

High Efficiency Waste Heat Recovery from Dual Fuel Marine Engines

M. ALTOSOLE^a, U. CAMPORA^{b1}, M. LAVIOLA^a and R. ZACCONE^a

^a*Department of Electrical, Electronic, Telecommunications Engineering and Naval Architecture (DITEN), University of Genova, Via all'Opera Pia 11A, I-16145 Genova, Italy.*

^b*Department of Mechanical, Energy, Management and Transportation Engineering (DIME), University of Genova, Via Montallegro 1, I-16145 Genova, Italy.*

Abstract. The paper presents an original variable layout steam plant for the Waste Heat Recovery (WHR-VL) from marine dual fuel (DF) engines, designed considering the different exhaust gas stack temperature limits, depending on the engine fuel type: generally no less than 160 °C for diesel oil and without limits for natural gas (NG). Using diesel oil, a single pressure superheated steam plant configuration is adopted, while a dual pressure system is proposed for the NG fuel mode of the engine. In both WHR-VL plant layouts, the produced steam is mainly delivered to a steam turbine for the ship electric energy production, while a smaller amount meets the ship thermal loads. Starting from data of a marine four stroke DF engine, the WHR-VL steam plant components are sized and optimized through a mathematical code, according to a numerical procedure described in the paper. The WHR-VL plant data and performance, for different engine fuel modes and loads, are compared with those of a more traditional WHR single pressure steam plant, developed and optimized for the waste heat recovery of the same dual fuel engine.

Keywords. Dual fuel engine, waste heat recovery, natural gas fuel, ship power plant efficiency

1. Introduction

The International Maritime Organization (IMO) is introducing more and more stringent regulations regarding sulphur oxides (SO_x) and nitrogen oxides (NO_x), emitted from marine engines [1, 2]. These two pollutant substances are subject to strict limits in the Emission Control Areas (ECA), currently located in North America east and west coasts, European North Sea, Baltic Sea and English Channel, but soon extended to further areas [3], including Mediterranean sea. For new ships, the theme of the carbon dioxide (CO₂) emissions abatement is also considered with increasing attention to reduce the greenhouse effect on the planet [4].

A CO₂ emissions reduction can be obtained adopting low carbon content fuels, as natural gas (NG). The engines fed by NG fuel strongly reduce the typical HFO pollutant emissions: -25÷30% CO₂, -25% CO, -85% NO_x, -98÷100% SO_x, -90÷99% particulate matter [5].

¹ Corresponding Author, Ugo Campora, Department of Mechanical, Energy, Management and Transportation Engineering (DIME), University of Genova, Polytechnic School, Via Montallegro 1, I-16145 Genova, Italy; E-mail: campora@unige.it.

A marine pollutants abatement could be also achieved by using more efficient engines, better if combined with Waste Heat Recovery (WHR) steam plants to produce electric energy through a steam turbine and satisfy the ship thermal users. Thus the on board diesel generators power and emissions can be reduced. In this regard, several marine WHR steam plants schemes are proposed in literature [6-14]. Two basic configurations are used: single or double steam pressure. A further emissions reduction can be achieved combining dual fuel (DF) engines with WHR steam plants. As authors' knowledge, very few papers [15,16] refer to marine DF engines combined with WHR steam plants.

The sulphur content in diesel engine fuels requires a Heat Recovery Steam Generator (HRSG) outlet gas temperature that is not less than about 160 °C; on the contrary this gas temperature limit cannot be applied to engines fed by NG. This allows a (potential) greater HRSG efficiency, and thus a greater efficiency for NG engines–WHR steam plants in comparison with the diesel engines–WHR systems. In marine DF engines waste heat recovery, the same WHR plant is used in both the fuel modes, while the engine exhaust gas mass flow and temperature are different for the same engine load and speed. In accordance with these considerations, the authors decided to propose a WHR plant with a variable steam plant layout (WHR–VL), in order to satisfy the HRSG exhaust gas temperature limit if the engine is fed by heavy fuel oil (HFO), and to reduce this temperature to a minimum possible value in the engine gas mode (to obtain a more efficient WHR plant when the NG fuel is used). DF engines are usually fed by NG (diesel oil is normally used just in particular conditions), therefore in this work the HRSG heat exchangers are designed and optimized for the NG fuel mode of the engine.

