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Thermo–economic optimization of air bottoming cycles

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Abstract

In this work a thermo–economic optimization analysis is performed on two air bottoming cycle (ABC) configurations with and without intercooler in the bottoming cycle. Thermo–economic optimization modeling is developed and the effect of the mass flow rate ratio of bottoming cycle air mass flow rate with respect to the topping cycle air mass flow rate is examined in terms of both ABC plant efficiency and total operation cost.

Keywords: gas turbine; air bottoming cycle; thermo–economic optimization

1. Introduction

Improvements in turbine and compressor industries have enhanced gas turbine efficiency by increasing the pressure ratio and turbine inlet temperature (TIT). Due to the high temperatures of gas turbine exhaust gases, it was proposed to implement a bottoming cycle to recover a portion of the heat in the exhaust gases instead of releasing it into the environment. Currently, conventional combined cycles are delivered in gas-steam power plants. Steam bottoming cycle (SBC) is the most thermodynamically efficient bottoming cycle for large-scale power plants with capacities greater than 50 MWe [1]. For capacities less than 50 MWe, the complexity and high expense of the heat recovery steam generator (HRSG) and steam turbine argue in favor of seeking alternatives. Moreover, SBC requires water treatment facilities with increased capital investment cost [2].

One option is to integrate a gas turbine cycle as the bottoming cycle. The main advantage of implementing an air bottoming cycle (ABC) over SBC is the absence of a HRSG and condenser. Consequently, ABC can be competitive in small-scale power plants for capacities less than 50 MWe. The benefits of ABC are: low capital investment cost, low operating and maintenance cost, quick delivery, high flexibility, short start up time, and compact size [3]. Additional advantages are: low pollutant emissions and short construction time [3]. Besides, having an air intercooler, water consumption is reduced to a minimum and the plant can be implemented in regions with water shortages [4]. ABC is simple in terms of operation because there is no combustion process in the bottoming cycle and consequently no toxic media to cause erosion on bottoming cycle turbine blades [5].

ABC was patented in 1988 by W. Farrell of the General Electric Company [6]. The theoretical concept of ABC was first realized by Wicks in [7]. The concept of ABC was developed from the theory of the ideal fuel-burning engine in [8]. During the last two decades, various studies have been conducted to analyze a range of aspects of ABC.

The industrial applications of ABC for the cogeneration of heat and power in industrial bakery, milk and whey powder drying and the glass industry was investigated in [4]. Based on the economic analysis, utilizing hot air in these industries was cost effective and the payback time was calculated at three to four years. Allison 571K and GE LM2500 gas turbines were chosen for an ABC analysis in [8] resulting in an increase of more than

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18% in overall efficiency compared to an open gas turbine cycle.

The effect of intercooling in ABC was studied in [5]. The results showed an increase in the efficiency of the cycle with intercoolers. Economic analysis of ABC was reported in [3]. The economic estimation showed that the investment expenditure of ABC is much lower than a conventional combined cycle power plant. A complete description of ABC and the design procedure for various components of ABC are given in [9]. Cost analysis of a 22 MWe ABC was also carried out with a capital investment estimate of US$9.4 million. It was concluded that the use of ABC is economic.

A hybrid solar ABC was studied in [1] by comparing its performance with conventional ABC and SBC for small unit power plants. It was concluded that SBC has the highest thermal efficiency whereas solar ABC has the lowest. In [10] ABC was coupled with a hybrid solar gas turbine. A solar tower was selected to heat the air to 1223 K prior to the combustion chamber. Multi-objective optimization was employed to reduce CO_2 emissions and at the same time minimize cost. The results showed an increase in cycle efficiency and significant reduction in CO_2 emissions and levelled the cost of the solar gas turbine with ABC compared to a simple gas turbine station.

The result of steam injection in ABC and an increase in cycle efficiency was reported in [11]. The effect of compression ratio and TIT on ABC performance was analyzed in [12]. The optimum design point for the ABC plant was found to be: a topping cycle pressure ratio of 10 and a bottoming cycle pressure ratio of 2 with TIT at 1673 K. The effect of gas flow and TIT on the efficiency of ABC was investigated in [13]. An increase in efficiency with higher TIT and lower bottoming cycle compression ratio was reported.

The integration of a reversed Brayton cycle into ABC with power, heat and cooling output was studied in [14]. A comparison between ABC and the organic Rankine cycle (ORC) was conducted in [15]. Both ABC and ORC are considered to be competitive for small scale power stations. It was reported that ORC is superior in the cogeneration of heat and power at temperature levels up to 823 K, whereas ABC performs better at higher temperatures.

In this work, a thermo–economic optimization analysis is performed on two ABC configurations with and without intercooler in the bottoming cycle. In particular, thermo–economic optimization modeling is developed and the effect of the mass flow rate ratio of bottoming cycle air mass flow rate with respect to the topping cycle air mass flow rate is examined in terms of both ABC plant efficiency and total operation cost.

