DTU Library

Design and optimization of a power hub for Brazilian off-shore oil production units

Vidoza, Jorge ; Andreasen, Jesper Graa; Haglind, Fredrik; Reis, Max ; Gallo, Waldyr

Published in:
Proceedings of ECOS 2018: 31st International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems

Publication date:
2018

Document Version
Peer reviewed version

Design and optimization of a power hub for Brazilian off-shore oil production units

Jorge Vidoza[a], Jesper Graa Andreasen[b], Fredrik Haglind[c], Max Reis[d], Waldyr Gallo[e]

[a]*Mechanical Engineering Institute, University of Campinas, Campinas, Brazil, jorgeavg@fem.unicamp.br*
[b]*Department of Mechanical Engineering, Technical University of Denmark, Kongens Lyngby, Denmark, jgan@mek.dtu.dk*
[c]*Department of Mechanical Engineering, Technical University of Denmark, Kongens Lyngby, Denmark, frh@mek.dtu.dk*
[d]*Mechanical Engineering Institute, University of Campinas, Campinas, Brazil, maxmauro@fem.unicamp.br*
[e]*Mechanical Engineering Institute, University of Campinas, Campinas, Brazil, gallo@fem.unicamp.br*

Abstract:

A worldwide trend to reduce greenhouse gases emissions has encouraged researchers to study more efficient solutions in diverse sectors, including Oil and Gas Industry. Most of offshore units are energized by redundant equipment operating at low loads, turning their energy consumption inefficient and increasing environmental impact. This work aims at identifying the optimal design of a gas and steam turbine combined cycle tailored for offshore oil production applications. The Brazilian pre-salt basin is taken as a case study to improve operational efficiency and reduce CO_2 emissions of floating oil production units. The idea is to concentrate the power supply to a floating power plant, composed of combined cycle power blocks. A model is developed, integrating the design of the gas turbine, heat recovery steam generators (single pressure and double pressure), steam turbine and condenser. Genetic algorithms are applied in two optimization approaches, single-objective and multi-objective. Three parameters are evaluated: equipment purchase cost, thermal efficiency and total weight. The results of the multi-objective optimization indicate that dual-pressure arrangement steam cycles, featuring 3 gas turbines, 1 HRSG and 1 steam cycle, could be an attractive design solution for power hubs. This arrangement has a low cost and weight, while the thermal efficiency is maintained at a reasonable high level (around 53.2 %). Moreover, the results indicate that by introducing a power hub, the CO2 emissions may be reduced by 18.7 % to 27.2 % compared with a conventional FPSO design.

Keywords:

Offshore Power Systems, Floating Power Plants, Multi-objective Optimization, Offshore Grids.

1. Introduction

In the past decades, environmental awareness supported changes in diverse economic sectors. Oil and gas activities in some regions are making efforts to adapt to these trends. With the depletion of traditional oil fields, offshore oil activities increased, particularly in deep-water oil fields. This is the case of Brazilian oil industry with oil activities focusing on the exploitation of the Pre-salt Basin. Several studies analyzed opportunities to mitigate environmental impacts in the Brazilian case, by introducing either CO_2 emissions forecasts [1,2] or proposing alternatives such as carbon capture [3] and the use of bottoming Organic Rankine Cycle (ORC) power systems [4].

Alternatives for the Norwegian case have been widely studied. Pierobon et al [5] performed several analyses for Rankine, Organic and Compressed Air bottoming cycles. Nguyen et al [6] realized exergetic analyses also for bottoming cycles. These studies, along with the work presented by Nord et al [7] compose an important state of art for the analysis of combined cycles (CC) in offshore facilities. Other approach of optimization was performed by Orlandini et al [8] with the introduction of wind power links through offshore grids. In the context of offshore grids, Hetland et al. [9] introduced the concept of a floating power hub, which would centralize power supply for offshore oil production units, by means of a dedicated floating power plant. Each oil production unit usually contains an over dimensioned power module for electricity supply to be used for the whole lifetime of the production unit. In the floating power hub, individual power modules would be removed, and a centralized power plant would supply energy through submersible cables. Submarine pipelines would supply the fuel required by the power plant, by using the associated gas from the oil production.

This study aims to analyze the performance of a floating power hub for the Brazilian Pre-salt basin, in order to reduce CO_2 emissions and improve the plant thermal efficiency. In the study by Hetland et al [9] the power hub concept is introduced along with pre-selected Combined Cycle components. In this case the component design is tailored to the Brazilian Pre-salt characteristics. Previous studies in the field of optimizing Combined Cycle components focused on the Waste Heat Recovery Unit (WHRU) design, either based on Once Through Boiler (OTB) designs or drum-type Heat Recovery Steam Generators (HRSGs). Manassaldi et al. [10] applied optimization procedures for onshore combined cycles, while Rovira et al. [11] considered off-design parameters for similar optimization objectives.

A literature review indicates that most studies regarding offshore power generation in oil platforms relate to already operating or built facilities, restricting the options to improve the plant performance. There are only a few studies concerning offshore power generation hubs based on associated gas extraction, and these are limited to the use of fixed commercial equipment sets [9,12]. Moreover, in previous works, gas turbines in CC optimizations for off-shore applications were generally considered as “black boxes”. The present study integrates the gas turbine design in the optimization procedure, in order to increase the number of CC layout options, potentially leading to more cost-effective and efficient combined cycle designs, which was not performed in references [5,7].

