Received: 16.02.2021
Accepted: 13.04.2021
Published Online: 30.06.2021
DOI: 10.18613/deudfd.881570
Research Article

Dokuz Eylül University
Maritime Faculty Journal
Vol:14 Issue1: Year:2022 pp:99-121
E-ISSN: 2458-9942

THERMODYNAMIC ANALYSIS AND PAYBACK PERIOD INVESTIGATION OF POWER TURBINE GENERATOR FOR MARINE DIESEL ENGINES

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ABSTRACT

Because of the heat flow and temperature, marine diesel engines exhaust gas energy is by far the highest desirable among the waste heat sources of a ship. Waste heat recovery systems generate electrical power by using this exhaust gas energy. This study measures how much the power turbine generator improves the system's efficiency for three engine loads and three ambient conditions. This study presents a power turbine generator using a six-cylinder low-speed marine diesel engine. Thermodynamic analyses are performed by considering three ambient conditions and three diesel engine loads, and the exergy destruction of each component, exergy efficiency, exergy destruction rate, efficiency increases, and power output of the system are calculated. In addition, the payback period is calculated according to the installation cost of the power turbine generator and the cost of the fuel saved annually. Based on the analysis, the highest power output for the power turbine generator occurred under 100% engine load and winter ambient conditions, and the lowest power output occurred under 75% engine load and tropical ambient conditions. Alternatively, for the same ambient conditions, the highest efficiency increase occurred at 90% engine load, and the lowest efficiency increase occurred at 100% engine load. In the power turbine generator, it is observed that the shortest payback period is 100% engine load and winter ambient conditions.

Keywords: *Power Tturbine Generator, Marine Diesel Engine, Engine Load, Ambient Condition, Payback Period*

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DENİZ DİZEL MOTORLARI İÇİN GÜÇ TÜRBİNİ JENERATÖRÜNÜN TERMODİNAMİK ANALİZİ VE GERİ ÖDEME SÜRESİ ARAŞTIRMASI

ÖZ

Isı akışı ve sıcaklık nedeniyle, deniz dizel motoru egzoz gazı enerjisi, bir geminin atık ısı kaynakları arasında açık ara en çok arzu edilen enerjidir. Atık ısı geri kazanım sistemleri, bu egzoz gazı enerjisini kullanarak elektrik enerjisi üretir. Bu çalışma, güç türbini jeneratörünün 3 motor yükü ve 3 ortam koşulu için sistemin verimliliğini ne kadar iyileştirdiğini ölçmektedir. Bu çalışma, altı silindirli düşük hızlı deniz dizel motoru kullanan bir güç türbini jeneratörünü göstermektedir. Termodinamik analizler, üç ortam koşulu ve üç dizel motor yükü dikkate alınarak yapılır ve her bir bileşenin ekserji yıkımı, ekserji verimi, ekserji yıkım oranı, verimlilik artışları ve sistemin güç çıkışı hesaplanır. Ayrıca geri ödeme süresi, güç türbini jeneratörünün kurulum maliyeti ve yıllık tasarruf edilen yakıt maliyetine göre hesaplanmaktadır. Analize göre, güç türbini jeneratörü için en yüksek güç çıkışı %100 motor yükü ve kış ortam koşullarında, en düşük güç çıkışı ise %75 motor yükü ve tropikal ortam koşullarında gerçekleşmiştir. Öte yandan aynı ortam koşullarında en yüksek verimlilik artışı %90 motor yükünde, en düşük verimlilik artışı ise %100 motor yükünde gerçekleşmiştir. Güç türbini jeneratöründe en kısa geri ödeme süresinin %100 motor yükü ve kış ortam koşulunda olduğu görülmüştür.

Anahtar Kelimeler: *Güç Türbini Jeneratörü, Deniz Dizel Makinesi, Makine Yükü, Ortam Koşulu, Geri Ödeme Süresi*

1. INTRODUCTION

Recently, maritime transport has been expanding rapidly and is now responsible for 80 to 90 percent of global trade. The importance and role of maritime transport in the economy are evident today, as we observe that the amount and value of goods transported in global trade are a hundred times greater than in the 1950s. Shipping, however, is currently vulnerable to major difficulties. Bunker fuel costs are now triple the costs in the 1980s and, depending on the ship type, fuel costs are expected to be 43 to 67 percent of overall operating costs (Baldi and Gabrielii, 2015: 654-665).

In almost all large ocean-going ships, two-stroke, low-speed diesel engines are used. Their efficiency approaches 48-51%, while the exhaust gas leaves the engine with large amounts of heat (Nielsen et al. 2014: 687-693). The amount of exergy, which can be defined as the work potential of energy, enables us to make predictions about the maximum power that can be gained with waste heat recovery systems. According to Figure 1, exhaust gas, which constitutes approximately 50% of the waste heat after the combustion process, is the resource most worth saving in both energy and exergy content. Although sweeping air has a 35% share and is second to exhaust gas in terms of energy content according to the chart, this ratio is approximately just 25% in the exergy chart. There is a difference of about 50% between the energy and exergy values of the lubricating oil and jacket-cooling water (Akman, 2016). By applying the waste heat recovery systems, it is possible to recuperate energy from this source (Man Diesel & Turbo, 2014: 6).

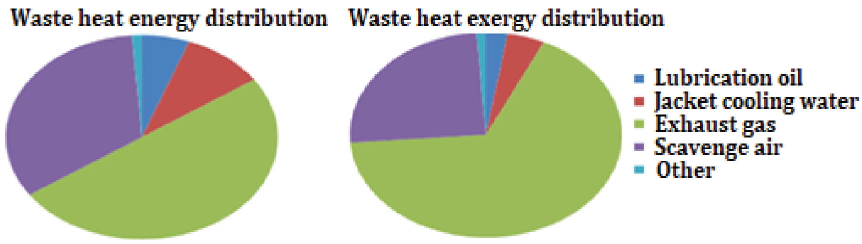


Figure 1: The distribution of Energy and Exergy in a Marine Diesel Engine
Source: Bellolio et al., 2015: 1-11.

