Ship Weight Reduction And Efficiency Enhancement Through Combined Power Cycles

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Abstract

This work addresses the problem of configuration of gas and steam turbine combined cycles for ships by simultaneously considering increased efficiency and reduced weight as design objectives. The performed analysis provides basic information to produce systems with simultaneous advantage in both aspects. The combined cycle considered, with total constant power of 20 MW, is modeled as a gas turbine in standard configuration coupled to a simple Rankine cycle. Calculation of system's weight includes the machinery as well as the fuel required to guarantee a given time at sea. To estimate the machinery components weight, some scaling relations have been developed and used. The results presented include an analysis of the predicted weight and efficiency of the combined cycle respect to varying design parameters such as amount of heat recovered, trip duration, steam turbine exit quality, steam generator pinch point, and gas turbine performance. When compared against gas turbines in simple cycle mode, combined cycles produce a fuel requirement reduction that can overcome, in terms of weight, the size increase experienced by the plant. However, it is in general observed that minimum weight and maximum efficiency configurations do not necessarily coincide, as both objectives compete at intermediate values of heat recovery. Therefore, the particular choice of the final design depends on the relative importance assigned to each objective for the considered system. Notably, minimum weight and maximum efficiency solutions are very different for short trip periods but become basically the same for very long ones. Regarding gas turbine operation parameters, they have a strong influence on both total weight and efficiency. An interesting consequence is that a low efficiency gas turbine could produce better results than a high efficiency one, given a large enough temperature for the exhaust gases.

Keywords

Combined cycles; Power plant weight; Power generation in ships; Power plant optimization

Nomenclature

c_p	specific heat (J/kgK)
f_{CC}	combined cycle's load factor
h	enthalpy (J/kg)
k	weight constant ($\text{kg}/(\text{kg/s})^{3/2}$ for turbines, kg/m^2 for heat exchangers)
$\dot{m}$	mass flow rate (kg/s)
s	entropy (J/kgK)
t_{SEA}	time at sea (d)
x	steam quality
A	heat transfer area (m^2)
LHV	fuel's low heating value (J/kg)
M	weight (t)
P	pressure (Pa)
$\dot{Q}$	heat flow (W)
T	temperature ($^{\circ}\text{C}$)
U	heat transfer coefficient ($\text{W/m}^2\text{K}$)
$\dot{W}$	power (W)
δ	auxiliary components' weight fraction
ε	gas turbine's heat loss fraction

η	efficiency
ρ	density (kg/m ³)
χ	power fraction
ΔP_{cw}	condenser's cooling water pressure drop (Pa)
ΔT_{min}	HRSG's minimum temperature difference (°C)
ΔT_{cw}	condenser's cooling water temperature increase (°C)
$\Delta T_{min,COND}$	condenser's minimum temperature difference (°C)

Subscripts

0	identifies fixed component of weight
1-12	identify fluid states represented in Fig. 1
avg	identifies the average temperature for a given range
cw	cooling circuit water or cooling circuit water pump
crit	identifies a qualitative behavior change point in Figs. 4-6
gas	flue gases
loss	identifies gas turbine's heat loss
min	identifies the minimum weight configuration point
max	identifies the maximum efficiency/heat recovery configuration points
w	Rankine cycle water or Rankine cycle water pump
CC	combined cycle
COND	condenser
EC	economizer component of the HRSG
EQUIP	equipment, meaning all components of combined cycle machinery

EV	evaporator component of the HRSG
FUEL	ship's fuel system
GT	gas turbine
HRSG	heat recovery steam generator
RK	Rankine cycle portion of the combined cycle
SH	superheater component of the HRSG
ST	steam turbine
TOTAL	identifies total combined cycle's weight including equipment and fuel
OTHER	identifies the overall weight of auxiliary components of the combined cycle
∞	identifies the ambient temperature

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1. Introduction

Combined cycles based on gas and steam turbines have been increasingly considered as an alternative to the technologies commonly used to provide power in ships. For a detailed account on the prospects of using combined cycles for large ships until 2008, see the review paper by Haglind [1-3]. In general, the advantages of using this type of power plant are apparent for different kind of applications. In commercial ships, where two- and four-stroke diesel engines are largely dominant [4], combined cycles can be considered as a viable alternative to conform to the stricter environmental regulations while offering similar or higher efficiencies. This has been considered in a recent report about the future of ship propulsion [5] and discussed in [6] in the light of environmental requirements for navigation in the Baltic Sea and in [7] under the context of part-load operation. In particular, combined cycles seem to be very suitable to some types of merchant vessels as LNG carriers. Due to the ability to burn boil-off gas, this option has been included in the analyses of propulsion alternatives presented in [8] and [9]. Also, the use of combined cycles on cruise ships can significantly lower the level of pollutants emitted, provide steam for the auxiliaries, and reduce the engine mass and volume (see [10] for an evaluation of the performance of several combined cycle arrangements in cruise ships and [11, 12] for a documented implementation of this powering solution). In navy ships, where gas turbines are very common due to their high power density, excellent reliability, and fast operational response, combined cycles offer a natural way to improve the intrinsically low efficiency of the power plant. This results in an extension of the ship cruising range or a reduction of the required fuel capacity [13, 14]. Furthermore, in the related field of power production for offshore platforms, combined cycles have been also recently revisited as an option to increase power production efficiency of the prevailing gas turbines. In this sense, several papers have been published

including an analysis of challenges and possibilities for the technology [15], the design and performance simulation of the system for a particular platform [16], the system integration on a platform and its thermoenviromonic evaluation [17], its comparison to other waste heat recovery options [18], and the simultaneous optimization in terms of weight and power [19].

Two basic configurations have been commonly considered for the installation of combined cycles in ships. In the COGAS configuration (COmbined Gas turbine And Steam turbine) the turbines are directly connected to the propeller shafts, while in the COGES configuration (COmbined Gas turbine Electric and Steam) the turbines are used to produce electricity through an electric generator and an electric motor is used to drive the propeller shafts [1]. In particular, the idea of using an Integrated Power System where all onboard prime movers are coupled to generators, and the electrical power is distributed through the ship for propulsion and service loads, has been increasingly considered and implemented due to the extra design and operational flexibility it supposes (see [20-23] for a chronological account of the development of the idea in the last two decades). Also, related to the use of combined cycles based on gas and steam turbines, there has been recently some interest to develop other heat recovery systems in ships (mainly for two-stroke diesel engines) with both power and non-power production purposes [24]. This includes absorption refrigeration systems [25, 26], cogeneration systems [27, 28], steam Rankine cycles coupled to two-stroke diesel engines [29], ORCs (both coupled to two-stroke diesel engines [30, 31, 32] or gas turbines [33]), Kalina cycles [34], and dual loop systems [35].

Inherent to the problem of power generation in ships is the need for compactness. In fact, compared with a land-based power plant where efficiency is a main factor and size is not in general an issue, volume and weight of the required system in a ship is always a major

concerning in the design, which often prevails over efficiency. In general, the information reported suggests that by using combined cycles, interesting weight and volume savings can be achieved when compared to diesel systems [2, 6, 10-12]. On the other side, when compared to gas turbine-based power systems, the combined cycle machinery clearly requires extra volume and weight to be allocated in the ship. However, if the fuel storage requirement for a given cruising range is included in the comparison, the potential for volume and weight savings can be important. As will be shown in this work, the larger efficiency of the combined cycle produce a fuel requirement reduction that can overcome the larger size demanded by the plant. Being such an important point of discussion, the volume and weight implications of using a combined cycle in a ship have been mentioned in almost every paper dealing with the subject [2, 6-8, 10-13]. However, the availability of specific data is scarce and a direct consideration of the problem during the design of a plant, as conducted here, has not been published. Very useful in this respect are some recent studies addressing the problem in the context of offshore platforms [15-19].

This paper addresses the problem of configuration of a combined cycle for ship applications by simultaneously considering efficiency and weight as objectives of the design. To the authors' knowledge, this type of analysis has not been published yet. The main goal of the study is to get basic information about the proper design choices that have to be made in order to produce systems with simultaneous advantage in both senses. To model the combined cycle, a standard gas turbine configuration and a simple Rankine cycle have been chosen, which seems to be a convenient option according to previous analyses [10, 15, 16]. When considering the weight associated to the power generation system, inclusion has been made of the machinery as well as of the fuel required to guarantee a given time at sea; to estimate the weight of the machinery

components, some scaling relations have been developed and used. Once the modeling framework is elaborated, the predicted weight and efficiency are analyzed respect to varying design parameters such as amount of heat recovered, trip duration, steam turbine exit quality, steam generator pinch point, and gas turbine performance.

