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## Optimization of advanced liquid natural gas-fuelled machineries for a high-speed ferry

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# Optimization of advanced liquid natural gas-fuelled machineries for a high-speed ferry

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November 2011

**Optimization of advanced liquid natural gas-fuelled machineries for a high-speed ferry**

2011

By

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## Summary

This report is aimed at designing and optimizing combined cycles in order to define the most suitable machinery system for the future high-speed Incat ferry operated by Mols-Linien. For this purpose, an in-house numerical simulation tool called DNA (Dynamic Network Analysis) and a genetic algorithm-based optimization routine are used. The top cycle is modeled as the aero-derivative gas turbine LM2500, while the following five options for bottoming cycles are modeled:

- Single pressure steam cycle
- Dual-pressure steam cycle
- ORC using Toluene as the working fluid with an intermediate oil loop
- ABC with inter-cooling
- CO<sub>2</sub> transcritical Rankine cycle

The combined cycles are simulated with a power requirement of 18 MW, operating the gas turbine at part load while optimizing the bottoming cycle in regards to combined cycle efficiency. For comparison, a second scenario is simulated operating the gas turbine at full load, thus maximizing the power output.

Considering combined cycle operation with a load of 18 MW, the results suggest that the thermal efficiencies of the combined gas and steam cycles are 46.3 % and 48.2 % for the single pressure and dual pressure steam cycles, respectively. The combined cycles with ORC, ABC and transcritical CO<sub>2</sub> as bottoming cycles obtained thermal efficiencies of 45.6 %, 41.9 % and 42.1%, respectively.

The dual pressure steam cycle is a complex and spacious system. The single pressure steam cycle relieves some complexity and components, but has lower efficiencies as well as both systems require water treatment and manual handling. The ORC is generally viewed as a less complex system, and can be automatically controlled. The pressure levels are also substantially lower than for a steam cycle, reducing mechanical stresses on components and pipes. The oil loop ensures more stable operation of the bottoming cycle during part load, compared with steam, where the reduced temperature of the heat source has adverse effect on the steam cycle power output. Still, the ORC technology is less mature, and the components are likely to be more costly as it is less developed. The ABC system applies well-known principles with low complexity, but is less competitive considering efficiency. The CO<sub>2</sub> bottoming cycle showed results comparable with the ABC solution. While the efficiencies are comparable, the CO<sub>2</sub> cycle demands very high pressures and is not a developed technology

Some preliminary evaluations of component size are done, but no conclusion could be made concerning heat exchanger dimensions. The combined cycle solution with ORC as bottoming cycle can be recommended as a machinery solution that will provide good thermal efficiencies and is less complex. While this being said, it is necessary that further work is done evaluating the component sizes for the ORC system, assuring that the weight and space requirements are held, while the thermal efficiency is kept high. If the combined cycle efficiency drops as a consequence of adapting the heat exchanger, the ABC system should be considered as an alternative.

# Numenclature

## *Abbreviations*

|                 |                               |
|-----------------|-------------------------------|
| ABC             | Air Bottoming Cycle           |
| C               | compressor                    |
| CC              | combined cycle                |
| CO <sub>2</sub> | Carbon Dioxide                |
| DNA             | Dynamic Network Analysis      |
| EC              | economizer                    |
| EV              | evaporator                    |
| FW              | feed-water                    |
| GA              | Genetic Algorithm             |
| GT              | gas turbine                   |
| HP              | high-pressure                 |
| HPEC            | high-pressure economizer      |
| HPEV            | high-pressure evaporator      |
| HPSH            | high-pressure super-heater    |
| HRSG            | heat recovery steam generator |
| IC              | inter-cooler                  |
| IHE             | internal heat exchanger       |
| LNG             | liquefied natural gas         |
| LP              | low-pressure                  |
| LPEC            | low-pressure economizer       |
| LPEV            | low-pressure evaporator       |
| LPSH            | low-pressure super-heater     |
| ORC             | Organic Rankine Cycle         |
| P               | pressure                      |
| T               | temperature                   |
| pl              | pressure level                |

## *Greek letters*

|        |            |
|--------|------------|
| $\eta$ | efficiency |
|--------|------------|

## *Subscript*

|    |               |
|----|---------------|
| ai | auto-ignition |
| b  | boiling       |
| c  | critical      |
| m  | melting       |
| th | thermal       |

# 1. Introduction

The marine industry has stringent demands on emission reduction to comply with, as further limitations on emission from Marpol Annex VI will come into force in the near future. This has increased the interest in machineries operated with liquefied natural gas (LNG). Gas turbines fuelled with LNG reduce the emissions of nitric oxides, sulfuric oxides and particles, but a gas turbine has a lower thermal efficiency than a diesel engine. To achieve higher efficiencies and higher net power output, a bottoming cycle extracting heat from the flue gas of the gas turbine is a viable solution. Gas turbines are much lighter and compact than the diesel engine, which allows better space utilization on ships. By increasing the efficiency with combined cycle technology, one will have both a compact and efficient alternative competitive with the diesel engine as prime movers for ships.

This report is aimed at designing and optimizing combined cycles in order to define the most suitable machinery system for the future high-speed Incat ferry operated by Mols-Linien. For this purpose, an in-house numerical simulation tool called DNA (Dynamic Network Analysis) and a genetic algorithm-based optimization routine are used. The bottoming cycles under consideration include a steam cycle, an Organic Rankine Cycle (ORC), an Air Bottoming Cycle (ABC) and a transcritical  $\text{CO}_2$  Rankine cycle. The combined cycles are simulated considering two different scenarios;

- (1) Total combined cycle power output of 18 MW
- (2) Maximum combined cycle power output

The first scenario reflects a machinery solution for the anticipated construction of a high-speed Incat catamaran planned to be installed with two separate machinery packages, each representing a combined cycle solution. A requirement for each machinery package is to achieve the highest efficiency at a power output of 18 MW, being the main operation load. The selection of gas turbines designed for marine application is limited; therefore the gas turbine selected for the topping cycle is not an optimal fit with the power output requirements. To achieve a combined cycle power output of 18 MW, the gas turbine is operated in part load condition. For simplicity, this report will only evaluate one machinery package, as they are presumed identical. In this way the scenario (1) simulation will evaluate the thermal efficiency and fuel savings by applying different combined cycles, as well as allow comparison with operation by gas turbine alone. The second scenario evaluates the potential power output and efficiency of a combined cycle, when the gas turbine is operated at full load, thereby without restrictions of total maximum output. These simulations will present how much power output is obtainable with the different combined cycle technologies.

The bottoming cycle concepts evaluated in this report have previously been subject for studies, while there are none or few papers evaluating the use of these combined cycles as prime movers for ships. The interest of this study is to simulate the different combined cycle configurations under equal operational circumstances, and to evaluate the combined cycle's applicability as machinery solutions for marine vessels.



## 1.1 Background

A combined gas and steam cycle is most commonly utilized today in larger power plants, and with increased oil prices and more stringent emission regulations there is increased focus on more energy efficient machinery solutions for ships. A gas turbine alone has lower thermal efficiency than that of the more traditional marine diesel engine, but when combined with a steam cycle, efficiencies of up to 58 % have been reported for land-based units [1]. Though showing great benefits in increased energy efficiency, the steam cycle is also a large bulky system, with a high level of complexity.

The Organic Rankine Cycle is a topic that has received a lot of attention the past decade, while the circumstance most considered is low-grade heat recovery such as geothermal or solar thermal power plants [2-5]. More recently there has been increased interest in a medium to high-temperature application of ORC [6, 7]. The ORC works in principle as a steam Rankine cycle, where a heat source evaporates the working fluid which is then expanded through a turbine, then condensed and pressurized before again being vaporized. Today there are manufacturers specialized in complete ORC systems, targeting ORC applied for waste heat recovery, geothermal and solar power production [8]. A plan to implement an ORC heat recovery system on a ship was recently presented, where it is intended to integrate the ORC system with a two-stroke diesel engine. The goal is to reduce fuel consumption of the vessel by 5-10 % [9, 10].

An ABC is an open Brayton cycle where the combustion chamber is replaced by a gas-to-air heat exchanger, extracting heat from the gas turbine exhausts. The concept was patented in 1988 by Farrell [11], but in contrast to the amount of research of bottoming cycles with other working fluids, the ABC has received limited attention. There are still a number of studies done evaluating the ABC and its potential for improving the performance of a gas turbine with emphasis on small to medium-scale power production. One incentive of researching the ABC is to model a bottoming cycle that is simpler, cheaper and easier to operate than the more common steam bottoming cycle. Bolland [12] evaluated a combined cycle consisting of the gas turbine LM2500 and ABC with a net combined cycle efficiency of 46.2%, whereas a similar evaluation by Kambanis [13] showed results of up to 47 % combined cycle efficiency. More recent studies have also been done, evaluating more complex ABC solutions, including steam injection, with promising results [14].

