ADVANCED EXERGY ANALYSIS OF A MARINE DIESEL ENGINE WASTE HEAT RECOVERY SYSTEM

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ABSTRACT

Carbon emissions, as one of the important GHG sources, are responsible for the damages on the nature by various ways such as air pollution, climate change, health issues etc. Hence, the number of studies aiming the reduction of greenhouse emissions in industrial activities has gained momentum in recent years.

The studies have revealed that transport has a high share of GHGs production. Since most of the transportation of commercial goods are being shipped by sea trade. Marine engines, mostly marine diesel engines have been subject to ever more tight measures for efficient operation. As the world ship fleet continues to grow, the ways to reduce the carbon footprint of the sector as well as to increase fuel economy are being thoroughly researched.

In this paper, a review of some methods and applications of reducing carbon emissions of ships through energy and exergy efficiency improvement is presented. The waste heat recovery method, by adding an economizer at the exhaust of marine engine, in combination with a steam cycle is considered. An energy and exergy analysis has been carried out to identify the key parameters, which affect the efficient operation of the system. Then, a parametric study has been conducted to determine the optimum range of operation for the engine and waste heat recovery system considering different load conditions. Finally, exergy destruction of each component is calculated to give further insight information about interactions of all system components and the potential of improvement for the efficient operation.

Keywords: Marine Engines, Combined Cycle, Rankine Cycle, Advanced Exergy Analysis

1. INTRODUCTION

Greenhouse gas emissions, GHGs, cause hazardous problems such as pollution driven health problems and climate change. The latter affects human, nature and other species in more harmful ways like melting arctic ices, shifting seasons, changing acidities in oceans, differing atmospheric conditions and regimes etc (Pachauri, Allen et al. 2014). As a well-known issue, GHG emissions, mostly anthropogenic CO_2 with around %80 of total, continue to ascend under the influence of economic, industrial and population growth and have accelerated more and changed climate in 20th century (figure 1) (Pachauri, Allen et al. 2014).

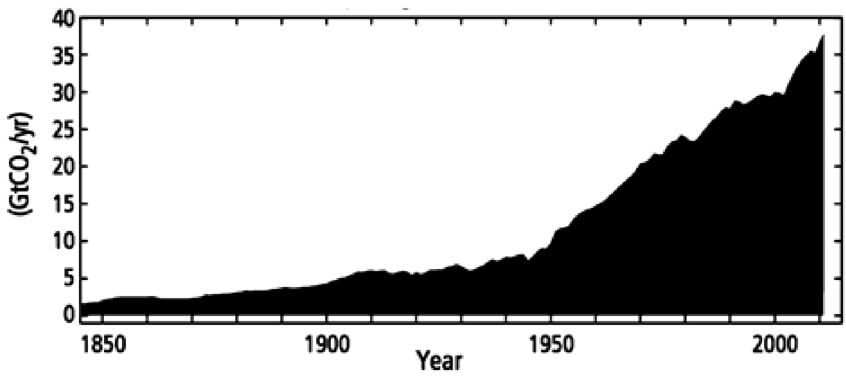

Figure 1: Global Anthropogenic CO2 emissions (Pachauri, Allen et al. 2014)

An outlook on GHG emitters reveals that transport has 14% share among all economic sectors (Pachauri, Allen et al. 2014). Seaborne trade has an average share of 3.1% GHG emitting (Smith, Jalkanen et al. 2014), thus, 22.14%, almost a quarter, of transport sector's GHG share belongs to shipping. Parallel to the economic growth, world fleet and seaborne shipments expand and world fleet has reached 1.69 billion DWT in January 2014 (Secretariat 2014). It is obvious that total shipping increase will raise its effect on climate change and growth of shipping may lead CO2 emissions to grow 2 to 3 times more by 2050 if no actions are taken (Buhaug, Corbett et al. 2009). To keep nature in a peaceful condition controlling and reducing GHGs emissions are obligatory.

To fulfil the need of GHG reduction some mandatory or voluntary rules and regulations have been put into action such as Energy Efficiency Design Index (EEDI) for new ships, Ship Energy Efficiency Management Plan (SEEMP) for all ships and Energy Efficiency Operation Indicator (EEOI) (Buhaug, Corbett et al. 2009, Bazari and Longva 2011). These legal actions and needs for fuel economy lead changes in operational conditions, hull design, engine improvements, and propeller selections. DNV has carried out a study that expresses CO2 reduction methods, how much reduction could be made (width of bar) and how much it would cost (height of bar) shown in Figure 2 (Alvik, Eide et al. 2009).

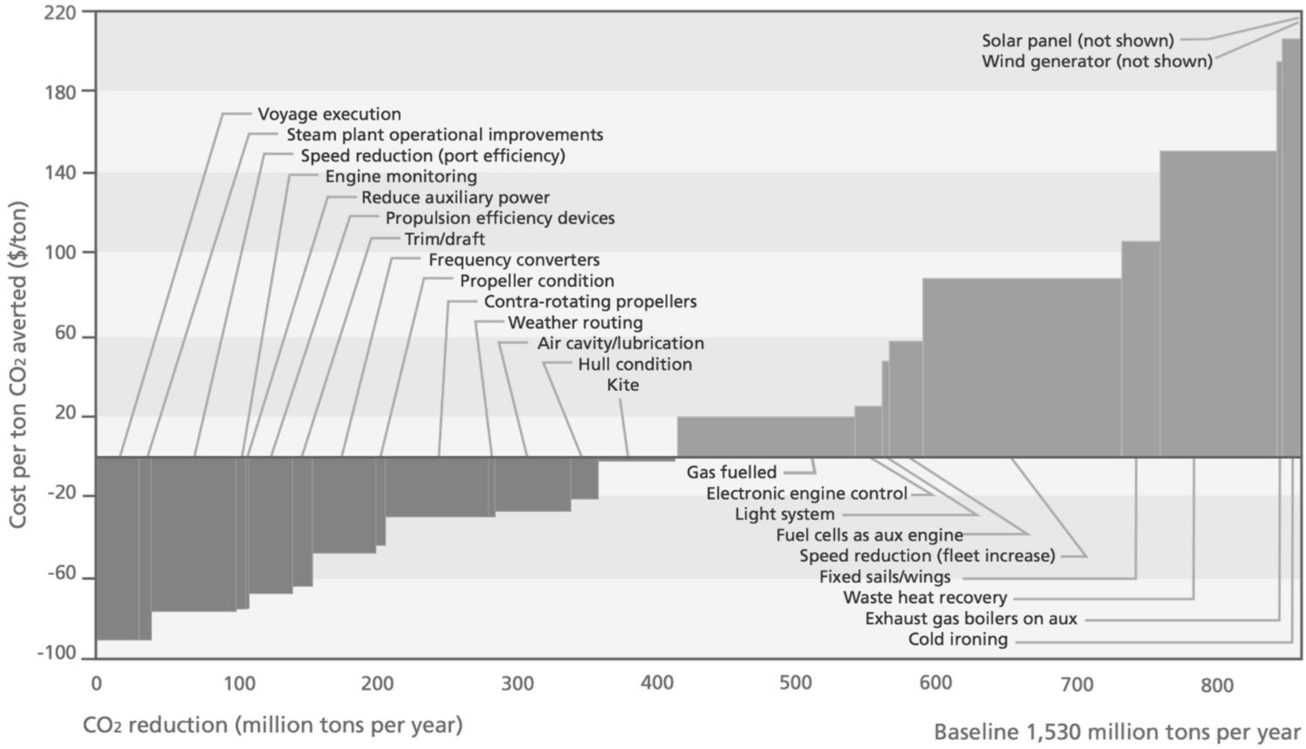

Figure 2: Average CO2 reduction cost per option (Alvik, Eide et al. 2009)