This paper is organized as follows. The general thermal model for the combined cycle is presented in section 2, while the model for weight estimation is presented in section 3. Then, the main results are presented in section 4, where the behavior of efficiency and weight for the combined cycle are analyzed under the influence of different design parameters. Finally, the most important and general findings are summarized in section 5.

2. Combined cycle modeling

Figure 1a shows a schematic representation of the combined cycle considered with its major components and main flows indicated. As can be seen, the configuration for the gas turbine is standard and the Rankine cycle chosen is the simplest possible, with just one pressure level for the steam, no reheating, and no steam extractions for water preheating considered. The consideration of this basic configuration will allow the development of a general picture for the weight-efficiency relationships of the system. It is expected that some degree of increase on the configuration's complexity could lead to extra savings on efficiency at the expense of a larger weight. However, the consideration of these effects is beyond the scope of this work and would be the focus of a future study.

Figure 1. Combined cycle considered.

For the overall combined cycle system, power ($\dot{W}_{CC}$) and efficiency (η_{CC}) can be expressed in the following traditional way:

$$\dot{W}_{CC} = \dot{W}_{GT} + \dot{W}_{RK} \quad (1)$$

$$\eta_{CC} = \frac{\dot{W}_{GT} + \dot{W}_{RK}}{\dot{Q}_{GT}} \quad (2)$$

where $\dot{W}_{GT}$ and $\dot{W}_{RK}$ are, respectively, the power delivered by the gas turbine and the Rankine cycle, and $\dot{Q}_{GT}$ is the heat input to the gas turbine. The fraction of power produced by the Rankine cycle (χ_{RK}) is an important indicator of the combined cycle configuration and is defined as:

$$\chi_{RK} = \frac{\dot{W}_{RK}}{\dot{W}_{CC}} \quad (3)$$

From Eqs. (1) and (3) is clear that a value $\chi_{RK} = 0$ corresponds to a simple cycle configuration and a higher value corresponds to a configuration with larger power coming from the Rankine cycle. The high limit value for χ_{RK} depends on the particular features of the configuration but typically has a maximum value of around 0.4 [36, 37].

In this study, $\dot{W}_{CC}$ will be considered as a constant design parameter imposed by the power requirements of the ship, whereas $\dot{W}_{GT}$ and $\dot{W}_{RK}$ will be adjusted to meet such condition through Eq. (1). In particular, an arbitrary value of 20 MW will be fixed as the required constant power of the combined cycle in the results presented ahead (this choice has been done due to the availability of data to work out the design at that power). As a reference, this is the power level required by a ~2,200 TEU container vessel or a ~220,000 DWT bulk carrier at their corresponding average design speed [38, 39]. As no use is being done here of particular equipment models or brands, it will be assumed that the required features and parameters for the system' components can be scaled continuously according to $\dot{W}_{GT}$ and $\dot{W}_{RK}$. This and next section will outline the model used to relate these variables with the thermodynamic performance and weight of the cycle. In practice, the results obtained will be useful to determine which particular set of existing commercial equipment is best suited for the purpose of the plant or to establish design guidelines for new equipment.

2.1. Gas turbine

For the gas turbine, an overall energy balance can be written as:

$$\dot{Q}_{GT} = \dot{W}_{GT} + \dot{m}_{gas} c_{P, gas} (T_4 - T_\infty) + \dot{Q}_{loss} \quad (4)$$

which expresses that the heat input from the burned fuel $\dot{Q}_{GT}$ is transformed into power $\dot{W}_{GT}$, sensible energy carried by the flue gases $\dot{m}_{gas} c_{P, gas} (T_4 - T_\infty)$, and heat delivered directly to the surroundings $\dot{Q}_{loss}$. In writing the sensible energy term on Eq. (4), an average specific heat $c_{P, gas} = 2.2496 \times 10^{-4} T_{avg} + 1.0367$ has been used for the flue gases with mass flow rate $\dot{m}_{gas}$, where T_{avg} , in °C, is the average between the ambient temperature T_∞ and the exit temperature T_4 . Note

that the objective of the combined cycle is to recover part of this sensible energy to produce extra power. Accordingly with the energy balance, the efficiency of the gas turbine η_{GT} is defined as:

$$\eta_{GT} = \frac{\dot{W}_{GT}}{\dot{Q}_{GT}} \quad (5)$$

Importantly, for almost every gas turbine available in the market [36, 37] T_4 and η_{GT} typically take values in the 450-550 °C and 25-35% ranges respectively, while $\dot{Q}_{loss}$ can be approximated as a constant small fraction ε (usually between 2 and 5%) of $\dot{Q}_{GT}$:

$$\dot{Q}_{loss} = \varepsilon \dot{Q}_{GT} \quad (6)$$

Using Eqs. (5) and (6) in Eq. (3), an expression for the scaling of $\dot{m}_{gas}$ according to $\dot{W}_{GT}$ can be written as:

$$\dot{m}_{gas} = \frac{\dot{W}_{GT}}{\dot{Q}_{GT}} \frac{1 - \varepsilon - \eta_{GT}}{c_{p,gas} (T_4 - T_\infty)} \quad (7)$$

Equation (7) shows a linear relationship between gas flow and gas turbine power, and will be useful to determine (given the gas turbine's operating parameters η_{GT} , ε , and T_4 at the ambient temperature T_∞ , and the specific heat $c_{p,gas}$) the amount of flue gases available to feed the Rankine cycle.

2.2. HRSG and condenser

Figure 1b shows schematically a thermodynamic diagram for the Rankine cycle, with state labels identified according to Fig. 1a. Included in the diagram are the cooling process 4-5 experienced in the Heat Recovery Steam Generator (HRSG) by the flue gases exhausted from the gas turbine and the heating process 11-12 experienced in the Condenser by the cooling water.

Whereas the inlet temperature of the gases to the HRSG T_4 is determined by the gas turbine, the outlet temperature T_5 depends on how much heat is recovered. Clearly, T_5 could take any value lower than T_4 but larger than a minimum established, given by limitations in the stack of the HRSG and taken here as 100 °C. Through this work, T_5 will have the role of a proxy variable determining the configuration of the combined cycle, directly related to the amount of recovered heat as just explained, and through it, to the power fraction χ_{ST} , the total efficiency η_{CC} , and the weight of the power plant.

The HRSG is considered here to have a counter-flow configuration comprising three successive sections: the economizer, the evaporator, and the superheater. The economizer is assumed to heat liquid water from state 6 to the saturated liquid state 6' by cooling hot gases from state 4'' to state 5, the evaporator takes saturated liquid water at 6' to produce saturated vapor at 6'' using hot gases from 4' to 4'', while the superheater produces superheated steam 7 out of saturated vapor 6'' using the hot gases coming from the gas turbine at 4 and cooling it down to 4'. More detailed aspects about the possible configurations for the HRSG, including discussion of the advantages of Once-Through Boilers for ship applications, the consideration of different sequences of sections, and more precise descriptions of the economizer-evaporator transition, are not considered here for the sake of simplicity. Future work in this subject would complement the results found here by including these aspects.

Energy balances for the different sections of the HRSG allow calculation of the corresponding heat flows $\dot{Q}_{EC}$, $\dot{Q}_{EV}$, and $\dot{Q}_{SH}$, and the linking of the gas temperatures T_i and the water enthalpies h_i at every intersection with the mass flow rate of gases coming from the gas turbine $\dot{m}_{gas}$ and the mass flow rate of water inside the Rankine cycle $\dot{m}_w$:

$$\dot{Q}_{EC} = \dot{m}_{gas} c_{p,gas} (T_{4''} - T_5) = \dot{m}_w (h_{6'} - h_6)$$

$$\dot{Q}_{EV} = \dot{m}_{gas} c_{p,gas} (T_{4'} - T_{4''}) = \dot{m}_w (h_{6''} - h_{6'}) \quad (8)$$

$$\dot{Q}_{SH} = \dot{m}_{gas} c_{p,gas} (T_4 - T_{4'}) = \dot{m}_w (h_7 - h_{6''})$$

Furthermore, the addition of heat for the three sections gives the total heat input into the Rankine cycle, through the HRSG:

$$\dot{Q}_{HRSG} = \dot{Q}_{EC} + \dot{Q}_{EV} + \dot{Q}_{SH} = \dot{m}_{gas} c_{p,gas} (T_4 - T_5) = \dot{m}_w (h_7 - h_6) \quad (9)$$

In the same way as for Eq. (4), the average specific heats in Eqs. (8)-(9) are calculated as $c_{p,gas} = 2.2496 \times 10^{-4} T_{avg} + 1.0367$, where T_{avg} are the average temperatures in °C for the corresponding ranges.