CO<sub>2</sub> is viewed as a promising working fluid for supercritical Brayton cycles applied as power conversion plants for nuclear reactors [15], as well as solar thermal power plants [16]. More recently, interest has increased in utilizing CO<sub>2</sub> in lower-temperature applications as a transcritical Rankine cycle often compared with ORC cycles [17-19]. The results of these studies have suggested both better and worse performance of the transcritical CO<sub>2</sub> cycle compared to ORC. While the research is limited and inconsistent, it is of interest to make an assessment of the CO<sub>2</sub> cycle performance for the present study. The transcritical CO<sub>2</sub> Rankine cycle is a closed Rankine cycle, but in contrast to the conventional steam Rankine cycle, the heat addition to the occurs in supercritical state. The CO<sub>2</sub> is then expanded through a turbine and condensed. By condensing the working fluid, the compression can be done by a pump reducing the compression work.

## 1.2 Scope and Objective

The objective of the project is to determine the most efficient system being able to deliver a maximum required output of 22.5 MW per machinery package, while having the highest possible efficiency at a load of 18 MW. The complete machinery system needs to be within a weight limitation of 100 metric tons per package. The five combined cycles evaluated are simulated as described in Table 1.

For all simulations ambient conditions are assumed to be 15°C and 1.01325 bar. The cooling source is assumed to be seawater with temperatures at inlet and outlet to be 10 and 20°C, respectively. For the modeling, heat losses in heat exchangers are neglected.

**Table 1 Combined cycle simulation plan**

|  |
|--|
| <b>Combined cycle scenario (1):</b>                    |
| Gas turbine, part load and single pressure steam cycle |
| Gas turbine, part load and dual pressure steam cycle   |
| Gas turbine, part load and ORC                         |
| Gas turbine, part load and ABC                         |
| Gas turbine, part load and CO <sub>2</sub>             |
| <b>Combined cycle scenario (2):</b>                    |
| Gas turbine, full load and single pressure steam cycle |
| Gas turbine, full load and dual pressure steam cycle   |
| Gas turbine, full load and ORC                         |
| Gas turbine, full load and ABC                         |
| Gas turbine, full load and CO <sub>2</sub>             |

Preliminary component evaluation is conducted, focusing on main heat exchanger between the topping and the bottoming cycle, and the bottoming turbine. Weight and size limitations of the heat recovery unit is set to a maximal diameter of 5 meters, a maximal height of 10 meters, and a weight limitation of about 40 tons (D. Nielsen, Mols-Linien A/S, Aarhus, Denmark, 2010, private communication).

For the purpose of simplicity, the report will hereon only evaluate or discuss one machinery package, as the two packages that complete the machinery system is identical, being two separate gas turbines with each their bottoming cycle.

## 1.3 Report Outline

The report is structured as follows; Chapter 2 provides a short description of the simulation tools that have been used during this study. In chapter 3, the gas turbine selected for the study is presented, followed by an overview of the evaluated bottoming cycles selected, and the operational parameters applied for simulation in chapter 4. Chapter 5 presents the result of the combined cycle simulation and a discussion of the findings, including discussion regarding system components. A conclusion of the work is drawn in chapter 6.

## 2. Simulation tools

Energy analysis and simulations are carried out by means of DNA (Dynamic Network Analysis), a simulation program developed at the Department of Mechanical Engineering, Technical University of Denmark [20, 21]. In DNA the physical model is formulated by connecting the relevant component models through nodes and by including operating conditions for the complete system. The physical model is converted into a set of mathematical equations to be solved numerically. The mathematical equations include mass and energy conservation for all components and nodes, as well as relations for thermodynamic properties of the fluids involved. In addition, the components include a number of constitutive equations representing their physical properties, e.g. heat transfer coefficients for heat exchangers and isentropic efficiencies for compressors and turbines. The program includes a component library with models for a large number of different components existing within energy systems. During the course of this research, further development of DNA was done in order to extend the fluid library for simulations.

A genetic algorithm (GA) is used by means of Matlab software, to optimize selected parameters for the ABC model, by enabling iterations with several variables at a time. A genetic algorithm is an algorithm based on the mechanism of natural selection, evaluating which parameter combinations give the best solution [22].

### 3. The Gas Turbine

The LM2500 manufactured by General Electric is decided as the topping cycle of the combined cycles. The LM2500 is an aero-derivative gas turbine with two-shaft design. The LM2500 consists of an axial flow compressor with 16 stages, an annular combustor, a two-stage compressor turbine and a six-stage power turbine. The LM2500 is designed for both on and offshore applications. Estimated average performance data for the LM2500 gas turbine, as well as fuel characteristics was provided by General Electric (I. Bach, General Electric, Hinnerup, Denmark, 2010, private communication).

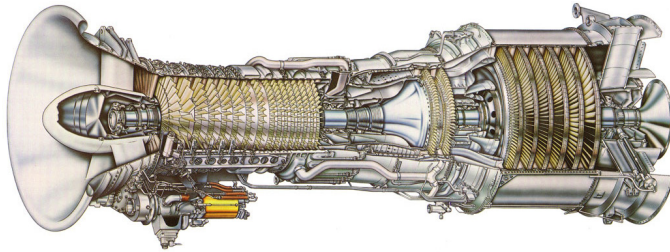


Fig. 1 LM2500 gas turbine (by courtesy of General Electric)

A turbine model at full-load operation is constructed based on the available performance data. The essential parameters for full-load used to create performance models in DNA are listed in Table 2.

Table 2 Full load performance of LM2500

|                                    |        |
|------------------------------------|--------|
| Inlet mass flow [kg/s]             | 68.6   |
| Fuel flow [kg/s]                   | 1.391  |
| Compressor outlet temperature [°C] | 485    |
| Exhaust temperature [°C]           | 579.8  |
| Relative inlet pressure loss [%]   | 1.23   |
| Relative outlet pressure loss [%]  | 3.69   |
| Compressor pressure ratio [-]      | 19.75  |
| Power output [kW]                  | 24 941 |
| Thermal Efficiency [%]             | 36.4   |

The gas turbine is assumed to consist of an inlet, compressor, combustion chamber, compressor turbine, power turbine, and exhaust. A combustion chamber pressure loss of 3% and a mechanical efficiency of 99% are assumed. Inlet mass flow, pressure ratio, generator efficiency, and inlet and outlet pressure drops are defined according to the specifications. Bleed is simulated by extracting 1% of the flow after the compressor. Cooling is used to enable higher turbine entry temperatures than the maximum allowable metal temperature. This is accomplished by bleeding off a part of the compressed air which then passes through cooling passages inside the blades. Effects of cooling are simulated by bleeding off 10% of the air flow after the com-

pressor. Depending on where the expansion work takes place, this air is mixed with the hot gases at different points along the expansion.

In practice, for modern gas turbines, the turbine cooling system generally is more complex than modeled here; air is usually extracted from the compressor at a number of places, matching the pressure condition where it should be injected in the turbine.

Steady state models describing the part-load characteristics of compressors and turbines of real gas turbine engines are used to simulate the part-load performance of the gas turbine. These models are based on component maps. Further information about the use of component maps in DNA is provided in Haglind and Elmegaard [23].

## 4. Combined Cycle Designs

### 4.1 Gas Turbine and Steam Cycle

Two bottoming steam cycles are modeled: one single pressure steam cycle (Fig. 2), and one dual pressure steam cycle (Fig. 3). While a dual pressure steam cycle is more efficient than a single pressure steam cycle, it also involves more components and increased complexity. According to Poullikkas [1], it is more cost efficient with a power plant of low complexity for small-scale power generation (less than 50 MW). In this report the performance of both a single pressure and a dual pressure steam cycle is evaluated.

