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Combined heat and power plant on offshore oil and gas installations

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Abstract

Implementation of energy efficient technologies is an issue of growing importance for the offshore oil and gas industry. Rising awareness of increasing levels of CO_2 in the atmosphere is a major driver in this move, with a main aim being to reduce the amount of released CO_2 per unit of oil or natural gas produced. Subsequently, the addition of steam bottoming cycles to exploit exhaust heat from gas turbines offshore has become an attractive alternative. This paper will investigate two different combined cycle configurations, namely the extraction steam turbine- and the backpressure steam turbine-cycle. Both cycles were modelled using the process simulation software Ebsilon Professional with a single GE LM2500+G4 gas turbine as a topping cycle, and a once-through heat recovery steam generator to exploit GT exhaust heat.At design, the steam turbines produced 8.2 MW and 6.0 MW respectively, achieving net thermal efficiency of 45.5%/42.1%. This was a 12.3 pp/8.9 pp increase compared to the simple cycle GE LM2500+G4 configuration.

The findings suggest that a backpressure steam turbine could be an attractive option for oil producing facilities with high demand for process heat, while an extraction steam turbine configuration is more suited to gas producing facilities with lower heat requirements and a higher demand for power and flexibility. Additionally, both cycles displayed a substantial reduction in emitted CO_2 per MWh produced, with reductions 26% and 21% compared to the simple cycle configuration achieved for the extraction and backpressure cycle respectively.

Keywords: combined cycle; process simulation; heat recovery; compact steam cycle; cogeneration; off-design; extraction steam turbine; back-pressure steam turbine

1. Introduction

Implementation of energy efficient technologies is an issue of growing importance for the offshore oil and gas industry. Rising awareness of increasing levels of CO_2 in the atmosphere is a major driver in this move, with a main aim being to reduce the amount of released CO_2 per unit of oil or natural gas produced. In Norway, 26% of the total CO_2 emissions originate from petroleum related activities on the Norwegian Continental Shelf (NCS), where gas turbine (GT) emissions account for 79% [1]. GTs are important installations offshore, as they provide mechanical drive, power generation and process heat. As an incentive to implement energy efficient technologies, the Norwegian parliament introduced a CO_2 tax on combustibles from petroleum related activities. As of 2015, this taxation is set at 1 NOK/Sm^{3}of burnt Natural Gas (NG) or approximately 428 NOK/ton of released CO_2 (about 52 $/ton released CO_2).

Given GTs' contribution to the total offshore CO_2 emissions, their use has attracted much public and academic attention. One result of the high taxation cost enforced by the Norwegian government is that efficiency enhancing technologies are starting to become economically feasible. The majority of offshore electric power generation is done by simple cycle GTs, with approximately one third of them being fitted with a waste heat recovery unit (WHRU). As a result, hot exhaust gases are in many cases released directly to the surroundings and represent a substantial loss in exergy. Exploiting this heat is considered to be one of the most environmentally friendly acts available for the purpose of reducing offshore CO_2 emissions [2].

A co-generative steam bottoming cycle, utilizing GT exhaust heat to produce both heat and power, is a promising solution to reduce CO_2 cost per generated MW offshore. Implementation of combined cycles (CCs) was already being discussed, and to some extent implemented, in the 90s [2–4]. However, challenges related to weight and size limitations has limited the use of steam bottoming cycles offshore. These issues were later addressed [5], and weight and efficiency compromises were evaluated [6]. Once-through heat recovery steam generators (HRSGs) were identified as attractive options for offshore installations, and the perfor-

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mance of a cycle was tested with design and off-design simulations [7]. Net plant efficiency improvements of 26-33% and emitted CO_2 reduction of 20-25% were indicated as achievable goals for compact CCS. A refinement of the design for such steam bottoming cycle was also presented, based on a detailed combined cycle model and numerical optimization tools [8]. Another study evaluated the site-scale integration of steam bottoming cycles for offshore applications [9]. A methodology to optimize such systems is presented and the most critical variables influencing the process configuration and the performance were pinpointed. Nguyen et al. [10] pointed out that the introduction of steam bottoming cycles is a complex design task. The best layout is strongly influenced by the specific characteristics of the selected offshore facility.Working fluids other than steam have also been considered in the literature. Bhargava et al. [11] carried out a comprehensive evaluation of bottoming organic Rankine cycle (ORC) for offshore applications, showing the potentials of the technology. Three process configurations were studied, and the relative advantages and limitations discussed. In a specific case study concerning an offshore facility in the North Sea, different waste heat recovery technologies were assessed [12]. The ORC demonstrated to slightly outperform the steam Rankine cycle, while the air bottoming cycle was discarded as the less attractive option. The optimal design of the ORC was further studied through a multi-objective optimization methodology [13]. The utilization of an ORC was also evaluated for a Brazilian floating production, storage and offloading (FPSO) unit [14]. An exergy analysis at different field conditions showed that the combined cycle was beneficial independently of the variation of the chosen production parameters.

The power-heat demand of an offshore installation changes during its lifetime. The amount of process heat and power needed is highly dependent on reservoir conditions and the corresponding processing equipment necessary for offshore production. A generalized topside processing system can consist of the following: production manifold, separation, oil treatment, gas treatment, condensate treatment, gas re-compression and water treatment. All of these processes have different requirements to power and heat consumption. Based on a case study by Nguyen et al. [15] the required temperature for process heat in different processing equipment is given in Table 1.

