# COMPARISON BETWEEN AIR BOTTOMING CYCLE AND ORGANIC RANKINE CYCLE AS BOTTOMING CYCLES

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## ABSTRACT
*Two bottoming cycles are analysed in this paper, the Air Bottoming Cycle (ABC) and the Organic Rankine Cycle (ORC). A comparison of thermodynamic performance between the cycles is given. Special attention is paid to choosing the component specifications on a realistic basis and the configurations so that an economic optimum for the cycles can be anticipated. Two cases for topping engines are investigated: a small-scale (7.8 $MW_e$) gas turbine with the exhaust temperature of 534 °C, and a large-scale (16.8 $MW_e$) diesel engine with 400 °C exhaust temperature. For both cases, two applications are considered: power generation only, and cogeneration of heat and power. The sensitivity of the performance against the main cycle parameters is also presented for both cycles. Considerations of applying high speed technology to both cycles are given. The term high speed technology refers here to a design where the turbomachine(s) and the electric machine (generator in this case) have a common shaft that is rotating at an optimum speed determined by the turbomachine(s).*

## KEYWORDS
*Air bottoming cycle, Organic Rankine cycle, Gas turbine, Diesel engine, High speed technology*

## 1. INTRODUCTION

In the quest for higher energy conversion efficiencies, combined-cycle technology offers an unbeaten concept. As a formulation of the Second Law of Thermodynamics, the operation of all heat engines is based on higher-temperature heat addition and lower-temperature heat rejection. Consequently, the rejected heat from the topping cycle can be implemented as a source for the bottoming one.

The conventional gas turbine combined cycle utilising the gas turbine as the topping "cycle" and the steam Rankine cycle as the bottoming one has become established technology nowadays. With advanced gas turbines and multiple pressure levels in the steam cycle, power generating efficiencies up to 60 % are quoted [1]. However, this concept is not being offered in small-scale applications, which gives the floor to alternative technologies. This paper considers the use of a gas turbine and a diesel engine as the topping engines for the Air Bottoming Cycle (ABC) and the Organic Rankine Cycle (ORC). The ABC is generally considered the most competitive for small and medium-scale power generation at sufficiently high temperature levels, whereas the ORC is considered at its best when utilising waste heat that is released at a low temperature. Consequently, the selection of the higher exhaust gas temperature-level gas turbine and lower temperature-level diesel engine as topping engines is expected to give information about the competitiveness between the ABC and the ORC at conditions close to their typical application areas.

### 1.1 Air Bottoming Cycle

The air bottoming cycle is a Brayton cycle where the working fluid is air. Differently from the externally-fired gas turbines with an exothermic reaction as a heat source, the ABC utilises heat that 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. Figure 1 shows as an example a schematic layout of an ABC with one intercooler.

The working fluid in the ABC is air and thus it can be utilised for external purposes at different temperature levels, having the extraction before or after the recuperator for instance. Having been exposed to high temperature (up to 550 °C) in the recuperator, the air is sterilised, which makes it suitable for sensitive processes, as well. Pure air as the working fluid in the ABC enables lower temperatures without a risk of condensation, compared to the topping cycle exhaust gases. Letting a fraction of the compressed and cooled air expand in a separate turbine yields low-temperature air, which makes it possible to configure the ABC also for cooling purposes.

In cogeneration, heat from the exhaust air after the bottoming cycle as well as high enough temperature-level heat from intercooling can be utilised for heating water, for instance. This is the case for the heat from the topping cycle exhaust gases after the recuperator, as well.

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 [4], Hirs et al. [5] and Arriagada et al. [6]. Kaikko et al. [7] have presented a thermodynamic analysis for a trigenerative ABC (for power, heating and cooling) where a reversed Brayton

cycle is integrated into an intercooled ABC to provide cold airflow. A study by Kaikko [8] has demonstrated the competitiveness of the ABC against steam Rankine cycle in small and medium-scale applications.
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 [9]. Since 1997, this concept has been developed by Kværner Energy in Norway [10]. 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 [11].