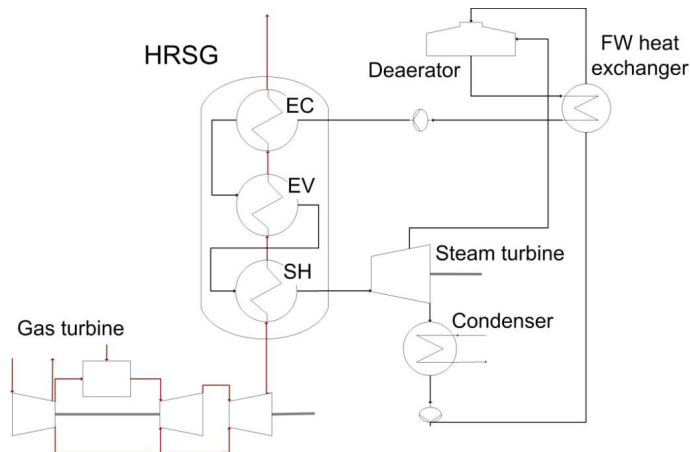


Fig. 2 Combined cycle with single pressure steam bottoming cycle

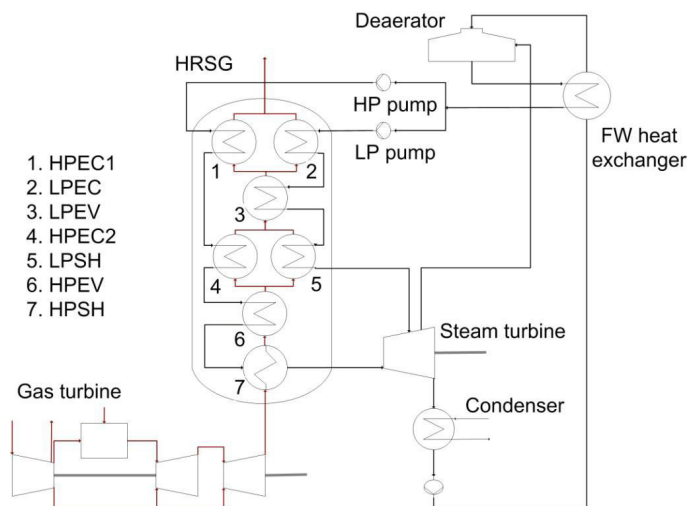


Fig. 3 Combined cycle with dual pressure steam bottoming cycle

The layout of the dual pressure steam cycle is modeled following the dual pressure cycle for low sulfur fuels presented by Kehlhofer et al. [24]. By including more economizers in the HRSG and splitting the exhaust gas stream, the gas temperature at the HRSG outlet is lowered, utilizing more of the exhaust gas energy. Assumptions made for both models are listed in Table 3.

**Table 3 Assumptions for steam cycle**

|   |       |
|---|-------|
| Pinch point, HRSG [°C]                    | 10    |
| Superheat approach [°C]                   | 30    |
| Cooling water inlet temperature [°C]      | 10    |
| Cooling water outlet temperature [°C]     | 20    |
| Condensing pressure [bar]                 | 0.032 |
| Turbine isentropic efficiency [%]         | 84    |
| Pump isentropic efficiency [%]            | 80    |
| Pressure drop steam cycle [%]             | 10    |
| Pressure drop exhaust gas side HRSG [bar] | 0.03  |

Steam is extracted from the steam turbine to the deaerator to ensure removal of non-condensable gases from the water. The deaerated water is then cooled by the condensed water entering the deaerator, lowering the feed-water temperature before being pressurized.

Low temperature corrosion is a concern when recovering heat from exhaust gas. Though there is no or little sulfur present while firing natural gas, low temperature corrosion can still occur in the HRSG if the feed-water temperature is below the water dew point which is typically between 40 – 45°C. The feed-water temperature is for the steam cycle simulations set to 60°C assuring that no low temperature corrosion can occur.

The pinch point in the HRSG is a parameter that greatly affects the cost and size of the unit. Kehlhofer et al. [24] state that the pinch points are typically between 8 -15°C. Lowering the pinch point increases the necessary heating surface of the HRSG, increasing both the size and cost of the system. For the gas-steam modeling, a pinch point of 10°C is chosen for the evaporators.

While the steam leaving the turbine will have started to condense, the steam quality needs to be kept above a certain minimum level to prevent erosion of the turbine blades. Steam turbine manufacturers recommend a steam quality within a range of 0.86 and 0.89 [25]. Modeling the steam cycles, the pressure levels were decided based on optimal power output while steam quality is held at 0.88. The condenser pressure was set to 0.032 bar resulting in a saturation temperature of 25°C, giving a 5 degree margin to the cooling water outlet temperature.

The assumption of the isentropic efficiency of the steam turbine is based on that of the Siemens SST-120 steam turbine (K. Brandelev, Siemens Fossil Power Generation Division, Ballerup, Denmark, 2011, private communication). The SST-120 is not particularly adopted for marine applications, however, applicability is a matter related to practical installations and not turbine performance. The same turbine isentropic efficiency is used for all steam model simulations. This is a simplification, as the mass flow and the pressure level of the different cycles vary. In

practice this would result in variations of the turbine isentropic efficiency for the different steam cycles.

The given assumptions for the steam cycles are applied for all steam cycle simulations, whereas the pressure of the steam cycles varies with configuration and scenario. The single pressure steam cycle is modeled with pressures of 56 bar and 88 bar for scenarios (1) and (2), respectively. Both pressure levels, under each of the circumstances, correspond to a steam quality at turbine outlet of 0.88.

The dual pressure cycle is modeled with a high-pressure (HP) of 60 bar and a low-pressure (LP) of 4 bar when optimized for scenario (1). By iteration, a combination of pressure levels that give optimal combined cycle thermal efficiency is found, while staying within the steam quality limit. This resulted in a steam quality of 0.87 at given pressure levels. Trying to obtain a quality of 0.88 for this steam cycle has such a detrimental effect on the power output as the pressure would have to be decreased. Optimizing the dual pressure steam cycle for scenario (2) resulted in a HP of 80 bar and a LP of 6 bar, corresponding to a steam quality of 0.88.

## 4.2 Gas Turbine and ORC

An important part of modeling an ORC is choosing the working fluid. Stability, safety and environment have to be considered while still selecting a working fluid with satisfying results for the given heat source temperature. There is a significant amount of literature dedicated to evaluation of working fluids for ORC processes [6, 7, 18, 26, 27], which also reflects the complexity of the matter and the varying circumstances of the evaluation. Many articles have considered the use of refrigerants such as R113, R123, R134a, or R152a for low-temperature ORC applications, but as many of them consist of chlorofluorocarbon (CFC) compounds, several have been phased out or will be in the near future [28] due to regulations on the use of CFC-gases.

This report is only considering subcritical Rankine cycles with a pure organic fluid. Supercritical Rankine cycles have shown higher efficiencies, but also demand higher pressures in the evaporator. While subcritical cycles have been more researched and implemented, there is none or little practical experience with a supercritical ORC [5].

The temperature of the waste heat source is a defining parameter when choosing the working fluid. The waste heat source is often divided into low ( $< 230^{\circ}\text{C}$ ), medium ( $230 - 650^{\circ}\text{C}$ ) or high ( $> 650^{\circ}\text{C}$ ) temperature heat source [5]. The flue gas temperature from the gas turbine can easily reach temperatures of about  $550 - 600^{\circ}\text{C}$  at full load. These temperatures exceed the maximum operating temperature or self-ignition temperature of most organic fluids. To prevent local overheating of the organic fluid, a heat transfer oil is used in an intermediate loop, also allowing the heat exchange to be operated close to the atmospheric pressure. Though the oil loop increases the irreversibility of the heat exchange, it also ensures higher stability with load variations for the ORC process due to the thermal inertia of the oil.

The heat transfer oil has its own temperature limitations with a common maximum temperature of  $340^{\circ}\text{C}$ . Also, to maintain the viscosity of the oil during operation, one would need to keep the oil temperature above a certain point. This lower temperature may vary with dimensions, flow



and oil characteristics (M. Bates, Duratherm Fluids, 2011, Lewiston, NY, USA, private communication). For the purpose of this project a minimum temperature of 130°C is assumed, whereas the maximum temperature is set to 335°C. While the temperature of the gas turbine exhaust can be considered a high-grade energy source, the use of an intermediate oil loop lowers the maximum temperature of the energy source to be “harvested” by the ORC, due to the temperature limitations of the heat transfer oil. The organic fluid should therefore be chosen, evaluated by how it “matches” the heat transfer slope of the oil. In Fig. 4, this is illustrated by a T-Q diagram, where the red line represents the hot exhaust gas, the yellow line represents the heat transfer oil, and the blue line then illustrates the organic fluid.

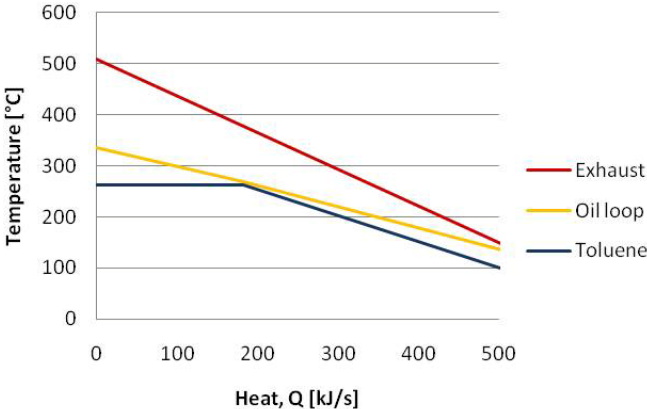


Fig. 4 T-Q diagram ORC

It is desirable to have a working fluid with a high critical temperature and a relatively low critical pressure which lessens the pump work. It is also favorable to use a dry or isentropic fluid having a positive or zero slope of the saturated vapor line, meaning that there will be no condensation in the fluid during expansion (see Fig. 5).