The waste heat recovery systems to be installed on vessels can be chosen depending on the complexity level of the systems to be installed and the actual electrical energy consumption on the ship. The waste heat recovery systems that can be selected for use in ships are divided into three. The first of these systems is the power turbine generator, which is the simplest to install and the cheapest. With the power turbine generator, depending on marine engine power, the system efficiency can be improved between 3% and 5%. The application of power turbine generators on ships with diesel engine power below 15,000 kW is technically and economically viable. The second of these systems is the steam turbine generator. With the steam turbine generator, it is possible to increase the system efficiency between 5% and 8% depending on the diesel engine power. The application of steam turbine generators on ships with diesel engine power below 25,000 kW is technically and economically viable. The third system is combined systems, in which the power turbine and steam turbine are used together and which can be applied to ships with high electricity needs such as container ships. When combined systems are used in ships, it is possible to achieve an efficiency increase of 8% to 11% depending on the diesel engine power. The application of combined systems on ships with diesel engine power above 25,000 kW is technically and economically viable. The selection of the most appropriate system for a specific ship project, therefore, needs careful consideration because of specifications such as fuel efficiency, regulatory restrictions, pollution requirements, the ship operating profile, and payback time (MAN B&W. 2014a: 9).

The Thermo Efficiency System (TES) consists of a fired boiler system for exhaust gas, a steam turbine (often referred to as a turbo generator), an exhaust gas turbine (often referred to as a power turbine), and a common generator for generating electricity. The turbines and the generator are placed on a common bedplate. Man Diesel & Turbo (2014: 5) has implemented the 'Thermo Efficiency System' at 100% SMCR for the 12K98ME / MC engine under ISO ambient conditions, achieving a gain of 11.2% for double pressure of 9.9% for single pressure. Also, the overall engine efficiency improves from 49.3% to 54.2% for single pressure and 54.8% for double pressure. The simple single pressure steam system uses only the heat of the exhaust gas. For the preheating of the feedwater, the more complicated dual pressure steam system requires additional waste heat recovery (WHR) sources (jacket water and scavenge air heat) (Man Diesel & Turbo, 2014: 10-12). Wartsila (2014: 7) has achieved a 12% gain by implementing a waste heat recovery system called 'Total Heat Recovery Plant' for the Sulzer 12RTA96C engine. The overall efficiency of the engine increased from 49% to 54.9%. Ersayın and Özgener (2015) conducted a performance analysis of the combined cycle power plant with the actual operation data received from the power plant control unit. They calculated the energy and exergy efficiency of each component of the power plant and found the combined cycle power plant's energy and exergy efficiency to be 56% and 50.04%, respectively. They made a parametric analysis for each component according to the ambient temperature and showed that the ambient temperature has significant effects on the performance of each component. Abuşoğlu and Kanoğlu (2009) performed exergy and thermoeconomic analyses of a real diesel engine cogeneration plant established in Gaziantep. They found that the diesel engine is the component with the highest rate of exergy breakdown among the components in the plant. They calculated the exergy destruction rate of the diesel engine as 83.32% and the exergy efficiency of the facility as 40.6%. They stated that small improvements in engine design and operation would increase plant efficiency more than major improvements in other components. Kanoğlu and Dinçer (2009) did performance analysis of various cogeneration systems by calculating energy and exergy efficiency. The cogeneration facilities they considered included a steam turbine system, gas turbine system, and diesel engine system. They investigated the effects of parameters such as steam pressure and water temperature on energy and exergy efficiency. They found the exergy efficiency of the gas turbine cogeneration system to be 22.6%, the steam turbine exergy efficiency to be 23.1%, and the exergy efficiency of the diesel engine cogeneration system to be 47.7%.

There are many studies in the literature on the thermodynamic analysis of waste heat recovery systems. However, studies on thermodynamic analysis of waste heat recovery systems for different engine loads and ambient conditions are limited. In addition, there are not many studies on the economic analysis of waste heat recovery systems. This study presents a power turbine generator using a six-cylinder low-speed marine diesel engine. Thermodynamic analyses are performed by considering three different ambient conditions and three different diesel engine loads, and the exergy destruction of each component, exergy efficiency, exergy destruction rate, efficiency increases, and power output of the system are calculated. Besides that, economic analysis is carried out according to the installation cost of the power turbine generator and the cost of the fuel saved annually.

2. POTENTIAL WASTE HEAT SOURCES IN SHIPS

Heat is lost from the engine to the environment in many ways. Approximately 5% of the total energy production of the engine goes into the cooling water system of a marine engine, and approximately 25% goes into the exhaust gas. In either form, heat is useful as a heat source for other systems (ABS, 2013: 45). Although the exhaust gas temperature is primarily subject to the nominal power of internal combustion engines, the jacket water temperature for each engine is almost identical. The exhaust gas temperature is usually above 673 K, the outlet temperature is between 363 K and 368 K for jacket water, and the return temperature is between 343 K and 358 K. Waste heat recovery from the exhaust gas and jacket water is recognized as providing useful opportunities that can improve engine efficiency dramatically and provide major economic and environmental benefits (Ma et al. 2016: 218-226). The comparison of engine heat balances with and without waste heat recovery systems is demonstrated in Figure 2. It is shown that about 50% of the total fuel heat energy is released by various streams into the atmosphere without doing any useful work. The total efficiency of the engine will increase to about 55% accompanied by the waste heat recovery method (Man Diesel & Turbo, 2014: 6; MAN B&W, 2014a: 9).

