation, due to its high solar insolation, is the main driving factor in the constant rise in hybrid power plants’ popularity. Therefore, construction of hybrid SBC power plants are strongly advised for the UAE to enhance the share of renewable energy contribution in power generation and abate the rate of CO₂ emission associated with power generation.

It was decided to provide a comparative analysis between the acquired results in this study and the results presented by Spelling [122] due to the similarities between the investigated plants. A CCC power plant with thermal energy storage is optimized by selecting the cost of avoided CO₂ emission as the objective. The optimum plant annual solar share is 8.8% which is similar to the annual solar share presented in this research work. Furthermore, the plant’s total investment cost is 128.9 MSUS$ with solar components responsible for 27.9 MUS$. In addition, the optimum power block LCOE is 82.0 US$/MWh. Taking into account that the plant capacity factor is 0.62, the reported LCOE by Spelling [122] is greater than the calculated LCOE in this research work. Finally, the acquired specific CO₂ emission and water consumption are 368 kgCO₂/MWh and 45.6 ltrH₂O/MWh, respectively.

4.3.3. Steam bottoming cycle hybridization.

Bearing in mind that already operating SBC power plants can be hybridized by providing a portion of their required thermal input from a renewable source of energy such as solar, it is of high interest to investigate the hybridization of an already existing power plant in the UAE both thermodynamically and economically. Initially, a non-hybrid reference SBC power plant is taken into consideration whose main characteristics are listed in Table 21.
The presented SBC power plant is the non-solar version of the hybrid SBC plant which was thoroughly assessed in the previous section. As a result, the main advantages and disadvantages of the SBC power plant’s hybridization can be indicated by comparing the results tabulated in Table 20 and Table 21. Firstly, plant hybridization will degrade its annual generated electricity from 420.9 GWh to 419.3 GWh which can be traced back to the additional pressure losses in the central tower and receiver. Additionally, it should be noted that heat losses in the combustion chamber are insignificant compared with heat losses in the central receiver. Consequently, a non-hybrid plant is capable of converting a greater portion of the provided thermal input into electricity as compared with hybrid power plants. In other words, hybrid power plants display poorer annual thermal efficiency such that the plant hybridization worsens the annual thermal efficiency from 48.44% to 47.16%. The most important contribution of the solar components’ integration is their considerable impact on annual fuel consumption and specific CO$_2$ emission. Plant specific CO$_2$ emission is reduced by 8.8% from 407.6 kgCO$_2$/MWhe to 371.9 kgCO$_2$/MWhe. Moreover, plant fuel consumption is decreased by 5675 tonsCH$_4$/year. In other words, plant hybridization with an annual solar share of 8.4% prevents about 15,600 tons of CO$_2$ from entering the atmosphere, annually.

Table 21: Non-hybrid reference SBC power plant’s main characteristics

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Assigned value</th>
</tr>
</thead>
<tbody>
<tr>
<td>TCPR</td>
<td>12.7</td>
</tr>
<tr>
<td>BCPR</td>
<td>1000</td>
</tr>
<tr>
<td>GTIT (K)</td>
<td>1519</td>
</tr>
<tr>
<td>DOSH</td>
<td>0.63</td>
</tr>
<tr>
<td>Annual generated electricity (GWhe)</td>
<td>420.9</td>
</tr>
<tr>
<td>Annual overall thermal efficiency</td>
<td>48.44%</td>
</tr>
<tr>
<td>Specific CO$_2$ emission (kgCO$_2$/MWhe)</td>
<td>407.6</td>
</tr>
<tr>
<td>Specific water consumption (ltrH$_2$O/MWhe)</td>
<td>41.9</td>
</tr>
<tr>
<td>Annual fuel consumption (tonne)</td>
<td>62375</td>
</tr>
<tr>
<td>Bottoming cycle share of electricity generation</td>
<td>41.13%</td>
</tr>
<tr>
<td>Capacity factor</td>
<td>1.0</td>
</tr>
<tr>
<td>Annual operating cost (MUS$)</td>
<td>14,794</td>
</tr>
<tr>
<td>Total capital cost (MUS$)</td>
<td>112,43</td>
</tr>
<tr>
<td>Net present value (MUS$)</td>
<td>12,824</td>
</tr>
<tr>
<td>LCOE (US$/MWh)</td>
<td>67.214</td>
</tr>
</tbody>
</table>

In addition, the plant’s operating cost is reduced as a lower amount of fuel is consumed annually. Reduction in fuel consumption and operating cost can be considered as an annual profit obtained by plant hybridization. However, additional maintenance and labor cost is not considered in the analysis since the annual plant...
hybridization profit was no longer positive and the payback period approach was not applicable. Thus, it is decided to keep the payback period analysis simple by only taking into account the variation in annual operating cost and additional capital investment. Plant annual operating cost is reduced by 1.339 MUS$/year. Accordingly, additional capital investment for power plant hybridization is estimated to be about 39.94 MUS$. Hence, the plant hybridization payback period is approximately 30 years. Noting that the plant will be operational for 25 years only, one may argue that the additional investment required for plant hybridization is never going to be fully compensated.

Furthermore, the life cycle saving approach was utilized to assess the feasibility of an investment which is plant hybridization. Unlike the payback period approach, additional maintenance and labor costs are taken into consideration for the life cycle saving analysis. As a result, the power plant hybridization life cycle savings is -43.465 MUS$ which implies that about 43.5 MUS$ of the initial investment is not compensated during 25 years of operation. One may argue that the initial investment required for plant hybridization is less than the acquired life cycle saving. As already discussed, plant hybridization will not yield any annual profit as the additional maintenance and labor costs are greater than the reduction in annual operating cost. Nonetheless, plant hybridization might be the most effective and economically justified solution for reducing the rate of global warming and climate change. Finally, the non-hybrid SBC power plant LCOE is 67.2 US$/MWh whereas hybrid SBC LCOE is reported to be 77.7 US$/MWh. Thus, generating 1 MWh of electricity with 8.9% of solar energy will cost 10.5 US$ more than generating the same amount of electricity without solar energy contribution.

It should be noted that prior to this section, hybrid power plant analyses were carried out with a specific optimum heliostat field layout. It is of high interest to study the impacts different heliostat field design variables have on the hybrid power plant thermo-economic performance. Initially, heliostat field capacity was varied by utilizing different numbers of mirrors in the field. The field capacity was stated based on its instantaneous thermal output during spring equinox noon. It should be noted that a maximum of 1460 mirrors were considered in the initial field layout. The optimum field layout which contains 1460 mirrors is capable of providing 105.8 MWth of thermal
energy during spring equinox noon. Consequently, the aforementioned value is considered as the maximum heliostat field capacity.

Results concerning the effect of heliostat field capacity on the plant’s annual fuel consumption and specific CO₂ emission are depicted in Figure 57. Clearly, increasing the heliostat field capacity results in a drop in the rate of fuel supplied at the combustion chamber. Bearing in mind that the non-hybrid SBC power plant annual fuel consumption and specific CO₂ emission are 62,375 tonneCH₄ and 407.6 kgCO₂/MWh, respectively. Consequently, fuel consumption and plant specific CO₂ emission are dropped by 1.7% and 1.6% to 61332 tonneCH₄ and 401.1 kgCO₂/MWh, respectively, through the integration of a 10 MWth heliostat field. Furthermore, the plant’s annual fuel consumption and specific CO₂ emission are further decreased by utilizing larger heliostat fields. For instance, a hybrid SBC power plant with a 105.8 MWth heliostat field consumes about 53,296 tons of CH₄ throughout a year. Consequently, more than 9000 tons of natural gas are saved annually. Furthermore, the plant specific CO₂ emission is abated to 350.3 kgCO₂/MWh. In other words, the amount of CO₂ emitted by the plant for generating 1 MWh of electricity is lowered by 57.3 kg.

![Figure 57: Effects of the heliostat field rate of thermal energy output during spring equinox noon on the plant annual fuel consumption and specific CO₂ emission](image)

Additionally, heliostat field expansion influences on the plant’s solar multiple and solar share are shown in Figure 58. Solar multiple represents the capability of the solar collector to achieve the nominal outlet temperature during the design period. In our case, a solar multiple of one means that the central receiver outlet temperature is 1223 K during spring equinox noon. Note that a solar multiple greater than one implies
that a portion of the supplied solar energy is not utilized in the power plant. With a maximum heliostat field capacity of 105.8 MWth, a solar multiple of 1.6 is achieved, as the heliostat field supplied thermal input of 105.8 MWth is significantly greater than the required thermal energy to raise the air temperature to the nominal temperature. Consequently, a section of the heliostat field must be defocused to avoid damaging the central tower piping.

![Figure 58: Effects of the heliostat field rate of thermal energy output during spring equinox on the plant’s solar multiple and annual solar share](image)

Another substantially informative performance indicator for hybrid power plants is the solar share. Solar share is a representative of the solar energy annual contribution in power generation. Clearly, increasing the field capacity results in growth in the plant’s annual solar share. Nonetheless, the annual solar share for hybrid power plants that do not utilize thermal energy storage is restricted, as the rate of growth in the plant’s annual solar share abates by constantly increasing the heliostat field capacity. This matter can be clearly seen from the results presented in Figure 58 as the solar share curve begins to flatten for high values of heliostat field capacity. The maximum solar share of 14.2% is acquired by utilizing all 1460 mirrors in the heliostat field.

Referring to Figure 59, heliostat field capacity effects on the plant’s LCOE and hybridization payback period are depicted. It should be noted that additional maintenance and labor costs associated with the solar components are disregarded in the payback period analysis since the annual profit becomes negative. These factors’ impacts are further elaborated in the life cycle savings analysis. It is obvious that
increasing the heliostat field capacity and annual solar share will only result in a constant rise in the plant’s LCOE. In general, the difference between the non-hybrid and hybrid power plants’ LCOE values are not substantial. In other words, hybrid power plant economic performance is poorer but competitive with non-hybrid plants. Plant LCOE varies from 72.9 US$/MWh for a 10 MWth heliostat field to 86.1 US$/MWh for a 105.8 MWth heliostat field. To further elaborate on the presented results, one may note that the plant specific CO$_2$ emission for the 105.8 MWth heliostat field is 350.3 kgCO$_2$/MWh. As a result, generating 1 MWh of electricity will cost an additional 18.9 US$ whereas 57.3 kg of CO$_2$ is blocked from entering the atmosphere. In other words, for each kilogram reduction in the CO$_2$ emission, an additional 0.330 US$ must be invested.

![Figure 59: Effects of the heliostat field rate of thermal energy output during spring equinox on the plant’s annual LCOE and payback period](image)

Additionally, the plant’s hybridization payback period is presented. Though the payback period is not the most accurate economic indicator, it is an informative evaluator for the power plant’s hybridization economic assessment. It is interesting that a low level of hybridization is less appealing than larger capacity heliostat field incorporations. For instance, a 10 MWth heliostat field payback period is 71.6 years whereas a 105.8 MWth heliostat field payback period is about 33.7 years. Furthermore, the shortest payback period of 28.6 years is reported for the integration of a 70 MWth heliostat field. It is interesting that the corresponding solar multiple for the integration of a 70 MWth heliostat field is 1.07. Consequently, one may argue that the payback periods for the heliostat fields with solar multiples close to one are shorter.
As already mentioned, the life cycle saving approach is another economic assessment approach considered in this research work. Results concerning the effect of the heliostat field capacity on the life cycle saving, annual profit, and additional investment required for plant hybridization are shown in Figure 60. We note that the negative values acquired for all the presented results indicate that plant hybridization is not a profitable investment. The significance of the presented result is the clear economic improvement accomplished by increasing the capacity of the heliostat field. The presented results indicate that the reduction in plant operating cost cannot compensate for the additional cost of maintenance and labor cost as the annual plant hybridization profit is constantly less than zero. Nonetheless, annual profit is optimized to -0.228 MUS$ by integrating a 70 MWth heliostat field. Accordingly, additional investment required for the power plant hybridization and life cycle saving approach each other as the annual profit gets closer to zero. Nonetheless, life cycle saving is constantly degraded as the heliostat field capacity increases. For the heliostat field capacity of 105.8 MWth, life cycle saving, additional investment and annual profit are -78.50 MUS$, -72.05 MUS$, and -0.553 MUS$, respectively. We must bear in mind that the negative value for the presented results is an indication of the economic losses in the considered investment.

Figure 60: Effects of the heliostat field rate of thermal energy output during spring equinox on the plant’s additional investment, annual profit, and life cycle saving

One should note that the heliostat field optimum configuration is kept constant for all previously presented results. Furthermore, the slightest variation in the heliostat field’s initial control variables, including the mirror sizing, central tower height, and
receiver’s dimensions, will result in a totally different optimum field layout. Consequently, to investigate the impact of the aforementioned control variables, the heliostat field must be continuously optimized to solely focus on its impact on the power plant’s performance. Moreover, investigating the integration of different heliostat field layouts with distinctive design variables provides valuable information on power plant hybridization in the UAE, bearing in mind that heliostat field layout optimization is carried out for every single point presented in the following results. Furthermore, heliostat fields’ sizing is not considered, unlike the previous analysis, as it might weaken the significance of the presented results. Consequently, the heliostat fields contain 1460 mirrors.

Initially, the effects of the central tower height on the field annual thermal output and plant solar share are depicted in Figure 61. Results indicate that implementing a taller central tower enhances the field thermal output. Correspondingly, power plant solar share increases as more solar energy can be provided by the heliostat field. Bearing in mind that solar share represents the percentage of thermal input provided by the heliostat field, as solar share increases the plant’s annual fuel consumption and specific CO₂ emission drops, accordingly. Nevertheless, one may notice that the plant solar share is not continuously rising, unlike the field annual thermal output, such that the maximum solar share of 14.2% is reported for a 125m central tower. At the maximum reported solar share, the plant’s annual fuel consumption and specific CO₂ emission are 53,293 tons CH₄ and 350.3 kgCO₂/MWh, respectively.

Furthermore, the effects of the central tower height on the payback period and plant LCOE are presented in Figure 62. Due to the greater pressure drop in a taller central tower piping, integration of a taller central tower degrades the plant’s annual thermal efficiency. Moreover, a greater amount of initial investment is required for the heliostat field incorporation. On the contrary, higher solar share implies that the plant’s annual fuel consumption and operating cost are decreased. Consequently, the plant’s LCOE and payback period experience an initial reduction in their values as the positive impacts of the taller central tower consideration outweigh its negative influences. The lowest LCOE value of 85.5 US$/MWh is acquired by choosing an 85m central tower. Nevertheless, it should be noted that implementation of a taller central tower is a more environmentally friendly alternative since the plant’s annual solar share is increased.
Regarding the payback period analysis, the minimum payback period of 33.3 years is acquired by utilizing a 105m central tower.

Figure 61: Effects of the central tower height on the heliostat field’s annual thermal output and plant solar share

Figure 62: Effects of the central tower height on the plant LCOE and payback period

Results concerning the influence of the central tower height on the life cycle saving, annual profit, and additional investment required for the plant hybridization are depicted in Figure 63. The presented economic indicators display a rather similar pattern to the previously discussed LCOE and payback period results. Annual losses associated with plant hybridization is minimized by implementing an 85m central tower. Note that the minimum reported annual loss is 0.525 MUS$/year. It is worth mentioning that the lowest additional investment required for plant hybridization is
reported to be 70.2 MUS$ for a central tower with a height of 85m. Accordingly, the minimum life cycle saving (loss) is -76.3 MUS$. Taking into account that the minimum difference between the required additional investment and life cycle saving is attained as the annual loss is minimized. At first, reduction in additional capital investment cost for an 85m central tower compared with a 75m central tower may seem unrealistic. Nonetheless, it should be noted that utilizing a taller central tower may result in a more compact heliostat field. Consequently, a cheaper solar component with a superior thermal performance results in a more thermo-economically justified configuration.

![Figure 63: Effects of central tower height on the plant’s additional investment, annual profit, and life cycle saving](image)

Another essentially effective design variable is the surface area of the mirrors utilized in the field to concentrate the solar radiation on the receiver. The effects of the mirror surface area on the field annual thermal output and plant’s annual solar share are shown in Figure 64. One should take into account that a greater solar heat flux is attained at the receiver’s surface by implementing a larger set of mirrors. Consequently, heliostat thermal performance is enhanced. Nonetheless, increasing the size of the mirrors degrades the field shading and blocking factor. As a result, the heliostat field optimization will provide a more expanded heliostat field to alleviate the field shading and blocking deterioration. Results indicate that increasing the mirror size leads to a rise in the field annual thermal output. Accordingly, the plant’s solar share is improved as the integrated heliostat field is capable of providing a greater amount of thermal input. In the analysis, maximum reported annual thermal output and solar share are 237.0 GWh and 15.0%, respectively.
Figure 64: Effects of the mirror surface area on the heliostat field annual thermal output and plant’s solar share

Referring to Figure 65, the effects of the mirrors’ surface area on the plant’s LCOE and payback period are illustrated. As already discussed, implementing the use of mirrors with greater surface area requires greater initial investment. Furthermore, the optimum heliostat field layout is more expanded; therefore, more land is required. In addition, the wiring and annual maintenance costs are increased. Nonetheless, the plant’s annual operating cost is reduced by having larger mirrors as the plant’s annual fuel consumption is abated. In general, LCOE is constantly rising by increasing the mirror surface area. Nonetheless, the aforementioned economic effects are more evident in the payback period approach. Note that the payback period was initially shortened prior to its substantial rise for larger mirrors’ utilization. A minimum payback period of 28.8 years is acquired by implementing mirrors with a surface area of 75 m².

Like the previous analysis, life cycle saving is utilized to further evaluate the economic aspects of plant hybridization in the UAE. Results concerning the effects of the mirror surface area on the life cycle saving, annual profit, and additional investment required for the plant hybridization are depicted in Figure 66. Life cycle losses and additional investment required for hybridization are constantly increasing. Moreover, the reported annual losses are considerably similar to the discussed payback period. To minimize annual losses to 0.234 MUS$/year, a mirror surface area of 75 m² is advised to be used in the heliostat field. Nonetheless, it must be taken into consideration that the plant’s annual fuel consumption and specific CO₂ emission are reduced by utilizing a larger set of mirrors to increase the plant’s annual solar share.
The last design variable studied in this research work is the central receiver’s dimensions. Results concerning the effect of the receiver’s dimensions on the field annual thermal output and plant’s annual solar share are presented in Figure 67. We must take into account that the heliostat field’s annual spillage factor is considerably improved by employing larger central receivers. The spillage factor, which is also referred to as the interception factor, is enhanced as the receiver is capable of intercepting a greater portion of the reflected solar radiation. Nonetheless, the
aforementioned improvement slowly fades away as the rise in the field annual thermal output becomes negligible. Furthermore, the reported solar share experiences the same rise in its value. Initially, utilizing larger receivers considerably enhances the plant’s annual solar share. Nonetheless, the rate of the rise in the annual solar share is constantly reducing. The maximum annual thermal output and solar share are 204.8 GWh and 14.4% for a receiver with a radius of 7m and height of 12m, respectively.

Figure 67: Effects of the receiver’s dimensions on the heliostat field annual thermal output and plant’s solar share

To evaluate the influence of the receiver’s dimensions economically, the plant’s hybridization payback period and LCOE are determined and presented in Figure 68. Results indicate that the LCOE is constantly rising. Nonetheless, the improvement in the heliostat field thermal output almost outweighs its additional capital investment for small receivers (first two points). By comparing the presented results for the receivers with dimensions of 2×7m and 3×8m, one can notice that the reported LCOE is only increased by 0.15 US$/MWh from 83.48 US$/MWh to 82.63 US$/MWh. The presented payback period’s results tell a similar story. The shortest payback period of 33.7 years is figured for the implementation of a receiver with radius of 3m and height of 8m.

To further analyze the economic performance of the receiver’s dimension, its influence is investigated on the additional equipment cost, annual profit, and life cycle saving. The acquired results are depicted in Figure 69. Unlike payback period analysis, annual losses associated with the power plant’s hybridization is minimized by having a
4×9m receiver, bearing in mind that the minimum annual hybridization loss is 0.556 MUSS/year. Furthermore, results indicate that increasing the receiver’s dimensions requires greater initial investment and results in a more significant life cycle loss.

