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Research Article

Thermo-economic assessment of air bottoming cycle technology for waste heat recovery purposes from the preheater tower of an Algerian cement plant

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ABSTRACT

The present work deals with the deployment of an air bottoming cycle as a technology for thermal energy valorization. This solution is integrated within the preheater tower on an Algerian cement plant. A combined MATLAB- Coolprop optimization tool has been developed to carry out a thermo-economic calculation allowing to design and optimize the air bottoming cycle and also check its economic returns. The developed code employs the genetic algorithm through a multi-objective function aiming to maximize the cycle's net power output as a priority objective, then minimize the heat exchanger's area of exchange, and also maximize the cycle's net present value. through the implemented models on it, the code gives the ability to confirm the applicability of this technology through the Algerian market which is characterized by one of the cheapest prices of electricity in the world that doesn't exceed 0.040 $ kWh^{-1}. The obtained results highlight that the implementation of such technology for energy valorization purposes from the preheater tower of the selected cement plant can be a profitable and attractive solution, particularly in scenarios of water scarcity. Since the implementation of this technology presented a power capacity of 1.2 MW covering then around 7.8% of the electricity demand of the investigated cement plant. The economic analysis of the proposed ABC cycle pointed out a net present value of 6.59 M$, and a payback time less than 3.6 years, besides a Levelized cost of energy less than 0.017 $ kWh^{-1} which is a comparable value to the subsidized prices of electricity in the Algerian market.

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INTRODUCTION

In the last decades, most of the researches related to the environment and energy field has been directed towards reducing greenhouse gas (GHG) emissions and decreasing energy consumption. Mainly because the high-energy consumption of different industrial sectors that leads to GHG emissions that have a direct impact on global warming and environmental pollution. In the midst of this, the massive emissions associated with production processes in various industries represent more than 30% of global GHG emissions [1]. The cement industry, in turn, occupies a key position in these emissions making it recognized as a highly energy-intensive sector. In addition, it is also characterized by a huge loss of energy that can achieve 40% of the total energy input as reported by previous work [2]. Simultaneously, worldwide cement manufacturing is responsible for about 8 % of global GHG emissions [3].

In this context marked by global warming and industrial pollution, the reduction of GHG emissions has become a priority for the scientific community. The Waste Heat Recovery (WHR) technologies are among the worldwide attractive solutions and can be used for heat process integration [4], activating absorption chillers [5] or electricity production [6]. Power WHR applications are mainly based on air bottoming cycles [7], both the steam and the organic Rankine cycles [8], the Kalina as well the combined cycles [9,10]. The air bottoming cycle (ABC) is considered among the simplest technologies due to the fact that it is characterized by a simple configuration besides to a low number of devices. This technology has been introduced by general electric in 1988 and was expected to be an alternative to the conventional steam cycles [7]. After being patented by Anderson and Ferrell, the exhausts from the coal fuel were employed to activate the ABC cycle [11]. Later on, Wicks et al. [12], elucidated its framework and concept. The interest in ABC cycles grows over the years as reported by the different scientific literature works. Najjar et al. [13], analyzed and investigated the performances of a combined gas turbine and air bottoming cycle system. They found that the implementation of this cycle can improve the net power output by about 30% and the efficiency by about 23% compared to the use of only the gas turbine. Besides that, the conducted study pointed out that ABC cycle is profitable from an economic point of view. In addition, Korobitsyn et al. [14], examined and conducted a techno-economic analysis of an ABC cycle for Combined Heat and Power-CHP-operating mode for different industrial applications such as bakery, milk industry, whey powder drying process as well as glass industry. The results outlined that the implementation of this technology for the industries requiring hot air results in reduction of fuel consumption with a payback period ranging between 3 to 4 years. As well as Saghafifar et al. [15], they conduct a techno-economic analysis for an ABC cycle for turbine's exhaust gas WHR. Two different configurations have been investigated: with and without intercooler. The results show that the first configuration without the intercooler guarantees an overall system efficiency of 41.89% with an operation cost of 0.7316 $\$ s^{-1}$ while the use of intercooler, as a second configuration, increases the system's overall efficiency to 43.17% and decreases the total operation cost to 0.7273$\$ s^{-1}$. Moreover, Khaldi [16], evaluated the thermodynamic feasibility of using ABC cycle for solar gas hybrid power plants. For his study, Khaldi proposed a thermodynamic scheme of solar ABC cycle. The performed work focused on a comparative study between the proposed hybrid model and two different models presented in literature (without solar energy). The results show that the solar-ABC cycle and the steam bottoming cycle had comparable produced power; whilst the conventional ABC cycle had the smaller generation capacity. The opposite trend has been observed in regard to efficiency meaning that the proposed model is specified by a small efficiency compared to the other models. In the majority of existing works, the authors have focused on the applications of the cycle at large scales, where the net power output can achieve 50 MW or more while very limited works studied the possibility of application of the this technology at the small scale. Among these works we found the interesting study performed by Pierobon et al. [17] wherein an ABC cycle was investigated for off-shore platform WHR. The authors carried out an multi-objective optimization with the aim of maximizing both the produced power by the cycle and the corresponding net present value, while also, minimizing the size of the recuperator. Results show that the maximum value of power output that can be achieved for this application is estimated at about 1.98 MW while the economic analysis revealed a total investment cost of 2.02 M$ associated with a net present value of 2.8 M$. Finally, it was concluded that the implementation of the ABC for this application combined with the proposed approach can boost the engine power output by 16% leading to a thermal efficiency improvment of 5.2%.

However, to the authors' best of knowledge, no existing literature assesses the viability of ABC cycles for recuperating waste heat from the preheater tower in the cement plants. Although, it could be that such an approach holds significant promise as an efficient and effective solution, particularly in situations where water scarcity poses a challenge.

In the midst of this, Algeria is considered as one of the main cement producers in the MENA region, where over 17 industries operate and produce up to 40 million tons annually [18]. However, this level of production is accompanied by substantial heat losses at various stages of the manufacturing process, resulting in significant carbon dioxide emissions. Specifically, each ton of clinker production yields a CO2 emission rate of 0.5071 ton [19]. Anywise, Algeria undertake a new strategy aims to reduce energy consumption and GHG emissions, for a plan of actions proposed for the decade 2021-2030, in which waste valorization is considered as one of the intended objectives.