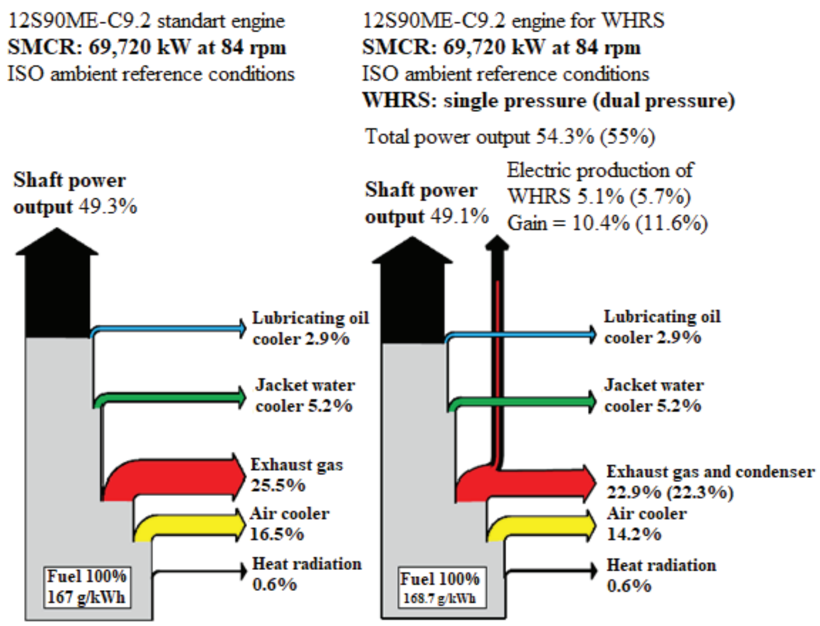


Figure 2: Heat Balance for Low-speed MAN B&W Engine Types without and with Waste Heat Recovery System
Source: MAN B&W, 2014a: 5.

2.1. Factors Influencing the Applicability of Waste Heat Recovery

Two waste stream parameters should be determined to test the applicability of waste heat recovery.

- Heat quantity
- Heat source temperature/quality

These parameters allow analyzing the quality and quantity of the stream. The following briefly discusses the principles that determine the applicability of waste heat recovery (U.S. Department of Energy, 2008: 6).

2.1.1. Heat Content

Heat content is an evaluation of how much energy the stream of waste heat produces. The heat quality is an indicator of waste heat's usefulness. Both the mass flow rate and flow temperature are a function of the amount of waste heat present in the waste stream.

$$E = \dot{m} \cdot h(T) \tag{1}$$

E is loss of waste heat, $\dot{m}$ is mass flow rate of the waste stream, and h (T) is the specific enthalpy of the waste stream. Although the amount of waste heat available is an important parameter, waste heat recovery alone is not an efficient measure of its applicability. It should be remembered that the waste heat quality is determined by the waste heat source temperature (U.S. Department of Energy, 2008: 6).

2.1.2. Heat Source Quality

The temperature of the waste heat source is a significant element that determines the applicability of waste heat recovery. Waste heat quality is classified as low for 232 °C and below, medium for 232-650 °C, and high for 650 °C and above (U.S. Department of Energy, 2008: 7).

2.2. Impact of Ambient Conditions on Temperature of Exhaust Gas

Marine diesel engines are generally two stroke and turbocharged types to provide high power. These engines have high thermal efficiency, which provides an important advantage over all other thermal engines. However, due to the presence of the turbocharger, its operations are complex and influenced by various parameters. The temperature level of the exhaust gas at the cylinder exits and the turbine upstream usually limits the power output of these engines. Therefore, it is essential to keep both at acceptable levels. Alternatively, the efficiency of the engine is greatly affected by the combustion mechanism in the engine cylinder. Any parameter influencing this mechanism has a remarkable impact on the engine's efficiency. Ambient conditions have a considerable effect on the operation of diesel engines and particularly on turbocharged engines. The combustion mechanism in the engine cylinder is highly influenced by the temperature and pressure of the charge air. Therefore, all conditions affecting these two parameters are crucial to engine operation (Hountalal et al. 2012: 2). The operating conditions for a standard main engine are divided into three: winter, ISO and tropical ambient conditions. Table 1 shows the values accepted for winter, ISO and tropical ambient conditions.

Table 1: Winter, ISO and Tropical Ambient Conditions

	Winter	ISO	Tropical
Barometric pressure (bar)	1	1	1
Air inlet temperature of turbocharger (°C)	10	25	45
Cooling water temperature (°C)	10	25	32
Relative humidity rate (%)	60	30	60

Source: MAN B&W, 2014b: 11.

2.3. Exhaust Gas Components

Two-stroke low-speed diesel engines pioneered the use of low-quality fuels, low fuel consumption and high reliability. From an operational viewpoint, it is inevitable that harmful gases are present in the exhaust gas. The emission values of NO_x, SO_x, CO, HC and particles, which are among the most discussed pollutants, are shown in Figure 3 (MAN B&W, 2014c: 14).

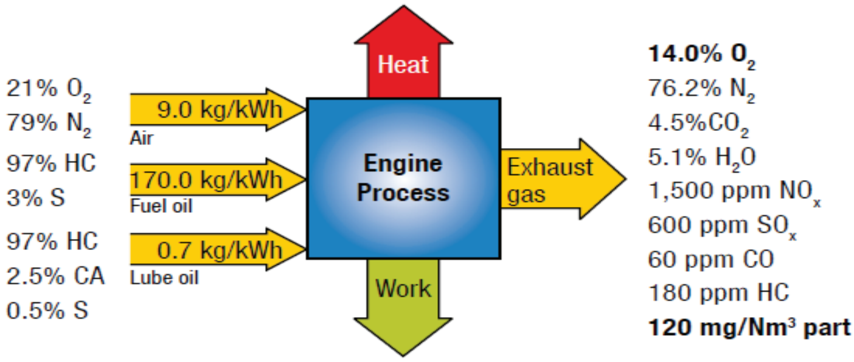


Figure 3: Typical emissions from low-speed diesel engines of the MC/ME type
Source: MAN B&W, 2014c: 14.