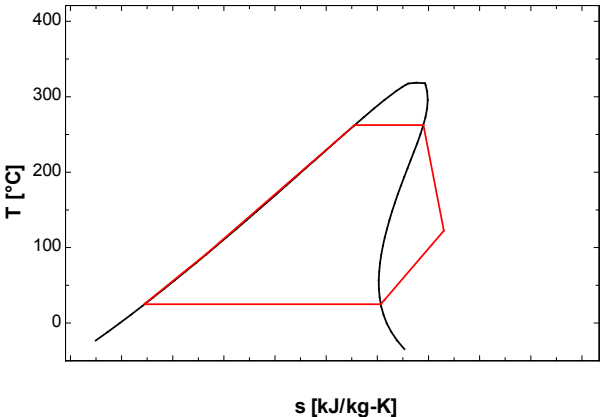


Fig. 5 T-s diagram of dry fluid, Rankine cycle

Quoillin [29] suggests that toluene and silicone oils, having a high critical temperature or boiling point, are compatible with a heat source of temperatures around 300°C. Whereas refrigerants or hydrocarbons like pentane or butane are well-suited with a heat source temperature lower than 200°C. Larjola [30] tested several fluids in an ORC prototype and concluded that toluene was the most suitable working fluid for high-temperature processes.

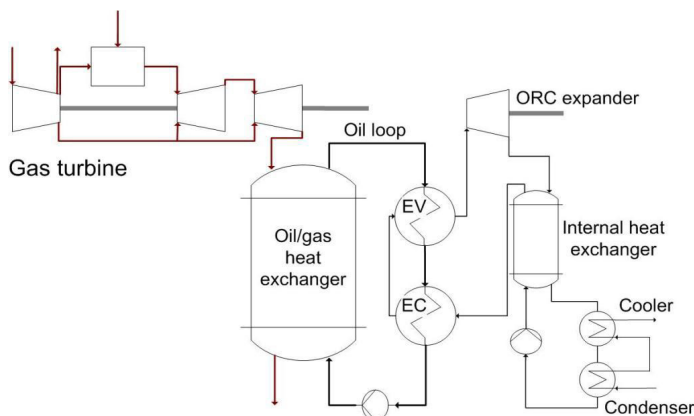
Also Chacartegui [27] reported that toluene ORC achieved maximum combined cycle efficiency, when coupled with a recuperated gas turbine. The choice of an optimal working fluid would necessarily be a compromise with all the parameters involved, such as power output, safety, cost, stability, and availability. Based on the given studies with similar circumstances, toluene (C<sub>7</sub>H<sub>8</sub>) is chosen as the working fluid for the ORC simulations, having a low critical pressure and a suitable critical temperature for the given application (see Table 4).

**Table 4 Characteristics of toluene**

|         | T <sub>m</sub> [°C] | T <sub>b</sub> [°C] | T <sub>c</sub> [°C] | T <sub>ai</sub> [°C] | P <sub>c</sub> [bar] | Molecular weight |
|---------|---------------------|---------------------|---------------------|----------------------|----------------------|------------------|
| Toluene | -95.15              | 110.6               | 318.6               | 480                  | 41.26                | 92.14            |

Organic fluids in contrast to water are subjected to chemical deterioration with high temperatures; therefore chemical stability is an important factor for the lifetime of the working fluid. Toluene is regarded as a stable fluid up to temperatures of 350°C [7]. When considering the safety of using organic working fluids, both hydrocarbons and aromatic fluids are flammable and toxic and need to be handled accordingly.

The thermodynamic properties for toluene were obtained by FPROPS, an open-source library of thermodynamic properties of a number of fluids. The thermodynamic state equations are based on the Helmholtz energy equation of state, as described by Lemmon et al.[31]. For the simulations a combined cycle system consisting of the topping gas turbine and a bottoming ORC was modeled as illustrated in Fig. 6.



**Fig. 6 Combined cycle with ORC for bottoming cycle**

Through simulations it is found that the best suited configuration is an ORC loop without superheat, but with an internal heat exchanger. The heat from the exhaust gas is absorbed in the

heat transfer oil which then heats up and evaporates the organic fluid toluene. Toluene is then expanded through the turbine. Toluene, being a dry fluid, means that there is no need to superheat the fluid prior to expansion. The excess heat at the turbine outlet is exploited in an internal heat exchanger (IHE) before the organic fluid is cooled down to saturation temperature and condensed.

The temperature of the heat transfer oil being fixed both at inlet and outlet of the heat exchanger, results in higher power output and improved thermal efficiency with the use of IHE. This is due to increased working fluid mass flow, as the difference in enthalpy of toluene is reduced while the amount of heat transferred is constant. The regeneration by the IHE reduces the cooling load on the condenser, but the fluid is still superheated at the regenerator outlet prior to condensation, making it necessary to cool the fluid until it reaches the condensation temperature. A pinch point between the toluene and the cooling source of 5°C is assumed for the cooling and condensation, resulting in a condensation temperature of 24.9°C with a corresponding pressure of 0.038 bar.

In order to assume a reasonable figure for the isentropic efficiency for the turbine, previous work is reviewed. Though there are manufacturers specialized in ORC systems, there is little information readily available about component performance. There are some variations in literature where Chen [18] assumes turbine and pump isentropic efficiencies of 70 % and 80 %, respectively, while Chacartegui [27] assumes a pump efficiency of 80 % and a turbine efficiency of 87 %. Mago [32] applies efficiencies of 80 % for turbine and 85 % for pump. ORC manufacturers claim to achieve isentropic turbine efficiencies of up to 90 % [33]. Based on the mentioned studies, the ORC turbine isentropic efficiency and pump isentropic efficiency for the present ORC cycle were both assumed to be 0.80 %.

A pinch point of 10°C was assumed for the oil – exhaust gas heat exchanger, whereas the pinch point in the evaporator and the IHE was set to 5°C. Similar assumptions were also presented by Chen [18]. The oil pressure is assumed to be 2 bar. For simplicity the pressure drop in the oil loop as well as heat losses in the bottoming ORC cycle were neglected. All assumptions are given in Table 5.

**Table 5 Assumptions for ORC**

|  |       |
|--|-------|
| Pinch point, exhaust - oil heat exchanger [°C]       | 10    |
| Pinch point, oil - toluene heat exchanger [°C]       | 5     |
| Pinch point, internal heat exchanger [°C]            | 5     |
| Cooling water inlet temperature [°C]                 | 10    |
| Cooling water outlet temperature [°C]                | 20    |
| Condensing pressure toluene [bar]                    | 0.038 |
| Turbine isentropic efficiency [%]                    | 80    |
| Pump isentropic efficiency [%]                       | 80    |
| Pressure drop ORC cycle [%]                          | 10    |
| Pressure drop exhaust gas side, heat exchanger [bar] | 0.03  |
| Heat transfer oil hot temperature [°C]               | 335   |
| Heat transfer oil cold temperature [°C]              | 130   |

Combined cycle simulations were carried out with the given assumptions for scenario (1) and scenario (2). In contrast to the steam cycle, the pressure level of the ORC is not affected by the lowered exhaust gas temperature of the gas turbine in scenario (1) and gives an optimal result with the pressure of 22 bar. The oil circuit with constant temperatures enables the ORC to operate at the same pressure and temperature, while the mass flow is the variable parameter influencing the heat transfer, and thus the power output.

### 4.3 Gas Turbine and ABC

A combined cycle model is constructed with the topping gas turbine LM2500 and a bottoming ABC cycle. The air compression of the ABC is divided into three compression levels with inter-cooling in-between to enhance the bottoming cycle performance (see Fig. 7).