Figure 68: Effects of the receiver’s dimensions on the plant’s LCOE and payback period

Figure 69: Effects of the receiver’s dimensions on the plan’s additional investment, annual profit, and life cycle saving

It was decided to provide a comparative analysis between the acquired results in this study and the results presented by Spelling [122] due to the similarities between the investigated plants. A CCC power plant with thermal energy storage is optimized by selecting the cost of avoided CO$_2$ emission as the objective. The optimum plant’s annual solar share is 8.8% which is similar to the annual solar share presented in this
research work. Furthermore, the plant’s total investment cost was 128.9 MSU$ with solar components responsible for 27.9 MUS$. In addition, the optimum power block LCOE was 82.0 US$/MWh. Taking into account that the plant capacity factor is 0.62, the reported LCOE by Spelling [122] is greater than the calculated LCOE in this research work. Finally, the acquired specific CO$_2$ emission and water consumption are 368 kgCO$_2$/MWh and 45.6 ltrH$_2$O/MWh, respectively.

As we have constantly mentioned throughout this research work, there is no surprise that plant hybridization is not an economically justified investment. Though hybrid power plants are competitive with non-hybrid power plants, non-hybrid configurations are more cost effective. In particular, plant hybridization is more economically unfavorable with the recent drop in the prices of fossil fuels and natural gas. However, climate change and the global warming crisis will most likely lead to more restrictions on power plants’ CO$_2$ emissions. Furthermore, due to constant technological advancement, the cost of solar components’ integration is expected to reduce, eventually. Moreover, the unexpected reduction in fossil fuel and natural gas costs will eventually fade away. With all the aforementioned factors, it is anticipated that solar hybridization will turn out to be a more economically justified configuration in the near future. Furthermore, it should be noted that environmental advantages of power plant hybridization justify its additional capital investment and operating cost, even with the current financial disadvantages of hybrid power plants.

### 4.3.4. Concluding remarks.

In this section, a thermo-economic optimization was accomplished for the installment of a new 50 MWe hybrid SBC power plant in the UAE. Another significant contribution of this research work is to separately investigate the hybridization of an already existing power plant. The aforementioned analyses were conducted with significant details implementing four distinctive economic assessment approaches including LCOE, payback period, net present value, and life cycle saving methods.

The hybrid SBC power plant’s minimum reported LCOE and net present value are 77.7 US$/MWh and -34.918 MUS$, respectively. Furthermore, the annual solar share and specific CO$_2$ emission of 8.87% and 371.9 kgCO$_2$/MWhe were obtained for the optimum power plant configuration. In addition, the optimum heliostat field weighted efficiency was 58.61% with 197.7 GWh of thermal energy delivered annually.
at the receiver surface. Noting that heliostat field sizing was performed within the thermo-economic optimization to employ a subset of mirrors providing the necessary thermal input, the integrated heliostat field contained 669 mirrors with weighted efficiency and annual thermal output of 64.25% and 99.3 GWh, respectively.

Further analysis of an already existing plant hybridization indicated that reduction in fuel consumption does not justify the additional capital investment, maintenance, and labor cost required for the solar components. In particular, plant hybridization is more economically unfavorable with the recent drop in the price of fossil fuels and natural gas. Nevertheless, it is believed that the environmental advantages of power plant hybridization justify its additional capital investment and operating cost considering the worrisome rate of CO$_2$ emission and global warming.

**4.4. Hybrid Air Bottoming Cycle Thermo-Economic Optimization**

In this section a hybrid ABC power plant is comprehensively investigated. Initially, the thermo-economic procedure which is utilized for the optimization of the power plant is presented. Afterward, a comparative analysis between hybrid SBC and hybrid ABC power plants is given. In the next section, the thermo-economic optimization results are provided and discussed. Finally, hybridization of an already existing ABC power plant is fully evaluated.

**4.4.1 Thermo-economic optimization procedure.**

Thermo-economic optimization of hybrid power plants is a considerably complex process consisting of several inner stages. An extensively simplified description of the developed thermo-economic model is depicted in Figure 70. Heliostat field optimization is the first step which is conducted based on the tabulated design variables in Table 6 and available meteorological data. The optimization is carried out with the discussed method presented by Collado [115] and Collado and Guallar [56, 116] with annual weighted efficiency as its objective. After the optimization is accomplished, mirrors are ranked based on their annual thermal performance.

Simultaneously, steady state analysis is conducted for the ABC power plant. Four design variables considered for the optimization of the plant are topping and bottoming cycle pressure ratios, turbine inlet temperature, and MFRR. Based on the assigned values for the abovementioned design variables, steady state analysis determines the topping and bottoming cycle air mass flow rate at ISO condition,
required solar heating for heliostat field sizing, tower piping dimension, and turbomachinery equipment initial investment.

Figure 70: Thermo-economic optimization of the hybrid air bottoming cycle power plant

The optimized heliostat field is not the final field layout employed in the optimization of the power plant. Noting that thermal energy storage is not considered, a portion of the provided solar input might be wasted due to temperature limits in the receiver and central tower piping. Therefore, heliostat field sizing is necessary to obtain a more economically justified outcome. In other words, it is essential to design a heliostat field that can provide a specific amount of thermal output at a specific time of year. Required solar heating obtained from the steady state analysis of the power plant is considered as the designed heliostat field thermal output. Moreover, the ranked array of all the mirrors within the field is employed to select a subset of the mirrors providing the required thermal output at the selected design period. Spring equinox noon is a popular choice for the design time. Consequently, the rate of thermal energy supplied by each mirror at spring equinox noon is calculated. Afterward, the calculated power is
added in sequence from the most efficient mirror to less efficient ones until the design thermal output is reached.

In the next step, the final heliostat field layout transient analysis is conducted for a whole year. Furthermore, heliostat field, tower, and receiver’s capital investments are determined. After acquiring the necessary results for the heliostat field annual thermal output and power plant steady state analysis, hybrid cycle transient analysis can be accomplished. Based on the instantaneous compressors’ air mass flow rates and heliostat field thermal output, hybrid configuration’s annual performance is evaluated. Transient analysis provides notable results for the thermo-economic optimization such as annual water and fuel consumptions, annual CO₂ emission, and annual generated electricity. The acquired information is utilized to calculate the plant’s annual operating cost. At this point, all the necessary information to evaluate the power plant LCOE are identified. Thus, the hybrid power plant LCOE is determined with the assigned design variables’ values.

Traditional gradient based optimization approaches are not advised for hybrid power plant optimizations considering the abovementioned multi-stages procedure. Hybrid power plant models are discontinuous, nonlinear, and non-differentiable. Therefore, other optimization approaches must be considered. The most suitable optimization method for the problem on our hands is the genetic algorithm. Bearing in mind that the genetic algorithm only requires the final value of the objective function; thus, objective function continuity and differentiability do not matter [159]. Consequently, the MATLAB optimization toolbox genetic algorithm is employed [160]. There is no guarantee that the optimum point acquired by the genetic algorithm is the global optimum. As a result, hybrid optimization (two successive methods of optimization) approach is considered for the analysis of this research work. It was decided to employ the optimum point attained by the genetic algorithm as the starting point in another MATLAB optimization toolbox built-in function. The other optimization built-in function, which does not employ the gradient approach, is “fminsearch”. To sum up, the acquired optimum conditions set by the genetic algorithm are utilized as the initial point for the “fminsearch” optimizer to ensure that the results are representing the most economically justified situation.
In the final stage, a line searching optimization approach is developed in MATLAB employing the procedures suggested by Ramos and Ramos [119]. The optimization algorithm adopts the method provided by Powell [125] to optimize each design variable separately. The optimization is needed to be initiated with a relatively good approximation of the optimum point. Consequently, acquired optimum design variables are considered as the starting point for the developed line search optimization code. Afterward, the optimum value for the first design variable is calculated by keeping other design variables values constant. This procedure is repeated for the other design variables to complete the first iteration. To find the optimum operating condition, multiple iterations are performed until satisfactory accuracy for all design variables is reached.

The hybrid power plant optimization is conducted with the plant’s LCOE as its objective. The selected design variables are topping cycle pressure ratio (TCPR), bottoming cycle pressure ratio (BCPR), topping cycle gas turbine inlet temperature (GTIT), and MFRR. The hybrid power plant optimization is accomplished by the black box approach recommended by Spelling [122]. In the black box approach, MATLAB optimizers provide the set of values for the assigned design variables. All the required analyses are conducted in the thermo-economic model, and the plant LCOE for the corresponding design variables is determined. Afterward, the only piece of information passed to the MATLAB optimizer is the plant LCOE. Similarly, this procedure is repeated until the optimum condition is acquired.

4.4.2. Comparative analyses.

In this section, a comparative analysis between ABC and SBC hybrid power plants is considered. For the comparative analysis, an identical topping cycle is considered. In other words, an already existing hybrid gas turbine power plant is taken into consideration. Two bottoming cycles, i.e. air turbine cycle and steam turbine cycle, are nominated to improve the simple hybrid gas turbine cycle efficiency. Therefore, the bottoming cycles’ heat recovery capacity is evaluated by having similar topping cycles with identical solar share, capital investment, available waste heat, and power generation capacity. The already existing hybrid gas turbine power plant main characteristics are listed in Table 22. The cycle pressure ratio and turbine inlet temperature are assumed to be 14 and 1500 K, respectively. At ISO conditions, the gas turbine air mass flow rate is 115.5 kg/s whereas the required fuel mass flow rate for
increasing the air temperature from 1223 K to 1500 K is 1.3035 kg/s. Additionally, the
nominal rate of solar thermal input which is utilized for the sizing of the heliostat field
is 73.331 MWth. In addition, the simple gas turbine cycle capacity at ISO conditions is
40.5 MWe. In addition, the hybrid power plant is capable of generating 337.12 GWhe
of electricity within a year. The plant’s annual fuel consumption is 76,424 tonne of
natural gas while 134.95 GWh of thermal energy is supplied by the heliostat field
annually. Annually, about 9.54% of the required thermal input for the plant’s operation
is provided by solar energy.

Table 22: Simple hybrid gas turbine characteristics

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Assigned value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure ratio</td>
<td>14</td>
</tr>
<tr>
<td>ISO air mass flow rate (kg/s)</td>
<td>115.5</td>
</tr>
<tr>
<td>Turbine inlet temperature (K)</td>
<td>1500</td>
</tr>
<tr>
<td>Nominal power output (MWe)</td>
<td>40.503</td>
</tr>
<tr>
<td>ISO fuel mass flow rate (kg/s)</td>
<td>1.3035</td>
</tr>
<tr>
<td>Annual fuel consumption (tonne)</td>
<td>76424</td>
</tr>
<tr>
<td>Annual generated Electricity (GWhe)</td>
<td>337.12</td>
</tr>
<tr>
<td>Annual overall energy efficiency</td>
<td>28.11%</td>
</tr>
<tr>
<td>Designed heliostat field thermal output at spring equinox noon (MWth)</td>
<td>73.331</td>
</tr>
<tr>
<td>Collector’s annual efficiency</td>
<td>73.77%</td>
</tr>
<tr>
<td>Annual solar share</td>
<td>9.54%</td>
</tr>
<tr>
<td>Solar multiple</td>
<td>80.4%</td>
</tr>
<tr>
<td>Annual thermal energy delivered by the heliostat field (GWh)</td>
<td>134.95</td>
</tr>
<tr>
<td>Heliostat field averaged cosine factor</td>
<td>81.50%</td>
</tr>
<tr>
<td>Heliostat field averaged attenuation factor</td>
<td>94.76%</td>
</tr>
<tr>
<td>Heliostat field averaged spillage factor</td>
<td>96.87%</td>
</tr>
<tr>
<td>Heliostat field averaged shading and blocking factor</td>
<td>93.71%</td>
</tr>
<tr>
<td>Heliostat field weighted efficiency</td>
<td>62.15%</td>
</tr>
<tr>
<td>Heliostat field averaged unweighted efficiency</td>
<td>58.79%</td>
</tr>
<tr>
<td>Heliostat field’s number of mirrors</td>
<td>940</td>
</tr>
<tr>
<td>Turbomachinery equipment cost (MUS$)</td>
<td>21.33</td>
</tr>
<tr>
<td>Solar equipment cost (MUS$)</td>
<td>37.03</td>
</tr>
<tr>
<td>Total capital cost (MUS$)</td>
<td>92.08</td>
</tr>
<tr>
<td>LCOE (US$/MWh)</td>
<td>89.507</td>
</tr>
</tbody>
</table>

The integrated heliostat field for the hybrid gas turbine plant is depicted in
Figure 71. The field contains 940 mirrors with annual weighted efficiency of 62.15%
and annual unweighted efficiency of 58.79%. The heliostat field layout is capable of
delivering 134.95 GWh of thermal energy. It should be noted that a portion of the
thermal energy provided by the field is wasted due to heat loss in the central receiver
and tower piping. Only 73.77% of the annual thermal energy provided by the field is
utilized for air preheating which is about 99.5 GWh. The field nominal thermal output
during spring equinox noon (design instance) is 73.3 MWth while the necessary thermal
input for achieving the maximum air temperature at the central receiver during spring
equinox noon is 91.2 MWth. Consequently, the plant’s solar multiple can be determined to be 80.4%. Economic analysis of the aforementioned hybrid gas turbine cycle is of high interest. The gas turbine cycle turbomachinery equipment capital investment cost is estimated to be about 21.33 MUS$ whereas the solar equipment initial investment is 37.03 MUS$. Furthermore, the total capital investment for the simple hybrid gas turbine power plant is 92.08 MUS$. Finally, the plant LCOE is reported to be 89.507 US$/MWh.

Figure 71: Heliostat field layout for the hybrid gas turbine cycle

Initially, integration of an air bottoming cycle to the aforementioned simple hybrid gas turbine power plant is investigated. Air bottoming cycle pressure ratio and air mass flow rate can be utilized in designing the most thermo-economically efficient bottoming cycle. Therefore, the effects of BCPR and MFRR on the plant’s different performance indicators are studied. Results concerning the effects of BCPR and MFRR on the plant’s total capital investment are displayed in Figure 72. It should be noted that the total capital investment for the hybrid gas turbine power plant is 92.08 MUS$. In
general, increasing the BCPR value results in a rise in the plant’s total capital investment. This is not the case for the MFRR as the plant’s total capital investment is maximized when the optimum value of MFRR is achieved. We need to bear in mind that the optimum value of MFRR is obtained by having equal heat capacities for hot and cold fluids in the air heat exchanger. By achieving equal heat capacities in the air heat exchanger, bottoming cycle heat recovery is maximized; however, larger heat transfer surface area is required. Consequently, the plant’s total capital investment is increased. The maximum total capital investment of 119.21 MUS$ is reached by setting MFRR and BCPR to 1.25 and 6. In other words, bottoming cycle maximum capital investment is 27.13 MUS$. The most expensive bottoming cycle is capable of producing 87.15 GWhe of electricity annually. On the other hand, the most inexpensive bottoming cycle configuration, which will only cost 9.68 MUS$, is associated with the lowest MFRR and BCPR values considered in the analysis. This bottoming cycle can only generate 31.63 GWhe of extra electricity throughout the whole year.

![Figure 72: Effects of MFRR and BCPR on the ABC power plant’s total capital investment](image)

Results concerning the effects of MFRR and BCPR on the plant’s annual generated electricity are depicted in Figure 73, taking into account that the hybrid gas turbine plant generates about 337.12 GWhe of electricity annually. In addition, the plant’s rate of thermal input (fuel mass flow rate and rate of solar energy) is not varied in the analysis. Consequently, it can be perceived that annual generated electricity is directly proportional to annual overall energy efficiency. Consequently, a rise in the plant’s generated electricity can be translated into an enhancement in the plant’s overall
energy efficiency. It can be seen that the ABC power plant’s annual generated electricity is optimized by enhancing the air heat exchanger heat recovery capacity. This matter is accomplished by selecting an appropriate bottoming cycle air mass flow rate which equalizes the bottoming cycle air heat capacity with the topping cycle exhaust gases’ heat capacity.

Additionally, increasing BCPR has diverse impacts on the plant’s generated electricity and annual energy efficiency. On one hand, bottoming air turbine cycle efficiency is improved by increasing its pressure ratio. On the other hand, increasing BCPR degrades the air heat exchanger’s heat recovery capability as hotter air stream enters the air heat exchanger. The plant’s annual generated electricity is maximized by setting MFRR and BCPR to 1.3 and 4, respectively. The maximum annual generated electricity is 432.72 GWhe, implying that the bottoming cycle has generated 95.6 GWhe of electricity annually. Thus, the bottoming cycle share of power generation is determined to be 22.1%. Moreover, the plant’s overall energy efficiency is enhanced from 28.11% to 36.08%. On the other hand, the minimum annual generated electricity by the bottoming cycle is 31.63 GWhe which results in the plant’s overall energy efficiency of 30.74%.

The most informative thermo-economic indicator presented in this research work’s analysis is the plant LCOE. The levelized cost of electricity approach determines the cost of electricity in which the plant’s total profit within its life span neutralizes its initial investment with loan interest rate consideration. The effects of
MFRR and BCPR on the ABC power plant’s LCOE are presented in Figure 74. It is worth mentioning that the hybrid gas turbine power plant’s LCOE is 89.5 US$/MWh. The minimum LCOE is reported to be 77.7 US$/MWh with MFRR and BCPR values of 1.35 and 3.75, respectively. Obviously, the addition of a bottoming cycle will indisputably increase the plant’s annual power generation and overall energy efficiency. Nonetheless, it should be noted that the integration of a bottoming cycle does not always improve the plant’s thermo-economic performance by reducing its LCOE. However, air bottoming cycle incorporation enhances the plant’s LCOE within the selected domain as the maximum reported LCOE is 86.2 US$/MWh.

![Figure 74: Effects of MFRR and BCPR on the ABC power plant’s LCOE](image)

Another bottoming cycle alternative is the integration of a Rankine cycle. Two main design variables in Rankine cycles are the steam turbine inlet temperature and steam mass flow rate. These design variables are manipulated by varying the value of BCPR and DOSH. Results concerning the effects of BCPR and DOSH on the SBC plant’s total capital investment are depicted in Figure 75. By comparing the results presented in Figure 72 and Figure 75, one can conclude that the ABC power plant’s total capital investment is considerably lower than the SBC required capital investment. The minimum capital investment obtained for the SBC plant is 195.4 MUS$. Hence, the minimum additional investment required for a steam bottoming cycle is 103.3 MUS$ which is even greater than the reference hybrid gas turbine cycle’s initial investment. High expenses of SBC power plants are due to the required investments for its HRSG, condenser, cooling tower, water treatment facility, and steam turbine.
The effects of BCPR and DOSH on the SBC plant’s annual generated electricity are investigated and the acquired results are presented in Figure 76. As can be seen, SBC power plants’ performance is substantially superior over the ABC power plants in terms of extra power generation. The lowest amount of electricity generated annually by the steam bottoming cycle is 207.1 GWhe while the maximum additional electricity provided by an air bottoming cycle is only 95.6 GWhe. The maximum annual generated electricity is 561.0 GWhe which is obtained by setting BCPR and DOSH to 1000 to 0.5, respectively. Consequently, the hybrid SBC power plant’s annual energy efficiency is between 45.37% and 46.77% within the investigated search domain.
Total capital investment and annual generated electricity are the two most influential factors in power plants’ LCOE calculation. Taking into account that SBC power plants are capable of generating more electricity annually and have higher annual energy efficiency, it is interesting to investigate the thermo-economic performance of hybrid ABC and SBC power plants by utilizing the levelized cost of electricity approach. Results concerning the effects of BCPR and DOSH on the SBC power plant LCOE are depicted in Figure 77. We note that the hybrid SBC power plant minimum reported LCOE is 76.4 US$/MWh while the minimum LCOE achieved for the hybrid ABC power plant is 77.7 US$/MWh. Furthermore, it must be noted that integration of a steam bottoming cycle into a hybrid gas turbine cycle constantly improves the plant’s LCOE as the maximum reported LCOE is 79.7 US$/MWh. However, proper optimization is advised to be accomplished to maximize its extra generated power and minimize the additional required capital investment, simultaneously.