An important parameter in the design of the HRSG is the minimum temperature difference ΔT_{min} between hot gases and water, which will be observed at the pinch point. This parameter is directly related with the size of the HRSG and provides a simple way to control the heat transfer area of its components while keeping them and the temperature profile of the whole equipment among feasible limits. Important to remark is that ΔT_{min} affects directly the pressure of process 6-7 (P_{HRSG}), and that typical values for ΔT_{min} are between 10 and 30 °C in HRSGs [36, 37]. For the particular configuration chosen in this analysis, the pinch point could be located at the economizer-evaporator intersection or at the extreme end of the superheater, and consequently, the minimum temperature difference can be calculated as:

$$\Delta T_{min} = \min(T_{4''} - T_{6'}, T_4 - T_7) \quad (10)$$

In the case of the condenser, it is considered here that the cooling water is taken and disposed directly to the sea, which, in terms of equipment, requires just the use of the circulation pump illustrated in Fig. 1 between states 10 and 11. Different condensing alternatives (including the use of cooling towers or air condensers), which would be appropriated in particular cases, are not considered here but could be the focus of a future work.

In a way similar as for the HRSG, the energy balance for the condenser allows calculation of the corresponding heat flow $\dot{Q}_{\text{COND}}$ and the linking of the cooling water temperatures (T_{11} and T_{12}) and the steam enthalpies (h_8 and h_9), with the cooling water mass flow rate $\dot{m}_{\text{cw}}$ and the mass flow rate of water inside the Rankine cycle $\dot{m}_w$:

$$\dot{Q}_{\text{COND}} = \dot{m}_{\text{cw}} c_{P,\text{cw}} (T_{12} - T_{11}) = \dot{m}_w (h_8 - h_9) \quad (11)$$

Important parameters to consider for the condenser design are the condenser pressure on the steam side P_{COND} (determining the condensation temperature T_{COND} and typically taking values as low as 9 kPa(a)), the quality of the steam inlet x_8 (determined by the steam turbine and with common values between 0.9 and 1), the cooling water temperature increase ΔT_{cw} (usually at values between 5 and 10 °C), and the minimum temperature difference between the cooling water and the condensing steam $\Delta T_{\text{min,COND}}$ (with typical values also in the 5-10 °C range). The following definitions can be used for these last two parameters, where it is implicit that 9 is a saturated liquid state at T_{COND} :

$$\Delta T_{\text{cw}} = T_{12} - T_{11} \quad (12)$$

$$\Delta T_{\text{min,COND}} = T_9 - T_{12} \quad (13)$$

2.3. Steam turbine and pumps

According to the Rankine cycle represented in Fig. 1, the power produced by the steam turbine $\dot{W}_{\text{ST}}$ can be calculated as:

$$\dot{W}_{\text{ST}} = \dot{m}_w (h_7 - h_8) \quad (14)$$

with $\dot{m}_w$ already included in Eqs. (8)-(11) and h_7 determined by P_{HRSG} and T_7 (the temperature at the exit of the superheater). The enthalpy at the turbine's outlet h_8 (determined by P_{COND} and x_8)

is additionally related to the inlet enthalpy h_7 through the definition of isentropic efficiency for the steam turbine η_{ST} :

$$\eta_{ST} = \frac{h_7 - h_8}{h_7 - h_{8s}} \quad (15)$$

where η_{ST} is taken as an operational parameter of the particular equipment used (typically taking values in the 75-85% range) and 8s is a reference state at P_{COND} based on the isentropic expansion defined by:

$$s_{8s} = s_7 \quad (16)$$

The powers required by the internal water pump $\dot{W}_w$ (associated to process 9-6) and by the cooling water pump $\dot{W}_{cw}$ (associated to process 10-11), can be calculated as:

$$\dot{W}_w = \dot{m}_w \eta_w \frac{P_6 - P_9}{\rho_w} \quad (17)$$

$$\dot{W}_{cw} = \dot{m}_{cw} \eta_{cw} \frac{\Delta P_{cw}}{\rho_w} \quad (18)$$

where η_w and η_{cw} are the corresponding pumping efficiencies (with common values around 85%), ρ_w and ρ_{cw} are the corresponding water densities, $P_6 - P_9$ is the pressure increase $P_{HRSG} - P_{COND}$ inside the Rankine cycle, and ΔP_{cw} is the pressure drop for the cooling water through the condenser (with usual values in the 300-600 kPa range).

Clearly, the effective power produced by the Rankine cycle can be obtained by subtracting the pumping power consumption from the power delivered by the steam turbine. Therefore:

$$\dot{W}_{RK} = \dot{W}_{ST} - \dot{W}_w - \dot{W}_{cw} \quad (19)$$

which can be used, together with the heat recovered by the HRSG, to define efficiency for the Rankine cycle through:

$$\eta_{\text{RK}} = \frac{\dot{W}_{\text{RK}}}{\dot{Q}_{\text{HRSG}}} \quad (20)$$

Equations (2), (5), and (20) can be used to produce the following relationship between the efficiencies of the combined cycle, the gas turbine, and the Rankine cycle:

$$\eta_{\text{CC}} = \eta_{\text{GT}} + \eta_{\text{RK}} \frac{\dot{Q}_{\text{HRSG}}}{\dot{Q}_{\text{GT}}} \quad (21)$$

From Eq. (21) can be seen that increasing the amount of heat recovered by the HRSG increases the combined cycle efficiency, for a constant value of the Rankine cycle efficiency. However, an excessive increase in the recovered heat could lead to a counterproductive decrease on the Rankine cycle efficiency, via a reduction in the boiling pressure. This effect, broadly known in the design of combined cycles [36, 37], will be addressed together with the corresponding implications on weight, in section 4 of this work.

2.4. Solution approach

The model just described (Eqs. (1)-(21) plus the required relationships between thermodynamic properties of water, i.e. $h_{6'} = h_f(T_{6'})$, $P_{\text{HRSG}} = P_{\text{sat}}(T_{6'})$, $h_7 = h(P_{\text{HRSG}}, T_7)$, $h_9 = h_f(P_{\text{COND}})$, $s_7 = s(P_{\text{HRSG}}, T_7)$, $h_{8s} = h(P_{\text{COND}}, s_7)$, $x_8 = (P_{\text{COND}}, h_8)$, $h_{6''} = h_g(P_{\text{HRSG}})$, $T_8 = T(P_{\text{COND}}, h_8)$, and $T_6 \approx T_9 = T_{\text{sat}}(P_{\text{COND}})$) constitutes a nonlinear system of algebraic equations for the defining variables of every combined cycle configuration. The input parameters for the model and the corresponding values assumed for them according to the guidelines presented above are presented in Table 1. As can be seen for some of the parameters, a range of values is shown next to the primary constant value. This is an indication that such variable will

be changed within that range in the respectively indicated section of this work, with the objective of analyzing efficiency and weight for the corresponding series of configurations. For any other sections the input parameters will be kept constant at the primary value.

The solution for the system of equations, which determines the operating parameters for the corresponding combined cycle configuration, is obtained through an iterative procedure implemented in MATLAB, where three variables are designated as target (x_8 , $\Delta T_{\min}$, and $\dot{W}_{CC}$, included in Table 1) and three more as control variables ($\dot{W}_{RK}$, T_6 , and T_7 , presented earlier in the model) that are adjusted until a solution is found. In brief, the procedure begins by assigning values to the parameters in Table 1, and guessing values for the three control variables. At each iteration, interim values for the targets variables can be found explicitly from the model presented in this section. Then, the control variables are adjusted and reapplied based on the match between the calculated and the originally assigned values for the target variables. Once the desired precision is achieved, every single feature of the configuration can be calculated, which includes important performance indicators as the overall efficiency (η_{CC}), the Rankine cycle efficiency (η_{RK}), and the Rankine cycle power fraction (χ_{RK}), as well as the required information to perform the weight estimation in next section.