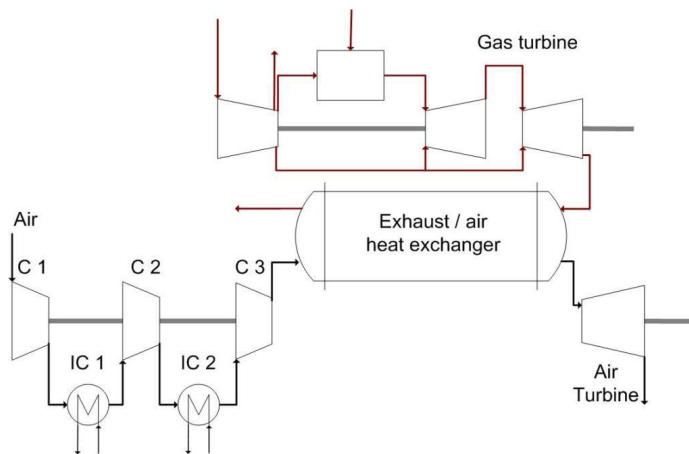


Fig. 7 Combined cycle with ABC for bottoming cycle

The number of inter-cooling units is a compromise of cost and efficiency. Bolland [12] argued that when considering construction and practicality of an ABC, it would be reasonable to evaluate whether to have one or two intercoolers, whereas the benefit with more units would be reduced by the increased parasitic losses. For the heat exchanger, a regenerator is chosen. Although there can be some leakage or carryover between the hot side and cold side in a regenerator that may cause extra pressure losses, a regenerator offers a much more compact heat transfer surface than a recuperator, reducing both size and cost of the component [12]. In order to demonstrate a more compact and simple alternative to the steam cycle, keeping the size and weight at a minimum is desirable. Considering availability of cycle components, Bolland et al. [12] concluded that a suitable compressor and turbine for an ABC cycle could be built applying conventional design practice.

While the cooling source was used for condensation in the previous bottoming cycle models, here the cooling is between a gas and a fluid where the heat-transfer will not be equally efficient. The pinch point is therefore set to 10°C in the intercoolers in comparison to a 5°C pinch

point in the condensers. In order to achieve the lowest possible temperature of the compressed air, the cooling source is assumed split into two streams, giving the inlet and the outlet temperature of the seawater as the same for both inter-coolers.

Based on the studies of Bolland [12], Kaikko [34] and Ghazikhani [14], polytropic efficiencies of the turbine and compressor were selected, as well as efficiency of the regenerator, and are listed in Table 6.

**Table 6 Assumptions for ABC**

|  |      |
|--|------|
| Effectiveness regenerator [%]                        | 93   |
| Inlet air temperature [°C]                           | 15   |
| Turbine polytropic efficiency [%]                    | 85   |
| Compressor polytropic efficiency                     | 91   |
| Compressor mechanical efficiency [%]                 | 98   |
| Pressure drop air side, heat exchanger [%]           | 2    |
| Pressure drop exhaust gas side, heat exchanger [bar] | 0.03 |
| Pinch point, intercoolers [°C]                       | 10   |
| Cooling water inlet temperature [°C]                 | 10   |
| Cooling water outlet temperature [°C]                | 20   |

The pressure ratio for the ABC compression is reported by Bolland [12] to be optimal at a 8:1 ratio when the compression is divided into three segments. Kaikko found an optimal compression ratio of 7.3 for the same system configuration. In order to determine the optimal pressure in each of the three segments of the compression, the genetic algorithm (GA) is applied. The GA enables iteration where the optimal combination of pressure ratios is found, resulting in the compression ratio for C1, C2 and C3 to be 2.15, 2.05 and 2.04, respectively, giving an overall compression ratio of 9. These pressure ratios were applied to both scenario simulations for the gas turbine and ABC combined cycle.

#### 4.4 Gas Turbine and Transcritical Carbon Dioxide Cycle

Carbon dioxide (CO<sub>2</sub>) has a critical temperature of 31°C, allowing for heat transfer to CO<sub>2</sub> in superheated state for low temperatures. The advantage this gives is a closer heat transfer curve between the heat source and working fluid in comparison to a working fluid that undergoes evaporation, giving enhanced heat recovery. This is illustrated by comparing the T-s diagrams of a simple steam cycle and a transcritical CO<sub>2</sub> cycle in Fig. 8 and Fig. 9.

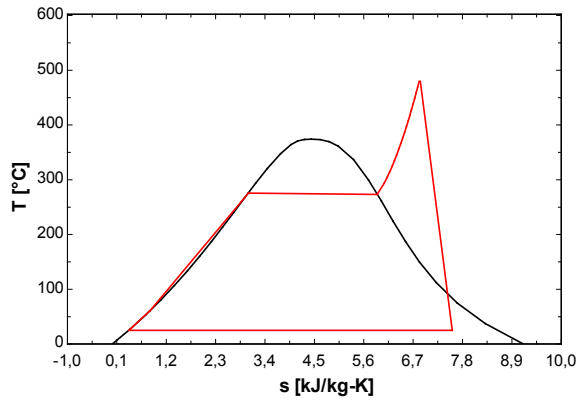


Fig. 8 T-s diagram of simple steam cycle

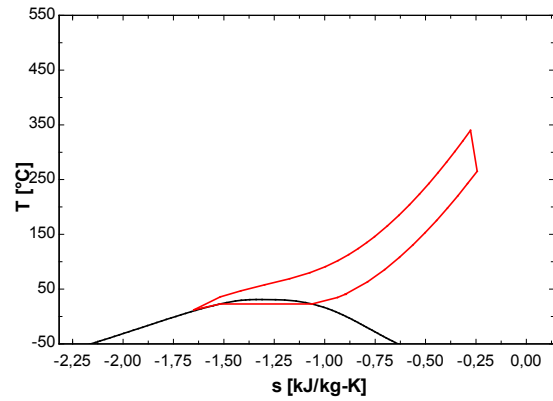


Fig. 9 T-s diagram of transcritical CO<sub>2</sub> cycle

CO<sub>2</sub> has also a high critical pressure of 73.8 bar. The high density of the gas ensures high volumetric efficiency which reduces the necessary heat transfer surface, furthermore, Ladam et al. [35] concluded in their study that CO<sub>2</sub> technology, with the characteristics of CO<sub>2</sub>, showed potential for more compact components both compared with steam and organic working fluids.

For this study, a model was constructed of a combined cycle with LM2500 as topping cycle, and a transcritical CO<sub>2</sub> cycle as bottoming cycle as illustrated in Fig. 10.

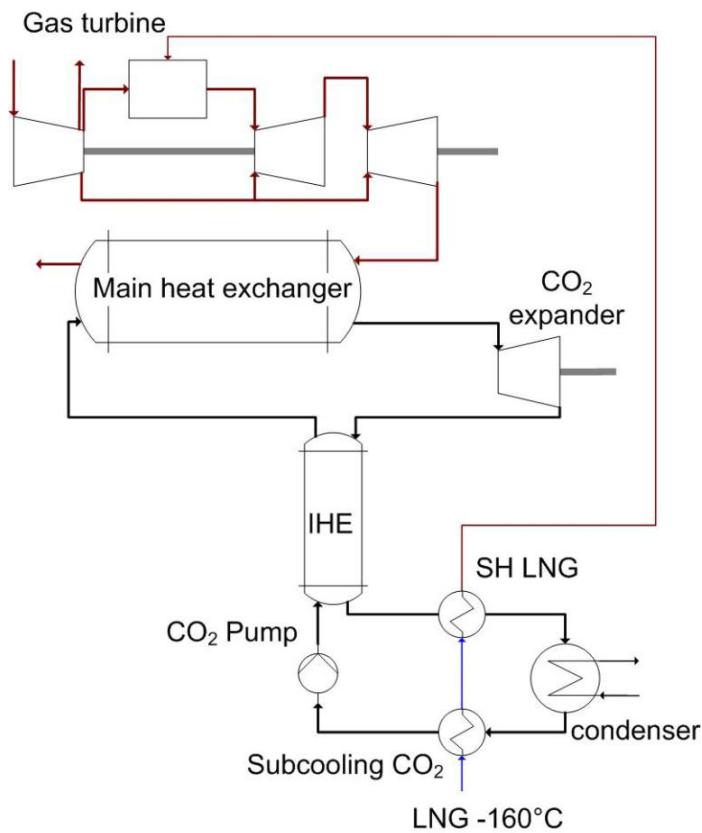


Fig. 10 Combined cycle gas turbine with CO<sub>2</sub> transcritical cycle as bottoming cycle

The CO<sub>2</sub> is heated by the exhaust gas in a main heat exchanger, followed by expansion. The CO<sub>2</sub> is then cooled in an internal heat exchanger, before superheating LNG, while CO<sub>2</sub> is cooled further down before condensation. Condensation is done by means of sea-water as cooling source.

The LNG that is injected in the combustion chamber of the gas turbine is stored at a temperature of -160°C. Prior to injection, the LNG needs to be superheated to a minimum of 28°C (I. Bach, General Electric, Hinnerup, Denmark, 2010, private communication). With CO<sub>2</sub> being an inert gas, it can be used to heat the LNG to the necessary temperature while cooling the CO<sub>2</sub>. The available LNG is limited to the fuel consumption of the upper cycle, resulting in a LNG mass flow of LNG to little to condense the CO<sub>2</sub>. Therefore sea-water is used for condensation. A pinch point between the sea-water and CO<sub>2</sub> is set to 5°C, from which a condensation pressure of 61 bar is derived, corresponding to a condensation temperature of 23°C.