Therefore, The objective of this study is to evaluate the potential of implementing the air bottoming cycle (ABC) for recuperating waste heat in Algerian cement plants. As the integration of such technology faces several challenges especially the subsidized price of electricity which is very cheap compared to European countries. Such a factor can make the ABC technology non-profitable for the Algerian Cement market and compromise seriously its implementation as an energy efficiency measure. Thus, contrarly to our previous work [20], this work aims to undertake a comprehensive analysis of the ABC technology for energy valorization in Algerian cement plants, taking into consideration both thermodynamic and economic factors. The study seeks to evaluate the feasibility and practicality of adopting these technologies in the local cement industry. To achieve this objective, a MATLAB-based optimization code has been developed, utilizing a genetic algorithm and interfaced with Coolprop database for calculating thermodynamic points and parameters. Subsequently, a multi-objective optimization was performed to prioritize maximizing net power output, minimizing exchange area, and maximizing the net present value.

METHODOLOGY

In this section, a description of the selected case study is discussed first. Then, the thermodynamic model and the power maximization theory are expressed. Afterwards, the adopted methodologies dealing with the estimation of the overall heat transfer coefficient in addition to the economic analysis are presented.

Selected Case Study

In the present work, the data of a selected cement plant located in the northern region of Algeria was employed to assess the potential of waste heat recovery at its level. This plant has been in operation since 1973, producing approximately 0.95 million tons of cement annually with an operational factor of up to 92% per year. The selection of this plant was based on the availability of comprehensive data and the possibility of conducting a follow-up analysis. Furthermore, the findings from this study can be extrapolated to other cement plants in Algeria, given that the majority of these plants employ similar technologies, fuel types, and operational conditions.

The investigation of heat loss within the selected cement revealed the existence of two main sources of waste heat with a considerable flow rate. The main source is the preheater tower exhaust gas whilst the other one represents the hot air leaving the cooler section. The different parameters of these sources are shown in Table 1.

In this study, only the waste heat from the preheater tower is taken into consideration as a first energy efficiency approach. This selection has been motivated by the fact that the preheater tower available energy is greater than that of hot air (52% higher) since it presents a relatively high temperature compared to the hot air. Such higher temperatures are more suitable for ABC as waste heat recovery technology applications. Figure 1 illustrates the source of the waste heat at the preheater tower level.

Table 1. Properties of exhaust gas and hot air in the cement industry

Parameter		Preheater tower	Cooler
Mass Flow rate (kg s^{-1})		47.113	56.321
Pressure (bar)		1.013	1.013
Temperature (°C)		382	216
Components mole fraction (%)	CO_2	26	0
	O_2	3	21
	N_2	69	79
	H_2O	2	0

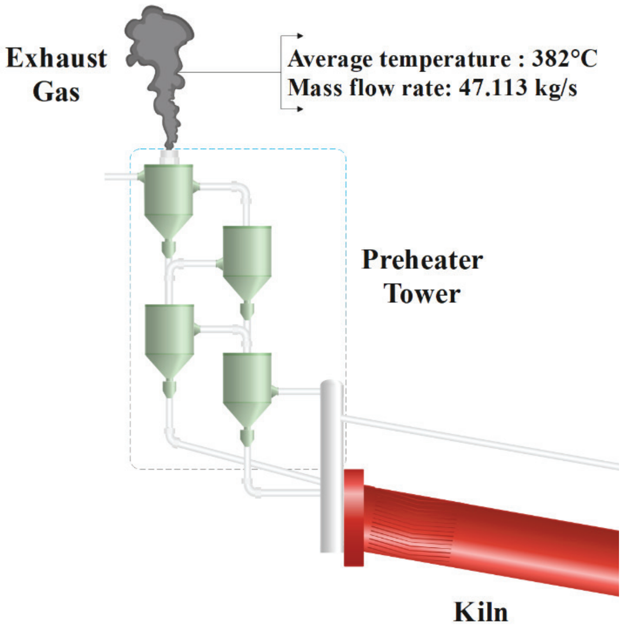

Figure 1. Exhaust gas at both preheater tower.

Thermodynamic Model and Technology Description

The ABC is an open thermodynamic cycle for waste heat recovery that was proposed since 1988s as a promising alternative for the stream bottoming cycle [7]. The ABC cycle is composed of a compressor, a heat exchanger and an expander, and looks like the Brayton cycle that we find in the open gas turbine cycle. The main distinction is that the working fluid crossing the turbine for the Brayton open gas cycle is a mixture of flue combustion gas while only air goes through the turbine for the ABC cycle. Additionally, the ABC cycle operates at moderate and low pressure and recovers waste heat energy as a heat source in place of the combustion chamber for the Brayton cycle open gas turbine cycle. Comparing now to the closed Brayton gas turbine cycle using air as working fluid as well as ABC cycle, the main difference between those two cycles remains in the fact that the closed Brayton cycle is activated by combustion of natural gas and the working fluid is cycled within a closed-loop while ABC cycle is open cycle using air as working fluid and recovering waste heat initially discharged to the atmosphere. Consequently, both open and closed Brayton cycles have positive carbon balance while the ABC cycle presents a net-zero carbon balance.

Furthermore, the absence of water as a medium fluid in the ABC cycle eliminates the need for steam equipment such as condensers and piping. This simplifies the configuration of the ABC cycle and allows for a combined architecture with reduced maintenance expenses and quick start-up time, as reported by Korobitsyn. [14]. On the other hand, this technology can be applied where water resources are scarce.

Operating principle of the ABC consists in compressing the ambient air through a compressor (1-2). The compressed air is subsequently directed to a heat exchanger for thermal energy exchange with the waste heat, prior to flowing through the turbine section (2-3). The air undergoes an expansion process in the turbine, resulting in the generation of shaft work, while simultaneously releasing heat and pressure (3-4). Finally, the air is discharged and can be useful for alternative utilization [21]. Figure 2 represents the basic diagram of the ABC cycle.

Then, for the computation of the diverse thermodynamic points and parameters, numerous equations are used, they can be summarised as follows:

Required compression power

$$\dot{W}_c = \dot{m}(h_2 - h_1) \tag{1}$$

Heat added to the system

$$\dot{Q}_{HEX} = \dot{m}(h_3 - h_2) \tag{2}$$

Turbine shaft power

$$\dot{W}_{Turb} = \dot{m}(h_4 - h_3) \tag{3}$$

The cycle's produced power

$$\dot{W}_{net} = \dot{W}_{Turb} - \dot{W}_{Comp} \tag{4}$$

The cycle's thermal efficiency

$$\eta = \frac{\dot{W}_{net}}{\dot{Q}_{HEX}} \tag{5}$$

In return, since the objective of the industry is to reduce electricity consumption, the power maximization concept is employed in this work, where the optimum pressure ratio value for maximizing the power output is expressed according to Boyce, [22] as follows:

$$r_p = \left[\left(\frac{T_3\, \eta_C \eta_T}{2\, T_1}\right) + \frac{1}{2}\right]^{\frac{\gamma}{\gamma-1}} \tag{6}$$

