AIR BOTTOMING CYCLE FOR COGENERATION OF POWER, HEAT AND COOLING

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ABSTRACT

The Air Bottoming Cycle (ABC) is an economical concept for increasing the power generating efficiency of small and medium-scale gas turbines. The ABC is a Brayton cycle that utilises the exhaust heat from the topping gas turbine via a heat exchanger. As output, power is yielded, as well as heat from optional intercooling and in the form of exhausted hot air. In this paper, a thermodynamic analysis is presented for a cogenerative system where a Reversed Brayton Cycle (RBC) is integrated into an intercooled ABC to provide cold airflow. System optimisation is discussed and characteristics are presented for power, heat and cooling output for the selected configuration. For the ABC and RBC, the sensitivity of the performance is presented against the main cycle parameters.

KEYWORDS

Air bottoming cycle, Reversed Brayton cycle, Gas turbine

1. INTRODUCTION

Combined-cycle technology is a frequently used concept for increasing the generating efficiency of energy systems. As the operation of all heat engines is based on higher-temperature heat addition and lower-temperature heat rejection, the rejected heat from the topping cycle can be implemented as a source for the bottoming one. The Air Bottoming Cycle (ABC) is a Brayton cycle that utilises the exhaust heat from a topping engine (gas turbine in this study). The heat to the ABC is introduced by using a heat exchanger and consequently the working fluid in the ABC is air. As output, power is yielded, as well as heat from the optional intercooling and in the form of exhausted hot air. The integration of a Reversed Brayton Cycle (RBC) into the ABC provides cold airflow. The emerging concept is especially suitable for small-scale distributed energy conversion with the need for all three products: power, heat and cooling.

From the cycle perspective, the ABC is a Brayton cycle with an external heat source, similarly to externally fired gas turbine (EFGT) cycles. The difference between these is that the heat source for EFGT is the exothermic reaction itself, whereas for the ABC the heat source is the rejected heat from the topping cycle. The configuration where the ABC recuperates heat from topping gas turbine cycles is also referred to as a dual gas turbine combined cycle (DGTCC) [1].

Heat for the ABC is transferred from the topping cycle exhaust flow to the compressor discharge air of the ABC. To maximise the heat recovery, it is favourable to have a high temperature difference between these flows and thus the optimum compression would be isothermal. In practice, this contributes to the use of intercooling between the compressor sections. When the RBC is integrated into the ABC, a fraction of the compressed air will be extracted, cooled down and let to expand to lower pressure and temperature. Depending on the available outlet temperature level in the air cooler and the required temperature for the cold air flow, the extraction can be implemented using a bleed or between compressor sections. Figure 1 shows as an example a schematic layout of a dual gas turbine combined cycle where the ABC has one intercooler and the RBC uses a bleed from the low-pressure compressor.

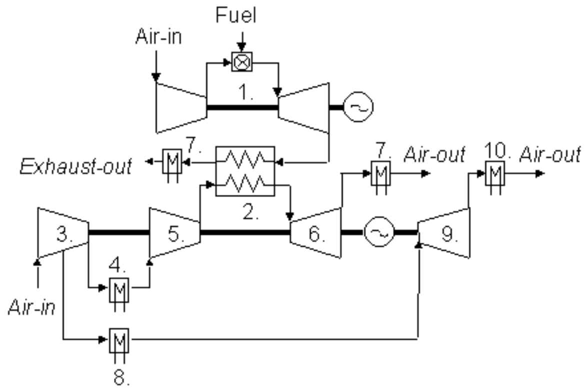

Figure 1: Cycle layout of dual gas turbine combined cycle with RBC integrated into ABC. 1. Standard gas turbine 2. Recuperator 3. Low-pressure compressor 4. Intercooler (optional heating duty) 5. High-pressure compressor 6. Hot-air turbine 7. Heating duty 8. Air cooler 9. Cold-air turbine 10. Cooling duty.

The ABC has been described and patented by Farrell [2]. Wicks [3] has derived the concept of the ABC from the theory of an ideal fuel-burning engine by comparing it with the Carnot cycle. During the last decade, several performance analyses on the cycle have been presented, for example by Weston [1], Hirs et al. [4], and Arriagada et al. [5]. A study by Kaikko [6] has demonstrated the competitiveness of the ABC against the steam Rankine cycle in small and medium-scale applications. A comparison by Kaikko et al. [7] between the ABC and the organic Rankine cycle (ORC) as small-scale bottoming cycles has indicated the potential of the ABC especially for cogeneration of power and heat, even with diesel engines as topping cycles.