The heating of LNG (and cooling of CO<sub>2</sub>) is divided into two parts, whereas LNG is heated partly by the condensed CO<sub>2</sub>, and partly by the superheated CO<sub>2</sub> before condensation. This enables superheat of LNG to 30°C, as well as sub-cooling CO<sub>2</sub> down to about 12°C prior to compression. While the heat transfer to the CO<sub>2</sub> cycle occurs with CO<sub>2</sub> being in supercritical state, the phase change occurs in the pump. This will likely cause cavitations problems, but a solution of sub-cooling the CO<sub>2</sub> prior to compression reduces the change of specific volume of CO<sub>2</sub> during compression, reducing the stresses on the pump.

The CO<sub>2</sub> transcritical Rankine cycle for power generation is, as far as literature goes, only evaluated in theory. The assumptions for the cycle are therefore based on previous studies [17, 18, 35, 36], and are listed in Table 7. The turbine inlet temperature, giving best result in power output and efficiency, is found by iteration for both scenario (1) and (2), and is 340°C and 360°C, respectively. All assumptions are applied for both scenario simulations.

**Table 7 Assumptions for CO<sub>2</sub> transcritical cycle**

|  |      |
|--|------|
| Turbine inlet pressure [bar]                               | 140  |
| Turbine isentropic efficiency [%]                          | 80   |
| Pump isentropic efficiency [%]                             | 80   |
| Pinch point, exhaust - CO <sub>2</sub> heat exchanger [°C] | 20   |
| Pinch point, internal heat exchanger [°C]                  | 5    |
| Cooling water inlet temperature [°C]                       | 10   |
| Cooling water outlet temperature [°C]                      | 20   |
| Condensing pressure CO <sub>2</sub> [bar]                  | 61   |
| LNG inlet temperature [°C]                                 | -160 |
| LNG outlet temperature [°C]                                | 30   |

Considering bottoming cycles and their working fluids for maritime application, presents particular safety restrictions, such as the working fluid preferably being non-flammable and non-toxic. In contrary to most organic fluids, CO<sub>2</sub> is not a flammable gas or considered toxic, still, it does present a health risk if inhaled, primarily due to oxygen deficiency. On the other hand, CO<sub>2</sub> is a cheap and available working fluid.

## 5. Results and Discussion

The results of the simulations for scenario (1) are shown in Table 8, and the simulation results for scenario (2) are presented in Table 9.

Scenario (1) considered operating the gas turbine at part load. The load on the gas turbine is therefore adjusted to achieve the wanted total power output of 18 MW. The effect of this is a lowered gas turbine exhaust outlet temperature and mass flow. For the bottoming cycles this meant an exhaust outlet temperature of around 515°C in comparison to an exhaust temperature of about 580°C when the gas turbine is operated at full load.

The results of the combined cycle simulation for scenario (1) are to be compared to a machinery solution consisting of the gas turbine without a bottoming cycle. The gas turbine LM2500 operated with a power output of 18 MW constitutes an operational load level of 72%, corresponding to a thermal efficiency of the gas turbine of 34%.

**Table 8 Results of scenario (1) simulations**

|                                 | GT and 1 pl<br>steam cycle | GT and 2 pl<br>steam cycle | GT and<br>ORC | GT and<br>ABC | GT and<br>CO <sub>2</sub> |
|---------------------------------|----------------------------|----------------------------|---------------|---------------|---------------------------|
| % GT load                       | 46.6%                      | 43.8%                      | 47.7%         | 54.3%         | 53.9%                     |
| GT load [kW]                    | 11 432                     | 10 735                     | 11692         | 13 307        | 13213                     |
| Exhaust outlet temperature [°C] | 178.0                      | 104.0                      | 140.0         | 95.3          | 166.0                     |
| bottom cycle load [kW]          | 6 568                      | 7 265                      | 6 308         | 4 693         | 4 787                     |
| Total load [kW]                 | <b>18 000</b>              | <b>18 000</b>              | <b>18 000</b> | <b>18 000</b> | <b>18 000</b>             |
| $\eta_{th}$ GT                  | 29.4%                      | 28.8%                      | 29.7%         | 31.0%         | 30.9%                     |
| $\eta_{th}$ secondary           | 35.5%                      | 33.4%                      | 31.9%         | 19.0%         | 23.1%                     |
| $\eta_{th}$ combined cycle      | <b>46.3%</b>               | <b>48.2%</b>               | <b>45.6%</b>  | <b>41.9%</b>  | <b>42.1%</b>              |
| Fuel savings [%]                | 26.5%                      | 29.4%                      | 25.4%         | 18.7%         | 19.1%                     |

**Table 9 Results of scenario (2) simulations**

|                                 | GT and 1 pl<br>steam cycle | GT and 2 pl<br>steam cycle | GT and<br>ORC | GT and<br>ABC | GT and<br>CO <sub>2</sub> |
|---------------------------------|----------------------------|----------------------------|---------------|---------------|---------------------------|
| % GT load                       | 100%                       | 100%                       | 100%          | 100%          | 100%                      |
| GT load [kW]                    | 24 514                     | 24 514                     | 24 514        | 24 514        | 24 514                    |
| Exhaust outlet temperature [°C] | 167.5                      | 104.8                      | 140           | 129.3         | 180.5                     |
| bottom cycle load [kW]          | 12 198                     | 13 436                     | 10 433        | 8 260         | 7 523                     |
| Total load [kW]                 | <b>36 712</b>              | <b>37 950</b>              | <b>34 947</b> | <b>32 774</b> | <b>32 037</b>             |
| $\eta_{th}$ GT                  | 35.7%                      | 35.7%                      | 35.7%         | 35.7%         | 35.7%                     |
| $\eta_{th}$ secondary           | 37.8%                      | 36.4%                      | 31.9%         | 23.6%         | 24.0%                     |
| $\eta_{th}$ combined cycle      | <b>53.5%</b>               | <b>55.3%</b>               | <b>51.0%</b>  | <b>47.8%</b>  | <b>46.7%</b>              |



## 5.1 Thermodynamic Performance

The combined cycle simulation results show little difference in the performance of the single pressure steam cycle compared to the ORC as bottoming cycle for scenario (1).

With regards to the thermal efficiency, the dual pressure steam cycle gave the best combined cycle result with an efficiency of 48.2%, whereas the combined cycle results of the single pressure steam cycle and the ORC as bottoming cycle are 46.3% and 45.6%, respectively (illustrated in Fig. 11). With the lowered temperature of the exhaust gas, the combined cycle models with ABC and CO<sub>2</sub> as bottoming cycle represent the least efficient systems. It should still be noted that even with the poor outcomes for these model, the efficiency is still around 8%-points better than the gas turbine being operated alone at the given load.

By simulating for a constant combined cycle output, the load on the gas turbine varies for each cycle. Lowering the load on the gas turbine reduces the gas turbine thermal efficiency, but can benefit the total combined cycle efficiency, as well as reducing the fuel consumption.

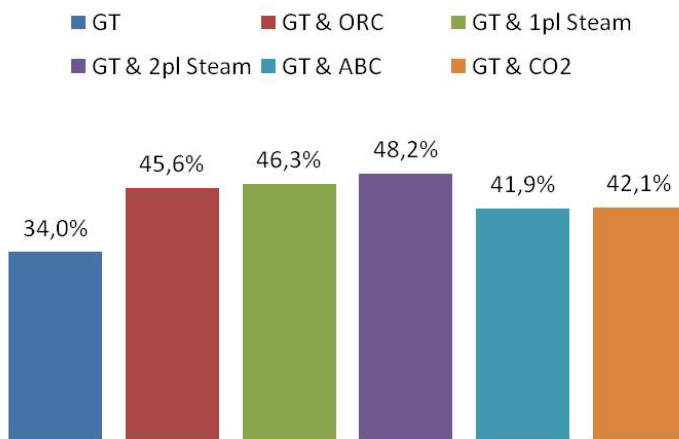


Fig. 11 Result comparison for scenario (1)

The reduced gas turbine exhaust outlet temperature of scenario (1), effects the steam cycles in the sense of lower steam pressures in order to maintain an acceptable steam quality. In comparison, the ORC cycle is not affected in the same matter. The lowered exhaust gas temperature does not reduce the temperature of the heat transfer oil, thereby allowing the ORC temperatures to be kept constant. This is illustrated by the constant thermal efficiency of the ORC bottoming cycles. The variable of the ORC bottoming cycle is the mass flow of the system that is less for scenario (1), resulting in a lower power output compared to scenario (2).