This work aims at identifying the optimal design of a gas and steam turbine combined cycle tailored for an off-shore power hub. Specifically, a design tailored to satisfy the Brazilian pre-salt basin specific needs is proposed. For this purpose, single-objective and multi-objective optimization studies are carried out, incorporating the performance, weight and cost of the combined cycle. The methodology presented could be used also for other off-shore applications for which the trade-offs among performance, weight and cost parameters need to be addressed.

2. Floating Production Storage and Offloading (FPSO) installations

Pre-salt "replicant FPSOs" are equipped with an energy module, consisting in four GE LM2500 gas turbines. Under the specific temperature, pressure and humidity conditions of the Brazilian pre-salt basin, the energy module total capacity reaches approximately 75 MW. Production forecasts indicate the possibility of obtaining an associated gas with a high percentage of CO_2, along with the oil and water mixture. Two additional GE LM2000 turbines provide mechanical drive for a CO_2 compression section. A more specific overview of pre-salt FPSOs operation scenarios can be found in Gallo et al.[1]. Combining both demands from the main power module and the CO_2 compression module, total electric demand in a replicant pre-salt FPSO may reach up to 80 MW. The proposed floating power hub would cover the electricity demand for three FPSOs, aiming to concentrate supply in one power plant. Fuel gas being produced and treated in the vessels is sent to the power hub, and in turn, it sends electricity via submersible cables to each vessel. FPSOs needs a heat source to perform treatment and separation processes, therefore, at least one cogeneration turbine must be left locally in each FPSO in order to fulfill such requirements. The remaining three turbines could be hypothetically removed and the turbines used to energize CO_2 compressors could be replaced by electrical drives. The total demand of three FPSOs would sum up to 240 MW and considering that at least 25 MW must be left in each vessel (and adding a reserve margin), it is estimated that the minimum installed capacity to be supplied by the power hub is 165 MW.

3. Methodology

3.1. Optimization method

Optimization procedures were performed using the Genetic Algorithm approach. Genetic algorithm (GA) optimization was selected, because their versatility and capacity to deal with non-smooth objective functions. Two approaches were considered in this study, single-objective and multi-objective optimization. In onshore power plants, usually space and weight requirements are omitted. However, in an offshore facility, space and weight may influence the overall capital costs. Furthermore, as the main objective of a floating power hub is to improve efficiency in FPSOs energy supply and thus, reduce CO_2 emissions, the design procedure becomes a trade-off among costs, size and efficiency.

In order to assess such trade-offs, three objectives were established, see Eq. (1): 1 - Minimizing costs, which will be evaluated as the total purchasing equipment costs for the power plant: the sum of the individual costs of gas turbines, steam turbines, HRSGs, pumps and condensers (PEC). 2 - Minimizing the total mass, resulting from the sum of the individual weights of the power plant aforementioned equipment (M). 3 - Maximizing thermal efficiency, having as consequence reduction of fuel consumption and CO2 emissions (η_{CC}).

$$[\mathrm{PEC}, \mathrm{M}, \eta_{CC}] = f(\overline{X}) \qquad (1)$$

The three objectives were assessed separately (single-objective optimization) and as a three-dimensional objective for a trade-off analysis (multi-objective optimization). As seen in Equation 1, a function containing thermodynamic, dimensional and economic modeling for the combined cycle is applied to evaluate the established objectives. $\overline{X}$ is a vector of decision variables, each input of this vector corresponds to design parameters of each considered component.

A series of combined cycle blocks compose the floating power hub. A combined cycle block can have diverse arrangements, one to four gas turbines are combined with a single Rankine cycle. In the same way, one waste heat recovery unit can be shared by two or four gas turbines, in order to reduce space and weight requirements. The proposed model accounts with integer variables to find an optimum layout. Depending on the optimization approach, integer and binary variables represent the size (GTI) and quantity (NTR) of gas turbines, and whether if each is coupled to a HRSG or one HRSG is shared by the GT group (HXI), as displayed in Fig. 1. The introduction of such variables to the model makes it necessary to establish a Mixed Integer Non-Linear Programming (MINLP), in order to select among these discrete values. This implementation is particularly useful when modeling the options for the gas turbine technologies.

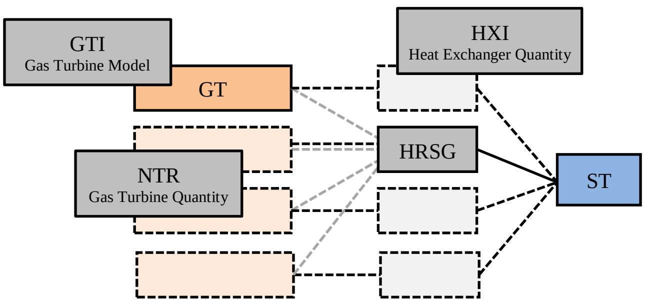


Fig. 1. Integer variables which define the combined cycle layout

The model starts establishing design parameters for the GT, which derive in a specific set of characteristics of the exhaust gases. Such characteristics are used to apply a Pinch Analysis, in order to determine heats and duties in each HRSG section. When the mass flows and enthalpies are determined, the steam turbine and condenser thermodynamic parameters are calculated. In this stage, the thermodynamic equilibrium in each component is verified, to avoid temperature crosses and fluid phase inconsistencies. Solutions not meeting thermodynamic equilibrium in all nodes are discarded.