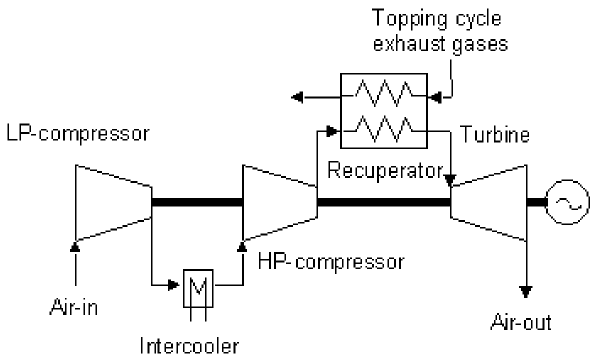


**Figure 1 : Layout of an open air bottoming cycle with one intercooler.**

### 1.2 Organic Rankine Cycle

An organic Rankine cycle is a Rankine cycle in which an organic fluid, such as toluene, is used instead of water. Figure 2 shows the sketch of the cycle in a T,s - diagram. The shape of the saturation curve of toluene differs greatly from that of water. The state of toluene vapour moves further away from the saturation curve during the expansion in the turbine. For this reason only small superheat is required since no droplets can condense in the low-pressure part of the turbine. The cycle can also be operated close to the critical pressure even at modest temperatures. Therefore, the heat needed for evaporation is small in relation to the heat needed for preheating and superheating. This allows efficient heat recovery and high turbine entry temperature even with only one pressure level vaporiser. This is shown in Figure 3. The corresponding curve for water vapour is shown for comparison. In addition, reasonably high efficiency can be achieved in the ORC even with a compact single stage turbine, since the specific enthalpy drop of toluene in the turbine is lower than that of water steam. For the reasons stated above ORC is especially well-suited for small scale power plants and for the utilisation of waste heat released at a low temperature.
The main process components of the ORC are shown in Figure 4. They include a once-through vaporiser, condenser, pre-feed pump and a high speed turbogenerator, in which turbine, generator and main feed pump are combined to form a single-shaft hermetic unit. Since the toluene vapour is highly superheated after the turbine (Figure 2), a recuperator can be used to preheat the toluene which is fed to the vaporiser, increasing the thermal efficiency of the cycle. The heat released from the cycle in the condenser can be utilised as district or process heat.

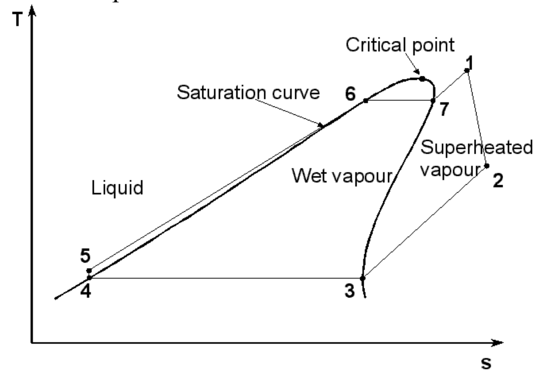


**Figure 2 : Sketch of the organic Rankine cycle in T,s diagram. 1→2 expansion in the turbine; 2→3 desuperheating; 3→4 condensation; 4→5 pressure increase in the feed pump; 5→6 preheating; 6→7 evaporation; 7→1 superheating.**

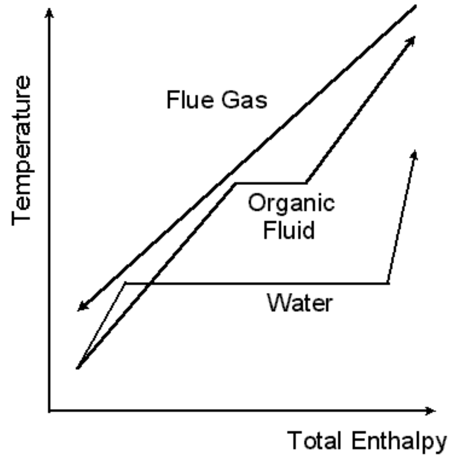


**Figure 3 : Temperature diagram of the vaporiser. The temperature of the organic fluid follows well the temperature of the flue gas, since the heat needed for vaporisation is small. The corresponding curve for water is shown for comparison.**

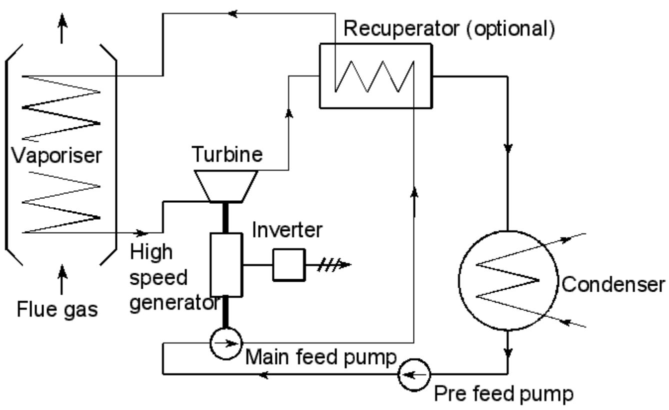


**Figure 4 : The main components of ORC.**

### 1.3 High Speed Technology

The term high speed technology is here used to refer to a system where the turbomachine and the electric machine (motor or generator) have a common rotor rotating faster than the synchronous speed. Many applications, such as kinetic compressors, small gas turbines and small expansion turbines require high speeds. Usually such speeds are produced by using a gearbox and a conventional electric machine. High speed technology is a solution that does away with the gearbox. A high speed electric machine is directly coupled to the rotor of the turbine, compressor or pump. An inverter is used to connect the electric machine to the network. By using process-fluid lubricated gas- or hydrodynamic bearings or magnetic bearings it is possible to make the high-speed application completely oil-free and hermetic.