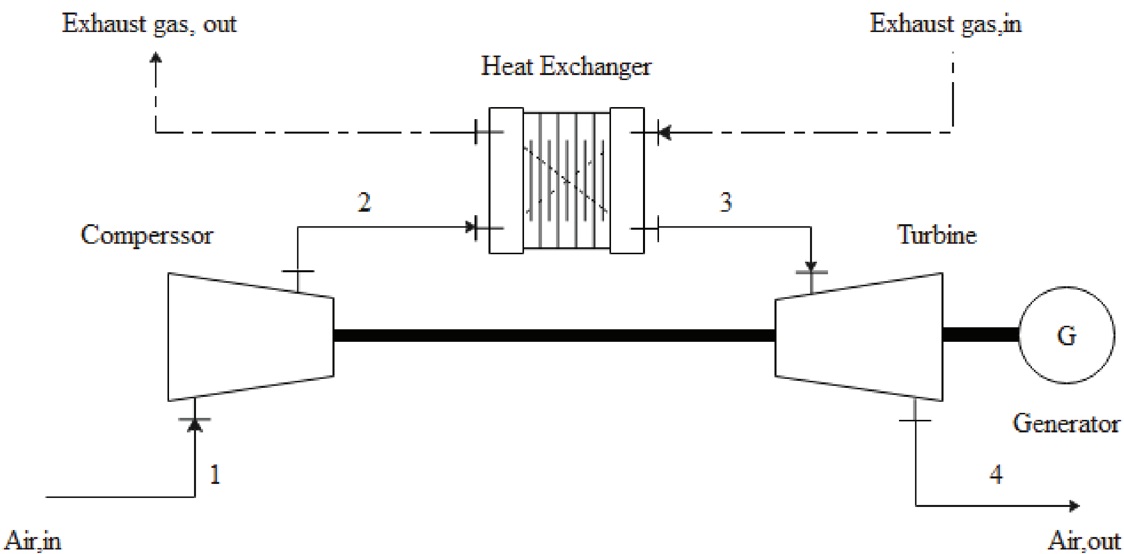

Figure 2. Schematic diagram of the basic air bottoming cycle [20].

Then, the estimation of isentropic efficiencies for the compressor and turbine is performed by utilizing the proposed values of pressure ratio and polytropic efficiency, in accordance with the methodology suggested by Ghojel [23], expressed as:

$$\eta_{is,c} = \left(\frac{r_{p\_c}^{\frac{(\gamma-1)}{\gamma}} - 1}{r_{p\_c}^{\frac{(\gamma-1)}{\gamma\,\eta_{pc}}} - 1} \right) \quad (7)$$

$$\eta_{is,t} = \left(\frac{r_{p\_t}^{\frac{\eta_{pt}\,(\gamma-1)}{\gamma}} - 1}{r_{p\_t}^{\frac{(\gamma-1)}{\gamma}} - 1} \right) \quad (8)$$

γ represents the specific heat ratio and it can be computed according to Kennedy et al. [24] as

$$\gamma = 1.42592 \; - \; 8.03974 \; \times 10^{-5} \times T \quad (9)$$

Where the temperature is taken at the average value of the compression and expansion processes, respectively.

Heat Transfer Soefficient

According to Douglas et al [25] and Pierobon and Haglind [17], there are two heat exchanger technologies that can be applied in case that the fluids in exchange are gases. These technologies are presented in the flat type besides the shell and tube heat exchangers. In this work, the authors choice fell on the shell and tube heat exchangers, as this type features by wide application and advantage of handling application in case there is a significant difference in pressure of the fluids in exchange as mentioned by Pierobon and Haglind [17], besides to their adaptability where there is a large difference between the heat transfer coefficients of the fluids as pointed in [26]. So all heat exchangers in this study are assumed to be shell and tube types.

So as a way to design and calculate their area of exchange, the correlations proposed in Coulson and Richardsons [27], and pointed out in [17] are employed.

Thus, the general heat transfer equation is given as the multiply of various parameters reperesented in the exchange area A besides the logarithmic mean temperature difference ΔT_{lm} and the overall heat transfer coefficient U, and it is given as:

$$\dot{Q} = UA\Delta T_{lm} \quad (10)$$

Where ΔT_{lm} is the logarithmic mean temperature, and it has the following formula:

$$\Delta T_{lm} = \frac{(T_{hot,in} - T_4) - (T_{hot,out} - T_3)}{\ln \frac{(T_{hot,in} - T_4)}{(T_{hot,out} - T_3)}} \quad (11)$$

And the overall heat transfer coefficient U is given as:

$$\frac{1}{U} = \frac{1}{h_{gas}} + \frac{1}{F_{gas}} + \frac{d_o \ln\left(\frac{d_o}{d_i}\right)}{2\,k_w} + \frac{d_o}{d_i}\frac{1}{F_{air}} + \frac{d_o}{d_i}\frac{1}{h_{air}} \quad (12)$$

F_{gas} and F_{air} are the fouling factors, they are fixed to be 5000 Wm^2K^{-1} and 7500 Wm^2K^{-1} for gas and air respectively. In addition, the coefficient of the heat transfer for each side h is calculated as a function of the Nusselt number Nu, in which it is expressed as:

$$Nu = \frac{d\,h}{\lambda} \quad (13)$$

Where d presents the diameter (inner diameter d_i for the tube side, and equivalent diameter D_e for the shell side), h referes the local heat transfer coefficient, while λ presents the thermal conductivity. Therefore, the Nusselt number of the hot gas in the shell side is computed using the correlation proposed by Zhukauskas et al [28], [29] as follows:

For laminar flow, where $Re_{gas}<10^3$

$$Nu_{gas} = 0.71 \times Re_{gas}^{0.5} \times Pr_{gas}^{0.36} \left(\frac{Pr_{gas}}{Pr_{gas,wall}}\right)^{0.25} \quad (14)$$

Fir transition region, where $10^3<Re_{gas}<2x10^5$

$$Nu_{gas} = 0.35 \times \varepsilon^{0.2} \times Re_{gas}^{0.6} \times Pr_{gas}^{0.36} \left(\frac{Pr_{gas}}{Pr_{gas,wall}}\right)^{0.25} \quad (15)$$

where

$$\varepsilon = \frac{a}{b} \quad (16)$$

Where Re presents the Reynolds number, Pr presents the Prandtl number.

$$Re_{shell} = \frac{\rho_{gas} \times V_{gas} \times D_e}{\mu_{gas}} \quad (17)$$

$$Re_{shell} = \frac{\dot{m}_{gas} \times D_e}{A_c \times \mu_{gas}} \quad (18)$$

A_c is the crossflow area, and is calculated as

$$A_c = \frac{D_s \times C_t \times B}{P_t} \quad (19)$$

And C_t is the tube clearance is determinate as

$$C_t = P_t - d_o \quad (20)$$

$$Pr_{gas} = \frac{\mu_{gas} \times C_P}{k} \quad (21)$$

The triangular pitch arrangement is adopted to calculate the equivalent diameter. The reason is that this type of arrangement provides a higher heat transfer coefficient and leads to more compactness in the exchange area. Therefore, this results in a cheaper heat exchanger compared to other ones using the square arrangement as mentioned in [30].

