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Thermodynamic and technoeconomic comparative justification of a waste heat recovery process with integration of multifluid and indirect evaporative cooler

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ABSTRACT

For a well-developed, efficient and feasible system, it is necessary to produce power generation enormously with a reduction in harmful emissions like Carbon dioxide (CO_2), Carbon monoxide (CO), Nitrogen (N), Nitrogen oxide (NOx), and Sulphur dioxide (SO_2). Waste heat gases emit directly into an environment, it has many adverse effects on the environment including global warming, environmental pollution, and effect on human health as well. Researchers believe that a thermally efficient system could be achieved by converting waste heat gases into net power output. From this system, the efficiency obtained is 5% to 8% unable to meet the space and cost demands of this waste heat recovery (WHR) system. For waste heat recovery, the most typical cycles used for this are the Rankine cycle and Brayton cycle. Even though these are the best cycles but their efficiency is not as such maximum. By observing all these aspects, there is a different way of recovering waste heat and that is an indirect evaporative cooler (IDE). An indirect evaporative cooler is beneficial in terms of enormous power generation, getting maximum efficiency, low operating cost, and acquiring a sustainable system. The focus of current research was to recover industrial waste heat gases exhausted from SP boilers in the cement industry. ASPEN HYSYS software is used for generating a waste heat recovery model that further operates on the Maisotsenko cycle (M cycle). The topping cycle and bottoming cycle are used in this model. Both the working fluid air and binary mixture CO_2-C_7H_8 operated in a model. By manipulating the model with working fluid air, this system generated a net power output of 68.53 MW with 35.44% thermal efficiency. Integrating the model with a binary mixture of CO_2-C_7H_8 permits 48.59 MW output power with a 38.57% efficiency value. Comparison analysis is performed for extracting the best optimal parameters with extreme power generation and the greatest efficiency value. The industrial operating parameters of the Bestway cement industry operated in this developed model present 38.04 MW and 30.63 MW of power generation with 27.78% and 27.77% efficiency by executing both fluids air and CO_2-C_7H_8 mixture. A techno-economic analysis (TEA) is performed for this entire waste heat recovery system which exhibits a cost of $30/MWh along 3 years payback period.

1. Introduction

With the increase in demand for energy and current lifestyle, electricity plays a vital role in the life of human beings. In recent days a variety of methods are proposed in power cycles to enhance electricity production. The most proposed cycle for electricity production is the Rankine cycle and Brayton cycle all over the world. Gas turbine (operates on Brayton cycle) needs an open cycle power plant just because of natural gas lower price. Rankine cycle demands a closed cycle power

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plant in which working fluid steam or water is used. A combined cycle power plant is used for getting maximum efficiency. Non-renewable

Nomenclature

AMC	Annual Maintenance Cost
CRF	Capital Recovery Factor
FAC	First Annual Cost
HMX	Heat and Mass Exchanger
IEC	Indirect Evaporative Cooler
LCOE	Levelized Cost of Energy
MHABC	Maisotsenko Humid Air Bottoming Cycle
ORC	Organic Rankine Cycle.
O&M	Operation and Maintenance
PWCC	Present Worth of Capital Cost
PP	Payback Period
SP	Suspension Preheater
TEG	Thermoelectric Generators
TEA	Techno-Economic analysis
TAC	Total Annual Cost
WHR	Waste Heat Recovery
WF	Working Fluid

sources such as the conventional cycle use fossil fuels that contribute to polluting the environment and are not a reliable system. In a wind turbine, manipulating photovoltaic cells with a combination of wind energy and solar irradiation is a beneficial method for electricity production. These are sustainable systems but they are limited in terms of high cost, storage issues, etc. For producing electricity in extensive amounts, the organic Rankine cycle (ORC) is the best choice without fossil fuel burning and needs high temperatures. The organic Rankine cycle is a Rankine cycle actually, but it differs in the working fluid. This uses organic fluids as working fluids rather than steam or water. ORC is growing nowadays because of minimum maintenance demand, operational pressures are vital and self-governing operation [1].

In the present decade, the engineering industry's focus is to remove hazardous emissions of greenhouse gases and efficiency improvement of the plant. By taking into consideration waste heat gas utilization is favorable in terms of lower fuel consumption, reduction in hazardous emissions, and efficiency improvement. A variety of methods are proposed for waste heat recovery. Waste heat rejection is possible at any temperature. Higher waste heat temperature is favorable in terms of efficiency [2]. Literature review on this waste heat presents the analysis of heat transfer of Maisotsenko Humid air Bottoming cycle (MHABC) by utilizing energy recovery and degree of humidification as well. An indirect evaporative cooler is used in this system and an air saturator, heat, and mass exchanger (HMX) model is generated. The extracted results highlight that 58 MW output power is obtained using MHABC along with 30% efficiency at the exit of the air saturator with 70% relative humidity. The effectiveness of the dew point achieved is 0.8 (80%) with a relative humidity of 32% at the air saturator exit. Using a 50% extraction ratio between the lower air saturator and heat exchanger waste heat is recovered more than 8 MW [3].

By using CO_2 as a working fluid, the relative assessment and the exergetic performance of the bottoming power cycle are analyzed. Air condenser modelling is done in it by using the outlet temperature equations. The energy balance equations are carried out by using Aspen Plus followed by the computation of the first law efficiency and then, the second law is calculated using different equations. The maximum gain of the exergetic efficiency for bottoming preheating cycle (BPHC) and the bottoming simple regenerative cycle (BSRC) are observed to be 18.71% and 26.83% respectively with a working temperature of 313 Kelvin. The overall efficiencies gained for the BPHC and for the BSRC are observed to be 10.12% and 28.92% respectively. For the greater ambient conditions of the temperature, BPHC is assessed as a great choice by considering all the factors of thermodynamic performance, investment cost, and combination of the area of heat transfer and overall heat transfer coefficient [4].

The waste heat recovery generation (WHRG) model is designed and simulation performed in Aspen HYSYS software. The simulations that HYSYS software gives are steady-state simulation and dynamic process simulation. HYSYS also provides advanced process control, the results of pinch analysis, heat transfer area, and rough cost estimation. The design of the waste heat recovery generation initializes from the collection of data from the cement plant such as operating data, flue gas properties, and water resources. By using the performance analysis, WHRG boiler (heat exchanger) performance analysis is performed using simulations. By using the distributed parameter model and Hysys petroism, icon software package the heat exchanger is modelled. By applying the pinch point, approach point to software and defining the inlet, outlet temperature provides simulation results mapping of Suspension Preheater (SP) and Air quenching cooler (AQC) boiler was presented in results with and without the heat loss of 10% and bypass of 40% because of change in the environment. Using this way, simulation results present that the power generation is around 20 Mega Watt (MW) and WHRG has the ability to minimize the 14 Mega Ton (MT) CO_2 on yearly basis [5].

There are extensive waste heat recovery methods proposed in the literature. The Brayton cycle is one of those methods and this is a thermodynamic cycle that works on the constant heat engine. Brayton cycle has three main components including a compressor, mixing chamber, and expander. The Brayton cycles are mostly used in gas power plant and airplanes because of less maintenance cost [6]. Another such method is the Stirling engine which works on the compression and expansion of the working fluid at various operating temperatures. At a low temperature of 300° F, the organic Rankine cycle system can harvest energy. Heat sources from the steam system could not show temperatures less than 500° F. In compressor heat recovery systems, Organic Rankine cycle is mostly used where it is mainly used in geothermal plant as a renewable energy harvesting [7]. A huge amount of electricity is produced from the hydropower sources or nuclear power sources. Each of these has a negative effect on the environment. The Nuclear power plant is a cause of nuclear meltdown. On the other hand, fossil fuel burning produces CO_2 emissions and global warming. Thermoelectric generators are used to overcome these problems [8]. Overall total industrial energy, the cement industry consumes 11% (720 Mega Watt) on annual basis. Depending upon the advancement of the cement plant and age the annual consumption of electricity lies within the range of 90 kWh to 130 kWh. Out of 24 existing cement plants, 9 of them installed waste heat recovery systems. According to the waste heat recovery, World Bank's 100–200 Mega Watt waste heat recovery electricity potential is for the cement industry [9].

Thermoelectric generators contain an array of p-type along with n-type semi-conductors as well as by providing heat on one end the cooler heat will sink on the other end generating electricity. P and N type materials are arranged in such a way that they should be thermally connected in parallel and electrically these are in series. The main advantage of the waste heat power system is cost saving economically. There are too many high electricity rates in New York, Connecticut, and Alaska. At a temperature difference of 30 °C, the thermoelectric generator economic effect includes the emission of NO_2 and SO_2 which is the major component in the coal power plant [10]. At a temperature difference of 30 °C and by using 24 modules, the thermoelectric generators 12.41 W of power is generated and it's designed by Hsu et al. The world's most efficient thermoelectric modules depend upon Bismuth-Tellurium and this was introduced by Sano et al. [11]. For enhancing the efficiency of a power plant, the combined cycle power plant is also used. The two main operations are performed in this cycle. A simple gas turbine cycle and heat recovery steam generator. From the turbine exhaust waste heat is recovered in this simple cycle and then that steam is provided to the water for the steam generation and further this steam is utilized in the steam turbine for producing electricity [12]. By using software ASPEN HYSYS Combined cycle power plant of triple pressure is modelled and simulated for the off-design operation by Zuming Liu et al. By comparing it with the equivalent model by Gate Cycle, they checked the performance of their model and simulation results. The obtained result presents the average difference between net power output and thermal efficiencies [13]. Saddiq H. A et al. uses Aspen Hysys software to study the modelling of turbine exhaust turbine and simple gas turbine for various possible configurations. On simple and the combined cycle power plant they studied the effect of operating parameters [14].

By using the ASPEN HYSYS simulator and with and without air intake cooling air, E.N Achimnole simulated a gas turbine power plant. They utilized the actual operating data. At Operating parameters like thermal Efficiency, heat rate, specific fuel consumption, and net power output their analysis present that the simulation results provide high reality if it is compared with the actual operating data [15]. Jawad et al. use ASPEN HYSYS software for modelling and simulating the Al Kayrat Power Plant. By comparing the obtained results with the actual operating data and operating conditions a high validity of results is obtained [16]. By improving the performance of a simple gas turbine in the Al kayrat Power Plant, waste energy is utilized in a huge amount and that is linked in the turbine unit with the outlet exhaust gases. A Steam generator for heat recovery is installed in a combined cycle power plant for utilizing this waste energy. For a simple gas turbine, the results of modelling are compared with the actual operating data that gives high validation and reliability to use ASPEN HYSYS software for the combined cycle power plant [17]. Chan et al. presented a review paper based on different approaches by utilizing industrial waste recoveries like adsorption pumps, absorption pumps, storage of thermal energy, and heat pumps (chemical) for generating power [18]. Kwak et al. presented a framework and model by using the various kind of technologies for industrial waste heat such as organic Rankine cycle system, heat pumping, and heating of boiler feed water. The main key objective was such that to present the accuracy of the suggested method for optimization. During winter using ORC reduces the cost by almost 6.4% [19]. An Off design model is proposed by Mazzi et al. to find out the manner of organic Rankine cycle (ORC) linked with the industrial waste heat recovery at steady and transient state conditions. At cold sink constant temperature, the obtained output results show that the system efficiency minor decreases by about 1% [20].