The research on high speed technology was initiated at Lappeenranta University of Technology in 1981. The developed technology has so far been applied to a low pressure ratio compressor [12], to an ORC power plant [13], to high pressure pumps [14], to a refrigeration process [15] and to a reversed Brayton process [16]. The rotational speed for these applications varies between 15.000 and 250.000 rpm.

The turbogenerators of the ORC power plant investigated in this study are proposed to be built using high speed technology. Toluene lubricated hydro-dynamic tilting pad bearings were selected, as was the case in the earlier ORC plants [13]. There is no risk of contamination of the cycle fluid with oil, which is important, since oil would decrease the thermal stability and heat transfer properties of toluene. Hermetic operation ensures that the process fluid will not leak out and cause any fire or environmental risk. In larger plants several standard size turbogenerator units are connected in parallel to get the desired output.

The same technology could be applied to the ABC as well. The size and cost of the ABC could be reduced by integrating the compressor, turbine and generator into a single-shaft high speed unit, since the high speed generator is compact and no gearbox would be needed. In the ABC magnetic and/or gas bearings would come into question.

## 2. CASES THAT HAVE BEEN STUDIED

In the analysis, two cases for topping engines have been investigated: a small-scale gas turbine (Alstom Tempest) with the power of 7.8 $MW_e$ and exhaust temperature of 534 °C, and a large-scale intercooled turbocharged diesel engine (Wärtsilä 18V46) with 16.8 $MW_e$ power and 400 °C exhaust temperature. For each topping engine, both the ABC and ORC have been studied as bottoming cycle alternatives. Altogether, this makes four basic configurations. Since for each configuration two applications have been considered: power generation and cogeneration of (district) heat and power, the total number of cases amounts to eight. This is illustrated in Figure 5. The specifications for the selected equipment are given in the following chapter.

For the selected cases, thermodynamics and steady-state performance have been studied at design-point operation. The sensitivity of the performance against main cycle parameters is also presented for both bottoming cycles. Comparisons are made between the bottoming cycle variants.

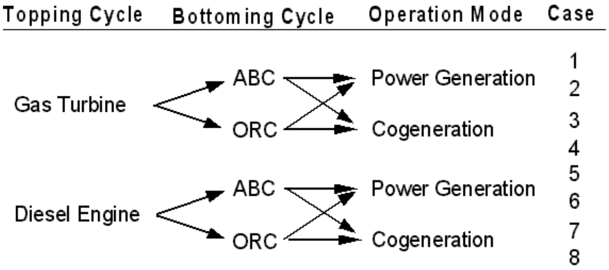


**Figure 5 :Cases that have been studied in the analysis.**

## 3. MODELLING AND OPTIMISATION

All calculations are based on the ISO operating 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.

Specifications for the topping engines are given in Table 1. For the diesel engine, these values are valid for pressure losses up to 30 mbar on the exhaust gas side. For the gas turbine, the values assume zero outlet losses and, consequently, any excess pressure loss (from the ABC recuperator, ORC vaporiser, or district heat exchanger) has a deteriorating impact on the performance of the topping engine.

**Table 1 : Performance specifications for the topping engines. The gas turbine (Alstom Tempest) data from GateCycle [17], the diesel engine (Wärtsilä 18V46) data from the manufacturer. The $CO_2$ emission data is calculated.**

| | | Alstom Tempest | Wärtsilä 18V46 |
|---|---|---|---|
| Net power output | MW | 7.75 | 16.8 |
| Net el. efficiency | % | 31.4 | 45.5 |
| Exhaust gas flow | kg/s | 28.6 | 28.0 |
| Exhaust gas temp. | °C | 534 | 400 |
| Spec $CO_2$ emission | g/kWh | 669 | 581 |

As fuel, the diesel engine uses light fuel oil. It contains sulphur and, consequently, the condensation-risk of sulphuric acids in the exhaust gases has been taken into account by limiting the minimum temperature to 120 °C. The gas turbine uses natural gas, which is virtually sulphur-free and, therefore, the minimum temperature of the exhaust gases in this case has been selected to be lower, 65 °C. Although the exhaust gas temperatures are well beyond these limits in power generation, these values limit the amount of heat to be recovered from the exhaust gases in cogenerative operation.

The performance of the (gas-liquid) heat exchangers for district heating is determined by the minimum temperature differences. For these non-contact heat exchangers of the counterflow type, the difference has been selected to be 15 °C. The return and supply temperature for district heating water has been assumed to be 50 °C and 80 °C, respectively. The return temperature and the minimum temperature difference allow the utilisation of the heat from the gas turbine exhaust gases down to 65 °C, which is also the selected minimum for the gas turbine. For the diesel exhaust gases, the temperature difference has no relevance since the minimum allowable temperature remains so much higher.