Thus, the equivalent diameter of the triangular pitch arrangement is determined according to [31] as follow:

$$D_e = \frac{4 \times (\frac{\sqrt{3}p_t{}^2}{4} - \pi \times \frac{d_o^2}{8})}{\pi \times \frac{d_o}{2}} \tag{22}$$

And the shell diameter can be calculated according to [32] by:

$$D_s = 0.637\sqrt{\frac{CL}{CTP}}\left[\frac{A_o \times ({}^{P_t}/_{d_o})^2 \times d_o}{L}\right]^{1/2} \tag{23}$$

Where A_o is the outer surface of the tubes, and is calculated as:

$$A_o = \pi \times d_O \times N_t \times L \tag{24}$$

$$\text{CL=0.93 for one pass} \tag{25}$$

$$\text{CTP=0.87 for triangular pitch} \tag{26}$$

Where, CL represents the tube layout constant, while CTP represents the tube count calculate constant.

On the other side, the Nusselt number of the Air in the tube side is calculated as follows:

For turbulent flow, $Re > 10^4$

$$Nu_{tube} = 0.021\, Re_{tube}{}^{0.8} Pr_{tube}{}^{0.33} (\frac{\mu}{\mu_{tube}})^{0,14} \tag{27}$$

For transition region $2100<Re<10^4$, the Hausen correlation mentioned in [33] is used, where the exponent of the Prandtl number is set to 0.45 according to Koch [34]

$$Nu_{tube} = 0.116\left[(Re_{tube}{}^{\frac{2}{3}} - 125)\, Pr_{tube}{}^{0.45}[1 + \frac{D_e}{L}]^{2/3}\right](\frac{\mu}{\mu_{tube}})^{0,14} \tag{28}$$

Where

$$Pr_{tube} = \frac{\mu_{air} \times C_P}{k} \tag{29}$$

$$Re_{air} = \frac{\dot{m}_{air} \times d_i}{A_c \times \mu_{air}} \tag{30}$$

$$A_c = \frac{\pi \times d_i{}^2}{4} \times \frac{N_t}{N_p} \tag{31}$$

Where N_t refers to the number of tubes, while N_p represents the number of passes.

Economic Evaluation

The various investment costs of the different devices of the ABC cycle are estimated using numerous equations which are selected to guarantee the coverage of the size range. So, the method of preliminary cost estimating used for chemical plants is adopted as a means to conduct the economic analysis, named the Module Costing Method (MCT). The mentioned method provides the ability to compute the purchase cost, including its related charges.

In this work, the purchase costs of the compressors and the turbines are computed using the equations suggested by Valero et al. [35] while the heat exchangers cost is computed using the equation suggested by Guo-Yan et al. [36]. Whereas, the generator cost is estimated employing the equation introduced by Lazzerato et al. [37].

The value of the chemical engineering plant cost index (CEPCI 2020: 596.2) is employed as a means to compute the new purchase cost for each device. Therefore, the new purchase cost of each device can be determined as proposed Zhang et al. [38]:

$$PEC_i = PEC_i^0 \frac{CEPCI_{new}}{CEPCI_{ref}} \tag{32}$$

Table 2 lists the equations used in this study, while Table 3 resumes the different factors used for these equations.

Table 2. The different equations of the purchase costs

Equipment	Cost equation	Ref
Air Compressor	$PEC_{AC}^0 = \left(\frac{c_1 \dot{m}_{ABC}}{c_2 - \eta_C}\right)\left(\frac{p_2}{p_1}\right) ln\left(\frac{p_2}{p_1}\right)$	[35]
Air Turbine	$PEC_{AT}^0 = \left(\frac{c_3 \dot{m}_{ABC}}{c_4 - \eta_T}\right) ln\left(\frac{p_3}{p_4}\right)[1 + \exp(c_5\, T_3 - c_6)]$	[35]
Shell and Tube heat exchanger	$PEC_{HE}^0 = c_7 A^{c_8}$	[36]
Generator	$PEC_{Gen}^0 = c_9 P_{ABC}^{c_{10}}$	[37]

Table 3. Coefficients used in the equations of Table 1

Equipment	Coefficients	Ref
Air Compressor	$c_1 = 39{,}5\ \$/(kg/s)$ $c_2 = 0.9$	[35]
Air Turbine	$c_3 = 266.3\ \$/(kg/s)$ $c_4 = 0.92$ $c_5 = 0.036\ (K^{-1})$ $c_6 = 54.4$	[35]
Shell and Tube heat exchanger	$c_7 = 2768\ \$/m^2$ $c_8 = 0.573$	[36]
Generator	$c_9 = 60\ (\$/kW)$ $c_{10} = 0.95$	[37]

In return, the total cost related to the ABC cycle, PEC_{ABC}, is computed using equation (33)

$$PEC_{ABC} = \sum_{1}^{n} PEC_i \tag{33}$$

Where PEC_i is the investment in each device.

Moreover, the total cost of the site is predicted through the multiplication of the total cost of ABC by 1.1. This value has been selected in accordance with the industrial partners which has experience in the domain of management and economic part.

$$C_{site} = 1.1 \times PEC_{ABC} \tag{34}$$

While the operation and maintenance cost is assumed to be 2% of the site cost according to Pezzuolo et al [39].

$$C_{O\&M} = 0.02 \times C_{site} \tag{35}$$

In return, the cash flow from electricity generation is estimated using equation (36), where tax exemption is considered according to the executive Decree No. 02-01 of 6 February 2002 (Algerian official journal) [40], This decree introduced the concept of self-electricity generation for fulfilling specific needs and waived the requirement of an exploitation license for power generation up to 25 MW. Therefore, as the objectives of this study align with these requirements, the resulting cash flow can be determined as:

$$CF = (S_{annual} - C_{O\&M}) \tag{36}$$

Where:

$$S_{annual} = E_{annual} \times s_E \tag{37}$$

$$E_{annual} = H_{Op,annual} \times W_E \tag{38}$$

$$H_{Op,annual} = f_a \times 365 \times 24 \tag{39}$$

f_a refers to the operational factor, s_E presents the electricity price, while E_{annual} represents the annual electricity production.

