Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-46236

GAS TURBINE BOTTOMING CYCLES FOR COGENERATIVE APPLICATIONS: COMPARISON OF DIFFERENT HEAT RECOVERY CYCLE SOLUTIONS

Rakesh K. Bhargava
Turbomachinery Consultant
Houston, USA

Michele Bianchi
University of Bologna - DIEM
Bologna, Italy

Andrea De Pascale
University of Bologna - DIEM
Bologna, Italy

ABSTRACT

With all the advancements made in the gas turbine technologies in the last 7 decades, a large amount (approximately 60%) of the thermal energy in the gas turbine exhaust is released in to the environment. This discharged heat could be profitably used not only in thermal utilities but also as an intermediate temperature heat source for the bottoming cycles producing electric power. This paper provides a systematic thermodynamic performance evaluation and comparison among the three different waste heat recovery solutions, namely, the Inverted Brayton Cycle, the Bottoming Brayton Cycle and the Organic Rankine Cycle. The results obtained from the parametric analyses of the CHP systems clearly identify advantages and limitations of the gas turbine technology and its size when combined with the three bottoming cycles evaluated in this study. A detailed discussion on the obtained results is presented in this paper.

INTRODUCTION

Modern gas turbine based power generation systems are characterized by simple cycle thermal efficiency values typically in the range of 30-40% [1]. One of the undesirable factors with the use of gas turbines and the other power generation equipment is a loss of considerable amount of thermal energy in their exhaust stream unless cogeneration or combined cycle arrangements are used to recover such waste heat. Nevertheless, for small power size gas turbines (GT) with limited conversion efficiency, the combined cycle arrangement is questionable (as the adopted simplified HRSG layout grants limited performance enhancement), while the other strategies can be introduced. The Exhaust Gas Temperature (EGT) values of some of the commercially available GT units with power output rating up to 50 MW (for both Heavy Duty and Aeroderivative) are shown in Fig. 1, where the EGT values for Micro Gas Turbines (MGT) and reciprocating Internal Combustion Engines (ICE) are also shown for comparison purpose. The presented data in Fig. 1 shows that the EGT values for gas turbines, ICEs, and MGTs are in the range of 400°C-600°C, 400°C-500°C, and 200°C-300°C, respectively and implying a potential for recovering electrical and thermal energy.

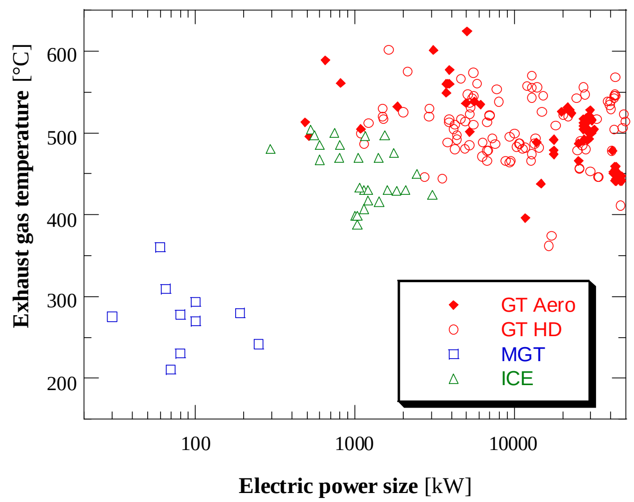

Figure 1: Exhaust gas temperatures for selected machines with power rating up to 50 MW (ISO conditions)

The aim of the present study is to investigate the selected bottoming cycles for recovering waste heat energy available from the small/medium power (100 kW to 30 MW) rating gas turbines and MGTs with particular emphasis to Combined Heat and Power (CHP) applications requiring low temperature (50°C-70°C) thermal energy.

THE INVESTIGATED BOTTOMING CYCLES

Among the available bottoming cycle solutions, the present study is focused on the following three cycles:

(i) The Bottoming Brayton Cycle (BBC), also named Air Heat Recovery Turbine by Romanov et al. [2] or Air Bottoming Cycle by Korobitsyn [3];

(ii) The Inverted Brayton Cycle (IBC), described by Wilson [4] and others [5,6]; and
(iii) The Organic Rankine Cycle (ORC), discussed by many researchers [7-15].

In a comprehensive parametric study, Romanov et al. [2] investigated effects of various design parameters on the performance of systems consisting of GT and BBC with different configurations and identified benefits of such a system. Exhaust gas heat recovery with BBC showed increase of approximately 11%-20% and 4%-6% for power output and efficiency, respectively [2].

Wilson [4] presented a historical development of IBC concept and showed an encouraging return on investment of 34% (based on 1970 energy and investment cost) for power output gain with the implementation of IBC to the standard gas turbine having exhaust gas temperature (EGT) of 560°C. It is interesting to note that the IBC arrangement allows exhaust heat recovery with no additional back pressure losses to the main gas turbine (topping cycle). Tsujikawa et al. [5] proposed and conducted a parametric evaluation of IBC system with intercooling and showed efficiency improvement of about 7%. While evaluating waste heat recovery utilizing IBC for the recuperated MGTs, Bianchi et al. [6] showed electrical and thermal performance improvement of the system including reduction in thermal stresses in the recuperator heat exchanger.

Invernizzi et al. [7] investigated performance of energy system consisting of ORC combined with the MGT in addition of presenting a detailed discussion on the thermodynamic characteristics of potential working fluids including their limitations and advantages as it relates to their use in the bottoming ORC system. Whereas, implementation of ORC with high efficiency and low EGT gas turbines investigated by Chacartegui et al. [8], showed significant performance gains which are higher than the conventional Rankine cycles. It became evident from their studies that proper selection of the working fluid in the ORC system is critical in achieving improved performance of the GT+ORC system.

While the non-cogenerative thermodynamic performances of these cycles have been analyzed in different available studies [2-8], the CHP layout arrangements and their comparative performance are investigated in this paper. Furthermore, the CHP arrangements with the bottoming cycles are examined considering recovery of thermal energy at low temperatures (50°C-70°C) useful for applications such as residential heating systems.

For a complete comparison of the bottoming cycles based CHP systems, an economic analysis in combination with the thermodynamic performance evaluation will be required. This aspect has not been addressed in this paper, but the evaluation of the electric power generation, the thermal energy recovery, and the primary energy savings can be used as a basis for the future economic investigation.

The Brayton Bottoming Cycle (BBC)

The BBC based CHP system, shown in Fig. 2, is a particular gas turbine where the combustor is substituted by a heat exchanger, here named recuperator (REC), which recovers thermal energy from the available hot gas stream of the topping GT cycle. In the BBC system, an external stream of air, compressed in a compressor (C), heated in the REC and expanded in the turbine (T) as shown in Fig. 2, undergoes a simple Brayton cycle process.