The ABC has been proposed to increase the efficiency of the simple-cycle gas turbine units on Norwegian oil platforms, motivated by CO_2-tax introduction [8]. Since 1997, this concept has been developed by Kværner Energy in Norway [9]. The application of the ABC on the platform is expected to give 25 % reduction in CO_2 and NO_x emissions and fuel consumption for a given plant output. In the Netherlands, the ABC is being considered for industrial implementation in two projects: as a hot-air cogeneration plant and as a heat recovery unit of an industrial furnace. On the basis of the results, two demonstration projects are scheduled for realisation [10].

2. MODELLING BASIS AND SYSTEM OPTIMISATION

In this study, the heat from the dual gas turbine combined cycle is implemented as district heat while the cooling duty is used for direct space cooling (refrigeration). All the calculations are based on the ISO ambient conditions (15 °C , 101.325 kPa , 60 %) and nominal performance. Hence, no inlet filter or exhaust duct pressure losses are taken into account for the topping or the bottoming cycle. The reference temperature for the cooling rate calculations is 20 °C, a temperature level that is typical for human applications.

Alstom Tempest has been selected as the topping gas turbine for the cycle analysis. Its nominal specifications for natural gas as fuel are given in Table 1. Considering the power range, the efficiency of the engine is high, and the exhaust gas temperature is at a high level. Both these factors contribute to a high power generating efficiency in combined cycle operation.

Table 1: Performance specifications for the topping gas turbine (Alstom Tempest) while operating on natural gas. Data from GateCycle [11]. The CO_2 emission is calculated.

Alstom Tempest		
Net power output	MW	7.75
Net el. efficiency	%	31.4
Exhaust gas flow	kg/s	28.6
Exhaust gas temp.	°C	534
Spec CO_2 emission	g/kWh	669

The specifications for the ABC and RBC have been selected to reflect the current state for advanced turbomachinery in small and medium class using axial equipment. This is also the case for the compact plate-fin recuperator for transferring heat from the topping cycle to the bottoming one. These specifications are presented in Table 2.

Table 2: Computational parameters for ABC and RBC calculations.

Compressor (unit) polytropic efficiency	91 %
Turbine (unit) polytropic efficiency	86 %
Recuperator effectiveness	93 %
Recuperator air-side pr. loss (rel. to inlet)	2 %
Recuperator exhaust gas-side pressure loss	2 %
Intercooler air-side pressure loss	2 %
Air cooler air-side pressure loss	2 %
District heat exchanger air-side pressure loss	2 %
Mechanical efficiency	99.5 %
Gearbox efficiency	98.5 %
Generator efficiency	98 %
Auxiliary power (relative to net power)	1 %

Intercooling in the ABC is implemented separately from the air-cooling in the RBC. In this way, the intercooler can be designed for non-condensing operation to reduce costs, whereas condensation may occur in the air cooler, depending on the pressure ratios and temperature levels to be used. Also, different cooling fluids may be applied to the heat exchangers: for the cogenerative application in this study, the heat from the intercooling is transferred to district heating water while the heat from the air cooler goes into the cooling water and remains unutilised.

Aside from intercooling, other sources for district heat are the topping gas turbine and ABC exhaust flows. The performance of the district heat exchangers is characterised by minimum temperature differences. For these non-contact liquid-gas heat exchangers of the counterflow type, the difference has been selected to be 15 °C while the return and supply temperatures for district heating water are 50 °C and 80 °C, respectively. This allows the utilisation of the heat down to 65 °C, which has also been selected as the minimum temperature for the exhaust gases from natural gas that is virtually sulphur-free. For the air cooler, air outlet temperatures of 10 °C, 20 °C and 30 °C have been used in the calculations to reflect different cooling water temperatures and minimum temperature differences.

To enhance power generation, intercooling could be implemented using the cooling water. The condensation risk at elevated pressures should then be taken into consideration by determining the minimum allowable air temperature in the intercooler by the saturating temperature increased by a temperature margin of, say, 20 °C. The ultimate lower limit for the temperature would be set by the cooling water inlet temperature added by the selected temperature difference. Although the air could be further cooled down by using a second-level intercooler and cooling water even for the current application, this was not considered feasible due to the small temperature decrease available for further cooling.