It is found that by enhancing the pressure of the boiler to increase the efficiency of the cycle the recovered energy amount decreases. The optimum point is found at a pressure value of 1398 kPa where maximum energy and exergy efficiencies are achieved. By changing some of the operating factors like environmental temperature, maximum cycle efficiency, and condenser pressure there is no effect on the boiler pressure and boiler pressure remains constant. From the rotary kiln, the exhaust gases of the cyclone preheater and Kalina cycle utilization for generating electricity in the Brazilian cement industry are estimated. Utilization of waste heat properly gives the net power generation ultimately 38,400 Mega Watt per year. Its main benefit is cost saving in terms of electricity and 32.98% of electricity savings on an average basis. Green House gas emissions are also minimized by about 104,965 tons per year which results to make the cement plant sustainable. Moreover, the benefits achieved a decrease in cement production Kilo watt hour per ton from 122 kWh/ton to 112 kWh/ton. Fauji cement contributes 10 MW and Askari cement contributes up to 7 MW to utilize the hot flue gases not only to improve performance but also to minimize power sources based on fossil fuels [21]. To utilize waste heat gases properly Fecto cement industry installed 6 MW power plants. The hot gases with the steam production rate of 47 tons per hour passed through two sets of boilers (heat exchangers). One is an SP boiler and another one is the AQC boiler. Waste heat recovery systems are designed to utilize the waste heat gases efficiently [22]. Operating parameters like clinker, the temperature of cooling air, masses of cooling air, and grate speed to improve efficiencies of grate cooling system optimized by Ahmed et al. Energy cooling system and exergy cooling system recovered efficiencies and also exhaust air was enhanced by 21.5% and 9.5% by recovery of waste heat. By changing clinker mass flow rate and cooling air flow rate 38.10% and 30.86% of energy cost are saved respectively [23]. There is a possibility of generating electrical energy by converting waste heat by using the micro thermoelectric generator. By using integrated circuit technology the thermoelectric generator could be fabricated [24].

There are many technologies for commercial waste heat. If a process utilizes a waste energy source or generates waste heat then, the main criterion is temperature. For waste heat, the low temperature is not preferred directly but the high- temperature waste heat could be recycled and it creates a broad processing region. Waste heat is divided into three main components. More than 650 °C is high temperature, a range of 230°-650 °C is medium temperature and low temperature is ultimately equal to 230 °C [25]. A power generation system that's based on the coupling of Organic Rankine cycle and PCM- based technologies for enhancing the efficiency of waste heat recovery system was modelled by Dal Magro et al. Waste heat recovery PCM-based minimizes fluctuations. The obtained results present that by using technology based on PCM the system thermal efficiency increases from 15.5% to 16.4% [26]. The System of power generation depends on coupling the Organic Rankine cycle with waste heat recovery of chemical company flourished by Lee et al. The working fluid that was used in it was R134a. The results show that the system generated 193.75 Kilo watts of net power output and 4.701% thermal efficiency [27]. Gas waste heat combined with solar energy generated from parabolic trough collectors is proposed by Bellos and Tzivanidis. They put waste heat into an economizer and then solar heat into a superheater. In a few cases, the waste heat is combined used with the evaporator. The obtained results highlight that the best working fluid is toluene for investigation. When the temperature of gas enhances from 150 °C to 300 °C then the system efficiency also increases in the range of 11.6% to 19.7% [28]. The waste heat of low grade changes into a waste heat of high grade by using the technology of ORC. Industrial waste is mostly used as heat in this technology. For waste heat of low-temperature ORC has many benefits like it requires a small pressure, better economy, an easy mechanism, and better efficiency. By modifying some of the components of ORC the thermodynamic performance of this performance will be increased. Organic Rankine Cycles that include plate-type heat exchangers perform betters comparatively than shell and tube-type heat exchangers. In this recent times, the combined cycles based on ORC are more attractive to researchers. To overcome both the needs for power and cooling the system of Organic Rankine cycle - Absorption Refrigeration cycle (ORC-ARC) is proposed [29].

The CO_2 emission's main source is fossil fuels that are running on desalination processes. About 76 million tons of CO_2 emits every year due to desalination capacities and this amount of CO_2 emissions will be increased to 218 million tons per year in 2040. The CO_2 emissions per year in 2019 were 43.2 Giga tons which was 20% more than the year 2013. The value of CO2 emissions in 2013 was 36.1 Giga tons on annual basis. Using the coefficient of performance (two-thirds) goal to maintain the temperature of the environment below 2 °C will be exhausted in 2050. In 2040 the amount of CO2 emissions will be increased from 1500% to 2200% (expected). Likewise, the consumption of water and its withdrawal will also be increased to about 1000%. The amount of global water consumption for generating power was just 15% in 2010 which was produced by the process of desalination at the cost of 75 Tera Watt hour (TWh) and 76 million tons of CO_2 emissions. The current efficiency of 10–15% is not suitable for future supplies of water. Membrane materials are suggested for the future but it demands 5–10 years to produce this on a commercial basis. Moreover, the desalination technologies (thermally driven) could attain 20–25% of efficiency near to zone of production in just 1–2 years of experience. These proposed solutions would be helpful in the future to protect the environment and save energy as well.

The top priority of researchers is to find out a more optimized, energy efficient, and environment-friendly method for recovering the waste heat. There are different methods used to recover the waste heat but these methods are not economically sustainable and energy efficient. The indirect evaporative cooler is the most recently proposed system for waste heat recovery because it is environment friendly, low cost, low NOx production, energy efficient, and better efficiency. Numerous studies have been done in the literature on the cooling applications of the indirect evaporative cooler but no such type of work is done that is based on power generation using an indirect evaporative cooler. There is a research gap in that no real-time simulations are performed for power generation using an indirect evaporative cooler. The ongoing research deals with the utilization of industrial waste heat gases with indirect evaporative cooler by using industrial operating data to enhance the performance of the plant and increase work output and efficiency. There is a novelty of work that an indirect evaporative cooler is integrating using its real time operating data (real-time power simulations performed) for its power generation application using multi-fluids. No such type of work is done that is based on a component level to analyze the performance of the plant. Different working fluids will be used in bottoming cycle compressor to examine the behavior of the power plant. In this research, the results of both models will be compared to make sure which model produces high efficiency and enhances the plant performance. Furthermore, in this research, only boiler waste flue gases will be used in the Topping cycle instead of the whole Gas Turbine cycle and that will be further used with an indirect evaporative cooler to analyze the enhancement in performance and efficiency.

The development of parametric analyses of each component of power plant is performed to examine the effect of the power generation and efficiency on the power plant by recovering the waste heat. There is extensive research work performed on the cooling applications of an indirect evaporative cooler but still, there is a huge research gap to integrate indirect evaporative with multiple working fluids and real-time industrial simulations. In this research industrial waste heat gases of SP boiler operated with an indirect evaporative cooler. A model is developed in ASPEN HYSYS software and each component of the power plant is accomplished using experimental values to analyze the power generation and efficiency at the turbine. Moreover, for improving power plant performance bottoming cycle is re-integrated using a mixture of carbon dioxide and toluene (CO_2-C_7H_8) working fluid. Performance analysis of net power output and efficiency is compared for all the working fluids. A model is validated by resembling simulation results with the experimental data. Additionally, the model is optimized using real-time industrial data. Real-time industrial simulation presents improvement in power generation and efficiency value of power plant installed in the cement industry. Last but not the least, the techno-economic analysis of the entire waste heat recovery system is performed as well to find out the implementation and running cost of this power plant based on the M cycle.

The main aim of this research work is to examine the performance analysis of the waste heat recovery process by using different working fluids and indirect evaporative cooler. First of all, there is a model development of waste heat recovery system in ASPEN HYSYS software. After this, model integration using experimental values. Then, model execution using the different working fluid and comparison analysis of both fluids. After all this, a model is validated with the reference model. Model optimization is performed under real-time operating conditions and in the last Techno-economic analysis of the whole waste heat recovery system is performed as well.

2. Methodology

In the cement sector the waste heat gases are usually from a kiln or preheater. There are various methods preferred for recovering this waste heat and utilizing the Kalina cycle or Organic Rankine cycle to recover this waste heat to produce electricity is one of the methods. Every researcher presented this title of waste heat recovery [30]. In the current research, a waste heat recovery model is developed in the ASPEN HYSYS software. Different operating parameters are manipulated in this software to examine the plant performance (power generation and efficiency). Also, a model integrated is with other working fluid to inspect the behavior of the power plant. A techno economic cost is performed for this whole waste heat recovery system.

2.1. Component list

First of all, in developing a model there is an addition of components in the ASPEN HYSYS list that are involved in the whole simulation process. Table 1 presents a list of components that are used in developing a model of waste heat recovery.

2.2. Fluid package

In the process simulation tool (ASPEN HYSYS) it is necessary to select a fluid package. The Fluid package is used to calculate properties in process simulation. Here, Soave Redlich Kwong (SRK) is the selected thermodynamic model for all components in developing a model of waste heat recovery. SRK has the almost the same accuracy as Peng Robinson (PR) thermodynamic model. SRK model is used where there are ideal liquids and real gases. Fig. 1 displays a property package selection.

2.3. Waste heat recovery model

Fig. 2 presents the model of waste heat recovery. In the topping cycle, there is a Suspension preheater (SP) boiler installed in Bestway cement industry Farooqia and waste heat gases are emitted from that boiler. In bottoming cycle, there is a compressor that compresses the air and further, the splitter splits that compressed air. The upper stream of the splitter, (stream 5) takes all the temperature of waste heat gases from the boiler in a mixer 1. The bottom stream (stream 6) undergoes with humidification process in mixer 2 as water is supplied from the pump here. Moreover, the humid air (stream 9) and hot air (stream 10) are mixed in a mixer 3, and then stream 11 undergoes toward a turbine for power generation. 68.53 MW power generation extracted at turbine by using air as a working fluid in the compressor.

2.4. Mathematical equations

The mathematical equations for the fluid package Soave Redlich Kwong (SRK) are presented below,

$$P = \frac{RT}{V_m - b} - \frac{a\,\alpha}{V_m\,(V_m + b)} \tag{1}$$

- **P** is the gas pressure
- **R** is gas constant
- **T** is temperature

Table 1
Components list with their composition.

Components	Mole fraction
Air	0.0000
CO_2	0.8313
Nitrogen	0.1111
Oxygen	0.0327
H_2O	0.0100
Argon	0.0134
NO	0.0009
CO	0.0004
SO2	0.0002
NO2	0.0001

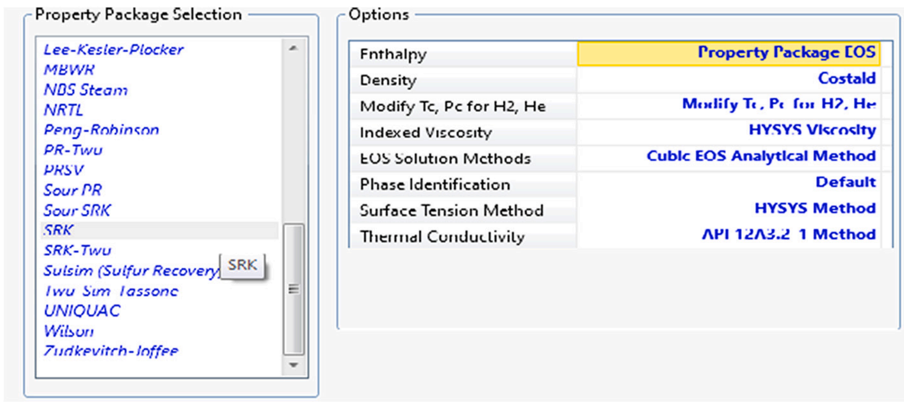

Fig. 1. Property package.