In the combustion process, the three main ingredients, air, fuel oil, and lubrication oil, go into the engine. 21% Oxygen (O_2) and 79% Nitrogen are in the air (N_2). Ninety-seven percent hydrocarbon (HC) groups and 3% sulfur (S) are found in fuel oil. Some lubrication oil is also included in the combustion phase, but the rate is low. Lubrication oil consists of 97% HC groups, 2.5% CA groups, and 0.5% sulfur (Gunes et al. 2012: 4). A small number of incomplete combustion products can be ignored such as CO, unburned hydrocarbon and NO_x emission components, and the mixture of CO_2, H_2O, N_2 and O_2 can be assumed as the composition of the exhaust gases (Wang et al. 2013: 414-419).

3. SYSTEM DEFINITION

3.1. Specified Marine Diesel Engine

For the three engine loads of the Hyundai-Wartsila 6RT-flex58T-E model marine diesel engine with 13.94 MW power output used with the power turbine generator, the mass flow rate and temperature of exhaust gas and mass flow rate and temperature of scavenge air values are given in Table 2 (Mito et al. 2018: 264-276).

Table 2: Mass Flow Rate and Temperature Values of Exhaust Gas and Scavenge Air at Various Engine Loads

Engine load (%)		75	90	100
Engine speed (rpm)		95	101	104.6
Exhaust gas	Mass flow rate (kg/s)	24.1	28.8	32.7
	Temperature (°C)	249	251	261
Scavenge air	Mass flow rate (kg/s)	23.5	28.2	31.9
	Temperature (°C)	190	213	230

Source: Mito et al. 2018: 264-276.

3.2. Power Turbine Generator

The simplest and cheapest power turbine generator to install among the waste heat recovery systems, as seen in Figure 4, comprises a power turbine located on the side path of the exhaust gas and a generator connected to this turbine. A compressor and turbine assembled on a common shaft constitute the turbocharger portion of the system. To do the shaft work needed for the compressor, exhaust gases from the diesel engine flow through the turbocharger's turbine (Abuşoğlu and Kanoğlu, 2009: 234-241). Using part of the exhaust gas which is not sent to the turbocharger, the power turbine produces output electrical power. Since the bypass valve of exhaust gas is turned off when an engine load is less than 50 percent SMCR (specified maximum continuous rating), the calculations consider three engine loads of more than 50 percent SMCR (Man Diesel & Turbo, 2014: 6; Saeed, 2014: 2). It is technically appropriate to apply a power turbine generator to engines with a main engine power below 15,000 kW (Bellolio et al.2015: 1-11). By using a power turbine generator on ships, it is possible to achieve an efficiency increase of 3% to 5% depending on the main engine power in the system (MAN B&W, 2014a: 7).

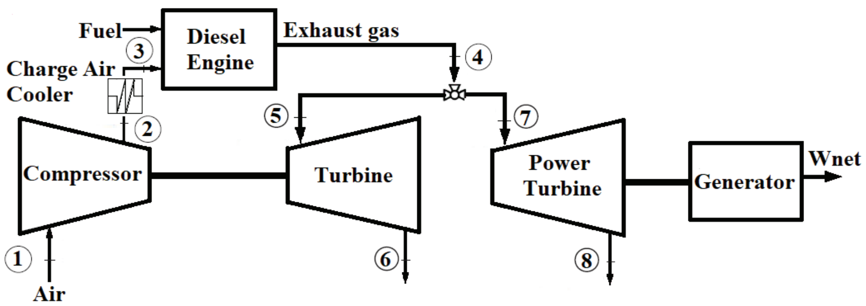


Figure 4: Power Turbine Generator
Source: Cengel and Boles, 2006: 507.

4. THERMODYNAMIC ANALYSIS

In this study, thermodynamic analyses are conducted using the EES package software. Energy and exergy equations used in thermodynamic analysis are given below. The EES package program is used for energy and exergy analyses. Temperature and pressure values at each point of the cycle are entered into the program and enthalpy and entropy values are obtained. The obtained enthalpy and entropy values are written in the equations for the power turbine generator components given in the third section, and the exergy destruction, exergy destruction rate, exergy efficiency and efficiency increase of each component are calculated (F-Chart Software, 2018).

4.1. Energy and Exergy Equations Used in Thermodynamic Analysis

In this study, the steady-state physical and thermodynamics equations are used based on the first and second thermodynamic laws. The steady state equations are viable during the ship's voyage. The assumptions made during this study are listed below (Abuşoğlu and Kanoğlu, 2009: 234-241).

- The power turbine generator is assumed to be at a steady state.
- Ideal gas principles are valid for exhaust gases and air.
- Combustion reaction is completed in the marine engine.
- Potential and kinetic energy changes are neglected.
- The dead state's pressure and temperature are taken as real ambient conditions.
- The fuel lower heating value (LHV) is used since the state of the water in the exhaust is usually steam in marine diesel engines.

For any steady state system, the mass balance equation (2), energy balance equation (3), and exergy balance equation (4) can be written with the above assumptions:

$$\sum \dot{m}_i = \sum \dot{m}_e \tag{2}$$

$$\dot{Q} + \dot{W} = \sum \dot{m}_e \, h_e - \sum \dot{m}_i \, h_i \tag{3}$$

$$\dot{E}_{heat} + \dot{W} = \sum \dot{m}_e\, e_e - \sum \dot{m}_i\, e_i + \dot{E}_{dest} \tag{4}$$

Where $\dot{W}$ and $\dot{Q}$ are the work and heat inputs, $\dot{m}$ is the mass flow rate of the fluid flow, h is the enthalpy, $\dot{E}_{dest}$ is the exergy destruction rate, and $\dot{E}_{heat}$ is the exergy transfer by heat at temperature T, which is presented by equation (5) (Abuşoğlu and Kanoğlu, 2008: 2026- 2031).