In the model, the bottoming-cycle flow rate has been determined by setting the heat capacity flows equal at both sides of the recuperator. This contributes to a minimised heat-transfer area for a given heat-transfer rate. For the configurations with multiple compressor units, the pressure ratios of the units have been assumed equal for simplicity. The bleeding mass flow to the RBC has no effect on the determination of the optimum pressure ratios, as this compression work can be regarded to be done by an external compressor unit.

Characteristic for Brayton cycles, the optimum values for the work and efficiency of the ABC are attained at different pressure ratios. The efficiency at which the recuperated heat is converted into work in the ABC is considered irrelevant while power generation (work) is prioritised to gain the highest possible generating efficiency for the combined cycle. Consequently, power generation from the ABC has been selected as the optimisation criterion instead of ABC cycle efficiency for further analysis in this study. For the RBC the optimisation is straightforward: to find the minimum pressure ratio that yields the required cold air flow temperature for the specified temperature after the air cooler and the efficiency for the cold-air turbine.

The modelling software GateCycle [11] has been implemented to simulate the performance of different cycle configurations. Performance data for the topping gas turbine has been here gained from the software database.

In the RBC, possible condensation of water vapour in the air cooler has been taken into account by a module in the GateCycle that separates condensed water (if any) from the air stream that exits the air cooler at specified temperature.

In the turbine, the temperature (and the saturating pressure) decreases more rapidly than the pressure of the water vapour. Consequently, the relative humidity increases in the expansion and part of the water vapour may condense to water or sublime to ice in the RBC turbine [12]. The latent heat that is released in the phase change has an increasing effect on the end temperature of the expansion process. In the modelling, this effect has been taken into account by using an iterative method where the end temperature is determined by adding into the expanding flow an amount of heat that equals to the corresponding latent heat. The amount of water vapour that undergoes phase change has been determined using the difference in the humidity ω (mass ratio of water vapour to dry air) as defined by Eq. (1). The equation has been derived using the Dalton gas mixture model as well as the definitions for relative humidity and mole fractions in the gas phase.

$$\omega = \frac{M_{vap}}{M_{da}} \frac{\varphi\, p_{sat}}{p - \varphi\, p_{sat}} \tag{1}$$

The maximum allowable humidity at the turbine outlet is calculated using a value of 100 % for the relative humidity φ. If the maximum amount is less than the actual humidity at the turbine inlet, part of the water vapour experiences a phase change. The corresponding amount of air mass flow has also been excluded in the determination of the cooling rate.

Although condensation and ice formation have a strong effect on the outlet temperature, the accuracy of the above simplified method is expected to be high. A more elaborate calculation procedure to determine the outlet temperature has been given by Backman [12]. In his method, the outlet temperature for the isentropic expansion is first iterated using changes in the specific entropy. This temperature is then applied to get the corresponding isentropic enthalpy and the actual enthalpy. For a case with 22 °C and 50 % ambient conditions, an RBC pressure ratio of 1.5 and air-cooler temperature level of 8 °C, for instance the difference in outlet temperatures between the simplified method and the method by Backman was only 0.2 °C while the increasing effect of the phase change on the turbine outlet temperature was exceeding 7 °C. The actual outlet temperature was -9.2 °C.

3. CYCLE ANALYSIS AND SENSITIVITY STUDIES

The implementation of intercooling between the compressor units helps to increase the heat transfer in the recuperator by reducing the overall compressor discharge temperature for a given pressure. Furthermore, the intercooling reduces compressor work and increases the available work from the turbine in this way. The optimum number of intercoolers is a case for optimisation with respect to increased heat recovery in the recuperator, exergy losses in the intercoolers, and costs.

Figure 2 presents the net power output of the ABC against the pressure ratio for a different number of intercoolers. The figure applies to a cogenerative case where no cold flow is being produced. As the figure indicates, the optimum pressure ratios for power generation increase with the increasing number of intercoolers, being in this case 4.3 when no intercooling is implemented, 6.2 for the case with 1 intercooler and 7.1 with two intercoolers. The curves become also flatter, which decreases the effect of the pressure ratio on the power output around the optimum. The figure also points out the rapidly damping effect of intercooling due to pressure losses in the intercoolers: implementing the non-intercooled cycle increases the power output by 22.3 % when compared to the topping gas turbine in simple-cycle operation. The first intercooler adds 2.1 %-points more but the second one only 0.6 %-points. Based on this, the configuration with one intercooler and ABC pressure ratio of 6.2 has been selected for further studies. The corresponding optimised ABC mass flow becomes 30.0 kg/s.