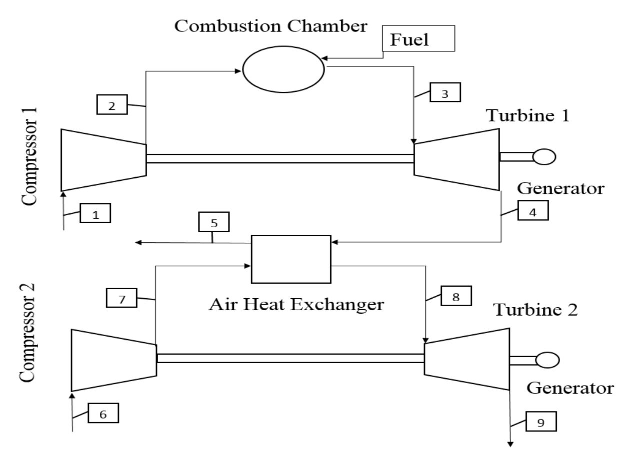

Figure 1: ABC without intercooling integration in the bottoming cycle

In section 2 the two ABC configurations examined are presented. In section 3 the thermo–economic optimization modeling developed for this investigation is presented and the results of the analysis are discussed in section 4. The conclusions are summarized in section 5.

2. ABC configurations

In the present paper, two different ABC configurations were investigated: (a) ABC without intercooling integration in the bottoming cycle and (b) ABC with intercooling integration in the bottoming cycle.

Referring to Fig. 1, the case of no intercooling integration in the bottoming cycle, ambient air is drawn at state 1 into the topping cycle compressor after passing through a filter. Air is compressed from state 1 to state 2 adiabatically in the compressor. After compression, air enters the combustion chamber at state 2 and heat is added to the air fuel mixture in an isobaric process up to state 3. At the final stage of the topping gas turbine, the exhaust gases from the combustion chamber are expanded adiabatically from state 3 to state 4 in the gas turbine to produce mechanical work, which will be converted into electricity in the generator. In a simple gas turbine, exhaust gases are released to the atmosphere, but in the case of ABC, the exhaust gases enter a heat exchanger at state 4 and exchange heat for the requirements of bottoming cycle air heating. The exhaust gases are released to the atmosphere at state 5. In the bottoming cycle ambient air enters the compressor at state 6 and is compressed adiabatically from state 6 to state 7. Then the compressed air enters the heat exchanger at state 7 recovering a portion of heat from the

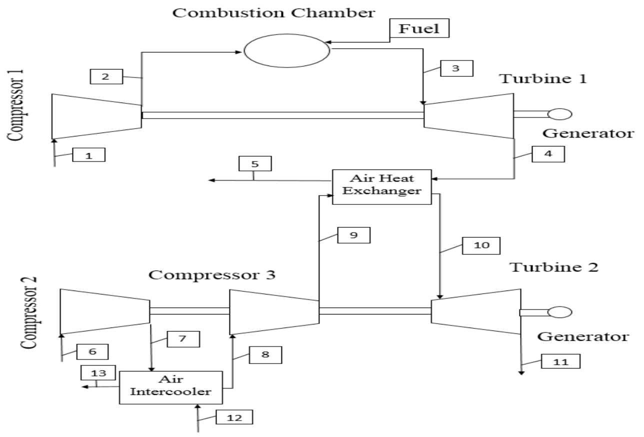

Figure 2: ABC with intercooling integration in the bottoming cycle

topping cycle turbine exhaust gases and exiting the heat exchanger at state 8. The hot air is expanded in the turbine from state 8 to state 9, which consequently enhances cycle performance.

Referring to Fig. 2, the case of intercooling integration in the bottoming cycle, the hot compressed air at state 7 is cooled in an intercooler before entering the compressor second stage at state 8. Using intercooling in the bottoming cycle the work consumed by the compressor is less, leading to lower back-work ratio and consequently higher overall cycle efficiency.

3. Thermo–economic modeling of ABC

This section presents the thermo–economic modeling developed for the analysis of ABC technology without or with intercooling integration in the bottoming cycle. First the thermodynamic modeling is discussed and then the economic modeling is illustrated. The formulations developed were implemented in MATLAB code for the purposes of the simulations.

3.1. Thermodynamic analysis

For this analysis ISO conditions are used with the relevant specific heat capacities evaluated at cold air standard conditions. All thermodynamic assumptions, design parameters and constraints used are tabulated in Table 1. The ABC technology consists of two compressors, one for the topping cycle and the other for the bottoming cycle. The topping cycle compressor outlet temperature and specific work input are given by:

$$T_{O,cT} = T_{i,cT}\left[1 + \frac{1}{\eta_{cT}}\left(r_{cT}^{\frac{\gamma_a-1}{\gamma_a}} - 1\right)\right] \tag{1}$$