The price of electricity s_E is calculated employing the data which listed by the Algerian ministry of energy [41], where the produced electricity is assumed to cover the rush hours consumption, and the rest used for the normal hours, and it is given as:

$$s_E = ((P_{el} \times 24 - P_{AvC} \times 4) \times s_E + 4 \times P_{AvC} \times s_E)/(P_{el} \times 24) \tag{40}$$

The net present value is calculated according to Bejan et al [42] as follows:

$$NPV = CF \times RF - C_{site} \tag{41}$$

Where

$$RF = \sum_{1}^{n} \frac{1}{(1+i)^n} \tag{42}$$

RF: capital recovery factor
i: annual interest rate
n: number of years

Then, the profitability index can be calculated through deviding NPV by C_{site}

$$PI = \frac{NPV}{C_{site}} \tag{43}$$

The Levelized Cost of Energy $LCOE$ refers to the associated cost of each produced electrical energy unit. It is estimated as

$$LCOE = \frac{C_{site} + C_{O\&M} \times RF}{E_{annual} \times RF} \tag{44}$$

Finally, the simple payback period is given by

$$SPB = \frac{C_{site}}{CF} \tag{45}$$

OPTIMIZATION SETTINGS AND PARAMETRIZATIONS

As mentioned before, To undertake this study, an optimization tool was developed and utilized to design and optimize the ABC cycle for waste heat recovery. The tool was implemented in the MATLAB environment and employed the Coolprop database for accurate thermodynamic calculations [43]. In addition, this tool employs one of Matlab toolboxes represented in the genetic algorithm as a means to estimate the optimum parameters based on the type of heat source. The code provides the flexibility to select single or multi-objective optimization approaches to optimize parameters such as net power output, efficiency, and net present value. The developed optimization code offers a valuable tool to investigate thermal energy valorization at the level of cement industries.

In this work, the tool employs a multi-objective function to design and optimize the ABC cycle. The optimization objective is to simultaneously maximize the produced power besides the net present value while minimizing the exchange area for the heat exchanger.. In this optimization, the net power output is set to be a priority objective, where the code gives the priority to optimize this variable, while the two rest objectives are linked to the first objective, in which they are optimized according to the first variable. The schema represented in Figure 3 explains this feature.

Table 4. Adopted parameters for the ABC cycle's different devices

Parameter	Value
Thermodynamic Part	
Polytropic efficiency of the Compressor (-)	0.88
Polytropic efficiency of the Turbine (-)	0.88
Electric generator efficiency (-)	0.98
Economic Part	
Interest rate (%)	2.5
System lifetime (year)	20

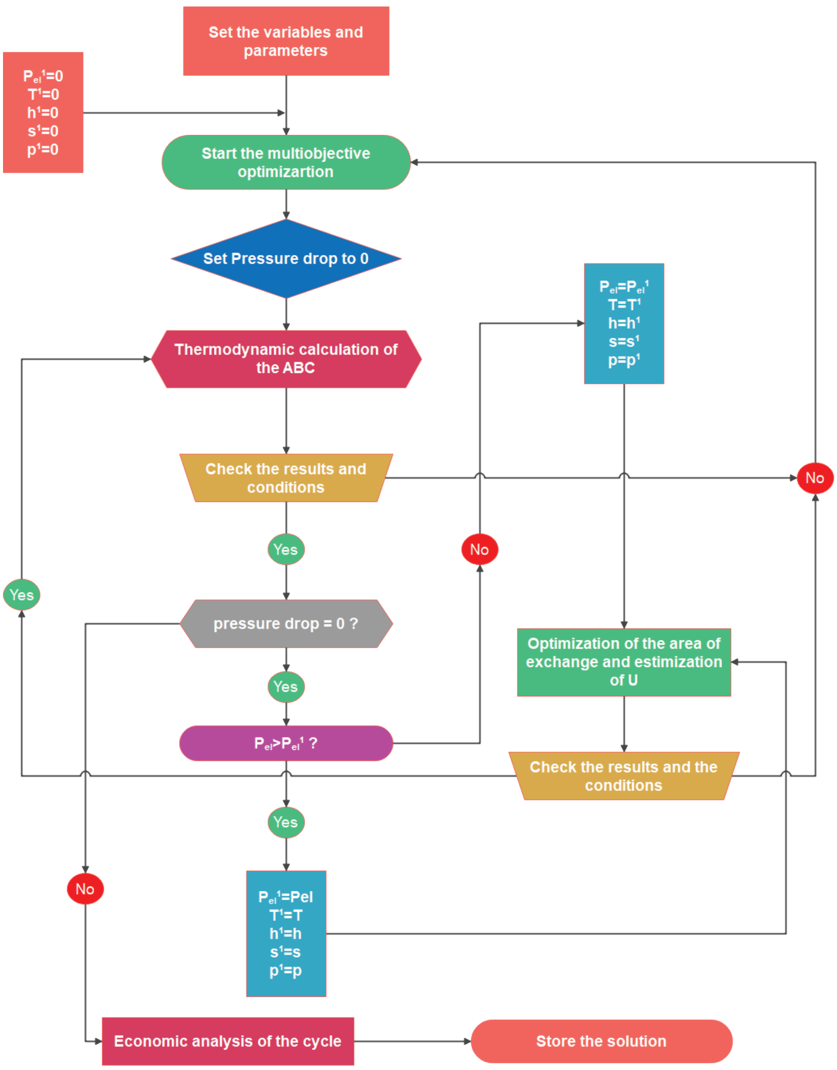

Figure 3. Methodology scheme involving the multi-objective optimization.

In addition, to perform this optimization, the basic architecture of the ABC cycle has been taken into cosideration in the tool, the choice fell on this configuration as it leads to decreased plant expenditures and operational intricacy, Figure 2 illustrate the examined configuration.

In return, as a means to select and assume the parameters and the properties of the various devices used in the cycle, the authors relied on their own expertise, literature sources, as well as data and knowledge provided by plant manufacturers. The selected parameters are presented in Table 4.

On the other side, in the present study, the provided upper boundaries and also lower boundaries for the various optimized parameters and variables, are shown in Table 5.

Table 5. Optimization upper and lower boundaries

Parameters to be optimized	LB	UB
Outlet temperature of the heat source, $T_{Hot,out}$ (°C)	70	T_{hot}
Inlet Temperature to the Turbine, TIT (°C)	100	TIT_{max}
Minimum PP in the Heat exchanger, $\Delta T_{pp,HE}$ (°C)	20	100
Pressure ratio (-)	1.5	4.5
Number of tubes (-)	5	1000
Length of the tube (m)	1	4
Inner diameter of the tube (mm)	16	50
Number of baffles (-)	1	10
Rapport pitch / outer diameter (P_t/d_o)	1.15	2.5
Thickness of the tube (mm)	3	/
Allowed pressure drop(%)	2	/

The maximum turbine inlet temperature TIT_{max}, in this case, is set to be T_{hot}-$\Delta T_{pp,HE}$, while the value of pressure ratio is limited between 1.5 and 4.5 according to Pierobon et al. [17].