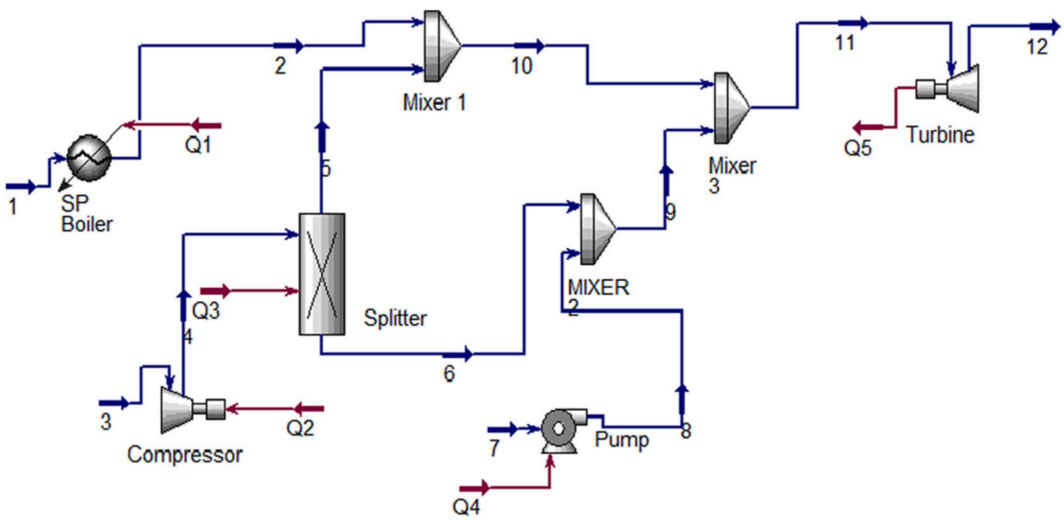

Fig. 2. Waste heat recovery model.

- $\mathbf{V_m}$ is the molar volume
- $\boldsymbol{\alpha}$ is the attractive term in original equation of Redlich-Kwong
- **a** is a constant that attracts for attractive potential of molecules
- **b** is a constant that corrects for volume

$$\alpha = \left(1 + \left(0.480 + 1.574\omega - 0.176\,\omega^2\right)\left(1 - \sqrt{T_r}\right)\right)^2 \tag{2}$$

- $\mathbf{T_r}$ is reduced temperature of compound
- $\boldsymbol{\omega}$ is acentric factor

$$a = \frac{1}{9\left(\sqrt{32} - 1\right)} \frac{R^2 T_C^2}{P_C} = 0.42748 \frac{R^2 T_C^2}{P_C} \tag{3}$$

$$b = \frac{\sqrt[3]{2} - 1)}{3} \frac{RT_c}{P_C} = 0.08664 \frac{RT_c}{P_C} \tag{4}$$

3. Schematic diagram of the system

Fig. 3 highlights the schematic diagram of a system. In this system, waste heat gases emit from a topping SP boiler at state 2. The air is used as a working fluid in the compressor at state 4 in bottoming cycle. The compressed air at point 5 releases its temperature and becomes cool when it reaches a state of 5´. The compressed air splits in two ways here. The long horizontal bar air saturators are used to inject water into the air (humidification) so that, the downward part of compressed air gets humidified at a state of 5´´. Water in the air saturator is coming from pump and it reaches to air saturator at a state of 5´´´. The upward part of compressed air takes all the temperature of waste heat emitting from a state 2 to 3 and this compressed air now becomes heated. This heated air and humid air both streams meet together at a state 5´´´´ and then this stream undergoes at turbine at state 6. The generator is installed next to a turbine which converts mechanical energy into electrical energy. The very low-temperature waste heat gases are emitted from a turbine at state 7.

4. Results validation

4.1. Validation of air saturator (mixer)

In this section, there is a validity of air saturator carried out. At ambient pressure conditions, the air saturator could cool air to its sub-wet bulb temperature which is visualized by most researchers both numerically and experimentally [38,39]. For benchmarking and validation purpose, the working ambient air temperature at the inlet of the air saturator changes from 5 °C to 45 °C and the temperature at the exit of the air saturator is noted. The obtained output results are validated by using the numerical results of Tariq [3] and experimental results of Riangvilaikul [39]. This current research is accomplished on two different relative humidity values 6.9 g/kg and 14.2 g/kg. The simulation results with a relative humidity value of 6.9 g/kg and working fluid air are compared with the bottoming two results of Riangvilaikul [39]

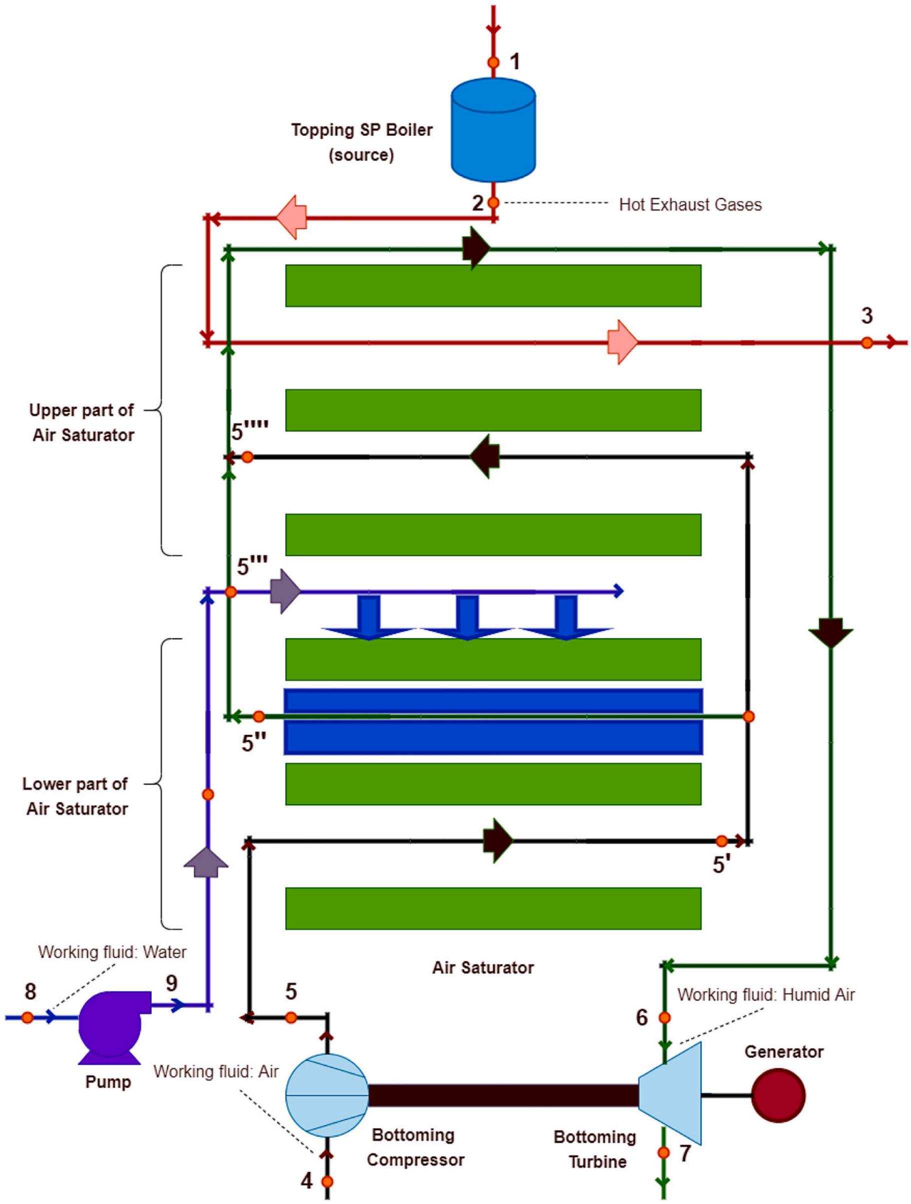

Fig. 3. MHABC working based on current research work.

and Tariq [3] at the same relative humidity values presented in Fig. 4. Further, this research study executed results at a relative humidity of 14.2 g/kg and a mixture of CO_2-C_7H_8 working fluid is compared with the results of Riangvilaikul [39] and Tariq [3] at a relative humidity value of 11.2 g/kg and a temperature difference of 2 °C (error 2 °C) is noted. Furthermore, this research study did not execute simulation results at a relative humidity of 26.4 g/kg however, Riangvilaikul [39] and Tariq [3] have experimental results at a relative humidity value of 26.4 g/kg presented in Fig. 4. Fig. 4 presents the best validation of the air saturator (mixer). The more detailed model verification is also presented in supplementary material heading 1.

4.2. Validation of overall heat and mass exchanger (HMX)

After the validation of the air saturator, there is a need for complete model validation. For comparison and matching purpose, the bottoming cycle flow of the air stream is divided into three streams so that's why the extraction ratio is 0.33. The amount of water injected for this current simulation is 1.4 kg/s. Overall efficiency obtained is 35.44% and 38.57% for this system. In this way, different sections of model validation and simulation strategy are achieved. Table 2 presents a detailed overview of validation.

5. Results and discussion

5.1. Boiler inlet parameters (State 1)

The nominal conditions at the inlet of boiler are: $T_1 = 1273\ ^{\circ C}$, $P_1 = 11.36$ bar, $\dot{m}_1 = 96.98$ kg/s. Parametric analyses are to be performed to ensure thermodynamic arrangements and thermo-economic analysis for a more optimal system. For sustainable recovery of waste heat, the parametric analysis should be performed on a priority basis. In Fig. 5 (a) the result presents that the increase in boiler inlet temperature increases the net power generation. This is because by relation,

$$Q = m^*Cp^*\Delta T \tag{5}$$

Heat content (Q) is in direct relation to boiler inlet temperature so, by increasing the boiler inlet temperature heat content (Q) increases and

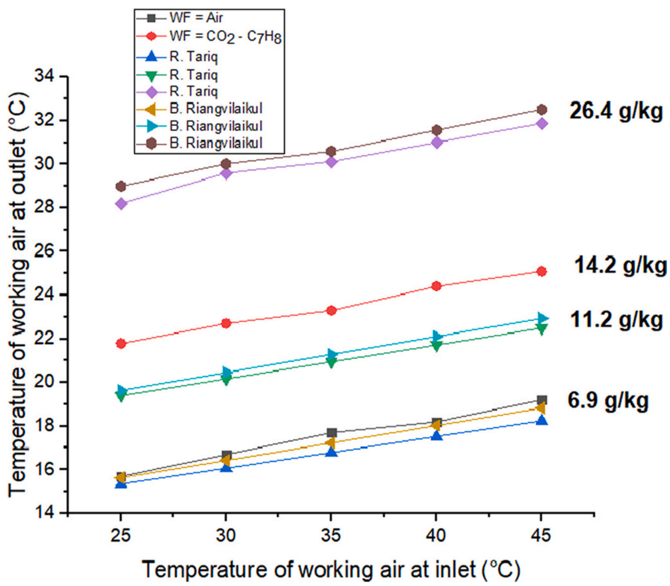

Fig. 4. Validation of air saturator (mixer). A comparative analysis of current research results with the numerical results of Tariq [3] and experimental results of Riangvilaikul [39].

Table 2
Results validation. Comparison of current numerical results with the results of R. Tariq [3].