In the paper, Rankine cycles and DF engine–WHR–VL combined plant (DF–WHR–VL CP) efficiency values are compared for each engine fuel type (diesel oil or NG). The HRSG size and weight are also reported. The WHR–VL steam plant is compared in terms of performance with a simpler WHR single pressure system [16], characterized by the same DF engine. The examined engine loads are comprised between 50% and 100 % of the Maximum Continuous Rating (MCR), in both fuel modes.

2. WHR steam plant modeling and optimization

A WHR single pressure saturated steam plant of a DF four stroke marine engine, whose scheme is shown in Figure 1, was already presented in a previous paper [16]. This WHR steam plant is basically composed by a HRSG (composed by an economizer (E) and an evaporator (EV)) that feeds a steam turbine (ST) to drive an electric generator (EG). The turbine exhaust steam is extracted by a condensing pump (SCP) from the condenser (SCO), and preheated in the Jacket Water (JW). Then it is delivered to the Heat Water Tank (HWT), from which it is moved by the main feed pump (MFP) to the HRSG economizer (E). Meanwhile the water is warmed in the engine scavenger (SC) by the turbocharger compressor outlet hot air. To satisfy the ship steam services, a saturated steam part is taken from the HRSG steam drum (SD), as illustrated in Figure 1.

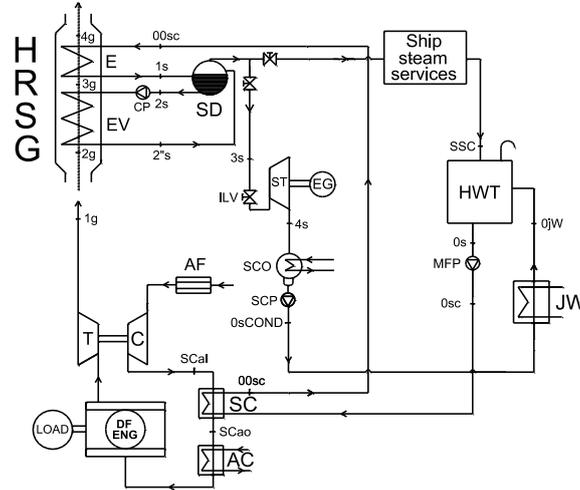


Figure 1. Marine engine WHR saturated steam plant layout

Starting from the DF engine exhaust gas temperature and mass flow rate, the main data and performance of the WHR plant Rankine cycle and components (HRSG economizer, evaporator, steam drum, steam turbine, heat water tank, etc.) are determined and optimized by a mathematical code developed in MATLAB[®] language, as already applied in [16]. The numerical procedure is hereinafter summarized, while more details are reported in [11]. The WHR steam plant design procedure is based on the steady state continuity and energy equations:

$$\sum (M_i - M_o) = 0 \quad (1)$$

$$\sum (M_i h_i - M_o h_o) = Q' - P \quad (2)$$

where M_i and M_o are the fluid mass flow rate in the plant components inlet and outlet sections respectively, h is the fluid specific enthalpy, Q' and P are respectively the heat and mechanical power exchanged with the outside.

The equation for the computation of the HRSG finned pipes wall overall heat exchange coefficient (k_e) is:

$$\frac{1}{k_e} = \frac{1}{h_e} + R_e + \frac{s}{k} \frac{A_e}{A_{ml}} + R_i \frac{A_e}{A_i} + \frac{1}{h_i} \frac{A_e}{A_i} \quad (3)$$

In the equation, h_e and h_i are the external and internal pipes convective heat transfer coefficients, R_e and R_i the external and internal pipes thermal resistances, s and k the pipes wall thickness and the wall thermal conductivity, A_e , A_i and A_{ml} the pipes wall external, internal and logarithm areas respectively. From Eq. (3), the HRSG components (economizer, evaporator and superheater) design can be carried out through the finned tube correlations [11]. The pipes heat exchange area is evaluated by solving the gas (_g) wall (_w) heat exchange balance Eq. (4) and the wall steam (_s) heat exchange Eq. (5):

$$Q'_g = k_g A_e (\Delta T)_{g-w} \quad (4)$$

$$Q'_s = k_s A_i (\Delta T)_{w-s} \quad (5)$$

In Eqs. (4) and (5), ΔT values are the fluid-pipe wall logarithmic temperature differences [11].

