Energy 230 (2021) 120634

Contents lists available at ScienceDirect

Energy

journal homepage: www.elsevier.com/locate/energy

Achieving 50% weight reduction of offshore steam bottoming cycles

Marit J. Mazzetti [a, *], Brede A.L. Hagen [a], Geir Skaugen [a], Karl Lindqvist [a], Steinar Lundberg [b], Oddrun A. Kristensen [b]

[a] SINTEF Energy Research, Kolbjørn Heijes vei 1 D, 7034, Trondheim, Norway
[b] Marine Aluminium AS, Husøyveien 165, 4262, Avaldnes, Norway

ARTICLE INFO

Article history:
Received 29 October 2020
Received in revised form
26 February 2021
Accepted 11 April 2021
Available online 23 April 2021

ABSTRACT

Adding a bottoming cycle to the gas turbines powering offshore oil and gas production plants allows additional power to be produced from recovered excess heat. Hence, the power demand of the platform can be met by burning less natural gas, and the CO_2 emissions reduced by up to 25%. However, the weight of the current bottoming cycles must come down to enable widespread implementation. This work presents a thorough weight minimization of a steam bottoming cycle utilizing gas turbine exhaust heat. Unconventional, but feasible designs of heat exchangers, ductwork and structural components are considered along with materials switching. Overall weight reductions of 38% and 52% were achieved for a 16 MW and a 12 MW offshore bottoming cycle respectively when compared to a 16 MW reference system. Key factors in achieving the weight reduction were the use of small steam generator tubes with an inner diameter of only 10 mm, improved condenser design and the use of aluminium structural framework replacing steel. By more than halving the weight of the bottoming cycle, it's implementation potential on offshore platforms has been greatly improved and can move the oil and gas industry towards significantly reduced CO_2 emissions.

1. Introduction

Offshore oil- and gas production is a very energy-intensive process: Typical installations offshore have power demands in the 60–80 MW range. The power required is usually supplied by LM2500 gas turbines in simple cycle configuration with efficiencies of only 38% [1]. The remaining energy is expelled as waste heat in the exhaust leading to unnecessarily high CO_2 emissions. There is a large potential for efficiency improvements through implementing a bottoming cycle that produces power from the hot exhaust gases, Fig. 1. Current offshore heat recovery is mostly limited to meeting field-specific heat demands. Only three of about 90 platforms on the Norwegian continental shelf have bottoming cycles installed for more efficient power production, primarily due to lack of available weight allowance on oil and gas platforms. Other factors can be available space or operational reliability issues [2]. In Norway the oil- and gas production emits 13.4 Mega tonnes/year of CO_2, accounting for about a quarter of total emissions in 2018 [5]. Widescale implementation of steam bottoming cycles could reduce CO_2-emissions from offshore platforms by 15–25% [2–4].

The reduction in CO_2 emission from adding a bottoming cycle will not be determined in this work as there is not enough process data available for the platform. However, the reduction in CO_2 emission from adding a bottoming cycle to an LM2500 +G4 turbine similar to the one used in this work LM2500, was found to be 25% or a reduction from 517 gCO_2/kWh to 388 g CO_g/kWh in previous work by Mazzetti et al. [3]. In addition, a case study was shown for a platform where adding a CO_2 bottoming cycle to an offshore gas turbine reduced the CO_2 emissions from the platform by 63 000 tonnes/year, a 22% reduction [3].

In a study of combined cycles for offshore oil and gas installations by Nord et al. [27] it was shown that the emitted CO_2 could be decreased by 20–25% by opting for a combined cycle rather than a simple cycle gas turbine for an offshore platform. Improving energy efficiency by adding a steam bottoming cycle can be a cost-effective climate measure. In Norway where there is a CO_2 tax offshore, the capital expenditure from installing a bottoming cycle can be repaid within 2–6 years due to saved operational costs from less fuel use and reduced CO_2 taxes [3].

Optimization of heat recovery bottoming cycles has been

* Corresponding author.
E-mail addresses: marit.mazzetti@sintef.no (M.J. Mazzetti), brede.hagen@ntnu.no (B.A.L. Hagen), geir.skaugen@sintef.no (G. Skaugen), karl.lindquist@barco.com (K. Lindqvist), slundber@online.no (S. Lundberg), oddrun@m-a.no (O.A. Kristensen).

https://doi.org/10.1016/j.energy.2021.120634
0360-5442/

Nomenclature

EDI	Electro-deionization
HRSG	Heat recovery steam generator
ORC	Organic Rankine Cycle

total layout volume of the HRSG was set constant and considered a constraint. In this study, only the pressure levels of the steam cycle were free process variables. They applied different objective functions like maximum net power production, maximum recovered heat, and minimum entropy generation. They used their model to show how the maximum net power and steam production varied, depending on the inlet gas temperature and the system configuration. Rezaie et al. [10] optimized an HRSG with minimum cost as