The power output from the ABC is penalised when producing a cold air flow with the RBC. This penalising effect has been studied by calculating the power output and cooling rates for different ratios of cold (RBC) flow to the total (ABC+RBC) mass flow. Here, three values have been used for the required cold flow temperature: 0 °C, -15 °C and -30 °C. Figure 3 presents the results for a case where the air-cooler outlet temperature is 20 °C. Changes in the cold air flow production have no effect on the district heat output.

With the specified ambient conditions, the total amount of water vapour that undergoes a phase change in the RBC is a function of cold flow temperature only. However, the temperature level in the air cooler outlet determines how this is divided among the air cooler and the turbine. This is demonstrated in Figure 4 for a case where the cold flow rate equals the optimised ABC mass flow rate (30.0 kg/s). At high enough air-cooler outlet temperatures, such as 30 °C in this study, no condensation occurs in the air cooler with the used cold flow temperatures. Consequently, condensation or sublimation to ice in the turbine increases with a decreasing cold flow temperature (and increasing pressure ratio). At low enough air-cooler outlet temperatures, such as 10 °C here, condensation occurs in the air cooler throughout the range of the cold flow temperatures. It increases with a decreasing cold flow temperature, decreasing correspondingly the amount of condensing or subliming mass flow in the turbine.

Figures 5a and 5b present the cold flow temperature and net coefficient of performance against the RBC pressure ratio for air-cooler outlet temperatures of 10 °C, 20 °C and 30 °C. The coefficient of performance (*COP*) is defined as a ratio of produced cooling rate to required power. As the figures indicate, the air-cooler outlet temperature has a strong impact on the RBC performance. Lowering this temperature not only lowers the cold flow temperature but also improves the *COP* for a specified pressure ratio, due to the increased cooling rate. On the other hand, a lower pressure ratio is required for a specified cold flow temperature when lower air-cooler outlet temperatures are being applied. This reduced pressure ratio will then further benefit the *COP* which is already more favourable towards lower air-cooler outlet temperatures. Figure 5c summarises the resulting impact on cycle performance and also presents the sensitivity of RBC performance against cold flow temperature. As an example, -15 °C cold air flow can be produced with a *COP* of 1.03 when the air-cooler outlet temperature of 30 °C is available, but with 1.85 if the air at the air-cooler outlet can be cooled down to 10 °C. Consequently, the lowest possible air-cooler outlet temperatures should be applied for favourable operation.

As a summary, Table 3 presents the performance of the specified dual gas turbine combined cycle with the RBC integrated into the ABC. The ABC has one intercooler and optimised pressure ratio and mass flow rate. An equal mass flow rate has been selected for the RBC. The air-cooler outlet temperature is 20 °C and the required cold flow temperature -15 °C.

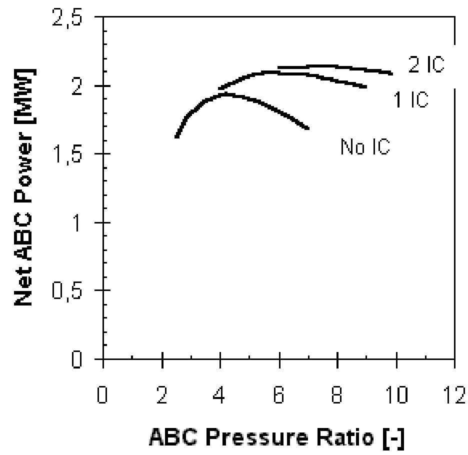

Figure 2: Impact of intercooling on net ABC power output. The curves represent the performance ratings of the bottoming cycle when no cold flow is being produced.

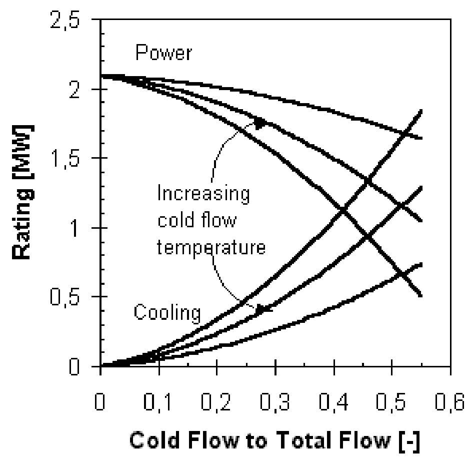

Figure 3: Impact of cold airflow production on net ABC power output. The three values for the cold flow temperature are 0 °C, -15 °C and -30 °C.