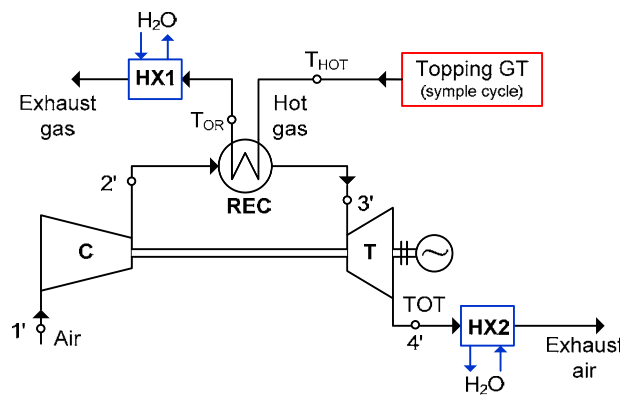

Figure 2: A CHP system layout with the BBC system

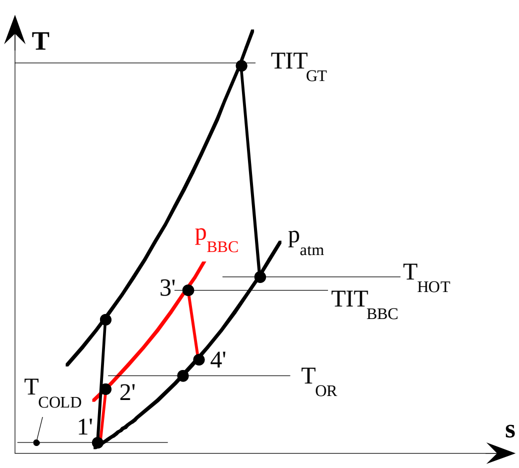

Figure 3: T-s diagram of the topping GT cycle and the BBC system

The thermodynamic T-s diagram of the BBC system is shown in Fig. 3 (cycle 1'-2'-3'-4' which operates between the BBC max pressure p_{BBC} and p_{atm}). The BBC is incorporated within the Brayton cycle of the topping gas turbine, which discharges a gas stream at temperature T_{HOT} (equal to 450°C in the case of Fig. 3). Thus the BBC turbine inlet temperature TIT_{BBC} is limited by the T_{HOT} value. The BBC net useful work is converted in the electric power, while a fraction of the unused heat can be further recovered in a CHP configuration. In the BBC system examined in our study, two dedicated heat exchangers are considered: heat exchanger HX1 recovers the available thermal energy form the topping hot gas stream downstream the REC; and the heat exchanger HX2 is used to extract the low temperature thermal energy from the BBC exhaust (the air state 4' at the turbine outlet is still above the ambient temperature, see Fig. 3). In order to design the CHP system utilizing the BBC, values of the following parameters must be defined: compressor pressure ratio, polytropic efficiency values for the compressor and turbine sections, compressor inlet air mass flow rate, effectiveness of heat exchangers HX1 and HX2, and effectiveness of the recuperator.

The Inverted Brayton Cycle (IBC)

The IBC based CHP system, shown in Fig. 4, consists of the following processes:

(a) The exhaust gas stream available from the topping GT cycle is expanded in the turbine (T) section of the IBC from the atmospheric pressure down to a below ambient pressure value (p_{low}) in order to convert into useful mechanical power the still available enthalpy content of the exhaust gas.

(b) The gas is cooled in the heat exchanger (HX1) of the IBC with an external fluid. A high temperature thermal energy production occurs in the heat exchanger HX1 as a part of the CHP system.

(c) The cooled gas leaving HX1 is compressed to ambient pressure in the compressor (C) section of the IBC.

(d) A low temperature thermal energy production occurs in the heat exchanger HX2 positioned at the compressor outlet, where exhaust gas leaving the IBC is still at a temperature above the ambient value as shown by the T-s diagram of this bottoming solution, (See cycle 1'-2'-3'-4' in Fig. 5).

The IBC minimum temperature (point 3' in Fig. 4), which influences the generated power, is limited by the cooling water source temperature T_{COLD}. The key design parameters of the IBC are: the IBC turbine exit pressure p_{low}, the IBC compressor inlet temperature ($T_{3'}$), polytropic efficiency values for the turbine and compressor sections, and effectiveness of the heat exchangers HX1 and HX2.

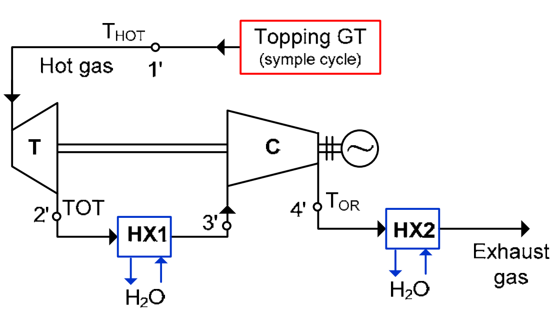

Figure 4: A CHP system layout with Inverted Brayton Cycle

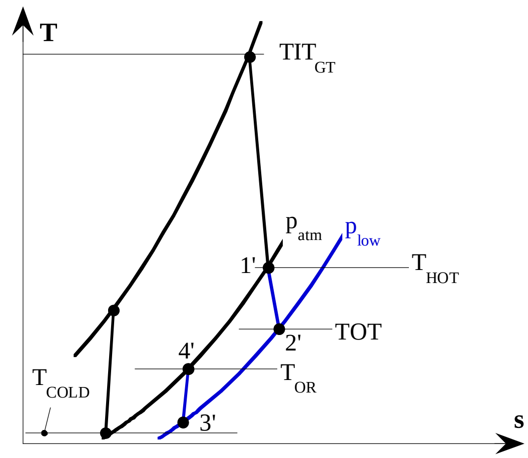

Figure 5: T-s diagram of the topping GT and the IBC cycle

The Organic Rankine Cycle (ORC)

The ORC based CHP system, shown in Fig. 6, is a bottoming cycle system using an external fluid which undergoes a thermodynamic closed cycle process resembling a water-based Rankine cycle. By integrating an ORC with a topping GT cycle, a binary fluid system is obtained in which the ORC's operating fluid receives heat from the topping GT cycle exhaust gas in a vaporizer heat exchanger (VAP) and discharges residual heat to the environment in a condenser (K). The CHP low-temperature thermal energy production is provided by the condenser output through a heat exchanger HX2. Whereas, high temperature thermal energy production occurs through an additional heat exchanger HX1 located downstream of the vaporizer (VAP) as shown in Fig. 6. A T-s diagram for the GT+ORC system, shown in Fig. 7, corresponds to the ORC based CHP system of Fig. 6, where a recuperator (REC) is used to preheat the working fluid before entering the vaporizer.