For the diesel engine, intercooling (from 205 °C to 65 °C) as well as cooling water cooling (from 90 °C to 80 °C) constitute an additional source for district heat.

In the power generation mode, 15 °C cooling water is applied to the ABC intercooling and the ORC condenser.

### 3.1 Air Bottoming Cycle

The specifications for the ABC 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 that has been selected for transferring heat from the topping cycle to the bottoming one. These specifications are presented in Table 2.

**Table 2 : Computational parameters for ABC calculations.**

| | |
|---|---|
| Compressor (unit) polytropic efficiency | 91 % |
| Turbine polytropic efficiency | 86 % |
| Recuperator effectiveness | 93 % |
| Recuperator air-side pressure loss | 2 % |
| Recuperator exhaust gas-side pressure loss | 2 % |
| Intercooler air-side pressure loss | 2 % |
| District heat exch. air/gas side pressure loss | 2 % |
| Mechanical efficiency | 99.5 % |
| Gearbox efficiency | 98.5 % |
| Generator efficiency | 98 % |
| Auxiliary power of ABC net power | 1 % |

In power generation, intercooling is implemented using cooling water. The condensation risk in the intercooler at elevated pressures has been taken into consideration by determining the minimum allowable air temperature in the intercooler according to the saturating temperature increased by a temperature margin of 20 °C. The ultimate lower limit for the temperature is set by the cooling water temperature (15 °C) added by the selected temperature difference of 15 °C, resulting to a value of 30 °C. In cogeneration, the heat from intercooling is utilised as district heat. In this case the intercooling temperature-level is restricted to 65 °C. Although the air could be further cooled down by using a second-level intercooler and cooling water, this was not considered feasible due to the small temperature decrease available for further cooling.

In the modelling, 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.

Characteristic for Brayton cycles, 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 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.

The modelling software GateCycle [17] has been implemented to simulate the performance of different ABC configurations. Performance data for the topping gas turbine has been gained from the software database. For the cases with the diesel engine, actual exhaust gas data has been simulated in the software.

### 3.2 Organic Rankine Cycle

The modelling of ORC has been performed using process calculation programs developed at Lappeenranta University of Technology. The properties of organic fluids have been calculated using exact real gas functions [18].

Common organic fluids that have previously been used in ORC power plants include various CFC-compounds (for example R11, R113 and R114), fluorinol, isobutane and toluene [19]. CFC-compounds were not considered at all in this study due to their harmful effect on the ozone layer. Toluene ($C_6H_5CH_3$) has been selected as a cycle fluid since its properties suit well the required temperature range. In addition its thermal stability is high and it is quite inexpensive. Toluene has also been successfully used in ORC power plants earlier [20].

A subcritical cycle has been selected for simplicity. Toluene easily withstands temperatures up to 400 °C, but to add safety and as there is no need for great superheat, a lower temperature can be used. Therefore the temperature and pressure of toluene vapour at the turbine inlet has been selected to be 340°C and 34 bar respectively. The performance of the components for the ORC has been chosen keeping the realistic and economic plant configuration in mind. Preliminary component design has been made by component suppliers and subcontractors for the vaporiser, heat exchangers, pre-feed pump, high speed turbogenerator and inverter. Using this information the component performance for the ORC calculation has been selected according to Table 3.

**Table 3 :Component performance for ORC calculations.**

| | |
|---|---|
| Turbine isentropic efficiency | 80% |
| Feed pump efficiency | 50% |
| Exhaust gas side pressure loss, in total | 3 kPa |
| Pipe system and vaporiser toluene side pressure loss, in total | 360 kPa |
| Recuperator vapour side pr. loss (if used) | 5 kPa |
| Condenser vapour side pressure loss | 5 kPa |
| Condenser temperature difference | 7.5 °C |

In addition the following losses are calculated and accounted for:

- bearing friction loss
- high speed generator gas gap friction loss
- generator and inverter efficiency
- internal consumption

Recuperator increases the cycle efficiency, but adds extra cost and produces undesirable pressure loss. It is thermodynamically feasible to use the recuperator only when the preheating part of the vaporiser becomes the limiting factor in the vaporiser design. This is the case when the heat source enters the vaporiser at a relatively high temperature, which is shown with the dashed line in Figure 6. If the pinch point is the limiting factor in the design of the vaporiser, then the recuperator is for little use, since it would only substitute the low temperature part of the vaporiser, where the temperature difference is already high. This occurs when the heat source enters the vaporiser at a low or modest temperature, which is shown with the continuous line in Figure 6. If a recuperator was used in this case, the heat source would leave the vaporiser at a higher temperature. On the other hand, the recuperator can easily be used to increase the flue gas minimum temperature, should this limit the vaporiser design. The choice to use or not to use the recuperator is further affected by economical considerations.