Sr #	Parameters	Saghafifar et al	R. Tariq [3]	Current research	
				WF = Air	WF = CO_2-C_7H_8
01	Extraction Ratio	0.33	0.33	0.33	0.33
02	Amount of water injected	–	1.4 kg/s	1.4 kg/s	1.4 kg/s
03	Turbine Inlet Temperature	1400 K	1400 K	1400 K	1400 K
04	Bottoming Cycle Pressure Ratio	4	4	4	4
05	Mass Flow Rate Ratio	1	1	1	1
06	Efficiency	33.5%	32.6%	35.44%	38.57%
07	Power Generation	50 MW	58 MW	68.53 MW	48.59 MW

that is why the temperature at the inlet of the turbine also increases. An increase in hot gas exhaust temperature of the topping cycle or an increase in temperature at the inlet of the turbine increases net power generation [31]. Basically by increasing the temperature across the boiler the enthalpy drop across the turbine increases which results in increase in power generation. The output result shows that by an increase in temperature from 1000 $^{\circ C}$ to 1500 $^{\circ C}$ net power output increases by 51 MW. The presented results are executed at a constant pressure of 11.36 bar and a constant mass flow rate of 96.98 kg/s.

In Fig. 5 (b) there is a pressure relation with net power. This graph presents that pressure is not a significant variable so, by increasing pressure, there is no such increase in net power generation. By increasing the pressure from 20 bar to 300 bar the net power output increases by just 0.57 MW. Some results in the literature also shows the same trend of a minor increase in power generation by increasing pressure [32]. The results are generated at a constant temperature of 1273 $^{\circ C}$ and a constant mass flow rate of 96.98 kg/s. This also highlights that a system that has low source pressure is fine enough to run that system.

In Fig. 5 (c) there is a trend of mass flow rate with net power generation. An increase in mass flow rate increases net power generation. This is because by increasing mass flow rate heat content (Q) increases so, it can recover more waste heat and increase net power output at the turbine. The increase in results of power generation is noted at about 37 MW at the turbine [31]. The results of mass flow rate with net power are generated at a constant temperature of 1273 $^{\circ C}$ and constant pressure of 11.36 bar.

5.2. *Compressor inlet parameters (State 3)*

The nominal conditions at the inlet of compressor are $T_3 = 25.05\ ^{\circ C}$, $P_3 = 1.013$ bar, $\dot{m}_3 = 73.47$ kg/s. Fig. 6 (a) presents that by increasing compressor inlet ambient temperature of air net power output decreases. The reason at the back end of this is that increasing ambient temperature results in decreases in mass flow rate through the turbine which ultimately decreases the net power output. Turbine output power is in direct proportion with the mass flow rate. The selected temperature range of ambient temperature is from 5 $^{\circ C}$ to 40 $^{\circ C}$. This is because the air ambient temperature is almost 5 $^{\circ C}$ in winter and 40 $^{\circ C}$ in summer in this Taxila (Khanpur) region [2017 ASHRAE Handbook-Fundamentals (SI)] so, that's why the trend is noted in between this temperature range. Fig. 1 in supplementary material shows ASHRAE handbook fundamentals for this Khanpur region. This current analysis shows that the ambient air temperature decreases power generation up to 8.85 MW [33]. These results of compressor air ambient temperature with the net power are generated at a pressure of 1.013 bar and a mass flow rate value of 73.47 kg/s.

From Fig. 6 (b) it is seen that increasing pressure ratio net power output increases. At lower values of pressure ratio from 0 to 10 net power increases significantly. This is because compressor work is less at that phase up to 10 pressure ratio so, the net power increases. At the highest pressure ratio after 10, the compressor work increases so that net output power is not increasing significantly and remains constant to some extent. Pressure ratio 10 is optimal for the highest work output as well as by considering its thermo-economic analysis [34]. 26 MW net power increased by increasing the compression ratio.

Fig. 6 (c) represents a relationship of mass flow rate with the net power. Net power increases with increasing mass flow rate. The reason behind this equation,

$$P = \Delta h_{actual} * \dot{m} \tag{6}$$

By increasing the mass flow rate enthalpy change increases so it means heat content (Q) increases and that's why net power generation increases [35]. Enthalpy change increases by increasing mass flow rate so, it means the heat content value increases at the turbine, which is the reason of an increase in power generation. Here, mass flow rate enhancement increases power up to 24.6 MW. The reason for selecting these mass flow rate ranges is that most of the compressor inlet mass flow rates are in this range in the case of waste heat recovery.

5.3. *Splitter top outlet (State 5)*

The nominal conditions at the top outlet of splitter (state 5) are:

$$T_5 = 256.9^{\circ C}, P_5 = 2.301\ \text{bar}, \dot{m}_5 = 51.41\ \text{kg/s}$$

There is a need of observing the splitter parameters. The top of the splitter goes for recovering waste heat so, it's important to find out its temperature and pressure. A good high temperature by considering power generation as well as economic analysis is beneficial for a better system of waste heat recovery.

Fig. 7 (a) presents a relationship of the splitter top outlet (state 5) temperature with the net power. The reason behind this is that by increasing compressor temperature the splitter outlet temperature increases. Increasing splitter temperature increases heat content (Q) so that its evaporation rate increases and it can recover more and more heat and so that's why net output power at the turbine increases. The trend is noted in between temperature range of 100 $^{\circ C}$ to 1000 $^{\circ C}$ because most of

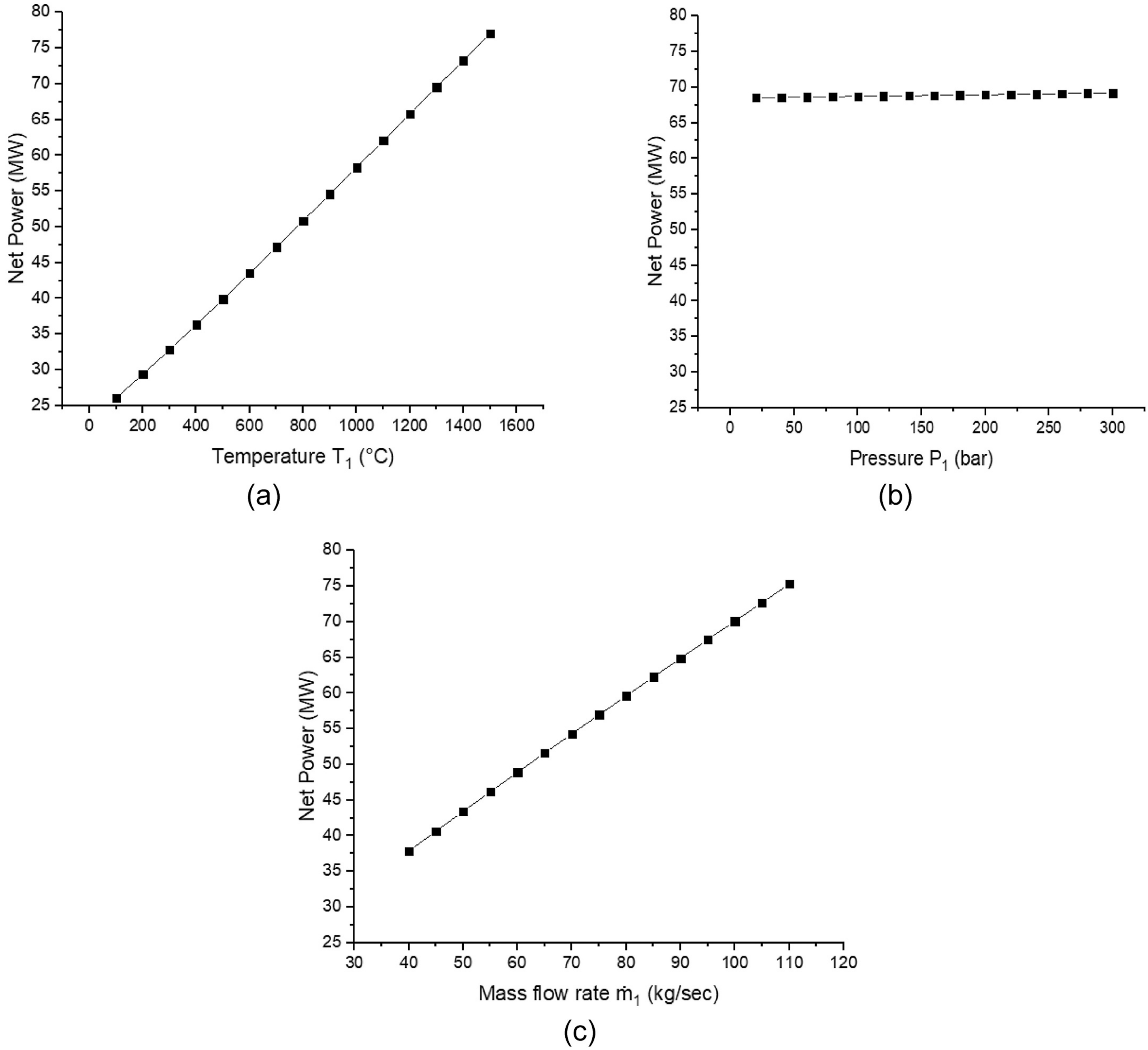

Fig. 5. Parametric assessment to note the influence of (a) boiler temperature, (b) boiler pressure and (c) boiler mass flow rate on the net power.

the waste heat from cement plant is in the range of 200 $^{\circ C}$ to 1000 $^{\circ C}$. The results present a power increase of 14.5 MW in a system.

Fig. 7 (b) presents a relationship of pressure with the output net power. By increasing pressure, there is no such increase in power generation and the increase in power generation is very small. The reason is that this outlet pressure of the splitter (state 5) increases by increasing the compression ratio so, by increasing more and more compression ratio there is no significant increase in power generation and it almost remains the same. That's why by increasing this pressure net power does not increases and ultimately it remains the same. The selected range for assessment is between 50 bar to 500 bar because the majority of pressure lies in this range. The increase in power generation is noted as only 0.16 MW.

From all these above discussions it is suggested to use a temperature of 500 $^{\circ C}$ and a pressure of 50 bar for the best net power output. These temperature and pressure values are critical and technically economical for a system of waste heat recovery.

5.4. Splitter bottom outlet (State 6)

The nominal conditions at the bottom outlet of the splitter (state 6) are:

$$T_6 = 256.9^{\circ C}, P_6 = 2.301 \text{ bar}, \dot{m}_6 = 22.04 \text{ kg}/\text{s}$$

A Parametric assessment of temperature and pressure at the splitter outlet is necessary. This parametric analysis makes sure the thermodynamics arrangements and thermo-economic analysis for a more optimal system. It is important to find out the effect of this air that undergoes for humidification process for sustainable waste heat recovery. The nominal conditions are available experimental values [3] from the literature.