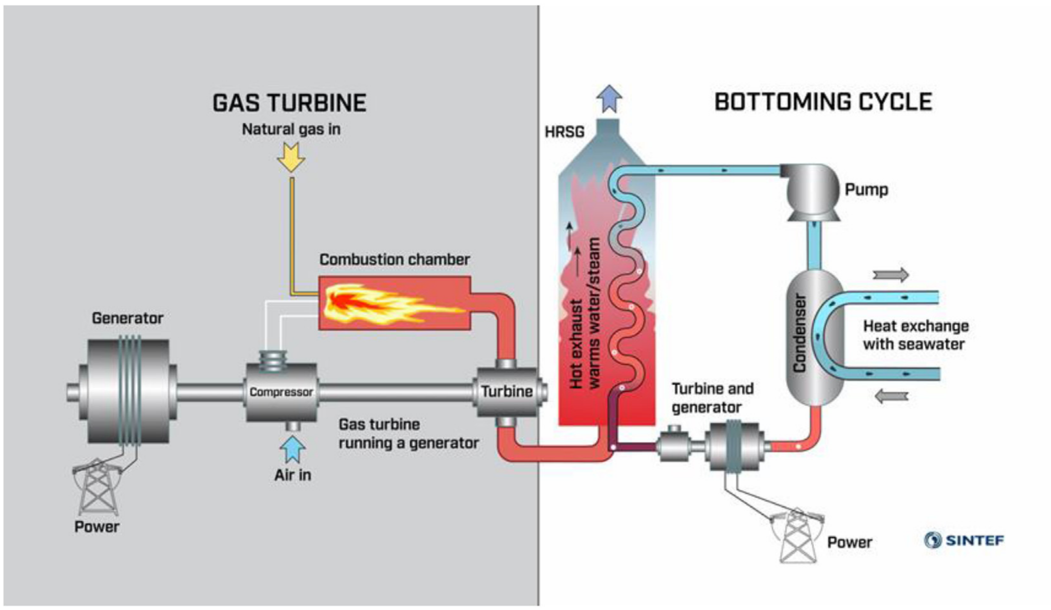

Fig. 1. Schematic of gas turbine with steam bottoming cycle.

considered previously in the open literature. Franko and Giannini [6] performed a two-level optimization of a HRSG. The optimum steam bottoming cycle operating conditions and system layout were found by minimizing the thermal exergy losses for given heat output through thermodynamic calculations. The optimized operating conditions from the first-level analysis were used as basis for the subsequent HRSG geometry optimization. The geometry of each section was optimized with respect to maximizing the compactness (Heat transfer area/Volume). Improvement over the existing design was shown by both reducing the exergy losses and increasing the compactness. However, the resulting HRSG size and weight was not reported. Manassaldi et al. [7] performed a similar two-level optimization of a HRSG. The independent variables in the second level optimization were the tube diameter, fin- (height, pitch, and thickness), and the tube layout (described by tube pitches and the number of tubes in across and along the gas flow) of the HRSG sections. Their objective was to either maximize the power output, maximize net power over material weight for a specified minimum net power output, or maximizing the recovered heat. They demonstrated how the choice of objective function impacted the overall size and volume of the HRSG. Mehrgoo and Amidpour [8] optimized the HRSG design with the objective of minimizing the entropy generation. They simultaneously optimized HRSG tube geometry (diameters, pitches, and the numbers in the transversal and longitudinal direction) and process variables like superheater outlet temperature and water/steam flow rate. They used the various values for the HRSG volume as a constraint parameter and showed how the entropy generation decreased with increasing volume. In Ref. [9] they did a similar study where the the objective based on tube and fin material cost. The independent variables governed the geometry for each section (economizer, evaporator, and superheater). The process conditions like capacity, steam temperature and flow rate were assigned values of an existing system. The weight of the finned tube bundle was reduced by 23% while having the same heat duty.

Refs. [6–9] considered the HRSG of multi-pressure level steam bottoming cycles, see Table 1. Despite these systems being the preferred solution for onshore applications, the single-pressure level systems have been recommended for offshore applications due to the limited space and weight on offshore platforms [4]. Indeed, the limited studies in the open literature concerning offshore bottoming cycles considered single-pressure level systems. Pierobon et al. [11] presented a comprehensive comparison of an organic and a steam Rankine and an air Brayton bottoming cycle. The organic and steam Rankine cycles outperformed the air bottoming cycle, with the steam cycle being economically superior to the ORC. This study evaluated maximum net present value (NPV) as the main objective function as a function of heat exchanger weight (OTSG + Condenser). The weight calculations only included the finned tube bundle and the tube layout (tube pitch), and fin geometry did not seem to be part of the optimization.

Nord et al. [12] used the automated component sizing feature of a commercially available modelling software, coupled with an optimization algorithm, to minimize the weight-to-power ratio of a steam bottoming cycle. Only a moderate decrease in the weight-to-power ratio (3.5%) was achieved compared to a knowledge-based design, indicating that onshore plant design methodology may be too restrictive to achieve the radical weight savings needed for

Table 1
Selection of studies in the open literature considering steam bottoming cycle optimization.

Reference	Environment	Number of pressure levels	HRSG tube diameter [mm]	Exhaust pressure drop [mbar]
Franko and Giannini [6]	Onshore	2–3	38–63.5	60
Manassaldi et al. [7]	Onshore	2	42.16–60.3	37
Mehrgoo and Amidpour [8]	Onshore	2	Not reported	14–86
Mehrgoo and Amidpour [9]	Onshore	1–3	33.4–88.9	Not reported
Rezaie et al. [10]	Onshore	1	38.1	13
Pierobon et al. [11]	Offshore	1	22.3–40	Not reported
Nord et al. [12]	Offshore	1	Not reported	15–35
This work	Offshore	1	10–20	13

offshore cycles.

In order to keep the cost of construction (CAPEX) down on oil and gas platforms there is limited allowance for additional weight that a platform can hold. The limited weight allowance therefore prohibits the installation of new heavy equipment. Also, weight distribution is important on the platform, the weight of each piece of equipment is therefore accurately tracked. The objective of this study is to therefore to reduce the weight of steam cycles significantly so that they can be implemented offshore.

The ambitious goal of the current investigation has been to achieve a 50% weight reduction of an offshore steam bottoming cycle compared to an existing 16 MW design. Based on previous unpublished work in the projects EFFORT [13] and COMPACTS [14] steam cycle technology was selected over a CO_2 cycle due to the potential of achieving lower weight, high technology maturity and the ability to compare results with existing designs. The novelty in the current approach, compared to earlier work, is three-fold:

- Simultaneous optimization of overall power cycle parameters and component design (including geometry and structural framework), to obtain minimum net system weight for offshore conditions. Heuristics used for onshore equipment and specific supplier constraints has been relaxed or removed, while maintaining physical realizability, to allow for unconventional designs.
- Switching to lightweight materials such as aluminium and titanium wherever possible in relation to temperature limits, corrosion considerations and other factors defined in offshore construction standards.
- Increasing operational reliability by optimizing system design and improving condensate treatment routines

This work summarizes the critical component models that have enabled weight optimization and highlights the geometric differences between conventional and weight-optimal component designs. The paper is a new benchmark for offshore steam cycle weight optimization cases, and a starting point for component manufacturers seeking to offer products with reduced weight requirements. The article also gives recommendations for improvements to condensate treatment systems.