Table 1: Temperature of heating processes

Process	Temperature range, °C
Fuel gas heating	40–60
Crude oil heating 1^{st} separation stage	45–55
Crude oil heating 2^{nd} separation stage	80–90
Condensate stabilization column, reboiler	180–200
Gas dehydration, TEG reboiler	205

The primary heat demand was identified as crude oil heating in the separation process followed by the reboiler for condensate stabilization. These results are supported by Voldsund et al. [16] whose exergetic case studies identified heat requirements in separation trains as dominant. Nguyen et al. [17] also identified the three dominating power consumers on offshore facilities as being: the compression train, seawater injection pumps and gas recompression, in that order.

The result of these requirements is that different combined cycle plant configurations can be attractive depending on the needs of the installation. This paper will look at two different configurations, a back-pressure steam turbine and an extraction condensing steam turbine, identifying pros and cons with each configuration and based on processing needs, determine which is the preferred option.

2. Methodology

2.1. Process description

The extraction steam turbine cycle is best described as a steam Rankine cycle with an extraction configuration for the turbine. The layout of the extraction steam turbine cycle is illustrated in Fig. 1.

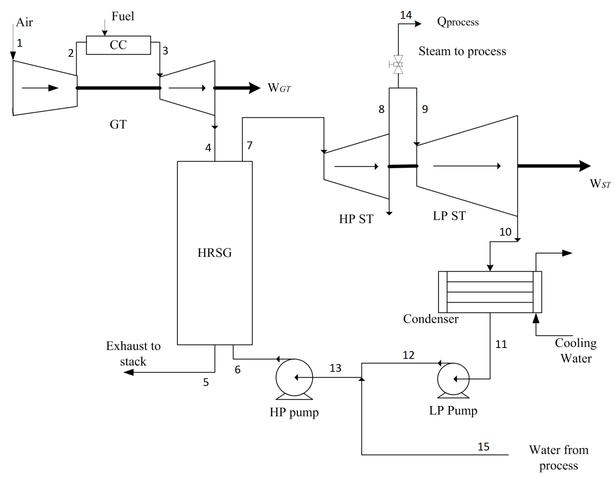

Figure 1: Layout of extraction steam turbine cycle

Extraction of steam (14) can be carried out by holes through the casing or from piping between stages as shown in the figure. The cycle has the option to vary the amount of process heat delivered, which is controlled by a control valve. This flexibility could potentially ease operation but comes at the price of limited power generation through the later turbine stages. The process heat will be equal to the latent heat extracted between points (14) and (15), available heat in the superheated region at (14) is considered as a loss. Saturated water returning from process is reintroduced (15) after the low pressure (LP) pump through a mixing valve.

The layout of the backpressure steam turbine cycle is different from the extraction steam turbine cycle in the sense that all steam passes through the steam turbine before available latent heat is extracted as process heat. An illustration of the cycle is given in Fig. 2.
This configuration results in limited power generation compared to the extraction case, as the pressure level at the turbine exit must be higher in order to deliver the required process heat temperature. Another consequence is that electric

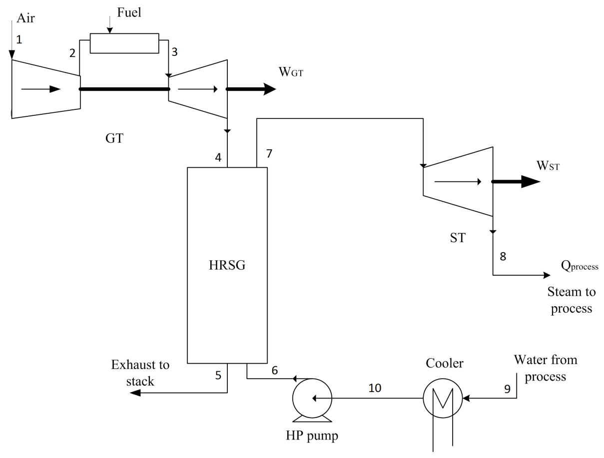

Figure 2: Layout of backpressure steam turbine cycle

power output will be fixed according to the specific amount of process heat delivered. However, an advantage of this configuration is that heat delivered to the steam cycle can be better utilized as there is no heat rejected in a condenser.

2.2. Model description

Simulations were performed in the process simulation software Ebsilon Professional version 10 patch 6 [18]. The software simulations were carried out at steady state conditions through calculative iterations with values from extensive fluid and material libraries. The following property relations/formulations were used:

- Steam Table IAPWS-97
- Saltwater Lib-SeaWa (2009)
- Real gas formulation of Stodolas Law
- Real gas formulation for gases (N_2, Ar, O_2, CO_2, SO_2, H_2O), for other gases the ideal gas approach was used

Cycles were designed in the graphical window with equipment components, measure points, boundary conditions and input values. The boundary condition assumptions were chosen based on recommendations for power plant modelling [19] and studies with similar simulations towards offshore power generation [6, 7] and are presented in Table 2.

Table 2: Boundary condition assumptions

Boundary conditions	Value
Ambient temperature, T_{amb} , °C	15
Ambient pressure, P_{amb} , bar	1.013
Ambient RH, %	60
Cooling inlet temperature, $T_{cw,in}$, °C	10
Max cooling temperature difference, $\Delta T_{cw,in}$, °C	10
Cooling water pressure, $P_{cw,in}$, bar	2
Cooling system	Direct water system
Cooling medium	Seawater

Power generation on offshore oil and gas installations may be subject to substantial variations. Knowledge of off-design behavior of the different co-generative concepts are therefore important. The following off-design simulations were carried out for this analysis:

- Four different process supply temperatures were investigated for the two cycles
  - 100°C
  - 120°C
  - 150 °C
  - 175 °C
- Calculations were carried out for each 10% step of GT load in the range 40-100%
- For the extraction steam turbine cycle, process heat was varied from 0 to the maximum allowed (further described in Section 3.2)

The simulation component specifications and off-design modelling behavior will be presented in the following sections, followed by the respective model descriptions.