Figure 77: Effects of MFRR and BCPR on the SBC power plant’s LCOE

To sum up, the presented results indicate the economic advantages of steam bottoming cycles for incorporation in small scale gas turbine power plants. Additionally, steam bottoming cycles can have a considerably greater impact on the plant’s annual efficiency and generated electricity. In general, hybrid ABC power plants display greater potential when the available initial investment is limited. Note that the reported LCOE for the ABC power plants are greater than the LCOE obtained for SBC power plants. Compared with hybrid ABC power plants, it can be said that the extra generated electricity by the hybrid SBC power plants justifies their additional capital investment. Thus, the steam bottoming cycle is a more cost effective waste heat
recovery alternative for small scale hybrid gas turbine power plants. However, the problem considered in this section must be carefully taken into consideration. It is important to note that the plants’ nominal power outputs are different for the case of SBC and ABC integrations. For the case of SBC integration, the combined configuration nominal power output rises to 66.9 MWe. Whereas, the combined configuration nominal power output for ABC integration is 50.1 MWe. Therefore, it is of higher interest to compare the ABC and SBC configurations’ thermo-economic performance at an identical capacity.

4.4.3. Thermo-economic optimization results.

A thermo-economic optimization of a hybrid ABC power plant in Abu Dhabi is carried out to minimize the plant LCOE. The complicated procedure for the thermo-economic optimization of the power plant utilized in this research work was thoroughly discussed in previous sections. The optimization design variables with their respective assigned search domains are listed in Table 23.

Table 23: Design variables and their respective search domains

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Assigned search domain</th>
</tr>
</thead>
<tbody>
<tr>
<td>Topping cycle pressure ratio (TCPR)</td>
<td>10.00 — 25.00</td>
</tr>
<tr>
<td>Bottoming cycle pressure ratio (BCPR)</td>
<td>2.00 — 8.00</td>
</tr>
<tr>
<td>Gas turbine inlet temperature (GTIT)</td>
<td>1223.0 K — 1673.0 K</td>
</tr>
<tr>
<td>Mass flow rate ratio (MFRR)</td>
<td>0.500 — 1.500</td>
</tr>
</tbody>
</table>

The thermo-economic optimization results are tabulated in Table 24. Optimization results indicate that a moderate TCPR is more economical. Consequently, one can conclude that extra power generated by increasing the compressor pressure ratio does not justify its additional capital investment. Moreover, increasing BCPR degrades its heat recover capability by feeding hotter air to the heat exchanger. Nonetheless, bottoming cycles with greater pressure ratios are more efficient. Therefore, the BCPR optimum value is achieved by finding a balance between the positive and negative impacts of its variation. Regarding the GTIT optimum value, economic analysis indicates that the turbine capital investment cost rises drastically for high values of inlet gas temperature. On the other hand, gas turbines are more efficient with higher inlet air temperatures. Therefore, the optimization task is to search for a condition in which the plant’s economic and thermodynamic performances are optimized simultaneously by selecting an appropriate value for GTIT. Furthermore, the optimum MFRR value is associated with the case that the topping cycle exhaust gases
and bottoming cycle air’s heat capacities are approximately identical which results in improving the bottoming cycle heat recovery.

Table 24: Hybrid ABC power plant thermo-economic optimization results

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Results</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Optimization result</strong></td>
<td></td>
</tr>
<tr>
<td>TCPR</td>
<td>15.17</td>
</tr>
<tr>
<td>BCPR</td>
<td>3.76</td>
</tr>
<tr>
<td>GTIT (K)</td>
<td>1525.4</td>
</tr>
<tr>
<td>MFRR</td>
<td>1.35</td>
</tr>
<tr>
<td><strong>Power plant thermodynamic result</strong></td>
<td></td>
</tr>
<tr>
<td>Annual electricity output (GWhe)</td>
<td>409.36</td>
</tr>
<tr>
<td>Annual overall energy efficiency</td>
<td>36.42%</td>
</tr>
<tr>
<td>Specific CO₂ emission (kgCO₂/MWhe)</td>
<td>483.88</td>
</tr>
<tr>
<td>Specific water consumption (lt/H₂O/MWhe)</td>
<td>12.43</td>
</tr>
<tr>
<td>Nominal solar share</td>
<td>64.44%</td>
</tr>
<tr>
<td>Annual solar share</td>
<td>8.12%</td>
</tr>
<tr>
<td>Annual fuel consumption (tonne)</td>
<td>72029</td>
</tr>
<tr>
<td>Bottoming cycle share of electricity generation</td>
<td>21.23%</td>
</tr>
<tr>
<td>Capacity factor</td>
<td>1.0</td>
</tr>
<tr>
<td><strong>Heliostat field result</strong></td>
<td></td>
</tr>
<tr>
<td>Designed heliostat field thermal output at spring equinox noon (MWth)</td>
<td>65.637</td>
</tr>
<tr>
<td>Collector’s annual efficiency</td>
<td>71.91%</td>
</tr>
<tr>
<td>Solar multiple</td>
<td>0.80</td>
</tr>
<tr>
<td>Annual thermal energy delivered by the heliostat field (GWh)</td>
<td>120.77</td>
</tr>
<tr>
<td>Heliostat field averaged cosine factor</td>
<td>82.21%</td>
</tr>
<tr>
<td>Heliostat field averaged attenuation factor</td>
<td>94.93%</td>
</tr>
<tr>
<td>Heliostat field averaged spillage factor</td>
<td>97.21%</td>
</tr>
<tr>
<td>Heliostat field averaged shading and blocking factor</td>
<td>93.60%</td>
</tr>
<tr>
<td>Heliostat field weighted efficiency</td>
<td>62.99%</td>
</tr>
<tr>
<td>Heliostat field averaged unweighted efficiency</td>
<td>59.56%</td>
</tr>
<tr>
<td>Heliostat field’s number of mirrors</td>
<td>830</td>
</tr>
<tr>
<td><strong>Economic result</strong></td>
<td></td>
</tr>
<tr>
<td>Annual operating cost (MUS$)</td>
<td>17.068</td>
</tr>
<tr>
<td>Solar Equipment cost (MUS$)</td>
<td>33.718</td>
</tr>
<tr>
<td>Bottoming cycle cost (MUS$)</td>
<td>13.845</td>
</tr>
<tr>
<td>Total capital cost (MUS$)</td>
<td>112.00</td>
</tr>
<tr>
<td>Net present value (MUS$)</td>
<td>-34.557</td>
</tr>
<tr>
<td>LCOE (US$/MWh)</td>
<td>77.763</td>
</tr>
</tbody>
</table>

Additional to the optimum values of the design variables, power plant thermodynamic performance is also presented in Table 24. Annual electricity generated by the plant is reported to be 409.36 GWhe. Bearing in mind that the plant is considered to operate for 8760 hours per year (no shut down) at full load, the plant capacity factor is one. Nonetheless, the ABC power plant’s power output is significantly dependent on the ambient air temperature and Abu Dhabi’s hot ambient air is constantly degrading the plant’s generated electricity. On average, the plant is generating 46.7 MWe of electricity. Out of the electricity generated by the power plant annually, 21.23% of it, which is equivalent to 86.9 GWhe, is produced by the air bottoming cycle. Additionally, annual energy efficiency of the power plant is 36.42%. Moreover, 483.88 kg of CO₂ is
emitted for generating 1 MWhe of electricity while only 12.43 liters of water is required for 1 MWhe of electricity generation. Two sources supplying the necessary thermal input for the plant’s operation are natural gas and solar energy. Annual fuel consumption is determined to be 72,029 tons whereas 8.12% of the thermal input is supplied by solar energy. The plant’s nominal solar share is 64.44% considering that the nominal receiver outlet temperature and GTIT of 1223 K and 1525.4 K are selected.

The heliostat field layout for the optimum hybrid ABC power plant is depicted in Figure 78. The presented heliostat field layout consists of 830 mirrors. In fact, the initial heliostat field contains 1460 mirrors. During optimization, a subset of mirrors containing the most efficient heliostats is considered to achieve the design thermal output. In our analysis, design thermal output is 65.637 MWth which is supplied by 830 mirrors during spring equinox noon, noting that heliostats closer to the tower are providing a greater share of thermal input. Additionally, heliostats located in the northern section of the field display superior thermal efficiency. Consequently, most of the heliostats are situated at the north of the central tower. The heliostat field solar multiple is calculated to be 0.8 implying that approximately 82.0 Mwth is required to achieve the design receiver outlet temperature of 1223 K during spring equinox noon. A solar multiple greater than one is not recommended for hybrid power generation without thermal energy storage as a portion of the supplied solar energy will not be utilized due to the central tower piping’s limited temperature.

Annual energy delivered by the heliostat field is 120.77 GWh utilizing 830 mirrors, whereas the complete field layout presented in the previous section is capable of supplying 197.66 GWh with 1460 mirrors. Bearing in mind that the number of mirrors utilized in the field are approximately halved (0.57) while the thermal energy delivered is reduced to 61.0%. In addition, only 71.91% of the supplied solar energy is utilized in the power plant due to the thermal losses in the central tower and receiver. Furthermore, other optical efficiency factors, including spillage, attenuation, shading and blocking, and cosine, are improved. Heliostat field unweighted and weighted efficiencies are enhanced as well to 59.56% and 62.99%, respectively.
Central receiver outlet temperatures throughout a day for a complete year are shown in Figure 79. In general, receiver outlet temperature does not reach the nominal temperature of 1223 K throughout the year which is understandable since the solar multiple for the optimum power plant is less than one. Nonetheless, receiver outlet temperature is constantly more than 1000 K during high insolation periods. On the other hand, heliostat field performance is unsatisfactory during low insolation months, i.e. January, February, and December. In the aforementioned months receiver outlet temperature does not reach 1000 K even during noon.
The hybrid ABC power plant capital investment cost breakdown is depicted in Figure 80. Note that 29% of the initial investment is associated with the heliostat field. Moreover, the central tower and receiver are responsible for 8% and 4% of initial investment. Consequently, solar components make up about 41% of the discussed hybrid ABC capital investment cost. Other significant contributors are the civil engineering and generators with 15% and 17% of the initial investment. Solar equipment cost is estimated to be 33.7 MUS$. Moreover, bottoming cycle cost is approximately 13.8 MUS$. Note that the presented total capital investment in Table 24 takes into consideration the installation, contingency, and other indirect factors. These factors are not considered in the economic breakdown depicted in Figure 80. Total capital investment of 112.0 MUS$ is reported with annual operating cost of 17.1 MUS$. Finally, the economic assessment indicators implemented for evaluating the thermo-economic performance of the power plants are tabulated in Table 24. The optimum value of LCOE for the analyzed power plant is 77.8 US$/MWh. Considering that the electricity sale price in the UAE is estimated to be 0.07 US$/kWh (70 US$/MWh), it was predictable that the plant’s net present value will be negative. Negative net present value implies that the investment for constructing a new hybrid ABC power plant is not compensated during its 25 years of operation. However, environmental advantages of hybrid power plants, not their economic performance, are the main driving factors in their constantly rising popularity among electricity
generation authorities. Therefore, an installment of a hybrid ABC power plant in the UAE is strongly recommended for increasing the share of renewable energy utilization in the power generation industry.

Figure 80: Hybrid ABC power plant equipment’s initial investment breakdown

4.4.4 Air bottoming cycle hybridization.

In this section, hybridization of an already existing ABC power plant is investigated. Initially, an ABC power plant is taken into consideration which operates with the optimum design variables obtained in the previous section. The considered ABC power plant’s main characteristics are tabulated in Table 25.

Table 25: ABC power plant’s main characteristics

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Assigned value</th>
</tr>
</thead>
<tbody>
<tr>
<td>TCPR</td>
<td>15.17</td>
</tr>
<tr>
<td>BCPR</td>
<td>3.76</td>
</tr>
<tr>
<td>GTIT (K)</td>
<td>1525.4</td>
</tr>
<tr>
<td>MFRR</td>
<td>1.35</td>
</tr>
<tr>
<td>Annual generated electricity (GWhe)</td>
<td>411.15</td>
</tr>
<tr>
<td>Annual overall energy efficiency</td>
<td>37.57%</td>
</tr>
<tr>
<td>Specific CO$_2$ emission (kgCO$_2$/MWhe)</td>
<td>525.52</td>
</tr>
<tr>
<td>Specific water consumption (ltrH$_2$O/MWhe)</td>
<td>0.271</td>
</tr>
<tr>
<td>Nominal power output (MWe)</td>
<td>50.0</td>
</tr>
<tr>
<td>Annual fuel consumption (tonne)</td>
<td>78570</td>
</tr>
<tr>
<td>Bottoming cycle share of electricity generation</td>
<td>21.20%</td>
</tr>
<tr>
<td>Capacity factor</td>
<td>1.0</td>
</tr>
<tr>
<td>Annual operating cost (MUS$)</td>
<td>18.610</td>
</tr>
<tr>
<td>Bottoming cycle cost (MUS$)</td>
<td>13.845</td>
</tr>
<tr>
<td>Total capital cost (MUS$)</td>
<td>65.76</td>
</tr>
<tr>
<td>LCOE (US$/MWh)</td>
<td>65.600</td>
</tr>
</tbody>
</table>

The described plant is a non-hybrid adaptation of the hybrid ABC power plant investigated in the previous section. Thus, a comparison between the results presented
in Table 24 and Table 25 indicates the advantages and disadvantages of ABC power plants’ hybridization. Initially, annual generated electricity is slightly reduced from 411.15 GWhe to 409.36 GWhe due to the additional pressure drops in the central tower piping and receiver. Furthermore, the combustion chamber thermal efficiency is considerably greater than the central receiver’s thermal efficiency. Consequently, the hybridization of the ABC power plant degrades the plant’s annual overall energy efficiency from 37.57% to 36.42%. Due to the implementation of solar energy in the hybrid ABC power plant, the specific CO$_2$ emission and annual fuel consumption are decreased markedly. It should be noted that the plant’s annual fuel consumption dropped by 8.3% from 78570 tonne to 72029 tonne. As a result, plant fuel consumption is reduced by about 6500 tonne of CH$_4$ which prevents approximately 18,000 tonne of CO$_2$ from entering the atmosphere during a complete year.

Due to the lower amount of fuel consumption, the hybrid power plant’s operating cost is reduced by 1.542 MUS$/year. Moreover, additional capital investment required for the plant’s hybridization is estimated to be about 46.24 MUS$. Finally, the non-hybrid ABC power plant’s LCOE is determined to be 65.6 US$/MWh whereas the hybrid power plant’s LCOE is 77.8 US$/MWh. In other words, the cost of producing 1MWh of electricity is raised by 12.2 US$ by the implementation of solar energy to provide about 8% of the power plant thermal input, annually. Finally, the ABC power plant hybridization payback period can be determined with the aforementioned economic information. With the recent fall in the cost of natural gas, hybridization of a power plant is even more economically unfavorable. The ABC power plant hybridization payback period is about 30 years. Considering that the plant operates for 25 years only, the hybridization of ABC power plants is not suggested economically. Nevertheless, it is strongly advised in order to reduce the rate of CO$_2$ emitted by fossil fuels and natural gas power plants.

It is of high interest to investigate the impact of integrating larger capacity heliostat fields on the plant’s thermo-economic indicators. Results concerning the effect of heliostat field rate of thermal output during spring equinox noon on the plant’s annual fuel consumption and specific CO$_2$ emission are depicted in Figure 81. In other words, the number of mirrors in the heliostat field are increased to enhance its thermal output.
Note that the number of mirrors is increased from 113 for the case of 10 MWth to 1460 for delivering 105.8 MWth of thermal energy during spring equinox noon.

![Graph showing annual fuel consumption and specific CO₂ emission vs. heliostat field capacity.](image)

**Figure 81**: Effects of the heliostat field rate of thermal energy output during spring equinox noon on the plant’s annual fuel consumption and specific CO₂ emission

Obviously, increasing the heliostat field capacity reduces the fuel mass flow rate required for the plant’s operation. Taking into account that the reference ABC power plant’s annual fuel consumption is 78,570 tonne, implementing a 10 MWth heliostat field will reduce the plant’s annual consumption to 77,584 tonne of CH₄. It should be mentioned that the maximum thermal energy supplied by the depicted heliostat field in Figure 40 during spring equinox noon is 105.8 MWth utilizing all 1460 mirrors. With the maximum field capacity, the annual fuel consumption is reduced by 12.6% to 68,685 tonne of CH₄. Increasing the heliostat field capacity has an identical effect on the plant specific CO₂ emission. Specific CO₂ emission is constantly reduced by utilizing a greater number of mirrors in the heliostat field such that the minimum specific CO₂ emission is 462.4 kgCO₂/MWh. Bear in mind that the reference ABC plant’s specific CO₂ emission is 525.5 kgCO₂/MWh. Consequently, the emitted CO₂ for generating 1 MWhe of electricity dropped by 63.1 kg.

The effects of increasing the heliostat field capacity on the field solar multiple and plant annual solar share are illustrated in Figure 82. Solar multiple evaluates the heliostat field’s ability to raise the air temperature in the receiver to the nominal temperature during the design period. Accordingly, a solar multiple of one implies that the heliostat field is capable of heating air within its central receiver to the assigned maximum temperature of 1223 K. It is worth mentioning that solar multiple values
greater than one are not advised for hybrid power plants without thermal energy storage, since a portion of the delivered thermal energy is wasted in the central tower and receiver due to the temperature limitation. At the maximum heliostat field capacity, a solar multiple of 1.26 is reported. Furthermore, solar share is a considerably important and informative indicator for hybrid power plants. It indicates the percentage of the annual thermal input which is provided by the heliostat field. For instance, a 10 MWth heliostat field contribution is only 1.2% whereas the other 98.8% of required thermal input is provided by natural gas in the combustion chamber. Additionally, there is a limit for a hybrid power plant’s solar share without thermal energy storage due to the aforementioned limit in maximum air temperature within the receiver and central tower, low insolation periods, and night times. With the maximum heliostat field capacity utilizing all 1460 mirrors, the plant’s annual solar share is increased to 12.3%.

![Figure 82: Effects of the heliostat field rate of thermal energy output during spring equinox on the plant’s solar multiple and annual solar share](image)

Referring to Figure 83, the impact of increasing the heliostat field’s thermal capacity on the plant’s LCOE and payback period are depicted. It is understandable that increasing the plant’s solar share will degrade its economic performance. It can be perceived that the reduction in the fuel consumption associated with a larger heliostat field does justify its additional capital investment as the plant’s LCOE is consistently rising. However, environmental concerns are the major motives in power plant hybridization. Additionally, the reported increase in the plant’s LCOE is not substantial, considering the notable reduction in CO₂ emission. For instance, the reference ABC power plant’s LCOE is 65.6 US$/MWh whereas the maximum LCOE reported in
Figure 83 is 84.3 US$/MWh. Thus, one can conclude that generating 1 MWh of electricity will cost an additional 18.7 US$ while the amount of CO$_2$ emitted to the atmosphere is reduced by 63.1 kg. Therefore, reducing each kilogram of CO$_2$ emitted costs about 0.3 US$.

Figure 83: Effects of the heliostat field rate of thermal energy output during spring equinox on the plant’s annual LCOE and payback period

A more interesting thermo-economic indicator is the payback period. A reference ABC power plant is considered for the hybridization. It should be noted that increasing the heliostat field capacity requires more capital investment; however, the plant’s annual fuel saving is abated. For the analysis of the payback period a simplified approach is considered by only taking into account the additional capital investment necessary for the plant’s hybridization and the plant’s annual profit from the reduction in its operating cost. It is noteworthy that a low level of hybridization has an extensively high payback period. With an integration of a 10 MWth heliostat field, the payback period is estimated to be about 74.6 years. Moreover, increasing the heliostat field capacity will cut down its payback period. Optimum heliostat field capacity lies in the neighborhood of 80 MWth with the payback period of 29 years.

Furthermore, the effects of heliostat field capacity on the hybridization cost, annual profit, and life cycle saving are presented in Figure 84. In this section, life cycle saving is considered to provide a more comprehensive analysis on the ABC hybridization feasibility. To determine the life cycle saving or loss associated with hybridization, additional investment required for the solar components are calculated. Furthermore, annual profit or loss is computed by finding the difference between the
reduction in the plant’s operating cost and growth in the maintenance and labor expenses. It should be noted that the negative values presented in the results represent losses or additional investment. It can be seen that the additional investment required for the hybridization is rising as the integrated heliostat field capacity increases. Similarly, hybridization continuously results in a negative saving throughout the plant’s operation. Furthermore, hybridization always leads to annual losses; nevertheless, the annual losses are minimized to 0.176 MUS$ by integrating a 90 MWth heliostat field.