2. Methodology

This section describes the methodology used for designing two steam bottoming cycles for offshore applications by minimizing their weight. The exhaust gas from two LM2500 Gas turbines operating at 90% load is the heat source of these bottoming cycles. Selected characteristics of the exhaust gas is shown in Table 2. The steam bottoming cycles differ in thermodynamic performance as shown in Table 3.

A third case, based on an existing 16 MW offshore steam bottoming cycle, is defined to quantify the weight reduction achieved in this study. Moreover, the total weight of this reference bottoming cycle, is subdivided into two Heat Recovery Steam Generators (HRSGs), condenser, steam turbine, and water treatment sections, enabling quantifying the weight reduction on a component level.

Table 2
Selected characteristics of the exhaust gas that is the heat source of the steam bottoming cycles.

Temperature at HRSG inlet [°C]	489
Pressure at HRSG inlet [bar]	1.065
Mass flow rate (from each turbine) [kg/s]	81.63
Available exhaust heat[a] [MW]	66.2
Available exhaust exergy[a,b] [MW]	32.6

[a] Assuming the heat source is cooled down to 120 °C.
[b] Assuming ambient temperature of 10 °C.

The bottoming cycle weight minimization is split into two steps. First, the geometry of the two HRSGs and the state points of the bottoming cycle are determined by an optimization with the aim of minimizing the total weight of the two HRSG units subject to the predefined net power output in Table 3. The bottoming cycle state points are used as basis for the subsequent design of lightweight steam condenser, steam turbine and condensate treatment systems. The two steps are described in detail in the following two subsections.

2.1. Combined process and HRSG optimization

The problem formulation for the combined process and HRSG design optimization is shown in Table 4 (fixed parameters and constraints) and Table 5 (independent variables). The fixed process parameters govern the condensation pressure and turbomachinery efficiencies. The condensation pressure was set in dialogue with a steam turbine manufacturer. The condensing pressure for the 12 MW cycle is set relatively high (see Table 4) such that the final stage of a conventional steam turbine can be omitted. The higher temperature difference in the condenser also contributes to a substantial weight saving due to less heat transfer surface requirement. The 16 MW cycle uses a more conventional, lower condensation pressure. The fixed HRSG geometry parameters govern fin thickness and geometry characteristics of the HRSG inlet- and exit transition ducts.

The independent variables govern both bottoming cycle process parameters and HRSG geometry parameters. The bounds used to constraint the independent variables are indicated in Table 5. Most

Table 3
Definition of two steam bottoming cycles (cases) and their thermodynamic performance.

Net power output/Case ID	12 MW	16 MW
First law efficiency (Net power/Available heat)	18%	24%
Second law efficiency (Net power/Available exergy)	37%	49%

Table 4
Fixed parameters and constraints for the bottoming cycle optimization.

Fixed process parameters	
Condensation pressure/temperature (12 MW) [bar]/[°C]	0.2/60.1
Condensation pressure/temperature (16 MW) [bar]/[°C]	0.05/32.9
Steam turbine isentropic efficiency	0.85
Feedwater pump isentropic efficiency	0.7
Process constraints	
Exhaust gas pressure drop in the HRSG core [Pa]	<1300
Turbine outlet vapour quality [%]	>95
Net power output [MW]	12/16
Fixed HRSG geometry parameters	
Fin thickness [mm]	1.05
Inlet duct cross-section [m^2]	4
Inlet duct diffuser angle [°]	25
Exit duct cross-section [m^2]	9
Exit duct nozzle angle [°]	75
HRSG geometry constraint	
Diagonal tube pitch/bend diameter	>3 X tube OD
Transversal tube pitch	>2 X tube OD

Table 5
Free variables for cycle optimization problem.

	Bounds	
Process variables	lower	upper
Water/steam mass flow [kg/s]	12	27
Feedwater pump outlet pressure [bar]	16	50
HRSG geometry variables		
Tube inner diameter[a] [mm]	10	20
Fin height[a] [mm]	5	15
Fin pitch[a] [mm]	3	15
Diagonal fin tip clearance[a] [mm]	2	15
Transversal fin tip clearance[a] [mm]	2	15
Tube length [m]	2	8
Number of tubes per row [–]	20	200

[a] Three variables – (economizer, evaporator and superheater).

Table 6
Condenser operating conditions (from cycle optimization).

Case ID	16 MW	12 MW
Condenser duty [MW]	41.7	37.6
Tube side		
Pressure [bar]	2	2
Inlet temperature [°C]	10	10
Outlet temperature [°C]	22	28
Max. velocity [m/s]	2.5	2.5
Mass flow [kg/s]	800	500
Shell side		
Pressure [bar]	0.05	0.2
Inlet temperature [°C]	33	60
Outlet temperature [°C]	33	33
Mass flow [kg/s]	18.1	16.0

of these values were set such that the bounds did not exclude the optimal solution. The exceptions are the lower bounds of the variables for tube diameter, fin pitch, fin height and fin tip clearance whose values were set to ensure a manufacturable design, while still allowing radically different designs compared to state-of-the-art.

The combined process and HRSG design optimization starts by computing the objective function which is the total weight of the two HRSGs. The HRSG core geometry is solely defined by the independent variables and the fixed HRSG parameters. A HRSG framework model is included to also account for beams, core support plates and casing plates in the objective function. The framework model uses the fixed parameters for the inlet- and exit transition ducts, see Table 4, and the size of the HRSG core to estimate the total weight of the framework covering the HRSG core. Further detail on the HRSG framework model is documented in Ref. [15].