Table 1: Data, assumptions, initial conditions and constraints

Parameter	Assigned value
Fuel low heating value	50000 kJ/kg
Reference temperature	298 K
Combustion chamber efficiency	98%
Combustion chamber pressure drop	2%
Turbine inlet temperature	1500K
Topping cycle pressure ratio	14
Bottoming cycle pressure ratio	4
Net power output	50 MWe
Compressor isentropic efficiency	85%
Turbine isentropic efficiency	87%
Heat exchanger pressure drop	2%
Heat exchanger pinch temperature	9 K
Intercooler pressure drop	2%
Intercooler outlet temperature	313 K

$$w_{cT} = \frac{c_{pa}T_{i,cT}}{\eta_{cT}}\left(r_{cT}^{\frac{\gamma_a-1}{\gamma_a}} - 1\right) \tag{2}$$

where$T_{i,cT}$ and $T_{o,cT}$ are the inlet and the outlet temperatures to and from the topping cycle compressor respectively in K, w_{cT} is the topping cycle compressor input specific work in kJ/kg, r_{cT} is the topping cycle compression pressure ratio in %, η_{cT} is the topping cycle compressor isentropic efficiency in %, c_{pa} is the air specific heat capacity in kJ/kgK and γ_a is the air specific heat ratio. Similarly, for the bottoming cycle compressor the outlet temperature and specific work input are given by:

$$T_{O,cB} = T_{i,cB}\left[1 + \frac{1}{\eta_{cB}}\left(r_{cB}^{\frac{\gamma_a-1}{\gamma_a}} - 1\right)\right] \tag{3}$$

$$w_{cB} = \frac{c_{pa}T_{i,cB}}{\eta_{cB}}\left(r_{cB}^{\frac{\gamma_a-1}{\gamma_a}} - 1\right) \tag{4}$$

where $T_{i,cB}$ and $T_{o,cB}$ are the inlet and the outlet temperatures to and from the bottoming cycle compressor respectively in K, w_{cB} is the bottoming cycle compressor input specific work in kJ/kg, r_{cB} is the bottoming cycle compression pressure ratio in %, η_{cB} is the bottoming cycle compressor isentropic efficiency in %, c_{pa} is the air specific heat capacity in kJ/kgK and γ_a is the air specific heat ratio.

The outlet temperature and specific work output from the topping cycle turbine can be determined from:

$$T_{O,tT} = T_{i,tT}\left[1 + \frac{1}{\eta_{tT}}\left(r_{tT}^{\frac{\gamma_g-1}{\gamma_g}} - 1\right)\right] \tag{5}$$

$$w_{tT} = c_{pg}T_{i,tT}\eta_{tT}\left(r_{tT}^{\frac{\gamma_g-1}{\gamma_g}} - 1\right) \tag{6}$$

where $T_{i,tT}$ and $T_{o,tT}$ are the inlet and the outlet temperatures to and from the topping cycle turbine respectively in K, w_{tT} is the topping cycle turbine output specific work in kJ/kg, r_{tT} is the topping cycle expansion pressure ratio in %, η_{tT} is the topping cycle turbine isentropic efficiency in %, c_{pg} is the exhaust gases specific heat capacity in kJ/kgK and γ_g is the exhaust gases specific heat ratio. Similarly, the outlet temperature and specific work output from the bottoming cycle turbine can be determined from:

$$T_{O,tB} = T_{i,tB}\left[1 + \frac{1}{\eta_{tB}}\left(r_{tB}^{\frac{\gamma_a-1}{\gamma_a}} - 1\right)\right] \tag{7}$$

$$w_{tB} = c_{pa}T_{i,tB}\eta_{tB}\left(r_{tB}^{\frac{\gamma_a-1}{\gamma_a}} - 1\right) \tag{8}$$

where $T_{i,tB}$ and $T_{o,tB}$ are the inlet and the outlet temperatures to and from the bottoming cycle turbine respectively in K, w_{tB} is the bottoming cycle turbine output specific work in kJ/kg, r_{tB} is the bottoming cycle expansion pressure ratio in % and η_{tB} is the bottoming cycle turbine isentropic efficiency in %.

For the topping cycle, the fuel and exhaust gases mass flowrates can be evaluated by:

$$\dot{m}_f = \frac{\dot{m}_{aT}c_{pg}\left(T_{o,cc} - T_r\right) - \dot{m}_{aT}c_{pa}\left(T_{i,cc} - T_r\right)}{\eta_{cc}LHV - c_{pg}\left(T_{o,cc} - T_r\right)} \tag{9}$$

$$\dot{m}_g = \dot{m}_{aT} + \dot{m}_f \tag{10}$$

where $\dot{m}_f$ is the topping cycle fuel mass flowrate in kg/s, $\dot{m}_{aT}$ is the topping cycle air mass flowrate in kg/s, $T_{i,cc}$ and $T_{o,cc}$ are the inlet and the outlet temperatures to and from the combustion chamber respectively in K, $\dot{m}_g$ is the exhaust gasses mass flowrate in kg/s, η_{cc} is the combustion chamber efficiency in %, LHV is the fuel low heating value in kJ/kg and T_r is reference temperature state associated with the fuel LHV in K.

The basic criteria in a heat exchanger optimization are high effectiveness and minimum pressure drop. Satisfying these criteria simultaneously is complicated, because of the low heat transfer coefficient and density of both fluids involved in the heat transfer process [10]. Moreover, high temperature gases and the high pressure difference between the two streams add more complexity to the heat exchanger optimization process.