If the candidate solution is consistent, then the physical properties of the components can be evaluated. The specific design for the HRSG is carried out and pressure drops are determined. If total pressure drops on the gas side exceed 0.1 bar the solution is also discarded. Physical properties such as weight and size are also calculated for the gas turbine and condenser. The calculated transfer areas, duties, thermodynamic and physical properties are main inputs for the cost estimation of the combined cycle equipment. Solutions are evaluated in an iterative process until reach a minimum or maximum value.

3.2. Gas turbines

3.2.1. Discrete analysis – commercial gas turbines

In order to span a considerable range of possibilities, four gas turbines were selected: (i) GE LM2500, (ii) SGT-700, (iii) GE LM6000, and (iv) SGT-800. GE LM2500 and SGT-700 are commonly used gas turbines for offshore applications like FPSO units and Floating Liquefied Natural Gas (FLNG) ships. Particularly LM2500 has been widely studied and it is installed in two of the currently operating offshore combined cycles in Norwegian platforms, Snorre-B and Oseberg. From a scale economy and efficiency point of view, larger turbines closer to what is actually used in on-shore applications are also considered: GE LM6000 is used for larger scale combined cycles and for cruise and ship propulsion. SGT-800, which has the largest power rating among the studied turbines, is mostly used in simple/combined cycle and cogeneration applications. The algorithm selects one of the four gas turbines to compose the combined cycle. Parameters such as temperatures and pressure ratios are inherent of each gas turbine. Gas turbine performances are obtained using the commercial software Thermoflex ® [13].

3.2.2. Continuous analysis – gas turbine parameter design

For the multi-objective optimization approach, the gas turbine parameters are modeled from the governing thermodynamic equations. The gas turbine is divided into four components: compressor, combustor, compressor turbine and power turbine, as seen in the simplified schemes in Fig. 2 and Fig. 3. In this case, the parameters defining the gas turbine design are the following: pressure ratios, turbine inlet temperature and compressor and turbine isentropic efficiencies. This allows the optimization search among continuous and smooth variables. Toffolo and Lazzaretto [14] applied multi-objective optimization for a single-shaft gas turbine. For this study, a similar optimization structure was proposed, considering a double shaft gas turbine. For this arrangement, there must be compatibility between the work delivered by the compressor turbine and the compressor power requirements. Additionally, there must be flow compatibility all along the expansion processes. Thermodynamic considerations and main design equations implemented follow Saravanamuttoo et al. [15].

In order to estimate the gas turbine weight a correlation was used, based on the methodology applied in Ref. [16]. A set of 9 commercial gas turbines and their weights were considered to obtain the correlation which depends on the turbine capacity in MW. Even though correlation does not reflect exact weight because of the particularities of each commercial design, it reflects an acceptable estimation for the overall cc mass estimation (R^2 = 0,77).

3.3. Steam Cycle

3.3.1. Single pressure heat recovery steam generator

In case of one pressure level the HRSG is divided in three main sections: economizer, evaporator and super-heater. The gas turbine arrangement established by the integer variables determines the exhaust gases mass flow rate into each HRSG. In Equation 2, NTR can take values from 1 to 4. The heat exchanger index is a binary variable used to determine the quantity of heat exchangers. If this index equals zero, HXI = 1, all gas turbines share a single HRGS. If the index is equal to one, HXI = NTR, each gas turbine is coupled to a separate HRSG. The mass flow rate of water is found by using the pinch point temperature difference:

$$\mathrm{m_{gas,total}} = \mathrm{m_{gas}} \mathrm{NTR/HXI} \qquad (2)$$

Fig. 2 shows a simplified system layout, with main flows and components. The steam cycle performance corresponding to one pressure level is determined by the temperature of steam exiting the HRSG, known as the live steam temperature, condenser pressure and steam turbine isentropic efficiency. Ambient conditions also affect the steam cycle, as the condensation pressure is limited by the ambient temperature. The condenser is assumed to be of shell-tube type, in which the cooling fluid would be treated seawater, and the operational pressure would be a decision variable set by the optimization procedure.

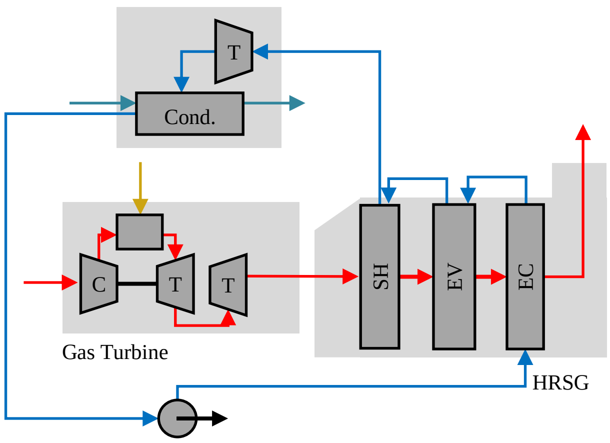


Fig. 2. Simplified one pressure level layout

3.3.2. Dual pressure heat recovery steam generator

The dual pressure arrangement is based on a typical configuration studied by Manassaldi [10]. In this case there are two economizers, super-heaters and evaporators, thus doubling the number of sections of the previous arrangement, and therefore adding more complexity to the system. The order and disposition of these sections is as seen in Fig.3. At the entrance, the low-pressure economizer handles one stream of water flow, which then divides at its outlet. A part of this main stream is sent to a pump to follow the high-pressure sequence to the evaporator and super-heater, the remaining mass flow is directed to the low-pressure sections.