Thereafter the state points of the bottoming cycle are determined. First the feedwater pump inlet state is computed as saturated liquid at the predefined condensation pressure. Thereafter the HRSG water inlet state is computed using the predefined pump efficiency and the independent variable for the pump outlet pressure. After that both inlet states of the HRSG is defined, and the outlet states are calculated by use of a thermal-hydraulic HRSG model. This model applies building blocks such as fluid nodes, thermal-hydraulic correlations, and solution algorithms from an inhouse heat exchanger library [16].

In addition, the variation in properties of the two fluids and the cross-counter current flow orientation are accounted for by discretizing the HRSG core into multiple sub-heat exchange models illustrated in Fig. 2. Modified ESCOA correlations [17] were used for predicting the heat transfer coefficient and pressure drop used on the exhaust side of all sub heat exchanger models. The modification involves multiplying the heat transfer coefficient by a correction factor of 0.826 and removes the observed bias of the ESCOA heat transfer correlation when compared to published experimental work. The correction factor was found through comparison with a comprehensive experimental database (479 data points for solid fin heat transfer) Holfeld [18]. The heat transfer coefficients on the water/steam side were predicted by the Gnielinski correlation [19] for single phase flow and by the method of Bennet and Chen [20] for evaporation.

The turbine inlet state is defined once the thermal-hydraulic HRSG model is finished. The turbine outlet state is computed using the predefined condensation pressure and turbine efficiency.

The design strategy takes advantage of constraints to ensure a feasible and consistent process design and a manufacturable HRSG design, Table 4. An equality constraint is imposed to ensure that the computed net power output (the difference between the generator power production and the pump motor power consumption) equals the predefined value of 12 MW and 16 MW. An inequality constraint is imposed to limit the amount of water droplets at the turbine outlet. The exhaust side pressure drop is governed by the allowable backpressure of the gas turbines. In this study a conservative limit was set at 3 kPa, of which 1.3 kPa is attributed to the heat exchanger core and the remaining pressure drop is distributed across ducts and silencer in the exhaust channel. Finally, two inequality constraints are imposed, based on personal communication with a HRSG manufacturer, to ensure manufacturable tube pitches.

The optimizations were carried out using NLPQL [21] which is a sequential quadratic programming method (SQP) for solving constrained optimization problems.

2.2. Condenser, turbine, water purification

The condenser and steam turbine are usually assembled as a single unit to minimize air infiltration. The size of the assembly is heavily influenced by the condenser and the low-pressure stages of the steam turbine, due to the large volume flows obtained at low steam pressure. This work has therefore focused on reducing the weight and volume of the condenser, thereby also minimizing the size of the support structure of the steam turbine skid. The 12 MW and 16 MW steam turbine/generator sets are designed by SIEMENS according to the condensation pressure specification given in section 2.1 (giving a much-reduced size of the 12 MW turbine). The

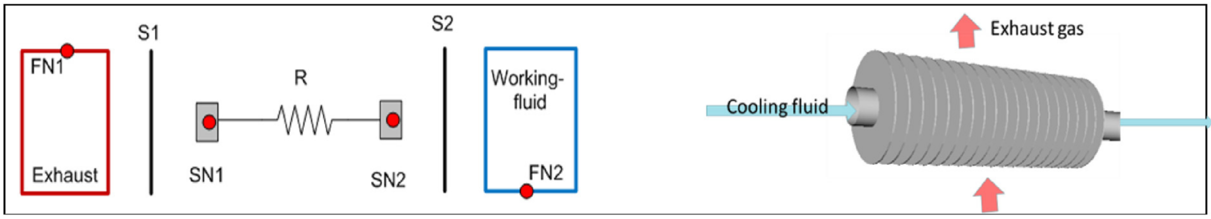

Fig. 2. The sub-heat exchanger model: Basic heat transfer elements (left) and geometric illustration (right).

condenser is optimized using the same in-house tool as for the HRSG [16].

The condenser is modelled as a standard shell and tube heat exchanger in U-tube configuration with two passes of cooling seawater and one pass of condensing steam. The number of parallel tubes and the tube length are optimized to obtain minimal total weight. The thermal design is described in uses correlations by Gnielinski [19] and Wolverine Tube inc [22] for the tube inside and outside heat transfer coefficient, respectively. Seawater resistant titanium is used for the tubes, while the remaining component estimations are based on stainless steel. The following component weights are estimated based on volume and density: Seawater tubes, shell with end-walls, tube sheets, bundle support plates. A few components use a fixed weight (nozzle/flanges, air cooling unit) or are scaled by the condenser footprint (condenser support, beams/pipes/other structural).

The feedwater purification system is a complex process consisting of two ion exchange beds (alternating between active and regeneration modes), a deaerator and several feedwater pumps. A relatively large freshwater holding tank is also required to provide steam cycle makeup water.

Electro-deionization (EDI) [23] was considered as a promising replacement for ion exchange resin beds to further decrease system weight. The required operational conditions of this technology (most notably for temperature) will be explored in further work and integrated in the weight optimization framework.

2.3. Uses of aluminium

A lightweight turbine- and generator module (skid) is developed for the 12 MW bottoming cycle in collaboration with Marine Aluminium [24]. Aluminium structures are installed offshore for several applications (e.g. helidecks and gangways) and an aluminium turbine skid represents a likely first use application with respect to offshore bottoming cycles. The module is built as a stress skin module with load bearing walls and decks.

The module is designed according to Eurocode 9 [25] and analysed for structural integrity in the STAAD.Pro (Structural Analysis and Design) software. Load cases for transport (1g vertical acceleration ± 0.3g acceleration in any direction) and single hook lifting according to NORSOK R-002 [26] are analysed. The structural model and the main skid content are shown in Fig. 3, with one side panel removed for visibility. Results from the structural analysis for the most critical load case (single hook lifting) indicated a maximum von Mises stress well below critical limits. This analysis verifies the structural integrity of the design.