Numerous other checks are implemented in the optimizer, in order to select values that give a good fitting regarding industrial practice. This is made possible by adopting the constraints discussed by Taborek [44], such as the baffling, which should be higher than 50 mm and the rapport between the length of the tube and the shell diameter should be between 3 and 15.

In return, in order to carry out this optimization, the parameters assumed to the genetic algorithm are:

- Number of populations: 500,
- Number of generations: 300,
- Crossover fraction: 0.8,
- Mutation fraction: 0.2.

The mentioned values has been tested and checked in terms of time and results, the outcomes confirm that higher values in population or generation sizes. E.g. adoption of 700 instead of 500 in generation size gives the same results but with extra time up to 30%.

RESULTS AND DISCUSSION

Validation of The Developed Code

As a way to ensure that the code gives confidence in thermodynamic results, the model has been validated according to literature by employing the case introduced by Pierobon et al. [17]. The choice fell on this case since the authors presented a case study in which they worked on the maximization of the net power output of an ABC technology for offshore application. So, to examine the developed tool, all the mentioned parameters and variables are adopted and used in the code, while several variables are fixed, being the model of power maximization represented in this study is different than the one represented in [17]. Thus, the exhaust gas outlet temperature and the recuperator inlet temperature difference are fixed according to Pierobon et al [17], while the value of the heat rate in the recuperator is used in order to estimate the composition of the exhaust gas. The optimization results pointed out a maximum deviation of about 1% in the pressure ratio of the cycle, while a deviation of less than 2% is recorded in the net power output. In contrast, a maximum deviation of about 2.34% is obtained in the value of the thermal efficiency, while the other parameters showed small deviations ranges between 0 and 0.57% as listed in Table 5. These small deviations can be justified by the fact that the gas components were estimated with insufficient accuracy, mainly because the authors did not share them due to industry privacy and confidentiality policy. Therefore, the obtained results confirm that the model presented in this study can be adopted as it gives confidence in results.

Table 6. Comparison between the results

variable	present	[17]	Deviation (%)
Pressure ratio p_2/p_1 (-)	2.77	2.8	1.07
Mass flow rate, $\dot{m}$ (kg s^{-1})	98.96	98.4	0.57
Exhaust gas outlet temperature, $T_{h,out}$ (°C)	145.5	145.5	0
Recuperator inlet temperature difference, ΔT_1 (°C)	26	26	0
Heat rate (recuperator), Q (MW)	22.98	22.97	0.04
Net power output, P_{el} (kW)	2848	2793	1.93
Thermal efficiency, η (%)	12.39	12.1	2.34

Optimization Outcomes

As mentioned in section 3, a multi-objective optimization aims for maximizing the net power output as a priority objective, then, minimizing the area of exchange of the heat exchanger, and also maximizing the net present value as a second priority is carried out. Table 7 contains the outcomes of this optimization.

Table 7. Optimization outcomes for the different variables

Optimized Variable	*Value*
Thermodynamic	
Pressure ratio p_2/p_1 (-)	2.57
Mass flow rate of the working fluid, $\dot{m}$ (kg s^{-1})	48.93
Outlet temperature of the Heat source, $T_{Hot,out}$ (°C)	153.16
Inlet Temperature to the Turbine, TIT (°C)	359.99
Minimum PP in the heat exchanger, $\Delta T_{pp,HE}$ (°C)	21.7
Isentropic efficiency of the Compressor (%)	86.3
Isentropic efficiency of the Turbine (%)	89.4
Produced Power, P_{ABC} (MW)	1.20
Thermal efficiency (%)	10.4
Economic	
Heat exchanger overall heat transfer coefficient, U (W m^{-2} K^{-1})	32.8
Cost of the ABC unit, C_{site} (M$)	1.97
Net Present Value (M$)	6.59
Payback Time (Years)	3.6
Levelized cost of energy, LCOE ($ kWh^{-1})	0.017
Profitability index, IP	3.35

Under the specified conditions and utilizing the power maximization theory equation (6), the results indicate that the ABC cycle has the potential to produce a net power output of 1.20 MW, accompanied by an optimal value for pressure ratio of 2.57 associated with a thermal efficiency value of 10.4%. To assess the performances of both devices; compressor and turbine, their isentropic efficiencies were determined through the use of equations (7) and (8), resulting in values of 86.3% and 89.4%, respectively. Additionally, the optimization analysis highlights the optimum heat source outlet temperature of 153.16 °C, a mass flow rate of 48.93 kgs-1, and a pinch point of 21.7 °C in the heat exchanger. These results suggest that the cycle has the potential to generate substantial power output with high efficiency while ensuring efficient heat transfer.

In addition, as a method of evaluating the effect of the compression ratio on the net power output and also efficiency, an examination of the cycle was performed by varying the compression ratio, in which the cycle has been optimized at different pressure ratio values ranging between 1.5 and 4. The outcomes of this investigation are shown in Figure 4 and Figure 5. For the net power output, the plot can be divided into two parts, in the first part, the results showed that there is a direct relationship between the produced power and the pressure ratio, in such a way the produced energy by the cycle increases with the increase of the pressure ratio to an optimal value, where the second part revealed a reverse relationship in such a way the produced power decreases with the increase of the pressure ratio until it reaches a minimum value corresponding to a pressure value of 4, the maximum value of net power output returns to an optimum compression ratio of about 2.6 and a net power output of 1.20 MW, this value coincides well with the result found by the first optimization using the power maximization theory as it is shown by the dot in the same plot. On the other side, the same approach can be applied to the efficiency where the maximum efficiency of the cycle can be obtained at a pressure ratio value of about 3. It should be noted, in addition, that the deployment of equation (6) results in a cycle efficiency lower since the optimization is set to maximize the net power output and not the efficiency.

On the other side, the estimation of the overall heat transfer coefficient highlights a small value of the overall heat transfer coefficient for the gas to gas heat exchanger. It was estimated at 32.8 W m^{-2} K^{-1} in this case. This value proves that the heat exchanger used for this application is a large one and is characterized by a large area of exchange. This value is validated according to Coulson and Richardsons [27] and also Roetzel [45], as they mentioned that the overall heat transfer coefficients of gas to gas exchange process are ranged between 5 and 50 W m^{-2} K^{-1}.