Some examples of available emission reduction methods may be shown to point out percentage of CO2 reduction potentials such as slow steaming 30%, hull coating 10%, propeller technology 5-10%, waste heat recovery 10% (Harrould-Kolieb and Savitz 2010).

Considering that 85% of world fleet have two-stroke marine diesel engine (Wärtsilä 2010), waste heat recovery (WHR) seems to be a good option to improve energy efficiency and reduce both CO2, other GHG emissions and fuel consumptions which are linked to each other. WHR could be defined as utilization of dissipated heat that is by-product of an energy conversion system. Therefore, it could be applied to exhaust systems, jacket water cooler and air intercooler as the three main heat sources of a marine engine (Shu, Liang et al. 2013). It should be noted that greatest share of waste heat of a main engine is exhaust which is almost half of total waste heat and around quarter of total fuel energy (MAN-Diesel&Turbo 2014a). Hence, It is a common application to utilize exhaust waste heat, especially in turbocharger to pressurize and increase mass of intake air (Shu, Liang et al. 2013). WHR systems could supply the total electricity demand of a ship excessively in many cases (MAN-Diesel&Turbo 2014b). It could be done with power turbine, thermoelectric generation, steam cycle, organic rankine cycle (ORC) (Shu, Liang et al. 2013). Power turbine is a gas turbine that is driven by excessive exhaust gas, which bypasses turbocharger and/or after turbocharger, to produce electricity or extra shaft power (MAN-Diesel&Turbo 2014a). Thermoelectric generation is converting heat directly to electricity, with heating the junction of two different conducting materials without a need of a heat engine (Sørensen 2004). Steam cycle is a power cycle and has water as working fluid that is alternatively vaporized and condensed based on rankine cycle (Cengel and Boles 2015). ORC is basically a rankine cycle, but instead of having water-steam as working fluid, it uses organic fluids and a useful option to produce power from low grade hat sources (Tumen Ozdil, Segmen et al. 2015). Some works that have been made on marine engine waste heat recovery systems, could be found in the review of Shu et al (Shu, Liang et al. 2013).

Exergy analysis, which is based on the first and the second law of thermodynamics (Bejan, Moran et al. 1996), is already applied almost all kind of energy systems (engines, WHRs, power plants etc.)(Dincer and Rosen 2013), however recently gained some importance and applied on ships and marine engines. It has been applied to a chemical tanker (Baldi, Johnson et al. 2014), a cruise ship (Baldi, Ahlgren et al. 2015), a generic ship (Marty, Hétet et al. 2015), and mostly on waste heat recovery systems (Yang and Yeh 2014, Song, Song et al. 2015).

Advanced exergy analysis is based on conventional exergy analysis and the difference starts with splitting exergy destruction of investigated component into *avoidable/unavoidable* and *endogenous/exogenous* exergy destruction parts (Morosuk and Tsatsaronis 2008). As far as is known, there are no applications that have been carried out on ships and marine energy systems yet.

In this paper, exergy and advanced exergy analyses of a designed single pressure steam cycle after a typical five-cylinder two-stroke low speed marine engine have been carried out. Fully loaded ship simulator data is used to perform analyses. Also, parametric analyses have been made due to lever position of main engine and upper pressure of steam cycle. Lastly, a new exergy efficiency description is introduced based on advanced exergy analysis and applied to the analyzed system.

2. METHOD

Exergy is the maximum useful work potential of a system that reversibly goes from a specified state to its environmental state (Cengel and Boles 2015). Exergy and a stream *j* could be formulated as below respectively while potential and kinetical exergies are neglected (Bejan, Moran et al. 1996):

$$\dot{Ex}_j \cong \dot{m}_j\left[\left(h_j - h_0\right) - T_0\left(s_j - s_0\right)\right] \tag{1}$$

Exergy balance for investigated *k*th component, general exergy efficiency and exergy destruction ratio could be written respectively as (Bejan, Moran et al. 1996):

$$\dot{Ex}_{D,k} = \dot{Ex}_{F,k} - \dot{Ex}_{P,k} \tag{2}$$

$$\varepsilon_k = \frac{\dot{Ex}_{P,k}}{\dot{Ex}_{F,k}} \tag{3}$$

$$y^*_{D,k} = \frac{\dot{Ex}_{D,k}}{\dot{Ex}_{D,tot}} \tag{4}$$

where $\dot{Ex}_{F,k}$ exergy rate of fuel, $\dot{Ex}_{P,k}$ exergy rate of product and $\dot{Ex}_{D,k}$ exergy destruction rate of the *k*th component; ε exergy efficiency, y^*_D share of exergy destruction of *k*th component amongst others; $\dot{Ex}_{D,tot}$ total exergy destruction of the overall system (Bejan, Moran et al. 1996). Exergy analysis could be used to determine sources, location and the magnitude of irreversibilities and to compare various systems (Morosuk and Tsatsaronis 2008). However, exergy analysis might not be adequate to reveal real improvement potential of investigated component, sources of exergy destructions, or interactions among other components, structure and simultaneous actions-reactions of overall system (Morosuk and Tsatsaronis 2009, Yang, Wang et al. 2013). Therefore, so-called advanced exergy analysis has been introduced to cover the gaps that left open by exergy analysis.

Advanced exergy analysis is a method to deepen understanding on thermodynamic inefficiencies and ease system improvement by splitting exergy destruction of components (Morosuk and Tsatsaronis 2008). *Endogenous* exergy destruction is a part that irreversibilities occur only in the investigated component while all other components work in minimum or non exergy destructive way; *exogenous* exergy destruction of investigated component is actually an effect of system on investigated component due to irreversibilities of other components and simultaneous operation of overall system (Yang, Wang et al. 2013). *Unavoidable* exergy destruction is a part that cannot be decreased due to technological limits, availability and cost of materials or manufacturing process in a foreseeable future; *avoidable* exergy destruction is the part that could be avoided due to improvements and main focus of the designer (Morosuk and Tsatsaronis 2008). These exergy destruction parts could be combined as e*ndogenous avoidable* that could be avoided by improvements on component itself*, endogenous unavoidable, exogenous avoidable* that could be avoided by improvements on interactions among components and structure of overall system and lastly *exogenous unavoidable* exergy destructions.