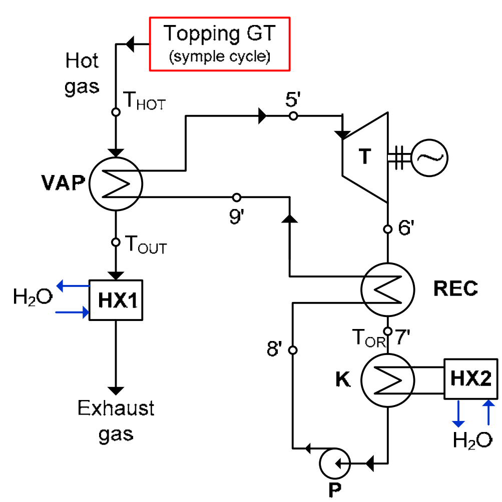

Figure 6: A CHP system layout with Organic Rankine Cycle

The ORC fluid selection, analyzed and published by many authors [8-10], is typically oriented to "dry" fluids with positive slope of the saturated vapor curve in order to avoid two-phase flows in the expander (T) section of the ORC and avoiding at the same time superheater boiler section. In this study, only a sub-critical ORC arrangement has been considered: this is common in the current state-of-the-art ORC applications, while advanced thermodynamic configurations with supercritical or high temperature superheated fluid [11] are currently under investigation but not implemented yet. The most appropriate fluid (meeting environmental and safety considerations) depends also on the available source temperature [10, 12], which affects the ORC evaporation pressure and temperature. In our study, benzene and MDM (also known as octamethyl-trisiloxane) are used as the working fluids for the ORC system, because the corresponding critical temperature is one of the highest (around 290°C for both fluids) among the fluids typically proposed for the ORC applications. Furthermore, benzene and MDM (which belongs to the siloxane family and used for high temperature commercial ORC systems) provide different thermodynamic performance [13-14], which are investigated in the paper.

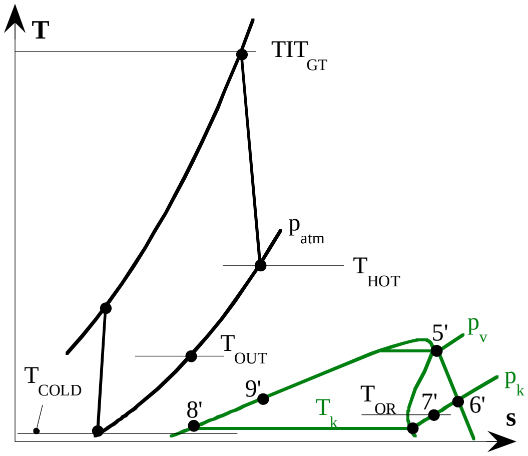

Figure 7: T-s diagram of the topping GT and the ORC cycle

In order to evaluate the ORC thermodynamic performance, the pressure levels of the working fluid in the evaporator and in the condenser (p_v and p_k), the turbine section's polytropic efficiency, and effectiveness values for various heat exchangers are the most important design parameters for the ORC based CHP system shown in Fig. 6.

ANALYSES METHODS AND ASSUMPTIONS

To compare the cogenerative performance of a topping GT cycle integrated with the bottoming cycles (BBC, IBC and ORC) described above and to identify the optimal thermodynamic configuration, a numerical parametric investigation has been carried out by varying the key design variables of the bottoming cycles. Moreover, the above described CHP systems based on the bottoming cycles have been analyzed with different topping gas turbines to assess the effect of the GT technology and power rating on the bottoming cycle and on the CHP system's overall performance. In particular, a Micro Gas Turbine (MGT), a very small and low-efficiency gas turbine (GT1), a small size heavy-duty medium technology gas turbine (GT2) and a modern aeroderivative gas turbine (GT3) have been considered as topping cycle gas turbines. Table 1 summarizes the main design data of the selected topping cycle gas turbines.

In case of the IBC system integration with the MGT, which is characterized by a recuperated cycle, a different layout following the work of Bianchi et al. [6] has been analyzed as shown in Fig. 8. In this case, the IBC is incorporated between the MGT expander and the recuperator. In this arrangement, the sub-atmospheric expansion in the IBC takes place using the high temperature gas at the MGT expander outlet as the gas temperature at the GT REC outlet would be too low to extract significant power with a sub-atmospheric expansion. For the other topping cycle GTs (GT1, GT2, & GT3) with the IBC, the CHP layout analyzed is as shown in Fig. 4.

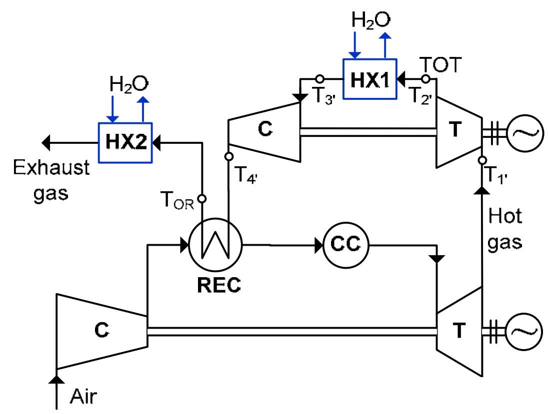

Figure 8 A CHP system layout with MGT and IBC

Table 1 Key design parameters of the selected topping GTs

		P_{el}	eff.	PR	TIT	T_{HOT}	m_{GT}
		[kW]	[%]	[-]	[°C]	[°C]	[kg/s]
MGT	Elliott TA100	100	29	4.0	950	298	0.84
GT1	Kawasaki M1A-23	2,043	24.3	11.4	1050	569	9.8
GT2	Siemens SGT-400	12,900	34.1	16.9	1250	555	40
GT3	GE LM2500+	28,000	38.4	22.2	1260	517	80

The thermodynamic parametric analyses of the CHP systems with the bottoming cycles considered in this paper have been based on the following additional assumptions:

- The values of polytropic efficiency of compressor and expander sections for the topping gas turbines and the IBC and BBC have been varied with the GT size as summarized in Table 2 in order to take into account the different levels of component technology [1, 6, 16].

- The ORC turbine's efficiency has been considered equal to 0.8, a value compatible with existing machines [15] and in line with a previous study of the authors [17].
- Due to the bottoming cycle components, additional pressure losses on the gas path have been considered; these losses affect the topping cycle GT performance. The gas side pressure losses have been assumed equal to 3% of the inlet pressure for the REC and equal to 2% of the inlet pressure for heat exchangers HX1 and HX2.
- The effectiveness value of various heat exchangers has been set equal to 85%.
- The gas exit temperature at HX1 (in BBC, IBC and ORC applications) and at HX2 (in BBC and IBC applications) has been fixed equal to 50°C, a value compatible with a CHP application operating with a low temperature thermal utility.
- The condensation temperature (T_k) of the ORC has been fixed equal to 70°C in order to consider a further low-temperature heat recovery by means of the external cooling fluid.
- The ORC simulations have been performed by assuming a fixed minimum temperature difference (ΔT) between the hot gas and the ORC fluid in the evaporator equal to 10°C.
- Water temperature at the inlet of HX1 and HX2 is assumed equal to 15°C.

Table 2: Assumed polytropic efficiency values for turbine and compressor sections of the selected topping gas turbines

	MTG	GT1	GT2	GT3
Compressor polytropic efficiency	0.85	0.89	0.90	0.91
Turbine polytropic efficiency	0.83	0.89	0.90	0.91

The numerical investigation is based on a lumped model approach which includes the following:

(i) The system is decomposed in the key sub-components (heat-exchangers, expanders, compressors, combustors, etc.)

(ii) Each sub-component of the energy system is modeled considering its specific design properties (internal efficiency, pressure losses, etc.)

(iii) The design operation is described by solving mass and energy balance for sub-components and for the overall system; this gives as output the thermodynamic state at inlet and outlet of each component.