Figure 84: Effects of the heliostat field rate of thermal energy output during spring equinox on the hybridization cost, annual profit, and life cycle saving

In the previous analyses, the optimum heliostat field layout and its characteristics are kept constant. Note that the smallest change in the heliostat field design variables, i.e. the central tower height, receiver surface area, and mirror size, demands a heliostat field layout optimization. In addition, investigating the effect of heliostat field design variables on the power plant performance provides valuable insights for hybrid power plant design and operation. Consequently, the impact variation in the aforementioned heliostat field design variables have on the power plant’s thermo-economic performance indicators are studied. It should be noted that for every single point presented in the following results, a new heliostat field layout optimization is accomplished to solely focus on the design variables’ influence with their maximum annual weighted efficiency. Furthermore, heliostat field sizing is not carried out and heliostat field transient analysis is conducted with 1460 mirrors.
Results concerning the effect of the central tower height on the field annual thermal output and the plant solar share are depicted in Figure 85. It is comprehensible that a taller central tower enhances the field optical efficiency and thermal performance. For instance, a heliostat field with a 75m long central tower can deliver 180.1 GWh of thermal energy annually while having a 155m long tower increases the field annual thermal output to 201.3 GWh. Due to improved heliostat field performance, the plant’s solar share is enhanced, too. Increase in the solar share implies that the plant’s annual fuel consumption and CO₂ emission dropped. Nonetheless, it should be noted that the rate of increase in both the field thermal output and plant’s solar share are constantly reducing. Furthermore, the maximum solar share is achieved with the central tower height of 145 m; although, the annual thermal energy delivered by the field is constantly rising. Understandably, central tower piping and receiver maximum operating temperature restrains the plant’s solar share. The minimum annual fuel consumption is calculated to be 68,649 tonne CH₄.

Figure 85: Effects of the central tower height on the heliostat field’s annual thermal output and plant’s solar share

The impact of the central tower height on the plant’s LCOE and payback period are displayed in Figure 86. It should be noted that increasing the central tower height has contradictory effects on the plant’s LCOE. On one hand, the plant’s annual generated electricity and overall energy efficiency are decreased due to the extra pressure drop in the central tower piping. Additionally, the plant’s capital investment is increased. On the other hand, lower fuel consumption abates the plant’s operating cost. The aforementioned contradictory effects are evident in the presented results as
the plant’s LCOE is initially reduced and increased afterward. Nonetheless, it should be taken into account that the main reason behind the plant’s hybridization is to increase the solar energy contribution. Thus, it can only be said with certainty that having an 85m long central tower is better than employing a 75m long tower, noting that an 85m long central tower will provide a greater amount of thermal energy and have a lower LCOE value. The LCOE value for the 85m long central tower is 83.8 US$/MWh.

Figure 86: Effects of the central tower height on the plant’s LCOE and payback period

The other interesting thermo-economic indicator is the plant hybridization payback period. The payback period evaluates the hybridization of an already existing ABC power plant. The minimum payback period of 30.7 years is reported for the case of employing a 105m central tower. Shorter or longer central towers will only prolong the time it takes to reimburse the additional capital investment required for plant hybridization by reducing fuel consumption and operating cost. Furthermore, effects of the central tower height on the hybridization cost, annual profit, and life cycle saving are presented in Figure 87. Initially, the additional investment required for the hybridization is reduced by integrating an 85m central tower instead of a 75m tower. The reduction in the additional expenses can be traced back to the optimum heliostat field layout. As a taller tower is utilized the optimum field layout becomes denser. Therefore, the field total capital investment drops. Nevertheless, further increase in the central tower height requires a greater amount of initial investment. The hybridization loss is minimized to 0.287 by utilizing a 105m tower.
The effects of the mirrors’ surface area on the field annual thermal output and plant’s solar share are shown in Figure 88. There is no surprise that larger mirrors are capable of concentrating a greater amount of solar energy on the central receiver. Nonetheless, increasing the size of the mirrors comes at a price as the plant’s optical efficiency reduces due to a degradation in its shading and blocking factor. In addition, increasing the mirrors’ surface area improves the plant’s solar share such that utilizing mirrors with surface area of 150 m$^2$ will improve the contribution of solar energy to 13.3%.
Results concerning the impact of the mirror sizing on the plant’s LCOE and payback periods are depicted in Figure 89. Employing larger mirrors in the heliostat field augments the field thermal output and abates the annual fuel consumption and CO₂ emission. Nevertheless, larger mirrors are more expensive and a greater amount of capital investment is required. Additionally, the heliostat field optimum layout is more expanded and requires a larger amount of land. It can be concluded from the results that the negative effects of increasing the mirrors’ size outdo its positive impacts as the plant’s LCOE is constantly rising. The maximum LCOE achieved with mirrors’ size of 150 m² is 88.5 US$/Mwh. Similar analyses are carried out for determining the plant’s hybridization payback period. A minimum payback period of 29.2 years is obtained for utilizing 100 m² mirrors. Furthermore, results concerning the effect of the mirror’s dimensions on the hybridization cost, annual profit, and life cycle saving are depicted in Figure 90. Results indicate that the cost of hybridization constantly rises by increasing the implemented mirror’s surface area. Additionally, the life cycle loss follows a similar pattern by continuously increasing. Nonetheless, the hybridization annual loss is minimized to -0.183 MUS$ by having a set of 100 m² mirrors.

Figure 89: Effects of the mirror surface area on the plant’s LCOE and payback period
Another important heliostat field design variable is the receiver surface area. We should consider that increasing the receiver surface area enhances the field spillage (intercept) factor as a greater portion of the reflected solar radiation is intercepted by the receiver. In other words, heliostat field mirrors have a larger target to concentrate the solar radiation on. The effects of employing a larger central receiver on the field’s annual thermal output and plant’s solar share are shown in Figure 91. Results indicate that increasing the receiver surface area to values greater than 50 m² has an insignificant impact on the field’s thermal performance and plant’s solar share. With a 50 m² central receiver, the field’s annual thermal output and plant’s solar share are 202.4 GWh and 12.4%, respectively.

Results concerning the effect of the receiver surface area on the plant’s LCOE and payback period are depicted in Figure 92. It is not advised to employ receivers with surface areas greater than 50 m² as the enhancement in the plant’s solar share are insignificant with respect to the growth in the plant’s LCOE. Moreover, the results indicate that the minimum payback period is 30.9 years for the installment of a receiver with surface area of 36 m². Finally, results concerning the effects of the receiver’s dimensions on the hybridization cost, annual profit, and life cycle saving are illustrated in Figure 93. Results indicate the annual loss associated with the power plant’ hybridization is minimized to 0.298 MUSS by utilizing a 36 m² (4m × 9m) receiver.
Figure 91: Effects of the receiver’s surface area on the heliostat field’s annual thermal output and plant’s solar share

Figure 92: Effects of the receiver surface area on the plant’s LCOE and payback period

4.4.5. Concluding remarks.

In this section, ABC power plant hybridization was extensively investigated. Primarily, air and steam bottoming cycles were selected for heat recovery from an already existing hybrid gas turbine cycle. Thermo-economic analyses were performed to identify the best bottoming cycle alternative. In addition, a thermo-economic optimization was carried out for the possibility of a hybrid ABC power plant installment in Abu Dhabi, UAE. Lastly, hybridization of an already existing ABC power plant was studied with LCOE and payback period methods.
To sum up, the presented results indicate the economic advantages of steam bottoming cycles for incorporation in small scale gas turbine power plants. Additionally, steam bottoming cycles can have a considerably greater impact on the plant’s annual efficiency and generated electricity. In general, hybrid ABC power plants display greater potential when the available initial investment is limited. Note that the reported LCOE for the ABC power plants are greater than the LCOE obtained for SBC power plants. Compared with hybrid ABC power plants, it can be said that the extra generated electricity by the hybrid SBC power plants justifies their additional capital investment. Thus, the steam bottoming cycle is a more cost effective waste heat recovery alternative for small scale hybrid gas turbine power plants. However, the problem considered in this section must be carefully taken into consideration. It is important to note that the plants’ nominal power outputs are different for the case of SBC and ABC integrations. For the case of SBC integration, the combined configuration nominal power output rises to 66.9 MWe. Whereas, the combined configuration nominal power output for ABC integration is 50.1 MWe. Therefore, it is of higher interest to compare the ABC and SBC configurations’ thermo-economic performance at an identical capacity.

Moreover, optimum value of LCOE for the hybrid ABC power plant is 77.8 US$/MWh. Considering that the electricity sale price in the UAE is estimated to be 0.07 US$/kWh (70 US$/MWh), the plant’s net present value is -34.6 MUS$. We note that a negative net present value implies that the investment for constructing a new
hybrid ABC power plant is not compensated during its 25 years of operation. Nevertheless, the main motives behind power plants’ hybridization are their environmental advantages. Therefore, power plant hybridization is strongly advised for expanding the share of renewable energy contribution in the power generation industry.

4.5. Hybrid Humid Air Bottoming Cycle Thermo-Economic Optimization

In this section a hybrid HABC power plant is comprehensively investigated. Initially, the thermo-economic procedure which is utilized for the optimization of the power plant is presented. Afterward, a comparative analysis between hybrid ABC and hybrid HABC power plants is given. In the next section, the thermo-economic optimization results are provided and discussed. Finally, hybridization of an already existing HABC power plant is fully evaluated.

4.5.1. Thermo-economic optimization procedure.

In the thermo-economic optimization of hybrid power plants, we have a rather complicated and multi-stage problem on our hands. A rather simplified but informative flow chart depicting the necessary steps in a thermo-economic optimization of a hybrid power plant is presented in Figure 94. The previously discussed heliostat field optimization by Collado [115] and Collado and Guallar [56, 116] is the initial stage of the optimization. The optimization is carried out utilizing the design variables, specifications, and meteorological data available in Table 6 to maximize the field’s annual weighted efficiency. After the heliostat field optimization, heliostats are sorted based on their annual weighted efficiency from the most efficient to the least efficient.

Separately, a steady state analysis is carried out for the power block. For the HABC configuration, five design variables are selected including topping cycle pressure ratio (TCPR), bottoming cycle pressure ratio (BCPR), gas turbine inlet temperature (GTIT), MFRR, and BCAH. Afterwards, the abovementioned design variables are assigned a specific value within the defined search domain. Steady state analysis is an essential step which provides a significant set of information to continue the thermo-economic optimization such as the topping and bottoming cycle air mass flow rates at ISO conditions, tower piping dimensions, necessary solar thermal energy at the design period, and turbomachinery components’ initial capital investment.
Initially, the heliostat field is optimized containing 1460 mirrors. Nonetheless, the optimized layout is not considered to be the final heliostat field configuration integrated in the power block. Bear in mind that thermal energy storage is not implemented into the plant and the central receiver’s operation is restrained to 950°C. Therefore, a part of the delivered thermal energy might not be utilized in the receiver to maintain the air temperature below the aforementioned limit. As a result, the field thermal energy output must be restrained to reduce the share of wasted thermal energy and make the hybrid power plant more feasible. Based on the calculated solar heating from the steady state analysis, heliostat field sizing is performed to only keep a specific number of mirrors able to provide the necessary thermal energy during spring equinox (design period). Employing the ranked array of the mirrors within the field, a subset of mirrors displaying the best annual performance are selected. Initially, the mirrors’ rate of thermal output during the design period is determined. Next, the calculated rates of thermal energy are added sequentially from the most efficient mirror to reach the design thermal capacity acquired from the steady state analysis.
In the next stage, the attained heliostat field layout thermal performance is assessed throughout a complete year. Additionally, the heliostat field’s initial capital investment along with the required investment for the central tower and receiver are determined. In the final stage of the developed thermo-economic model, a transient analysis is carried out for the power block utilizing the acquired data from the steady state analysis, heliostat field transient analysis, and weather data. In the transient analysis, the water temperature injected in the bottoming cycle’s air stream is assumed to be equal to the ambient air temperature. Furthermore, it is considered that the water mass flow rate provided in the evaporator is constantly regulated to keep the BCAH factor constant during the plant’s operation. Bearing in mind that the mass flow rate of the injected water is rather insignificant, it is believed that the aforementioned adjustment is achievable. Finally, the provided information from the plant’s steady state and transient analyses along with the heliostat field economic analysis enable us to determine the plant’s LCOE.

Clearly, a traditional gradient based optimization method cannot be utilized for the presented thermo-economic model. Bearing in mind that the objective function is nonlinear, non-differentiable, and discontinuous, heuristic optimization approaches must be employed. Genetic algorithms are appropriate for the optimization of the developed thermo-economic model. In the genetic algorithm approach, only the value of the objective function is needed [159]. Additionally, the MATLAB optimization toolbox provides a capable genetic algorithm built-in function which is employed in the analysis [160]. Consequently, the black box approach recommended by Spelling [122] can be implemented. In the black box optimization approach, the optimizer and developed thermo-economic model are only exchanging data in the form of the design variable and objective function values. In other words, the MATLAB optimizer assigns a value to each design variable in order to carry out the thermo-economic analysis. Afterward, the objective function value (LCOE) is determined and sent back to the MATLAB optimizer to continue the optimization and find the optimum condition.

A key disadvantage of genetic algorithms is that the attained optimum condition might be a local optimum rather than a global optimum. Consequently, it is decided to utilize the acquired optimum condition from the genetic algorithm as the initial guess in another optimization process. For this purpose, a line search optimization code based on the suggestion by Ramos and Ramos [119] is developed in MATLAB. The code is
developed based on Powell’s optimization algorithm [125] in which each design variable is optimized separately. Nonetheless, the optimization must be begun with a relatively good estimate of the optimum condition to ensure convergence. Afterward, design variables are optimized one by one. This procedure is repeated until the variation in the design variables’ optimal reaches a satisfactory level.

4.5.2. Comparative analysis.

In this section, integration of an HABC waste heat recovery bottoming cycle is investigated for the hybrid gas turbine presented in Table 22. It should be noted that ABC integration is investigated by utilizing two design variables, i.e. MFRR and BCPR, whereas HABC implementation can be studied by employing three design variables, MFRR, BCPR, and BCAH. In order to provide a better comparative analysis between these two configurations, it is decided to eliminate BCAH as a design variable in this section. For the analysis of this section, the BCAH value is kept constant at one implying that the air leaving the evaporator is fully saturated. In other words, implementation of a direct evaporative cooler in ABC configurations is evaluated by setting BCAH to one. Compressed air leaving the bottoming cycle compressor is cooled down and fully saturated in a direct evaporative cooler prior to the heat recovery heat exchanger. This way, the heat recovery capability of the bottoming cycle is considerably improved.

Results concerning the effects of the MFRR and BCPR on the plant’s initial capital investment are depicted in Figure 95. Results indicate that implementing a bottoming cycle with a greater compression ratio necessitates a higher initial investment. Furthermore, the required capital investment is maximized at an MFRR where the cold and hot fluids’ heat capacities are equal. At this specific point, bottoming cycle heat recovery is maximized; nonetheless, the cost of the heat exchanger becomes significant as well. Based on the presented results for ABC and HABC configurations, it can be concluded that HABC is a slightly more expensive power plant configuration. For instance, the maximum capital investment required for ABC and HABC power configurations are 119.2 MUS$ and 124.2 MUS$, respectively. Taking into account that the aforementioned maximum initial investment for the HABC plant is achieved by setting MFRR and BCPR to 1.15 and 6 whereas the maximum for ABC configuration is obtained at MFRR and BCPR of 1.25 and 6, respectively. At these points, ABC and HABC power plants are capable of generating 424.27 GWh and
447.05 GWh of electricity, annually. It can be calculated that the extra investment required and additional electricity generated with the HABC integration are 32.1 MUS$ and 109.93 GWh, respectively. Similarly, the extra investment required and additional electricity generated by the ABC implementation are 27.1 MUS$ and 87.15 GWh. As an approximate estimation, generating 1 MWh of additional electricity throughout the plant’s life cycle (for 25 years) with ABC and HABC employment costs about 12.44 US$/MWh and 11.68 US$/MWh, respectively.

Figure 95: Effects of MFRR and BCPR on the HABC plant’s total capital investment

Results concerning the impact of the MFRR and BCPR on the ABC and HABC power plant configurations’ annual generated electricity are shown in Figure 96. We take into account that the plant’s annual power output is directly proportional to the annual thermal efficiency because the rate of thermal input (solar and fuel) is constant. In general, the influences of MFRR and BCPR variations on the plant’s thermal performance are relatively similar to their effects on the plant’s initial investment. In other words, increasing the plant’s thermal efficiency and annual power output demands a greater initial investment. Maximum annual power output and thermal efficiency is obtained by setting MFRR and BCPR to 1.1 and 7, respectively. The maximum annual power output and thermal efficiency are 447.14 GWh and 37.28%, while, the maximum annual power output and thermal efficiency for the ABC configuration is 432.72 GWh and 36.08% at MFRR and BCPR of 1.3 and 4, respectively.
Figure 96: Effects of MFRR and BCPR on HABC plant’s annual generated electricity

Form the presented results, it can be perceived that HABC’s thermal performance is optimized at a lower MFRR than the ABC configuration due to the higher heat capacity of the humid air. Furthermore, HABC can operate at a higher pressure ratio. We note that increasing the bottoming compressor pressure ratio degrades the bottoming cycle heat recovery capability as the bottoming cycle air stream entering the heat recovery heat exchanger has a higher temperature. Nonetheless, employing a direct evaporative cooler (evaporator) in the HABC configuration enables it to utilize a compressor with a higher pressure ratio which enhances the bottoming cycle and plant’s overall performance.

The effects of the MFRR and BCPR on the plant’s LCOE are presented in Figure 97. In principle, LCOE combines the economic and thermal performance of the power plant; therefore, it is the most appropriate indicator to assess the proposed configurations’ performance. A minimum LCOE of 76.74 US$/MWh for the HABC configuration is attained at MFRR and BCPR of 1.1 and 5.5. Bearing in mind that the reference hybrid power plant’s LCOE is 89.51, implementation of HABC can improve its LCOE by 14.27%. Furthermore, it should be noted that the maximum LCOE reported within the assigned domain is 87.67 US$/MWh. Thus, it can be concluded that implementation of a HABC always leads to an enhancement in the plant’s LCOE. Furthermore, it should be noted that the optimum LCOE for ABC integration is 77.7 US$/MWh with MFRR and BCPR values of 1.35 and 3.75, respectively.
Figure 97: Effects of MFRR and BCPR on HABC plant’s LCOE

To sum up, the performances of HABC and ABC employment as a heat recovery bottoming cycle in an already existing hybrid gas turbine power plant were investigated. By making BCAH equal to one, the difference between the two investigated bottoming cycles is limited to the existence of an aftercooler (direct evaporative cooler) in the HABC configuration. Therefore, it can be perceived that integrating an aftercooler in ABC configurations, which cools down and humidifies the bottoming air stream, improves the plant’s thermal performance. While more investment might be required for the evaporative cooler, the plant LCOE is enhanced considerably. Hence, HABC is a more efficient and cost effective waste heat recovery bottoming cycle than ABC.

4.5.3. Thermo-economic optimization results.

Thermo-economic optimization is carried out for ABC and HABC power plants to provide a more detailed assessment on the advantages and disadvantages of water injection in ABC power plants recommended by Ghazikhani et al. [14]. Additionally, the optimization is carried out by taking Abu Dhabi as a case study. The optimization design variables associated with the HABC configuration are presented in Table 26 along with their respective assigned search domains.