It is observed from Fig. 8 (a) that the temperature range selected for analysis is 100 $^{\circ C}$ to 1000 $^{\circ C}$. The increase in power generation is noted at the turbine by increasing the temperature. The reason behind this is that increasing splitter outlet temperature increases the temperature at the turbine so, that turbine extracts more and more work of waste heat

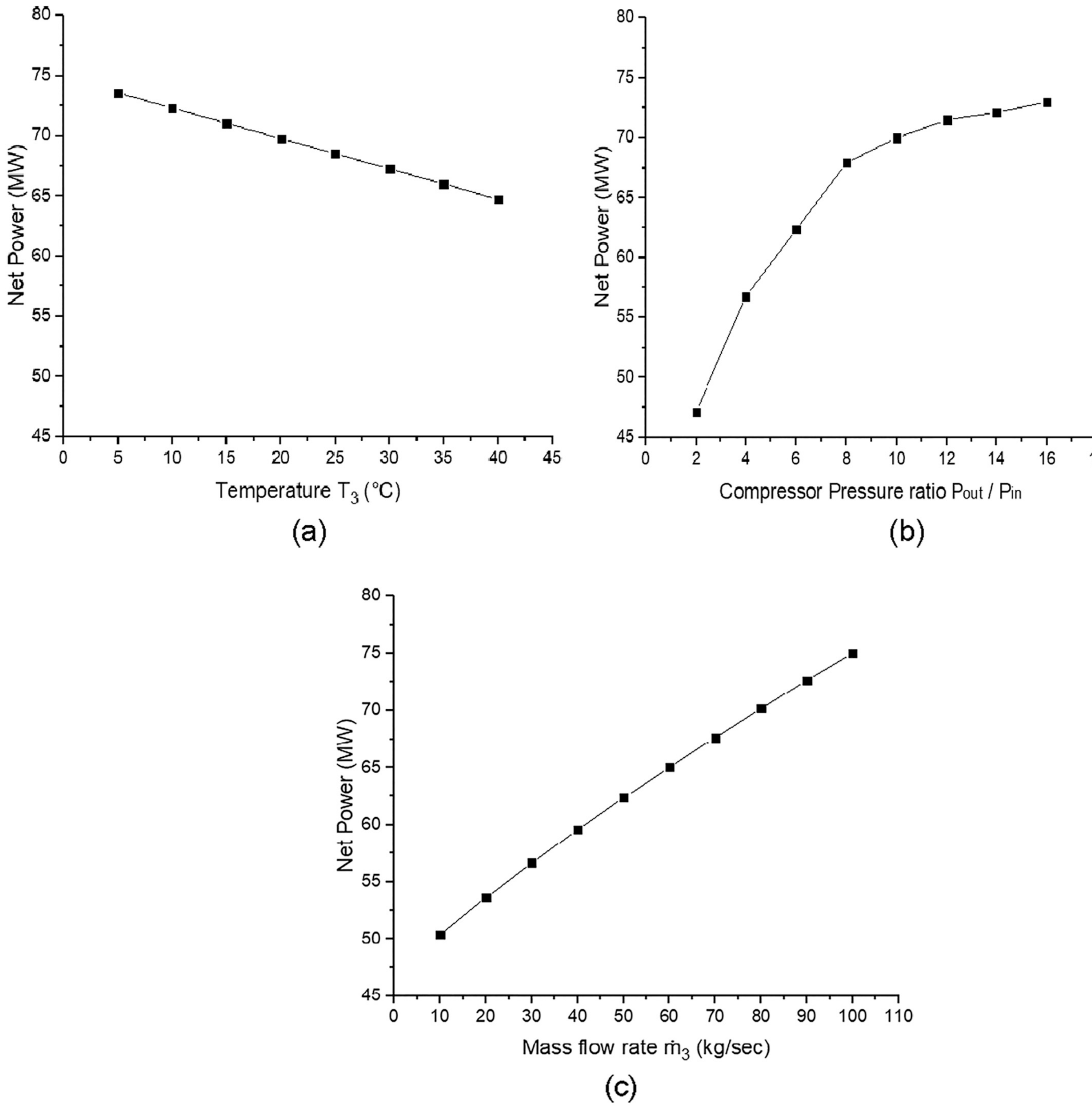

Fig. 6. Parametric assessment to note the influence of (a) compressor temperature, (b) compressor pressure ratio and (c) compressor mass flow rate on the net power.

recovery, and in return net work output at the turbine increases. In this, the increase in temperature from 100 $^{\circ C}$ to 1000 $^{\circ C}$ increases power generation by 6.25 MW. Here the temperature of 500 $^{\circ C}$ is optimal for this system. It would be beneficial in terms of cost as well as in terms of sustainability.

Fig. 8 (b) represents that pressure is the least significant variable and it has no such high impact on net output work. In this stage power at turbine output increases only 0.6 MW with increasing pressure. It is great enough to use low pressure of up to 50 bar for the sustainability of this system.

Here, in this stream, the recommended values of temperature and pressure are 500 $^{\circ C}$ and 50 bar respectively.

5.5. *Pump inlet parameters (State 7)*

The nominal conditions at the inlet of pump are $T_7 = 25.05\ ^{\circ C}$, $P_7 =$ 9.81 bar, $\dot{m}_7 = 1.4$ kg/s. It is observed from Fig. 9 (a) that increasing water temperature at the inlet of the pump increases net power. The purpose behind this increasing net power is that by increasing water temperature the rate of humidification increases and the space between water molecules increases. This increase in space between water molecule increases surface area (stronger intermolecular forces) so that the evaporation rate increases and in return net power output increases. The selected water temperature range is from 5 $^{\circ C}$ to 40 $^{\circ C}$ because water temperature for cooling or humidification purposes lies in between this range [36]. Higher temperature changes its phase. The increase in net

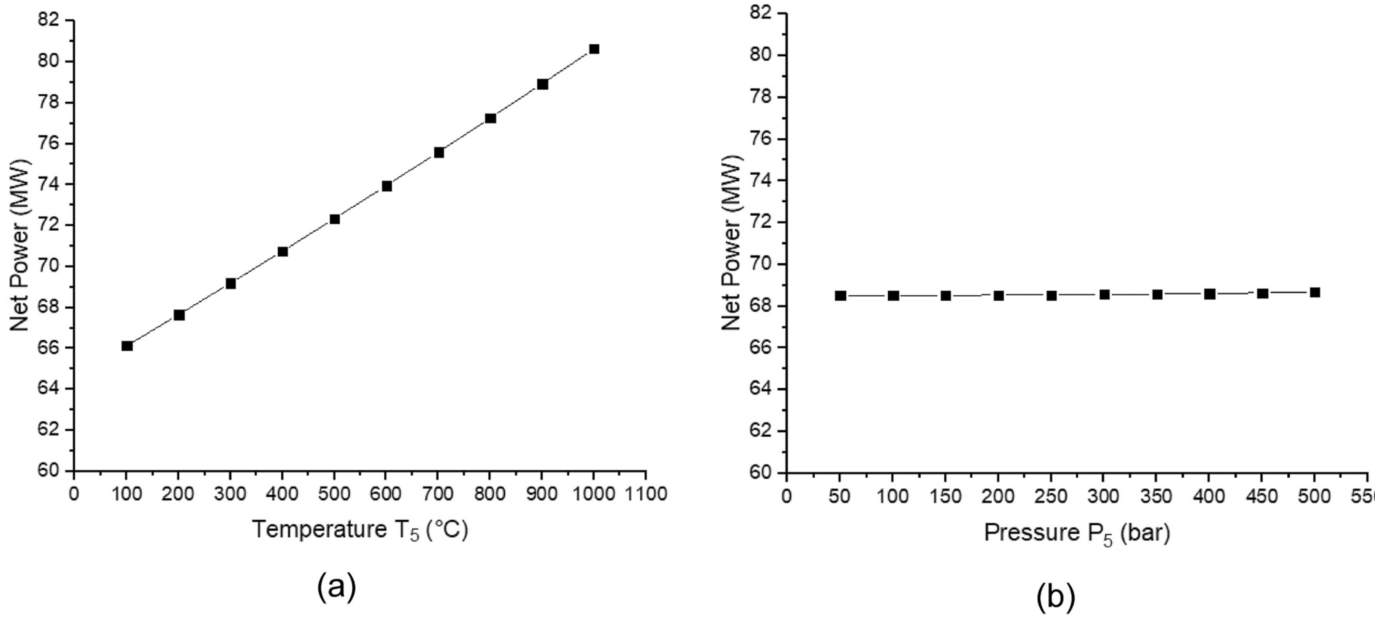

Fig. 7. Parametric assessment to note the influence of (a) splitter top outlet (state 5) temperature and (b) splitter top outlet (state 5) pressure on the net power.

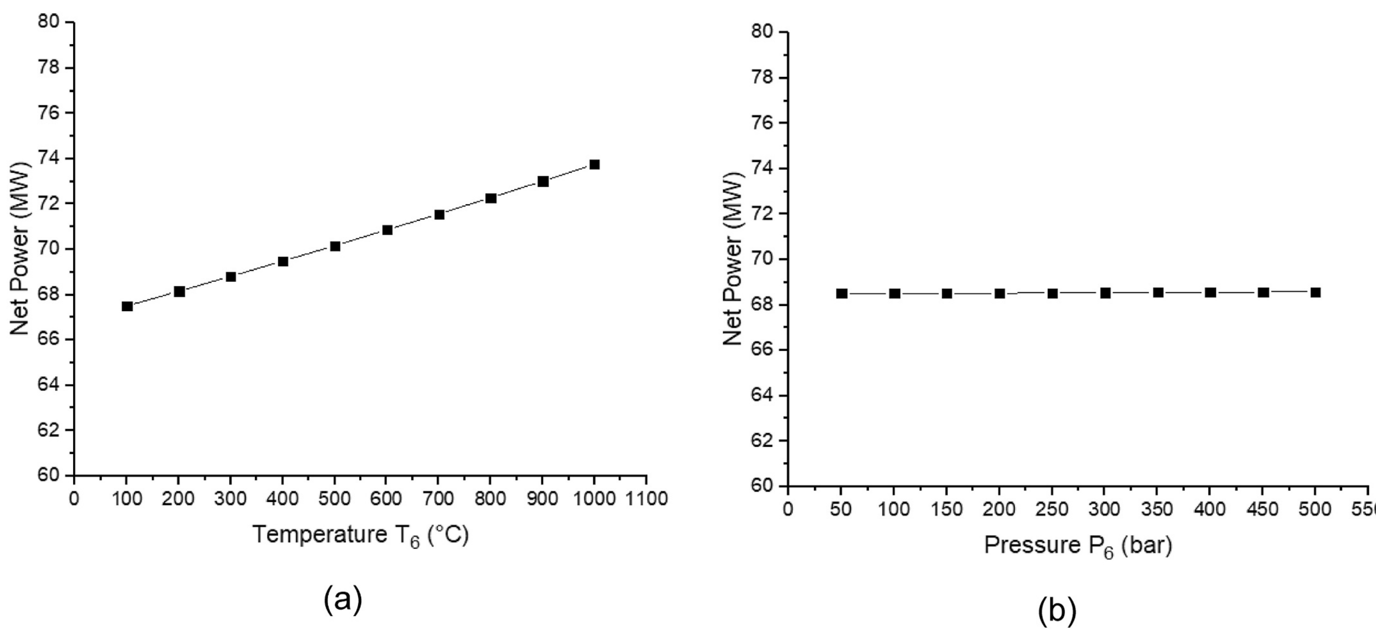

Fig. 8. Parametric assessment to note the influence of (a) splitter bottom outlet (state 6) temperature and (b) splitter bottom outlet (state 6) pressure on the net power.

power is observed to be 0.7 MW.

From Fig. 9 (b) it is observed that pressure is the least significant variable here and it has no major impact on net power generation. The reason at the back end of this is that the enthalpy change increases by increasing pressure so work output increases to some extent. Here in this case the work output increases only 0.1 MW. The reason is that the highest compression ratio also does not increases as such enthalpy change at the turbine so it could not extract most of the heat and that's why net power output decreases.

It is observed that from Fig. 9 (c) mass flow rate decreases net power output. This is because by increasing too much flow rate there is an excess of water in a mixing chamber. This excess of water minimizes the temperature of working air in a mixing chamber and cools down its temperature to some extent. This results in a decrease in temperature at the inlet of the turbine so, that is why the net power generation decreases in this case. Increased mass flow rate decreases power generation up to 25 MW. The assessment of parameter mass flow rate was between 5 and 50 kg/s. The reason was that at these temperatures, the pressure ranges of water most of the flow rate lies in this range.