3. Results & discussion

An overview of the results for each optimized sub-system, including the overall weight saving, is shown in Fig. 4. The total weight reduction is substantial when compared to the reference system at 52% for the 12 MW system and 38% for the 16 MW system. The HRSG module (blue colours) is the heaviest module in the optimized 16 MW system (51% of the weight), followed by the turbine/condenser skid (grey colours, 30% of the weight). The largest weight reduction (up to 70% reduction in the 12 MW design) is obtained for the HRSG core and ductwork, followed by the condenser. Results for each module are given in the following sections.

The power output per unit of system weight is 6% higher for the 16 MW system compared to the 12 MW system, indicating some "economies of scale" of selecting a larger power output. The difference, however, is not dramatic in the range considered.

3.1. HRSG and cycle optimization results

The optimized HRSG and cycle parameters are given in Table 7. A unique feature of the optimized design is the low tube inner diameter, which reaches its lower bound (10 mm) for both cycle capacities. This indicates that the optimal tube diameter is limited by manufacturing constraints rather than the steam-side thermal-hydraulics. A separate optimization of the 12 MW case without tube diameter boundary was run to confirm this hypothesis, which led to a tube diameter of 6.4 mm. This result is not surprising; Large tubes take up space and the main heat transfer resistance is on the gas side, leading to a need for the largest possible heat transfer area per unit cross section. Meanwhile, a large steam pressure drop carries little efficiency penalty. Identical results were found in previous work with CO_2 as the working medium [15]. However, the steam bottoming cycle optimization studies from the open literature reports much larger HRSG tubes, see Table 1. Whether the proposed HRSG designs in the open-literature is weight-optimal could therefore be questioned. Note that the steam side pressure drop is only indirectly constrained in this work (by the net power output constraint).

Exhaust side pressure drop is kept below the accepted maximum by compensating small tube diameters by optimizing remaining geometry parameters.

Remaining parameters in Table 7 are within the pre-defined bounds. Though it should be noted that the fin pitch is relatively large and the fin height is moderate compared to the tube diameter and the available space. This can be explained by the relatively constant minimum cross section mandated by the pressure drop constraints,[1] and the significant weight of the fins, Table 7 and Fig. 6.

The casings of the reference and optimized HRSG's, Fig. 5 give an illustration of the weight reduction of the optimized HRSG's. The weight reduction is primarily due to a smaller heat exchanger core and a squarer footprint, resulting in shorter transition ducts.

The detailed weight distribution of the HRSG core is compared to the reference HRSG module in Fig. 6. The weight of the inlet and

[1] The pressured drop scales linearly with the Euler number, but quadratically with the flow cross section through the average flow velocity.

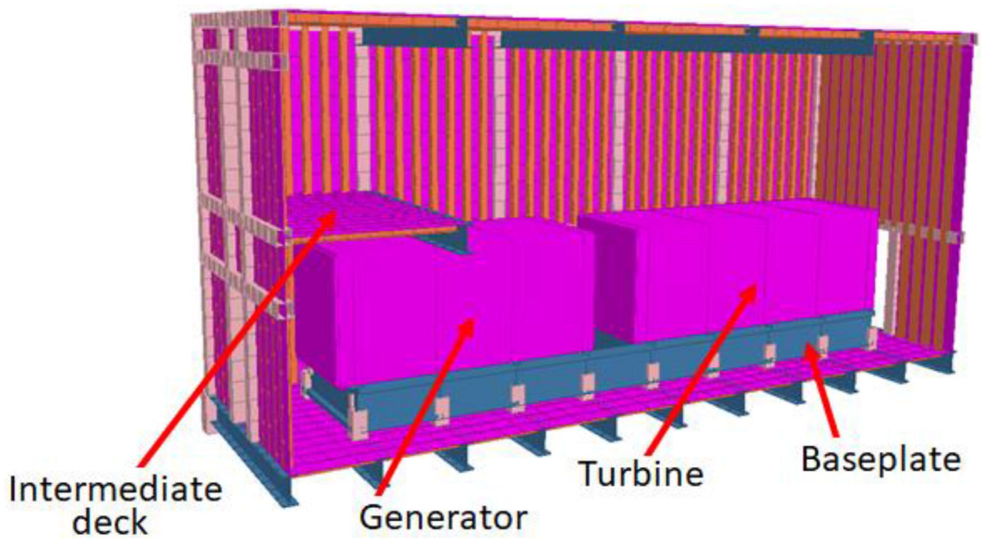

Fig. 3. Structural model of steam turbine- and generator skid design (with side panel removed). Drawing by Marine Aluminium.

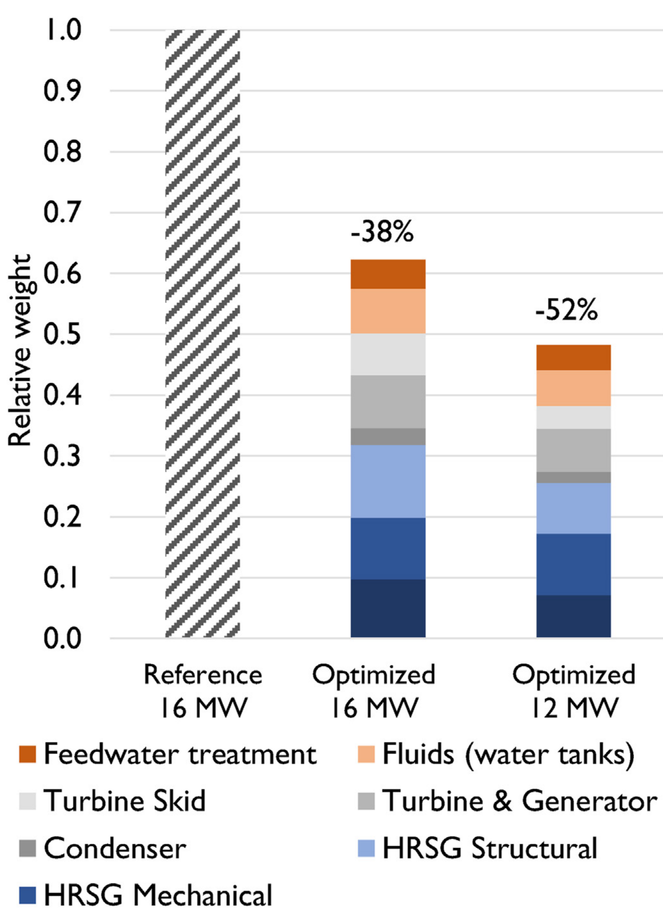

Fig. 4. Bottoming cycle weight distribution & weight savings compared to reference cycle (Weight reduction in %).

exit ducts are significant. A weight reduction of 57% is achieved for the HRSG for the 16 MW system.