2.2.1. Gas turbine

The aeroderivative gas turbine General Electric LM2500 +G4 was chosen as the GT topping cycle. The +G4 model is the latest addition in the LM2500-series which is one of the most commonly used GTs on the NCS. Operational values for the gas turbine were obtained from the "Gas Turbine Library" developed by VTU Energy for the Ebsilon Professional software. The library contains individual gas turbine operation characteristics in accordance with industry standards for gas turbine acceptance, namely ISO2314 and ASME PTC22.

In order to maintain operational flexibility offshore, the co-generative plant is designed for a gas turbine load lower than maximum load. For the subsequent simulations, a gas turbine operating load of 70% was chosen for the steam bottoming cycle, allowing for flexibility in the power generation. In addition, the outlet pressure was set to 1.045 bar in order to compensate for pressure loss in the OTSG. All GT parameters chosen for the simulation can be seen in Table 3.

Table 3: GE LM2500 +G4 Parameters

Parameter	Value
Model type	GE LM2500 +G4
GT fuel	Methane
Lower Heating Value, kJ/kg	50047
GT inlet pressure drop, $\Delta P_{GT,inlet}$, bar	0.010
GT outlet pressure, $P_{GT,outlet}$, bar	1.045

During off-design simulations GT performance was regulated by pre-determined turbine characteristics in Ebsilon's VTU GT library. Models and characteristics in the library are developed in cooperation with GT manufacturers and are based on real operation data. This eliminates the need for additional validation of the characteristic curves.

2.2.2. Heat recovery steam generator

As mentioned in Section 1, once-through HRSGs were identified as an attractive option for offshore installations [6]. Based on these findings, an OTSG was chosen for the subsequent simulations. Ebsilon Professional V-10.6 does not include a specific component for this type of heat exchanger, but other components were reconfigured to achieve an OTSG model. Three components were specified to act as economizer, evaporator and superheater respectively, with a floating vaporization point within the three units, thus allowing similar operation as an OTSG. At design point the economizer was configured to deliver saturated water at the outlet, the evaporator to deliver saturated steam on the hot side, and the superheater to have an upper terminal difference of 30 K. The upper terminal difference of 30 K was chosen in order to secure sufficient thermal driving forces. In addition, based on the argumentation of Nord and Bolland [6], a minimum pinch point temperature of 25 K was specified to cohere with offshore size and weight limitations. Pressure loss through the OTSG was set to 25 mbar corresponding to recommendations from Bolland [19]. In Table In Table 4 the assumptions and specifications are summarized.

Table 4: OTSG simulation parameters

Parameters	Value
Pressure loss exhaust-side, $\Delta P_{OTSG,ex}$, bar	0.025
Pressure loss water-side, $\Delta P_{OTSG,H_2O}$, bar	0.050
Min. pinch point temperature ΔT_{PP}, °C	25
Terminal difference at the hot end ΔT_{SH}, °C	30

2.2.3. Steam turbine

Ebsilon has a steam turbine component which was selected for the simulation. Live steam pressure was set to 25 bar according to an offshore steam bottoming cycle optimization study [8]. All chosen parameters are given in Table 5.

Table 5: Steam Turbine simulation parameters

Parameters	Value
Live steam pressure, $P_{HP,steam}$, bar	25
Minimum steam quality	0.90
Turbine first stage isentropic efficiency, $\eta_{s_{ST,HP}}$	0.92
Turbine second stage stage isentropic efficiency, $\eta_{s_{ST,LP}}$	0.88

During part-load operation, the waste heat from the GT exhaust will vary. In order to match this change in delivered heat, most steam turbines in bottoming cycles are operated with a sliding pressure mode [20]. A sliding pressure mode allows the live steam pressure to gradually decrease down to approximately 50% of design pressure. A control valve fixes the pressure level for operation below 50%. Sliding pressure mode was used for the steam turbine component in all Ebsilon Professional simulations.

The swallowing capacity for a turbine stage in off-design operation with no extraction was calculated by Stodolas Law, Equation 1:

$$\frac{\dot{m}}{\dot{m}_0} = \frac{P_{inlet}}{P_{outlet}} \sqrt{\frac{P_{inlet,0}\upsilon_{inlet,0}}{P_{inlet}\upsilon_{inlet}}} \sqrt{\frac{1-\left(\frac{P_{outlet}}{P_{inlet}}\right)^{\frac{n_v+1}{n_v}}}{1-\left(\frac{P_{outlet,0}}{P_{inlet,0}}\right)^{\frac{n_v+1}{n_v}}}} \tag{1}$$

where suffix 0 represents the ST design point and $(n_v+1)/n_v$ is the relation for polytropic pressure-volume exponent. All steam turbines in the simulation were configured to use inlet pressure correction by Stodolas law in process calculations. The ST efficiency will change slightly during part load. Ebsilon Professional captures this change by adjusting performance based on volume flow variations. The correction characteristic used by Ebsilon is illustrated in Fig. 3.

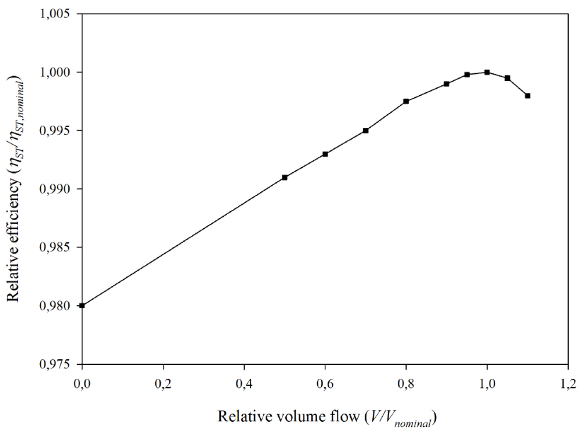

Figure 3: ST efficiency correction characteristic

2.2.4. Additional components

The condenser component is only valid for the extraction steam turbine cycle. It was configured to maintain a constant mass flow of cooling water during off-design simulations. By this, the variation in outlet pressure of the second turbine stage is dependent on the cooling water return temperature. Pressure drop on the cooling water side was set to 0.1 bar. Generator efficiency and isentropic pump efficiency were set according to recommendations from Bolland [19]. The parameters are summarized in Table 6.