(iv) Real fluid behavior is used for all the working fluids.

The thermodynamic analysis has been performed on the BBC and IBC based CHP systems with the commercial software GateCycle^{TM} [18], while the ORC based CHP system performance has been obtained with an externally developed calculation worksheet, which uses the fluid properties calculated by means of the thermodynamic database of the FluidProp program [19].

RESULTS AND DISCUSSION

The obtained numerical results are presented in the following paragraphs for each investigated bottoming cycle, by analyzing the effect of the main design parameter on the electric and thermal power production. The aim is to identify the best performing bottoming cycle configuration, depending on the topping cycle gas turbine, and to compare the CHP performance.

Performance of CHP System with BBC

Figure 9 presents the calculated values of the BBC additional electric power (P_{BBC}) with reference to the topping GT power (P_{GT}) and the calculated total efficiency, defined as:

$$\eta_{GT+BBC} = \frac{P_{GT} + P_{BBC}}{F} \tag{1}$$

where, F is the energy content, expressed in kW, of the total fuel consumed by the topping GT cycle based on the fuel's LHV. For our study, natural gas with LHV value of 48 MJ/kg is used.

In particular, Fig. 9-a shows the effect of the BBC inlet air flow (m_{BBC}) for different BBC pressure ratio values ($PR_{BBC,}$ ranging from 4 to 6 in case of GTs, from 1.5 to 2 in case of MGT) and for the different considered GTs. For a given topping GT and a fixed PR_{BBC} value, the change in m_{BBC} affects the BBC TIT value (an increase in m_{BBC} causes a decrease in TIT_{BBC}, because the amount of discharged heat from the topping GT is constant) which, in turn, affects the bottoming specific power (which reduces with TIT_{BCC} decrease): the numerical analysis shows that the obtained BBC power is maximum if m_{BBC} equals the topping GT outlet mass flow (m_{GT}). The BBC increase in power is more significant in small and low-efficiency topping GTs, reaching values up to 30% of the GT power. However, the electric power recovered is small (approximately 2-3% of the topping cycle) with the MGT, mainly due to their low T_{HOT} values.

Figure 9-b shows that an optimum value of PR_{BBC} can be found equal to about 4 for all the three analyzed GT machines, while for the MGT the value of PR_{BBC} providing maximum power is close to 2.

The cogenerative performances obtained for the CHP system with the BBC are presented in Fig. 10. The ratio of the electric to thermal power for the CHP system (in Fig. 10-a) changes with the investigated machines, ranging from about 1.0 in case of GT3 to 0.6 for GT1 and even less in case of MGT. The gas temperature levels of the thermal energy available at inlet to the heat exchangers HX1 and HX2 are shown in Fig. 10-b. The values of turbine outlet temperature (TOT) at inlet to the heat exchanger HX1 decrease in case of GT1, GT2, and GT3 from less than 350°C down to about 200°C as the value of PR_{BBC} increases in the investigated range from 3 to 8 (the decrease of TOT with increase in the value of PR_{BBC} can be justified observing the T-s diagram of Fig. 3). The REC outlet temperature (T_{OR}) instead rises in the same range of values with increasing values of PR_{BBC} because of decreased amount of heat exchange in the REC. Similar trends are obtained for the MGT case, but the temperature values are lower (see Fig. 10-b).

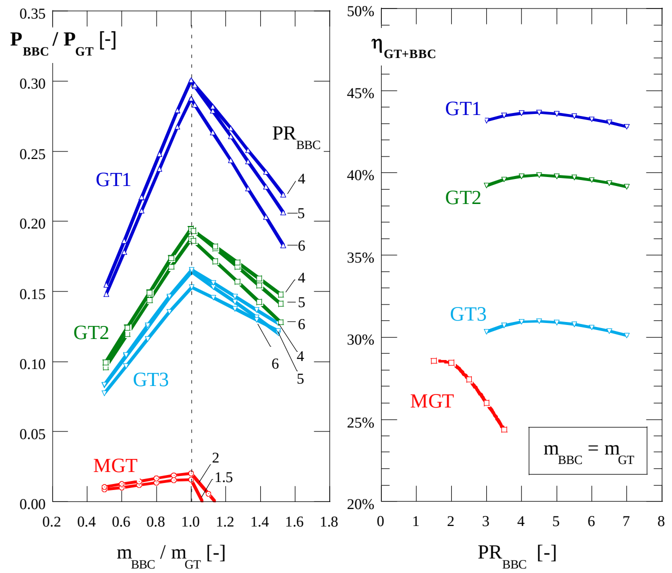

Figure 9: (a) BBC electric power versus normalized air mass flow; (b) GT+BBC electric efficiency versus BBC pressure ratio.

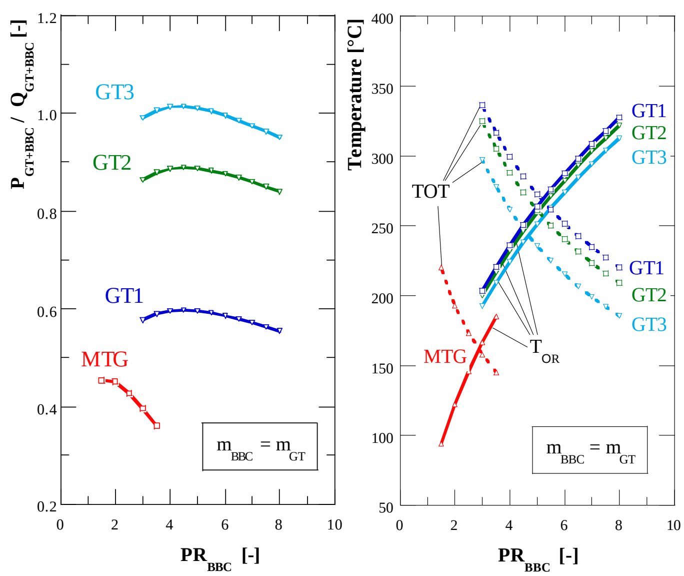

Figure 10: (a) BBC electric to thermal power ratio versus BBC pressure ratio; (b) BBC TOT and REC exit temperature versus BBC pressure ratio

Figure 11 shows the thermal energy recovery for the two heat exchangers HX1 and HX2, with reference to the GT discharged thermal energy. The amount of energy recovered by HX1 (Q_{HX1}) is larger than the energy recovered with HX2 (Q_{HX2}) when the PR_{BBC} value is low. Very similar values of Q_{HX1} (and also of Q_{HX2}) are obtained for GT1, GT2 and GT3 cases.

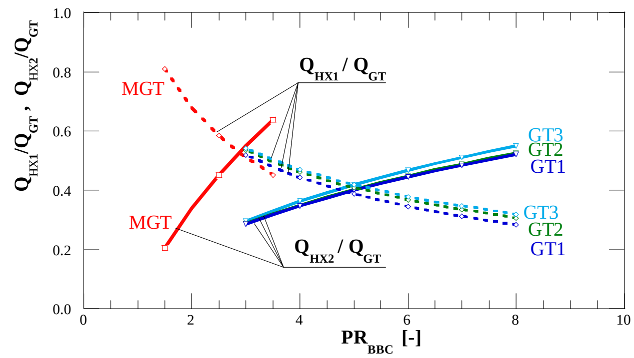

Figure 11: BBC thermal energy split between HX1 and HX2.