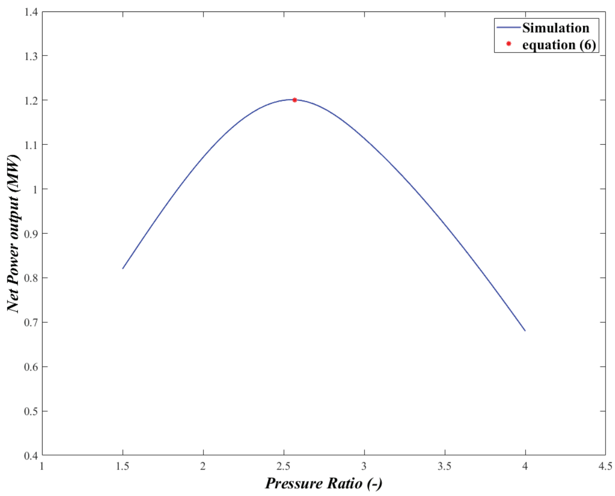

Figure 4. Net power output versus pressure ratio, the maximum point (red star) is obtained by equation (6).

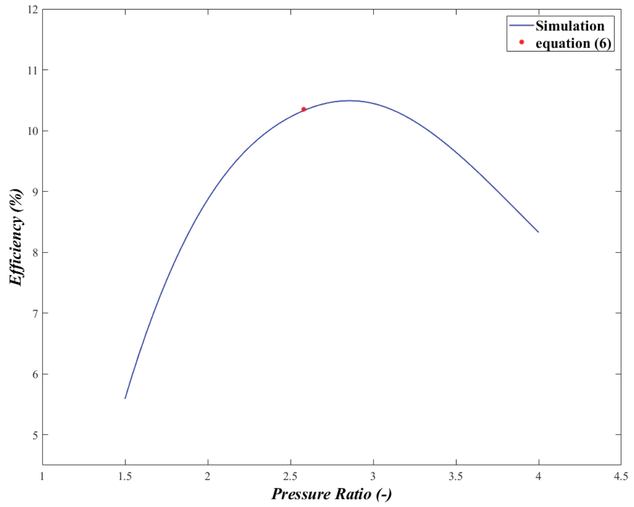

Figure 5. Efficiency versus pressure ratio, the maximum point (red star) is obtained by equation (6).

In return, the economic analysis using the model explained in section 2.4 and employing the parameters listed in Table 2 show that the total price of the ABC unit cycle is estimated at about 1.97 M$. Such technology is associated with a payback period of 3.6 years for a net present value of 6.59 M$. In addition, the estimation of the Levelized cost of energy for this cycle results in a value that doesn't exceed 0.017 $ kWh^{-1}, whilst the installation of this technology can give a value of profitability index that can exceed 3.35.

Moreover, the investigation of the investment cost in the different devices of the cycle revealed that the turbine and the heat exchanger are the main equipment in terms of investment costs, in which they represent around 84% of the total cost of the ABC unit. The value of the purchase cost of the heat exchanger is relatively high due to its large size to balance the small values of the heat transfer coefficient. In addition, the compressor and the electric generator represent low investment costs that don't exceed 11% and 5%, respectively as shown in Figure 6.

Therefore, The results of this study demonstrate that the introduction of the ABC technology at the preheater tower level in this cement industry is a favorable option from both economic and thermodynamic standpoints. This approach has the potential to reduce electricity consumption by generating approximately 1.2 MW of power, which represents about 7.8% of the total electricity demand for the investigated cement plant. Furthermore, the implementation of the ABC technology in this industry proves to be financially viable, with a Net Present Value exceeding 6.59 M$ and a Profitability Index of 3.35. Moreover, the daily electricity production can lead to a reduction of electricity bills by about 25.8% if the electricity is consumed during the rush hours. The cycle's design parameters are tabulated in Table 8, while the ABC cycle T-s diagram besides the pinch point position are presented in Figure 7.

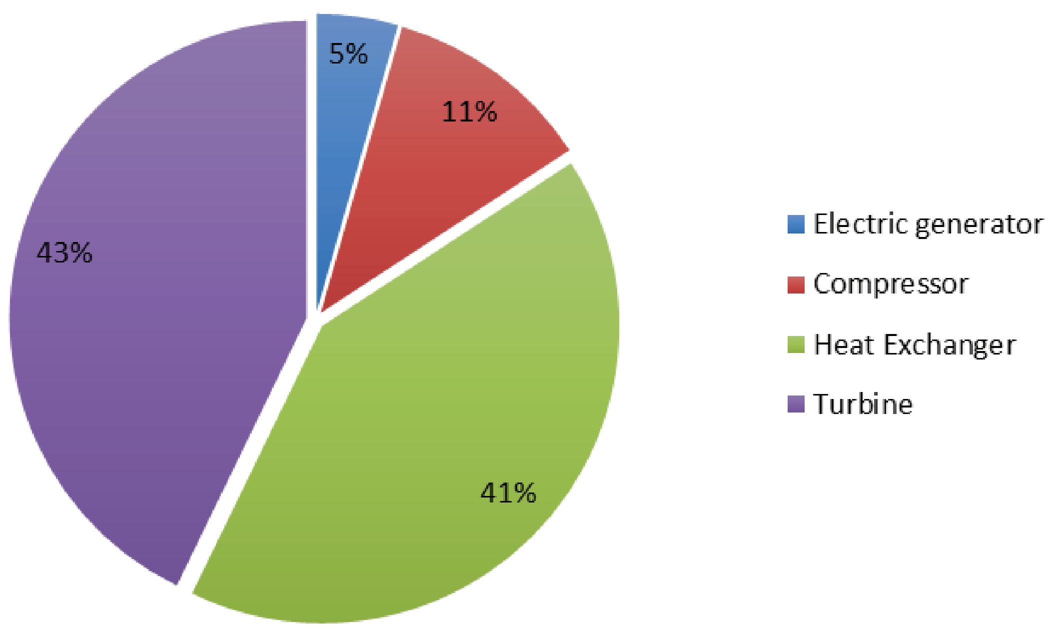

Figure 6. Investment costs in each device.