$$\dot{E}_{heat} = \sum(1 - \tfrac{T_0}{T})\dot{Q} \tag{5}$$

The specific flow exergy and total exergy can be calculated from equations (6) and (7). In equation (6), T_0 is the temperature at ambient conditions. Exergy destruction owing to irreversibility in the system is presented in equation (8). The exergy destruction rate (λ), which can be calculated using Equation (9), is calculated to evaluate the energy loss and irreversibility rate generated in each component according to the total exergy destruction. Exergy efficiency, which stands for the ratio between the work output ($\dot{W}_e$) and the fuel exergy ($\dot{E}_{fuel}$) of the waste heat recovery system, is another significant parameter in conducting an exergy analysis. From equation (10), the exergy efficiency (ε) can be calculated (Mito et al. 2018: 264-276; Kanoğlu and Dinçer, 2009: 76-81).

$$e = (h - h_0) - T_0(s - s_0) \tag{6}$$

$$\dot{E} = \dot{m}e \tag{7}$$

$$\dot{E}_{dest} = \sum \dot{m}_i\, e_i - \sum \dot{m}_e\, e_e + \left(1 - \tfrac{T_0}{T}\right)\dot{Q} + \dot{W} \tag{8}$$

$$\lambda = \frac{\dot{E}_{dest,i}}{\dot{E}_{dest,t}} \tag{9}$$

$$\varepsilon = \frac{\dot{W}_e}{\dot{E}_{fuel}} = 1 - \frac{\dot{E}_{dest}}{\dot{E}_{fuel}} \tag{10}$$

4.2. Thermodynamic Performance Analysis of Power Turbine Generator

Thermodynamic analysis of a power turbine generator often involves evaluating the performance of system components. The isentropic efficiencies of an adiabatic turbine and an adiabatic compressor can be stated, respectively as (Cengel and Boles, 2006: 512):

$$\eta_{turb} = \frac{w_a}{w_s} = \frac{h_i - h_e}{h_i - h_{e,s}} \tag{11}$$

$$\eta_{comp} = \frac{w_s}{w_a} = \frac{h_{e,s} - h_i}{h_e - h_i} \tag{12}$$

w_a and w_s are the real and isentropic specific works in equations (11) and (12), respectively, and the subscript s represents the isentropic condition (Cengel and Boles, 2006: 507). The energetic and exergetic formula for the components of the power turbine generator are given in Table 3.

The chemical exergy value e_f of the fuel is calculated using equation (13). Where ξ is the ratio of the fuel exergy flow, and LHV_f is the lower calorific value of the fuel (İbrahim et al. 2017: 977-985). For the HFO (heavy fuel oil) used in marine diesel engines, the ratio of the fuel exergy flow is taken as 1.065 (Abuşoğlu and Kanoğlu, 2009: 234-241) and the LHV value is taken as 40,420 kJ/kg (Ntziachristos et al. 2016: 456-465).

$$\xi = \frac{e_f}{LHV_f} \tag{13}$$

The total exergy destruction ($\dot{E}_{dest,t}$) can be calculated from equation (14). The overall system exergy efficiency is found by equation (15) (Hazar, 2019: 42).

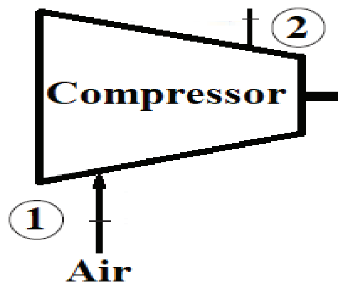
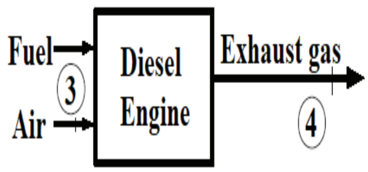
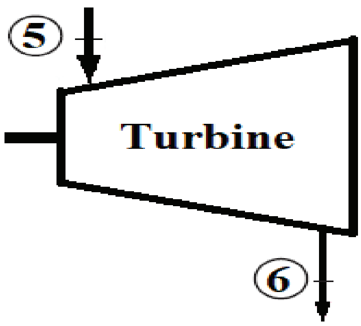
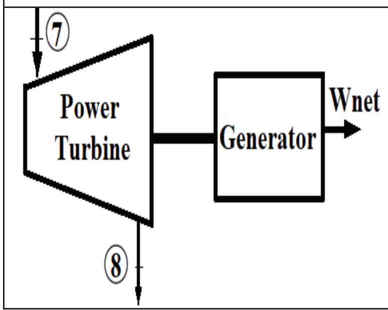
$$\dot{E}_{dest,t} = \dot{E}_{dest,comp} + \dot{E}_{dest,DE} + \dot{E}_{dest,T} + \dot{E}_{dest,PTG} \tag{14}$$

$$\varepsilon_{system,PTG} = \frac{\dot{E}_6 + \dot{E}_8 + \dot{W}_{PTG} + \dot{W}_{DE}}{\dot{E}_1 + \dot{E}_{fuel}} \tag{15}$$

The efficiency values used in the calculations are 84% for the compressor isentropic efficiency, and 89% for the turbine and the power turbine isentropic efficiency (Larsen et al. 2014: 260-268). At each engine load, 12% of the exhaust gas coming out of the combustion chamber is given to the power turbine, the rest is sent to the turbine of the turbocharger (MAN B&W, 2014a: 15).

In the equations in Table 3, T_1 is taken as the turbocharger air inlet temperature according to the ambient conditions and T_2 as the scavenge air temperature according to the engine loads. P_1 is taken as the barometric pressure and P_2 is calculated according to T_1, T_2, P_1, and k constant. k constant is 1.4 for air in the calculations. The same calculations are repeated for the compressor and the turbine at three engine loads and three ambient conditions. The abbreviations used in the tables are diesel engine (DE), turbine (t), power turbine generator (PTG), and compressor ($comp$).