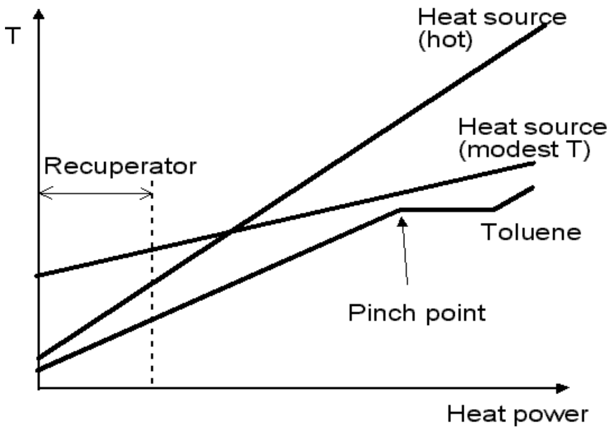


**Figure 6 : Temperature diagram of the ORC vaporiser. When the heat source is at a high temperature (dashed line) the recuperator can be used to increase the output of the ORC, but when the heat source enters the vaporiser at a low temperature (solid line), the recuperator is of little use.**

In the following calculations the whole ORC power plant has been optimised for each case, so the main process variables vary slightly. The recuperator has been used in cases where it was thermodynamically feasible. The minimum temperature difference in the vaporiser (at pinch point or in the preheater depending on the case) has been selected to be as high as about 30 - 40 °C in order to keep the vaporiser compact and cheap. In some cases the turbine entry temperature has been lowered to 326 °C to enable this.

## 4. SENSITIVITY STUDIES

### 4.1 ABC

The implementation of intercooling between the compressor units helps to increase the amount of heat to be recovered in the ABC. Furthermore, intercooling reduces compressor work and increases the available work from the turbine. 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.

As a result from simulations with the gas turbine and the diesel engine as the topping engine in power generation, Figure 7 presents the net power output of the ABC against the pressure ratio for a different number of intercoolers. For the cases with the diesel engine, the minimum pressure ratios are limited by the selected minimum temperature (120 °C) for the topping cycle exhaust gases. The superior performance of the ABC when using the gas turbine as the topping engine is mainly due to the higher exhaust gas temperature. The optimum pressure ratios for power generation increase with the increasing number of intercoolers for both topping cases. For the case with gas turbine, for instance the optimum pressure ratio is 4.2 when no intercooling is implemented, 6.2 for the case with 1 intercooler and 7.3 with two intercoolers. The curves become also flatter, which decreases the effect of the pressure ratio on the power output around the optimum. Figure 7 also points out the rapidly damping effect of intercooling due to pressure losses in the intercooler: Implementing the non-intercooled ABC increases the power output by 24.3 % in the gas-turbine case. The first intercooler adds 4.5 % points more but the second one only 1.7 % points. Based on this, the configuration with 1 intercooler was selected for further studies.

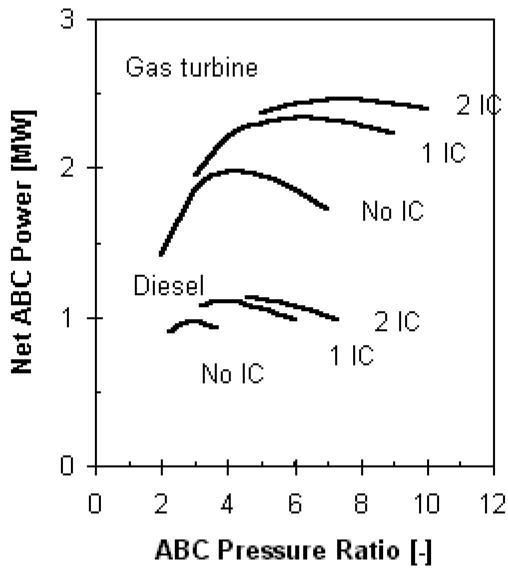


**Figure 7: The impact of intercooling on the net ABC power output. The figures represent the performance ratings of the bottoming cycle only.**

The sensitivity of the ABC performance in power generation against turbomachinery efficiencies, topping cycle exhaust gas temperature and recuperator effectiveness was examined for the configuration with one intercooler and the gas turbine as the topping engine. The results of the simulations are presented in Figure 8.

The turbine inlet temperature to the ABC and, consequently, the topping cycle exhaust gas temperature has a dominating impact on the ABC performance (as also indicated in Figure 7). This suggests favouring topping engines or cycles with a high exhaust gas temperature. Another implication is that the use of low-temperature heat will inevitably lead to low cycle performance, sometimes too low to justify the utilisation of the air bottoming technology.

Turbomachinery efficiencies have also a strong impact on the cycle performance, as a consequence of 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 exhaust gas temperature, while the power output becomes less sensitive towards the pressure ratio around the optimum. For recuperator effectiveness, the increase in the optimum pressure ratio and the impact on cycle performance are less evident.