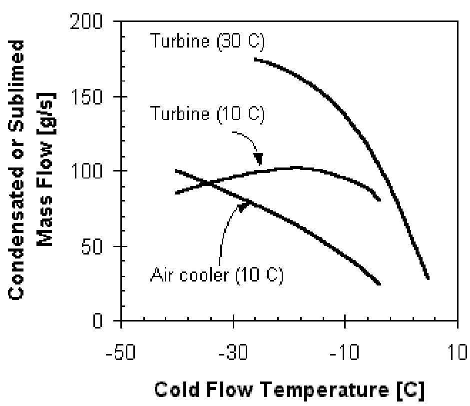

Figure 4: Division of phase change phenomena between the RBC air cooler and turbine for air-cooler outlet temperatures of 10 °C and 30 °C. At 30 °C, all condensation and sublimation occurs in the turbine.

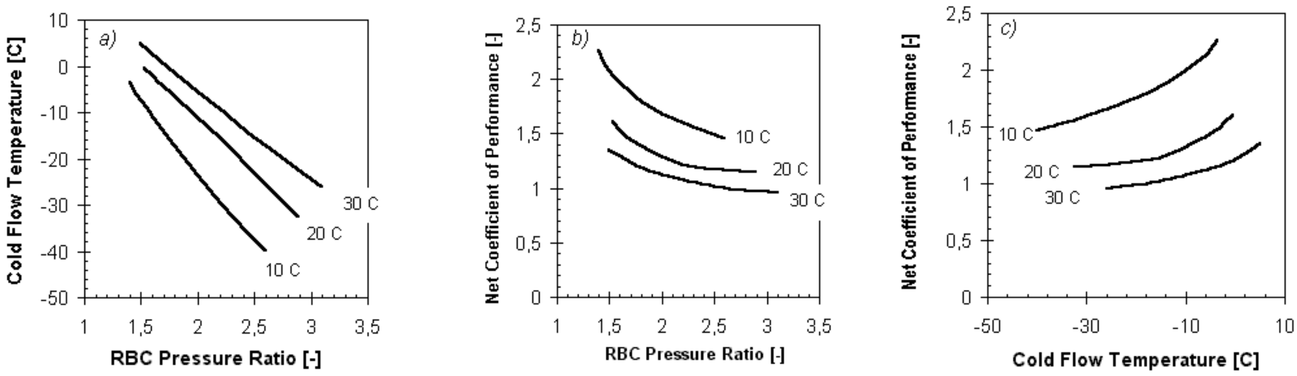

Figure 5: Impact of RBC pressure ratio on a) cold flow temperature and b) net coefficient of performance, and c) the resulting *COP* against cold flow temperature when the air-cooler outlet temperature is 10 °C, 20 °C and 30 °C.