Table 1. Input Parameters for Thermodynamic Configuration Calculation

Parameter	Value
HRSG Exhaust Gas Temperature, T_5	100-500 °C (Sec. 4.1-4.5)
Steam Turbine Exhaust Quality, x_8	95 % (90-102 % in Sec. 4.3)
HRSG Pinch Point, $\Delta T_{\min}$	20 °C (5-40 °C in Sec. 4.4)
Gas Turbine Exhaust Temperature, T_4	500 °C (450-550 °C in Sec. 4.5)
Gas Turbine Efficiency, η_{GT}	30 % (25-35 % in Sec. 4.5)
Miscellaneous Gas Turbine Losses, ε	4 %
Steam Cycle Condenser Pressure, P_{COND}	0.1 bar(a)
Steam Turbine Isentropic Efficiency, η_{ST}	80 %
Combined Cycle Power, $\dot{W}_{CC}$	20 MW
Flue Gases Average Specific Heat, $c_{p,gas}$	1.07-1.16 ^a kJ/kgK
Cooling Water Specific Heat, $c_{p,cw}$	4.18 kJ/kgK
Ambient Temperature, T_{∞}	15 °C
Cooling Water Inlet Temperature, T_{11}	25 °C
Cooling Water Temperature Increase, ΔT_{cw}	5 °C
Water Pumps Efficiency, η_w and η_{cw}	85 %
Cooling Water Circuit Pressure Drop, ΔP_{cw}	450 kPa
Water density, ρ_w and ρ_{cw}	1,000 kg/m ³

^a Actually calculated through $c_{p,gas} = 2.2496 \times 10^{-4} T_{avg} + 1.0367$, which take values within the indicated range.

3. Weight estimation

There are five main weight components related with the use of a combined cycle in a ship: the gas turbine (M_{GT}), the HRSG (M_{HRSG}), the steam turbine (M_{ST}), the condenser (M_{COND}), and the required fuel to operate the system (M_{FUEL}). In broad terms, for a fixed total power of the combined cycle M_{GT} and M_{FUEL} decrease while M_{HRSG} , M_{ST} , and M_{COND} increase with higher waste heat recovery and system efficiency. According to this, the total weight associated to the system will be calculated here as:

$$M_{TOTAL} = M_{GT} + M_{HRSG} + M_{ST} + M_{COND} + M_{FUEL} + M_{OTHER} \quad (22)$$

where the extra term M_{OTHER} represents the added weight of auxiliary components such as pumps, pipes, fans, water, or instruments that are essential to the operation of the system but are too small to be accounted individually. To deal with it, a small constant percentage δ (around 5%) of the total weight for the equipment is considered:

$$M_{OTHER} = \delta (M_{TOTAL} - M_{FUEL}) \quad (23)$$

Analysis of weight implications due to the installation of a combined cycle in a ship requires ability to calculate the previously described weight components as a function of the configuration variables. Even when the exact calculation of weight should be a complicated process involving a big number of factors, the effort here will be directed to the development of methods to estimate it in the most precise way possible while keeping the dependence on the variables discussed in previous section as simple as possible. In that sense, is very important to understand the approach here as a first approximation to the weight of a combined cycle, simple and accurate enough to observe how the general design behaves under different choices.

Weight decomposition of different types of ship places the power plant machinery and fuel weight in the approximated ranges 2 to 6 % and 3 to 14 % of the total ship weight respectively [40]. This means the present analysis deals with the possibilities of reducing a fraction corresponding to between 5 and 20 % of the total weight of a ship.

3.1. Gas and steam turbines

Same analytical approach is used here to calculate weight of both the gas turbine and the steam turbine, showing that in general terms and under certain (similarity) conditions, M_{GT} and M_{ST} can be expressed as simple increasing functions of $\dot{m}_{gas}$ and $\dot{m}_w$ respectively (and through them of $\dot{W}_{GT}$ and $\dot{W}_{ST}$, cf. Eqs. (7) and (14)). In more detailed descriptions, factors such as operating pressures and temperatures, internal components efficiency, and particular auxiliary parts and manufacturing details should have a more direct influence in the determination of weight. However, in this work they will play a role just through their influence over the corresponding mass flow rate.

It is important to clarify that the considered weights for the gas and steam turbines just include the mechanical parts for the equipment itself, disregarding any required gearbox, electric generator, or powered system. All these additional equipment, in fact, could add a very significant amount of weight (for a typical industrial gas turbine-generator package, the gas turbine itself could have a weight as low as 15% of the total). However, this work is focused in the possibilities to produce a prescribed amount of power for a ship, no in the alternatives to transform or use that power in an optimal way. An all-electric ship, for example, could have different options to change the weight of the mechanical-to-electrical conversion system by combining electric conversion technologies and adjusting electrical operation parameters. The

analysis of solutions to take a prefixed amount of power to the end users inside a ship is an important problem closely related to the one presented here, but at the same time independent because of its presence regardless the use of a combined cycle. Future analysis should address it.

If similarity is assumed for the entire set of possible gas and steam turbines to be employed in the combined cycle, then scaling laws can be used to relate their corresponding operating parameters, including weight and mass flow rate. This assumption implies that for any configuration, the particular gas and steam turbines will be scaled versions of their respective predefined designs. This is especially reasonable if, for example, a given manufacturer product series has been chosen to supply the power generation equipment for a fleet, or a particular turbine model is trying to be adjusted to service a particular ship design. As described in [41] for the gas turbine case, application of scaling allows the development of new models out from existing ones by keeping temperatures, pressures, blade angles, velocities, and stresses constant. As a result, pressure ratio, efficiency and blade tip speed are also maintained; linear dimensions such as diameter or length vary directly with the scale factor; shaft speed varies as the inverse of the scale factor; mass flow and power vary with the square of the scale factor; and volume and weight vary with the cube of the scale factor. Therefore, for a similar set of turbines, weight M and mass flow rate $\dot{m}$ can be related according to the following simple formula:

$$M = k \cdot \dot{m}^{3/2} \quad (24)$$

where k is a constant determined from the corresponding base design turbine.

It is important to notice that the weight considered in the previous scale analysis corresponds to the portion varying with the size of the equipment. Some generality is gained if a fixed weight portion is added to the estimation formula, so that any non-size dependent component of the system can be taken into account in a better way. Consequently, gas and steam

turbine weights will be calculated as functions of the gas and steam mass flow rates respectively through the expressions:

$$M_{GT} = k_{GT} \cdot (\dot{m}_{gas})^{3/2} + M_{GT,0} \quad (25)$$

$$M_{ST} = k_{ST} \cdot (\dot{m}_w)^{3/2} + M_{ST,0} \quad (26)$$

From public available information for gas turbines in the 4 to 25 MW range, constants k_{GT} and $M_{GT,0}$ have been estimated to be between 27.3 and 33.8 kg/(kg/s)^{3/2}, and 0 and 573 kg respectively. In the same way, for steam turbines in the 3 to 20 MW range the corresponding constants k_{ST} and $M_{ST,0}$ have been estimated between 202 and 238 kg/(kg/s)^{3/2}, and 654 and 2,641 kg respectively. Figure 2 shows the areas covered by the proposed expressions with the estimated constant values for both gas and steam turbines in a weight-mass flow rate diagram, together with the individual data found. It is important to keep in mind that all the reported values, even when obtained from real equipment, must be used carefully and mostly for indicative purposes. A series of turbines designed, for example, with the particular goal of having an increased power density, could be located well outside the area defined in Fig. 2.

Figure 2. Weight vs. mass flow rate for gas and steam turbines.

3.2. HRSG and condenser

In a similar way as for turbines weight estimation, an analytical approach is used here to calculate weight of both the HRSG and the condenser. It will be seen that, as particular forms of heat exchangers, HRSG and condenser weights can be modeled in a simplified way as linear functions of their corresponding heat transfer areas, which can be calculated from the modeling information of the combined cycle obtained in section 2. In a more detailed description, weight could be found a direct function of factors as pressures, temperatures, flow rates, and/or manufacturing or design details, however, in this approach every variable plays its role solely through the influence over the corresponding heat transfer area.