The thermo-economic optimization results for the HABC power plant’s configuration are tabulated in Table 27. By comparing the optimum operating conditions for ABC and HABC configurations, the impact of water injection in the bottoming cycle air can be perceived. Initially, the optimum value of TCPR is
considerably low for the HABC configuration compared to ABC’s optimum TCPR. On the contrary, optimum BCPR for the bottoming cycle is greater for the HABC configuration which can be traced back to the cooling capability of the evaporator. One should note that increasing the bottoming cycle compressor pressure ratio leads to a rise in the air temperature entering the heat recovery heat exchanger. Therefore, the bottoming cycle heat recovery capability is degraded. Nonetheless, the evaporator which is devised in the HABC configuration operates as an aftercooler reducing the negative impact of increasing the BCPR. This is the reason that HABC’s optimum BCPR is greater. Another interesting aspect of the presented results is HABC’s notably lower optimum MFRR, bearing in mind that the optimum MFRR is obtained as the bottoming cycle heat recovery capability is maximized. Water injection increases the bottoming cycle air stream heat capacity; thus, lower bottoming cycle air mass flow rate is required for the HABC configuration.

Table 26: Design variables and their respective search domains

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Assigned search domain</th>
</tr>
</thead>
<tbody>
<tr>
<td>Topping cycle pressure ratio (TCPR)</td>
<td>10.0 — 25.0</td>
</tr>
<tr>
<td>Bottoming cycle pressure ratio (BCPR)</td>
<td>2.0 — 8.0</td>
</tr>
<tr>
<td>Gas turbine inlet temperature (GTIT)</td>
<td>1223 K — 1673 K</td>
</tr>
<tr>
<td>Mass flow rate ratio (MFRR)</td>
<td>0.50 — 1.50</td>
</tr>
<tr>
<td>Bottoming cycle air humidification (BCAH)</td>
<td>0.10 — 1.00</td>
</tr>
</tbody>
</table>

Additionally, the plant’s thermal performance indicators are presented. Since both ABC and HABC power plant configurations are designed to generate 50 MWe at ISO conditions, their annual power outputs are approximately identical. Nevertheless, HABC’s annual thermal efficiency is considerably greater implying that the plant can generate the aforementioned electricity with a notably lower thermal input (solar and fuel). Noting that ABC and HABC’s annual thermal efficiencies are 36.42% and 37.56%, respectively. Hence, water injection can enhance the plant’s thermal efficiency by 1.14% points. Additionally, water injection abates the plant’s annual fuel consumption by 2.9% from 72,029 tonne to 69,968 tonne of CH₄. In other words, utilizing the HABC configuration instead of ABC can save more than 2000 tonne of natural gas annually. Accordingly, the plant’s specific CO₂ emission dropped by 3.3% from 483.88 kgCO₂/MWh to 468.15 kgCO₂/MWh. Thus, annual CO₂ emission is reduced by about 5700 tonne by implementing the water injection power augmentation approach.
Table 27: Hybrid HABC power plant thermo-economic optimization results

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Optimization result</strong></td>
<td></td>
</tr>
<tr>
<td>TCPR</td>
<td>13.4</td>
</tr>
<tr>
<td>BCPR</td>
<td>1527</td>
</tr>
<tr>
<td>GTIT (K)</td>
<td>5.7</td>
</tr>
<tr>
<td>MFRR</td>
<td>1.11</td>
</tr>
<tr>
<td>BCAH</td>
<td>0.98</td>
</tr>
<tr>
<td><strong>Power plant thermodynamic result</strong></td>
<td></td>
</tr>
<tr>
<td>Annual electricity output (GWh)</td>
<td>411.01</td>
</tr>
<tr>
<td>Annual overall energy efficiency</td>
<td>37.56%</td>
</tr>
<tr>
<td>Specific CO\textsubscript{2} emission (kgCO\textsubscript{2}/MWhe)</td>
<td>468.15</td>
</tr>
<tr>
<td>Specific water consumption (ltrH\textsubscript{2}O/MWhe)</td>
<td>548.96</td>
</tr>
<tr>
<td>Annual solar share</td>
<td>8.57%</td>
</tr>
<tr>
<td>Annual fuel consumption (tonne)</td>
<td>69968</td>
</tr>
<tr>
<td>Bottoming cycle share of electricity generation</td>
<td>25.25%</td>
</tr>
<tr>
<td>Capacity factor</td>
<td>1.0</td>
</tr>
<tr>
<td><strong>Heliostat field result</strong></td>
<td></td>
</tr>
<tr>
<td>Designed heliostat field thermal output at spring equinox noon (MWth)</td>
<td>65.113</td>
</tr>
<tr>
<td>Collector’s annual efficiency</td>
<td>74.73%</td>
</tr>
<tr>
<td>Solar multiple</td>
<td>0.81</td>
</tr>
<tr>
<td>Annual thermal energy delivered by the heliostat field (GWh)</td>
<td>119.84</td>
</tr>
<tr>
<td>Heliostat field averaged cosine factor</td>
<td>82.26%</td>
</tr>
<tr>
<td>Heliostat field averaged attenuation factor</td>
<td>94.94%</td>
</tr>
<tr>
<td>Heliostat field averaged spillage factor</td>
<td>97.24%</td>
</tr>
<tr>
<td>Heliostat field averaged shading and blocking factor</td>
<td>93.56%</td>
</tr>
<tr>
<td>Heliostat field weighted efficiency</td>
<td>63.04%</td>
</tr>
<tr>
<td>Heliostat field averaged unweighted efficiency</td>
<td>59.60%</td>
</tr>
<tr>
<td>Heliostat field’s number of mirrors</td>
<td>823</td>
</tr>
<tr>
<td><strong>Economic result</strong></td>
<td></td>
</tr>
<tr>
<td>Annual operating cost (MUS$)</td>
<td>16.885</td>
</tr>
<tr>
<td>Solar Equipment cost (MUS$)</td>
<td>33.540</td>
</tr>
<tr>
<td>Bottoming cycle cost (MUS$)</td>
<td>15.931</td>
</tr>
<tr>
<td>Total capital cost (MUS$)</td>
<td>112.29</td>
</tr>
<tr>
<td>Net present value (MUS$)</td>
<td>-31.617</td>
</tr>
<tr>
<td>LCOE (US$/MWh)</td>
<td>77.075</td>
</tr>
</tbody>
</table>

On the other hand, power plant configurations that employ steam/water injection to improve their thermal performance consume a considerable amount of water [58]. It can be seen from the presented results that the specific water consumption for the HABC configuration is 548.96 ltrH\textsubscript{2}O/MWh. Therefore, one can conclude that the HABC configuration consumes 536.53 liters of additional water for generating 1 MWh of electricity which reduces the rate of CO\textsubscript{2} emission by 15.73 kg. Nevertheless, deploying a condenser is an effective solution for the implementation of the HABC configuration in regions with water shortage. In addition, the HABC configuration’s annual solar share is 8.57%. In other words, 8.57% of the required thermal input is provided by the heliostat field annually. Finally, the results presented for the bottoming cycle share of power generation indicate the effectiveness of water injection in the bottoming cycle in order to improve its thermal performance. HABC’s bottoming cycle
share of power generation is 25.25% whereas ABC’s bottoming cycle only contributes to 21.23% of the generated electricity, annually.

The optimum heliostat field layout integrated in the HABC topping cycle for air preheating is depicted in Figure 98. The presented field layout selected a subset of 823 mirrors displaying the best annual thermal performance. As a result, the field weighted and unweighted efficiencies are improved by 4.43% points and 4.70% points compared to the optimum field layout presented in Figure 40. Moreover, the optimum field design capacity (thermal output during spring equinox noon) is 65.113 MWe. In addition, the field delivers 119.84 GWh of thermal energy during a complete year with only 74.73% of it being utilized in the power block. Consequently, 30.285 GWh of thermal energy is wasted due to the losses and operating temperature restrictions in the central tower and receiver. Finally, the fields integrated in ABC and HABC configurations are almost identical, implying that the hybrid configurations can be properly assessed against each other.

The solar multiple is the ratio of the required thermal input to raise the air temperature leaving the receiver to the assigned maximum of 950°C during the design period. The integrated heliostat field solar multiple is 0.81; thus, the air temperature leaving the receiver is less than the assigned limit. In the case of having a field with a solar multiple greater than one, a part of the field must be directed away from the central tower to avoid damaging the receiver and tower piping during high insolation periods. Consequently, the field delivered thermal energy cannot be fully utilized. In general, solar multiples greater than one are only advised if thermal energy storage is integrated within the configuration. Figure 99 presents the receiver’s outlet temperature during a day for a complete year. It can be seen from the presented results that the air temperature leaving the receiver does not exceed the assigned limit of 950°C which is an indication of the implemented heliostat field capacity appropriateness.
Figure 98: Heliostat field layout for the optimum hybrid HABC configuration

Figure 99: Central receiver’s outlet temperature throughout the day for a complete year, a) winter, b) spring, c) summer, d) fall
Finally, the hybrid HABC power plant equipment’s capital investment breakdown is depicted in Figure 100. One should note that only plant equipment costs are considered in the presented results, bearing in mind that the total capital investment tabulated in Table 27 includes installation cost, contingency, and other indirect factors as well as the equipment’s initial investment. The depicted results indicate that solar equipment’s capital investment accounts for 41% of the total equipment cost, whereas the generator and civil engineering are responsible for 17% and 15% of the total equipment cost. Furthermore, HABC’s bottoming cycle initial investment is 15.931 MUS$ whilst ABC’s bottoming cycle only costs 13.845 MUS$. Nevertheless, HABC’s total capital investment is almost equal to ABC’s with only 0.29 MUS$ additional investment required. It should be noted that HABC’s annual operating cost is lower due to lower fuel consumption. Hence, implementing an HABC configuration instead of ABC leads to saving 0.183 MUS$, annually.

Figure 100: Hybrid HABC power plant equipment’s initial investment breakdown

Other economic indicators considered for the thermo-economic assessment of the proposed configurations are net present value and LCOE. HABC’s configuration optimum LCOE is 77.075 US$/MWh while ABC’s optimum LCOE is 77.763 US$/MWh. Therefore, water injection in the bottoming cycle air can improve the plant’s LCOE by 0.9%. Additionally, net present value for HABC and ABC configurations are -31.617 MUS$ and -34.557 MUS$, respectively. Hence, 2.940 MUS$ is saved during 25 years of plant operation by employing the HABC configuration instead of the ABC configuration. In general, the hybrid HABC net
present value is negative implying that the initial investment is not fully reimbursed during 25 years of the plant’s operation, taking into account that the LCOE is the minimum necessary electricity sale price to fully compensate the initial investment during the plant’s operation. Consequently, with an LCOE of 77.075 US$/MWh and electricity sale price of 70 US$/MWh, it is understandable that the net present value is negative.

4.5.4 Humid air bottoming cycle hybridization.

In this section, hybridization of an already existing HABC power plant is investigated. Initially, a reference HABC power plant is considered, operating with the optimum design variables attained in the previous section. The aforementioned non-hybrid HABC power plant’s main characteristics are tabulated in Table 28.

Table 28: HABC power plant’s main characteristics

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Assigned value</th>
</tr>
</thead>
<tbody>
<tr>
<td>TCPR</td>
<td>13.4</td>
</tr>
<tr>
<td>BCPR</td>
<td>5.7</td>
</tr>
<tr>
<td>TIT (K)</td>
<td>1527</td>
</tr>
<tr>
<td>MFRR</td>
<td>1.11</td>
</tr>
<tr>
<td>BCAH</td>
<td>0.98</td>
</tr>
<tr>
<td>Annual generated electricity (GWhe)</td>
<td>412.51</td>
</tr>
<tr>
<td>Annual overall energy efficiency</td>
<td>38.60%</td>
</tr>
<tr>
<td>Specific CO₂ emission (kgCO₂/MWh)</td>
<td>511.44</td>
</tr>
<tr>
<td>Specific water consumption (ltrH₂O/MWh)</td>
<td>535.00</td>
</tr>
<tr>
<td>Annual fuel consumption (tonne)</td>
<td>76717</td>
</tr>
<tr>
<td>Bottoming cycle share of electricity generation</td>
<td>25.14%</td>
</tr>
<tr>
<td>Capacity factor</td>
<td>1.0</td>
</tr>
<tr>
<td>Annual operating cost (MUS$)</td>
<td>18,476</td>
</tr>
<tr>
<td>Bottoming cycle cost (MUS$)</td>
<td>15,931</td>
</tr>
<tr>
<td>Total capital cost (MUS$)</td>
<td>66,01</td>
</tr>
<tr>
<td>Net present value (MUS$)</td>
<td>+21,713</td>
</tr>
<tr>
<td>LCOE (US$/MWh)</td>
<td>65,174</td>
</tr>
</tbody>
</table>

It is better to mention that the only difference between the presented plants in Table 27 and Table 28 is the integration of a heliostat field for power plant hybridization. Therefore, advantages and disadvantages of power plant hybridization can be perceived by performing a comparative analysis between the aforementioned data. Possibly, a negligible effect of power plant hybridization is the additional air pressure drop in the solar equipment. Therefore, the plant’s annual generated electricity dropped by 1.50 GWh on an annual basis. Additionally, power plant hybridization degrades the plant’s overall thermal efficiency by 1.04% points from 38.60% to 37.56%. The aforesaid drop in plant efficiency can be traced back to the thermal efficiency of the solar receiver as compared with the combustion chamber, taking into
account that the combustion chamber’s efficiency is 98% whereas the solar components’ efficiency is reported to be only 74.73%. On the other hand, power plant hybridization can significantly reduce the plant’s annual fuel consumption and CO₂ emission. Bearing in mind that the plant’s annual solar share is 8.57%, consumption of 6750 tonneCH₄ is replaced by the delivered solar energy from the integrated heliostat field. Accordingly, more than 18,500 tonne of CO₂ are prevented from entering the atmosphere annually.

Moving on to the economic aspects of power plant hybridization, annual operating cost is reduced by 1.591 MUS$ due to lower fuel consumption of the hybrid configuration. Additionally, hybridization requires 46.28 MUS$ of additional investment; therefore, a preliminary payback period calculation can determine that it takes about 29 years to compensate for the hybridization’s additional investment. Furthermore, power plant hybridization increases the cost of generating 1 MWh by 18.3% from 65.174 US$/MWh to 77.075 US$/MWh. In other words, generating 1 MWh of electricity costs an additional 11.9 US$ in order to reduce the rate of CO₂ emission by 43.3 kg. Based on the presented net present values for hybrid and non-hybrid configurations, it can be concluded that hybridization’s total additional expenses during 25 years of operation is 53.330 MUS$.

During the previous analysis, the integrated heliostat field capacity was determined based on the power block requirement. Nonetheless, it can be noteworthy to study the effect of implementing heliostat fields with different capacities on the plant’s thermal and economic performance indicators, noting that the presented heliostat field layout in Figure 40 can deliver 105.8 MWth of thermal energy during the design period (spring equinox noon) by utilizing all 1460 mirrors. Therefore, the maximum heliostat field capacity considered in this study is 105.8 MWth. Results concerning the effect of the heliostat field capacity on the plant’s annual fuel consumption and specific CO₂ emission are depicted in Figure 101. Annual fuel consumption and specific CO₂ emission are strongly dependent on the integrated field capacity. It is no surprise that increasing the field capacity leads to a drop in the field’s annual fuel consumption and specific CO₂ emission. As compared with the non-hybrid configuration, employing a 10 MWth heliostat field can reduce the annual fuel consumption and CO₂ emission by more than 1000 tonneCH₄ and 2800 tonneCO₂,
respectively. Similarly, integrating a 105.8 MWth field further abates the plant’s annual fuel consumption and CO2 emission by about 10,000 tonneCH4 and 28,000 tonneCO2, respectively.

Figure 101: Effects of the heliostat field rate of thermal energy output during spring equinox noon on the plant’s annual fuel consumption and specific CO2 emission

Similarly, the effects of the heliostat field capacity on the field solar multiple and annual solar share are presented in Figure 102, bearing in mind that fields with solar multiple values greater than one are only advised for the plants utilizing thermal energy storage to store the additional supplied thermal energy for employment during low insolation periods and night. By integrating the 105.8 MWth heliostat field, the plant’s solar multiple and annual solar share are 1.32 and 12.8%, respectively. Furthermore, it should be noted that the plant’s solar share is restricted by the solar component’s operating temperature limits and solar energy inaccessibility during night time. Therefore, two approaches can be considered to increase a plant’s annual solar share beyond its limit. First, plant operation during nights can be limited. Therefore, solar energy can have a more significant role in power generation. Second, thermal energy storage can be integrated within the power block.

Referring to Figure 103, heliostat field capacity influences on the plant’s LCOE and hybridization payback period are depicted. The presented result is an indication of the power plant hybridization’s economic disadvantages. In other words, generating electricity gets more expensive as the annual solar share increases and specific CO2 emission reduces. The maximum LCOE of 83.6 US$/MWh is reported for integrating
the heliostat field with the maximum capacity. As compared with the non-hybrid configuration, generating 1 MWh of electricity costs an additional 18.4 US$ for reducing the rate of CO₂ emission by 64.8 kg. Therefore, one can conclude that an extra 0.284 US$ is spent for reducing the rate of emitted CO₂ by 1 kg. Additionally, payback period analysis indicates that integrating heliostat fields with relatively small capacities is notably uneconomical. A minimum payback period of 28.1 years is reported for the case of employing an 80 MWe heliostat field.

Figure 102: Effects of the heliostat field rate of thermal energy output during spring equinox on the plant solar multiple and annual solar share

Figure 103: Effects of the heliostat field rate of thermal energy output during spring equinox on the plant’s annual LCOE and payback period
To further investigate the economic impact of the heliostat field capacity, hybridization additional cost, annual profit, and life cycle saving are depicted in Figure 104. Obviously, more investment is required as the field capacity increases. The maximum additional investment associated with the 105.8 MWth heliostat field is 72.1 MUS$. Nonetheless, annual profit obtained by the power plant hybridization is optimized to -0.113 MUS$ as an 80 Mwth is incorporated within the power block. Noting that the negative value implies that the power plant hybridization leads to economic losses, one may question that the payback period approach could not be applied when an investment is not profitable. Nonetheless, in order to present an informative ranking of the plants with different levels of hybridization using the payback period approach, it was decided to only include the additional investment required for the hybridization and the associated reduction in the plant’s annual operating cost. In other words, variation in the plant’s maintenance and labor cost are not considered in payback period analysis. However, these factors are considered in life cycle saving analysis. Overall, life cycle losses are constantly rising as the incorporated heliostat field capacity increases. In particular, 75.0 MUS$ of additional expenses is required for the integration of the 105.8 MWth heliostat field during 25 years of operation.

![Figure 104: Effects of the heliostat field rate of thermal energy output during spring equinox on the hybridization cost, annual profit, and life cycle saving](image)

To analyze the effect of the heliostat field capacity, only the number of mirrors utilized within the field are varied. Therefore, the optimum heliostat field layout presented in Figure 40 is employed. Nevertheless, it is of high interest to investigate the
influence of other heliostat field design variables such as the mirrors’ surface area, receiver’s dimensions and central tower height. Taking into account that the slightest variation in the above-mentioned design variables affects the optimum heliostat field layout, the heliostat field must be re-optimized each time that a new set of values is assigned to these design variables. In other words, the optimum heliostat field layout integrated in the power block is calculated for each single point presented in the following results. Additionally, it must be noted that heliostat field sizing is not implemented to better depict the effect of the heliostat field design variables on the power plant’s thermo-economic indicator. In other words, the implemented heliostat fields contain 1460 mirrors.