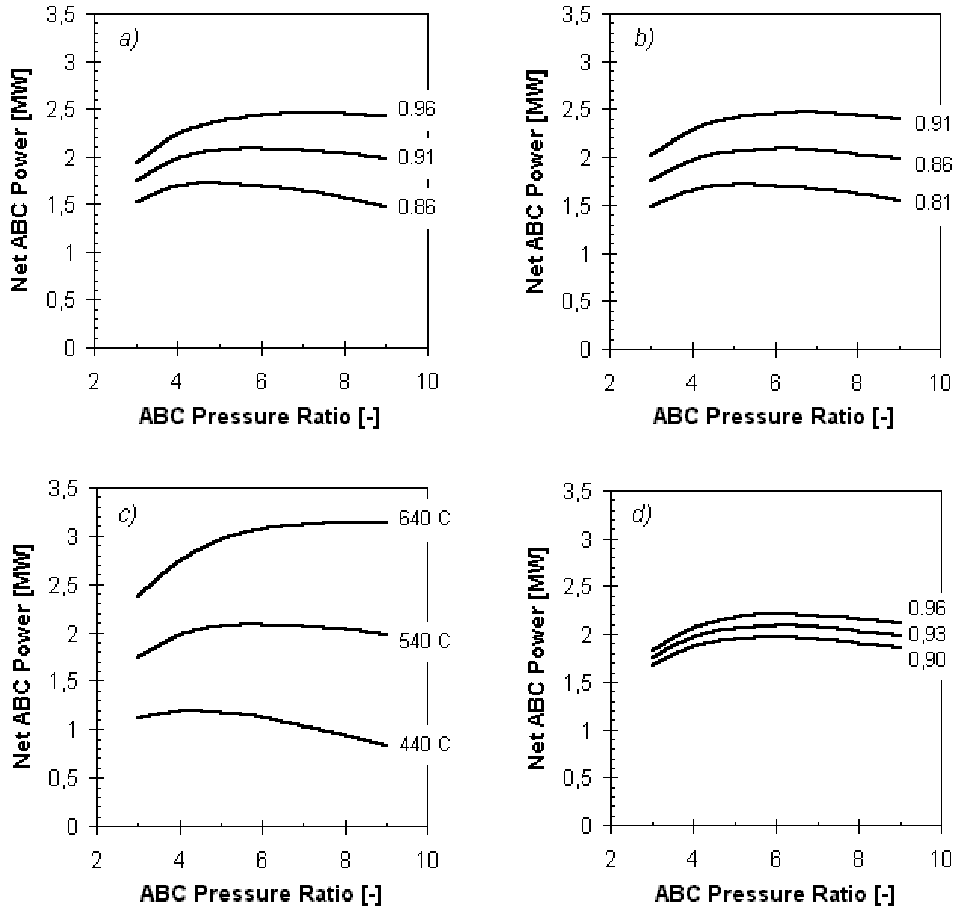

Figure 6: Sensitivity of the performance of ABC with one inter cooler against a) polytropic efficiency of the compressor units b) turbine polytropic efficiency c) topping cycle exhaust gas temperature d) recuperator effectiveness. The curves are valid for applications when no cold flow is being produced, similar to Fig. 2.

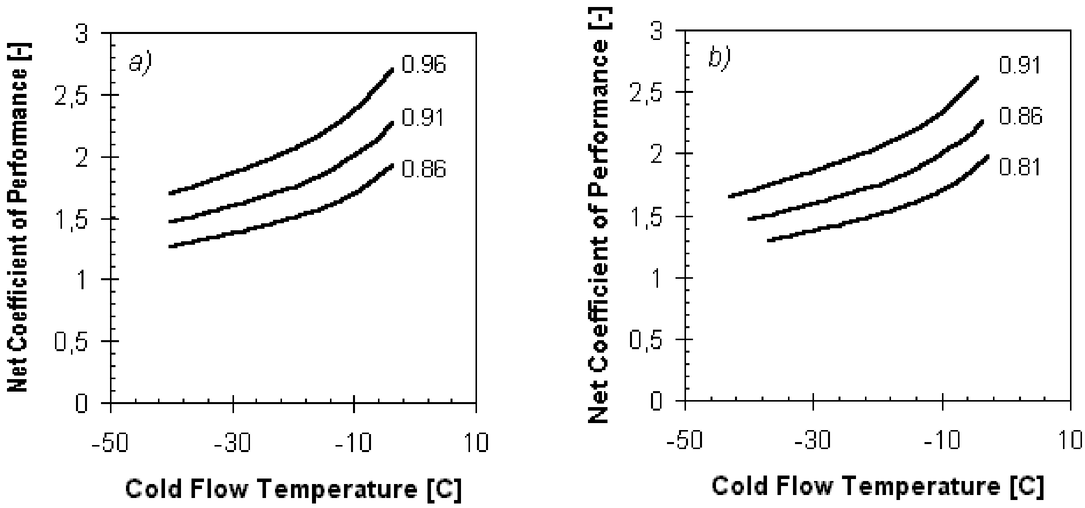

Figure 7: Sensitivity of RBC performance against a) compressor polytropic efficiency and b) turbine polytropic efficiency. The air-cooler outlet temperature is 10 °C.

Table 3: Performance of the specified dual gas turbine combined cycle with RBC integrated into ABC.

Net power output (gas turbine)	MW	7.55
Net power output (ABC)	MW	1.24
District heat rate	MW	11.19
Cooling rate (-15 °C)	MW	1.05
Net electric efficiency	%	35.6
CO_2 emission for power	g/kWh	590
Reduction in CO_2 due to ABC	%	11.8
Power-to-heat ratio	-	0.79
Total eff. for power and heat	%	80.8

Apart from the efficiency of the low-pressure compressor, changes in the ABC specifications have no effect on RBC performance. On the other hand, changes in the RBC specifications affect only the net power output from the ABC by altering RBC power need. Based on this, sensitivity studies have been performed separately for both cycles.