Table 6: Additional machinery simulation parameters

Parameters	Value
Pump isentropic efficiency, $\eta_{s,pump}$	0.7
Generator efficiency, η_{gen}	0.985
Mechanical efficiency, η_{mech}	0.996
Condenser cooling-water pressure drop, bar	0.1

2.2.5. Extraction steam turbine model

In Fig. 4 the extraction steam turbine cycle is shown at design point from the Ebsilon Professional graphical window. The extraction steam turbine is built up of two steam turbines working in series, where steam can be extracted from piping between the two. Live steam pressure was set to 25 bar and steam was extracted at 10 bar. Extracted steam was further expanded through a valve to match the desired extraction supply temperature, 150°C at design, corresponding to steam saturation pressure of 5 bar. Process heat at design point was set to 5.0 MW to ensure flexibility in heat extraction. The decision was based on maximum allowed extraction in the system, calculated to be 10.5 MW. The LP pump and motor was set to operate in local off-design, corresponding to maximum mass flow at 100% GT load. This was in order to ensure operation within the pump characteristics during simulation. Given the 10°C allowed temperature increase in the cooling water through the condenser, a 20°C upper temperature difference between the condensing steam and cooling water was chosen. This provided sufficient driving forces and reduced the size of the condenser compared to an onshore installation. This resulted in condensation at 40°C, corresponding to a pressure level of 0.07 bar at design.

2.2.6. Backpressure steam turbine model

The backpressure steam turbine cycle at design point is shown in Fig. 5. The steam turbine was modelled with one ST component and backpressure of 2 bar, corresponding to a supply temperature of 120°C, at design point. An exit pressure of 2 bar was chosen after evaluation of off-design modelling showing higher supply temperatures giving a large penalty in power output. The reduction in electrical power output of backpressure of 5 bar (a supply temperature of 150°C) was 30% compared to the 2 bar case. All steam from the turbine continues directly to the process heat extraction. The returning saturated water has a temperature of 120°C, which is unrealistically high if the water is coming from a water treatment facility, storage tank or the process. An aftercooler was installed in the closed cycle design to lower the temperature to 60°C. Feedwater temperatures above 60°C is a rule of thumb in the industry, ensuring no condensation of water in the exhaust which could lead to corrosion in the OTSG [19].

2.3. Model validation

As cogenerative facilities are highly site specific it was determined to perform validation on an existing plant. The facility chosen for validation was the Oseberg D cogenerative offshore facility. Reference values for the simulation were taken from "Energy optimization on Offshore Installations with emphasis on Offshore Combined Cycle plants" by Kloster [2], and provided validation work on the same facility by Nord et al. [7], who acquired validation results within 0.1% error of actual data.

In Table 7 the power output from the validation is presented along with references. The results correspond to reference values with an accuracy above 99%. It is thus concluded that the systematic method for simulation is valid within a reasonable error limit.

Table 7: Validation results at design point, max power output, no process heat

Reference	Value
P.Kloster [2], kW	15800
L.O.Nord [7], kW	15800
Ebsilon Professional V-10-6 simulation, kW	15811

2.4. Definitions

In this section the relevant performance indicators used in this paper will be defined.

2.4.1. Power outputs

Gas turbine power output was defined as net work delivered to the shaft multiplied by the generator and mechanical efficiencies. The expressions for GT shaft power and power output are given in Equation 2 and Equation 3.

$$\dot{W}_{shaft_{GT}} = \dot{W}_{turbine} - \dot{W}_{compressor} \tag{2}$$

$$\dot{W}_{GT} = (\dot{W}_{shaft}\eta_{gen}\eta_{mech})_{GT} \tag{3}$$

The steam turbine power output was defined as the work produced by the steam turbine multiplied with generator and mechanical efficiencies, and is given in Equation 4.

$$\dot{W}_{ST} = (\dot{W}_{turbine}\eta_{gen}\eta_{mech})_{ST} \tag{4}$$

To avoid liquid formation in the last stages of the steam turbine a minimum required steam quality was determined. The steam quality is defined as the ratio of vapor mass flow rate to the total mass flow rate.

$$x = \frac{\dot{m}_{vapour}}{\dot{m}_{vapour} + \dot{m}_{liquid}} \tag{5}$$

The pump work is defined by:

$$\dot{W}_{pump} = (\dot{m}H)\eta_{pump}\eta_{motor} \tag{6}$$

where $\dot{m}$ is the mass flow through the pump and H is the pump specific head.