To calculate splitted exergy destructions, new cycles should be created in accordance with definitions. All created cycles should produce overall system's total product exergy rate as constant. Endogenous exergy destruction of *k*th component, $\dot{E}x_{D,k}^{EN}$ is calculated as exergy destruction of *k*th component by creating hybrid cycles where *k*th component is in its own conditions and the others in a theoretical condition. Exogenous exergy destruction, $\dot{E}x_{D,k}^{EX}$ is the difference between real exergy destruction and endogenous exergy destruction (Morosuk and Tsatsaronis 2008):

$$\dot{E}x_{D,k} = \dot{E}x_{D,k}^{EN} + \dot{E}x_{D,k}^{EX} \tag{5}$$

Unavoidable exergy destruction of *k*th component, $\dot{E}x_{D,k}^{UN}$ is calculated as exergy destruction of the *k*th component in created unavoidable cycle where all components are in their unavoidable conditions. Avoidable exergy destruction of *k*th component, $\dot{E}x_{D,k}^{AV}$ is the difference between real exergy destruction and unavoidable exergy destruction (Morosuk and Tsatsaronis 2008).

$$\dot{E}x_{D,k} = \dot{E}x_{D,k}^{UN} + \dot{E}x_{D,k}^{AV} \tag{6}$$

Endogenous unavoidable exergy destruction of *k*th component, $\dot{E}x_{D,k}^{EN,UN}$ is calculated as exergy destruction of *k*th component by creating hybrid cycles where *k*th component is in its unavoidable conditions and the rest is in a theoretical condition(Morosuk and Tsatsaronis 2008). Endogenous avoidable $\dot{E}x_{D,k}^{EN,AV}$, exogenous unavoidable $\dot{E}x_{D,k}^{EX,UN}$ and exogenous avoidable $\dot{E}x_{D,k}^{EX,AV}$ exergy destructions could be calculated as differences shown respectively below (Yang, Wang et al. 2013):

$$\dot{E}x_{D,k}^{EN,AV} = \dot{E}x_{D,k}^{EN} + \dot{E}x_{D,k}^{EN,UN} \tag{7}$$

$$\dot{E}x_{D,k}^{EX,UN} = \dot{E}x_{D,k}^{UN} + \dot{E}x_{D,k}^{EN,UN} \tag{8}$$

$$\dot{E}x_{D,k}^{EX,AV} = \dot{E}x_{D,k}^{EX} + \dot{E}x_{D,k}^{EX,UN} \tag{9}$$

A new exergy efficiency is introduced due to the basis of advanced exergy analysis as:

$$\varepsilon_k^{+} = \frac{\dot{E}x_{P,k}}{\dot{E}x_{F,k} - \dot{E}x_{D,k}^{AV}} \tag{10}$$

The necessity for new defined exergy efficiency is to evaluate the effect of advanced exergy analysis on *k*th component, to see improvement potential directly. When a system or component is built, operated and evaluated, the expectation is to have same desired output at all times. Thus, in this new efficiency term, desired output, exergy rate of product, of *k*th component is kept constant while avoidable exergy destruction is affected required input, exergy rate of fuel directly by decreasing the need of exergy rate of fuel. So, it might be defined that the avoidable exergy destruction could be a sort of fuel saving potential of the *k*th component.

3. SYTEM MODEL DESCRIPTION

The system model depends on an open exhaust system and a single pressure closed steam cycle. Exhaust system has an economizer, evaporator with pump and drum and superheater in common usage with the steam cycle. Steam cycle includes feedwater pump, feedwater preheater, turbine and condenser. In summary, feedwater is pumped, heated in feedwater preheater by using the energy of main engine cooling water. Then, it enters to economizer and gets heated by exhaust gas until outlet is saturated water. Evaporator takes in the saturated water and turns into saturated steam. Superheater, generates superheated steam that comes as saturated from evaporator, later it expand in the turbine to condenser pressure is obtained. Working steam is condensed to saturated water at condenser pressure by seawater and cycle is completed. The system model is designed and run in Ebsilon Professional Software (STEAG 2015) and can be seen in Figure 3.

For analysis, a general cargo ship simulation with its cargo, speed, main engine RPM and power output, exhaust gas after turbocharger etc. information has been evaluated. Ship simulator general information that is used in this study could be found in table 1.

Table 1: General information of investigated ship and main engine

Ship		Main Engine	
Length overall	305 m	Cylinder bore	90 cm
Draught	19 m	Stroke	292 cm
Breadth	47 m	MCR Power	18 MW
Depth	30.4 m	MCR Engine Speed	74 RPM
Deadweight	178720 tons	Specific Fuel Consumption	168 g/kWh

Starting with navigational full ahead position of the lever, all available data has been taken for lever positions 100%, 75%, 50% and 25% from simulator. However, in this study, only given data in table 2 has been used that includes main engine effective power, ship speed, main engine jacket water (ME JW) condition, exhaust gas condition after turbocharger (TC) and main engine speed. Seawater inlet conditions is also taken from simulator and kept constant as 20 ℃ and 1.313 bar for all analyses.

Table 2: Simulator data used for this study

Ahead Lever (%)	Engine Speed (RPM)	Ship Speed (knts)	Mean Effective Power (MW)	TC Out Temp (C)	TC mass Flow (ton/h)	ME JW Out Temp (C)	ME JW Pres (bar)	ME JW Mass Flow (ton/h)
100	74	15,37	17,03	240,72	171,65	79,98	3,08	220,45
75	67,1	13,98	12,59	224,33	131,43	80	3,13	224,6
50	49,2	10,03	4,82	232,08	68,54	80,01	3,16	229,46
25	29,6	5,78	0,99	226,36	37,8	80	3,3	191,31

Great importance related to GHG emissions belongs to specific fuel consumption (SFC) and exhaust gas content of the investigated ship's main engine. Simulator directly gives NOx, SOx gases amount, however, an assumption, that exhaust gas only contains NOx, SOx and CO2, has been made to calculate amount of CO2 emission from the difference between SFC and other GHGs. Values of emissions could be found in table 3.