Table 3: Energy and Exergy Formulas for the Components of a Power Turbine Generator

Components	Energy and exergy formulas
	$\dot{m}_1 = \dot{m}_2 = \dot{m}_a \quad \lambda_{comp} = \frac{\dot{E}_{dest,comp}}{\dot{E}_{dest,t}}$ $\dot{W}_{comp} = \dot{m}_1(h_2 - h_1)/\eta_{comp}$ $P_2 = P_1[\frac{\eta_{comp}(T_2-1)+1}{T_1}]^{\frac{k}{k-1}}$ $\dot{E}_1 = \dot{m}_1[(h_1 - h_0) - T_0(s_1 - s_0)]$ $\dot{E}_2 = \dot{m}_1[(h_2 - h_0) - T_0(s_2 - s_0)]$ $\dot{E}_{dest,comp} = \dot{E}_1 + \dot{W}_{comp} - \dot{E}_2$ $\varepsilon_{comp} = \frac{\dot{E}_2}{\dot{E}_1+\dot{W}_{comp}} = 1 - \frac{\dot{E}_{dest,comp}}{\dot{W}_{comp}+\dot{E}_1}$
	$\dot{m}_a + \dot{m}_f = \dot{m}_g \quad \dot{E}_{fuel} = e_f\dot{m}_f$ $\varepsilon_{DE} = \frac{\dot{W}_{DE}}{\dot{E}_{fuel}} \quad \dot{E}_{dest,DE} = \dot{E}_{fuel} - \dot{W}_{DE}$
	$\dot{m}_5 = \dot{m}_6 \quad \dot{W}_T = \dot{m}_5(h_5 - h_6)\eta_T$ $T_6 = T_5[1 - \eta_T(1 - \left(\frac{P_5^{\frac{k-1}{k}}}{P_6}\right))]$ $\dot{E}_5 = \dot{m}_5[(h_5 - h_0) - T_0(s_5 - s_0)]$ $\dot{E}_6 = \dot{m}_5[(h_6 - h_0) - T_0(s_6 - s_0)]$ $\dot{E}_{dest,T} = \dot{E}_5 + \dot{W}_T - \dot{E}_6$ $\dot{m}_7 = (\%12)\,\dot{m}_4 \quad \dot{m}_5 = (\%88)\,\dot{m}_4$ $\varepsilon_{turb} = \frac{\dot{E}_6+\dot{W}_T}{\dot{E}_5} = 1 - \frac{\dot{E}_{dest,T}+\dot{W}_T}{\dot{E}_5}$ $\lambda_{turb} = \frac{\dot{E}_{dest,T}}{\dot{E}_{dest,t}}$
	$\dot{m}_7 = \dot{m}_8 \quad \dot{W}_{PTG} = \dot{m}_7(h_7 - h_8)\eta_{PTG}$ $\dot{E}_7 = \dot{m}_7[(h_7 - h_0) - T_0(s_7 - s_0)]$ $\dot{E}_8 = \dot{m}_7[(h_8 - h_0) - T_0(s_8 - s_0)]$ $\dot{E}_{dest,PTG} = \dot{E}_7 + \dot{W}_{PTG} - \dot{E}_8$ $\varepsilon_{PTG} = \frac{\dot{E}_8 + \dot{W}_{PTG}}{\dot{E}_7} = 1 - \frac{\dot{E}_{dest.PTG} + \dot{W}_{PTG}}{\dot{E}_7}$ $\lambda_{PTG} = \frac{\dot{E}_{dest,PTG}}{\dot{E}_{dest,t}}$

Source: Ohijeagbon et al. 2013: 153-164; Ersayın and Özgener, 2015: 832-842; Ganjehkaviri et al. 2015: 231-243.

5. ECONOMIC ANALYSIS

Because of the additional power of waste heat recovery systems, the economic analysis shows a decrease in annual fuel consumption and fuel cost. It is highly advantageous to minimize engine running costs, as this decreases shipping costs for shipping companies, thereby enhancing their role in the competition to ship freight. Using equation (16), the specific fuel consumption (SFC) is determined, noting the engine power and the additional power of the waste heat recovery system (Mito et al. 2018: 264-276). The decrease in specific fuel consumption reflects the fuel quantity that is saved when waste heat recovery systems are applied. The amount of fuel saved per year ($AFSY$) is calculated by equation (17), the cost of fuel savings per year ($CFSY$) in equation (18), and the payback period (PP) of the power turbine generator in equation (19) (Man Diesel & Turbo, 2014: 14).

$$SFC = \frac{m_{fuel}}{\dot{W}_{DE}+\dot{W}_{PTG}} \tag{16}$$

$$AFSY = (\frac{m_{fuel}}{\dot{W}_{DE}} - \frac{m_{fuel}}{\dot{W}_{DE}+\dot{W}_{PTG}})(\dot{W}_{DE} + \dot{W}_{PTG}) \tag{17}$$

$$CFSY = (AFSY) * (HFO\ price) \tag{18}$$

$$PP = \frac{Installation\ cost}{C.F.S.Y.} \tag{19}$$

HFO price is on average \$316/ton in global markets as of December 2020 (Ship and Bunker, 2020). The power turbine generator is assumed to operate 300 days a year as a continuous service rating (Mito et al. 2018: 264-276). When calculating the specific fuel consumption, the fuel burned by the engine is taken as 40 tons/day for 75% engine load, 49 tons/day for 90% engine load, and 56 tons/day for 100% engine load (WIN GD, 2017: 26).

The installation costs of waste heat recovery systems depend on the marine engine power. The installation cost of the power turbine generator for the main engine used in this study is determined as \$1,098x10^6 (Hazar, 2019: 56).

6. RESULTS AND DISCUSSION

6.1. Thermodynamic Analysis

For the power turbine generator, three diesel engine loads and three ambient conditions have been taken into consideration, and the exergy efficiency, exergy destruction, exergy destruction rates, power outputs, efficiency, and efficiency increase in each component have been calculated. In addition, the payback period has been determined by calculating the fuel savings obtained from the power turbine generator. The values obtained from the calculations are shown in Table 4.