Performance of CHP System with IBC

Figure 12 provides the calculated performance of the CHP system obtained by integrating a topping gas turbine cycle with the IBC. In this bottoming cycle case, the key design parameter which has been varied is the sub-atmospheric expander outlet pressure (p_{low}). The performance results show that, for the GT cases examined, the lower is the value of p_{low} the higher the benefit in terms of electric power recovery with increased CHP system efficiency. The adoption of the IBC can provide up to 20-30% more electrical power if the GT1 machine is used and the value of p_{low} is in the range of 0.3-0.5 bar. Only the MGT+IBC case (with a different plant layout, according to Fig. 8) shows a different behavior; in this MGT case, the value of p_{low} providing the maximum power increase (up to about 40-45%) is 0.3 bar, while the optimum p_{low} value which provides the maximum efficiency is close to 0.6 bar. It is interesting to notice in Fig. 12-b that the overall electric efficiency can become as high as 44% in case of integration of the IBC with the aeroderivative GT3, while the efficiency can reach values of 31% in case of GT1 and close to 33% in case of the MGT.

Figure 13-a shows the calculated values of the electric to thermal power ratio. The maximum values are obtained for low p_{low} values in case of GTs+IBC (larger values up to 0.95 for case GT3), while in the MGT+IBC case the maximum value equal to about 0.6 is obtained in case of p_{low} = 0.6 bar.

The available heat is discharged to the thermal utilities at different levels of temperature, as shown in Fig. 13-b: the temperature at the turbine outlet ranges from 300-500°C, in case of GTs+IBC (the lowest values are obtained with low p_{low}), and it is close to 250°C in the case of MGT+IBC. The temperature at the HX2 inlet (T_{OR}), instead, ranges between 50 and 200°C, with lower values when p_{low} is high.

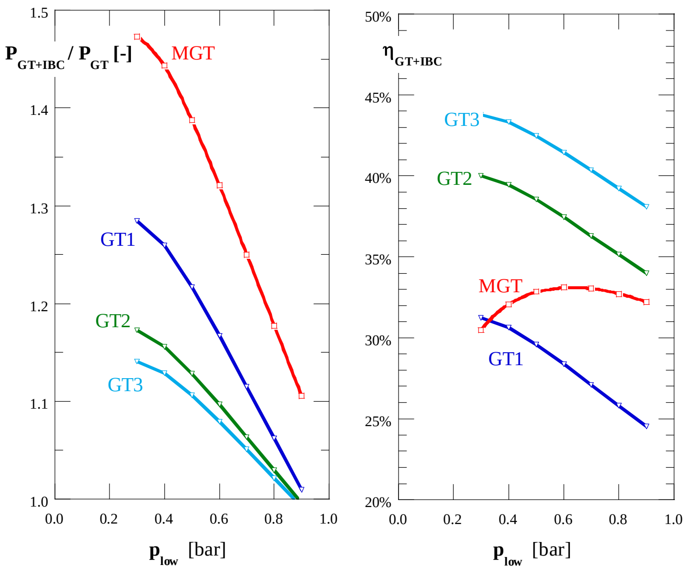

Figure 12: (a) normalized GT+IBC electric power versus IBC lower pressure; (b) GT+IBC electric efficiency versus IBC lower pressure

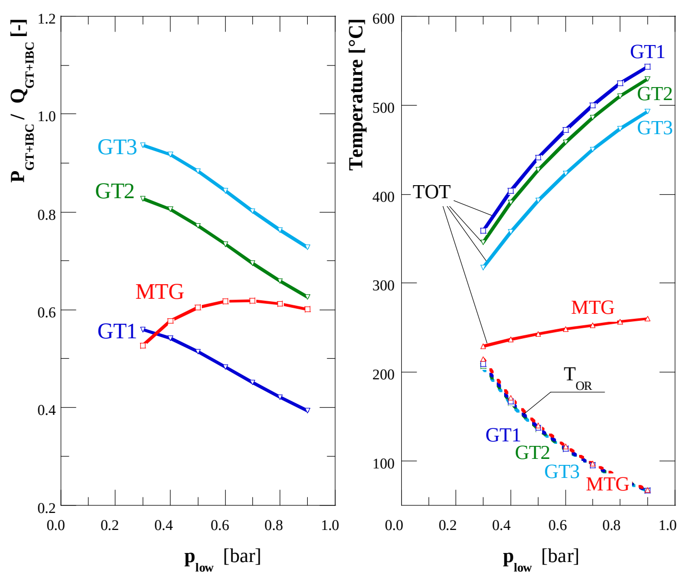

Figure 13: (a) IBC electric to thermal power ratio versus IBC lower pressure; (b) IBC TOT and HX2 inlet temperature versus IBC lower pressure

Figure 14 shows the thermal energy recovery for the two heat exchangers HX1 and HX2 in case of IBC; in this case the amount of energy recovered by HX1 can be equal to about 60-95% of the GT discharged thermal energy. The trends of thermal energy recovery are observed same for the three gas turbines examined. In the MGT case the HX2 thermal energy can become more significant, especially at low p_{low} values.

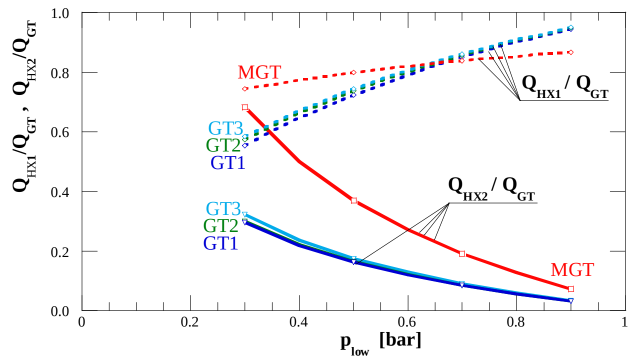

Figure 14: IBC thermal energy split between HX1 and HX2

Performance of CHP System with ORC

The ORC bottoming system performance have been investigated by considering different values of the vaporization temperature (T_v), the key design parameter which strongly affects both the electric power generation and the thermal energy recovery.

The obtained thermodynamic performance results are shown in Figs. 15 and 16, for the two investigated organic fluids, namely Benzene (continuous lines) and MDM (dotted lines).

In particular, both the electric power production and the electric efficiency of the integrated GT+ORC system increase with increase in the T_v value with both ORC fluids. Whereas, the values of net power output and electric efficiency of the MGT+ORC system decrease with increase in the T_v value of both ORC fluids. This observed behavior with different topping machines is mainly due to the variation of the ORC fluid mass flow rate and the related power generation in the bottoming cycle. In particular, in the case of GT1, GT2 & GT3, which have similar performance trends and have high value of T_{HOT} compared to the MGT, the power output of the bottoming cycle and the evaporated organic fluid mass flow rate increase and decrease, respectively with increase in the T_v value leading to enhancement in the net power output and efficiency of the CHP system. However, in the case of MGT with reduced value of T_{HOT}, both the amount of organic fluid required and the power output in the bottoming cycle reduce with increase in the T_v value and thus contributing to the reduction in the net power output and the efficiency of the GT+ORC based CHP system.