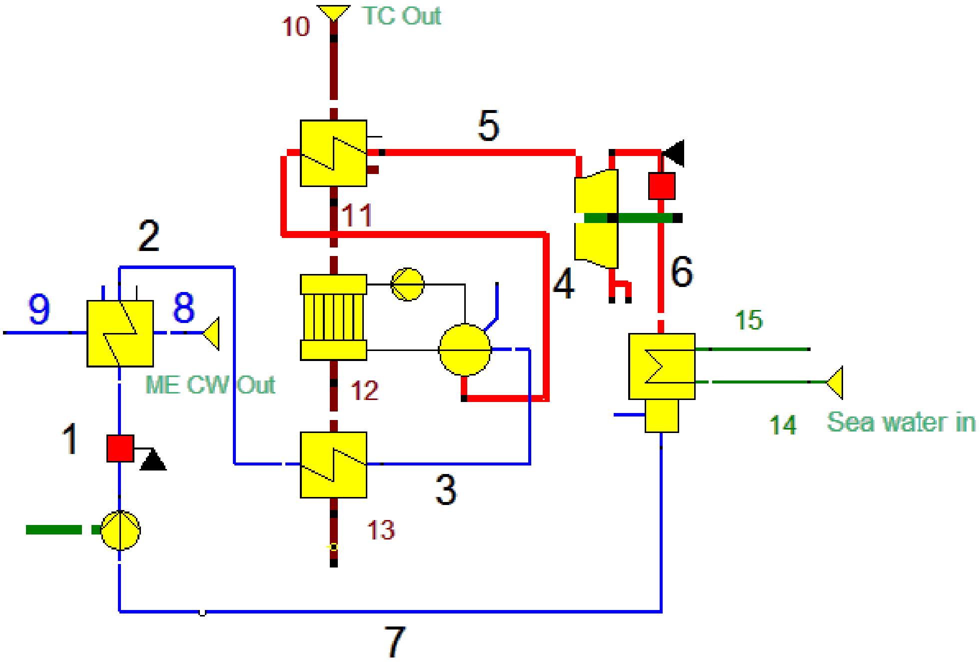

Figure 3: General schematic diagram of WHR system with single pressure closed steam cycle

Table 3: specific fuel consumption and exhaust gas content of the main engine

Ahead Lever (%)	SFC (g/kWh)	NOx(g/kWh)	SOx (g/kWh)	CO2 (g/kWh)
100	184,44	15,14	12,91	156,39
75	181,42	11,47	12,7	157,25
50	213,83	11,94	14,96	186,93
25	460,56	19,84	32,24	408,48

Lever position is also used as parameter for parametric study with feedwater pressure of closed steam cycle for 6, 7, 7.5, 8 bar. The reason to choose this four-pressure condition is that simulator has an exhaust boiler working at 7.5 bar. Therefore, optimum condition parametric study is held around this value. In table 4, properties of firstly designed real closed steam cycle for navigational full 100% ahead lever position could be found as an example.

Table 4: Properties of working fluids at 100% lever position 7.5 bar closed steam cycle

Pipe	Mass kg/s	Pressure bar	Temperature °C	Enthalpy kJ/kg	Entropy kJ/kgK
1	1,51456	7,5	41,5772	174,787	0,593125925
2	1,51456	7,3	71,9800	301,880	0,978667492
3	1,51456	7,15	162,810	687,831	1,970736504
4	1,51456	7,15	165,810	2763,643	6,699784581
5	1,51456	7	228,720	2908,488	7,018057818
6	1,51456	0,1	45,8076	2291,729	7,232915368
7	1,51456	0,08	41,5101	173,852	0,592531584
8	61,2361	3,08	79,98	335,072	1,074981795
9	61,2361	2,88	79,2344	331,929	1,066129123
10	47,6806	1,033	240,72	244,378	7,411929929
11	47,6806	1,028	236,265	239,777	7,404330093
12	47,6806	1,023	172,022	173,844	7,267398161
13	47,6806	1,013	160	161,584	7,242300292
14	60,3263	1,313	20	83,4512	0,296804467

15	60,3263	1,013	32,8076	136,623	0,473492984

For exergy and advanced exergy analyses, real, theoretical and unavoidable conditions are also needed to be described. Due to lack of data on components that is used to create closed steam cycle, all data is assumed according to expert opinion. In addition, the aim of this paper is to show that analyses and new introduced efficiency term could be used to mitigate GHG emissions and improve fuel economy. Conditions data is given in table 5.

As it could be seen, in theoretical conditions minimum upper terminal temperature difference between cold fluid outlet and hot fluid inlet is 0.1 ℃ instead of 0. In theory, reversible processes which is for instance heat transfer at non temperature difference condition, take infinite time to be completed, however a small irreversibility makes processes finish in a finite time (Bejan, Moran et al. 1996).

Table 5: Assumed real, theoretical and unavoidable component conditions

Component	Conditions		
	Real	Theoretical	Unavoidable
1 Pump	$\eta = 0{,}8$	$\eta = 1$	$\eta = 0{,}9$
2 Feedwater preheater	$\Delta P_{12} = 0{,}2\ bar$ $\Delta P_{89} = 0{,}2\ bar$ $\Delta T_{Min} = 8$ ℃	$\Delta P_{12} = 0\ bar$ $\Delta P_{89} = 0\ bar$ $\Delta T_{Min} = 0{,}1$ ℃	$\Delta P_{12} = 0{,}1\ bar$ $\Delta P_{89} = 0{,}1\ bar$ $\Delta T_{Min} = 3$ ℃
3 Economizer	$\Delta P_{32} = 0{,}15\ bar$ $\Delta P_{1312} = 0.01\ bar$ $x_3 = 0$	$\Delta P_{32} = 0\ bar$ $\Delta P_{1312} = 0\ bar$ $x_3 = 0$	$\Delta P_{32} = 0{,}1\ bar$ $\Delta P_{1312} = 0.003\ bar$ $x_3 = 0$
4 Evaporator	$\eta_P = 0{,}8$ $\Delta P_{Circ} = 0{,}1\ bar$ $\Delta P_{1211} = 0.005\ bar$ $x_4 = 1$	$\eta_P = 1$ $\Delta P_{Circ} = 0\ bar$ $\Delta P_{1211} = 0\ bar$ $x_4 = 1$	$\eta_P = 0{,}9$ $\Delta P_{Circ} = 0{,}06\ bar$ $\Delta P_{1211} = 0.003\ bar$ $x_4 = 1$
5 Superheater	$\Delta P_{54} = 0{,}15\ bar$ $\Delta P_{1110} = 0.005\ bar$ $\Delta T_{Min} = 12$ ℃	$\Delta P_{54} = 0\ bar$ $\Delta P_{1110} = 0\ bar$ $\Delta T_{Min} = 0{,}1$ ℃	$\Delta P_{54} = 0.08\ bar$ $\Delta P_{1110} = 0.003\ bar$ $\Delta T_{Min} = 5$ ℃
6 Turbine	$\eta_T = 0{,}9$	$\eta_T = 1$	$\eta_T = 0{,}92$
7 Condenser	$P_6 = 0{,}1\ bar$ $\Delta P_{67} = 0{,}02\ bar$ $\Delta P_{1415} = 0{,}3\ bar$ $\Delta T_{Min} = 13$ ℃	$P_6 = 0{,}04\ bar$ $\Delta P_{67} = 0\ bar$ $\Delta P_{1415} = 0 bar$ $\Delta T_{Min} = 0{,}1$ ℃	$P_6 = 0{,}05\ bar$ $\Delta P_{67} = 0{,}01\ bar$ $\Delta P_{1415} = 0{,}1\ bar$ $\Delta T_{Min} = 5$ ℃

4. ANALYSIS

Real cycles are designed due to the information given in table 5. Temperature at atmosphere outside of exhaust gases after economizer is fixed at 160 ℃ due to low temperature corrosion risk (Deniz 2015) only for real cycles to harvest most of exhaust gas waste heat. Produced power of each cycle is calculated and noted to be used in other created cycles.