Table 7
Optimized bottoming cycle process and HRSG geometry for 12 MW and 16 MW.

		16 MW	12 MW
Optimization variables			
Pump outlet pressure	[bar]	18.3	26.4
Working fluid mass flow	[kg/s]	18.1	16.0
Tube inner diameter	[m]	10.0	10.0
Core width	[m]	3.8	3.6
Number of tubes per row	[–]	117	75
Economizer geometry			
Number of tube rows	[–]	10	9
Transversal fin tip clearance	[mm]	8.1	20.9
Diagonal fin tip clearance	[mm]	8.2	8.3
Fin pitch	[mm]	13.5	9.9
Fin height	[mm]	8.3	8.4
Evaporator geometry			
Number of tube rows	[–]	21	18
Transversal fin tip clearance	[mm]	5.5	20.0
Diagonal fin tip clearance	[mm]	5.6	7.4
Fin pitch	[mm]	8.7	6.4
Fin height	[mm]	9.5	8.9
Superheater geometry			
Number of tube rows	[–]	8	6
Transversal fin tip clearance	[mm]	5.5	20.2
Diagonal fin tip clearance	[mm]	5.6	7.6
Fin pitch	[mm]	9.3	7.4
Fin height	[mm]	9.6	8.8
Objective function			
(2 X HRSG weight)	[ton]	84.9	61.9
Constraints			
Net power production	[MW]	16.0	12.0
Expander outlet quality		95%	95%
Exhaust pressure drop	[Pa]	1300	1300

3.2. Condenser

The geometrical details of the optimized steam condenser are given in Table 8. The detailed weight distribution and comparison with the reference condenser is shown in Fig. 7. The weight reduction for the condenser for the 16 MW and 12 MW cases is 52% and 27% respectively.

The weight of the condenser for the 12 MW case is much smaller than for the 16 MW case, primarily due to the higher condensing pressure and the resulting larger temperature difference to the

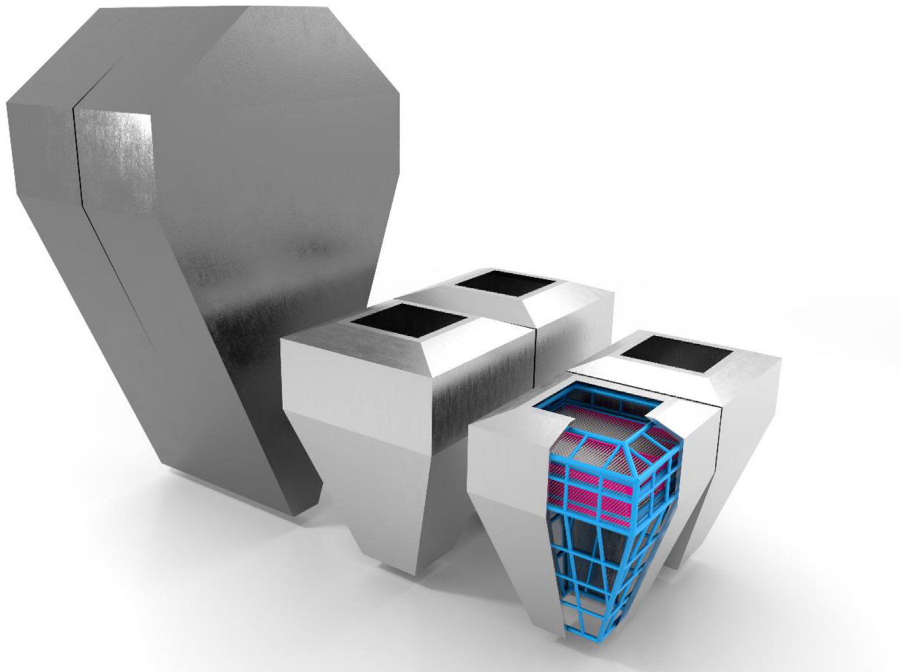

Fig. 5. Casing around the reference HRSG and optimized HRSG's for the 16 and 12 MW cases. Cutaway indicates duct, insulation and beam model included in the HRSG core calculations.