Net plant power output was defined as the sum of work from the GT and ST less the auxiliary power requirements. Auxiliary power requirements here represent the combined power consumption of all pumps in the system.

$$\dot{W}_{net,plant} = \dot{W}_{GT} + \dot{W}_{ST} - \dot{W}_{aux} \tag{7}$$

2.4.2. Plant efficiencies

GT thermal efficiency was defined as GT power output divided by the energy output of the fuel. The specific energy output is here expressed by the Lower Heating Value (LHV).

$$\eta_{GT} = \frac{\dot{W}_{GT}}{\dot{m}_{fuel}LHV_{fuel}} \tag{8}$$

Thermal efficiency of the combined cycle was defined as the plant net power output, Equation 7, divided by the fuel energy output. The relation is given in Equation 9.

$$\eta_{net,plant} = \frac{\dot{W}_{net,plant}}{\dot{m}_{fuel}LHV_{fuel}} \tag{9}$$

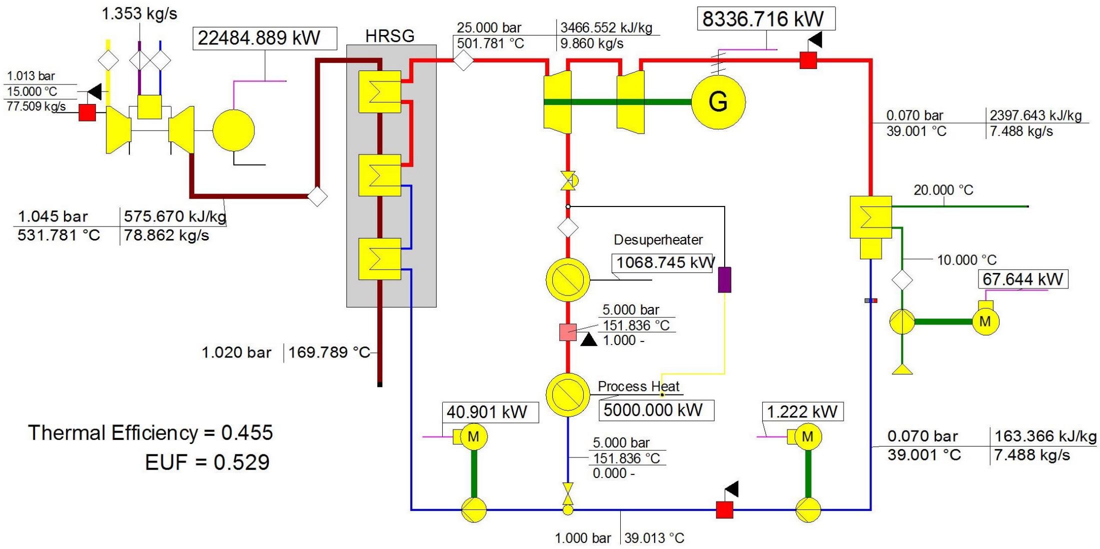

Figure 4: Layout of extraction steam turbine cycle

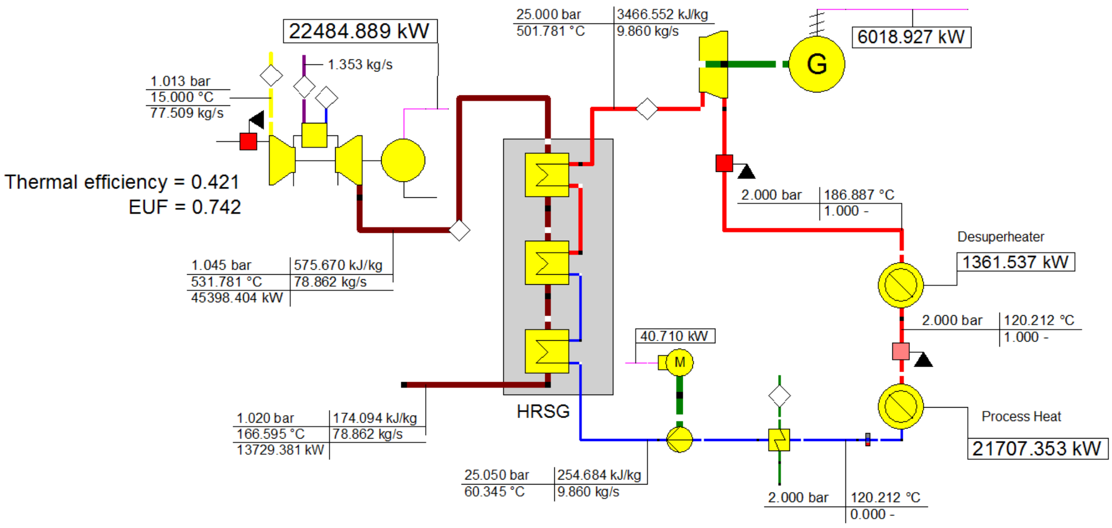

Figure 5: Layout of backpressure steam turbine cycle

In addition to the net plant power output, process heat is extracted and utilized. Useful process heat from the cycles was defined as the latent heat of steam, Equation 10. The available superheat was extracted in the simulations and regarded as a loss.

$$\dot{Q}_{process} = \dot{m}_{process} \Delta h_{vap} \tag{10}$$

This heat can be added to the net plant power output in Equation 9 providing an Energy Utilization factor. The Energy Utilization Factor (EUF) is a parameter which tells how much of the energy from the burnt fuel is being utilized either by power generation or by process heat.

$$EUF = \frac{\dot{W}_{net,plant} + \dot{Q}_{process}}{\dot{m}_{fuel} LHV_{fuel}} \tag{11}$$

2.4.3. Emission rate

Reducing CO_2 emissions is one of the major driving forces behind offshore combined-cycle installations. A useful measure for the achieved reduction is the CO_2 emissions rate. CO_2 emission rate (ER) from the plant was defined as the annual emitted CO_2 in the exhaust divided by the annual production of power in megawatt hours.

$$ER = \left(\frac{\dot{m}_{CO_2}}{\dot{W}_{net,plant}} \right)_{annual} \tag{12}$$

3. Results and discussion

3.1. Reference case

The reference case for the simulations was modelled as a simple cycle using the GE LM2500+G4 turbine operating at 70% load. The power output was calculated to be 22.5

MW equivalent to a thermal efficiency of 33.2%. In order to calculate the CO_2 emission rate, annual operating hours were assumed to be 8000 hours. This resulted in an annual electric power output of 18000 MWh. At a GT load of 70% the simulations showed the exhaust CO_2 content to be 3.79 kg/s, resulting in an ER equal to 606 kg/MWh. Given that no exhaust heat is utilized in the simple cycle configuration, the calculated EUF is equal to thermal efficiency at 33.2%.