Figure 16 provides the effect of T_v on the calculated cogenerative performance: the P_{GT+ORC}/Q_{GT+ORC} ratio (Fig. 16-a) is larger in case of GT3 and increases with T_v (from 1 up to 1.1 in case of Benzene), while the lowest electric to thermal power ratio values (in the range 0.6-0.7) are obtained with the MGT.

For all the GTs examined and the range of T_v considered, the calculated value of the gas temperature T_{OR} at the REC outlet (thermodynamic state 7' in Fig.6) is nearly 80°C and 90-100°C for Benzene and MDM, respectively. Whereas, the gas temperature T_{OUT} at the vaporizer exit varies significantly with T_v from 130-250°C in the MGT case for both ORC fluids, but

T_{OUT} values are relatively lower for all the other GTs. These observed changes can be attributed to the ORC fluid thermodynamic state in which heat exchange occurs and its curve slope.

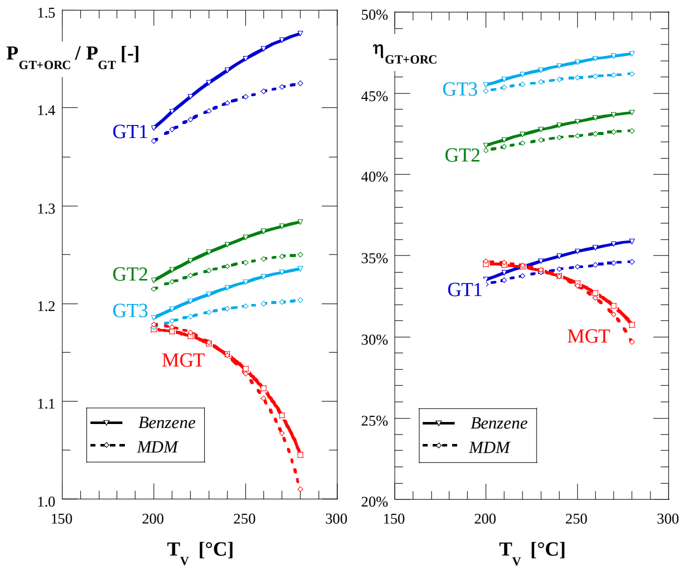

Figure 15: (a) normalized GT+ORC electric power versus ORC vaporization temperature; (b) GT+ORC electric efficiency versus ORC vaporization temperature

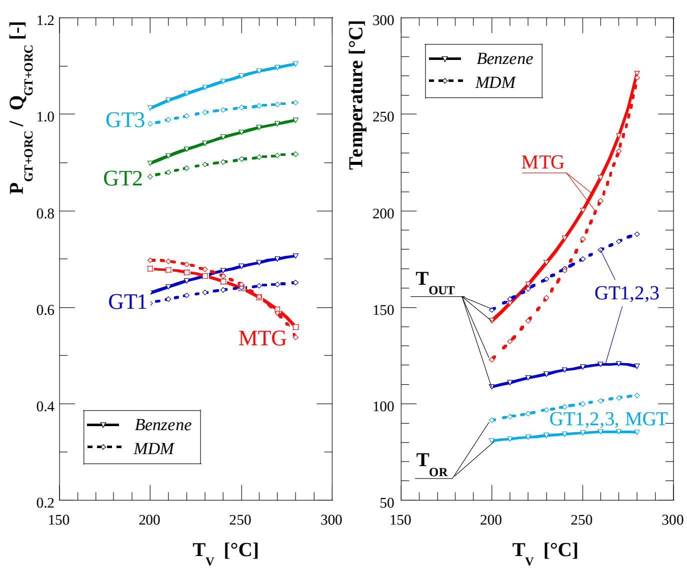

Figure 16: (a) ORC electric to thermal power ratio versus ORC vaporization temperature; (b) gas exit temperature at the ORC vaporizer (T_{OUT}) and ORC fluid temperature at the REC outlet versus (T_{OR}) ORC vaporization temperature

Further illustration on variation of the gas temperature T_{OUT} for one gas turbine (GT3) and the working fluids Benzene and MDM are presented on T-s diagrams in in Figs. 17 and 18, respectively. In the MGT case and with Benzene as the ORC fluid, the minimum ΔT value occurs at the beginning of the vaporization process at p_v, whereas, in the GT3 case (and similarly for GT1 and GT2 cases not shown in the figure) the minimum ΔT occurs at the beginning of the liquid phase (in correspondence of point 9'), downstream of the REC. The location of the minimum ΔT is important because it affects the exit gas temperature (T_{OUT}), and consequently the amount of thermal energy in the exhaust gas, which can be recovered downstream, with HX1. Similar conclusions can be obtained for ORC operating with MDM (Fig. 18).

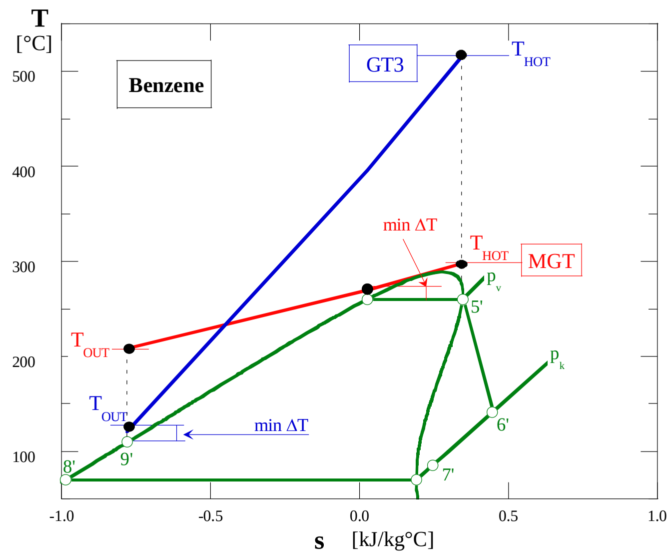

Figure 17: T-s diagrams of the ORC (Benzene, T_v=260°C, T_k=70°C) and corresponding gas temperature decrease at the outlet of GT3 (blue) and of MGT (red)

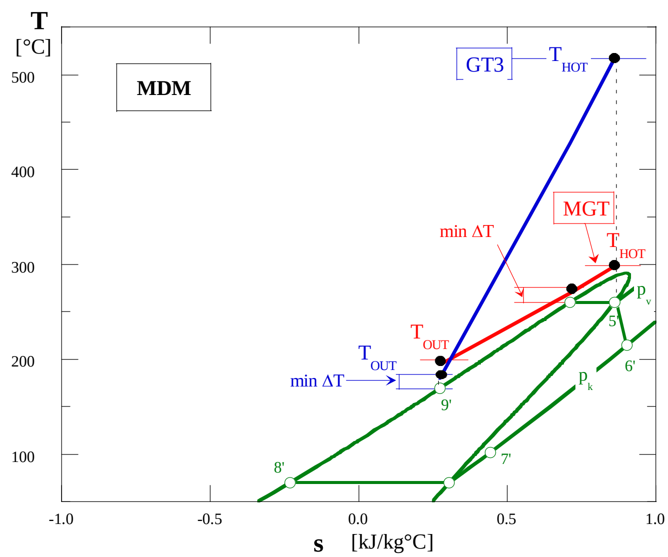

Figure 18: T-s diagrams of the ORC (MDM, T_v=260°C, T_k=70°C) and corresponding gas temperature decrease at the outlet of GT3 (blue) and of MGT (red)

The obtained results show that the fluid selection affects the final CHP performance. In particular, in comparison with MDM, Benzene provides larger values of electric efficiency and electric power, but lower values of thermal to electric power ratio and lower inlet temperature at the thermal utilities (T_{OUT}).