Unavoidable cycles are created with constant power outputs derived from real cycles. Hence mass flow rate and exhaust gas exit temperature are released free to maintain desired output power. It is seen that exhaust temperature at economizer outlet is above 160 ℃. Unavoidable exergy destruction of each component is calculated by exergy analyses of each cycle.

Theoretical cycles are built first to analyze endogenous and endogenous unavoidable exergy destructions of components with same amount of power output of real cycles. Then, all theoretical cycles are hybridized one by one for each component whether with real conditions for endogenous exergy destruction and unavoidable conditions for endogenous unavoidable exergy destruction. Lastly, endogenous and endogenous unavoidable exergy destructions calculated by exergy analyses of each hybrid cycle.

Table 6: Mass, energy balances and exergy analysis equations

Component	Mass Balance	Energy Balance	Exergy Analysis	
			$\dot{E}x_F$	$\dot{E}x_P$
1 Pump	$\dot{m}_7 = \dot{m}_1$	$\dot{W}_P = \dot{m}_1(h_1 - h_7)$	$\dot{W}_P$	$\dot{E}x_1 - \dot{E}x_7$
2 Feedwater preheater	$\dot{m}_1 = \dot{m}_2$ $\dot{m}_9 = \dot{m}_8$	$\dot{m}_1(h_2 - h_1) = \dot{m}_8(h_8 - h_9)$	$\dot{E}x_8 - \dot{E}x_9$	$\dot{E}x_2 - \dot{E}x_1$
3 Economizer	$\dot{m}_3 = \dot{m}_2$ $\dot{m}_{13} = \dot{m}_{12}$	$\dot{m}_3(h_2 - h_3) = \dot{m}_{12}(h_{12} - h_{13})$	$\dot{E}x_{12} - \dot{E}x_{13}$	$\dot{E}x_3 - \dot{E}x_2$
4 Evaporator	$\dot{m}_4 = \dot{m}_3$ $\dot{m}_{11} = \dot{m}_{12}$	$\dot{m}_3(h_4 - h_3) = \dot{m}_{12}(h_{11} - h_{12})$	$\dot{E}x_{11} - \dot{E}x_{12}$	$\dot{E}x_4 - \dot{E}x_3$
5 Superheater	$\dot{m}_4 = \dot{m}_5$ $\dot{m}_{10} = \dot{m}_{11}$	$\dot{m}_4(h_5 - h_4) = \dot{m}_{11}(h_{10} - h_{11})$	$\dot{E}x_{10} - \dot{E}x_{11}$	$\dot{E}x_5 - \dot{E}x_4$
6 Turbine	$\dot{m}_5 = \dot{m}_6$	$\dot{W}_T = \dot{m}_5 h_5 - \dot{m}_6 h_6$	$\dot{E}x_5 - \dot{E}x_6$	$\dot{W}_T$
7 Condenser	$\dot{m}_6 = \dot{m}_7$ $\dot{m}_{14} = \dot{m}_{15}$	$\dot{m}_6(h_6 - h_7) = \dot{m}_{14}(h_{15} - h_{14})$	$\dot{E}x_6 - \dot{E}x_7$	$\dot{E}x_{15} - \dot{E}x_{14}$

All cycles are created for each ahead lever position and closed steam cycle pressure combination. Mass and enegy conversation laws are applied. Exergy and advanced exergy analyses are executed for a total of 288 cycles and all components. Equations are shown in table 6. As it can be seen, all exergy rates of streams should be calculated to apple exergy and advanced exergy analyses.

5. RESULTS AND DISCUSSION

For being consistent with the given data on table 4, results of exergy and advanced exergy analyses are given in table 7 and table 8 with exergy destruction ratio, exergy efficiency and new exergy efficiency calculations for 7.5 bar feedwater pressure and navigational full 100% lever position.

Table 7: Exergy analysis of steam cycle at 7.5 bar pressure and 100% lever conditions

Component	$\dot{E}x_{F,k}$	$\dot{E}x_{P,k}$	$\dot{E}x_{D,k}$	ε_k	$y^*_{D,k}$
Pump	1,419	1,171	0,249	0,825	0,000
Feedwater Heater	44,415	32,991	11,424	0,743	0,021
Economizer	257,672	174,124	83,548	0,676	0,151
Evaporator	1360,340	1187,524	172,816	0,873	0,312
Superheater	120,397	87,707	32,690	0,728	0,059
Turbine	1023,005	934,118	88,887	0,913	0,160
Condenser	460,512	296,157	164,355	0,643	0,297
Overall System	1782,825	934,118	553,969	0,524	

Table 8: Advanced Exergy analysis of steam cycle at 7.5 bar pressure and 100% lever conditions

Component	$\dot{E}x^{UN}_{D,k}$	$\dot{E}x^{AV}_{D,k}$	$\dot{E}x^{EN}_{D,k}$	$\dot{E}x^{EX}_{D,k}$	$\dot{E}x^{EN,UN}_{D,k}$	$\dot{E}x^{EN,AV}_{D,k}$	$\dot{E}x^{EX,UN}_{D,k}$	$\dot{E}x^{EX,AV}_{D,k}$	ε^+_k
Pump	0,100	0,148	0,194	0,054	0,086	0,108	0,014	0,040	0,921
Feedwater Heater	17,341	-5,917	15,680	-4,256	15,403	0,277	1,939	-6,195	0,655
Economizer	53,971	29,577	76,360	7,188	50,896	25,464	3,075	4,113	0,763
Evaporator	156,359	16,457	152,987	19,829	145,444	7,542	10,914	8,915	0,884
Superheater	21,733	10,957	28,892	3,798	20,283	8,609	1,450	2,348	0,801
Turbine	72,504	16,383	93,844	-4,957	73,443	20,400	-0,940	-4,017	0,928
Condenser	73,795	90,560	139,674	24,681	65,120	74,555	8,675	16,006	0,801
Overall System		158,16							0,575

Figure 4 shows that 75% lever positions have best exergy and new exergy efficiencies for all pressures. Power outputs at 100% lever position are the highest as expected. 75% lever positions for all feedwater pressure conditions have the highest exergy and new exergy efficiencies among all. This might be related to main engine characteristics. At 8 bar steam cycle power outputs, exergy and new exergy efficiencies are highest amongst all conditions. Original exhaust boiler pressure, 7.5 bar, should be changed and efforts should be focused on 8 bar feedwater pressure condition. This is the first step of improvement.