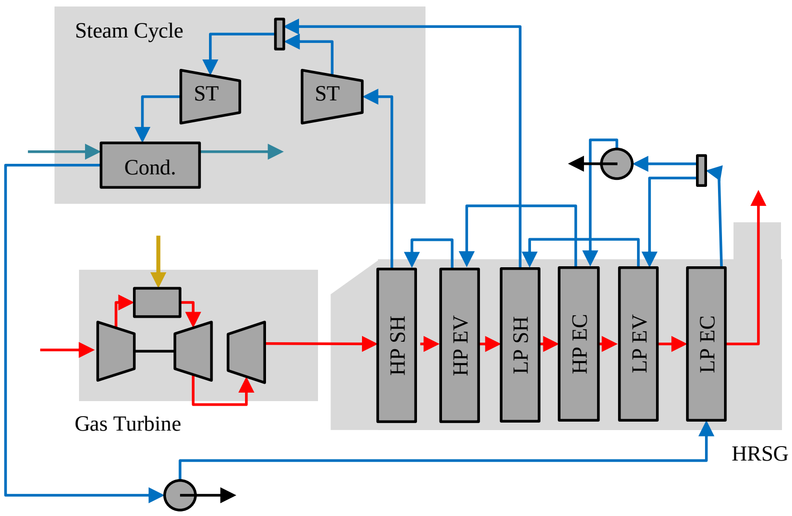


Fig. 3 Two pressure HRSG simplified layout

This system has a low-pressure pump, handling the complete stream of water flow, and a high-pressure pump which increases the pressure of the high-pressure water mass flow. The calculations for estimating mass flows are similar to the single-pressure arrangement. This case considers two different streams, a low-pressure and a high-pressure stream. The mass flow rates are calculated based on assumed pinch point values in the low-pressure and high-pressure sections, and the energy balances of each boiler section.

In this case, there are two steam turbines, one for each pressure level. The high-pressure steam turbine receives the live steam from the HRSG and expands down to the low-pressure level. At this point the high-pressure stream is mixed with the stream coming from the low-pressure super-heater, to enter the second turbine, and finally expanding to the condenser operational pressure. The power output of the steam cycle will be the sum of both high and low-pressure steam turbines.

3.3.3. Heat recovery steam generator dimensioning

The dimensioning methodology follows the same procedure whether the HRSG has a single or dual pressure arrangement. A bundle of finned tubes disposed vertically conforms each section. Exhaust gases flow horizontally through the tube bundle and enclosure. Each section; economizer, evaporator and super heater, for high and low pressure are calculated separately. Dimensional values cannot be assumed arbitrarily, as there is a strict trade-off among size, heat transfer and costs. The overall heat transfer coefficient method is applied to determine heat transfer areas in each HRSG section. An inline arrangement with plain fins is considered to calculate the heat transfer coefficient, adapting the methodology applied by Dumont [17].

The method is based on the heat transfer area outside the tubes. Calculation of the heat transfer coefficient for the exhaust gases depends on an average outside heat transfer coefficient and on the dimensional inputs, which are used to calculate the Colburn Factor, and a term including the efficiency of the heat transfer through the fins. The heat transfer coefficient and friction factors for fully developed turbulent flow inside the tubes were determined according to the correlations by Gnielinski [18].

The obtained heat transfer area through the dimensional set also results in specific pressure drops, which are evaluated in each HRSG section. A high backpressure for the gas turbine is detrimental for its performance. Likewise, excessive pressure drops in the water/steam affects the steam cycle power output. Pressure drop in exhaust gases side is determined by a set of non-dimensional values detailed in Refs. [17] and [19]. Pressure drops in the Steam/Water side are calculated as in [20].

3.3.4. Weight estimation of steam cycle components

Dimensional characteristics obtained for each heat transfer section are the base for estimating piping material weight, which accounts for an important share of the total HRSG and condenser weight. For the condenser case it was estimated that the total weight is governed by the piping installed in it, by establishing a fixed overall heat transfer coefficient. Nevertheless, for the HRSG, additional components have an important impact on its total weight. When using the weight correlation for the HRSG proposed by Rivera-Alvarez [16], weight results are, in average, larger than those obtained by calculating the tubing weight; the main reason being that the correlation by Rivera-Alvarez [16] includes a rough estimation of other components, such as drums and casing. Therefore, the correlation by Rivera-Alvarez [16] was applied for the HRSG in the optimization model of this work. The steam turbine and generator weight were calculated using the methodology followed by Rivera-Alvarez [16] and Haug [21], assuming that the steam turbine weight is proportional to the steam mass flow rate passing through the turbine.

3.4. Economic evaluation

The implementation of a power hub entails a complex economical evaluation, as it gathers diverse generation technologies along with various operational alternatives. In this study, the economical evaluation is limited to the equipment costs of the combined cycle power plant (PEC), as they are expected to be the most important share of the total power hub capital costs. Equipment costs are strongly dependent on the combined cycle configuration and gas turbine rating. Updated gas turbine costs for commercial devices were obtained using Thermoflex®, while cost estimations for the gas turbine dimensioning approach was made using the continuous correlation described in Ref. [22]. Costs for steam turbine, HRSG, pumps, and electric generator were calculated as stated in Frangopoulos [23], and updated by Carapellucci and Giordano [24] and Roosen et al. [25]. The condenser cost was estimated using the approach suggested by Shah [26]. All correlations result values in USD.