Accordingly with the description in section 2, the heat transfer area for the considered HRSG (A_{HRSG}) can be decomposed in the areas of its individual components, economizer (A_{EC}), evaporator (A_{EV}), and superheater (A_{SH}):

$$A_{HRSG} = A_{EC} + A_{EV} + A_{SH} \quad (27)$$

where the required area for every heat exchanger can be calculated from the common expression relating heat rate to the corresponding mean logarithmic temperature difference:

$$\begin{aligned} \dot{Q}_{EC} &= U_{EC} A_{EC} \frac{(T_{4''} - T_{6'}) - (T_5 - T_6)}{\ln((T_{4''} - T_{6'}) / (T_5 - T_6))} \\ \dot{Q}_{EV} &= U_{EV} A_{EV} \frac{(T_{4'} - T_{6''}) - (T_{4''} - T_{6'})}{\ln((T_{4'} - T_{6''}) / (T_{4''} - T_{6'}))} \\ \dot{Q}_{SH} &= U_{SH} A_{SH} \frac{(T_4 - T_7) - (T_{4'} - T_{6''})}{\ln((T_4 - T_7) / (T_{4'} - T_{6''}))} \end{aligned} \quad (28)$$

In Eqs. (28), every heat rate and temperature is known from the modeling of the combined cycle presented in section 2, while the heat transfer coefficients for the economizer

(U_{EC}), evaporator (U_{EV}), and superheater (U_{SH}) have typical values in the ranges 30 to 50, 25 to 65, and 50 to 90 W/(m²K) respectively [42].

Following the same procedure, the heat transfer area for the condenser (A_{COND}) can be calculated from:

$$\dot{Q}_{COND} = U_{COND} A_{COND} \frac{(T_8 - T_{12}) - (T_9 - T_{11})}{\ln((T_8 - T_{12}) / (T_9 - T_{11}))} \quad (29)$$

where again, the heat transfer rate and the temperatures come from the thermodynamic modeling of the combined cycle, and the heat transfer coefficient (U_{COND}) takes a value in the typical range 2,000 to 4,000 W/(m²K) [43].

Importantly, under this approach the values for all heat transfer coefficients (U_{EC} , U_{EV} , U_{SH} , and U_{COND}) will be assumed constant for the entire sets of possible heat exchangers to be used in the combined cycle. Also, such values are considered to include the effect of the corresponding heat exchanger correction factor. Both things are reasonable if, for example, predetermined HRSG and condenser designs are intended to adjust to the required size of a combined cycle, as usually dimensions are modified maintaining the same flow configuration and trying to keep heat transfer coefficients at the same level. However, it is important to keep in mind that sizing a heat exchanger implies a trade-off between a high heat transfer rate and a low pressure-drop. Because of that, variations in the heat transfer coefficient could be observed to satisfy pressure drop constraints, when the same heat exchanger design is used. These considerations are very important and may have a considerable impact in the calculated heat transfer area and in the approach used to estimate weight, as will be discussed. However, to include them in the analysis is beyond the scope of this work and should be the focus of a future work.

In general terms, the weight of a heat exchanger can be decomposed in the weight of the heat exchange surface, the weight of the external structure, the weight of the fluid, and a fixed (in most cases small) weight coming from any non-size dependent component. The weight of the heat exchange surface can be considered proportional to the heat transfer area if its thickness is fixed, as usually common for tubes and fins. The weight of the filling fluids is proportional to the volume of the heat exchanger, which, at the same time, can be considered proportional to the heat transfer area if internal spacing (diameter, pitch) is fixed, as is also usually the case.

The weight of the external structure, to complete the picture, is the most complex part. It can be considered proportional to the external area of the heat exchanger if its thickness (shell, refractory, insulation, structural components) is fixed. But such external area can be related to the heat transfer area in several different ways depending on the criteria employed to accommodate the internal components of the heat exchanger. For example, if external aspect ratios of the equipment are kept constant (which, however, can modify fluid velocities and consequently heat transfer coefficients), it can be shown that external area is proportional to the heat transfer area raised to $2/3$. If fluid velocity is maintained constant instead (which, however, can affect pressure drop through the heat exchanger when large variations in the device dimensions are experienced), it can be shown that external area is a linear function of heat transfer area, with constants depending in the mass flow rate. If pressure drop for the fluid on the external side of the tubes is fixed (which affects heat transfer coefficients as explained before), dependence between external area and heat transfer area becomes more complicated, with raised-to-the $1/3$ and $5/6$ terms and important influence of the mass flow rate.

The detailed discussion of these considerations is intimately related with the discussion about the heat transfer coefficient and is beyond the scope of this work. Due to the linear

dependence of the heat exchange surface and fluid weights on the heat transfer area, and the almost-linear dependence of the external structure weight on the same, it will be assumed here that the total heat exchanger weight can be expressed as a simple linear function of the heat transfer area. Therefore, the following expressions will be used for HRSG and condenser weight:

$$M_{\text{HRSG}} = k_{\text{HRSG}} \cdot A_{\text{HRSG}} + M_{\text{HRSG},0} \quad (30)$$

$$M_{\text{COND}} = k_{\text{COND}} \cdot A_{\text{COND}} + M_{\text{COND},0} \quad (31)$$

In Eq. (30), constants k_{HRSG} and $M_{\text{HRSG},0}$ have been estimated to be in the 8.5 to 9.5 kg/m² and 520 to 4,650 kg ranges respectively, while values for constants k_{COND} and $M_{\text{COND},0}$ in Eq. (31) have been estimated in the 12.5 to 13.5 kg/m² and 5,000 to 9,000 kg ranges. Figure 3 shows the areas covered by the proposed expressions Eq. (30)-(31) with the estimated value ranges for the constants, in a weight-heat transfer area diagram. Included in Fig. 3 are some individual data found for different HRSGs and condensers manufactured by two companies. It is important to note that the data shown for the HRSGs have areas spanning from around 1,500 m² to 7,000 m², which are capable to recover from around 5 MW to 19 MW of thermal energy out from the flue gases of a GT, and therefore have the proper size for a combined cycle to be configured with a GT in the approximated 5 to 20 MW range. At the same time, the data included for the condensers have areas between 500 and 1,500 m², capable of evacuating between 15 and 45 MW of thermal power corresponding to a steam turbine with a power in the approximated 5 to 15 MW range. As for the turbines weight, it is important to keep in mind that the values reported here must be used carefully and mostly for indicative purposes: data in Fig. 3 correspond to heat exchange equipment designed for terrestrial applications.

Figure 3. Weight vs. heat transfer area for HRSGs and condensers.

3.3. Fuel

Fuel weight represents a significant portion of the total weight associated to the power generation system in a ship. As stated before, by increasing the efficiency of the power generation system, the fuel weight required for a given mission in a ship is decreased. It will be shown in the results section of this paper that fuel weight savings can easily offset any required increase in the combined cycle equipment weight, producing a double ship-weight fuel-consumption reduction effect.

Fuel weight calculation follows straightforward from the operational conditions assumed for the ship, the combined cycle power ($\dot{W}_{CC}$), and the efficiency (η_{CC}) found in section 2. It can be performed by the formula:

$$M_{FUEL} = \frac{t_{SEA} f_{CC} \dot{W}_{CC}}{\eta_{CC} LHV} \quad (32)$$

where M_{FUEL} represents the weight of fuel required at embarkation, t_{SEA} and f_{CC} are the expected time at sea and the average load factor of the combined cycle for the mission, and LHV is the low heating value of the fuel (typically around 43,000 kJ/kg for liquid fuels used in marine

applications). At this point, it is important to remark that the time at sea variable t_{SEA} depends not only on the type of ship but in the kind of mission analyzed. Common values for it could range from 1 to 15 days. Methodologies to determine proper values for the equivalent time at sea as a function of mission duration and some other operational features can be found in the specialized literature [44] and are not further discussed here.