A similar procedure is applied to the others steam plant heat exchangers design (SCO, JW and SC). The steam turbine is modelled by the typical non dimensional performance map [17] while the HRSG steam drum (SD) and the heat water tank (HWT) are modelled as simple fluids mixing tanks.

In order to assess the WHR steam plant performance, in both design and off-design steady state working conditions, a second simulator has been developed. The previous WHR steam plant design procedure and the off-design performance code are validated in other works of the authors [11,12], by comparing the calculation results with data of similar existing systems. The difference between calculated and reference parameters is less than 1% in design conditions, while it is less than 4% in off-design operations.

3. WHR variable layout steam plant

The authors' proposal for a new WHR-VL steam plant layout starts from the WHR single pressure configuration of Figure 1, that represents the high pressure part (superheater) of the dual pressure steam plant (black lines in Figure 2a). A low pressure saturated steam system is added (blue lines in Figure 2a). The WHR-VL plant part is basically composed by: a low pressure economizer (E lp in Figure 2a), an evaporator (EV lp) and a steam drum (SD lp). When the engine burns NG fuel, in addition to the WHR-VL high pressure steam plant, the low pressure section is also activated. In fact the valve V1 is opened to take cold water through the steam condensing pump (SCP), and the gas deviator bulkhead (GD) allows to send the gas into the low pressure HRSG evaporator (EV_{LP}) and economizer (E_{LP}). By this solution, a greater cooling of the exhaust gas is possible.

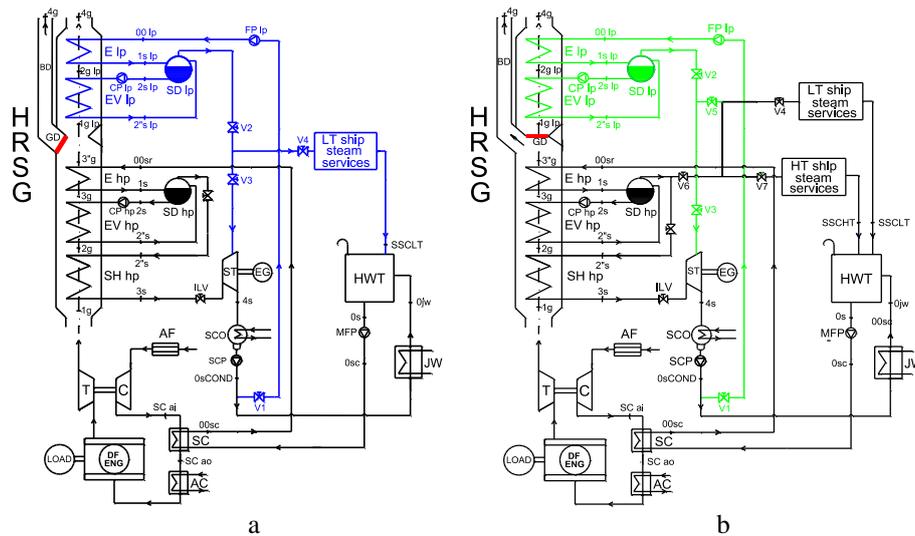


Figure 2. WHR-VL steam plant configurations in NG mode (a) and HFO mode (b)

In the HFO mode of the engine, only the WHR–VL high pressure steam plant (black lines in Figure 2b) is working, and the low pressure HRSG section (green lines in Figure 2b) is disabled by opening the gas deviator bulkhead (GD); in such a way, the exhaust gas exiting from the high pressure economizer (E hp) is sent to the bypass duct (BD), and the valves V1, V3 and V5 are closed. In this configuration the WHR–VL is a single pressure superheated steam plant, where the high pressure economizer (E hp) water is preheated by the hot air of the engine scavenger (SC).

4. Case study

For the performance comparison between the WHR–VL and the single pressure WHR systems, a MAN 51/60 18V dual fuel four stroke tier III engine, characterized by 17550 KW for the maximum continuous rating (MCR) power at 500 rpm, is considered.