The formulation of the air-to-air counter flow heat exchanger is carried out using pinch analysis. For a counter flow heat exchanger, if the heat capacitance of the hot stream is greater than the heat capacitance of the cold stream, the pinch point is located at the hot stream entry and the cold stream exit, where the minimum temperature difference between the hot and cold streams occurs. In this case the hot and cold streams exit temperature can be computed by:

$$T_{o,cs} = T_{i,hs} - \triangle T_{pinch} \tag{11}$$

$$T_{o,hs} = T_{i,hs} - \frac{\dot{m}_{aB}c_{pa}\left(T_{o,cs} - T_{i,cs}\right)}{\dot{m}_g c_{pg}} \tag{12}$$

however, if the heat capacitance of the cold stream is greater then,

$$T_{o,hs} = T_{i,cs} - \triangle T_{pinch} \tag{13}$$

$$T_{o,cs} = T_{i,cs} - \frac{\dot{m}_g c_{pg}\left(T_{o,hs} - T_{i,hs}\right)}{\dot{m}_{aB}c_{pa}} \tag{14}$$

where $T_{i,hs}$, $T_{o,hs}$, $T_{i,cs}$ and $T_{o,cs}$ are the hot and the cold fluid streams inlet and exit temperatures in K, ΔT_{pinch} is the designed pinch temperature difference in K and $\dot{m}_{aB}$ is the bottoming cycle air mass flowrate in kg/s.

Total net power output and overall efficiency of ABC can then be calculated by:

$$\dot{W}_{net} = \left(\dot{m}_g w_{tT} - \dot{m}_{aT}w_{cT}\right) + \left(\dot{m}_{aB}w_{tB} - \dot{m}_{aB}w_{cB}\right) \tag{15}$$

$$\eta = \frac{\dot{W}_{net}}{\dot{m}_f LHV} \tag{16}$$

where w_{tT} and w_{cT} are the topping cycle specific works of turbine and compressor respectively in kJ/kg, likewise, w_{tB} and w_{cB} are the bottoming cycle specific works of turbine and compressor respectively in kJ/kg, $\dot{W}_{net}$ is the total net power output in kW and η is the overall efficiency in %. Also, the mass flow rate ratio ($MFRR$) is defined as:

$$MFRR = \frac{\dot{m_{aB}}}{\dot{m}_{aT}} \tag{17}$$

therefore $\dot{m}_{aT}$ can be determined from Eq. (18).

$$\dot{m}_{aT} = \frac{\dot{W}_{net}\left[\eta_{cc}LHV - c_{pg}\left(T_{o,cc} - T_r\right)\right]}{w_{tT}\left[c_{pg}\left(T_{o,cc} - T_r\right) - c_{pa}\left(T_{i,cc} - T_r\right)\right] + \left(w_{netB}MFRR + w_{netT}\right)\left[\eta_{cc}LHV - c_{pg}\left(T_{o,C} - T_r\right)\right]} \tag{18}$$

where: w_{netT} and w_{netB} are the specific net works from the topping and bottoming cycles respectively, kJ/kg

3.2. Economic optimization

It is widely known that the most efficient cycle configuration may not be the most economic alternative. Therefore in this work cost estimation and economic analysis of the ABC technology is carried out as well. The objective function for capital and operating costs [16], which are minimized by varying the design variables, is given by:

$$C_{Total} = \dot{m}_f c_f LHV + \sum_{i=1}^{n} \frac{Z_i CRF \varphi}{3600N} \tag{19}$$

where C_{Total} is the total operation cost of the power plant in US\$/s, c_f is the fuel cost per energy unit in US\$/GJ (assumed to be 5.5 US\$/GJ for this analysis), CRF is capital recovery factor in % (assumed to be 19.1% for this analysis), φ is maintenance factor (assumed to be 1.03 for this analysis) and N is the annual number of hours of operation of the plant (assumed to be 8110 h/year for this analysis). $Z_i, i = 1, 2, \ldots, n$, is the capital investment cost of each component constituting the ABC technology under investigation, in US\$, as described by equations (20) - (26) below [16].

For the topping and bottoming cycles compressors the capital investment cost, Z_{cT} and Z_{cB}, respectively, are evaluated by:

$$Z_{cT} = \frac{39.5 r_{cT} \dot{m}_{aT}}{0.9 - \eta_{cT}} \ln(r_{cT}) \tag{20}$$

$$Z_{cB} = \frac{39.5 r_{cB} \dot{m}_{aB}}{0.9 - \eta_{cB}} \ln(r_{cB}) \tag{21}$$