It is recommended to use nominal conditions at the inlet of the turbine. The temperature of the water should be 25 $^{\circ C}$. Pressure should be 10 bar and the mass flow rate recommended value is 1.5 kg/s.

5.6. *Turbine parameter (change in pressure, State 11)*

The available conditions at the inlet of the turbine are:

$T_{11} = 898.9^{\circ C}, P_{11} = 2.301 \text{ bar}, \dot{m}_{11} = 171.86 \text{ kg/s}$

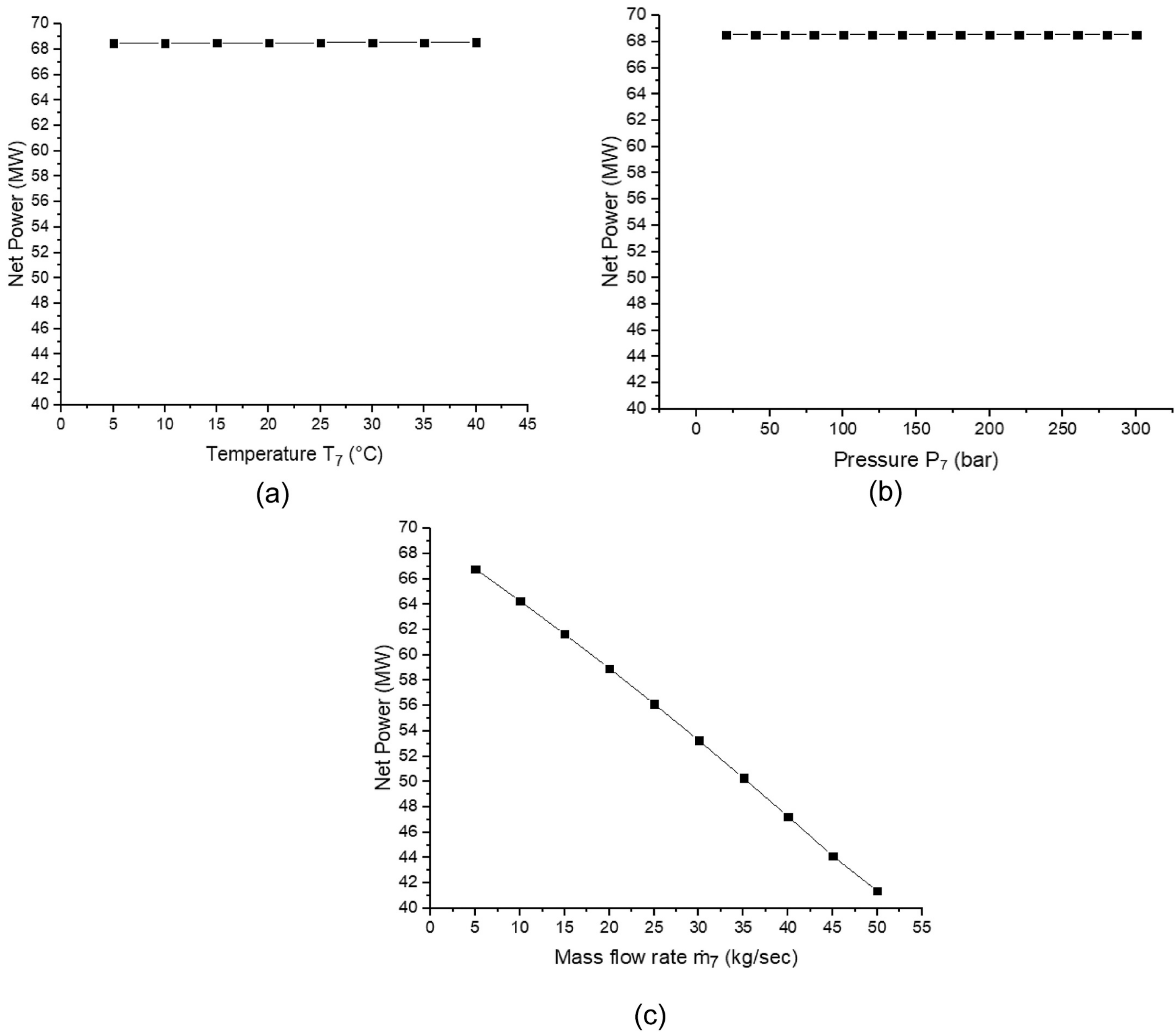

Fig. 9. Parametric assessment to note the influence of (a) pump temperature, (b) pump pressure and (c) pump mass flow rate on the net power.

The relationship presented in below Fig. 10 highlights the change in pressure at the turbine with the net power. Change in pressure at turbine is the pressure at inlet and outlet of turbine and by increasing this change net power output increases. This is because by relation,

$$P = \Delta h_{actual} * \dot{m}$$

Pressure change at the turbine increases the enthalpy drop at the turbine. An increase in enthalpy drop increases the heat content (Q) so that more and more waste heat is recovered and that's why net power output at the turbine increases. From this change, there is a significant increase in power generation by 65 MW. Enthalpy drop increases by varying pressure change this means that the difference of temperature at the inlet of turbine and the outlet of the turbine increases so, the turbine extracts more and more work and so in-return power generation value increases.

5.7. The mass flow rate ratio of topping and bottoming cycle (State 1, State 3 ratio)

The design conditions of mass flow rate are,

- Topping cycle mass flow rate $\dot{m} = 96.98$ kg/s
- Bottoming cycle mass flow rate $\dot{m} = 73.47$ kg/s

The relationship of this mass flow ratio of the topping and bottoming cycle in Fig. 11 displays that by increasing the mass flow rate ratio net power increases by 43 MW. The reason at the back end of this is increased heat content value. This relation is presented by the following equation,

$$Q = m * Cp * \Delta T$$

By increasing the mass flow rate the temperature drop across the turbine increases with the constant pressure ratio. The increase in mass flow rate value at the turbine increases the power generation. The higher value of flow rates gives more work. This discussion is also supported by

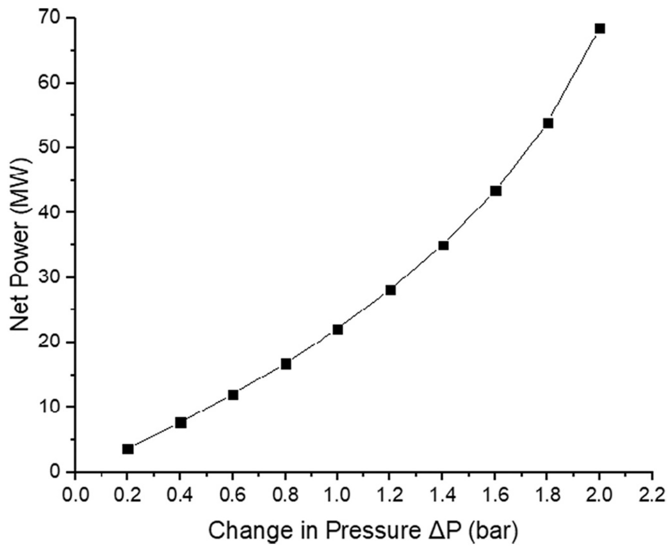

Fig. 10. Parametric assessment to note the influence of turbine change in pressure on the net power.

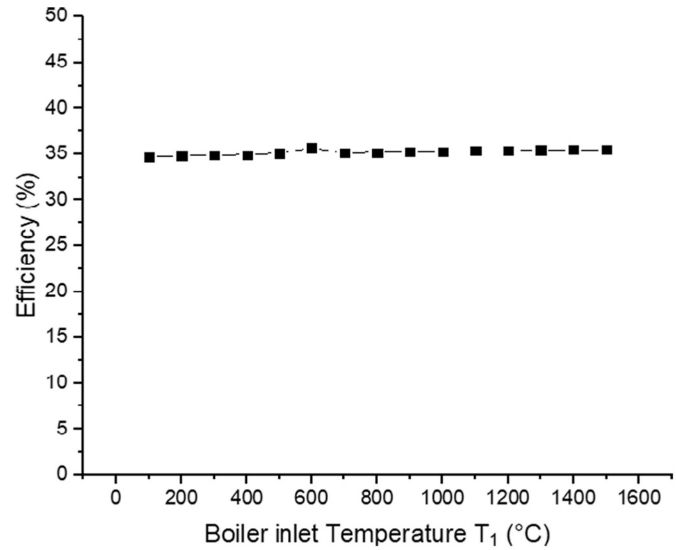

Fig. 12. Parametric assessment to note the influence of boiler inlet temperature on the net power.

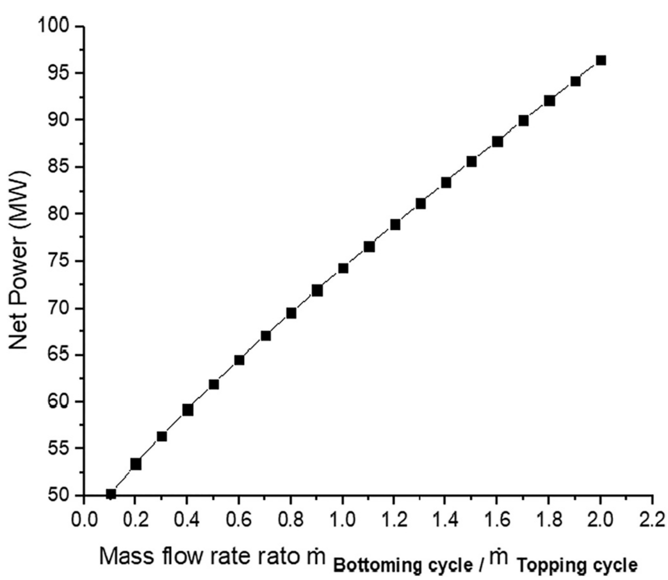

Fig. 11. Parametric assessment to note the influence of mass flow rate ratio on the net power.

following the equation presented below,

$$P = \frac{Q}{t} \tag{7}$$

5.8. Boiler inlet temperature T_1 versus efficiency

The relationship of boiler inlet temperature with the efficiency presented in Fig. 12 represents that with increasing boiler source temperature the efficiency does not increase and it ultimately remains the same. The reason is that by increasing boiler source temperature the temperature at the turbine inlet increases and net power output at the turbine not increases significantly to a large extent. From this, it could say by increasing boiler inlet temperature net power output increases but it may not increases significantly according to its input at the turbine so that is why efficiency almost remains the same and does not increase with increasing source temperature. The efficiency of this system could be improved by increasing power output by following this relation,

$$\text{Efficiency} = \eta = \frac{W_{net}}{Q_{in}} \tag{8}$$

The efficiency of this waste heat system could also be improved by preheating the temperature, using different working fluids, and varying temperature of cooling water [37]. Here, the increased in efficiency is observed only 0.8%. The efficiency is also improved by increasing the net output work at the turbine by using less heat content value.

5.9. Waste heat recovery model execution with a binary mixture of CO_2-C_7H_8

A binary mixture (CO_2-C_7 H_8) of Carbon dioxide CO_2 and toluene C_7 H_8 is used as a different working fluid in a model of waste heat recovery. Fig. 2 in a supplementary material shows using different working fluid net power output decreases and the net power produced is 48.59 MW. In Literature this binary mixture (CO_2-C_7 H_8) is also used as a working fluid. All the simulation results generated from model execution using working fluid CO_2-C_7H_8 are presented in the supplementary material from heading 2 to heading 9.