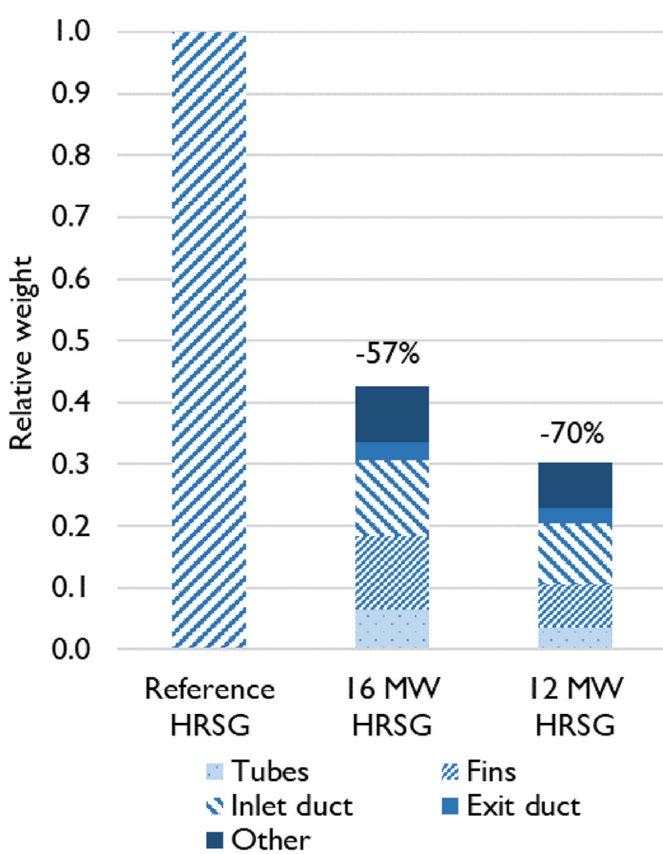

Fig. 6. HRSG core weight distribution & weight saving compared to reference. "Other" include headers and all structural components (beams, support plates etc.) around the heat exchanger core. (weight saving in %).

cooling seawater. Some parts of the condenser, however, do not scale with duty and/or temperature difference, most notably nozzles and structural components.

Table 8
Optimized condenser parameters.

Case ID	16 MW	12 MW
Tube length [m]	5.5	4.2
Shell diameter [m]	1.6	1.3
Number of tubes [−]	2520	2574
Total length [m]	7.6	6.3
Footprint [m^2]	12.4	8.0
Shell + tube weight [kg]	10 400	5300
Total weight [kg]	24 200	15 600

3.3. Skid

The use of structural aluminium reduces the empty skid weight by 27%, or 7metric tonnes, for the 12 MW system compared to the standard 16 MW solution. Part of the reason why the skid weight is reduced is that it is shorter, as the condenser is now mounted on the outside of the skid.

3.4. Condensate purification and operational reliability

Weight reduction in the feedwater treatment system was achieved by switching materials from steel to glass fibre reinforced plastic (GFRP) in all process tanks where temperature limits permit.

It was found that condensate purity may be a factor in operational reliability of such units. Weight reduction was achieved by switching the construction material of the condensate tanks from stainless steel to glass-fiber reinforced plastic giving a 60% weight reduction of the tank. Work has started to investigate Electro deionization (EDI) as a replacement for the ion exchange purifier. EDI has a much lower weight; hence a potential weight reduction of the treatment system could be achieved. The main potential

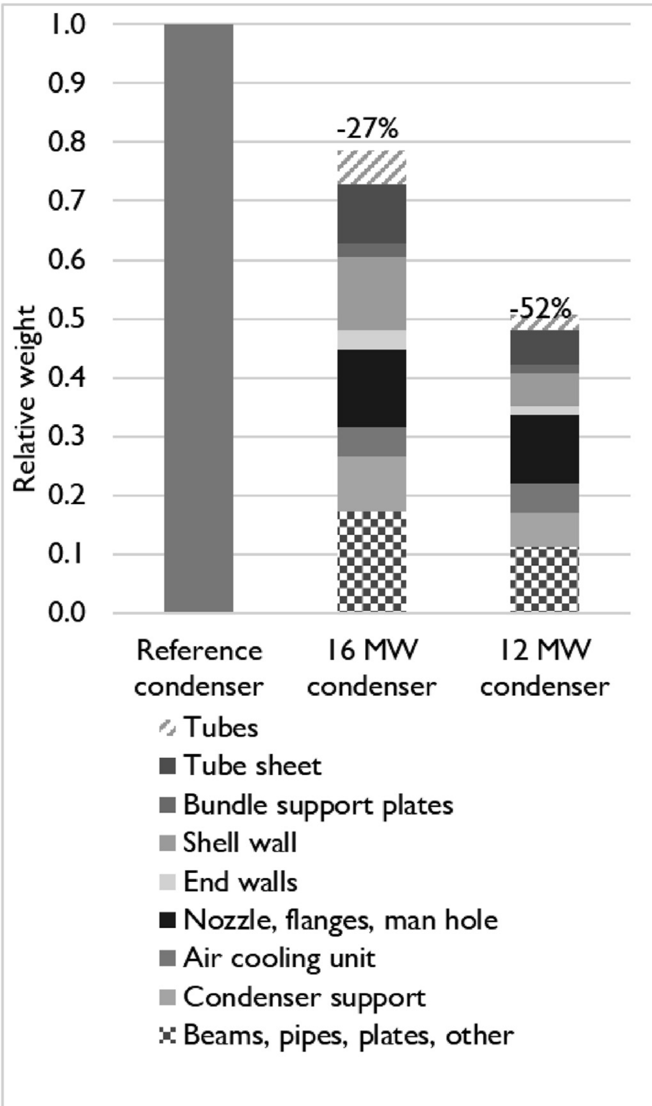

Fig. 7. Condenser weight distribution & weight saving compared to reference (weight saving in %).

benefits however are the lower operational costs and higher operational reliability of EDI technology which are key factors when considering widespread implementation.

4. Conclusions

The present study demonstrates the feasibility of drastic weight reduction of offshore bottoming cycles. Mathematical models for equipment weight are used to optimize HRSG (heat recovery steam generator) geometry and steam cycle parameters with respect to weight while maintaining net power output. The weights of the condenser, steam turbine skid and feedwater treatment system are then further reduced by geometrical changes and materials switching. The following specific conclusions can be drawn:

- Unconventional equipment geometries must be considered to reach the significant weight reductions needed in offshore applications. This requires careful re-evaluation of existing standards and best practices.
- A small tube diameter is an important component of the optimized HRSG designs, as this allows for a small cross section of the HX core which also lowers framework & duct weight.
- The use of aluminium enables significant reduction of framework weight. Further weight reductions are possible by materials switching (e.g. in ductwork) but this requires validation of the structural integrity and risk of exposure to high temperatures.
- The condenser weight can be reduced by having longer, but fewer tubes than the reference condenser. This results in a larger heat transfer coefficient on the sea water side and thus a reduced need for heat transfer area. In addition, the tube sheet weight was reduced due to the smaller shell diameter.
- Materials switching from steel to glass fiber reinforced plastic allows reduction in weight of condensate and resin holding tanks.
- The steam cycle power output has a significant implication for the system minimum weight, but the power output per kilogram of equipment is relatively constant when detailed optimization is performed, at least for the power ranges considered in this work.