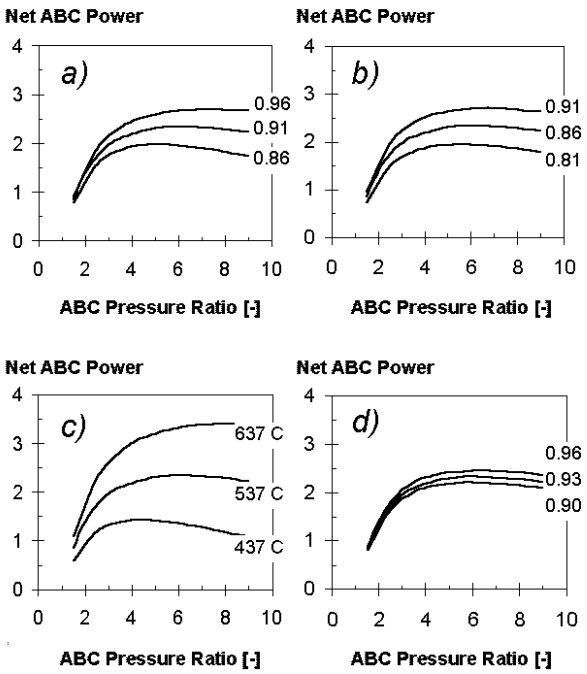


**Figure 8: Sensitivity of the performance of the ABC with one intercooler against a) polytropic efficiency of the compressor units b) turbine units c) topping cycle exhaust gas temperature d) recuperator effectiveness. The topping engine is the gas turbine.**

### 4.2 ORC

The sensitivity of ORC performance against condensing water outlet temperature (Figure 9a) and turbine isentropic efficiency (Figure 9b) was studied. In these analyses the heat source and the state of the toluene vapour at the turbine inlet were kept unchanged. The cycle mass flow was adjusted to keep the logarithmic mean temperature difference in the vaporiser nearly constant. This leads to an approximately equally tightly dimensioned vaporiser. A recuperator was used in the case of the gas turbine, but it was omitted for the diesel engine due to the lower heat source temperature.

Figure 9 shows how the output of the ORC decreases with increasing condensing water temperature due to increasing condenser pressure. The tendency is typical for any Rankine cycle. Furthermore, a nearly linear dependence on turbine efficiency is observed. High turbine efficiency is naturally desired, but turbine performance is clearly not as crucial for the ORC as it is for the ABC.

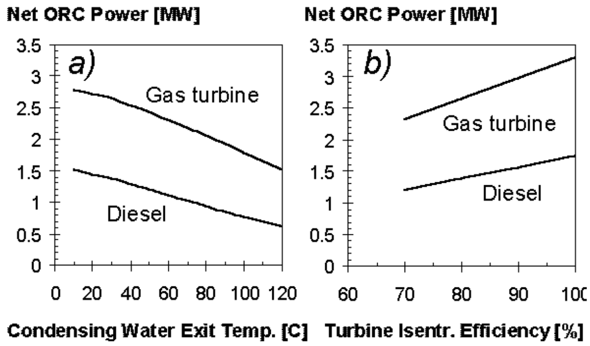


**Figure 9: Sensitivity of the ORC performance against a) condensing water exit temperature and b) turbine isentropic efficiency for both topping engines.**

## 5. PERFORMANCE COMPARISON

Figure 10 shows the impact of topping engine exhaust gas temperature on the net power output from the optimised bottoming cycles both in power generation and cogeneration. The figure has been produced using the mass flow and composition of the exhaust gases from the selected gas turbine and varying the temperature level. Although the typical range for the exhaust gas temperatures is up to 600 °C, higher temperatures are shown in Figure 10 to better characterise the nature of the bottoming cycles.

In the ORC, toluene vapour temperature cannot be increased over 340°C to maintain thermal stability. Consequently, at high enough exhaust gas temperatures, the cycle efficiency remains constant. Meanwhile, the heat input to the vaporiser is proportional to the inlet temperature of the exhaust gas, causing the power output from the ORC to increase only linearly with the increasing exhaust gas temperature. For the ABC, increasing the exhaust gas temperature increases the heat input to the cycle, but also the cycle efficiency due to increased heat input temperature. Consequently, power output from the ABC increases more at high exhaust gas temperatures than with the ORC. As a result, the ORC is superior in power generation at