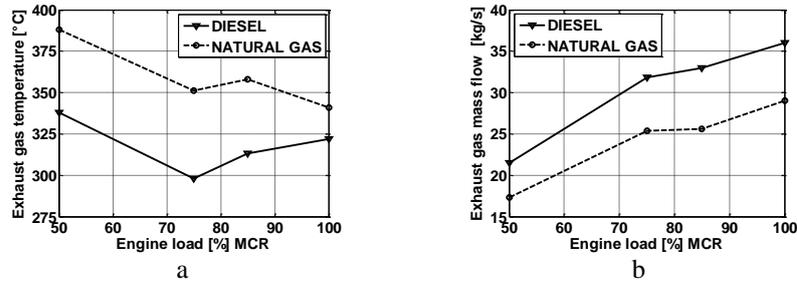


Figure 3. DF engine exhaust gas characteristics depending on engine load

In reference to the characteristics of a ferry equipped with this engine type [16], the ship services thermal power is assumed to be satisfied by 0.57 kg/s of saturated steam at least 7 bar in HFO mode, and 0.17 kg/s at least 2 bar when the fuel is NG. The data of Figures 3a and b, from the engine project guide [18], show remarkable gas temperature and mass flow rate differences between the two possible engine fuel types.

4.1. WHR–VR steam plant optimization procedure

The design and performance optimization of the DF engine WHR–VL steam plant are carried out similarly to those of the WHR system of Figure 1 and described in [11,16]. Considering that DF engines usually work in NG fuel mode, the DF–WHR–VL combined plant performance is optimized for NG application. Taking into account the Normal Condition Rating (NCR, 75% MCR power) of the engine, the Rankine cycle parameters (i.e.: pressures, temperatures, mass flow rate) and WHR plant components are optimized to maximize the combined plant efficiency (η_{CP}):

$$\eta_{CP} = \frac{P_{DFE} + P_{ST} - P_{MFP} - P_{SCP}}{M_f LHV_f} \quad (6)$$

where: P_{DFE} and P_{ST} are the DF engine and the steam turbine power, P_{MFP} and P_{SCP} the power values of the main feed and steam condenser pumps, M_f and LHV_f the fuel mass flow and its lower heating value.

The WHR–VL steam plant optimization procedure starts from the definition of a “fixed parameters” series: engine exhaust gas mass flow rate (26 kg/s) and temperature (352°C), SD low pressure (3 bar) and condenser pressure (0.06 bar), HRSG dimensions (2.40 m x 2.85 m). By numerically changing some “HRSG optimization parameters” to maximise the η_{CP} value, the HRSG main characteristics are found. The examined parameters for the optimization process are the pressure of the high pressure steam drum (variable range: 5÷30 bar), and the low and high pressure pinch point temperature differences (variable range: 5÷20°C). The second column of Table 3 shows the main results of the optimization for the gas engine WHR–VL configuration (Figure 2a).

5. WHR–VL and WHR combined plants comparison

Table 3 shows the WHR–VL plant performance compared with the WHR single pressure configuration. Both WHR systems are optimized for the waste heat recovery of the same DF marine engine, running at NCR load in NG fuel mode. However the comparison is carried out for both NG and HFO modes.

Table 3. WHR-VL and WHR combined plants design data and performance comparison

WHR-VL and WHR steam plants parameters	WHR-VL NG	WHR NG	WHR-VL HFO	WHR HFO
plant scheme	Figure 2a	Figure 1	Figure 2b	Figure 1
WHR plants optimization parameters				
HRSG _{HP} steam drum pressure [bar]	26.4	7.9	12.9	8.4
low press. pinch point temp. diff.: ΔT_{ppLP} [°C]	12.6	–	–	–
high press. pinch point temp. diff.: ΔT_{ppHP} [°C]	5.2	5.1	18.2	7.0
HRSG characteristics				
HRSG _{eff} [%]	69.1	58.3	36.0	46.0
HRSG _{LP} steam drum pressure [bar]	3	–	–	–
high press. approach point temp. diff.: ΔT_{ppAP} [°C]	25.8	–	14.6	–
HRSG outlet temperature [°C]	124	178	200	171
HRSG height [m]	6.9	4.3	6.9	4.3
HRSG weight [t]	68.5	28.6	68.5	28.6
Engine and WHR-VL steam plant characteristics				
η_E [%]	47.4	47.3	45.8	45.6
$\eta_{Rankine}$ [%]	28.4	26.4	30.0	27.1
η_{CP} [%]	52.1	50.2	47.5	47.7
engine power (75% MCR) [KW]	13162.5	13162.5	13162.5	13162.5
ST power [kW]	1285	851	510	637
ST efficiency [%]	73.5	71.1	74.0	70.8
steam turbine low press. mass flow rate [kg/s]	0.81	–	–	–
steam turbine high press. mass flow rate [kg/s]	1.38	3.52	1.37	2.61
ship steam service mass flow rate [kg/s]	0.17	0.17	0.57	0.57