5.10. Boiler inlet temperature versus net power output and efficiency with HMX for both working fluid

Fig. 13 exhibits the combination of results (power generation and efficiency) for both the working fluid by using a heat and mass exchanger (HMX). These results present that air power generation increases from 26.12 MW to 77.01 MW with an increase in efficiency value from 34.71% to 35.51%. For CO_2-C_7H_8 these results highlight that power generation increases from 21.18 MW to 53.75 MW with efficiency increase from 37.2% to 38.65%. This presented Fig. 13 is best for comparing power generation and efficiency for both of the working fluids.

5.11. Boiler inlet temperature versus net power output and efficiency without HMX for both working fluid

Fig. 14 displays a combined result of both working fluid for power generation and efficiency without using HMX. For air, the value of power generation increases from 1.4 MW to 5.87 MW with an increase in efficiency from 4.28% to 12.55% for both the working fluid.

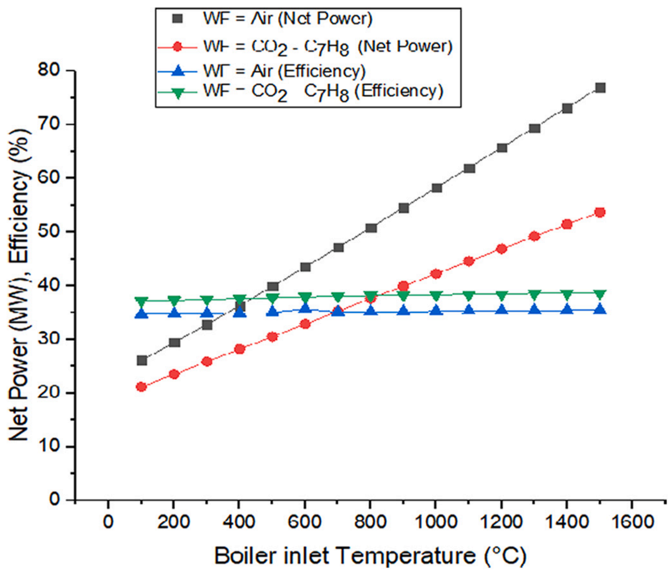

Fig. 13. Parametric assessment to note the influence of boiler inlet temperature on the net power output and efficiency

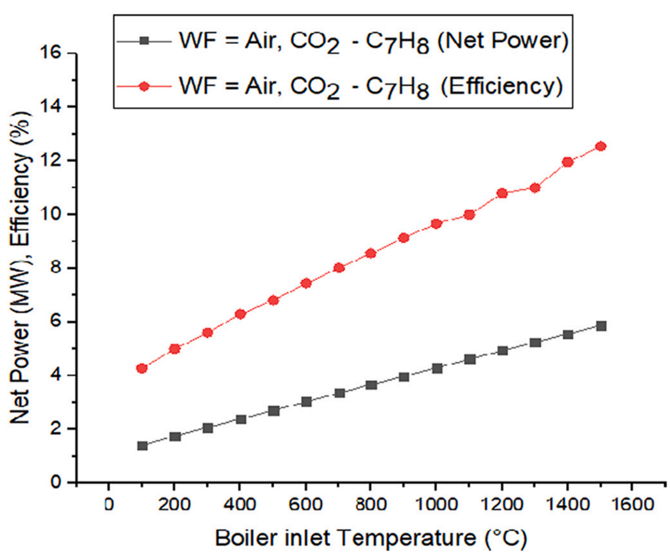

Fig. 14. Parametric assessment to note the influence of boiler inlet temperature on the net power output and efficiency.

5.12. Comparison analysis of both working fluids

Fig. 15 illustrates the comparison analysis of both the working fluid. The results of net power output and efficiency are presented for both working fluid with and without HMX. It is highlighted in the presented figure the increase in power generation is high with HMX like in the case of air this increase is from 26.12 MW to 77.01 MW and in the case of CO_2-C_7H_8 power generation increase is 21.18 MW to 53.75 MW. The increase in efficiency value for air and CO_2-C_7H_8 is just from 34.71% to 35.51% and 37.2% to 38.65 which is not a big increase. Without HMX increase in power generation value is small just from 1.4 MW to 5.87 MW but here there is a significant increase in efficiency value from 4.28% to 12.55% for both working fluids. It is concluded that HMX is best for increasing power generation and without HMX the efficiency of the system is increased.

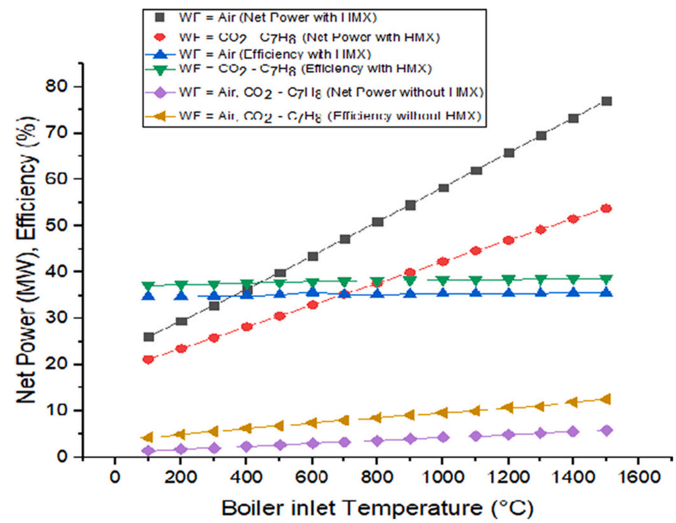

Fig. 15. Parametric assessment to note the influence of boiler inlet temperature on the net power output and efficiency with and without HMX for both working fluid.

5.13. Comparative analysis of both working fluids with respect to net power output produced

Table 3 presents a comparative analysis of both the working fluids by observing a temperature range, pressure range, and mass flow rate range. The net power produced from these ranges across each component is compared for both the working fluids. This Table is beneficial for finding out the best parameter range for maximum power generation.

The below-presented Table 4 manifests the mass flow rate ratio of the topping and bottoming cycle with the net power produced and boiler inlet temperature with the efficiency for working fluid air and CO_2-C_7H_8. The ratio of the topping and bottoming cycle is calculated in this way,

$$\text{Ratio} = \dot{m}_{\text{bottoming cycle}} / \dot{m}_{\text{topping cycle}} \tag{9}$$

(See Tables 5–7.)

This ratio comparison for both working fluid clearly describes that air is the best working fluid for power generation. Boiler inlet temperature with efficiency is mentioned in this table and compared for both fluids.

5.14. Industrial operating parameters

As the purpose of this research is carried out to recover the waste heat gases from Bestway cement Industry so operating parameters of the Bestway cement industry are used for analysis.

$T = 320\ ^{\circ C}$, $P = 2.85$ bar, $\dot{m} = 429{,}642$ kg/h $= 119.34$ kg/s.

These operating parameters of the industry present a power generation of 38.04 MW with an efficiency of 27.78% by using air as a working fluid in bottoming cycle presented in Fig. 11 in the supplementary material. Fig. 12 in the Supplementary material presents power generation at the turbine with efficiency values at the turbine.

The operating parameters of the industry present a power generation of 30.63 MW with efficiency of 27.77% by using CO_2-C_7H_8 as a working fluid in bottoming cycle. Fig. 13 in the supplementary material displays a complete model integrated with CO_2-C_7H_8. Fig. 14 in the supplementary material highlights the value of power generation at turbine along with isentropic efficiency and polytrophic efficiency.

6. Techno-economic analysis (TEA)

The techno-economic analysis is executed based upon the Levelized cost of energy (LCOE), life cycle cost, and payback period. The whole life

Table 3
Comparison analysis of both working fluids using the effect of (a) temperature range, (b) pressure range, and (c) mass flow rate on the net power production.

(a)

Component	Temperature range (°C)	Net power produced (MW)	
		Working fluid Air	Working fluid CO_2-C_7H_8
Boiler Inlet (1)	100–1500	26.12–77.01	21.18–53.75
Compressor Inlet (3)	5–40	73.58–64.73	54.2–44.9
Splitter Top Outlet (7)	100–1000	67.66–80.64	48.49–49.14
Splitter Bottom Outlet (8)	100–1000	67.51–73.76	45.65–65.83
Pump (13)	5–40	68.49–68.56	48.57–48.61
Turbine	–	–	–

(b)

Component	Pressure range (bar)	Net power produced (MW)	
		Working fluid Air	Working fluid CO_2-C_7H_8
Boiler Inlet (1)	20–300	68.55–69.17	48.63–48.98
Compressor Inlet (3)	2–16 (Pressure Ratio)	47.1–73	23.69–47.51
Splitter Top Outlet (7)	50–500	68.52–68.68	48.58–48.53
Splitter Bottom Outlet (8)	50–500	68.53–68.59	48.12–46.38
Pump (13)	20–300	68.53–68.54	48.59–48.6
Turbine	0.2–2 (Change in Pressure)	3.69–68.53	2.42–48.59

(c)

Component	Mass flow rate (kg/s) range		Net power produced (MW)	
	Working fluid Air	Working fluid CO_2-C_7H_8	Working fluid Air	Working fluid CO_2-C_7H_8
Boiler Inlet (1)	40–110	40–110	26.12–77.01	21.18–53.75
Compressor Inlet (3)	10–100	10–100	73.58–64.73	54.2–44.9
Splitter Top Outlet (7)	51.41 (Calculated)	3.57 (Calculated)	67.66–80.64	48.49–49.14
Splitter Bottom Outlet (8)	22.04 (Calculated)	69.88 (Calculated)	67.51–73.76	45.65–65.83
Pump (13)	5–50	5–50	68.49–68.56	48.57–48.61
Turbine	–	–	–	–

Table 4
Comparison analysis of both working fluids on the net power production using (a) mass flow ratio of topping and bottoming cycle and efficiency analysis of both fluids using (b) boiler inlet temperature T1.

(a)

The mass flow rate ratio of topping and bottoming cycle	Net power produced (MW)	
	Working fluid air	Working fluid CO_2-C_7H_8
0.2–2	53.44–96.45	46.32–59.91

(b)

Boiler inlet temperature T_1 (°C)	Efficiency %	
	Working fluid air	Working fluid CO_2-C_7H_8
100–1500	34.71–35.51	37.2–38.65

Table 5
Comparison Analysis of both working fluid with and without Heat and mass exchanger (HMX) to influence the effect on net power output and efficiency.

Sr #	Working fluid	Boiler inlet temperature (°C)	Net power output (MW)	Efficiency (%)
With HMX				
01	Air	100–1500 °C	26.12–77.01	34.71–35.51
02	CO_2-C_7H_8	100–1500 °C	21.18–53.75	37.2–38.65
Without HMX				
03	Air	100–1500 °C	1.4–5.87	4.28–12.55
04	CO_2-C_7H_8	100–1500 °C	1.4–5.87	4.28–12.55

Table 6
Initial Parameters for the simulation of waste heat recovery model.

Sr #		Parameters	Values
01	Topping cycle	Topping Cycle Mass Flow Rate	96.98 kg/s
02		Inlet Temperature of Air (T1)	1546 K
03		Inlet Pressure of Air (P1)	11.36 kPa
04		Components	CO_2, N_2, O_2, H_2O, Ar, NO, CO, SO_2, NO_2
05	Bottoming Cycle	Bottoming Cycle Pressure Ratio (BCPR)	16
06		Inlet Temperature of Air (P)	298.2 K
07		Inlet Pressure of Air	101.3 kPa
08		Isentropic Efficiency of Compressor	85
09		Mechanical Efficiency of Compressor	88.94
10		Isentropic Efficiency of Turbine	85
11		Mechanical Efficiency of Turbine	81.97

Table 7
Details of the industrial waste heat recovery power plant.