temperature levels up to 680 °C, while the ABC dominates at higher temperatures.
Cogeneration of power and heat penalises the power output from the ORC more than with the ABC. The reason for this is that in cogeneration, the turbine expansion in the ORC ends at essentially elevated pressure (when compared to the power generation mode) to gain an adequate temperature level for the district heating water, while for the ABC the reduction in turbine expansion work is caused by a minor pressure loss in the district heating heat exchanger. As a result, in cogenerative application power output levels are very close to each other in the ABC and ORC up to 550 °C, after which the ABC shows better performance.
Tables 4 and 5 present the performance of the studied eight cases (combined cycles) with optimised ABC and ORC as bottoming cycles. The results are well in line with Figure 10 in respect with power output: Compared to the ABC, the ORC applications feature slightly better electric efficiencies (up to 0.9 % points) in the power generation mode, whereas in cogeneration the electric efficiencies between the ABC and ORC applications are very close to each other. The implementation of bottoming cycle technology offers a means for increased power generation with the same amount of fuel, or decreased fuel consumption with the same power output. This can be indicated by the specific $CO_2$ emission for power generation as given in tables 4 and 5. The selected bottoming cycle concepts offer up to 24 % reduction in $CO_2$ emissions. With regard to the total efficiency of heat and power generation, more heat can be recovered as district heat from a Rankine-based ORC and the remaining topping cycle exhaust gases than from a Brayton-based ABC and the corresponding topping cycle exhaust gases. Consequently, ORC applications feature 4 to 5 % point better total efficiencies than ABC applications with both topping engines.

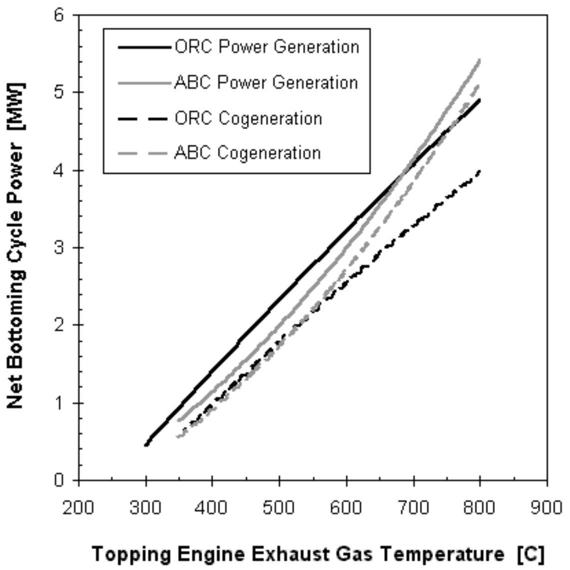


**Figure 10: Net power output from the bottoming cycle against the topping engine exhaust gas temperature.**

**Table 4 : Summary of the thermodynamic performance of the combined cycle with the gas turbine (Alstom Tempest) as the topping cycle and the ABC and ORC as the bottoming cycle.**

| *Gas turbine + bottoming cycle* | | ABC | ORC |
|---|---|---|---|
| ***Power generation*** | | | |
| Net power output | MW | 9.99 | 10.21 |
| Net electric efficiency | % | 40.4 | 41.3 |
| $CO_2$ emission for power | g/kWh | 519 | 508 |
| Reduction in $CO_2$ em. due to bottoming cycle | % | 22.4 | 24.1 |
| ***Cogeneration of power and heat*** | | | |
| Net power output | MW | 9.64 | 9.66 |
| District heat output | MW | 11.19 | 12.35 |
| Power-to-heat ratio | - | 0.86 | 0.78 |
| Net electric efficiency | % | 39.0 | 39.1 |
| Total efficiency | % | 84.3 | 89.0 |
| $CO_2$ emission for power | g/kWh | 538 | 537 |
| $CO_2$ emission for power and heat | g/kWh | 249 | 236 |

**Table 5 : Summary of the thermodynamic performance of the combined cycle with the diesel engine (Wärtsilä 18V46) as the topping cycle and the ABC and ORC as the bottoming cycle.**

| *Diesel engine + bottoming cycle* | | ABC | ORC |
|---|---|---|---|
| ***Power generation*** | | | |
| Net power output | MW | 17.90 | 18.11 |
| Net electric efficiency | % | 48.5 | 49.0 |
| $CO_2$ emission for power | g/kWh | 545 | 539 |
| Reduction in $CO_2$ em. due to bottoming cycle | % | 6.2 | 7.2 |
| ***Cogeneration of power and heat*** | | | |
| Net power output | MW | 17.67 | 17.78 |
| District heat output | MW | 11.85 | 13.07 |
| Power-to-heat ratio | - | 1.5 | 1.4 |
| Net electric efficiency | % | 47.8 | 48.1 |
| Total efficiency | % | 79.9 | 83.5 |
| $CO_2$ emission for power | g/kWh | 552 | 549 |
| $CO_2$ emission for power and heat | g/kWh | 331 | 317 |

## 6. CONCLUSIONS

This paper compared two bottoming cycle technologies that are considered to be competitive for small-scale energy conversion or for low temperature level applications: an Air Bottoming Cycle (ABC) and an Organic Rankine Cycle (ORC).
Using the selected specifications, the ORC is superior to the ABC in power generation mode both with the selected gas turbine and the diesel engine as topping engines. Accordingly, a combined cycle applying the ORC shows a better performance (electric efficiency) than the ABC in power generation. Cogeneration of power and heat penalises the power output from the ORC more than with the ABC. Consequently, the combined cycle electric efficiencies of ABC and ORC

applications come very close to each other in the cogenerative mode. With regard to the total efficiency of heat and power generation, ORC applications feature better performance than the ABC with both topping engine alternatives.
The sensitivity of the performance of the ABC and ORC is very high against the topping cycle exhaust gas temperature, and they are close to each other at the temperature range determined by the selected topping engines. The efficiencies of the ABC turbomachinery have a strong effect on the cycle performance. For the ORC, the impact of the turbine efficiency on cycle performance is weaker.