When the engine is fed by NG, the second and third columns of Table 3 show a greater WHR–VL high pressure SD pressure and a very similar pinch point temperature difference (ΔT_{ppHP}). Due to the dual pressure configuration, the WHR–VL steam plant strongly reduces the stack gas temperature (HRSG outlet temperature in Table 3), and increases the HRSG efficiency of nearly 11 percentage points. This better result is obtained against superior HRSG height and weight. The larger ST power of the WHR–VL plant (about 0.4 MW more powerful in comparison with the ST of the WHR single pressure system) allows a 2% greater combined plant efficiency (η_{CP}).

The DF–WHR–VL combined plant performance comparison between NG and HFO modes, shows that the HRSG high pressure steam drum (SD hp in Figure 2a) is about twice when the engine is fed by NG fuel. The same table shows a HRSG high pressure pinch point temperature difference (ΔT_{ppHP}) higher in the WHR–VL single pressure plant (Figure 2b, HFO fuel). These facts penalize the WHR–VL single pressure plant performance compared to the dual pressure WHR–VL one, in fact the HRSG single pressure stack gas temperature is much higher (200°C against 124°C). The higher WHR–VL dual pressure plant ST power, in comparison with the single pressure WHR–VL one, involve a greater combine plant efficiency. In the HFO mode, the WHR–VL and WHR single pressure systems comparison shows a smaller difference in ST power and combined cycle efficiency (η_{CP}). The comparison between the WHR–VL steam plant and the WHR single pressure one is also presented for different loads of the engine: 50%, 75%, 85% and 100% MCR [18].

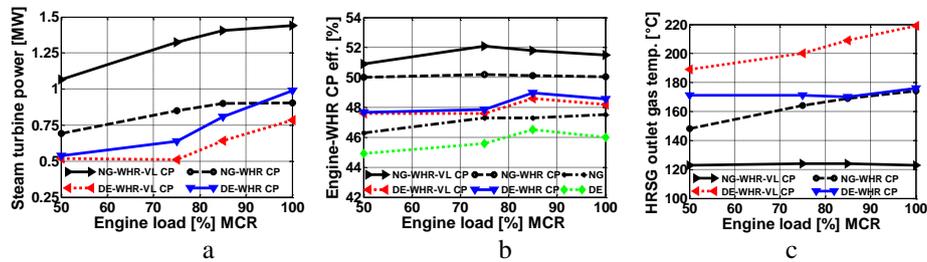


Figure 4. WHR plants parameters for different engine loads

Figure 4a shows that the WHR–VL plant steam turbine power is much greater in the NG fuel mode of the engine. This is true also for the WHR single pressure plant, but with a lower power difference.

Figure 4b reports the efficiency advantage of the WHR–VL system in NG fuel mode, in comparison with the single pressure WHR plant. The same figure shows almost the same efficiency behaviour when the engine is fed by HFO. For NG fuel application, the HRSG gas outlet temperature is the lowest for the WHR–VL plant and quite constant for all engine loads (Figure 4c), while in HFO mode, the effect of the low pressure section deactivation is evident. Minor HRSG outlet gas temperature differences between NG and HFO modes are observed in the WHR plant, due to its single configuration.

6. Conclusions

An original WHR variable layout steam plant for marine DF engines is presented. The main goal is to improve the waste heat recovery for NG fuel applications by reducing the HRSG outlet gas temperature value recommended by diesel engines manufacturers (160 °C), but at the same time respecting this temperature limit in diesel oil working conditions. By a numerical code, the proposed solution is compared with a conventional engines WHR single steam pressure system, equipped with the same DF engine. In NG fuel mode at the NCR running condition of the engine, the efficiency of the variable configuration is 10% greater than the single engine efficiency, and around 4% higher in comparison with the adoption of a WHR single pressure plant. On the

contrary, when the engine burns HFO, the two examined systems show the same combined plant efficiency. A similar behaviour is verified for the other engine loads considered in this study (50%, 75%, 85% and 100% MCR).

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