Table 4: Results from Power Turbine Generator

Engine load	75%			90%			%100		
Ambient conditions	Winter	ISO	Tropical	Winter	ISO	Tropical	Winter	ISO	Tropical
ε_{de} (%)	0.431	0.431	0.431	0.517	0.517	0.517	0.431	0.431	0.431
ε_{turb} (%)	0.633	0.654	0.693	0.622	0.636	0.665	0.608	0.639	0.661
ε_{comp} (%)	0.708	0.682	0.639	0.740	0.720	0.686	0.759	0.741	0.715
ε_{ptg} (%)	0.633	0.654	0.693	0.622	0.636	0.665	0.608	0.639	0.661
ε_{system} (%)	0.479	0.486	0.494	0.569	0.576	0.586	0.472	0.479	0.488
λ_{de} (%)	0.814	0.825	0.840	0.744	0.756	0.774	0.788	0.805	0.818
λ_{turb} (%)	0.086	0.075	0.06	0.124	0.112	0.094	0.108	0.092	0.079
λ_{comp} (%)	0.087	0.088	0.09	0.114	0.115	0.119	0.087	0.089	0.09
λ_{ptg} (%)	0.011	0.01	0.008	0.016	0.015	0.012	0.014	0.012	0.01
$E_{dest,de}$ (kW)	13,80	13,80	13,80	11,70	11,70	11,70	18,40	18,40	18,40
$E_{dest,t}$ (kW)	1,463	1,264	989.4	1,951	1,736	1,423	2,541	2,111	1,789
$E_{dest,comp}$ (kW)	1,487	1,485	1,489	1,795	1,791	1,800	2,040	2,040	2,040
$E_{dest,ptg}$ (kW)	200.1	172.9	135.3	266.3	237	194.3	347.1	288.4	244.3
W_T (kW)	1,686	1,376	997	2,396	2,007	1,537	3,153	2,683	2,115
W_{comp} (kW)	5,094	4,673	4,110	6,904	6,398	5,724	8,474	7,902	7,139
W_{ptg} (kW)	231	188	136	327	274	210	431	367	289
Efficiency increase (%)	0.048	0.055	0.063	0.051	0.059	0.068	0.041	0.048	0.056

- The lowest exergy destruction in the diesel engine that was obtained is 11,706 kW at 90% engine load, while the largest is 18,396 kW at 100% engine load.

- The highest exergy destruction rate in the diesel engine that was obtained is 84% at 75% engine load and tropical ambient conditions; the lowest is 74% at 90% engine load and winter ambient conditions.

- In the power turbine generator, the largest exergy destruction rate that occurs is 1.6% at 90% engine load and winter ambient conditions, while the lowest is 0.8% at 75% engine load and tropical ambient conditions.

- The highest exergy efficiency in the system is 58.62% that was obtained at 90% engine load and tropical environment conditions, while the lowest is 47.23% at 100% engine load and winter ambient conditions. Figure 5 demonstrates the exergy efficiency of the power turbine generator.

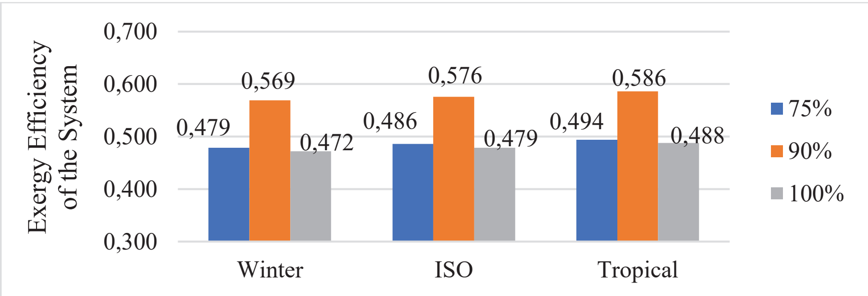


Figure 5: Exergy Efficiency of the System

- The highest efficiency is observed in tropical ambient conditions, and the lowest efficiency is seen in winter ambient conditions in all three engine loads.

- The highest efficiency increase in the system that occurs is 6.89% at 90% engine load and tropical ambient conditions; the lowest efficiency increase that occurs is 4.12% at 100% engine load and winter ambient conditions. The efficiency increase in the power turbine generator system is indicated in Figure 6. Because the exergy destruction of the diesel engine is the lowest at 90% engine load, the highest exergy efficiency and efficiency increase of the system is obtained at 90% engine load.

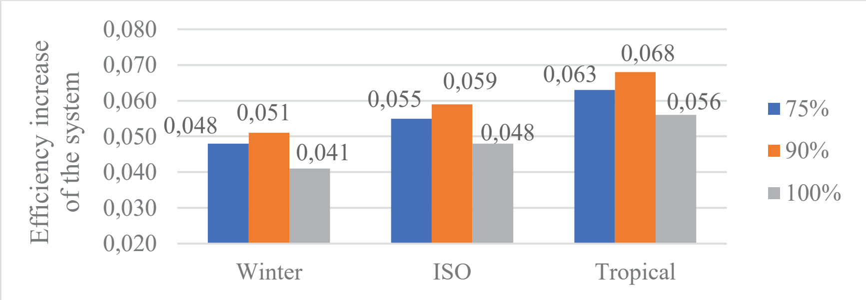


Figure 6: Efficiency Increases in the System

- The largest power output obtained from the power turbine generator is 430.8 kW at 100% engine load and winter ambient conditions, and the lowest is 136.4 kW at 75% engine load and tropical ambient conditions. The power output from the power turbine generator is shown in Figure 7. Because the mass flow rate and temperature values of the exhaust gas are highest at 100% engine load and the difference between the scavenge air and exhaust gas temperature is the highest in winter ambient conditions, the power outputs of the power turbine generator are highest in winter conditions and 100% engine load.