Regarding the load factor f_{CC} , it will depend on the mission's power demand curve and the size chosen for the combined cycle. The amount of total power required by a ship at some specific time is a function of changing factors as ship speed, sea state, and service loads usage, which makes it highly variable. Similarly to the calculation of the equivalent time at sea or the amount of fuel required in a mission, different methodologies (whose discussion is out of the scope of this work) have been developed to estimate the installed power requirement of a ship [44]. In any case, it is reasonable to assume that, in most cases, the combined cycle installed power will be just a fraction of the total installed power, which should supply the base power for the ship at a high efficiency. In that way, the combined cycle should operate close to a load factor $f_{CC} = 100\%$ and should deliver a representative portion, if not most, of the energy needs, whereas other complementary generation equipment with a simpler, cheaper, or lighter configuration (but not as efficient), will take care of the power demand peaks. Additionally and in connection with the assumption of f_{CC} close to 100%, is the fact that all the previous analysis (including the calculation of efficiency in section 2) was performed for such condition. If the load factor of the combined cycle goes well below that value, a decrease on the efficiency should be expected and additional elements should be considered in the analysis in order to make it valid [7].

3.4. Summary of input parameters

The model described in this section (Eqs. (22)-(32)) allows the estimation of the weight related to the installation of a combined cycle in a ship, after the corresponding configuration has been calculated in section 2. The input parameters for the weight estimation model and the corresponding values assumed for them according to the guidelines presented above are presented in Table 2. As can be seen for parameter t_{SEA} , a range of values is shown next to the primary constant value. This is an indication that such variable will be changed within that range in the correspondingly indicated section of this work. For any other sections the input parameters will be kept constant at the primary value.

4. Results

Following, the model developed in sections 2 and 3 will be used to analyze the efficiency and weight of a combined cycle under different configurations, with the input parameters to be used in the model listed in Tables 1 and 2. A base case will be presented first (section 4.1), which uses the primary values indicated in those tables, and which allows a varying value for the HRSG exhaust gas temperature T_5 in order to consider combined cycles with distinct degrees of waste heat recovery. Then, the influence of the following parameters on the behavior of the cycle will be analyzed: the time at sea t_{SEA} (section 4.2), the steam turbine exhaust quality x_8 (section 4.3), the HRSG pinch point ΔT_{min} (section 4.4), and the gas turbine efficiency η_{GT} and exhaust temperature T_4 (section 4.5). In each corresponding subsection the parameters of interest will be varied inside the value range indicated in Tables 1 and 2, while all the other parameters will remain fixed at the primary value used for the base case.

Table 2. Input Parameters for Weight Estimation

Parameter	Value
Weight of auxiliary components, δ	5 %
Gas Turbine weight constant, k_{GT}	31 kg/(kg/s) ^{3/2}
Gas Turbine fixed weight, $M_{GT,0}$	300 kg
Steam Turbine weight constant, k_{ST}	220 kg/(kg/s) ^{3/2}
Steam Turbine fixed weight, $M_{ST,0}$	1,700 kg
Economizer heat transfer coefficient, U_{EC}	40 W/m ² K
Evaporator heat transfer coefficient, U_{EV}	45 W/m ² K
Superheater heat transfer coefficient, U_{SH}	70 W/m ² K
Condenser heat transfer coefficient, U_{COND}	3,000 W/m ² K
HRSG weight constant, k_{HRSG}	9 kg/m ²
HRSG fixed weight, $M_{HRSG,0}$	2,500 kg
Condenser weight constant, k_{COND}	13 kg/m ²
Condenser fixed weight, $M_{COND,0}$	7,000
Time at sea, t_{SEA}	3 d (1.5-10 d in Sec. 4.2)
Combined cycle load factor, f_{CC}	100 %
Low Heating Value of the fuel, LHV	43,000 kJ/kg

4.1. Base case - The effects of varying heat recovery in the HRSG

An increasing heat recovery from the exhaust gases of the gas turbine implies a decreasing temperature T_5 at the outlet of the HRSG. At no heat recovery, T_5 will be equal to the

temperature of the exhaust gases coming from the gas turbine T_4 (fixed at 500 °C for the base case). On the other side, the maximum heat recovery will occur at the minimum temperature T_5 allowed in the stack, taken here as 100 °C. Following this idea, Figs. 4 and 5 present the behavior of several key parameters of the combined cycle configuration as a function of T_5 .

Figure 4. Combined cycle efficiency, Rankine cycle efficiency, and steam turbine power fraction vs. HRSG exhaust gas temperature for the base case.

Figure 5. Gas and steam temperatures at the gas turbine-superheater and the economizer-evaporator transitions vs. HRSG exhaust gas temperature for the base case.

In Fig. 4, the combined cycle and Rankine cycle efficiencies η_{CC} and η_{RK} , as well as the fraction of power produced by the Rankine cycle χ_{RK} , are shown as a function of the heat recovery in the HRSG indicated by T_5 . As can be seen, starting from $T_5 = 500$ °C and all the way down to the critical value $T_{5,crit} = 178.5$ °C (indicated in Fig. 4 with a vertical dashed line), a larger heat recovery (lower T_5) means an increasing fraction of the power produced by the steam turbine and an increasing efficiency for the combined cycle, as a consequence of the constant observed efficiency for the Rankine cycle. In this range, the power of the Rankine cycle goes from 0 to 5.84 MW (representing the 29.2 % of the total 20 MW produced) and the efficiency of the combined cycle goes from 30 % (the efficiency of the gas turbine in the simple cycle configuration) to 42.4 % as maximum value, while the efficiency of the Rankine cycle remains constant at 27.8 %. Below this range, further heat recovery (leading to exhaust gases flowing out of the HRSG colder than 178.5 °C) will produce a rapidly decreasing efficiency of the Rankine cycle, which on turn, will decrease the power produced by the steam turbine and the total efficiency of the combined cycle. At $T_5 = 100$ °C for example (the minimum temperature allowed for the exhaust gases here), the efficiency of the Rankine cycle has decreased to 15.9 %, reducing its power production to 4.52 MW (22.6% of the total power) and the efficiency of the combined cycle to 38.8 %. Even when more energy is been made available to the Rankine cycle through the increased heat recovery in this situation, the dramatic decrease on the efficiency to convert that energy into power actually causes a detriment on the steam turbine power and the resultant power plant total efficiency.

Figure 5 helps to clarify the effect that HRSG's heat recovery has on the efficiency of the Rankine cycle. Indicated in Fig. 5 are the temperature of both exhaust gases and water steam at the gas turbine outlet-superheater inlet transition (T_4 and T_7 respectively), as well as at the

economizer-evaporator transition ($T_{4''}$ and $T_{6'}$ respectively). As can be seen, for $T_5 > T_{5,\text{crit}} = 178.5\text{ }^{\circ}\text{C}$, the thermodynamic configuration of the Rankine cycle remains constant. In this case, the temperature of the superheated steam $T_7 = 480\text{ }^{\circ}\text{C}$ is $20\text{ }^{\circ}\text{C}$ below the fixed $T_4 = 500\text{ }^{\circ}\text{C}$ (a consequence of the pinch point condition Eq. (10)), while the constant evaporation temperature $T_{6'} = 249.4\text{ }^{\circ}\text{C}$ determines the high level of pressure for the cycle $P_{\text{HRSG}} = 3.937\text{ MPa}$. Also in this case, as T_5 decreases to $T_{5,\text{crit}}$ the temperature difference between the gases and the steam at the economizer-evaporator transition ($T_{4''} - T_{6'}$) continuously decreases to $\Delta T_{\text{min}} = 20\text{ }^{\circ}\text{C}$. On the other side, for $T_5 < T_{5,\text{crit}}$, the pinch point condition Eq. (10) imposes a continuous reduction of the evaporation temperature (with the consequential reduction of P_{HRSG}), which causes a substantial decrease in the superheating temperature T_7 . It can be observed in Fig. 5, how the temperature difference $T_{4''} - T_{6'}$ remains constant at $20\text{ }^{\circ}\text{C}$ in this case, creating a continuously changing thermodynamic configuration for the Rankine cycle, which has lower evaporation and superheating temperatures (and consequently lower efficiency) when T_5 is further reduced. At the minimum $T_5 = 100\text{ }^{\circ}\text{C}$, for example, the evaporation and superheating temperatures have been reduced to $T_{6'} = 139.3\text{ }^{\circ}\text{C}$ (corresponding to $P_{\text{HRSG}} = 355\text{ kPa}$) and $T_7 = 239.7\text{ }^{\circ}\text{C}$, which explains the reduction of the Rankine cycle efficiency to the previously reported low value of 15.9% .