Similarly, the capital investment cost for the topping cycle turbine, Z_{tT}, the topping cycle turbine, Z_{tB}, the combustion chamber Z_{cc}, the air-to-air counter flow heat exchanger, Z_{AHX}, and the bottoming cycle intercooler, Z_{Int}, can be calculated by:

$$Z_{tT} = \frac{266.3 \dot{m}_g}{0.92 - \eta_{tT}} \ln(r_{tT}) \left[1 + e^{(0.036 T_{i,tT} - 54.4)}\right] \tag{22}$$

$$Z_{tB} = \frac{266.3 \dot{m}_{aB}}{0.92 - \eta_{tB}} \ln(r_{tB}) \left[1 + e^{(0.036 T_{i,tB} - 54.4)}\right] \tag{23}$$

$$Z_{cc} = \frac{266.3 \dot{m}_{aT}}{0.995 - \left(\frac{P_{o,cc}}{Pi,cc}\right)} \left[1 + e^{(0.018 T_{o,cc} - 26.4)}\right] \tag{24}$$

$$Z_{AHX} = 2290 \left[\frac{\dot{m}_g c_{pg} \left(T_{i,hs} - T_{o,hs}\right)}{0.018 LMTD_{AHX}}\right]^{0.6} \tag{25}$$

$$Z_{Int} = 2290 \left[\frac{\dot{m}_{aB} c_{pa} \left(T_{i,hs} - T_{o,hs}\right)}{0.018 LMTD_{Int}}\right]^{0.6} \tag{26}$$

where $P_{i,cc}$ and $P_{o,cc}$ are the combustion chamber inlet and outlet pressures in kPa, $LMTD_{AHX}$ is the log mean temperature difference in the air heat exchanger in K and $LMTD_{Int}$ is the log mean temperature difference in the bottoming cycle intercooler in K.

3.3. Optimization procedure

A flowchart of the optimization algorithm developed and implemented in MATLAB code [17] in the case of the analysis of an ABC technology without intercooling integration in the bottoming cycle is illustrated in Fig. 3. Referring to both Fig. 1 and Fig. 3, the topping cycle compressor outlet temperature at state 2 and the compressor specific work are calculated based on the ambient air temperature entering the compressor and the cycle compression ratio. Then, the topping cycle turbine outlet temperature at state 4 is calculated as well as the respective specific work based on the TIT at state 3. The bottoming cycle compressor outlet temperature at state 7 and the compressor specific work are determined based on the ambient air temperature entering the compressor and the cycle compression ratio. The *MFRR* ratio is set at zero during the first iteration. For the air heat exchanger analysis, the location of the pinch point is determined by comparing the heat capacitance of the hot and cold fluid. Depending on the pinch point location, the outlet temperature of the air at state 8 and the exhaust gas temperature at state 5 from the air heat exchanger are calculated. Then both the temperature at state 9 and the bottoming cycle turbine specific work are computed. Air mass flow rates for topping and bottoming cycles, the fuel mass flow rate and the exhaust gas mass flow rate are then determined.

Finally, the economic analysis is carried out. The capital investment cost of each component is calculated and the total operation cost of the power plant is determined. The new total operation cost value is compared to the minimum total operation cost calculated during the previous steps, and the minimum value between the two is selected as the new minimum total operation cost. The MFRR ratio is then increased by a step of 0.01 and the iterations are