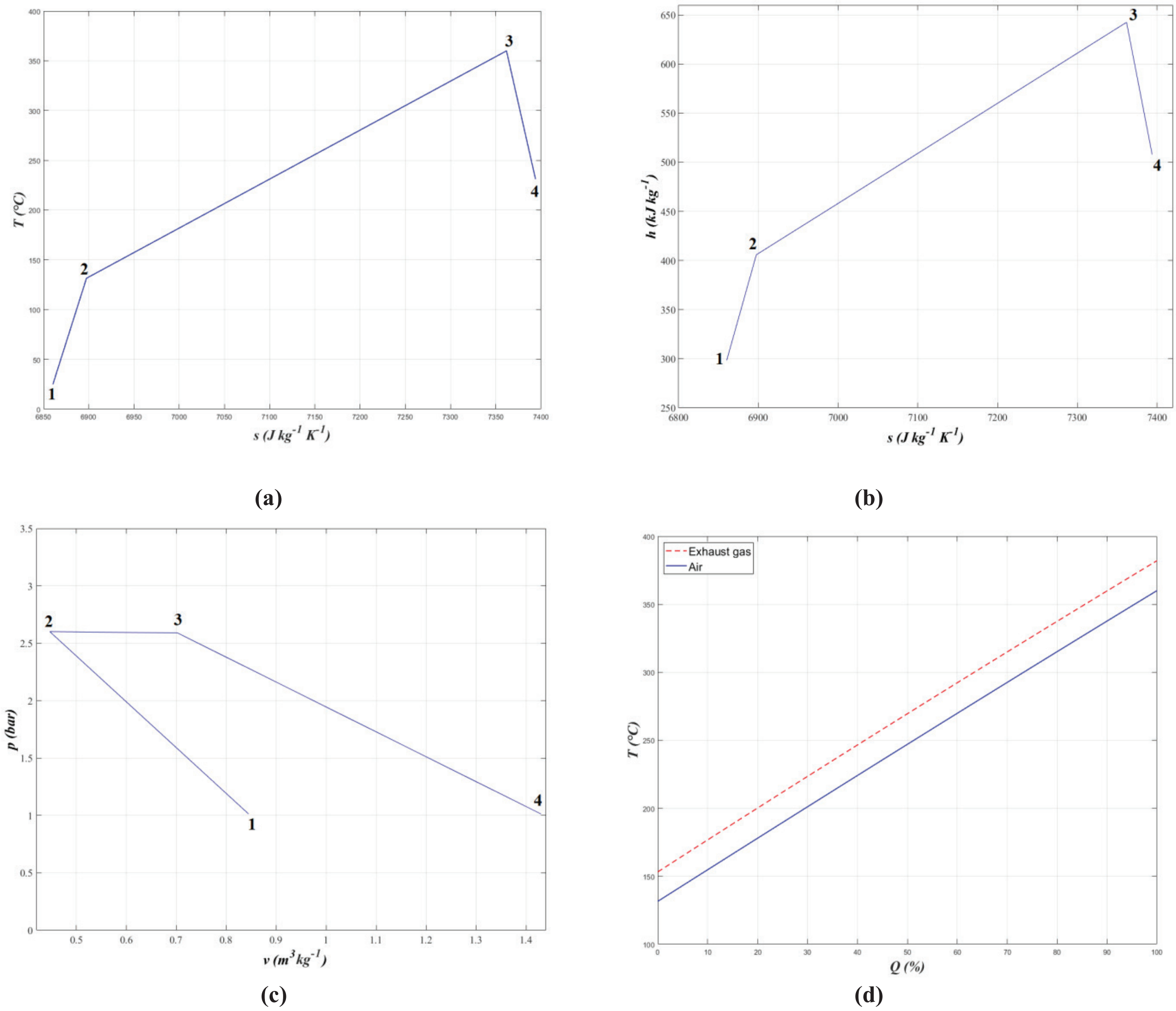

Figure 7. ABC cycle (a). T-s diagram, (b). h-s diagram, (c) p-v diagram, and (d) and heat exchanger pinch point.

Table 8. ABC cycle design point results

Points	T (°C)	s (J kg^{-1} K^{-1})	p (bar)	h (kJ kg^{-1})	v (m^3 kg^{-1})
1	25.00	6860.41	1.01	298.447	0.845
2	131.50	6897.52	2.60	405.841	0.447
3	359.99	7361.71	2.59	642.475	0.703
4	231.15	7393.82	1.01	507.882	1.430

CONCLUSION

This work proposed the integration of an Air bottoming cycle as a waste heat recovery technology in order to recover the waste heat discharged by a cement preheater tower characterized by a relatively high temperature compared to cooler's hot air. This technology aims to reduce energy consumption and greenhouse gas emissions. So, as a way to evaluate the adaptablitiy the of this technology, a real case study of a real Algerian cement plant has been selected and examined.

A comprehensive analysis of the application of the ABC technology for WHR in the preheater tower of the investigated cement plant demonstrated that this technology is capable of providing approximately 7.8% of the total electricity demand. The results also revealed that the proposed ABC cycle is economically viable and provides positive impacts on energy consumption, ultimately leading to a reduction in carbon emissions. The power output capacity and economic performance indicators of the designed ABC cycle suggest that its implementation would yield substantial benefits to the cement industry.

Thus, the ABC cycle has been demonstrated as an attractive solution for waste heat recovery purposes in the cement plant, particularly in situations where water scarcity is an issue, despite its relatively low efficiency. Results indicate that this technology offers favorable thermodynamic and economic outcomes, leading to reductions in energy consumption and greenhouse gas emissions through the reduction in natural gas consumption required for electricity production.

However, this technology is dedicated to medium and high-temperature sources, therefore, there is no guarantee for favourable results in case of the use of the ABC cycle for waste heat recovery purposes from low-temperature sources, mainly because lower temperature leads to lower pressure ratio constraints, and thus lower turbine work and cycle efficiency.

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

NOMENCLATURE

A	Surface area, m^2
a	Relative transverse pitch, m
b	Relative longitudinal pitch, m
CF	Cash flow, \$
C_p^0	Basic purchase cost, \$
C_t	Tube clearance, m
CL	Tube layout constant, nd
CTP	Tube count calculate constant, nd
d	diameter, m
E_{annual}	Annual electricity production, kWh.
f_a	Plant availability factor, nd
H	Fin height, m
$H_{Op,annual}$	Annual operating hours, hour
k	Thermal conductivity, W m^{-1} K^{-1}
$LCOE$	Levelized cost of energy, \$ kWh^{-1}
$\dot{m}$	Mass flow rate, kg s^{-1}
NPV	Net present value, \$
p	Pressure, bar
P_{avC}	Average consumed energy, kWh.
Pr	Prandlt number, nd
PI	Profitability index, nd
P	Power, kW
SPB	Simple payback time, year
S_{annual}	Annual incomes, \$
N_t	Number of tubes, nd
N_p	Number of passes, nd
T	Temperature, °C
TIT	Turbine inlet temperature, °C
N_t	Volumetric flow, m^3 s^{-1}
x	Vapor quality, nd

Abbreviations

CEPCI	Chemical engineering plant cost index
HEX	Heat exchanger
ABC	Air bottoming cycle
WHR	Waste heat recovery
PP	Pinch Point

Greek symbols

η	Efficiency, %
Δ	Difference
μ	Viscosity, N m^{-2}
ρ	Density, kg m^{-3}
v	Specific volume, m^3 kg^{-1}

Subscripts

t	tube
in	inlet
is	isentropic
out	outlet
o	outer
ref	reference
max	maximum
min	minimum

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