3.2. Extraction steam turbine cycle

3.2.1. Design

The extraction cycle obtained a power output of 30.8 MW, a 37% increase compared to a simple cycle GE LM2500+G4. As determined in Section 2.2, delivered process heat was set to 5.0 MW. The overall amount of energy extracted between the turbine stages was 7.5 MW. Losses due to desuperheating were calculated as 1.1 MW. This is a substantial amount of heat loss and in a real plant this energy could have been utilized. The last unaccounted 1.4 MW is re-injected in the cycle as saturated water. Other losses add up to a total of 1.1 MW, including all pump work throughout the cycle. Lastly, a substantial amount of heat is rejected in the condenser (16.7 MW) and lost in the exhaust (12.8 MW). However, the low temperature of these streams makes energy extraction difficult.

The results of the simulation yields a thermal efficiency of 45.5% at design, an increase of 12.2 pp compared to the simple cycle case. Adding the 5 MW of extracted process heat, the calculated EUF reaches a value of 52.9%. The key findings are summarized in Table 8.

Table 8: Extraction steam turbine cycle, key simulation results

Parameters	Value
Fuel mass flow, $\dot{m}_{CH4}$, kg/s	1.35
Gas Turbine power output, $\dot{W}_{GT}$, MW	22.5
Steam Turbine power output, $\dot{W}_{ST}$, MW	8.3
Net plant power output, $\dot{W}_{net,plant}$, MW	30.7
Process heat extraction, $\dot{Q}_{process}$, MW	5.0
Combined Cycle thermal efficiency, $\eta_{net,plant}$, %	45.5
EUF, %	52.9
CO_2 emission rate, ER, kg/MWh	438

3.2.2. Off-design

Off-design simulations were carried out according to specifications given in Section 2.2. At each GT load, heat extraction was varied from zero to the maximum allowed. Maximum heat extraction was limited by Ebsilon Professional's restriction on deviation from nominal mass flow in the ST, whose upper limit is set to 46%. In Fig. 6 the calculated window of operation is given for a supply temperature of 150°C. The major operational trends for other supply temperatures were found to be identical to the 150°C supply case, and the presented results will thus be limited to this configuration. In the operational chart net electrical power output $\dot{W}_{net,plant}$ is plotted against extracted process heat $\dot{Q}_{process}$. Each solid line represents a given GT load, equivalent to a constant fuel input. The white squares marks net thermal efficiency points of the plant, and the dotted lines illustrates the trend. The design point is indicated with a circle in the center of the chart.

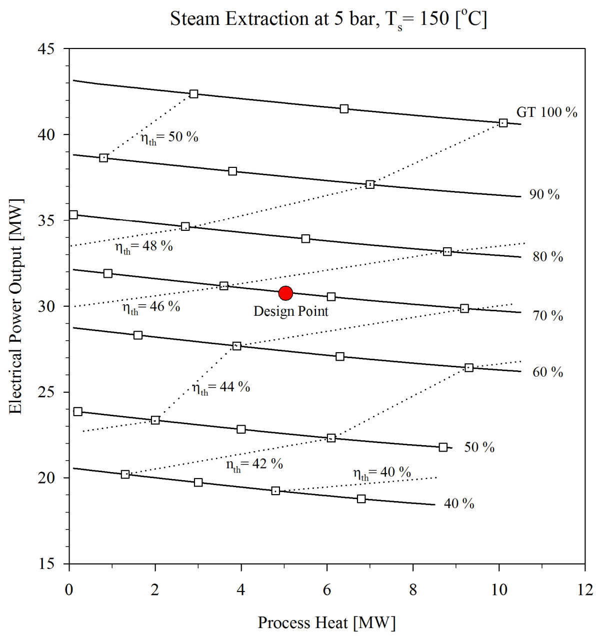

Figure 6: Calculated operational chart for extraction of steam at $T_s = 150°C$

It can be seen from the chart that a maximum process heat extraction of 10.5 MW is achievable. For 40 and 50% GT load, the extraction of process heat was limited by the pressure level in the piping between the two ST stages reaching 5 bar, the lower value of the sliding pressure mode in the ST. Further extraction is limited by a control valve, ensuring a minimum pressure level of 5 bar for the second ST stage. A bigger gap between the solid lines can be observed between 90-100% GT load. The reason for this deviation was an observed drop in the GT exhaust temperature in this range. The same phenomenon was observed in the range 50-60%. This drop in exhaust temperature resulted in a ST power output reduction of approximately 1.0 MW and 1.3 MW respectively.

In Table 9 detailed performance values for the plant are given for minimum and maximum heat extraction. A stable ST power output is observed in the GT 60-90% load range. The EUF for the system was stable at maximum extraction through the whole GT operating range, only experiencing a decrease of 4.3% between maximum and 40% GT load. Given the stable power output from the steam bottoming cycle, the variation in net thermal efficiency is credited to changes in GT efficiency at different loads.