Results concerning the effect of the central tower’s height on the field’s annual thermal output and plant’s solar share are shown in Figure 105. Increasing the central tower height ameliorates the field’s annual weighted efficiency which leads to an improvement in the field’s annual thermal output, though, the rate of growth in the field’s thermal output reduces as the central tower height increases. The maximum annual thermal output and weighted efficiency of 201.3 GWh and 59.7% are reported for employing a 155m central tower. Understandably, increasing the field’s annual thermal output improves the plant’s annual solar share and reduces its fuel consumption and specific CO₂ emission. Nonetheless, it is important to note that an increase in the field’s annual thermal output does not necessarily lead to an enhancement in the plant’s solar share. We take into consideration that the operating temperature limits in the central tower and receiver restrict the utilization of the supplied thermal energy. Thus, an increase in the field’s thermal output is only effective as long as the air temperature in the receiver has not reached the aforementioned limit. The maximum solar share of 12.8% is obtained for a 135m central tower which reduces the plant’s annual fuel consumption to 66,617 tonneCH₄/year.
Figure 105: Effects of the central tower height on the heliostat field’s annual thermal output and plant’s solar share

Referring to Figure 106, the effects of the central tower height on the plant’s LCOE and hybridization payback period are illustrated. Bearing in mind that employing a larger central tower improves the field shading and blocking factor, the optimum heliostat field layout will be more condensed. Therefore, additional expenses of increasing the tower height might be canceled out by the reduction in the land and wiring costs. The aforementioned scenario is evident as an 85m tower replaces a 75m tower. Nevertheless, one may argue that the LCOE is not the best economic indicator to assess the impact of the central tower height. While an 85m central tower presents the minimum LCOE value, integrating a larger central tower can further reduce the plant’s specific CO₂ emission. Therefore, payback period analysis is provided to assess the feasibility of an already existing HABC power plant. Results indicate that a minimum payback period of 30.2 years is attained by utilizing a 105m central tower. Additionally, the impact of the central tower’s height on the hybridization initial investment, annual profit, and life cycle saving are presented in Figure 107. As already discussed, hybridization’s initial investment is reduced by 0.8 MUS$ with utilization of an 85m central tower instead of a 75m tower. Additionally, the hybridization annual losses are minimized to 0.238 MUS$ by employing a 105m central tower.
Furthermore, heliostats with larger surface area can be utilized in the field. Bearing in mind that increasing the mirror’s dimensions degrades the plant shading and blocking factor, the field must be further expanded to improve the field’s annual weighted efficiency. Results concerning the effect of the mirror’s dimensions on the field’s annual thermal output and plant solar share are depicted in Figure 108. Results indicate that implementing larger mirrors improves the field’s annual thermal output. Nonetheless, it should be noted that the rise in the field’s thermal output cannot be extended to an enhancement in the field’s weighted efficiency. On the contrary, the field’s annual weighted efficiency is constantly reducing. For instance, increasing the
mirror’s size from 75 m$^2$ to 150 m$^2$ increases the field’s annual thermal output by 105.2 GWh/year while its weighted efficiency is reduced by 6.3% point. Furthermore, the result depicts the growth in the plant’s annual solar share as larger mirrors are employed within the field. Consequently, the plant’s annual fuel consumption is reduced. A maximum solar share of 13.83% is reported for 150 m$^2$ mirrors implementation with annual fuel consumption of 65,842 tonne CH$_4$.

Figure 108: Effects of the mirror surface area on the heliostat field’s annual thermal output and plant solar share

The effects of mirrors’ size on the plant’s LCOE and hybridization payback period are presented in Figure 109. We note that the plant’s annual operating cost reduces due to the reported abatement in the plant annual fuel consumption. Nevertheless, increasing the mirrors size always leads to a higher LCOE. In principle, it can be concluded that additional investment is always demanded in order to reduce the plant’s CO$_2$ emission. On the other hand, payback period analysis indicates that mirrors with surface area between 75 m$^2$ and 100 m$^2$ are more cost effective. Additionally, the impact that the mirrors’ surface area has on the hybridization cost, annual profit, and life cycle saving are shown in Figure 110. As previously discussed, increasing the mirrors’ dimension results in a more expanded field; therefore, the hybridization cost is constantly rising. On the other hand, the hybridization annual losses are minimized to 0.134 MUS$ by utilizing 100 m$^2$ mirrors. Furthermore, life cycle losses associated with the power plant’s hybridization is constantly increasing as more initial investment is required and hybridization only results in annual losses.
Finally, the effect of the central receiver’s dimensions on the field’s annual thermal output and the plant’s annual solar share are presented in Figure 111. It should be noted that increasing the receiver’s surface area enhances the field spillage (intercept) factor, as a higher percentage of the redirected solar irradiation reaches the receiver’s surface. As a result, the field’s annual thermal output and weighted efficiency are improved. Nevertheless, increasing the receiver’s dimensions beyond $5m \times 10m$ is not advised as the improvement in the field’s thermal performance is diminished. In particular, replacing a $5m \times 10m$ receiver with a $7m \times 12m$ receiver improves the field’s annual thermal output and weighted efficiency by $0.7 \text{ GWh/year}$ and $0.21\%$.
points, respectively. Similarly, the power plant’s annual solar share enhancement is insignificant for receivers larger than $5m \times 10m$.

Furthermore, results concerning the effect of the receiver’s dimensions on the plant’s LCOE and hybridization payback period are depicted in Figure 112. One should note that increasing the receiver’s dimensions beyond the aforementioned value is not advised as the improvement in the plant’s solar share and CO2 emission do not justify the additional investment required. Furthermore, the results indicate that the minimum payback period is 30.4 years for the installment of a $4m \times 9m$ receiver. In addition, the effects of the receiver’s dimensions on the hybridization cost, annual profit, and life cycle saving are presented in Figure 113. Results indicate that increasing the receiver’s size demands a greater amount of initial investment. For instance, increasing the receiver’s dimensions from $5m \times 10m$ to $7m \times 12m$ requires an extra 3.9 MUS$ whereas the plant’s specific CO2 emission is only abated by 0.37 kgCO2/MWh. Additionally, the hybridization’s annual losses are minimized to 0.252 MUS$ by utilizing a $4m \times 9m$ receiver.

![Figure 111: Effects of the receiver’s surface area on the heliostat field’s annual thermal output and plant’s solar share](image)

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In this section, water injection in hybrid ABC power plants were extensively studied. Initially, an already existing hybrid gas turbine plant was considered for air bottoming cycle implementation. The air bottoming cycle was studied with and without water injection to properly assess the effect of water injection on the bottoming cycle and the plant’s thermal and economic capabilities. Furthermore, a thermo-economic optimization was carried out to evaluate the installment of a hybrid HABC as compared with a hybrid ABC configuration. In addition, an already existing HABC power plant
hybridization was studied by assessing the effect of the heliostat’s field design variables including its capacity, central tower height, mirrors and receiver’s dimensions.

Acquired results for the bottoming cycle integration showed that water injection in the ABC power plants can improve their heat recovery capability and reduce the plant’s LCOE. In particular, integrating an aftercooler which cools down and humidifies the air stream prior to the heat recovery heat exchanger can be an effective approach to improve the plant’s performance. Additionally, the thermo-economic analysis and optimization indicated that the HABC configuration is a more economical power plant configuration. Results indicated that the optimum LCOE for HABC and ABC are 77.1 US$/MWh and 77.8 US$/MWh, respectively. Additionally, net present value for HABC and ABC configurations were -31.617 MUS$ and -34.557 MUS$, respectively. Hence, 2.940 MUS$ was saved during 25 years of plant operation by employing the HABC configuration instead of the ABC configuration. Power plant hybridization results showed the significant effectiveness of power plant hybridization to further reduce the plant’s fuel consumption and specific CO₂ emission. Nonetheless, a hybrid configuration requires more initial investment and has higher running expenses than a non-hybrid power plant. Therefore, hybridization is economically unfavorable.

As a final note, hybridization might not be an economically justified investment. Nevertheless, it can be one of the more cost effective solutions to fight against global warming with the current renewable energy technologies’ level of maturity. In particular, developed countries are beginning to put more constraints on power plants’ CO₂ emission by increasing the penalty cost. Furthermore, fossil fuels and natural gas are experiencing an unusual and temporary drop in their cost. In addition, it is expected that solar equipment’s capital cost reduces with the current rate of technological advancement and special attention given to this matter. Therefore, the power plant’s hybridization economic is anticipated to improve to give additional incentive for the implementation of this technology within power plants all around the world.

4.6. Maisotsenko Bottoming Cycle Thermo-Economic Optimization

In this section a hybrid MBC power plant is comprehensively investigated. Initially, the thermo-economic procedure which is utilized for the optimization of the power plant is presented. Afterward, a comparative analysis between hybrid ABC and hybrid MBC power plants are given. In the next section, the thermo-economic
optimization results are provided and discussed. Finally, hybridization of an already existing MBC power plant is fully evaluated.

4.6.1. Thermo-economic optimization procedure.

We must take into account that thermo-economic analysis and optimization of hybrid power plants has a complex nature. In order to present the necessary steps in hybrid power plant thermo-economic optimization, a simplified flow chart is depicted in Figure 114. At the first stage, heliostat field optimization is conducted by employing the abovementioned approach presented by Collado [115] and Collado and Guallar [56, 116]. Heliostat field optimization is carried out in order to maximize the field’s annual thermal output utilizing the design variables, specifications, and meteorological data tabulated in Table 6. At the final stage, the mirrors are arranged based on their annual performance in order to use them in the next steps of the thermo-economic optimization.

Concurrently, the power block’s steady state analysis is carried out. MBC configuration steady state analysis is performed with the consideration of five design variables, i.e. topping cycle pressure ratio (TCPR), bottoming cycle pressure ratio
(BCPR), gas turbine inlet temperature (GTIT), MFRR, and ASDH. Steady state analysis is a necessary step within the thermo-economic model because it provides essential information for completing the transient analysis. In principle, the power block is designed by performing the steady state analysis. In fact, transient analysis’ necessary design parameters are calculated based on the abovementioned design variables including the topping and bottoming cycle air mass flow rate at ISO conditions, heliostat field capacity, tower piping dimensions, and the power block’s turbomachinery components’ capital investment.

One should note that thermal energy storage is not considered for integration in the plant. Therefore, an oversized heliostat field might provide significantly greater thermal energy than required. Considering the central tower and receiver’s operating temperature restriction, the power block can only utilize a limited amount of thermal energy, and the rest will be wasted. Because of the aforementioned reasons, the heliostat field must be designed based on the required thermal input in the power block in order to have a more economical hybrid power plant. Steady state analysis can determine a rough approximation of the necessary heliostat field capacity which is employed to perform the heliostat field sizing by only keeping a specific number of mirrors able to deliver the designed heliostat field capacity during the design period. Moreover, spring equinox noon is selected to be the design period in this research work. Utilizing the ranked array of mirrors acquired in the heliostat field optimization, a subset of mirrors presenting the best annual thermal output are considered able to provide the required thermal output during spring equinox. At first, the field thermal performance during spring equinox is evaluated. Next, the calculated rates of thermal energy during spring equinox are totaled successively from the best heliostat until the necessary thermal output is reached.

Afterward, heliostat field transient analysis is carried out to determine its thermal performance throughout the year. Furthermore, the heliostat field’s economic analysis is conducted to estimate the initial investment required for the field, tower, and receiver. In the last stage, the power block’s transient analysis is performed by employing the information provided by the steady state analysis, heliostat field transient analysis, and available meteorological data for the UAE. For the transient analysis of the MBC configuration, it is assumed that the injected water temperature is always equal to the ambient air temperature. Additionally, the transient analysis is carried out
by keeping the value of ASDH constant. In other words, the rate of injected water is constantly regulated to keep the plant operation at its optimum. The adjustment in the rate of water injection at the upper and lower sections of the air saturator is similar to the accomplished regulation in the delivered fuel mass flow rate at the combustion chamber. Therefore, water regulation can be achieved based on the ambient air temperature and relative humidity. After finalizing all the discussed stages, the hybrid power plant’s LCOE can be determined.

It is obvious that traditional gradient based optimization methods are not appropriate for the optimization of the discussed thermo-economic model. Heuristic optimization methods such as genetic algorithms must be employed because of their compatibility with nonlinear, non-differentiable, and discontinuous problems. A genetic algorithm can accomplish the necessary optimization without any knowledge of the objective function except for its value [159]. Furthermore, a genetic algorithm’s built-in function is available in the MATLAB optimization toolbox [160] which is implemented for the thermo-economic optimization carried out in this research work. In order to carry out the optimization employing a genetic algorithm, a black box approach is implemented [122]. In other words, the optimizer will specify the assigned design variables’ value and pass them to the thermo-economic model. Thermo-economic analysis is carried out to calculate the selected optimization objective (LCOE). Afterwards, the objective value is given back to the optimizer to carry on the optimization.

A genetic algorithm is an appropriate method for the optimization of the above-mentioned thermo-economic model; nevertheless, it might result in a local optimum instead of global optimum. As a result, another optimization should be carried out to ensure that the acquired optimum conditions represent the global optimum. For this reason, a line search optimization code is prepared based on the recommendation by Ramos and Ramos [119]. The aforementioned line optimization code performs an optimization for each design variable separately based on the Powell’s optimization algorithm [125]. An important necessity for the line optimization method is to start the optimization with a relatively good estimation of the optimum point. Therefore, the optimum conditions acquired by the genetic algorithm are selected as the starting point for the developed line search optimization code.
4.6.2. Comparative analysis.

In this section, integration of an HABC waste heat recovery bottoming cycle is investigated for the hybrid gas turbine presented in Table 22. Taking into account that the topping cycle design variables are not eligible for this section because an already existing hybrid gas turbine power plant is selected, only two design variables, BCPR and MFRR, can be considered for the ABC analysis. Similarly, three design variables, including BCPR, MFRR, and ASDH, can be considered for the analysis of the MBC integration. Nevertheless, it is decided to only consider two design variables for the MBC configuration as well. Bearing in mind that the air saturator is not commercialized whereas the Maisotsenko cooling cycle is a commercialized and popular cooling technology, it is decided to replace the discussed air saturator with a Maisotsenko cooler and a simple heat exchanger. In principle, the lower section of the air saturator is a cooler which is an adoption of the Maisotsenko cooling cycle. Consequently, the aforementioned scenario can be simulated by setting the ASDH value to zero, implying that no water is injected into the upper section of the air saturator. Therefore, a simple heat exchanger replaces the upper section of the air saturator. To sum up, analysis is carried out by keeping ASDH at zero which is associated with the case of integrating an aftercooler (Maisotsenko cooler) and a simple heat recovery heat exchanger. In other words, the only difference between the MBC configuration considered in this section and the ABC configuration is the incorporation of a Maisotsenko cooler to cool down the bottoming cycle compressed air and recover a greater portion of available waste heat in the topping cycle exhaust gases.

Results concerning the effects of MFRR and BCPR on the power plant’s capital investment are depicted in Figure 115. Similar to the presented results for the ABC configuration, having a greater BCPR requires more initial investment. However, there exists a MFRR value which maximizes the plant’s initial capital investment. It should be noted that the optimum value of MFRR which maximizes the initial capital investment is associated with the case of having equal heat capacities at the heat exchanger. In order to compare the additional investment required for ABC and MBC implementation, capital investment at an MFRR of 0.6 and BCPR of 2 can be taken into consideration. At the aforesaid point, the MBC integration additional investment is 9.5 MUS$ whereas ABC’s required investment is 9.7 MUS$. A maximum additional capital investment of 27.1 MUS$ (total capital investment of 119.2) is required when
MFRR and BCPR are 1.05 and 8, respectively. While maximum additional investment for the ABC implementation is 27.1 MU$ as well, it is achieved by setting the MFRR and BCPR to 1.25 and 6, respectively. Furthermore, the most expensive MBC and ABC configurations are capable of generating 114.2 GWh and 87.2 GWh, annually.

Figure 115: Effects of MFRR and BCPR on the MBC power plant’s total capital investment

The effects of MFRR and BCPR on ABC and MBC power plants’ annual power outputs are presented in Figure 116. One should note that the provided rate of thermal input (fuel and solar thermal) is not affected by the variation in the MFRR and BCPR. Consequently, the depicted results present the influence of the aforementioned design variables on the plant’s annual thermal efficiency, too. In general, the plant’s thermal efficiency and power output are maximized when the topping cycle exhaust gases and bottoming cycle’s humid air heat capacities are equal. Furthermore, the results indicate that the optimum MFRR is lower for MBC as compared with ABC due to the added water. Therefore, the optimum MBC configuration is more compact as compared with the ABC’s optimum configuration. Furthermore, a rise in BCPR implies that a hotter air stream is utilized for heat recovery in the ABC power plant. Nonetheless, the MBC configuration cools down the compressed air in the lower section of the air saturator (aftercooler); therefore, the negative impact of implementing a bottoming cycle with a greater compressor pressure ratio is alleviated. As seen from the results presented in Figure 116, the optimum BCPR value for MBC is greater than the ABC configuration due to the aftercooler implementation.
Comparing the thermal performance of ABC and MBC configurations at an MFRR of 0.6 and BCPR of 2, additional electricity generated throughout a complete year by the bottoming cycle for MBC and ABC are 35.5 GWh and 31.6 GWh, respectively. Similarly, the MBC and ABC power plants’ annual thermal efficiencies are 31.07% and 30.74%, respectively. In general, it can be concluded that even integration of a small scale bottoming cycle can improve the plant’s thermal performance. Bear in mind that MBC displays a greater potential to be integrated as a bottoming cycle. Maximum thermal improvement for MBC integration is obtained by setting MFRR and BCPR to 1.05 and 6.5 resulting in 116.4 GWh of extra generated power and annual thermal efficiency of 37.81%. Whereas, the ABC plant’s thermal efficiency is optimized by setting MFRR and BCPR to 1.3 and 4, respectively. The bottoming cycle is capable of generating 95.6 GWh of additional electricity which improves the plant’s thermal efficiency to 36.08%. As previously discussed, the MBC optimum MFRR is lower while its optimum BCPR is greater as compared with ABC’s optimum operating conditions.

Results concerning the impact of MFRR and BCPR on ABC and MBC power plant configurations’ LCOE are shown in Figure 117. MBC’s optimum LCOE is 75.2 US$/MWh which is obtained at an MFRR of 1.05 and BCPR of 6. Therefore, the existing plant’s LCOE is improved by 16.0%. Taking into account that the maximum reported LCOE is 86.5 US$/MWh the integration of an MBC in an already existing gas turbine power plant only leads to an improvement in the plant’s LCOE. Bearing in mind
that ABC’s optimum LCOE is 77.7 US$/MWh with MFRR and BCPR values of 1.35 and 3.75, the MBC can provide a more significant enhancement in the plant’s LCOE.

Based on the results presented in this section, it can be concluded that MBC integration in an already existing hybrid gas turbine cycle might require a greater investment as compared with ABC integration. Nevertheless, the MBC configuration has a more significant impact on the plant’s annual generated electricity and LCOE. Take into consideration that the MBC configuration studied in this section is, in principle, an ABC power plant with the integration of a Maisotsenko cooler after the bottoming cycle compressor for aftercooling purposes. As a result, MBC configurations do not necessarily need to have a complicated air saturator to operate. In other words, the presented results indicate the notable advantages of Maisotsenko cooler implementation in ABC power plants.

Figure 117: Effects of MFRR and BCPR on the MBC power plant’s LCOE

4.6.3. Thermo-economic optimization results.

Thermo-economic optimization is conducted for the MBC power plant’s configuration. In addition, LCOE is considered as the thermo-economic optimization objective. Moreover, Abu Dhabi is selected as the plant’s location due to its high potential for solar thermal power plant implementation. Five design variables are considered for the MBC configuration’s thermo-economic optimization comprising TCPR, BCPR, MFRR, GTIT, and ASDH. The aforementioned design variables and their respective assigned search domains are tabulated in Table 29.
Table 29: Design variables and their respective search domains

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Assigned search domain</th>
</tr>
</thead>
<tbody>
<tr>
<td>Topping cycle pressure ratio (TCPR)</td>
<td>10.0 — 25.0</td>
</tr>
<tr>
<td>Bottoming cycle pressure ratio (BCPR)</td>
<td>2.0 — 8.0</td>
</tr>
<tr>
<td>Gas turbine inlet temperature (GTIT)</td>
<td>1223 K — 1673 K</td>
</tr>
<tr>
<td>Mass flow rate ratio (MFRR)</td>
<td>0.50 — 1.50</td>
</tr>
<tr>
<td>Air saturator degree of humidification (ASDH)</td>
<td>0.00 — 1.00</td>
</tr>
</tbody>
</table>

Thermo-economic optimization results for the hybrid MBC power plant’s configuration are tabulated in Table 30. Compared to the ABC configuration’s optimum operating conditions, the MBC power plant topping cycle has a lower pressure ratio, whereas its bottoming cycle employs a compressor with a greater pressure ratio. In general, the lower section of the air saturator acts as an aftercooler enabling the plant to have higher values of BCPR, bearing in mind that increasing BCPR leads to an improvement in the bottoming cycle efficiency. Additionally, the optimum value for GTIT is relatively similar between the considered configurations while the optimum MFRR is significantly lower for the MBC configuration. As already mentioned in the previous section, water addition increases the cold fluid (humid air) heat capacity in the heat recovery heat exchanger (upper section of the air saturator). Therefore, the optimum bottoming cycle air mass flow rate which maximizes the plant’s heat recovery capability is less. Finally, it can be concluded that no water addition in the upper section of the air saturator is the most cost effective setup. In other words, MBC’s thermo-economic performance is optimized by utilizing the lower and upper sections of the air saturator as an aftercooler and a conventional heat recovery heat exchanger, respectively. Therefore, an already commercialized Maisotsenko cooler and a simple heat exchanger can be employed in the MBC configuration instead of a complicated air saturator.