Combined cycle has long been the standard for land-based power plants. By reducing their weight, combined cycle power plants can be easier to implement and therefore become the standard for offshore power production as well, rather than the exception that they are today. This can lead to reduction in CO_2 emissions from offshore power plants by up to 25%.

Credit author statement

Marit J. Mazzetti: Conceptualization, Writing-Original Draft, Review & Editing, Investigation, Brede AL Hagen: Conceptualization, Methodology, Writing-Original Draft, Review & Editing, Software, Investigation, Geir Skaugen: Conceptualization, Methodology, Writing-Review & Editing, Software, Investigation, Karl Lindqvist: Writing-Original Draft, Steinar Lundberg: Investigation, Formal Analysis, Oddrun A. Kristensen: Investigation, Formal Analysis.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgement

The authors acknowledge the partners: Neptune Energy Norge AS, Alfa Laval, Equinor, Marine Aluminium, NTNU, SINTEF and the Research Council of Norway, strategic Norwegian research program PETROMAKS2 (#233947) and (#269349/E30) for their support.

References

[1] GE. LM2500 Aeroderivative gas turbine product specifications 2021.
[2] Kloster P. Energy optimization on offshore installations with emphasis on offshore combined cycle plants. Offshore Europe Oil and Gas Conference and Exhibition, Aberdeen, United Kingdom September, 1999.
[3] Mazzetti MJ, Nekså P, Walnum HT, Hemmingsen AKT. Energy-efficiency technologies for reduction of offshore CO_2 emissions. Oil Gas Facil 2014;3: 89–96. https://doi.org/10.2118/169811-PA.
[4] Nord L, Bolland O. Steam bottoming cycles offshore - challenges and possibilities. J Power Technol 2012;92:201–7.
[5] Norsk Petroleum. Emissions to air 2020.
[6] Franco A, Giannini N. A general method for the optimum design of heat recovery steam generators. Energy 2006;31:3342–61. https://doi.org/10.1016/j.energy.2006.03.005.

[7] Manassaldi JI, Mussati SF, Scenna NJ. Optimal synthesis and design of heat recovery steam generation (HRSG) via mathematical programming. Energy 2011;36:475–85. https://doi.org/10.1016/j.energy.2010.10.017.
[8] Mehrgoo M, Amidpour M. Constructal design and optimization of a dual pressure heat recovery steam generator. Energy 2017;124:87–99. https://doi.org/10.1016/j.energy.2017.02.046.
[9] Mehrgoo M, Amidpour M. Configurations and pressure levels optimization of heat recovery steam generator using the genetic algorithm method based on the constructal design. Appl Therm Eng 2017;122:601–17. https://doi.org/10.1016/j.applthermaleng.2017.04.144.
[10] Rezaie A, Tsatsaronis G, Hellwig U. Thermal design and optimization of a heat recovery steam generator in a combined-cycle power plant by applying a genetic algorithm. Energy 2019;168:346–57. https://doi.org/10.1016/j.energy.2018.11.047.
[11] Pierobon L, Benato A, Scolari E, Haglind F, Stoppato A. Waste heat recovery technologies for offshore platforms. Appl Energy 2014;136:228–41. https://doi.org/10.1016/J.APENERGY.2014.08.109.
[12] Nord LO, Martelli E, Bolland O. Weight and power optimization of steam bottoming cycle for offshore oil and gas installations. Energy 2014;76:891–8. https://doi.org/10.1016/J.ENERGY.2014.08.090.
[13] EFFORT. Offshore energy effiicency technologies. 2014.
[14] COMPACTS. Compact offshore steam cycles. 2018.
[15] Skaugen G, Walnum HTT, Hagen BAL, Clos DP, Mazzetti MJ, Nekså P. Design and optimization of waste heat recovery unit using carbon dioxide as cooling fluid. Am. Soc. Mech. Eng. Power Div. POWER 2014:1–10. https://doi.org/10.1115/POWER2014-32165.
[16] Skaugen G, Kolsaker K, Walnum HT, Wilhelmsen Ø. A flexible and robust modelling framework for multi-stream heat exchangers. Comput Chem Eng 2013;49:95–104.
[17] Ganapathy V. Industrial boilers and heat recovery steam generators. Industrial boilers and heat recovery steam generators. 2003.
[18] Holfeld A, Næss E. Influence of the fin type and base tube diameter of serrated and solid-fin tubes on the heat transfer and pressure drop performance. Proc 15th Int Heat Transf Conf IHTC- 2014;15:1–12.
[19] Gnielinski V. New equations for heat and mass transfer in turbulent pipe and channel flow. Int Chem Eng 1976;16:359–68.
[20] Bennett DL, Chen JC. Forced convective boiling in vertical tubes for saturated pure components and binary mixtures. AIChE J 1980;26:454–61.
[21] Schittkowski K. NLPQL: a fortran subroutine solving constrained nonlinear programming problems. Ann Oper Res 1986;5:485–500. https://doi.org/10.1007/BF02739235.
[22] Thome JR. Wolvrine engineering data book III. Wolver Tube Inc; 2009.
[23] Prato T, Gallagher C. Using EDI to meet the needs of pure water production. 2010. https://WwwGeCom/in/Sites/WwwGeCom.in/Files/GE-EDI-Pure-Water-ProductionPdf.
[24] https://m-a.no/, Marine Aluminium. 2019.
[25] EUROCODE. European Standards.
[26] NORSOK R002. Norwegian Shelf Standards.
[27] Nord, Olof Lars, Bolland O. Design and off-design simulations of combined cycles for offshore oil and gas installations. Appl Therm Eng 2013;54(1).