Turning the attention to the weight features of the combined cycle, Fig. 6 illustrates the weight composition as a function of T_5 . As explained in section 3, the total weight is the sum of equipment and fuel weights ($M_{\text{TOTAL}} = M_{\text{EQUIP}} + M_{\text{FUEL}}$), whereas the equipment includes all components of the combined cycle machinery ($M_{\text{EQUIP}} = M_{\text{HRSG}} + M_{\text{GT}} + M_{\text{COND}} + M_{\text{ST}} + M_{\text{OTHER}}$). For convenience, just the two more representative of those components (in terms of weight), HRSG and gas turbine, have been included in Fig. 6. As can be seen, the total

equipment weight increases with a larger heat recovery (lower T_5). In that case, with the increasing contribution of the Rankine cycle to the total size of the power plant, the additional weight required in HRSG, condenser, and steam turbine overpasses the weight decrease experienced by the gas turbine. In particular, for the simple cycle configuration ($T_5 = 500$ °C), the total equipment weight is 36.8 t, with 64 % of it (23.6 t) corresponding to the gas turbine and the rest coming from the residual weight of the other devices (the HRSG weight is 2.5 t in this case as a consequence of the assumed expression Eq. (30)). Comparatively, for maximum heat recovery ($T_5 = 100$ °C), the total equipment weight is 113.4 t, with 62 and 14 % of it (70.0 and 16.2 t) corresponding to the HRSG and gas turbine respectively. Interestingly, for the previously mentioned maximum efficiency configuration ($T_5 = 178.5$ °C), the weight of the HRSG (and consequently that of the total equipment) experiences a slope discontinuity as a result of the mean logarithmic temperature difference behavior for the HRSG.

Figure 6. HRSG, gas turbine, equipment, fuel, and total weights vs. HRSG exhaust gas temperature for the base case.

The fuel weight behavior included in Fig. 6 follows straightforward from Eq. (32), where a larger efficiency means a smaller fuel weight. For the base case assumed value of $t_{SEA} = 3$ d,

the fuel weight is reduced from 401.3 t in simple cycle configuration ($T_5 = 500\text{ }^{\circ}\text{C}$) to 284.4 t at maximum efficiency ($T_5 = 178.5\text{ }^{\circ}\text{C}$), and increases again to 311.1 t for maximum heat recovery ($T_5 = 100\text{ }^{\circ}\text{C}$). The total weight associated to the power plant, obtained from adding fuel and equipment weight, is also included in Fig. 6 to clarify the overall impact of a combined cycle for different degrees of heat recovery. As can be seen, for the base case there is always an overall weight reduction respect to the no heat recovery situation, which happens because the fuel weight savings are in all cases larger than required equipment weight increase. Specifically, the total weight goes down from 438.1 t for the simple cycle configuration ($T_5 = 500\text{ }^{\circ}\text{C}$), to 368.0 t (16 % decrease) at the minimum weight configuration (occurring when $T_5 = 233.5\text{ }^{\circ}\text{C}$), and goes up again to 380.2 t at maximum efficiency ($T_5 = 178.5\text{ }^{\circ}\text{C}$) and 424.5 t at maximum heat recovery ($T_5 = 100\text{ }^{\circ}\text{C}$). Most importantly here is that the minimum weight configuration does not coincide with the maximum efficiency one. In fact, for the minimum weight of 368.0 t the efficiency of the combined cycle is 40.3 %, while for the maximum efficiency of 42.4 % the corresponding weight is 380.2 t. Clearly, beyond the minimum weight configuration, the equipment weight rise required to increase heat recovery will be larger than the fuel weight reduction produced by the efficiency gain.

To better understand the previous observation about the competition of two different objectives (minimum weight, maximum efficiency) in the configuration of a combined cycle, Fig. 7 presents a combination of the data compiled in previous Figs. 4 and 6. In this figure, efficiency of the combined cycle (η_{CC}) is taken as abscissa while total weight divided by the fixed time at sea (M_{TOTAL}/t_{SEA}) is taken as ordinate. For the base case discussed in this section (with $t_{SEA} = 3$ d), the important information is clearly identified in the figure as a bold line on which T_5 plays a parametric role: every point in such curve corresponds to a different value of T_5 and

consequently to a different configuration. As can be seen, starting from the simple cycle configuration (labeled point SC in Fig. 7), the line goes to the minimum weight configuration (point $M_{TOTAL,min}$), to then go to the maximum efficiency configuration (point $\eta_{CC,max}$), and end up at the maximum heat recovery configuration (point $\dot{Q}_{HRSG,max}$). For the two sections of the line running from the simple cycle to the minimum weight and from the maximum efficiency to the maximum heat recovery, it is clear that a larger efficiency of the combined cycle also means a shorter total weight. However, for the section of the line running from the minimum weight to the maximum efficiency configurations, those objectives are in competition as any efficiency improvement implies a setback on the total weight of the combined cycle. As can be drawn from these observations, every optimal design must lie in the short section of the curve joining minimum weight and maximum efficiency, with the particular location of the final design depending on the relative importance of each one of the objectives for the considered system.

Figure 7. Combined cycle total weight per unit time at sea vs. efficiency for different trip durations.

4.2. The effects of trip duration

In addition to the base case ($t_{SEA} = 3$ d), Fig. 7 also includes the total weight vs. efficiency curves for the combined cycle at several other $t_{SEA} = 1.5, 2, 5,$ and 10 d. The calculation of such curves has been carried in the same way as described before, keeping all the remaining parameters fixed as indicated in Tables 1 and 2. As can be seen, the trip duration doesn't affect the efficiency at the simple cycle, maximum efficiency, and maximum heat recovery configurations, which keep being 30 %, 42.4 %, and 38.8 % respectively as in the base case situation. However, the total weight associated to the power plant decreases with longer trips for those three points and for any other prescribed efficiency configuration, which is a consequence of the diminishing participation of the total equipment weight respect to the fuel weight for longer periods. As a reference, in the case of the simple cycle, the value for M_{TOTAL}/t_{SEA} take values of 158.3, 152.2, 146.0, 141.1, and 137.5 t/d for 1.5, 2, 3, 5, and 10 d respectively. Regarding the minimum weight configuration, its efficiency increases with trip duration in such a way that approaches the maximum attainable efficiency when the period is too long. According to this, the efficiency at the minimum weight configuration take values of 37.9, 39.2, 40.3, 41.4, and 42.3 % for 1.5, 2, 3, 5, and 10 d respectively. Also, the percentage of weight reduction at the point of minimum weight (respect to the weight at simple cycle) increases with longer trips taking values of 9.4, 12.3, 16.0, 20.0, and 24.1 % for 1.5, 2, 3, 5, and 10 d respectively.

In summary, trip duration has an important effect on the total weight of the power plant and the efficiency for minimum weight. For short trip periods (shorter than 5 d) the locus of optimal designs (the section of the curve joining minimum weight and maximum efficiency), has a considerable size and the problem of determining the optimal configuration based on the

competence of the two objectives considered is important. For long trip periods (e.g. 10 d) minimum weight and maximum efficiency configurations are basically the same and the concept of competing objectives loses its importance.

4.3. The effects of steam quality

Figure 8 presents the total weight vs. efficiency of the combined cycle curves for several values of the steam turbine exhaust quality x_8 . The base case considered before ($x_8 = 0.95$) is included in Fig. 8 (identified as a bold line) together with three more values of $x_8 = 0.90, 1.00$, and 1.02 . All other parameters used for the calculation of these curves remain fixed at the values indicated in Tables 1 and 2. As can be appreciated, for low heat recovery configurations variable x_8 doesn't have any influence on the efficiency or weight of the combined cycle. Just for configurations having efficiency larger than around 37 % the difference starts being noticed, with the base case always producing a lower weight and, at the same time, having the maximum possible efficiency. The minimum weight configuration for the cases considered has always values around 123 t/d (the actual difference between the largest and the smallest minimum weight is just 1 t/d) for efficiency values around 40 % (all minimum weight points are inside a 1 % narrow region). Regarding the maximum efficiency configurations, there is a small difference between the worst (41.6 % for $x_8 = 1.02$), and the best cases (42.4 % for $x_8 = 0.95$ and 0.90). In summary, variable x_8 doesn't play a very important role in the increasing of efficiency or the decreasing of weight for the combined cycle considered.