Moving on to the power plant thermodynamic performance, it can be seen that the investigated configurations’ annual power outputs are approximately similar. Nonetheless, MBC can achieve the presented annual power output with lower initial investment. Furthermore, MBC’s annual thermal efficiency is 38.11% while ABC’s thermal efficiency is 36.42%. Taking into account that the annual power output is almost equal, it can be concluded that lower annual thermal input (solar and fuel) is demanded by the MBC configuration. As perceived from the annual thermal efficiency, the MBC power plant’s annual fuel consumption is notably lower. In other words,
implementing an MBC power plant instead of an ABC leads to about 2700 tons of CH$_4$ saving, annually. Hence, the plant rate of CO$_2$ emission is considerably lower. Note that MBC emits 460.77 kgCO$_2$ for generating 1 MWh of electricity whereas ABC’s specific CO$_2$ emission is 480.88 kgCO$_2$/MWh. On an annual basis, having an MBC configuration instead of an ABC prevents about 7400 tonne of CO$_2$ from entering the atmosphere.

Table 30: Hybrid MBC power plant thermo-economic optimization results

<table>
<thead>
<tr>
<th>Parameter</th>
<th>MBC optimization result</th>
</tr>
</thead>
<tbody>
<tr>
<td>TCPR</td>
<td>12.0</td>
</tr>
<tr>
<td>BCPR</td>
<td>6.1</td>
</tr>
<tr>
<td>GTIT (K)</td>
<td>1532</td>
</tr>
<tr>
<td>MFRR</td>
<td>1.06</td>
</tr>
<tr>
<td>ASDH</td>
<td>0.00</td>
</tr>
</tbody>
</table>

Power plant thermodynamic result

<table>
<thead>
<tr>
<th>Parameter</th>
<th>MBC power plant thermo-economic optimization results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Annual electricity output (GWh)</td>
<td>413.83</td>
</tr>
<tr>
<td>Annual overall energy efficiency</td>
<td>38.11%</td>
</tr>
<tr>
<td>Specific CO$_2$ emission (kgCO$_2$/MWhe)</td>
<td>460.77</td>
</tr>
<tr>
<td>Specific water consumption (ltrH$_2$O/MWhe)</td>
<td>604.98</td>
</tr>
<tr>
<td>Annual solar share</td>
<td>8.92%</td>
</tr>
<tr>
<td>Annual fuel consumption (tonne)</td>
<td>69338</td>
</tr>
<tr>
<td>Bottoming cycle share of electricity generation</td>
<td>27.91%</td>
</tr>
<tr>
<td>Capacity factor</td>
<td>1.0</td>
</tr>
</tbody>
</table>

Heliostat field result

<table>
<thead>
<tr>
<th>Parameter</th>
<th>MBC power plant thermo-economic optimization results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Designed heliostat field thermal output at spring equinox noon (MWth)</td>
<td>65.313</td>
</tr>
<tr>
<td>Collector’s annual efficiency</td>
<td>77.04%</td>
</tr>
<tr>
<td>Solar multiple</td>
<td>0.84</td>
</tr>
<tr>
<td>Annual thermal energy delivered by the heliostat field (GWh)</td>
<td>120.25</td>
</tr>
<tr>
<td>Heliostat field averaged cosine factor</td>
<td>82.24%</td>
</tr>
<tr>
<td>Heliostat field averaged attenuation factor</td>
<td>94.94%</td>
</tr>
<tr>
<td>Heliostat field averaged spillage factor</td>
<td>97.23%</td>
</tr>
<tr>
<td>Heliostat field averaged shading and blocking factor</td>
<td>93.58%</td>
</tr>
<tr>
<td>Heliostat field weighted efficiency</td>
<td>63.03%</td>
</tr>
<tr>
<td>Heliostat field averaged unweighted efficiency</td>
<td>59.59%</td>
</tr>
<tr>
<td>Heliostat field’s number of mirrors</td>
<td>826</td>
</tr>
</tbody>
</table>

Economic result

<table>
<thead>
<tr>
<th>Parameter</th>
<th>MBC power plant thermo-economic optimization results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Annual operating cost (MUS$)</td>
<td>16.769</td>
</tr>
<tr>
<td>Solar Equipment cost (MUS$)</td>
<td>33.645</td>
</tr>
<tr>
<td>Bottoming cycle cost (MUS$)</td>
<td>14.537</td>
</tr>
<tr>
<td>Total capital cost (MUS$)</td>
<td>108.85</td>
</tr>
<tr>
<td>Net present value (MUS$)</td>
<td>-23.969</td>
</tr>
<tr>
<td>LCOE (US$/MWh)</td>
<td>75.330</td>
</tr>
</tbody>
</table>

On the other hand, MBC’s specific water consumption is substantially higher. In principle, water addition is introduced to improve the plant’s thermal efficiency and reduce its fuel consumption. For generating 1 MWh of electricity, MBC consumes an additional 592.55 liters of water as compared with the ABC power plant. The main drawback of steam/ water injected cycles is the rise in the plant’s water consumption which is an issue in places with water limitations [58]. Nevertheless, it should be noted
that a condenser can be implemented for the MBC configuration in order to reduce its water consumption. In addition, the MBC power plant’s annual solar share is 8.92%, indicating the share of thermal energy delivered by the heliostat field. Furthermore, the presented results for the bottoming cycle share of power generation indicates the superiority of MBC as compared with ABC for bottoming cycle integration. MBC can provide about 28% of the generated electricity whereas ABC is only responsible for about 21% of the annually generated electricity.

Moreover, the optimum heliostat field layout utilized in the hybrid MBC power plant is depicted in Figure 118. The optimum heliostat field contains 826 mirrors which is capable of delivering 65.313 MW of thermal energy during spring equinox noon. In addition, the heliostat field provides 120.25 GWh of thermal energy annually which is about 8.92% of the thermal energy supplied to the power block. Moreover, only 77.04% of the delivered thermal energy is employed in the plant due to the thermal loss in the central tower and receiver. In addition, it should be noted that the topping cycle air stream is not preheated in the central tower during low insolation periods to avoid pressure losses in the solar components. In general, the heliostat fields incorporated in ABC and MBC power plants are very similar; therefore, a proper comparative analysis between these two hybrid plants can be accomplished.

The plant’s solar multiple is 0.84 implying that approximately 77.75 MW of thermal energy is required to raise the air temperature leaving the compressor to 950°C during spring equinox noon. Bear in mind that solar multiples greater than one are not suggested for implementation in plants without thermal energy storage. Therefore, a portion of the provided thermal energy cannot be employed in the plant due to the receiver’s operating temperature limit. For the case considered in this study, the presented solar multiple is reasonable in order to avoid surpassing the receiver’s temperature limit, frequently. To better illustrate the aforementioned constraint on the hybrid power plant’s performance, the central receiver’s outlet temperatures throughout a day for a complete year are depicted in Figure 119. Overall, the central receiver’s outlet temperature does not exceed the temperature limit of 950°C. However, the receiver’s outlet temperature approaches the limit during high insolation period. The presented results are indications of the integrated heliostat field’s appropriate capacity to provide the necessary thermal input without exceeding the assigned limits. One must
note that heliostat fields with greater capacities are not economically justified unless thermal energy storage is integrated within the hybrid configuration.

Figure 118: Heliostat field layout for the optimum hybrid MBC configuration

In addition, the hybrid MBC power plant equipment’s capital investment cost breakdown is presented in Figure 120. It should be noted that capital investment factors such as installation cost, contingency, and other indirect factors are not included in the initial investment breakdown depicted in Figure 120 to avoid crowding the presented result. In other words, only equipment costs are considered. The results indicate that the solar components are responsible for 42% of the equipment cost with the heliostat field demanding the most investment. Solar equipment capital investment cost is estimated to be 33.645 MUS$ whereas the bottoming cycle initial investment is 14.537 MUS$. Other noteworthy contributors are the generators (with auxiliaries such as transformers, control systems, and electrical components) and civil engineering with 18% and 15%, respectively. As compared with ABC, the bottoming cycle in MBC
configuration requires more investment. Nonetheless, MBC’s total capital investment is 108.85 MUS$ whereas ABC demands a greater initial investment of 112.0 MUS$. Furthermore, having a hybrid MBC power plant instead of an ABC can save up to 0.299 MUS$/year due to lower annual operating cost.

Figure 119: Central receiver’s outlet temperature throughout the day for a complete year, a) winter, b) spring, c) summer, d) fall

Figure 120: Hybrid MBC power plant equipment’s initial investment breakdown

Additionally, LCOE and net present value approaches are employed to assess the thermo-economic performance of the investigated configurations. Taking into consideration that the LCOE and net present value are correlated, optimizing one of
them (LCOE in this study) implies that the other indicator is optimized as well. The optimum LCOE for the MBC power plant configuration is 75.3 US$/MWh. Taking into consideration that the plant is considered to be located in the UAE and the electricity sale price is 0.07 US$/kWh (70 US$/MWh), it is expected to have a negative net present value. In other words, negative net present value is an indication that the total profit from 25 years of selling the generated electricity to the grid does not justify the necessary initial investment to construct a new hybrid MBC. Nonetheless, the MBC power plant is a more economical alternative as compared with the ABC configuration. Additionally, environmental significances of hybrid power plants justify their economical disadvantages.

4.6.4 Maisotsenko bottoming cycle hybridization.

In this section, hybridization of an already existing MBC power plant is studied. Initially, a reference MBC power plant is presented. It is decided to consider that the non-hybrid MBC power plant operates with optimum design variables acquired in the prior section in order to compare the results for hybrid and non-hybrid MBC power plants. The main characteristics for the reference MBC plant are listed in Table 31.

Table 31: Non-hybrid MBC power plant’s main characteristics

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Assigned value</th>
</tr>
</thead>
<tbody>
<tr>
<td>TCPR</td>
<td>12.0</td>
</tr>
<tr>
<td>BCPR</td>
<td>6.1</td>
</tr>
<tr>
<td>TIT (K)</td>
<td>1532</td>
</tr>
<tr>
<td>MFRR</td>
<td>1.06</td>
</tr>
<tr>
<td>ASDH</td>
<td>0.00</td>
</tr>
<tr>
<td>Annual generated electricity (GWhe)</td>
<td>415.35</td>
</tr>
<tr>
<td>Annual overall energy efficiency</td>
<td>39.07%</td>
</tr>
<tr>
<td>Specific CO\textsubscript{2} emission (kgCO\textsubscript{2}/MWhe)</td>
<td>505.32</td>
</tr>
<tr>
<td>Specific water consumption (ltH\textsubscript{2}O/MWhe)</td>
<td>590.83</td>
</tr>
<tr>
<td>Annual fuel consumption (tonne)</td>
<td>76322</td>
</tr>
<tr>
<td>Bottoming cycle share of electricity generation</td>
<td>27.79%</td>
</tr>
<tr>
<td>Capacity factor</td>
<td>1.0</td>
</tr>
<tr>
<td>Annual operating cost (MUS$)</td>
<td>18.417</td>
</tr>
<tr>
<td>Bottoming cycle cost (MUS$)</td>
<td>14.537</td>
</tr>
<tr>
<td>Total capital cost (MUS$)</td>
<td>62.42</td>
</tr>
<tr>
<td>Net present value (MUS$)</td>
<td>+28.986</td>
</tr>
<tr>
<td>LCOE (US$/MWh)</td>
<td>63.599</td>
</tr>
</tbody>
</table>

It should be noted that the presented MBC power plant in Table 31 is a non-hybrid version of the discussed plant in the previous section. Therefore, pros and cons of MBC power plant hybridization can be studied by comparing the results obtained for hybrid and non-hybrid power plants. A rather insignificant impact of power plant hybridization is a small abatement in the plant’s annual power output due to the pressure
losses in the solar components. In addition, one must note that the combustion chamber has a superior thermal efficiency as compared with the heliostat field’s central receiver. Therefore, a higher percentage of the supplied thermal input is efficiently utilized in the combustion chamber. Therefore, hybridization leads to a drop in the plant’s annual thermal efficiency. On the other hand, the hybrid power plant’s specific CO₂ emission and annual fuel consumption are notably lower as a portion of the required thermal input is delivered by the heliostat field. On an annual basis, consumption of 7000 tonne of CH₄ is substituted by solar energy with power plant hybridization. Consequently, the annual quantity of CO₂ emission is abated by 19,206 tonneCO₂.

As fuel consumption is reduced with power plant hybridization, the plant’s operating cost declines by 1.648 MU$/year. Furthermore, power plant hybridization demands 46.43 MU$ of additional investment. Besides, the non-hybrid MBC power plant’s LCOE is 63.599 US$/MWh while the hybrid configuration’s LCOE is 75.330 US$/MWh. Consequently, generating 1 MWh of electricity when 8.92% of the required thermal input is supplied by the heliostat field costs an extra 11.731 US$ as compared with the non-hybrid configuration. The presented results enable us to estimate the power plant’s hybridization payback period with solar share of 8.92%. Considering the annual revenue obtained from the reduction in the plant’s operating cost and additional investment needed for the plant hybridization, the payback period is about 28 years. One should note that with the recent reduction in fuel cost, power plant hybridization is even more economically unjustified. Bearing in mind that a power block’s lifetime is 25 years and the payback period is 28 years, power plant hybridization is not a profitable investment. Nonetheless, actions must be taken to reduce and restrain the amount of CO₂ emission that might be economically unfavorable.

Utilizing heliostat fields with different capacities for hybridization of an already existing MBC power plant can provide valuable insight in power plant hybridization’s thermo-economic performance. Results concerning the effect of the integrated heliostat field capacity (rate of thermal output during spring equinox noon) on the plant’s annual fuel consumption and specific CO₂ emission are depicted in Figure 121. It should be noted that the optimum heliostat field depicted in Figure 40 contains 1460 mirrors and is capable of supplying 105.8 MW of thermal energy during spring equinox noon. Therefore, the maximum capacity for the integrated heliostat field is obtained when all
1460 mirrors are considered for the final layout. There is no surprise that increasing the incorporated heliostat field’s capacity leads to a drop in the plant’s annual fuel consumption and specific CO$_2$ emission. Integrating a heliostat field with maximum capacity can reduce the plant’s annual fuel consumption by 13.5% from 76,322 tonne to 66,052 tonne of CH$_4$. Understandably, specific CO$_2$ emission is reduced as the plant’s annual fuel consumption decreases. By employing all 1460 mirrors in the field, the plant’s specific CO$_2$ emission is abated by 13.0% from 505.3 kgCO$_2$/MWh to 439.6 kgCO$_2$/MWh. Therefore, the rate of fuel consumption drops by 65.7 kgCO$_2$ for generating 1 MWh of electricity.

![Figure 121: Effects of the heliostat field’s rate of thermal energy output during spring equinox noon on the plant’s annual fuel consumption and specific CO$_2$ emission](image)

In addition, the effects of the heliostat field’s capacity on its solar multiple and plant’s annual solar share are presented in Figure 122. Clearly, increasing the field capacity implies that its solar multiple is increasing. Bear in mind that the solar multiple will be one when the air’s temperature leaving the central receiver reaches 950°C during spring equinox noon. Additionally, one can note that there is a strong correlation between solar multiple and annual solar share. Nevertheless, the aforementioned correlation cannot be further extended for fields with greater capacities since the plant’s solar share is restricted by the operating temperature limits and low insolation periods while the solar multiple can be increased to any desired value. This issue can be clearly seen in the presented results as the rate of increase in the plant’s solar share drops for fields with relatively large capacities. To further increase a plant’s solar share, thermal energy storage is a necessity. With the maximum heliostat field capacity utilizing all
1460 mirrors, the field’s solar multiple and plant’s annual solar share are 1.35 and 13.12%.

Figure 122: Effects of the heliostat field’s rate of thermal energy output during spring equinox on the plant’s solar multiple and annual solar share

Similar to the previous sections, thermo-economic analyses are conducted to further evaluate the impact of the integrated heliostat field’s capacity on the plant’s LCOE and payback period, and the results are depicted in Figure 123. Generally, a plant’s LCOE is constantly increasing as a larger heliostat field is incorporated in the plant. Therefore, one can conclude that the abatement in the plant’s fuel consumption does not justify the extra investment required for hybridization. In addition, the payback period approach is utilized to provide an extensive understanding of the power plant’s hybridization feasibility. For the payback period analysis, two factors are taken into consideration. Initially, the annual profit obtained as a result of annual operating cost reduction is determined. Additionally, the extra investment needed to hybridize the MBC power plant is calculated. Consequently, the payback period is computed by calculating the ratio of the additional investment over the annual profit, bearing in mind that the additional maintenance and labor expenses for the hybrid power plant were intentionally ignored. This issue is further discussed in Figure 124. Note that integrating small capacity heliostat fields is not advised due to their long payback periods. Furthermore, the optimum payback period of 27.35 years is achieved with an 80 MWth heliostat field. It is noteworthy that an 80 MWth heliostat field’s solar multiple is 1.02. Consequently, it is more economical to implement a heliostat field with a solar multiple value close to one.
Figure 123: Effects of the heliostat field’s rate of thermal energy output during spring equinox on the plant’s annual LCOE and payback period

Moreover, the life cycle saving approach is employed to further investigate the effects of the heliostat field capacity. The results concerning the effect of the heliostat field capacity on the hybridization cost, life cycle saving, and annual profit are depicted in Figure 124. It should be noted that the negative value for the hybridization cost is an indication of the additional investment demanded by hybridization. Similarly, the negative value for the annual profit implies that hybridization results in annual losses. It was previously mentioned that the payback period is calculated without considering the additional maintenance and labor expenses for a hybrid power plant, noting that the annual profit will become negative by considering the aforementioned factors and the payback period approach cannot be applied. In fact, the payback period is presented only to assess the impact of the heliostat field’s capacity variation. Results indicate that power plant hybridization results in a rise in the plant’s operating and maintenance cost. Therefore, life cycle loss is constantly rising. Nonetheless, the annual losses due to hybridization are minimized by integrating an 80 MWth heliostat field. The minimum reported value for the annual losses is 0.0565 MUSS. Note that as hybridization annual losses abate, life cycle saving value approaches the additional investment required for the plant’s hybridization.
In the previously presented results, the optimum heliostat field’s layout depicted in Figure 40 was implemented. Nonetheless, it should be noted that the most insignificant change in the heliostat field’s initial characteristics, such as the central tower, mirrors, and receiver’s dimensions will change the optimum orientation of the mirrors’ spread in the field. Therefore, variation in the aforementioned variables demands another field optimization. In other words, the integrated heliostat field is constantly optimized for every single point presented in the following sections. In addition, heliostat field sizing is not considered in this section to better present the impact these design variables have on the plant’s and field’s performances.

Results concerning the effect of the central tower height on the field’s annual thermal output and the plant’s solar share are depicted in Figure 125. Results indicate that implementing a taller central tower improves the field’s annual thermal output. In other words, the field weighted efficiency is enhanced. As a result, the plant’s solar share is increased, too. However, it is noteworthy that the plant’s solar share is not constantly rising. A maximum solar share of 13.14% is achieved with a 135m central tower, bearing in mind that taller central towers can provide more thermal output annually. Nevertheless, the additional thermal input cannot be employed in the plant due to the central tower and receiver’s operating temperature limits. Therefore, a reduction in the plant’s annual solar share is reported.
In addition, the central tower’s height impacts on the plant’s LCOE and payback period are displayed in Figure 126. We must take into consideration that implementing a taller central tower improves the plant’s solar share which leads to a drop in the plant’s operating cost. On the other hand, the pressure drop is more significant in tall central towers degrading the plant’s annual power output. These factors have opposing effects on the plant’s LOCE as can be seen in the presented results. A minimum LCOE of 81.33 US$/MWh is attained when the central tower height is 85m. However, one must note that while increasing the central tower height results in a more expensive generated electricity, the generated electricity is utilizing a higher percentage of solar energy. In other words, the generated electricity is cleaner as the solar share increases and the specific CO₂ emission drops.