As the power hub is divided in combined cycle blocks, it is expected that the quantity of blocks installed have a significant impact on the purchasing equipment costs. Quantity indexes for the combined cycle components and blocks are similar as those used in the total weight calculation.

4. Results and discussion

4.1. Model validation

An analogous combined cycle structure was modeled using Thermoflex in order to validate the model for the steam cycle design. The case of efficiency optimization for double pressure is taken as reference to validate the preliminary results. The optimization results in a vector with a set of dimensional, thermodynamic and integer variables. Dimensional variables and pinch points of the HRSG obtained in the optimization were introduced in Thermoflex. Resulting steam turbine performance parameters and design pressures from Thermoflex were compared with values of the proposed model.

The results feature a temperature profile and several output parameters; a comparison of these results between the model and Thermoflex is shown in Table 1. Deviations in the temperature nodes are below 1.8%. As may be seen in Table 1 there are no deviations larger than 6,5 %, indicating that model for steam cycle design developed in this work provides reasonable results for the established objectives.

Table 1. Results of the validation of the steam cycle design model

Parameter	Model	Thermoflex	Error (%)
Steam Turbine Power Output (kW)	11.03	10.96	0,68
Mass Flow HP (kg/s)	8.25	8.52	-3,21
Mass Flow LP (kg/s)	2.08	2.04	2,21
High Pressure (bar)	79.91	78.67	1,55
Low Pressure (bar)	10.60	9.91	6,51
Turbine Inlet Temperature (K)	749.32	742.40	0,92

4.2. Single-objective Optimization

For the following results, the nomenclature used for representing the layout of the combined cycle blocks is simplified as follows: the number of gas turbines, followed by the number of heat recovery steam generators, and finally the number of steam turbines. Therefore, a 2(1)x1 block configuration, has two gas turbines with a shared HRSG and one steam turbine.

This optimization procedure allows analyzing extreme points of the variable solution set. Table 2 and Table 3 highlight the most important parameters of the single-objective optimization procedure. There are similarities of the obtained arrangements for both pressure levels. The number of gas turbines and blocks are similar in both cases. Results in one objective optimization tend to reduce blocks in 1(1)x1 arrangements. This allows improving the heat exchange by increasing the heat transfer area in HRSGs delivering steam for a single steam turbine, however, such layout results in a considerable increase in weight.

Optimized dual-pressure arrangements result in more compact combined cycles and smaller HRSGs when compared with single-pressure configurations. These results are mainly because achieving maximum efficiencies requires greater heat transfer areas when dealing with one pressure configurations. Likewise, weight minimization is obtained to the detriment of the heat transfer area.

The results provide a less efficient system for the dual-pressure configuration, including one shared HRSG, and having the same model and quantity of gas turbines as the single-pressure case with three separated HRSGs.

Minimizing equipment costs results in the lowest performance levels. Besides the lowest heat transfer areas for the whole system, minimizing costs also follows the lowest operating pressures in both the single and dual-pressure configurations. Reduced areas and low pressures decrease size of equipment, reducing its efficiency but not necessarily diminishing total weight; the main reason being the selection of gas turbine SGT-800 which holds the lowest cost per MW and the largest weight.

Table 2. Single-objective single pressure optimization results

Optimization case	**Efficiency (η_{cc})**	**Costs (PEC)**	**Weight(M)**
Combined Cycle Design			
GT Model	LM6000 PA	SGT-800-50	LM2500+PV
CC Layout	1(1)x1	1(1)x1	3(3)x1
Groups	4	3	2
HTA (m^2)	43182	19340	9145
W_{ST} (MW)	13.60	11.67	28.09
Objective Values			
η_{CC} (%)	**53.28**	46.22	51.80
PEC (MMUSD)	122.02	**80.67**	130.00
M_{total} (ton)	2588	1895	**1262**

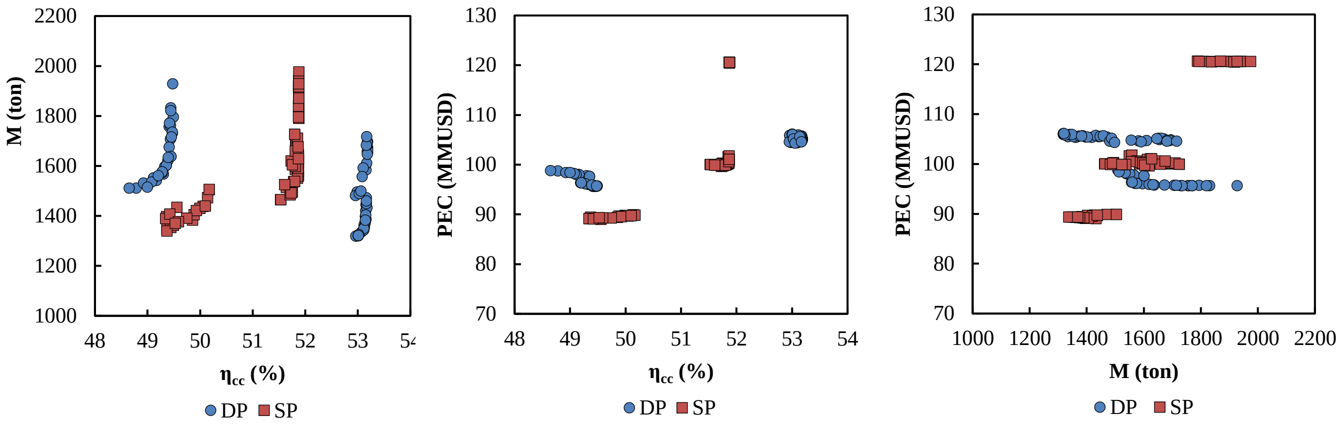
In all cases, there are important reductions in fuel consumption due to the application of combined cycles. Diminishing fuel consumption is directly related to reducing CO_2 emissions. Considering a business-as-usual scenario, in which the three FPSOs remain separated, estimated CO_2 emissions are approximately 455.686 ton/year at full load. The results obtained in this paper suggest that it would be possible to reduce those emissions in a range between 18.7 % and 27.2 % at peak demand for the dual-pressure steam cycle configuration by applying the weight minimization and efficiency maximization optimizations, respectively.