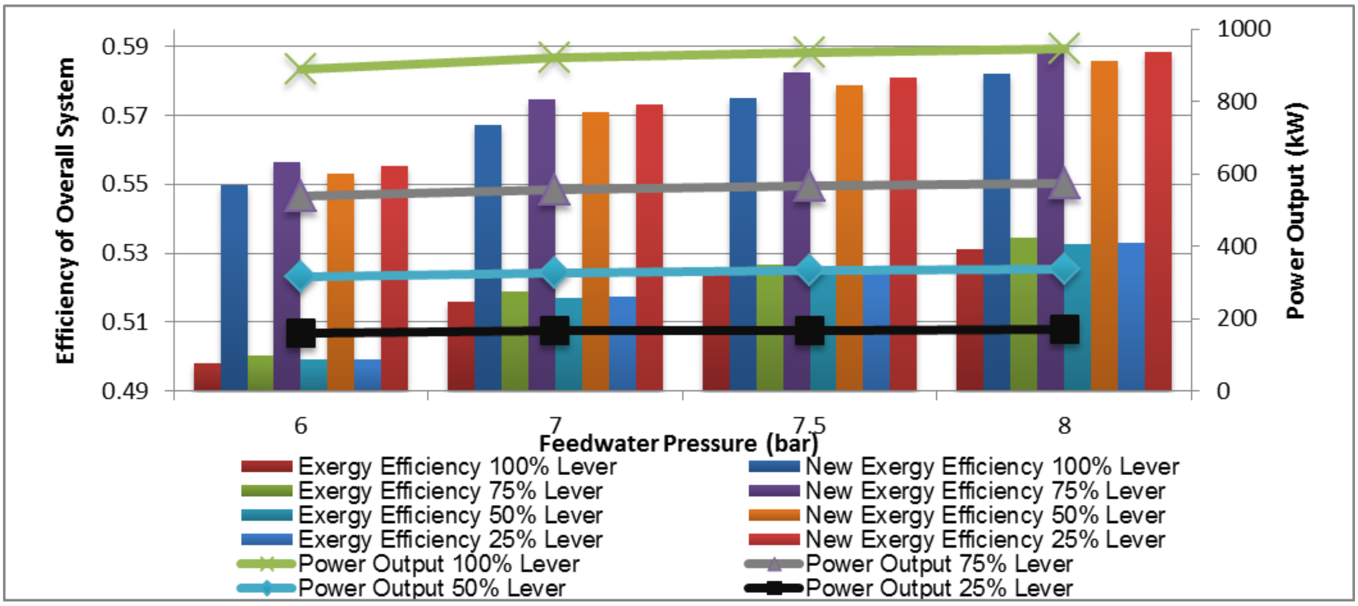

Figure 4: exergy, new exergy efficiencies and power output according to feedwater pressure and lever positions

Figure 5 shows exergy analysis results of exergy destruction ratios for all lever positions at 8 bar. Pump has the lowest share among all and condenser has the highest for all lower positions. Evaporator has the second highest share of exergy destruction. Trends of exergy destructions do not change significantly with respect to the lever position. In this table, exergy analysis expresses that improvement efforts should be focused on condenser first then evaporator. Least effort should be directed to pump and feedwater heater. However, there is no information about how much of this exergy destruction could be recovered or what is the effect of other components on investigated component.

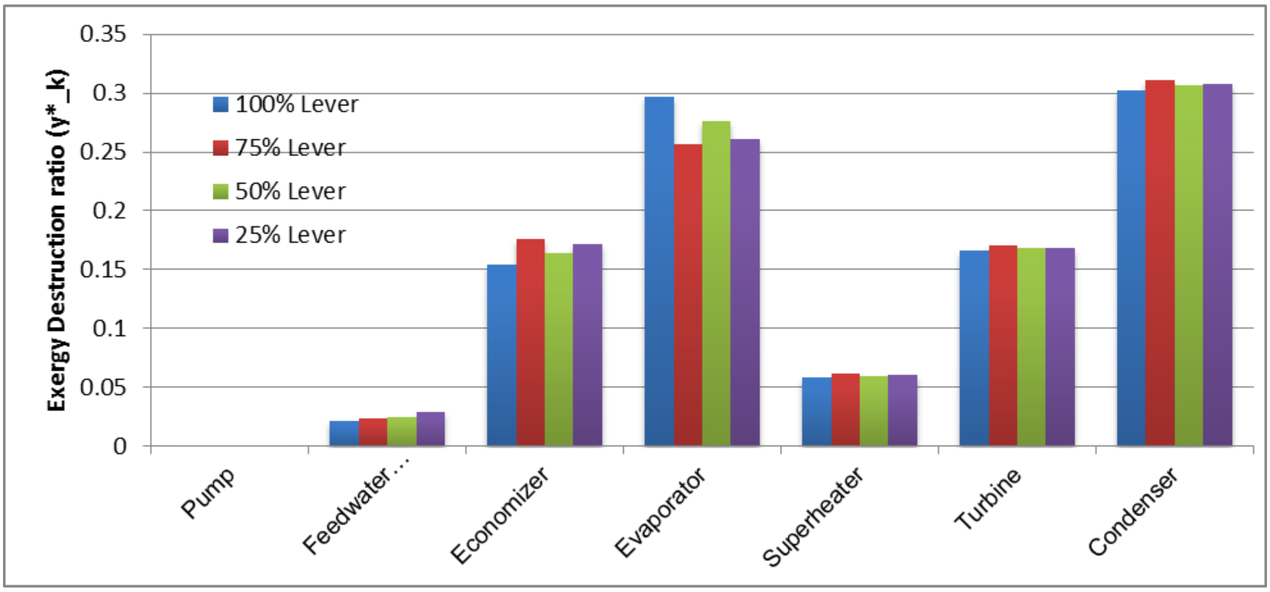

Figure 5: Exergy destruction ratios of components for all lever positions al 8 bar.