3.3. Backpressure steam turbine cycle

3.3.1. Design

The backpressure steam turbine cycle generated a total of 28.5 MW, an increase of 27% compared to the simple cycle configuration. The power output from the steam turbine accounted for 6.0 MW, a noticeable decrease from the extraction configuration. Loss due to desuperheating was cal-

Table 9: Off-design results for extraction steam turbine at $T_s = 150°C$

No extraction		
GT load, %	$\dot{W}_{ST}$, MW	$\eta_{net,plant}$, %
100	10.8	50.9
90	9.8	50.2
80	9.6	47.3
70	9.6	47.3
60	9.5	45.7
50	8.0	45.0
40	7.9	42.7

Maximum extraction				
GT load, %	$\dot{W}_{ST}$, MW	$\dot{Q}_{process}$, MW	$\eta_{net,plant}$, %	EUF, %
100	8.3	10.5	47.9	60.5
90	7.4	10.5	47.1	60.8
80	7.1	10.5	45.5	60.3
70	7.2	10.5	43.7	59.3
60	7.0	10.5	41.6	58.5
50	5.8	8.9	41.0	58.0
40	5.8	8.5	38.3	56.2

culated as 1.4 MW, but as previously mentioned, parts of this heat can be extracted in a real plant. A noticeable increase is seen in extracted heat which reaches a value of 21.7 MW at 120°C supply temperature. This is more than four times the amount of process heat delivered by the extraction steam turbine at design. The aftercooler reduced the returning process water temperature down to 60°C, accounting to a loss of 2.5 MW. In addition, exhaust losses were similar to the extraction case at 12.6 MW, and other losses added up to a total of 1.0 MW.

The results of the simulation yields a thermal efficiency of 42.1% at design, an increase of 8.9 pp compared to the simple cycle case. Adding the 21.7 MW of extracted process heat, the calculated EUF reaches a value of 74.2%. The key findings are summarized in Table 10.

Table 10: Key simulation results for backpressure steam turbine cycle

Parameters	Value
Fuel mass flow, $\dot{m}_{CH4}$, kg/s	1.35
Gas Turbine power output, $\dot{W}_{GT}$, MW	22.5
Steam Turbine power output, $\dot{W}_{ST}$, MW	6.0
Net plant power output, $\dot{W}_{net,plant}$, MW	28.5
Process heat extraction, $\dot{Q}_{process}$, MW	21.7
Combined Cycle thermal efficiency, $\eta_{net,plant}$, %	42.1
EUF, %	74.2
CO_2 emission rate, ER, kg/MWh	474

3.3.2. Off-design

In Fig. 7 the operational line for the extraction steam turbine cycle at a supply temperature of 120°C is given.

Noticeable shifts in the curve can be observed in the GT load regions 90-100% and 50-60%. Again, this is a result of the drop in GT exhaust temperature, similar to the extraction case. A major difference between the two cases is the additional drop in delivered process heat for the backpressure cycle, arising due to the fixed relation between power output and process heat extraction. A detailed summary of the CC results along the operational line can be found in Table 11.

In the GT load range of 60-90% the steam turbine output is seen to be very stable, providing a power output ranging from

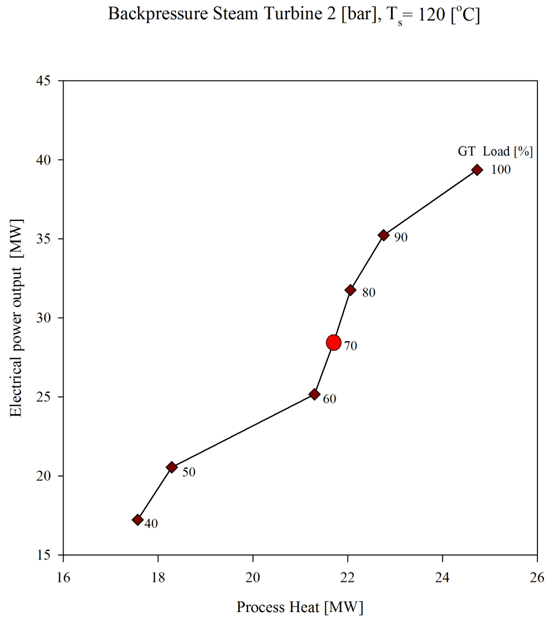

Figure 7: Operational line for backpressure steam turbine at 2 bar

6.0-6.4 MW. This suggests that deviations in power demand can be met by quick load variations of the GT, while maintaining a relatively stable power output from the bottoming cycle. The delivered amount of process heat is also fairly consistent for this operating range. The major changes in plant thermal efficiency can be credited to variations in the GT efficiency. For this range the EUF also remained stable at around 70%, which indicates good utilization of combusted fuel for an offshore application. At maximum GT load the system is able to deliver 7.1 MW of electricity and 24.7 MW of process heat. This results in a maximum thermal efficiency of 46.6% with an EUF of 72.2%. The thermal efficiency is lower compared to other designs, including the extraction case discussed in this paper, but the amount of process heat available for extraction can make up for this shortcoming, and can in some cases be sufficient to cover the entire heat requirement of an installation given supply temperature demand below 120°C.

3.4. Discussion

A premise to the discussion of the results needs to be made, in order to identify the limitations of the analysis presented. It is known that each offshore oil and gas installation displays specific characteristics, making difficult to outline universally valid evaluations. Despite that, this work tried to provide some general indications on the viability of the two combined cycle configurations studied. In accordance with this objective, some assumptions and simplifications had to be introduced. In the first instance, the plant heat requirements and the heat integration strategies were investigated in detail. The main parameter taken into account was the temperature at which the heat is made available. Neither the actual breakdown of heat requirements connected to different types of offshore installations nor the methods to sup-

Table 11: Results for backpressure steam turbine off-design simulation at 2 bar $T_s = 120°C$

GT load, %	$\dot{W}_{GT}$, MW	$\dot{W}_{ST}$, MW	$\dot{Q}_{process}$, MW	$\eta_{net,plant}$, %	EUF, %
100	32.3	7.1	24.7	46.6	72.2
90	29.0	6.3	22.8	45.7	71.6
80	25.8	6.1	22.1	44.2	71.0
70	22.5	6.0	21.7	42.1	70.1
60	19.2	6.0	21.3	40.1	69.8
50	15.9	4.7	18.3	38.9	69.1
40	12.7	4.6	17.6	36.0	67.9

ply heat to the various processes were analyzed. Another key assumption which is worth to point out regards the plant power output. The additional power, resulting from the addition of a bottoming cycle, was considered as a usable output. This may not always be true, as offshore installations can be isolated systems. If a specific power output is requested, the GT in a combined cycle arrangement would need to operate at a lower load compared to the reference case, resulting in lower gains in terms of thermal efficiency and CO_2 emissions reductions. Given this necessary premise, the results obtained can be discussed.