Table 3. Single-objective dual-pressure optimization results

Objective	**Efficiency (η_{cc})**	**Costs (PEC)**	**Weight(M)**
Combined Cycle Design			
GT Model	LM2500+PV	SGT-800-50	LM2500+PV
CC Layout	1(1)x1	1(1)x1	3(1)x1
Groups	5	3	2
HTA (m^2)	26297	18488	19306
W_{ST} (MW)	11.28	14.62	22.90
Objective Values			
η_{CC} (%)	**54.64**	48.67	49.23
PEC (MMUSD)	123.00	**90.70**	133.00
M_{total} (ton)	1536	1540	**858**

4.3. Multi-objective optimization

The results of the multi-objective optimization consist of a three-axis Pareto front, in which each axis represents one of the established objectives. In order to simplify the result visualization, three graphs are created, see Fig. 4. In each graph two objectives are analyzed, for both single-pressure and double-pressure arrangements. Each point of the Pareto front represents a feasible solution of the presented calculation. Thus, it also represents a set of variables and a specific arrangement for the combined cycle. As some variables are integer by nature, the Pareto front of both arrangements presents clusters of results with similar characteristics. CO_2 and fuel consumption reductions, for both single and double-pressure optimization cases, are expected to be between the extreme points of the single-objective optimization values presented previously.



(a) (b) (c)

Fig. 4 Pareto fronts results of the multi-objective optimization study: a) Efficiency (η_{cc}) vs. Weight (M), b) Efficiency (η_{cc}) vs. Costs (PEC), c) Weight (M) vs Costs (PEC).

The single-pressure optimization results in three differentiated solution clusters. This distinction is most appreciable for the equipment costs objective. Higher, medium and lower cost clusters are visible in Fig. 4 (b). The clusters are characterized by the total quantity of gas turbines in the whole power plant, being 6, 5 and 4, respectively for each cluster. As an overall trend, gas turbine efficiency increases, reducing the exhaust gas temperature. This characteristic produces a rise in heat transfer area, in order to obtain high overall combined cycle thermal efficiencies. This reflects in an increase of HRSGs size and cost along with downstream components. The gas turbine inlet temperature is in average 1506 K for all single pressure results. The three objectives show an overall positive correlation when considering all results, meaning that weight, cost and efficiency increase all together and vice versa. However, this trend is not visible among each cluster values, especially weight objective tends to be more disperse and less correlated with the other objectives. An overview of each cluster details is shown below.

Low cost solutions, seen in Fig. 4 (b) as a cluster around 90 MMUSD, are characterized by gas turbines with higher ratings and lower isentropic efficiencies; $\eta_{pt} = 87.1 - 87.3\%$ and $\eta_{co} = 88.5 - 89.6\%$. The power plant would be composed by four 35.52 MW gas turbines, in combined cycle blocks of 1(1)x1 or 2(2)x1. For this cluster, the influence of pinch point and isotropic efficiencies on the overall thermal efficiency are more noticeable, as they have larger variation spans when compared with medium and higher costs clusters. The combination of low pinch points, varying 5 K along the cluster, and high isentropic efficiencies results in overall higher thermal efficiencies.

The results indicate that the medium cost level power plants have five blocks, with gas turbine ratings ranging 32.29 to 32.92 MW in 1(1)x1 CC layouts. In addition to having an influence on the block arrangement, the power plant costs are influenced by small changes on the gas turbine rating. Larger ratings produce higher costs, and vice versa. The influence of the pinch point is similar as the one explained for the previous cluster with a narrower variation span (2 K). The mechanical efficiencies in the gas turbines present limited variations, thus having limited effects on the performances.

Finally, the results indicate that the three-block arrangements have the highest costs. In this cluster, combined cycle blocks are arranged in two gas turbines of 32.35 MW, i.e. 2(2)x1. Design parameters are more constant and their variation span is negligible, and thus their influence in result does not present a specific trend. Specifically, pinch point value averages 20 K, and mechanical efficiencies in the gas turbine are at the upper bound, $\eta_{pt} = \eta_{co} = 90\%$. The relation between gas turbine rating and overall costs is similar as described for the medium cost cluster.