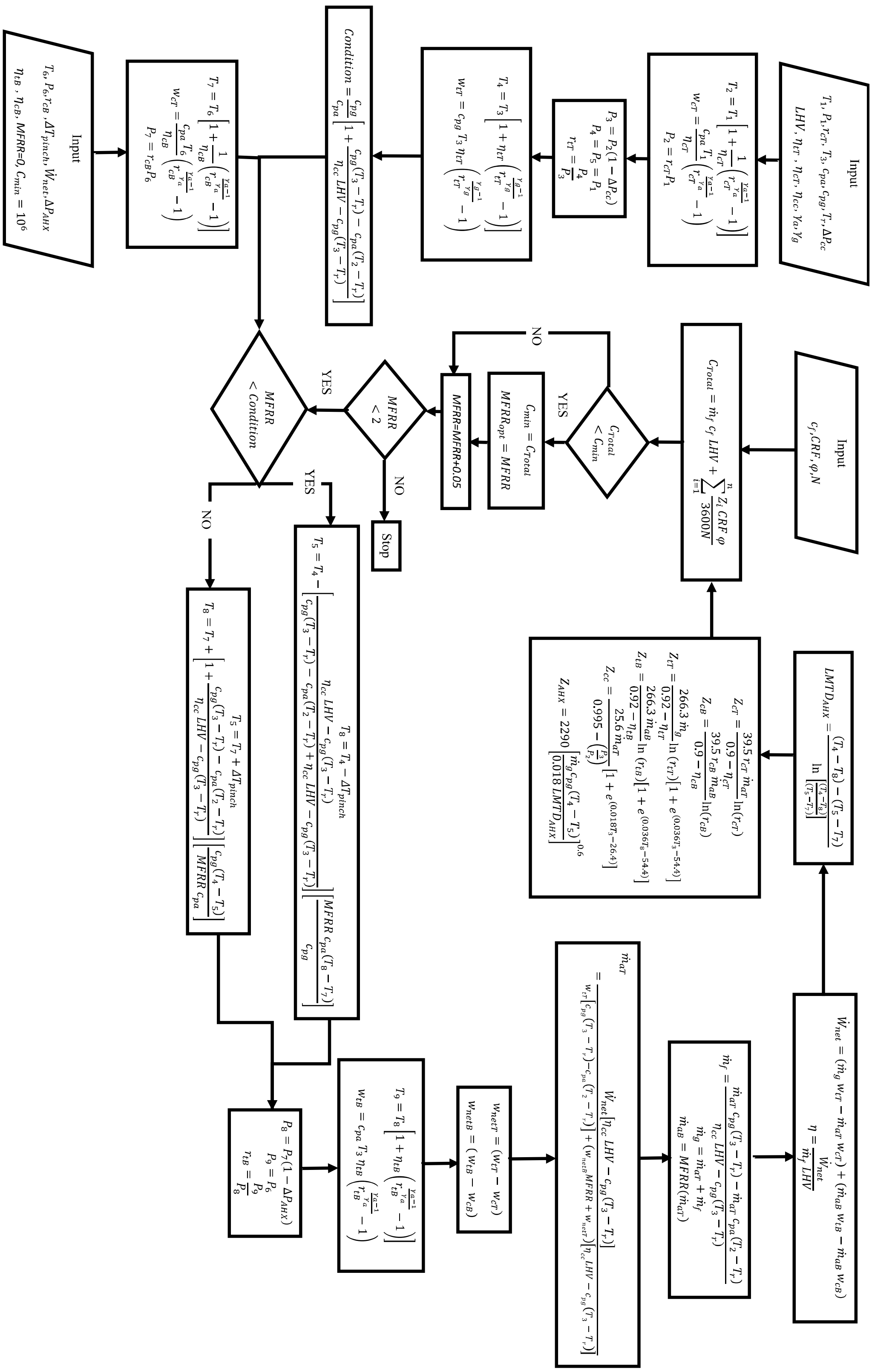

Figure 3: Optimization algorithm flowchart for the analysis of ABC technology without intercooling integration in the bottoming cycle

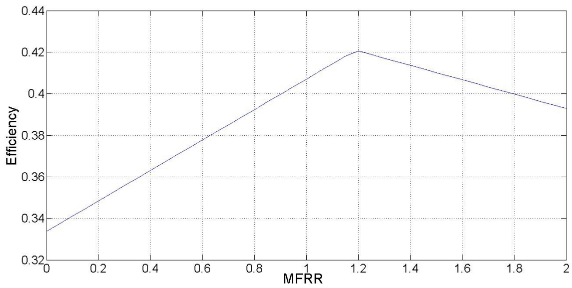

Figure 4: Effect of MFRR ratio on efficiency (r_{cT}=14, r_{cB}=4, TIT=1500 K)

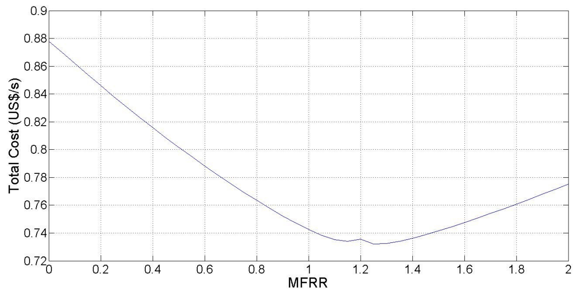

Figure 5: Effect of MFRR ratio on total operation cost (r_{cT} = 14, r_{cB} = 4, TIT=1500 K)

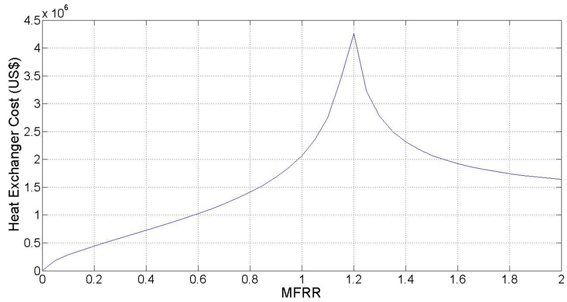

Figure 6: Effect of MFRR ratio on air heat exchanger cost (r_{cT} = 14, r_{cB} = 4, TIT=1500 K)

repeated until the MFRR ratio reaches the value of 2. The final output provides the design parameters in the case of the minimum total operation cost.

For the optimization of the second case examined, i.e., ABC technology with intercooling integration in the bottoming cycle, see Fig. 2, a similar algorithm was developed.

4. Results and discussion

The bottoming cycle turbine is sensitive to changes in the air mass flow rate and pressure ratio [9]. The air mass flow rate for the bottoming cycle can be varied to enhance cycle overall efficiency and performance. However, based on the available literature most of the studies implemented on ABC technology have taken the value of the bottoming cycle air mass flow rate to be equal to the topping cycle air mass flow rate [11]. Further, in [9] the mass flow rates were selected in such a way that the product of the mass flow rate and the specific heat capacity for both the air and the exhaust gases were equal to unity. Thus, a smaller heat exchanger can be utilized for a given power output [9]. Moreover, the required heat transfer area is minimized [14]. Since exhaust gases specific heat capacity is higher than air specific heat capacity, the bottoming cycle air mass flow rate must then be greater than the gas mass flow rate. Consequently, the $MFRR$ ratio must be greater than unity to achieve optimum conditions and better heat exchanger performance.