The thermal energy recovered in HX1 and HX2 is shown in Fig. 19 for the ORC system (using Benzene as the ORC fluid). In this case the Q_{HX2} value is larger than Q_{HX1} if the GTs are considered, while in case of MGT Q_{HX1} becomes prevalent at high T_V values.

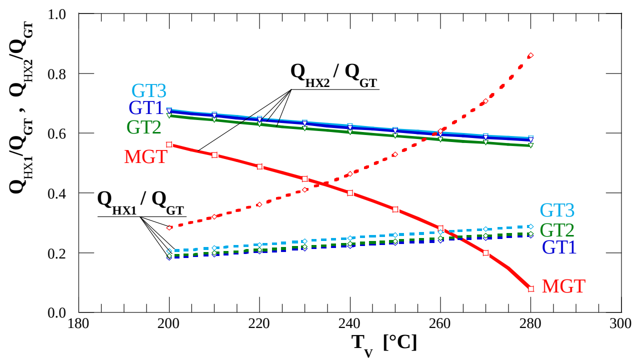

Figure 19: ORC thermal energy split between HX1 and HX2

Comparison of cogenerative performance

In order to quantify the cogenerative performance of a CHP system and to establish a criterion to compare different investigated CHP solutions, it is important to evaluate at the same time the electric and thermal production and the corresponding primary energy consumption. To this aim, the Primary Energy Saving (PES) index [20] can be introduced. The PES index is a non-dimensional parameter which represents the ratio between the amount of saved primary energy due to cogeneration and the primary energy consumption of the separate production of electric and thermal energy in reference conditions. The definition of PES is given below in Eq. (2):

$$PES = 1 - \frac{1}{\frac{\eta}{\eta_{ref}} + \frac{\tau}{\tau_{ref}}} \qquad (2)$$

where, the electric efficiency (η) and the thermal efficiency (τ) are defined according to Eqs. (1) and (3), respectively.

$$\tau = \frac{Q_{HX1} + Q_{HX2}}{F} \qquad (3)$$

Following the European Union cogeneration reference documents [21], the reference electric efficiency (η_{ref}) is here chosen equal to 52.5% (a mean value for currently installed combined cycles in Europe, considering operation at full and part load) and the reference thermal production efficiency (τ_{ref}) is fixed equal to 90%.

The calculated values of maximum PES are reported in Table 3 for each considered bottoming cycle; the corresponding values of the design key parameters varied during the investigation (PR_{BBC} for BBC, p_{low} for IBC and T_v for ORC) are also reported in Table 3. The maximum PES values are obtained in case of GT3 and with ORC (only the case of Benzene is considered in Table 3).

Table 3: A comparison of Primary Energy Saving Index for examined bottoming cycles

	BBC		IBC		ORC (Benzene)	
	PES_{max}	PR_{BBC}	PES_{max}	p_{low} (bar)	PES_{max}	T_v (°C)
GT3	0.237	4.5	0.261	0.3	0.276	280
GT2	0.205	4.5	0.231	0.3	0.247	280
GT1	0.142	4.5	0.177	0.3	0.199	280
MGT	0.114	1.5	0.187	0.5	0.181	200

The values of electric efficiency and thermal efficiency of the different investigated solutions with GT and bottoming cycles are provided in Fig. 20, where constant PES lines are also plotted.

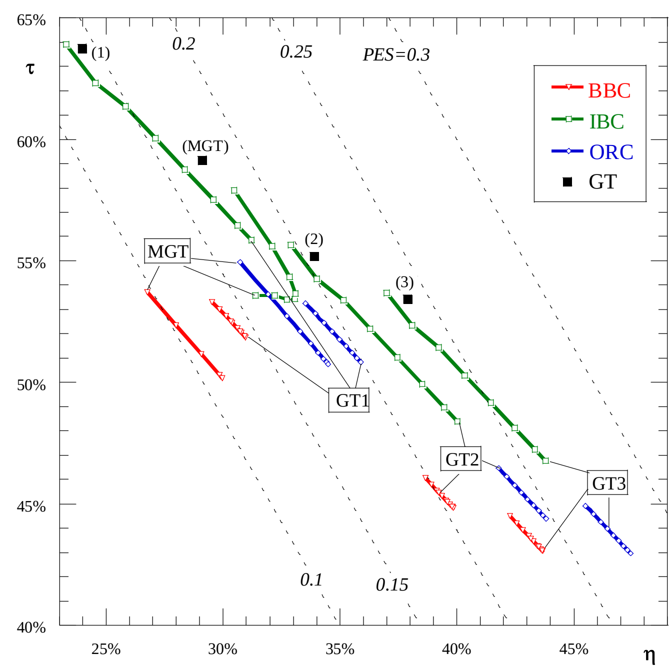

Figure 20: Thermal efficiency versus electric efficiency of BBC, IBC and ORC (with benzene) bottoming cycles, for different values of the key design parameter of the bottoming cycle

For each GT machine and each bottoming cycle, the line for PES index is obtained by varying the specific design parameter: PR_{BBC}, p_{low} and T_v for BBC, IBC, and ORC, respectively.

Figure 20 shows that for a given GT machine and a given bottoming cycle, the value of PES index does not change significantly as a function of the design parameter of the bottoming cycle. On the contrary, the electric efficiency and the thermal efficiency values are more influenced by the bottoming cycle's design parameter. Finally, it is also important to compare the introduced bottoming solutions with the simple GT in CHP mode (indicated with points in the figure); this reference case is based on the utilization of a heat exchanger which recovers thermal energy from the GT exhaust. Figure 20 highlights that the PES value increases, in comparison with the simple GT with heat recovery, if the GT+ORC based CHP system is used. The GT+IBC solution allows achieving PES values similar to the original GT with heat recovery. Whereas, the BBC integration provides PES values lower than the PES of the original GT with heat recovery.