Simulations of the extraction steam turbine cycle revealed promising operational characteristics. Firstly, an increase in thermal efficiency of 13.3 pp was achieved in comparison to the simple cycle GE LM2500+G4 case, when the GT is operating at the same load. The effect on thermal efficiency decreased for higher GT loads, but the ST delivered stable power outputs for a large operational window. The system displayed good flexibility in heat and power output, allowing for heat extraction from 0 to 10.5 MW for a range of GT loads. Such operational flexibility would potentially ease the implementation process in existing facilities. Calculated net thermal efficiency is in the expected range compared to previous work by Nord and Bolland[6], ranging from 38.3 to 50.9%. However, the maximum obtained EUF of 60.8% might not be enough to out-compete other configurations, such as the mentioned WHRU configuration, for installations with high heat demand. In Section 1 the different heat and power requirements for offshore installations were discussed. The primary heat demand was identified as crude oil heating in the separation process followed by heat to the reboiler for condensate stabilization, while the three dominating power consumers were identified as being the compression train, seawater injection pumps and gas recompression. It can thus be concluded that offshore production from a gas reservoir will require less process heat than production from an oil reservoir in addition to a higher demand for power and power flexibility. For such an installation the extraction steam turbine could be a suitable option. The current system design does not deliver the required steam temperature for reboiling of condensate and regeneration of TEG for dehydration, which requires temperatures above 200°C. It is assumed that the heat to that process could be provided by means of another heating system, such as an electric heater. In order to supply that temperature without additional systems, other configurations could be considered. For instance, steam could be extracted in front of the turbine or directly from the OTSG or, alternatively, a fraction of the GT exhaust gases could be directly used for providing process heat. Although these options could be interesting to study, they were out of the scope of this work and, thus, were not further investigated. With regard to the CO_2 emissions, the calculated CO_2 ER at design was reduced by 27% compared to the simple cycle configuration, making the system interesting from an environmental and, possibly, an economical point of view.

The backpressure steam turbine cycle did not deliver as much power as the extraction case with a ST power output of 6 MW and a thermal efficiency of 42.1% at design. However, delivered process heat was substantially higher at 21.7 MW with a supply temperature of 120°C, resulting in an EUF of 74.2%. This is an increase of 41 pp compared to the simple cycle case. Higher supply temperatures of 150 and 175°C resulted in a substantial power penalty, making the backpressure cycle unfavorable for these specifications. Looking back to Section 1, the dominating heat consumption in the separation trains require supply temperatures in the range 45–50°C. This makes the backpressure steam turbine cycle a good option for oil producing facilities with high demand for process heating. Additionally, the cycle could be used for water treatment (boiling of seawater) and flue gas heating. As for the extraction case, to fulfil heat requirements above the design temperature, either additional heating systems would need to be used or alternative process configurations would need to be applied. A large drawback for the backpressure steam cycle was the fixed relation between generated heat and power, which restricts operational flexibility. The calculated CO_2 ER was 474 kg/MWh, a reduction of 21% compared to the simple cycle GE LM2500+G4 configuration. This is a lower reduction than the extraction steam cycle generated, 27%, however the reduction is still substantial, especially when the increased EUF is taken into account.

4. Conclusions

This paper looks at two possible combined cycle configurations for off-shore oil and gas installations: the extraction steam turbine cycle and the backpressure steam turbine cycle. Compared to a simple cycle GE LM2500+G4 the results show that installation of a bottoming cycle will deliver a noticeable increase in net thermal efficiency. At design and at constant GT load, the most promising configuration was the extraction steam turbine, delivering a ST power output of 8.3 MW and achieving net thermal plant efficiency of

45.5%. This constitutes an increase of 13.3 pp compared to the simple cycle case. The backpressure steam turbine cycle produced a ST power output of 6.0 MW, resulting in net thermal efficiency of 42.1%. Despite achieving lower thermal efficiency than the extraction case, the backpressure configuration delivered approximately four times the amount of process heat, making it a highly attractive alternative for offshore installations with high heat demand below 120°C. Oil producing facilities with high heat demand in separation trains could be an ideal fit. The penalty in power output makes backpressure steam turbines unattractive for integration in systems with high temperature heat demands. For such installations, in addition to high demands of power and flexibility, the extraction steam turbine could be a more interesting alternative, although the amount of process heat required should be considerably lower and not exceed 150°C in supply temperature. A gas producing installation is thus found more fitting to the extraction steam turbine cycle. None of the investigated cycles was found suitable for delivering process heat at high temperatures without considering substantial modifications of the process configuration.

The extraction steam turbine demonstrated very good flexibility that could ease implementation in a real life system. The biggest drawback for the backpressure steam turbine was the fixed power/heat relation, making it more challenging to implement. Simulations showed a substantial reduction in emitted CO_2 per MWh produced for the two cycles investigated. Within the framework of the proposed analysis, a reduction of 26% and 21% in ER was seen for the extraction and backpressure steam turbine, respectively. These results illustrate that reduction in emitted greenhouse gases is possible with available bottoming cycle technology.

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