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## REFERENCES

[1] Gas Turbine World Handbook 1999-2000, Vol 20, Pequot Publishing Inc.,Fairfield, CT, USA (1999)
[2] Farrell W.M., Air Cycle Thermodynamic Conversion System, US patent 4.751.814A, General Electric Company, New York, USA (1988)
[3] Wicks F., The Thermodynamic Theory and Design of an Ideal Fuel Burning Engine, Proceedings of the 25th Intersociety Energy Conversion Engineering Conference IECEC'90, Vol 2, pp 474-481, Boston, USA (1991)
[4] Weston K.C., Dual Gas Turbine Combined Cycles, Proceedings of the 28th Intersociety Energy Conversion Engineering Conference IECEC'93, Vol 1, pp 955-958, Boston, USA (1993)
[5] Hirs G.G., Wagener M.T.P.A. and Korobitsyn M.A., Performance Analysis of the Dual Gas Turbine Combined Cycle, AES-Vol 35, pp 255-259, Thermodynamics and the Design, Analysis and Improvement of Energy Systems, ASME 1995 (1995)
[6] Arriagada J. and Assadi M., Air bottoming cycle for Gas Turbines, ISME 2000, Tehran, Iran (2000)
[7] Kaikko J. and Hunyadi, L., Air Bottoming Cycle for Cogeneration of Power, Heat and Cooling, Accepted for the 2nd International Heat Powered Cycles Conference HPC'01, Paris, France (2001)
[8] Kaikko J., Air Bottoming Cycle, an Alternative to Combined Cycles, Final Report, Department of Energy Technology, Royal Institute of Technology, Stockholm, Sweden (2000)
[9] Bolland O., Førde M. and Hånde B., Air Bottoming Cycle: Use of Gas Turbine Waste Heat for Power Generation, Journal of Engineering for Gas Turbines and Power, Vol 118, pp 359-368 (1996)
[10] Kværner Energy, Air Bottoming Cycle Development Project, Kværner Energy, Lysaker, Norway (1998)
[11] Korobitsyn M.A., Industrial Applications of the Air Bottoming Cycle, ECOS'99, June 8-10, Tokyo, Japan (1999)
[12] Larjola J., Backman J., Esa H., Pitkänen H., Sallinen P. and Honkatukia J., Centrifugal compressor design and testing in Finnish high speed technology, ASME International mechanical engineering congress & expo, November 5-11, Orlando, Florida, USA (2000)
[13] Jokinen T., Larjola J. and Mikhaltsev I., Power Unit for Research Submersible, Elecship 98; International conference on electric ship, proceedings pp 114-118, September 1st, Istanbul (1998)
[14] Larjola J., Alamäki J. and Sallinen P., High speed pumps in water jet cutting and in water jetting, Proceedings of the 4th Pacific Rim International Conference on Water Jet Technology, Part l: Future applications of the jetting technology (Invited papers), pp 47-56, April 20-22, Shimizu, Japan (1995)
[15] Kuosa M., Backman J., Talonpoika T., Sallinen P. Larjola J. and Honkatukia J., Refrigeration process with high speed technology, ASME '98 Turbo Expo, ASME paper No. 98-GT-532, 2-5 June, Stockholm, Sweden (1998)
[16] Backman J., On the Reversed Brayton Cycle with High Speed Machinery (dissertation), Research Papers 56, 130 p., Department of Energy Technology, Lappeenranta University of Technology, Lappeenranta, Finland (1996)
[17] GateCycle for Windows, Enter Software Inc., Version 5.22.0.r, Menlo Park, California, USA (1998)
[18] Honkatukia J., Static model of an ORC-process (ORC-prosessin staattinen malli, in Finnish), Series EN B-84, 22 p., Department of Energy Technology, Lappeenranta University of Technology, Lappeenranta, Finland (1994)
[19] Larjola J., Lindgren O. and Vakkilainen E., Electricity from waste heat (Sähköä hukkalämmöstä, in Finnish), Series D:194, Energy Department, Ministry of Trade and Industry, Finland (1991)
[20] Lacey D. and Prasad A, Five Organic Rankine Cycle systems installed at field test sites, Modern Power Systems, October (1981)

## NOMENCLATURE

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|---|---|
| ABC | air bottoming cycle |
| CFC | chlorofluorocarbons |
| ORC | organic Rankine cycle |