Scenario (2) was modeled with the gas turbine operated at full load, and the combined cycles were optimized for the highest power output and thermal efficiency. Again the steam cycles show the best results with a combined cycle thermal efficiency of 55.3 % with the dual pressure steam bottoming cycle.

The high gas turbine exhaust temperatures prove to be a good match with the steam bottoming cycle reflecting high thermal efficiencies for the gas and steam combined cycles. Another factor that influenced the combined gas and steam cycle efficiencies, especially the dual pressure steam cycles, is the low feed-water temperature allowing more energy to be extracted from the gas turbine exhaust. This cannot be exploited as good in the ORC as the lower oil temperature limits the lower exhaust gas temperature, rather than the water dew point of the exhaust.

The steam cycle is shown by comparing the results of scenario (1) and scenario (2) to be adversely affected by lowering the exhaust gas temperature. In Fig. 12 the relative difference between the combined cycle thermal efficiencies can be evaluated.

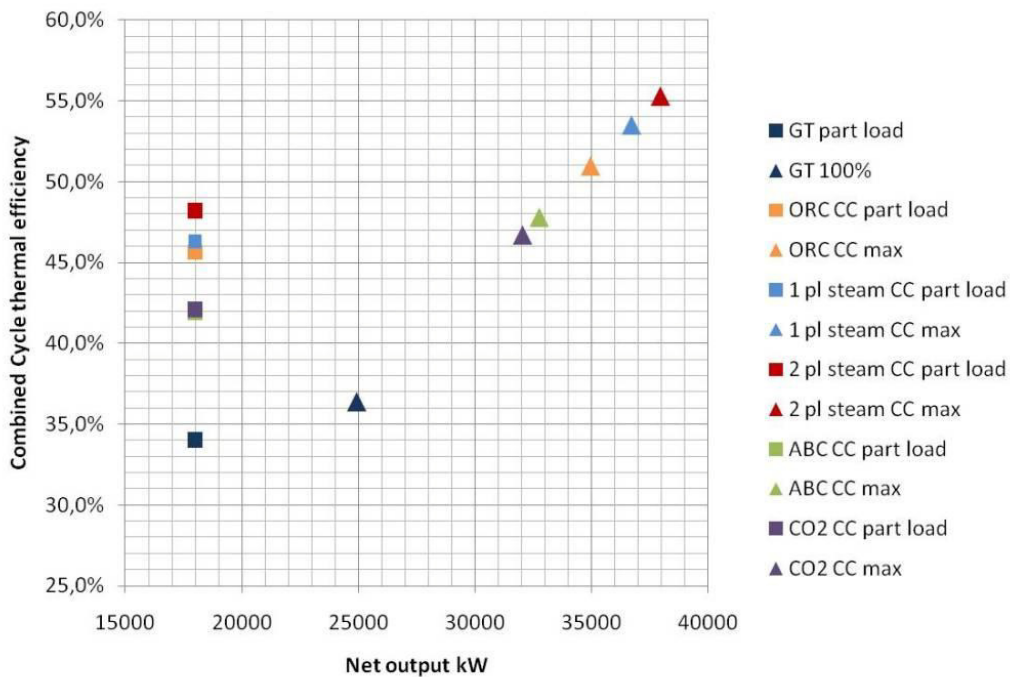


Fig. 12 Combined cycle comparison

The difference in thermal efficiency between the single pressure steam cycle and the ORC as bottoming cycles is for scenario (1) much less than for scenario (2). Part-load operation constitutes a great share of vessel operation for most ships, and the bottoming cycle sensitivity to lowered gas turbine exhaust temperature is imperative.

The combined gas turbine and ABC extracts a great amount of energy out of the exhaust gas, and the use of intercoolers allows for low exhaust outlet temperatures leaving the regenerator. But the compression work in the ABC cycle is a much higher energy demand than the pump work of either steam cycle or ORC, leading to a lot of the absorbed energy being utilized for the air compression. A bi-product of the ABC cycle is hot air. The air expanded in the ABC leaves the bottoming turbine with a temperature around 220°C. This hot air can be used for heating purposes, but is in this thermal evaluation a heat loss to the surroundings, resulting in a low bottoming cycle thermal efficiency of the ABC.

CO<sub>2</sub> does not present competitive results compared to either steam or ORC bottoming cycles. The high condensation pressure of the CO<sub>2</sub> transcritical cycle results in a relatively low pressure ratio of the turbine giving that the power output is limited. The CO<sub>2</sub> could potentially condense at much lower pressure and temperature, but this is limited by the available cooling source.

## 5.2 Component Evaluation

The component that comprises the largest volume and weight for any of the bottoming cycles is the heat exchanger between the exhaust gas and the given working fluid. Some assessments are made of the size of this unit, in particular for the gas and ORC combined cycle. This is done considering scenario (1) simulations. The turbine of the bottoming cycles is also evaluated based on own results of scenario (1) simulations, as well as previous research is reviewed.

### 5.2.1 Heat recovery unit

The conditions for the waste heat recovery unit was decided with Mols-Linien A/S, to correspond to the gas turbine operated at part load, with a combined cycle power output of 18 MW. If the power output would be increased beyond the 18 MW, the gas turbine would run separate from the bottoming cycle delivering the entire power output, while by-passing the waste heat recovery unit. Weight and size limitations of the heat recovery unit is set to a maximal diameter of 5 meters, a maximal height of 10 meters, and a weight limitation of about 40 tons (D. Nielsen, Mols-Linien A/S, Aarhus, Denmark, 2010, private communication).

Preliminary dimensions of a heat transfer unit were provided by Alfa Laval Aalborg A/S (O. Knudsen, Alfa Laval Aalborg A/S, Aalborg, Denmark 2011, private communication) and are listed in Table 10. The same dimensions is claimed to be applicable both with steam as working fluid, and with heat transfer oil as for the ORC bottoming cycle. The simulation output and assumptions considering the heat transfer, was also confirmed by the calculations done by Alfa Laval Aalborg A/S.

**Table 10 Heat exchanger measures**

|                            |     |
|----------------------------|-----|
| Height                     | 3.2 |
| Width [m]                  | 3.4 |
| Length [m]                 | 7.2 |
| weight, dry [ton]          | 41  |
| weight, in operation [ton] | 45  |

The proposed unit is a rectangular, horizontal oriented ribbed heat exchanger. Alterations can be made to the dimensions while holding the total areal constant. Increasing the height, and thereby relieving the length, can be done by adding more ribs per meter, but this will also increase the weight of the unit (O. Knudsen, Alfa Laval Aalborg A/S, Aalborg, Denmark 2011, private communication). The volume of the unit is within the dimensional limits, but the orientation, giving a length of 7.2 meters is beyond the limitation of 5 meters in diameter, as well as the maximum weight is exceeded.

Further assessment is necessary to dimension a heat exchanger within the limitations that are presented. It can be necessary to re-evaluate the pinch points used, in order to reduce the necessary heat transfer area in an effort to define a smaller and lighter heat exchanger, but this would also reduce the combined cycle efficiency.

By increasing the pinch point, the power output of the bottoming cycle is reduced, and to obtain the combined cycle output, the gas turbine load must be increased. Increasing the load on the gas turbine, will in return, increase the exhaust gas mass flow which also influences the heat exchanger dimensions. An iteration process is necessary to find the optimal combination of gas turbine load and pinch point, in order of designing a heat exchanger within the dimensional constraints.

Considering the ABC solution for bottoming cycle, Bolland et al. [12] proposed an estimate of a recuperator volume and weight, as well as a factor between recuperator and regenerator weight in a study of a combined gas turbine and ABC, of similar power output as in this study. According to his evaluation, a regenerator of around 10 tons in weight should be obtainable whereas it was found that a recuperator as heat exchanger would constitute a core volume of about  $54 \text{ m}^3$  and a core weight of 42.4 tons.

### 5.2.2 Turbine

Previous research of ORC and  $\text{CO}_2$  cycles generally suggest that components for ORC and  $\text{CO}_2$  cycles are more compact and simpler than the components of a steam cycle with similar power output. Chacartegui et al. [27], estimates that a turbine with similar power output would be about 20% smaller for toluene compared to steam as working fluid, in a study of similar combined cycle simulation as reported here.

Volume flow and enthalpy drop are factors that influence the turbine size, where volume flow affects the turbine exit area, and the enthalpy drop over the turbine affects the number of turbine blades necessary. These two working fluid characteristics is evaluated for the single pressure steam, ORC and  $\text{CO}_2$  bottoming cycles of the scenario (1) simulations, where the power output of the ORC and steam turbine is similar, while the  $\text{CO}_2$  power output is smaller.