The sensitivity of ABC performance has been examined against the following cycle parameters: turbomachinery efficiencies, topping cycle exhaust gas temperature and recuperator effectiveness. The results of the simulations are presented in Figure 6.

Inherent to Brayton cycles, the turbine inlet temperature to the ABC and, consequently, the topping cycle exhaust gas temperature has a dominating impact on ABC performance. This suggests favouring topping engines or cycles with high exhaust temperature levels. Turbomachinery efficiencies have also a strong impact on cycle performance because of the low temperature ratios in the ABC. Therefore, the need to use high-efficiency turbomachinery in ABC applications is apparent. The optimum pressure ratio increases with increasing component efficiencies and topping cycle exhaust gas temperature, while the power output becomes less sensitive towards the pressure ratio around the optimum. For recuperator effectiveness, the impact on cycle performance and the increase in the optimum pressure ratio are less evident.
In addition to Figure 5c, the sensitivity of RBC performance has been studied by examining the impact of turbomachinery efficiencies on the coefficient of performance when the air-cooler outlet temperature is 10 °C (Figure 7). As the figure indicates, compressor and turbine efficiencies have very similar effects. A comparison with Figure 5c shows that the cold flow temperature has a more dominating role in determining RBC performance than turbomachinery efficiencies.

4. CONCLUSIONS

In this paper, a thermodynamic analysis has been given for a system where a Reversed Brayton Cycle (RBC) is integrated into an intercooled Air Bottoming Cycle (ABC) to provide cold airflow. The resulting trigenerative Air Bottoming Cycle (TriABC) with power, heat and cooling output can be an attractive alternative for small-scale distributed energy conversion systems.

The power output from the TriABC depends on the cooling rate (required cold air mass flow rate and temperature level). For the specified combined cycle with Alstom Tempest as the topping gas turbine and the optimised TriABC as the bottoming cycle, the TriABC contributes the power output with 2.1 MW if no cooling is required. This corresponds to an increase of 24.4 % in the power output and an increase of 7.6 %-points in net electric efficiency as compared to the topping gas turbine in a simple-cycle operation. For a case where -15 °C cold air mass flow equals that of the basic ABC, a power output of 1.2 MW (an increase of 13.4 %) is attained while a cooling duty of 1.1 MW is being fulfilled. The air-cooler outlet temperature of 20 °C has been used for this case. The production of cold air affects the power output from the TriABC only by altering the power need of the RBC, and therefore the heat production is unaffected by the changes in the power and cooling ratings. For the specified combined cycle, the total amount of 11.2 MW is applicable as district heat.

The performance of the ABC is highly sensitive to the turbine inlet temperature. To enhance the performance, topping engines with high exhaust gas temperatures should be favoured. ABC turbomachinery efficiencies have also a strong effect on cycle performance. Therefore, the need to use high-efficient compressors and turbines is evident. For the RBC, the temperature of air that can be reached in the air cooler is decisive for cycle performance. Consequently, the air must always be cooled down as much as possible before the expansion.

This study has considered a case where the cold air flow is not returned. Should it be returned from the process at a conveniently low temperature, it could be utilised by cooling down the air further before expansion. This regenerative configuration would reduce the required pressure ratio for the RBC, and consequently, the power need for a specified cold flow temperature.

The TriABC offers an intriguing way to integrate the use of energy-efficient combined cycle technology and environmentally friendly (CFC-free) cooling technology for small-scale installations. Offshore industry and transportation can be regarded as potential platforms for TriABC applications, but also stationary industry and specific applications, such as hospitals and shopping centres. However, a systematic exploration of the potential for the concept, followed by detailed design and comprehensive economic evaluation is required before making final decisions of the feasibility of the specific applications.

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NOMENCLATURE

ABC	air bottoming cycle
CFC	chlorofluorocarbons
COP	coefficient of performance, -
DGTCC	dual gas turbine combined cycle
EFGT	externally-fired gas turbine
IC	intercooler
M	molecular weight, kg/kmol
ORC	organic Rankine cycle
p	pressure, Pa
RBC	reversed Brayton cycle
TriABC	trigenerative air bottoming cycle
φ	relative humidity, -
ω	humidity, -

Subscripts

da	dry air
sat	saturated
vap	water vapour