The payback period is a better indicator to select a more appropriate central tower for power plant hybridization. A minimum payback period of 29.54 years is obtained by having a 105m central tower. In addition to the payback period and LCOE approach, other economic indicators for the hybridization of an already existing MBC power plant including hybridization extra investment, annual profit, and life cycle savings are presented in Figure 127, noting that the heliostat field should be further expanded if a short central tower is implemented. Therefore, increasing central tower height does not always require more investment. Moreover, the hybridization annual loss is minimized by having a 105m central tower which is identical to the optimum
central tower height reported by the payback period, bearing in mind that the minimum annual loss is 0.19MUS$.

Figure 126: Effects of central tower height on the plant’s LCOE and payback period

Figure 127: Effects of central tower height on the hybridization cost, annual profit, and life cycle saving

Another important design variable is the mirrors’ surface area implemented in the field to redirect solar radiation toward the tower. Results concerning the effect of the mirrors’ size on the field annual thermal output and plant’s solar share are shown in Figure 128. It is predictable that increasing the mirrors’ surface area leads to a rise in the heliostat field’s thermal output and weighted efficiency. Accordingly, the plant’s solar share increases as well, implying that the specific CO₂ emission and annual fuel consumption are abated.
Furthermore, the effects of the mirrors’ sizing on the plant’s LCOE and payback period are depicted in Figure 129. The presented results indicate that implementing larger heliostats results in an increase in the plant’s LCOE. It is understandable that larger mirrors reduce the plant’s annual operating cost. Nonetheless, greater investment is required due to a larger set of mirrors and a more expanded field which is not justified by reduction in the plant’s operating cost and fuel consumption. Furthermore, the payback period analysis indicates that the most economical mirror size is 75 m² with a minimum payback period of 27.83 years. We must also take into consideration the fact that fields with larger mirrors are more environmentally friendly. Results concerning the effects of the mirror’s surface area on the hybridization cost, annual profit, and life cycle savings are shown in Figure 130. As it previously mentioned, increasing the employed mirrors’ surface area requires a greater initial investment. As a result, the hybridization life cycle losses are continuously increased. Nonetheless, the obtained results for the hybridization annual losses tell a different story. Results indicate that annual loss is minimized to 0.087 MUS$ by having 100 m² heliostats in the field.
The final design variable considered in this section is the central receiver’s dimensions. It should be noted that increasing the receiver’s dimension improves the field spillage factor, as the receiver is capable of intercepting a higher percentage of the redirected solar radiation. The impact of the receiver’s dimensions on the field’s annual thermal output and plant’s solar share are depicted in Figure 131. The presented result is an indication of the ineffectiveness of increasing the central receiver’s surface area to values greater than 50 m². For instance, increasing the receiver’s surface area from 50 m² to 84 m² improves the field’s annual thermal output by 2.4 GWh/year from 202.4
GWh to 204.8 GWh. Similarly, solar share variation for receivers with surface areas greater than 50 m$^2$ is insignificant. Implementing a receiver with 84 m$^2$ surface area instead of a 50 m$^2$ receiver enhances the plant’s solar share by 0.09 % points from 13.27% to 13.36%.

![Figure 131: Effects of the receiver’s surface area on the heliostat field’s annual thermal output and plant’s solar share](image)

Results concerning the effect of the receiver’s surface area on the plant’s LCOE and payback period are depicted in Figure 132. Clearly, central receivers with surface area greater than 50 m$^2$ should not be implemented as this only results in extra expenses without any tangible improvement in the field’s performance. Additionally, the presented results indicate that the hybridization payback period is minimized to 29.86 years by implementing a 4m × 9m central receiver. To further evaluate the impact of the receiver’s dimensions on the plant’s thermo-economic performance, additional investment required for the power plant’s hybridization along with the annual profit and life cycle saving are presented in Figure 133. Noting that larger receivers demand more investment, the extra investment required for the power plant’s hybridization is constantly rising. Nonetheless, this is not the case for the life cycle saving. For instance, increasing the receiver’s surface area from 14 m$^2$ to 24 m$^2$ abates the hybridization life cycle losses. Therefore, it can be perceived that the economic advantages of fuel saving outweigh the extra investment needed to increase the receiver’s dimensions. Nonetheless, this is the only case in which the life cycle saving is enhanced. In all other cases, increasing the receiver’s dimensions only leads to a more uneconomically
justified scenario. Moreover, results indicate that the optimum receiver’s dimensions which minimize the annual losses to 0.208 MUSS are $4m \times 9m$.

![Graph](image1)

**Figure 132:** Effects of the receiver’s surface area on the plant’s LCOE and payback period

![Graph](image2)

**Figure 133:** Effects of the receiver’s surface area on the hybridization cost, annual profit, and life cycle saving

### 4.6.5. Concluding remarks.

In this section, the MBC power plant hybridization was comprehensively studied. Initially, an already existing hybrid gas turbine power plant was considered for bottoming cycle integration. Two bottoming cycle alternatives, i.e. air and Maisotsenko bottoming cycles, were selected for heat recovery purposes. A comparative analysis between these two bottoming cycles was carried out. Furthermore, thermo-economic
optimization was performed to evaluate the possibility of a hybrid MBC power plant in Abu Dhabi, UAE. The optimization results were evaluated against the results for a hybrid ABC power plant to further discuss the advantages and disadvantages of MBC power plant configuration. In the final section, hybridization of an already existing MBC power plant was extensively evaluated by studying the effects of the heliostat field capacity, central tower height, central receiver’s dimensions, and mirrors’ surface area on the field’s and plant’s thermo-economic performances.

In general, results are in the favor of MBC’s superiority over ABC power plant configuration, both economically and thermodynamically. Bear in mind that two scenarios were selected to properly evaluate MBC’s performance against the ABC configuration. First, implementation of a bottoming cycle in an already existing hybrid gas turbine cycle was investigated. The optimum reported LCOE for MBC and ABC configurations are 75.2 US$/MWh and 77.7 US$/MWh, respectively. Next, thermo-economic optimization was carried out for installment of a new hybrid power plant. Results indicated that the optimum LCOE for MBC and ABC are 75.3 US$/MWh and 77.8 US$/MWh, respectively.

Additionally, hybridization of an already existing MBC power plant was considered by studying the effects of different heliostat field design variables such as field capacity, central tower height, mirrors’ surface area, and receiver’s dimensions. In general, it can be perceived that power plant hybridization is not a cost-effective modification. Nonetheless, the plant specific CO₂ emission can be notably reduced. Nowadays, climate change has become a global concern. Therefore, hybridization can be considered as one of the more economically justified solutions. Additionally, one must note that solar components’ required capital investments are expected to drop in the near future due to rapid technological improvement and special attention given to this issue.

4.7. Comparative Analysis between Two Technologies

Finally, it is decided to provide a general comparison between the proposed configurations’ results and the already operating Shams 1 solar thermal power plant using parabolic trough. It should be noted that in principle the proposed configurations are significantly different from Shams 1; nonetheless, it is of high interest to have a general comparison since both configurations are implemented in the Abu Dhabi. The comparative analysis between the proposed hybrid power plants and Shams 1 is
presented in Table 32. Drawing any significant conclusion from the presented results are not advised due to the notable differences between the configurations. Nonetheless, two important factors must be addressed. Initially, the reported reduction in annual CO$_2$ emission for Shams 1 is a rough approximation. It should be noted that the presented source [32] did not discuss the considered approach for calculating the amount of CO$_2$ emission reduction. It is unknown whether the reported result is calculated by considering the amount of thermal energy delivered by the integrated solar collector or by taking into account the specific CO$_2$ emission associated with the annual power generated. Secondly, the reported approximate electricity cost is the ratio of the initial capital investment over the total amount of electricity generated during the plant lifetime. Therefore, it is only an approximation of the economic aspects of the discussed configurations and is not an indication of the economic advantages of the proposed configurations over the Shams 1 power plant.

Table 32: Comparative analysis between the proposed hybrid power plants and Shams 1 [31, 32, 162, 163]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Shams 1</th>
<th>SBC</th>
<th>ABC</th>
<th>HABC</th>
<th>MBC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gross turbine capacity (MWe)</td>
<td>100</td>
<td>50</td>
<td>50</td>
<td>50</td>
<td>50</td>
</tr>
<tr>
<td>Integrated CSP technology</td>
<td>PTC</td>
<td>HFC</td>
<td>HFC</td>
<td>HFC</td>
<td>HFC</td>
</tr>
<tr>
<td>Configuration</td>
<td>Solar-only</td>
<td>Hybrid</td>
<td>Hybrid</td>
<td>Hybrid</td>
<td>Hybrid</td>
</tr>
<tr>
<td>Power block configuration</td>
<td>Rankine cycle</td>
<td>SBC</td>
<td>ABC</td>
<td>HABC</td>
<td>MBC</td>
</tr>
<tr>
<td>Heat transfer fluid</td>
<td>Oil</td>
<td>Air</td>
<td>Air</td>
<td>Air</td>
<td>Air</td>
</tr>
<tr>
<td>CSP nominal outlet temperature (°C)</td>
<td>393</td>
<td>950</td>
<td>950</td>
<td>950</td>
<td>950</td>
</tr>
<tr>
<td>Solar field aperture area (m$^2$)</td>
<td>627,840</td>
<td>80,229</td>
<td>99,537</td>
<td>98,698</td>
<td>99,058</td>
</tr>
<tr>
<td>Annual Solar Share</td>
<td>75%</td>
<td>8.87%</td>
<td>8.12%</td>
<td>8.57%</td>
<td>8.92%</td>
</tr>
<tr>
<td>Annual Operating hours</td>
<td>2200</td>
<td>8760</td>
<td>8760</td>
<td>8760</td>
<td>8760</td>
</tr>
<tr>
<td>Annual CO$_2$ emission avoided (tonne)</td>
<td>175,000</td>
<td>15,606</td>
<td>17,988</td>
<td>18,560</td>
<td>19,206</td>
</tr>
<tr>
<td>CO$_2$ emission Reduction per 1 m$^2$ of mirror (kgCO$_2$/m$^2$)</td>
<td>278.7</td>
<td>194.5</td>
<td>180.7</td>
<td>188.0</td>
<td>193.9</td>
</tr>
<tr>
<td>Capital Investment (MUS$)</td>
<td>350</td>
<td>152.37</td>
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<td>112.29</td>
<td>108.85</td>
</tr>
<tr>
<td>Annual Power generated (GWh)</td>
<td>220</td>
<td>419.27</td>
<td>409.36</td>
<td>411.01</td>
<td>413.83</td>
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<tr>
<td>Plant lifetime (years)</td>
<td>30</td>
<td>25</td>
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<tr>
<td>Approximate electricity cost (US$/MWh)</td>
<td>53.0</td>
<td>14.5</td>
<td>10.9</td>
<td>10.9</td>
<td>10.5</td>
</tr>
</tbody>
</table>

*Presented results for the Shams 1 power plant are approximate published data*
Chapter 5: Conclusion and Recommendations

In this research work, feasibility of a 50 MWe hybrid (solar and natural gas) combined cycle power plant with a topping gas turbine cycle and four different bottoming cycles were assessed. Power plant hybridization was accomplished by employing a solar tower collector (heliostat field collector). Three rather unconventional bottoming cycle configurations were chosen including gas turbine (air bottoming cycle), water injected gas turbine (humid air bottoming cycle), and the Maisotsenko cycle (Maisotsenko bottoming cycle). These three configurations along with the conventional combined cycle power plant (steam bottoming cycle) were optimized by conducting thermo-economic and transient analyses in MATLAB to identify the most economically justified plant configuration for the United Arab Emirates. Additionally, two different heliostat field layouts were taken into consideration including the radial-staggered and spiral layouts. Moreover, thermo-economic evaluation was accomplished by utilizing five different economic approaches, i.e. net present value, payback period, life cycle saving, Knopf objective function, and levelized cost of electricity.

5.1. Conclusion

Initially, heliostat field analysis indicates that the optimization method, developed by Collado and Guallar [116], improved the radial-staggered heliostat field layouts by determining the local radial increment between the rows of the mirrors. Consequently, the radial-staggered layout presented better performance than the spiral layout in all the considered objectives, i.e. annual weighted efficiency, annual unweighted efficiency, and levelized cost of energy. Results indicated that the radial-staggered and spiral field layouts’ optimum weighted efficiencies were 58.61% and 58.38%, respectively. Additionally, the minimum reported levelized cost of energy for the radial-staggered and spiral layouts were 33.88 US$/MWh and 34.03 US$/MWh, respectively.

Furthermore, a comparative analysis was conducted for a non-hybrid version of the proposed configuration to investigate the most thermo-economically efficient bottoming cycle. In general, SBC power plants can operate with the highest thermal efficiency whereas an optimized MBC power plant configuration can be the most economical alternative followed by the HABC configuration. Utilizing Knopf’s [135]
assessment approach, MBC displayed the best economic performance with the lowest total operating cost of 0.7799 US$/s followed by HABC, ABC, and SBC power plants’ configurations with total operating costs of 0.7901 US$/s, 0.7970 US$/s, and 0.8093 US$/s, respectively. On the other hand, the life cycle saving analysis for the integration of a waste heat recovery bottoming cycle within a 43.2 MWe gas turbine power plant indicated that SBC incorporation leads to the maximum revenue. Whereas, similar analysis for a 10.8 MWe gas turbine presented the MBC implementation as the most cost effective alternative. Finally, transient analysis showed that MBC is the most cost effective power plant configuration for hot and humid climates with an LCOE of 64.41 US$/MWh followed by HABC, ABC, and SBC with reported LCOE values of 65.75 US$/MWh, 66.36 US$/MWh, and 68.88 US$/MWh, respectively.

An extensive analysis was conducted for hybrid adaption of the proposed configurations considering all possible scenarios. In general, one can conclude that the MBC configuration is the most cost effective configuration for the installment of a 50 MWe hybrid power plant followed by HABC, SBC, and ABC configurations. It is important to mention that the optimized LCOE for MBC was 75.330 US$/MWh while the minimum achievable LCOE for HABC, SBC, and ABC configurations were 77.075 US$/MWh, 77.663 US$/MWh, 77.763 US$/MWh, respectively. Additionally, the positive impact of water injection in ABC configurations’ bottoming cycle air stream were repeatedly presented and discussed. Acquired results for the bottoming cycle integration showed that water injection in ABC power plants can improve their heat recovery capability and reduce the plant LCOE. In particular, integrating an aftercooler which cools down and humidifies the air stream prior to the heat recovery heat exchanger can be an effective approach to improve the plant’s performance. Moreover, advantages of air saturator and Maisotsenko cooler integrations in ABC power plants was clearly discussed. In particular, implementation of a bottoming cycle in an already existing hybrid gas turbine cycle was investigated. Taking into consideration that the MBC configuration studied in section 4.6.2 was, in principle, an ABC power plant with integration of a Maisotsenko cooler after the bottoming cycle compressor for aftercooling purposes. As a result, MBC configurations do not necessarily need to have a complicated air saturator to operate. In other words, the presented results indicated the notable advantages of Maisotsenko cooler implementation in ABC power plants.
5.2. Recommendations

As a final note, hybridization might not be an economically justified investment. Nevertheless, it can be one of the more cost effective solutions to fight against global warming with the current renewable energy technologies’ level of maturity. Furthermore, one may argue that the specific CO₂ emission for the non-hybrid adaption of an SBC power plant is 409.6 kgCO₂/MWh whereas the reported specific CO₂ emission for hybrid MBC, HABC, and ABC configurations are 460.8 kgCO₂/MWh, 468.2 kgCO₂/MWh and 483.88 kgCO₂/MWh, respectively. Taking into consideration that the only incentive for power plant hybridization is to reduce the rate of CO₂ emission, a non-hybrid SBC configuration is a cleaner technology than the hybrid version of the other proposed configurations. Accordingly, other proposed configurations’ economic advantages over the hybrid SBC might be deemed irrelevant as a non-hybrid SBC is cleaner and cheaper. Nonetheless, it should be noted that the capacity factor of one is considered for the analysis of this research work. Consequently, it can be said that solar energy is only utilized for power generation for about one third of the plant’s life cycle. This assumption significantly augmented the hybrid configurations’ CO₂ emission. In other words, the hybrid plant’s CO₂ emission is significantly greater during night time. Therefore, by limiting the plant’s operation to daytime, only a hybrid MBC or HABC plant can deliver clean technology as compared with the non-hybrid SBC and a more cost effective alternative than the hybrid SBC’s configuration. Furthermore, the presented analysis and results can be a platform for conducting research on solar-only power plants utilizing the proposed configurations. There is a possibility that the MBC and HABC configurations would be a more cost effective option for small scale solar-only power plants. Additionally, integration of thermal energy storage can further put MBC and HABC configurations in an advantage as compared with a hybrid SBC configuration. Nonetheless, the hybrid SBC configuration is the cleanest combined cycle configuration, and it is advised to be utilized as long as the economic aspect of the plant is not a concern.

There are quite a few possibilities to further extend the investigations, analyses, and results presented in this research work. First, integration of thermal energy storage can be investigated to further increase the plant’s solar share without reducing the capacity factor [164-168]. Alternatively, a solar-only configuration can be investigated utilizing thermal energy storage. Furthermore, other waste heat recovery bottoming...
cycles such as the organic Rankine cycle and Kalina cycle can be considered to provide a more comprehensive evaluation of all the waste heat recovery possibilities [169-172]. Additionally, power augmentation techniques can be utilized to improve the plant’s thermo-economic performance. The power augmentation approaches consist of inlet air cooling, supplementary firing, and water/steam injection in the topping cycle combustion chamber. For inlet air cooling, the desiccant based air cooling technique or absorption systems can be implemented to utilize waste heat for air cooling purposes. Additionally, supplementary firing can be achieved by either heating up the topping cycle exhaust gases in a supplementary combustion chamber or implementing a parabolic trough in the bottoming cycle to further heat up the bottoming cycle fluid. Finally, for the steam injection approach in the topping cycle, water can be heated by the available waste heat or a low temperature solar collector. Moreover, there are different techniques that can be employed for solar and hybrid power plants’ investigation such as exergo-economic and environomic analyses [173]. In addition, the integrated solar collector can be further investigated by considering alternative heliostat field design considerations. For instance, the aforementioned beam down tower design or hillside central receiver design by Noone et al. [174] and Slocum et al. [175] can be considered. In addition, solar reforming can be investigated instead of thermal energy storage integration. In other words, a heliostat field with solar multiples greater than one can be implemented to be utilized for both preheating and solar reforming. It is important to mention that solar fuel production and solar reforming are considerably popular topics [3, 4, 176-178].
References


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Vita

Mohammad Saghafifar attended Towheed High School in Shiraz, Iran, from 2004 to 2007 and graduated from Adab Iranian High School in Dubai, United Arab Emirates, in 2008. He continued his post-secondary education in the Mechanical Engineering Department at the American University of Sharjah in 2008. He received his bachelor’s degree in Mechanical Engineering in July of 2012. During his bachelor’s degree, he was a recipient of Merit and Petrofac Scholarships as well as Dean’s List and Chancellor’s awards.

Mr. Saghafifar began pursuing his graduate studies in September 2012 for a master’s degree in Mechanical Engineering at the American University of Sharjah where he received a Graduate Teaching Assistantship award for the complete duration of his master’s degree. During his master’s degree, he published seven journal papers in prestigious and peer reviewed international journals. One of the published journal papers was featured by the Advances in Engineering website as a Key Scientific Article contributing to excellence in engineering, scientific and industrial research. Advances in Engineering is a highly selective venue with the invited articles being less than 0.1% of all published literature. Additionally, he presented two papers in different international conferences. Mr. Saghafifar has also prepared and submitted seven journal papers which are currently under review in different scientific journals. Furthermore, he has conducted research on other areas including but not limited to gas turbine inlet cooling, solar air conditioning, desiccant air conditioning, Photo-voltaic/thermal collector, and pulse combustor integration.