Sr #	Item	Value
01	Electricity generation from Bestway cement power plant in 24 h	7700 kW
02	Electricity generation in 1 h	320.83 kW
03	Waste heat recovery system designed	30 years
04	Running time of power plant per year	5000 h
05	Electricity generation in 30 years[1]	48,124,500 kWh
06	Electricity generation in 1 year	1,604,150 kWh
07	Capital recovery factor[2]	0.11

Average interest rate of Pakistan from last 30 years (1992 to 2022) is 10.99% [42] = 11%.
LCOE = Total Annual Cost (TAC) / Electricity generation.
PWCC = Initial cost + O & M cost + Replacement cost – Salvage value [41].
Initial cost = Land allocation cost + Equipment cost + Installation cost + Transport cost + overhead cost [41].
[1] Electricity generation = Electricity generation/h * no of years * Running time per year.
[2] CRF = $i_{interest}\,(1 + i_{interest})^{number\ of\ years} / (1 + i_{interest})^{number\ of\ years} - 1$ [41].

cycle cost of a waste heat recovery system is estimated by balancing costs over the entire life period. It includes initial cost, operational and maintenance cost, and salvage value [40].

6.1. Initial cost

6.1.1. Land allocation cost

See Table 8.

Table 8
Details of land allocation cost.

Land allocation cost	
Equipment	**Area**
Steam turbine generator	31,300 mm * 35,000 mm = 31.3 m * 35 m = 1095.5 m^2
AQC boiler	9510 mm * 16,120 mm = 9.51 m * 16.12 m = 153.14 m^2
SP boiler	14,000 mm * 14,000 mm = 14 m * 14 m = 196 m^2
Circulating water tank and pump house	24,050 mm * 21,500 mm = 24.05 m * 21.5 = 517.075 m^2
Water treatment and pump station	32,800 mm * 26,900 mm = 32.8 m * 26.9 m = 882.32 m^2

Total Area = 1095.5 + 153.14 + 196 + 517.075 + 882.32 = 2844.035 m^2 = 30,612.93 ft^2 = 112.44 marla.
Price of Land = 2.3 Crore = Rs 23,000,000 for 8 Kanal [43].
1 Kanal Price = PKR 2,875,000 so, 1 Marla Price = PKR 143,750.
Price of 112.44 Marla Land = 143,720 * 112.44 = PKR 16163250 = 87,040 USD.
Land Allocation Cost = $ 87,040.

6.1.2. *Equipment cost*

See Table 9.

6.1.3. *Installation, transport and overhead cost*

See Tables 10 and 11.

6.2. *Operation and Maintenance (O&M) cost*

See Table 12.

6.3. *Replacement cost*

See Table 13.

6.4. *Salvage value*

See Table 14.

6.5. *Present worth of capital cost*

See Table 15.

Table 9
Details of equipment cost.

Equipment cost	
Item	**Cost**
Boiler	$ 80,000 [44]
Compressor	$ 5500 [45]
Splitter	$ 21 [46]
Mixer	$ 1885 [47]
Pump	$ 4700*
Turbine	$ 91,151**
Generator	$ 50,000 [48]
DCS and Instrumentation	$ 38,959**
Water treatment cooling tower	$ 9430**
Others (Ducts, Pipes)	$ 2000**

Total equipment cost = $ 80,000 + $ 5500 + $ 21 + $ 1,885 + $ 4700 + $ 91,151 + $ 50,000 + $ 38,959 + $ 9430 + $ 2000.
Total equipment cost = $ 283,646.

* Value calculated from the software. Fig. 15 is representing the cost of the pump in the supplementary material.
** Values obtained from the industry.

Table 10
Details of installation, transport and overhead cost.

Sr #	Item	Cost
01	Installation cost[1]	$ 14,182.3
02	Transport cost[1]	$ 2836.46
03	Overhead cost[1]	$ 14,182.3

[1] Quoted from the industrial manufacturers and by considering the literature.

Table 11
Details of initial cost.

Initial cost	
Land allocation cost	$ 87,040
Equipment cost	$ 283,646
Installation cost	$ 14,182.3
Transport cost	$ 2836.46
Overhead cost	$ 14,182.3

Total initial cost = $ 87,040 + $ 283,646 + $ 14,182.3 + $ 2836.46 + $ 14,182.3.
Total initial cost = $ 40,1888.

Table 12
The details of a first annual cost, annual maintenance cost and O & M cost.

Sr #	Item	Cost
01	First Annual Cost[1]	$ 44,207.68
02	Annual maintenance cost[2]	$ 2211
03	O & M cost[3]	$ 3667

[1] First Annual cost (FAC) = CRF * Initial cost [41].
[2] Annual maintenance cost (AMC) = 5% * First annual cost [41].
[3] O&M cost = AMC * $i_{\text{interest}}\,(1 + i_{\text{interest}})^{\text{number of years}} / i_{\text{interest}}\,(1+ i_{\text{interest}})^{\text{number of years}} - 1$ [41].

Table 13
The details of replacement cost of items.

Sr #	Item	Replacement	Cost
01	Splitter	After 2 years [46]	$ 32
02	Pump	After 5 years [49]	$ 32,461
03	Mixer	After 5 years [47]	$ 13,019
Total Replacement Cost			**$ 45,512**

Replacement cost = Equipment cost * $(1+ i_{\text{inflation}})^{\text{replacement year-1}} / (1+ i_{\text{interest}})^{\text{replacement year}}$ [41].
Average inflation rate of Pakistan from the last 30 years (1992 to 2022) is 8.47% [50].

Table 14
Salvage value cost.

Sr #	Item	Cost
01	Salvage value[1]	$ 2634

[1] Salvage value = 0.15 * initial cost / $(1+ i_{\text{interest}})^{\text{number of years}}$ [41].

Table 15
Details of present worth of capital cost.

Present Worth of Capital Cost (PWCC)	
Initial cost	$ 40,1888
O&M cost	$ 3667
Replacement cost	$ 45,512
Salvage value	$ 2634

PWCC = Initial cost + O&M cost + Replacement cost – Salvage value.
PWCC = $ 40,1888 + $ 3667 + $ 45,512 - $ 2634.
PWCC = $ 448,433.

6.6. Details of different items for finding Levelized cost of energy (LCOE) and payback period

See Table 16.

The Levelized cost of energy for a system operation is found to be 30 $/MWh or 0.03 $/kWh for one year with a payback period of 3 years [51,52]. Fig. 16 shows the techno-economic results. In this Figure, different cost percentages are mentioned for an operation of a whole system. The Levelized cost of energy of 30 $/MWh for this system is smaller than in comparison with literature [52]. The presented Fig. 16 displays that energy cost has a higher value than other costs. The reason is such that it includes all the costs of equipment as well as installation cost, transport cost, and overhead cost. The Operation and maintenance cost is negligible as compared to rest of all the costs so it displays as 0% in Fig. 16. The LCOE of $30/MWh is required to run this waste heat recovery system based upon an indirect evaporative cooler. The payback period of this cost is 3 years.

Table 16
The details of total annual cost, LCOE, initial investment cost, cash flow cost and payback period.

Sr #	Item	Cost
01	Total annual cost[1]	$ 49,328
02	Electricity generation	1,604,150 kWh
03	Levelized cost of energy[2]	0.030 $/kWh / 30 $/MWh
04	Initial investment	$ 448,433
05	Cash flow	$ 150,000
06	Payback period[3]	3 years

[1] Total Annual cost = CRF * PWCC.
[2] Levelized cost of energy (LCOE) = Total Annual Cost (TAC) / Electricity generation.
[3] Payback Period (PP) = Initial investment / Cash flow.

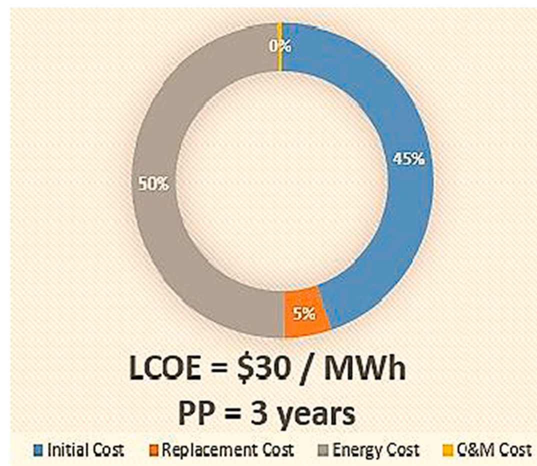

Fig. 16. Result of techno-economic analysis.

7. Conclusion

For a sustainable and low carbon emitting system, units of waste heat recovery that are thermally efficient undergo broad research. The best-proposed system for the recovery of waste heat in terms of both thermal performance and economic feasibility is the Maisotsenko Humid air bottoming cycle (MHABC) [3]. By using previous numerical and experimental results this model of heat and mass exchanger is validated.

In this research, plant performance is observed at different operating parameters and different working fluids. It is found that by using air as a working fluid, the waste heat recovery plant produces 68.53 MW with an efficiency of 35.44%. Using a mixture of Carbon dioxide and toluene (CO_2-C_7H_8) plant shows an output power of 48.59 MW with an efficiency of 38.57%. From this research, it is concluded that air is the best working fluid in terms of power generation.

The parameters of industry operated in a model which presents 38.04 MW power with an efficiency value of 27.78% by using air as a working fluid. By executing the model with a mixture of CO_2-C_7H_8 model exhibits a power generation of 30.63 MW with an efficiency value of 27.77%. Here, air again acts as the best suitable working fluid for power generation. This also highlights that a huge amount of net output power, as well as efficiency, increases by using an indirect evaporative cooler. Techno-economic analysis of this waste heat recovery system presents a cost of 30 $/MWh. This cost is economically feasible for this system.

7.1. Future recommendation

Use ammonia and a mixture of n-butane (60%) and n-hexane (40%) as a working fluid in bottoming cycle. Perform exergy analysis of the system using these multi-fluids. Another application would be to integrate the conventional desalination units [53–55] with the M-power cycles. Another interesting simulation tool would be to evaluate the M-power cycles using digital twin modelling [56–62].

CRediT authorship contribution statement

Haris Khan: Methodology, Software, Validation, Formal analysis, Investigation, Data curation, Writing – original draft, Writing – review & editing, Visualization. **Rasikh Tariq:** Methodology, Software, Validation, Formal analysis, Investigation, Data curation, Writing – original draft, Writing – review & editing, Visualization. **Syed Nasir Shah:** Conceptualization, Methodology, Validation, Investigation, Resources, Writing – original draft, Writing – review & editing, Visualization, Supervision, Project administration, Funding acquisition. **Muhammad Wakil Shahzad:** Supervision, Project administration, Funding acquisition, Writing – review & editing, Investigation. **Tanveer Ahmad:** Conceptualization, Methodology, Validation, Investigation, Resources, Writing – original draft, Writing – review & editing, Visualization, Supervision, Project administration, Funding acquisition. **Nadeem Ahmed Sheikh:** Conceptualization, Methodology, Validation, Investigation, Resources, Writing – original draft, Writing – review & editing, Visualization, Supervision, Project administration, Funding acquisition.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

Data will be made available on request.

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