Figure 8. Combined cycle total weight per unit time at sea vs. efficiency for different steam turbine exhaust qualities.

4.4. The effects of HRSG pinch point

Following the same idea of tracking the influence of a particular parameter on the overall behavior of the combined cycle power plant, Fig. 9 presents the total weight vs. efficiency curves for several values of HRSG pinch point $\Delta T_{\min}$. The base case considered before ($\Delta T_{\min} = 20$ °C) can be identified again in Fig. 9 as a bold line next to other three cases with values of $\Delta T_{\min} = 5$, 10, and 40 °C. Once again, all other parameters used for the calculation of these curves remain fixed at the values indicated in Tables 1 and 2. Similar to the situation with changing x_8 , for low heat recovery, variable $\Delta T_{\min}$ doesn't have an important influence on the efficiency or weight of the combined cycle. For values of efficiency lower than 41 %, the difference on weight between the best ($\Delta T_{\min} = 40$ °C) and the worst case ($\Delta T_{\min} = 5$ °C) is just around 1 t/d. A significant dependence on $\Delta T_{\min}$ can be appreciated, however, on the maximum efficiency of every case as this increases with lower pinch point, taking values of 41.3, 42.4, 42.9, and 43.2 % for $\Delta T_{\min} = 40, 20, 10$, and 5 °C respectively. Interestingly, the locus of optimal points joining minimum

weight and maximum efficiency configurations for all the considered cases, seems to form a single universal line that becomes steeper for the larger efficiencies attainable with smaller $\Delta T_{\min}$.

In conclusion, it could be said that the general weight vs. efficiency relationship is basically independent on the pinch point for the set of cases analyzed, with the pinch point just determining the extent of the particular minimum weight to maximum efficiency section of the curve. This result is very important as it allows some freedom in the choice of the desired weight-efficiency combination in the power plant design: selecting a different value for $\Delta T_{\min}$ will not affect substantially the selected values for those objectives, even when the general configuration of the plant will experience an important change. For example, the design objectives $M_{\text{TOTAL}} = 380.2 \text{ t}$ and $\eta_{\text{CC}} = 42.4 \%$ found for the base-case maximum efficiency configuration discussed before, with $P_{\text{HRSG}} = 3.937 \text{ bar}$ and $T_7 = 480 \text{ }^\circ\text{C}$ for $\Delta T_{\min} = 20 \text{ }^\circ\text{C}$, could be nearly obtained with a quite different thermodynamic configuration of $P_{\text{HRSG}} = 4.492 \text{ bar}$ and $T_7 = 495 \text{ }^\circ\text{C}$ for $\Delta T_{\min} = 5 \text{ }^\circ\text{C}$, as can be seen from the superposing of points in the corresponding curves of Fig. 9.

Figure 9. Combined cycle total weight per unit time at sea vs. efficiency for different HRSG pinch points.

4.5. The effects of gas turbine performance

A final important analysis to be made in this work is about the influence of the gas turbine operation parameters on the combined cycle design. Figure 10 shows the total weight vs. combined cycle efficiency curves for several combinations of gas turbine efficiency η_{GT} and exhaust gas temperature T_4 . The base case considered before ($\eta_{GT} = 30\%$ and $T_4 = 500^\circ\text{C}$) can be identified again in Fig. 10 as a bold line with no markers next to other eight cases with values of $\eta_{GT} = 25, 30,$ and 35% , and $T_4 = 450, 500,$ and 550°C . All other parameters used for the calculation of these curves remain fixed at the values indicated in Tables 1 and 2. As appreciated in Fig. 10, for a given efficiency, total weight decreases with larger T_4 and, much more substantially, with larger η_{GT} . As natural, the simple cycle configurations take efficiency values of 25, 30, or 35 % depending on the corresponding efficiency of the gas turbine. Regarding the maximum combined cycle efficiency for each case, this increases with both increasing η_{GT} and T_4 , from 36.6 % for $\eta_{GT} = 25\%$ and $T_4 = 450^\circ\text{C}$ to 47.9 % for $\eta_{GT} = 35\%$ and $T_4 = 550^\circ\text{C}$. Analogously, the minimum weight for each case decreases with both increasing η_{GT} and T_4 , from 145.7 t/d for $\eta_{GT} = 25\%$ and $T_4 = 450^\circ\text{C}$ to 106.5 t/d for $\eta_{GT} = 35\%$ and $T_4 = 550^\circ\text{C}$. However, if expressed as percentage of the corresponding simple cycle weight, the minimum weight configuration represents a reduction that increases with T_4 and decreases with η_{GT} , from 23.9 % for the case $\eta_{GT} = 25\%$ and $T_4 = 550^\circ\text{C}$ to 9.8 % for $\eta_{GT} = 35\%$ and $T_4 = 450^\circ\text{C}$.

Figure 10. Combined cycle total weight per unit time at sea vs. efficiency for different gas turbine operation parameters.

Summarizing, the gas turbine operation parameters have a strong influence on both the total weight associated to and the efficiency of the combined cycle. Specifically, both a larger efficiency of the gas turbine and a higher exhaust gas temperature contribute to improve those two indicators. Importantly, even when η_{GT} seems to play a more important role than T_4 in the improvement of the combined cycle design for the range of parameters analyzed, is the combined effect of both parameters what determines the overall performance of the system. According to this, a less efficient gas turbine could produce better results than a more efficient one, if the corresponding exhaust gas temperature is a high enough. The type of analysis included here could be used in the determination of objective design conditions for gas turbines, so that, when used in a combined cycle power plant for ship applications, produce the best possible results.

5. Conclusions

In this paper, a model for the analysis of combined cycles for shipboard applications has been developed, paying simultaneous attention to the thermal performance and the weight aspects of the power plant. To the authors' knowledge, this type of analysis has not been published before. In the model, a set of key input parameters has been defined (summarized in Tables 1 and 2), which define series of configurations with distinct degrees of waste heat recovery.

For fixed values of the input parameters, the combined cycle efficiency and weight are functions of the heat recovery experienced in the HRSG. Starting from the simple cycle configuration, the efficiency increases with a larger heat recovery, reaches a maximum value, and then decreases as a consequence of a rapidly decreasing efficiency of the Rankine cycle for excessive heat recovery. Similarly, the total weight associated to the power plant decreases at first, reaches a minimum value, and then increases as a consequence of the extra weight required for the HRSG. Importantly, the minimum weight configuration does not coincide with the maximum efficiency one. Beyond the minimum weight configuration, the equipment weight rise required to increase heat recovery will be larger than the fuel weight reduction produced by the efficiency gain.

The combined behavior for efficiency and weight of the combined cycle allows the distinction of three different regions of low, medium, and high heat recovery. For the low and high heat recovery regions (extending from simple cycle to the minimum weight configuration and from maximum efficiency to maximum heat recovery respectively), a larger efficiency of the combined cycle also means a shorter total weight. For the medium heat recovery region instead (running from the minimum weight to the maximum efficiency configurations), those objectives

are in competition as any efficiency increase implies an increase on the total weight of the combined cycle. The particular choice of the final design, lying in this medium heat recovery region, will depend on the relative importance of efficiency and weight for the considered system.

Complementary to the analysis of heat recovery, the influence of other parameters on the overall behavior of the cycle has been analyzed. Trip duration has been determined as an important factor on total weight and efficiency for minimum weight. For short trip periods (shorter than 5 d) the medium heat recovery region is considerable and the determination of the optimal configuration based on the competence of the two objectives is important. For long trip periods (e.g. 10 d) minimum weight and maximum efficiency configurations are basically the same. The pinch point has been determined important in defining the extent of the medium heat recovery region and the thermodynamic configuration of the Rankine cycle, with lower values allowing larger maximum efficiencies. The gas turbine operation parameters have been observed as having a strong influence on both total weight and efficiency of the combined cycle, with larger gas turbine efficiencies and higher exhaust gas temperatures contributing to improve both. Interestingly, an indirect consequence of this general behavior is that a less efficient gas turbine could produce better results than a more efficient one, if the corresponding exhaust gas temperature is high enough.

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