The Pareto front for dual-pressure arrangement presents two main result clusters, namely, power plants featuring four and six gas turbines. Four gas turbine arrangements appear in a cluster surrounding efficiencies around 49% in Fig. 4 (a). Six turbine arrangements are the cluster with average efficiency of 53% in the same figure. In this case, the cluster results are highly differentiated, due to the large gap between average gas turbine ratings between the two clusters, being 34.26 MW and 25.05 MW, respectively. The gas turbine inlet temperature also presents an approximate gap of 100 K between the clusters. Another differentiation feature is the presence of combined cycle blocks with a shared HRSG among the gas turbines, instead of a separate HRSG for each gas turbine. High HP pinch points and low LP pinch points are related to better thermal efficiencies in both clusters. Clusters details are presented as follows.

The first cluster of lower efficiencies is formed by two configurations containing four gas turbines, two CC blocks 2(2)x1 or one block 4(4)x1. In this cluster, objectives and variable trends are more correlated. As compressor isentropic efficiency decreases, so does the thermal efficiency and overall weight. On the contrary, it produces an increase in overall costs. An increase in steam cycle high pressure results in a weight reduction and in cost increase. Optimal gas turbine inlet temperature is about 1452 K with very small variation span. The results suggest that the cost increases when splitting four gas turbines into two blocks, due to the introduction of additional steam cycle components. This block distribution also affects the thermal efficiency, as in one CC block there are four HRSG, meaning a larger heat transfer area, thus increasing the thermal efficiency compared with the two-block arrangement. Even though block distribution is discrete, cost and thermal efficiency variations are smooth among the results and they also depend on gas turbine properties.

The second cluster is characterized by power plants having six gas turbines in two blocks of three gas turbines each. The cluster is divided into power plants having two types of blocks, 3(3)x1 or 3(1)x1. This cluster presents the highest thermal efficiencies of all multi-objective optimized results, ranging from 53.0 % to 53.2 %. The main difference between both clusters is the shared HRSG. This characteristic, along with fact that some design values are close to their bounds, make the relation among variables more disperse. Isentropic efficiencies in the gas turbine are in average $\eta_{pt} = \eta_{co} = 89\%$. The gas turbine inlet temperature varies between 1525 and 1537 K. As the isentropic efficiencies remain constant, small changes in the LP pinch point (varying from 16.8 to 17.6 K) affect the overall thermal efficiency (52.9% to 53.2%). The HP pinch point remains almost constant around 26.8 K for all solutions. As detailed for the previous cluster, also in this case, the cost is influenced by the block layout and gas turbine properties.

5. Conclusions

This paper presented a method for identifying the optimal design of a gas and steam turbine combined cycle tailored for an off-shore power hub. Specifically, a design tailored to satisfy the Brazilian pre-salt basin specific needs was proposed. Both a single-objective and multi-objective optimization studies were carried out, incorporating the performance, weight and cost of the combined cycle. The proposed method allowed identifying a wide range of alternative combined cycle design solutions, including the design and quantity of gas turbines.

The results of the single-pressure optimization suggest that there are not clear benefits of choosing a single-pressure HRSG. In order to reach the maximal thermal efficiency, larger heat transfer areas are required compared to dual-pressure arrangements, resulting in high costs and weight. The effects of introducing commercial gas turbines is clear; the highest combined cycle thermal efficiency was obtained with the high efficiency gas turbine (LM2500), while a lower combined cycle cost was obtained when using a lower cost-to-power ratio gas turbine (SGT-800-50). The results indicate that reducing compressor and gas turbine isentropic efficiencies have a large negative impact on combined cycle thermal efficiency, but a less pronounced effect on the plant weight and cost, since the also the size of the steam cycle is reduced due to drop in compressor and turbine performances.

The results of the multi-objective optimization indicate that dual-pressure arrangement steam cycles, featuring 3 gas turbines, 1 HRSG and 1 steam cycle, could be an attractive design solution for power hubs. Because of the high efficiency and cost-benefit it presents when compared to other results. This arrangement has a low cost and weight, while the thermal efficiency is maintained at a reasonable high level (around 53.2 %). Moreover, the results indicate that by introducing a power hub, the CO_2 emissions may be reduced by 18.7 % to 27.2 % compared with a conventional FPSO design.

Finally, it needs to be pointed out that it is of crucial importance to find out if a selected design will maintain an efficient operation while the heat and power demands are varying over time. Further work includes to extend the methodology presented in this paper to include also other operating conditions of the off-shore platform, by considering a forecasting analysis of the heat and power demands.

Acknowledgments

The authors acknowledge the Brazilian program for High-Education support (CAPES) for the economical funds, as well as the International Post-graduate Exchange program of Santander Bank. This paper is part of a Research and Development project supported by British Gas, part of Shell Group.

Nomenclature

PEC Purchase Equipment Cost

GTI Gas Turbine Index, n/d

NTR Number of Turbines per Block, n/d

HXI Heat Exchanger per Block, n/d

HTA Heat Transfer Area (m^2)

W Power (MW)

M Weight (ton)

MMUSD Million Dollars

Subscripts and superscripts

GT Gas Turbine

ST Steam Turbine

HRSG Heat Recovery Steam Generator

References

[1] Gallo WLR, Gallego AG, Acevedo VL, Dias R, Ortiz HY, Valente BA. Journal of Natural Gas Science and Engineering Exergy analysis of the compression systems and its prime movers for a FPSO unit. J Nat Gas Sci Eng 2017;44:287–98. doi:10.1016/j.jngse.2017.04.023.