Toluene at turbine outlet is about 4 times denser than steam, while having similar condensing pressure.  $\text{CO}_2$  has a very high condensing pressure resulting in  $\text{CO}_2$  being a staggering 2390 times more dense than steam at turbine outlet. The calculated mass flows of the cycles vary also dependent of working fluid. The mass flow of steam is 5.9 kg/s, while toluene and  $\text{CO}_2$  has mass flows of about 34.4 kg/s and 84.7 kg/s, respectively. Based on these values, the volume flow at turbine outlet for each of the cycles is found. While toluene is denser than steam, the greater mass flow of the system result in a volume flow at turbine outlet which is 1.4 times higher than the volume flow of steam, whereas the volume flow of steam at turbine outlet is 165 times larger than for  $\text{CO}_2$ . The greater volume flow of toluene at turbine outlet compared to steam, suggest a greater turbine exit area of the toluene turbine. The volume flow of toluene is about 40 % higher than for steam, and transferring this directly to increased turbine exit area, suggest an increased diameter of the turbine outlet by 20%.

While this seems to contradict the argument that ORC turbines can be made smaller than steam turbines, one also needs to evaluate the enthalpy drop over the turbine.

The enthalpy drop over the turbine differs greatly from steam to toluene and CO<sub>2</sub>. The calculated enthalpy drop for the single pressure steam cycle turbine is about 6 times higher than the enthalpy drop over the ORC cycle turbine, and almost 17 times higher than that of the CO<sub>2</sub> cycle turbine. According to Larjola [30], the low specific enthalpy drop typical for organic fluids eases the turbine design, resulting in that a single stage turbine can often be used instead of a multistage turbine as is required when operated with steam. The enthalpy drop of toluene over the turbine is only 1/6 of the enthalpy drop for the corresponding steam cycle. As a rough estimation, this can be converted to that the necessary stages of the toluene turbine is 1/6 of the necessary stages of steam turbine. This would mean a significantly shorter and lighter turbine. While the volume flow is larger suggesting an increased turbine exit area, the low enthalpy drop reduces significantly the length and weight of the turbine, supporting the claim that more compact turbines can be used for ORC systems.

The characteristics of the CO<sub>2</sub> cycle indicate that the turbine can be made very compact, which also agrees with other evaluations. Immediately, such a compact design can be considered to be beneficial, but the high pressure and high energy density of such a system can be a concern of practicality and safety of operation.

A concern with steam turbines is droplet formation as the steam quality drops during expansion. This can cause corrosion in the turbine and erosion of the turbine blades. Limit of steam quality is therefore given to prevent this. This is not an issue when operating with toluene or CO<sub>2</sub>, as there is no condensation during the expansion, easing the strains on the turbine in comparison to steam as working fluid.

### 5.3 Implementation on vessel

Legislation and component classification is a concern when new methods and technology is planned to be implemented on marine vessels. Components need to be certified according to class, and legislations regarding safety and fire protection are much more restrictive concerning offshore compared to onshore application.

Steam turbine systems have been used onboard ships for decades, and should not have any difficulty of being certified. Organic fluids such as toluene as working fluids, on the other hand, can impose stricter conditions for implementation being a flammable and toxic fluid. One concern is that double piping for the organic fluid could be demanded to achieve classification. This would increase the required space and complicate the use of ORC systems. No suggestions of regulations for an ORC onboard a vessel has been made known by any class society at present time. CO<sub>2</sub> in comparison to toluene is not flammable or toxic. CO<sub>2</sub> is also a widely used on vessels as fire extinguisher in machine rooms, and other rooms with sensitive instruments. The biggest concern with the CO<sub>2</sub> cycle is rather the high operating pressures, and how to ensure safe operation and handling of the system. The ABC bottoming cycle alternative applies well known technology, operate with low cycle pressures, and use air as working fluid. None of these factors raises concern for obtaining classification.

## 5.4 Summary of findings

In Table 11, a summary of the results and evaluations of the combined cycle for scenario (1) is given. Scenario (1) being the case of a combined cycle output of 18 MW in all the alternative solutions.

**Table 11 Summary of evaluations for scenario (1)**

|  | Efficiency | Fuel savings | Complexity | Availability | Part load characteristics |
|--|------------|--------------|------------|--------------|---------------------------|
| CC gas turbine and single pressure steam cycle                 | 46.3%      | 25.6%        | high       | high         | poor                      |
| CC gas turbine and dual pressure steam cycle                   | 48.2%      | 29.4%        | medium     | high         | poor                      |
| CC gas turbine and ORC   | 45.6%      | 25.4%        | low        | medium       | good                      |
| CC gas turbine and ABC   | 41.9%      | 18.7%        | low        | high         | poor                      |
| CC gas turbine and CO <sub>2</sub> transcritical Rankine cycle | 42.1%      | 19.1%        | high       | low          | medium                    |

This study has primarily focused on optimal power output and efficiency, but when evaluating the optimal machinery system, there are other important factors to be considered, such as complexity, availability, cost and size. Cost often correlates with availability of the system, where high availability corresponds to developed components available on the market, and low availability corresponds to a need for development.

Though the dual pressure steam cycle gives the best combined cycle efficiency, it is a complex and spacious system. The single pressure steam cycle relieves some complexity and components, but has lower efficiencies as well as the system requiring water treatment and manual handling.

The ORC is generally viewed as a less complex system, and can be automatically controlled. The pressure levels are also substantially lower than for a steam cycle, reducing mechanical stresses on components and pipes. The oil loop ensures more stable operation of the bottoming cycle during part load, compared with steam, where the reduced temperature of the heat source has adverse effect on the steam cycle power output. Still, the ORC technology is less mature, and the components are potentially more costly as it is less developed. The working fluids applied for an ORC also constitutes health and fire hazard if there were leakages, whereas steam and air are safe alternatives.

The ABC system applies well-known principles as well as being a simple system. The pressure levels are low, and there is no steam generation. While there is no need for a HRSG unit, the heat transfer between the exhaust gas and the air is less efficient than steam generation. But considering ease and maturity of the technology, it still represents a reasonable solution with relatively good efficiencies compared with operation by gas turbine alone. The ABC is an alternative that could be evaluated as a solution if regulations impose restrictions for organic fluids, making use of ORC difficult.

The CO<sub>2</sub> bottoming cycle showed results comparable with the ABC solution. While the results are comparable, the CO<sub>2</sub> cycle demands very high pressures and is not a technology that has been tried out. It is suggested that the components of the CO<sub>2</sub> cycle can be made very compact due to the high densities of the cycle, but components are not available, and will need development. Considering the results of the CO<sub>2</sub> cycle being similar to that of the ABC, as bottoming cycle, the ABC would be the preferred solution of the two based on technology maturity and simplicity of the ABC.

## 6. Conclusion

This report compared five different combined cycle configurations evaluating the most suitable machinery system for the future high-speed Incat ferry operated by Mols-Linien.

Based on the given assumptions and specifications, the results suggest that a combined gas and steam cycle is the most energy efficient solution, having the highest thermal efficiency. Still, the combination of gas turbine and ORC shows competitive results with a combined cycle thermal efficiency of 45.6 % versus 46.3 % for the gas and single pressure steam combined cycle.

The results for the combined cycles with ABC and CO<sub>2</sub> transcritical Rankine cycle as bottoming cycles are not equally competitive with gas and steam combined cycle efficiencies. Still, they presents an increase in thermal efficiency of about 8%-points compared with the thermal efficiency of the gas turbine for a combined cycle power output of 18 MW. The ABC cycle is also, in comparison to other combined cycles, a simple system applying well-known technology.

Defining the optimal solution is dependent on which characteristics are of considered most important, it being performance, size, low complexity or technological maturity. In terms of greatest efficiency, and thereby fuel savings, a dual pressure steam cycle shows the best result. An ORC, on the other hand, is less complex, offer good part load operation and automation of operation, while obtaining good combined cycle efficiency. While showing promising results, the ORC is a technology that is more researched than practiced, and class and regulations are not finalized on the application of organic fluids. The ABC cycle applies well-known technology and is a simple and compact system, but the combined cycle efficiency is not as good as steam or ORC. The CO<sub>2</sub> transcritical Rankine cycle as bottoming cycle obtained similar combined cycle results as the ABC solution. The cycle is operated with very high pressures and the results suggest that the components of the system can be made very compact, but it is not a developed technology and with the relatively poor results in efficiency, it must be evaluated if it is of economic interest to design the components needed for the system.

The combined cycle solution with ORC as bottoming cycle can be recommended as a machinery solution that will provide good thermal efficiencies and is less complex. While this being said, it is necessary that further work is done evaluating the component sizes for the ORC system, assuring that the weight and space requirements are held, while the thermal efficiency is kept high. If the combined cycle efficiency drops as a consequence of adapting the heat exchanger, the ABC system should be considered as an alternative.



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