The results concerning the effect of the MFRR ratio on overall efficiency of the ABC plant without intercooling integration in the bottoming cycle are illustrated in Fig. 4. Increasing the $MFRR$ ratio improves cycle efficiency. The optimum value for the $MFRR$ ratio is 1.19, which results in the highest overall efficiency of 42.09%. Efficiency decreases for values of the $MFRR$ ratio greater than 1.19. By setting the MFRR ratio to zero, a simple gas turbine cycle is modeled with overall efficiency of 33.37%.

Thermodynamic analysis of ABC mostly concentrates on efficiency as the criterion for optimization, whereas it is crucial to study the effect of the MFRR ratio on the total production cost of the ABC plant to determine the most cost effective design. The results concerning the effect of the MFRR ratio on the total production cost of the ABC plant without intercooling integration in the bottoming cycle are illustrated in Fig. 5. The addition of a bottoming cycle is an economic choice, since the cycle total cost falls from 0.8779 $/s (at MFRR=0) to a minimum of 0.7319 $/s at an MFRR ratio of 1.27. At an MFRR ratio of 1.2 a slight increase in total operation cost is observed. The explanation for this is provided in Fig. 6, where the cost for the air heat exchanger at an MFRR ratio of 1.19 is at a maximum; thus the extra cost of the air heat exchanger outweighs the cost reduction in the cycle.

The results concerning the effect of the compression ratio on overall efficiency and total operation cost of the ABC plant without intercooling integration in the bottoming cycle are presented in Fig. 7 and Fig. 8 respectively. Maximum efficiency of 42.34% is obtained for a topping cycle pressure ratio of 20.4 with a bottoming cycle pressure ratio of 4. However, in the case of total production cost, the minimum cost of 0.7318 $/s is obtained for a topping cycle pressure ratio of 13.5 with a bottoming cycle pressure ratio of 4.

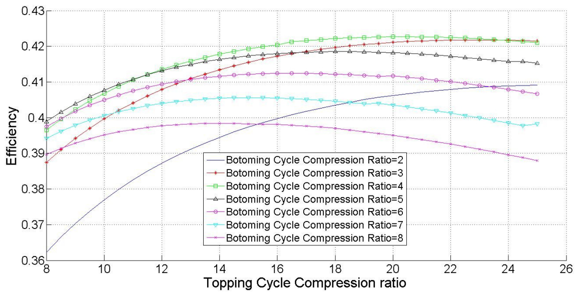

Figure 7: Effect of compression ratio on overall efficiency (TIT=1500 K, optimum MFRR ratio)

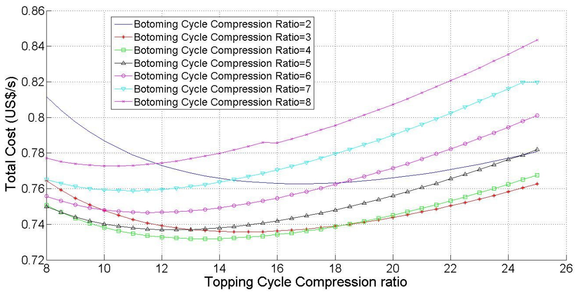

Figure 8: Effect of compression ratio on total operation cost (TIT=1500 K, optimum MFRR ratio)

Another key factor in thermo–economic optimization is TIT. The performance of ABC is extremely sensitive to changes in TIT [14]. It was stated that TIT can become as high as 1561 K and higher values can be achieved by implementing advanced blade material and cooling technology in [12]. TIT of 1673 K for the analysis of an ABC system was selected in [11] and as provided in [10] this corresponds to the combustion chamber outlet temperature in modern gas turbine power plants, which is considerably greater than the maximum turbine blade tolerance. Therefore, exhaust gases are cooled down by a cooling air stream from the compressor during expansion in the turbine [10].

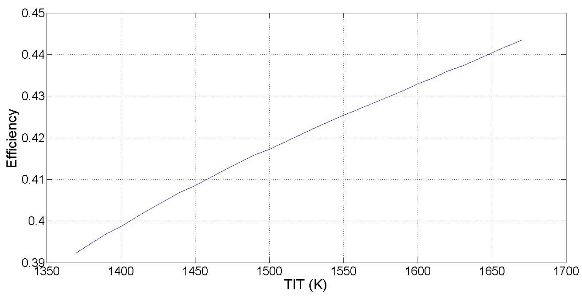

Figure 9: Effect of TIT on overall efficiency (r_{cT} = 13.5, r_{cB} = 4, optimum MFRR)

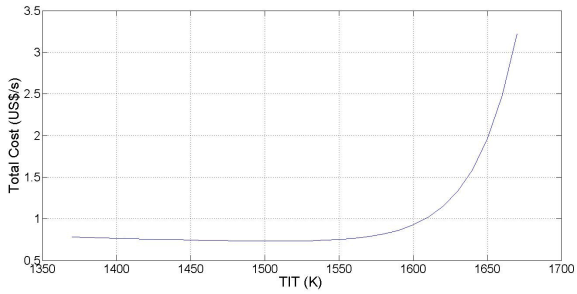

Figure 10: Effect of TIT on total operation cost (r_{cT} = 13.5, r_{cB} = 4, optimum MFRR)