[2] De Oliveira S, Van Hombeeck M. Exergy analysis of petroleum separation processes in offshore platforms. Energy Convers Manag 1997;38:1577–84. doi:10.1016/S0196-8904(96)00219-1.

[3] Carranza Sánchez YA, de Oliveira S. Exergy analysis of offshore primary petroleum processing plant with CO2 capture. Energy 2015;88:46–56. doi:10.1016/j.energy.2015.05.130.

[4] Barrera JE, Bazzo E, Kami E. Exergy analysis and energy improvement of a Brazilian floating oil platform using Organic Rankine Cycles. Energy 2015;88:67–79. doi:10.1016/j.energy.2015.03.091.

[5] Pierobon L, Nguyen T Van, Larsen U, Haglind F, Elmegaard B. Multi-objective optimization of organic Rankine cycles for waste heat recovery: Application in an offshore platform. Energy

2013;58:538–49. doi:10.1016/j.energy.2013.05.039.

[6] Nguyen T-V, Tock L, Breuhaus P, Maréchal F, Elmegaard B. Oil and gas platforms with steam bottoming cycles: System integration and thermoenvironomic evaluation. Appl Energy 2014;131:222–37. doi:10.1016/j.apenergy.2014.06.034.

[7] 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. doi:10.1016/j.energy.2014.08.090.

[8] Orlandini V, Pierobon L, Schløer S, De Pascale A, Haglind F. Dynamic performance of a novel offshore power system integrated with a wind farm. Energy 2016;109:236–47. doi:10.1016/j.energy.2016.04.073.

[9] Hetland J, Kvamsdal HM, Haugen G, Major F, Kårstad V, Tjellander G. Integrating a full carbon capture scheme onto a 450 MWe NGCC electric power generation hub for offshore operations: Presenting the Sevan GTW concept. Appl Energy 2009;86:2298–307. doi:10.1016/j.apenergy.2009.03.019.

[10] Manassaldi JI, Mussati SF, Scenna NJ. Optimal synthesis and design of Heat Recovery Steam Generation (HRSG) via mathematical programming. Energy 2011;36:475–85. doi:10.1016/j.energy.2010.10.017.

[11] Rovira A, Sánchez C, Muñoz M, Valdés M, Durán MD. Thermoeconomic optimisation of heat recovery steam generators of combined cycle gas turbine power plants considering off-design operation. Energy Convers Manag 2011;52:1840–9. doi:10.1016/j.enconman.2010.11.016.

[12] Windén B, Chen M, Okamoto N, Kim DK, McCaig E, Shenoi A, et al. Investigation of offshore thermal power plant with carbon capture as an alternative to carbon dioxide transport. Ocean Eng 2014;76:152–62. doi:10.1016/j.oceaneng.2013.07.016.

[13] Thermoflow. Thermoflex 2017.

[14] Toffolo A, Lazzaretto A. Evolutionary algorithms for multi-objective energetic and economic optimization in thermal system design. Energy 2002;27:549–67. doi:10.1016/S0360-5442(02)00009-9.

[15] Saravanamuttoo H, Rogers G, Cohen H, Straznicky P. Gas turbine theory. 6th ed. Dorset, UK: PEARSON Prentice Hall; 2009.

[16] Rivera-Alvarez A, Coleman MJ, Ordonez JC. Ship weight reduction and efficiency enhancement through combined power cycles. Energy 2015;93:521–33. doi:10.1016/j.energy.2015.08.079.

[17] Dumont M-N, Heyen G. Mathematical modelling and design of an advanced once-through heat recovery steam generator. Comput Chem Eng 2004;28:651–60. doi:10.1016/j.compchemeng.2004.02.034.

[18] Gnielinski V. On heat transfer in tubes. Int J Heat Mass Transf 2013;63:134–40. doi:10.1016/j.ijheatmasstransfer.2013.04.015.

[19] Escoa engineering manual. Fin Tube Technology. 2002.

[20] Mehrgoo M, Amidpour M. Constructal design and optimization of a dual pressure heat recovery steam generator. Energy 2017;124:87–99. doi:10.1016/j.energy.2017.02.046.

[21] Haug KC. Weight Estimation of Steam Cycle for CO2 Capture System on Offshore Oil and Gas Installation. NTNU, 2016.

[22] Gas Turbine World. Performance Specs 2016;46.

[23] Frangopoulos CA. Comparison of thermoeconomic and thermodynamic optimal design of a combined-cycle plant. In: Kouremenos DA, editor. Proc. Int. Conf. Anal. Therm. Energy Syst., Athens: 1991.

[24] Carapellucci R, Giordano L. A comparison between exergetic and economic criteria for optimizing the heat recovery steam generators of gas-steam power plants. Energy 2013;58:458–72. doi:10.1016/j.energy.2013.05.003.

[25] Roosen P, Uhlenbruck S, Lucas K. Pareto optimization of a combined cycle power system as a decision support tool for trading off investment vs. operating costs. Int J Therm Sci 2003;42:553–60. doi:10.1016/S1290-0729(03)00021-8.

[26] Shah R, Sekulić D. Fundamentals of Heat Exchanger Design. Hoboken, New Jersey: Wiley; 2003.