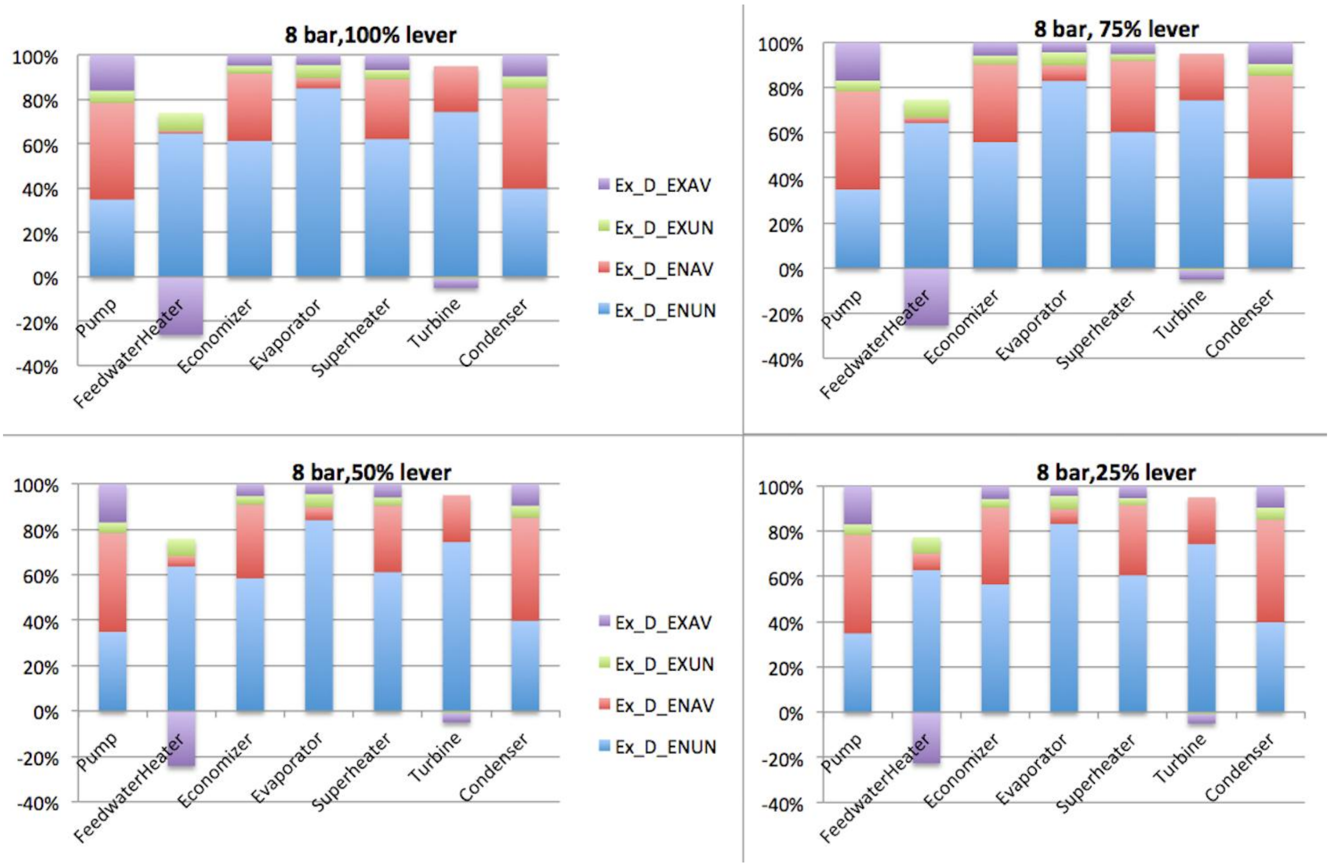

Figure 6: Percentages of exergy destructions as result of advanced exergy analyses

Figure 6 shows percentage parts of investigated components' exergy destructions after application of advanced exergy analysis. It could be appropriate to notice that in all lever position cases, exogenous avoidable destructions of feedwater heater and turbine and also exogenous avoidable exergy destruction of turbine are negative. This is because of change in condenser pressure. When condenser pressure is lower in endogenous, endogenous unavoidable hybrid and unavoidable cycles than real cycle, turbine produces more power, but there is also more exergy destruction. And also, inlet stream (pipe 1) would have lower temperature due to low condenser pressure, hence feedwater preheater's exergy rate of fuel and exergy rate of product increase, so does exergy destruction. Besides that, it is also observable that exergy destruction profiles for all conditions are similar for each investigated component. Even the values are slightly different, general evaluation may be made by only analysing one condition combination e.g. 8 bar, 100% lever position. The hhighest share is the endogenous unavoidable exergy destruction of all components, then endogenous avoidable exergy destruction. That means, interactions among components of the overall system are weak and mostly, it is the component itself that has irreversibilites to improve. Pump has the highest share of endogenous avoidable exergy destruction, the second is condenser and the third one is economizer. Exogenous avoidable exergy destruction is also the highest at pump, then condenser. With respect to the information on Figure 6, one would focus on firstly avoidable exergy destruction to recover more exergy, secondly, compare the highest percentage of avoidable exergy destructions to rank components, finally start improvement within the component itself or system topology.

Figure 7 shows avoidable and unavoidable parts of all components' exergy destructions as well as exergy and new exergy efficiencies of each component. In general, distribution trends of all variables are similar for all conditions. Maximum avoidable exergy destruction belongs to condenser, that means if any improvement might be done, it should start with condenser due to high recoverability of its exergy destruction. Second is economizer to focus on. Evaporator has the highest share unavoidable exergy destruction, even its avoidable share is higher than superheater, however comparing with Figure 6, it is obvious to say less effort should be made on evaporator to improve and also pump had the highest percentage of avoidable exergy destruction however the amount recoverable is the smallest among all.