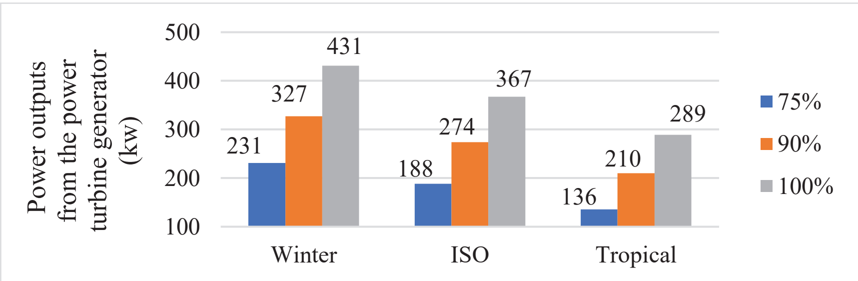


Figure 7: Power outputs from the Power Turbine Generator

6.2. Payback Period Investigation

Fuel savings calculated based on Equation (19) are shown in Figure 8. The highest fuel saving is achieved with 517 tons/year at 100% engine load and winter ambient conditions, while the lowest fuel saving is achieved with 159 tons/year at 75% engine load and tropical ambient conditions.

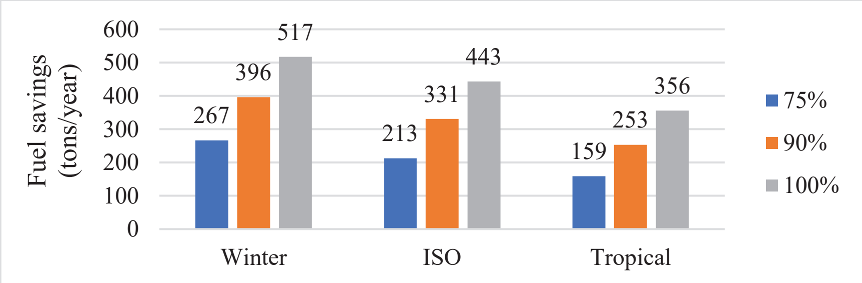


Figure 8: Fuel Savings from the Power Turbine Generator

The cost of fuel savings calculated for the power turbine generator based on Equation (20) is shown in Figure 9. The largest fuel cost saved is 163,372 USD/year at 100% engine load and winter ambient conditions, and the lowest fuel cost saved is 50,244 USD/year at 75% engine load and tropical ambient conditions.

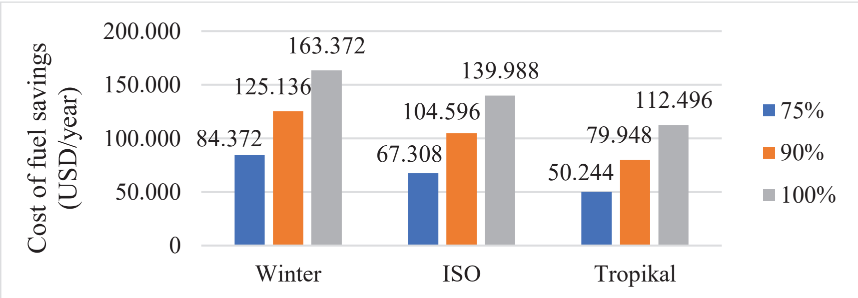


Figure 9: Cost of Fuel Savings from the Power Turbine Generator

The payback period calculated for the power turbine generator based on Equation (21) is given in Figure 10. The shortest payback period which was obtained is 6.7 years at 100% engine load and winter ambient conditions, and the longest payback period is 21.8 years at 75% engine load and tropical ambient conditions.

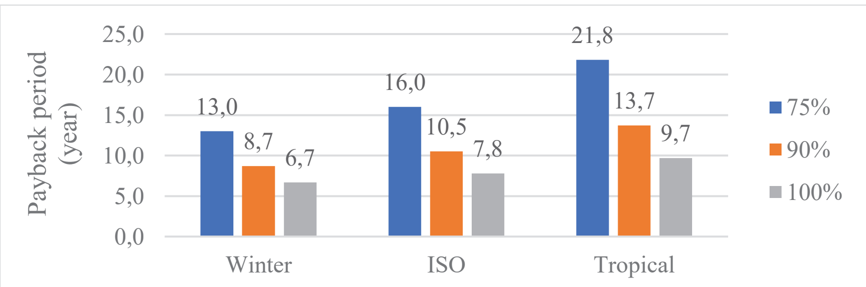


Figure 10: Payback Period of the Power Turbine Generator

7. CONCLUSIONS

In this paper, a six-cylinder low-speed marine diesel engine is used for a power turbine generator. Analyses are performed for the power turbine generator by considering three ambient conditions and three engine loads, and the exergy destruction of the system components, exergy efficiency, exergy destruction rate, efficiency increases, and power outputs of the system are calculated. In addition, the fuel savings obtained from the power turbine generator are calculated and the annual cost of the fuel saved is determined. The payback period is calculated according to the installation cost of the power turbine generator and the cost of the fuel saved annually.

The rise in the exergy destruction rate of the compressor and marine diesel engine as the ambient temperature rises and the occurrence of the largest exergy destruction rate in the marine diesel engine in all ambient conditions agree with the results of similar studies conducted before.

The results indicate that the highest power output for the power turbine generator occurred at 100% engine load and winter ambient conditions, and the lowest power output occurred at 75% engine load and tropical ambient conditions. On the other hand, for the same ambient conditions, the highest efficiency increase is obtained at 90% engine load, and the lowest efficiency increase is obtained at 100% engine load.

Annual fuel savings are determined by calculating the specific fuel consumption with the power outputs obtained from the power turbine generator, and the payback period is calculated according to the installation cost and the amount of fuel saved annually. In the power turbine generator, it is observed that the shortest payback period is 100% engine load and winter ambient conditions.

According to the results, increases in efficiency and power output obtained from the system do not occur uniformly under varying conditions. For this reason, it will be beneficial for shipowners who want to apply waste heat recovery systems to their ships to make an optimization study according to their own wishes and needs of the ship. This study will constitute a reference for ship operators according to the ambient conditions and diesel engine load of the ship.

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