The results concerning the effect of TIT on the overall efficiency of the ABC plant without intercooling integration in the bottoming cycle are illustrated in Fig. 9. As expected higher TIT values lead to the ABC system achieving higher efficiency. For a TIT value of 1670 K overall efficiency is 44.34%. However, we need to investigate whether an increase in TIT would justify the extra investment. Thus, the effect of TIT on the total operation cost of the ABC plant, without intercooling integration in the bottoming cycle, are presented in Fig. 10. Minimum total operation cost of 0.7316 $/s is achieved at the TIT value of 1510 K. For TIT values greater than 1600 K total operation cost rises enormously, mainly owing to the large increase in investment costs for the topping cycle turbine and combustion chamber.

The optimum results obtained in the case of the ABC plant without intercooling integration in the bottoming cycle are summarized in Table 2. Optimum overall efficiency is 41.89% and the overall heat rate is 8593.3 kJ/kWh, with the efficiency of the bottoming cycle being 18.76%. The power output of the bottoming cycle is 10.336 MWe, which is 20.67% of the total power output of the ABC plant. The optimum total operation cost is 0.7316 $/s, with the main cost related to the topping cycle investment.

Similar results were obtained in the case of the thermo–economic analysis of the ABC plant with intercooling integration in the bottoming cycle. However, as expected, both overall efficiency and total operation cost are improved compared to the no intercooling integration case. The optimum results obtained in the case of the ABC plant without intercooling integration in the bottoming cycle are summarized in Table 3. Optimum overall efficiency is 43.17% and the overall heat rate is 8319.2 kJ/kWh, with the efficiency of the bottoming cycle being 18.98%. The power output of the bottoming cycle is 11.495 MWe, which is 22.81% of the total power output of the ABC

Table 2: Optimum results for the case of the ABC plant without intercooling integration in the bottoming cycle

Parameter	Optimum result
Topping cycle:	
Pressure ratio	13.5
Turbine inlet temperature	1510 K
Air mass flowrate	105.0402kg/s
Fuel mass flowrate	2.3870 kg/s
Exhaust gases temperature	461.6663 K
Compressor cost	US$M2.9756
Turbine cost	US$M2.9752
Combustion chamber cost	US$M0.58604
Bottoming cycle:	
Pressure ratio	4
Turbine inlet temperature	876.9516 K
Air mass flowrate	137.0745 kg/s
Available heat recovery	55.090 MWth
Net power output	10.336 MWe
Overall efficiency	18.76%
Compressor cost	US$M0.60048
Turbine cost	US$M0.99733
ABC:	
MFRR	1.27
Air heat exchanger cost	US$M3.0061
Net power output	50 MWe
Overall efficiency	41.89%
Heat rate	8593.3 kJ/kWh
Total operation cost	0.7316 US$/s

plant. The optimum total operation cost is 0.7273 $/s, with the main cost related to the topping cycle investment.

5. Conclusions

In this work a thermo–economic optimization analysis was performed on two air bottoming cycle (ABC) configurations with and without intercooler in the bottoming cycle. Thermo–economic optimization modeling was developed and the effect of the mass flow rate ratio of bottoming cycle air mass flow rate with respect to the topping cycle air mass flow rate was examined in terms of both ABC plant efficiency and total operation cost. For a 50 MW capacity ABC plant the optimum results indicate that in the case of an ABC plant without intercooling integration in the bottoming cycle overall efficiency is 41.89% and total operation cost is 0.7316 $/s. In the case of a 50 MW ABC plant with intercooling integration in the bottoming cycle, overall efficiency increases to 43.17% and total operation cost drops to 0.7273 $/s.

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Table 3: Optimum results for the case of the ABC plant with intercooling integration in the bottoming cycle

Parameter	Optimum result
Topping cycle:	
Pressure ratio	13.7
Turbine inlet temperature	1510 K
Air mass flowrate	104.7905 kg/s
Fuel mass flowrate	2.3109 kg/s
Exhaust gas temperature	402.6486 K
Compressor cost	US$M 2.9685
Turbine cost	US$M 2.9049
Combustion Chamber Cost	US$M 0.5690
Bottoming cycle:	
First stage compressor pressure ratio	2.7
Second stage compressor pressure ratio	2
Turbine inlet temperature	857.4059 K
Air mass flowrate	129.9403 kg/s
Available heat recovery	60.562 MWth
Net power output	11.495 MWe
Overall efficiency	18.98%
First and second stage compressors cost	US$M 0.4176
Air intercooler cost	US$M 1.6342
Turbine cost	US$M 1.1391
ABC:	
MFRR	1.24
Air heat exchanger cost	US$M 3.4574
Net power output	50 MWe
Overall efficiency	43.27%
Heat rate	8319.2 kJ/kWh
Total operation cost	0.7237 US$/s