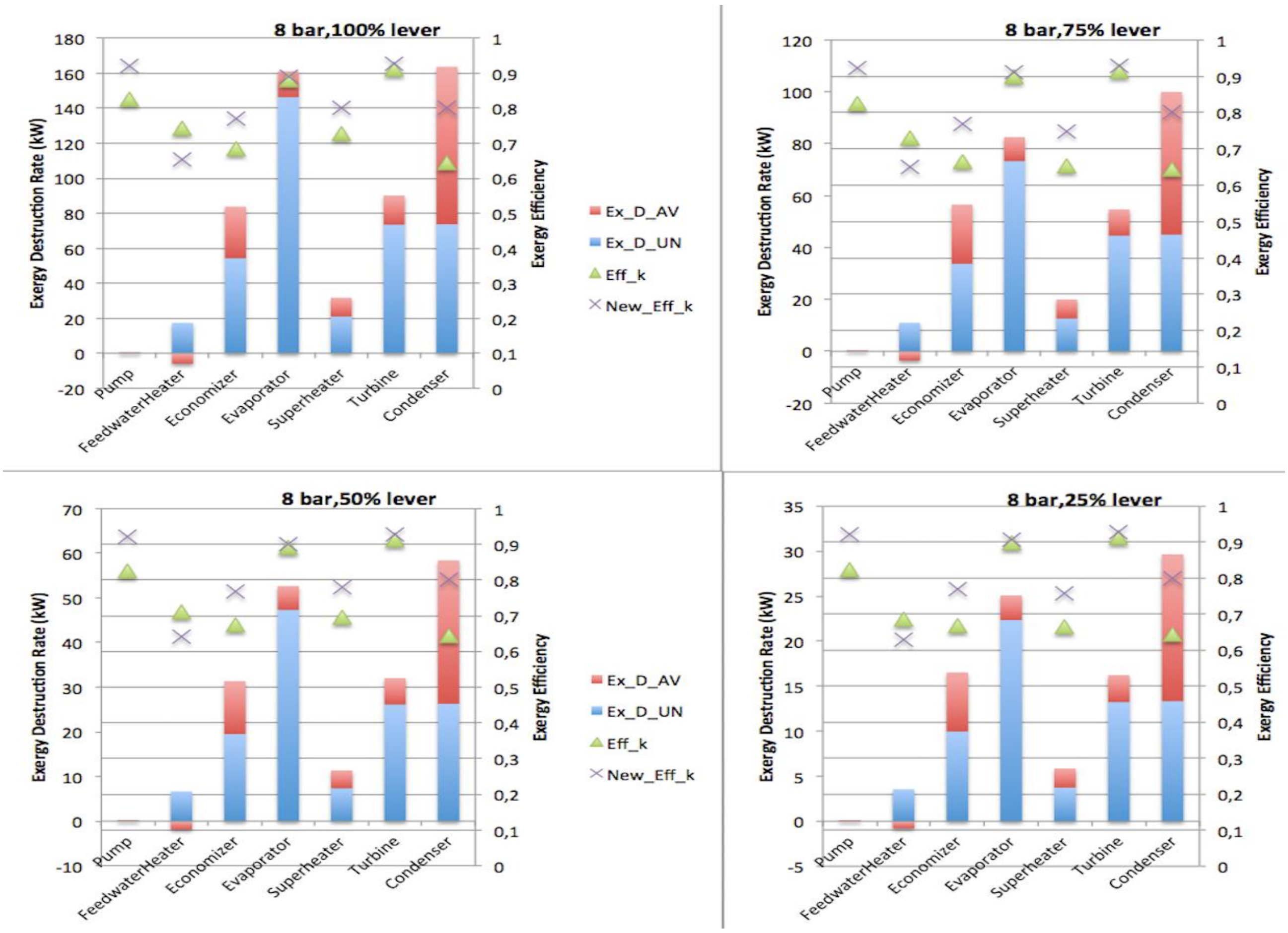

Figure 7: Shares of avoidable and unavoidable exergy destructions as result of advanced exergy analyses

Figure 8 shows specific fuel consumption, NOx, SOx and CO2 contents before and after steam cycle applied to all ahead lever positions at 8 bar feedwater pressure. It is easy to observe that all of the variables have decreased while main engine speed, ship speed, fuel consumption are constant and power output has increased. By increasing power output, diesel generator and shaft generator needs for electricity drop, and even excessive power output of steam cycle may be added to shaft by PTI/PTO gearbox when needed. Hence, inclusion of a steam cycle could be expressed as a large improvement.

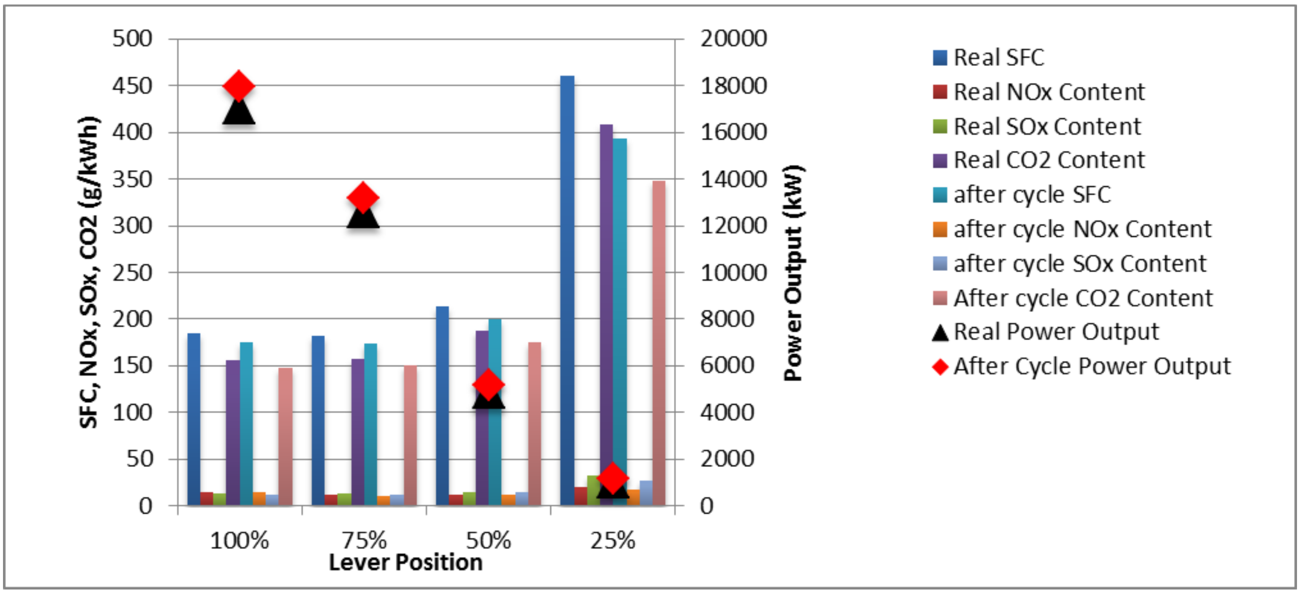

Figure 8: Specific fuel consumption, NOx, SOx, CO2 contents and power output before and after cycle

6. CONCLUSIONS

In this paper, waste heat recovery with a steam cycle is selected among other GHG reduction and efficiency improvement techniques. It has been applied to a typical five-cylinder two-stroke marine engine as a single pressure closed steam cycle. Then, it is analyzed by exergy and advanced exergy analyses with parameters of feedwater pressure and main engine lever position and optimum range of system operation determined as 8 bar feedwater pressure and all lever positions.

A detailed advanced exergy analysis revealed more information on system operation and improvement availabilities of investigated components compared to standard exergy analysis. While exergy analysis suggested that improvement focus and study should be on condenser, evaporator, turbine, economizer, superheater, feedwater heater and pump respectively, advanced exergy analyses presented a different order as condenser, economizer, turbine, evaporator, superheater, and pump. Also, effects of other components on investigated component have been obtained by the help of splitting exergy destructions. This information would help designer to decide best working conditions and topology of the overall system.

Newly introduced exergy efficiency term has been exposed the benefit of using advanced exergy analysis by showing the potential improvement and improved efficiency of components and overall system.

Adding a closed steam cycle after main engine increased power output of the ship. Hence, generator necessity is decreased or completely vanished. With a PTI/ PTO gearbox produced power could be canalized to the shaft directly when needed. Also adding WHR system decreased specific fuel consumption, NOx, SOx, CO2 emissions by range 5.25% up to 14,7% regarding to lever positions.

It would be useful to apply this methods and new exergy efficiency not only a part but also all ship systems may be more beneficial to exhibit potentials of improvement. Thus, more studies should be performed on different engines and ships for efficiency and protecting nature.

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