CONCLUSIONS

The carried out investigation offers a performance comparison of the three different innovative thermodynamic solutions to recover residual heat, which otherwise would be wasted, from the conventional small size gas turbines. The cogeneration of additional electric power and thermal energy has been evaluated with reference to a low-temperature thermal utility. Based on this investigation, the following concluding remarks can be made:

- The CHP systems with the examined bottoming cycles (BBC, IBC and ORC) can be optimized in order to maximize the electric power or the thermal energy production.
- Among the different bottoming cycles, the ORC seems to offer comparatively better results in terms of the primary energy saving index and of the overall electric efficiency for a given GT type examined except for the MGT. For the MGT, CHP system with IBC shows higher performance in terms of the PES index and the overall electric efficiency.
- Among the different topping gas turbine technologies examined, the GT3 technology shows higher performance in terms of the overall electric efficiency and the PES index mainly due to high efficiency of the aeroderivative topping gas turbine. However, the topping gas turbine GT1 has comparatively higher thermal energy recovery as a result of the highest exhaust gas temperature among the four GT technologies evaluated.

NOMENCLATURE

EGT	Exhaust Gas Temperature [°C]
F	input fuel power, with reference to LHV [kW]
LHV	fuel Lower Heating Value [kJ/kg]
m	mass flow [kg/s]
p	pressure [bar]
p_{low}	IBC below ambient pressure [bar]
P	electric power [kW]
PES	Primary Energy Saving index [-]
PR	Pressure Ratio [-]
Q_{HX1}	thermal energy recovered in heat exchanger HX1 [kW]
Q_{HX2}	thermal energy recovered in heat exchanger HX2 [kW]
Q_{GT}	thermal energy available from the topping cycle [kW]
s	specific entropy [kJ/kg°C]
T	temperature [°C]
TIT	Turbine Inlet Temperature [°C]
TOT	Turbine Outlet Temperature [°C]
ΔT	temperature difference between gas and ORC fluid

Symbols

η	electric efficiency [-]
τ	thermal efficiency [-]

Subscripts

atm	atmospheric value
BBC	Brayton bottoming cycle
COLD	cold thermal source
GT	gas turbine in topping cycle
HOT	hot thermal source
HX1	high temperature heat exchanger
HX2	low temperature heat exchanger
IBC	inverted Brayton cycle
k	at ORC condenser
OR	at REC outlet
ORC	organic Rankine cycle
ref	reference value
OUT	at evaporator outlet
v	at ORC evaporator

Acronyms

BBC	Brayton Bottoming Cycle
C	Compressor
CC	Combustion Chamber
CHP	Combined Heat and Power
EGT	Exhaust Gas Temperature
GT	Gas Turbine
HX	Heat-exchanger
IBC	Inverted Brayton Cycle
ICE	Internal Combustion Engine
K	Condenser
MGT	Micro Gas Turbine
ORC	Organic Rankine Cycle
P	Pump
REC	Recuperator
T	Turbine
VAP	Evaporator

REFERENCES

[1] Bhargava, R. K., Bianchi, M., Campanari, S., De Pascale, A., Negri di Montenegro, G., and Peretto, A., 2010, "A Parametric Thermodynamic Evaluation of High Performance Gas Turbine Based Power Cycles", Transaction of the ASME, Journal of Engineering for Gas Turbines and Power, Vol. 132, pp. 022001-1/14.

[2] Romanov, V. V., Movchan, S. N., Chobenko, V. N., Kucherenko, O. S., Kuznetsov, V. V., and Shevtsov, A. P., 2010, "Performance and Application Perspectives of Air Heat Recovery Turbine Units", ASME Paper No. GT2010-23129.

[3] Korobitsyn, M., 2002, "Industrial Applications of the Air Bottoming Cycle", Energy Conversion and Management, Vol. 43, pp. 1311–1322.

[4] Wilson, D. G., 1984, *The Design of High-Efficiency Turbomachinery and Gas Turbines*, The MIT Press, Cambridge, Massachusetts, London, England, pp. 135, Chap. 3.

[5] Tsujikawa, Y., Ohtani, K., Kaneko, K., Watanabe, T., and Fujii, S., 1999, "Conceptual Recovery of Exhaust Heat from a Conventional Gas Turbine by an Inter-cooled Inverted Brayton Cycle", ASME Paper No. 99-GT-0378.

[6] Bianchi, M., De Pascale, A., and Negri di Montenegro, G., 2005, "Micro Gas Turbine Repowering with Inverted Brayton Cycle", ASME Paper No. GT2005-68550.

[7] Invernizzi, C., Iora, P., and Silva, P., 2007, "Bottoming Micro-Rankine Cycles for Micro-Gas Turbines", Applied Thermal Engineering, Vol. 27, pp. 100–110.

[8] Chacartegui, R., Sánchez D., Muñoz, J.M., and Sánchez, T., 2009, "Alternative ORC Bottoming Cycles for Combined Cycle Power Plants", Applied Energy, Vol. 86, pp. 2162–2170.

[9] Liu, B. T., Chien, K. H., and Wang, C. C., 2004, "Effect of Working Fluids on Organic Rankine Cycle for Waste Heat Recovery", Energy, Vol. 29, pp. 1207–1217.

[10] Quoilin, S., Declaye, S., and Lemort, V., 2010, "Expansion Machine and Fluid Selection for the Organic Rankine Cycle", Proc. of HEFAT2010, 7th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics, July 19-21 July, Antalya, Turkey.

[11] Karellas, S. and Schuster, A., 2008, "Supercritical Fluid Parameters in Organic Rankine Cycle Applications", Int. J. of Thermodynamics, Vol. 11 (No. 3), pp. 101-108, September.

[12] Drescher, U. and Bruggemann, D., 2007, "Fluid Selection for the Organic Rankine Cycle (ORC) in Biomass Power and Heat Plants", Applied Thermal Engineering, Vol. 27, pp. 223–228

[13] Colonna, P., Guardone, A., and Nannan, N. R., 2007, "Siloxanes: A New Class of Candidate Bethe-Zel'dovich-Thompson Fluids", Physics of Fluids, Vol. 19, 086102.

[14] Spinelli, A., Dossena, V., Gaetani, P., Osnaghi, C., and Colombo, D., 2010, "Design of a Test Rig For Organic Vapours", ASME Paper No. GT2010-22959.

[15] Mago, P. J., Chamra, L. M., Srinivasan, K., and Somayaji, C., 2008, "An Examination of Regenerative Organic Rankine Cycles Using Dry Fluids", Applied Thermal Engineering, Vol. 28, pp. 998–1007.

[16] Campanari, S. and Macchi, E., 2003, "Technical and Tariff Scenarios Effect on Microturbine Trigenerative Applications", ASME Paper No. GT2003-38275.

[17] Bianchi, M. and De Pascale, A., 2011, "Bottoming cycles for electric energy generation: parametric investigation of available and innovative solutions for the exploitation of low and medium temperature heat sources", Applied Energy, Vol. 88, pp. 1500-1509.

[18] GateCycle™ , 2007, version 6.0, GE Energy.

[19] Colonna, P. and van der Stelt, T. P., 2004, "FluidProp: A Program for the Estimation of Thermophysical Properties of Fluids", software.

[20] Directive 2004/8/EC of the European Parliament and of the Council.

[21] Commission Decision, 21 December 2006 - Establishing Harmonised Efficiency Reference Values for Separate Production of Electricity and Heat in Application of Directive 2004/8/EC of the European Parliament and of the Council.