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Simplified exergy analysis of ship heating systems with different heat carriers and with the recovery of waste heat

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Abstract The application of waste heat from exhaust gas of ship's main engines has become widely practiced as early as in the 1930s. Thus the increase of ship's overall efficiency was improved. Nowadays all newly built ships of the 400 gross tonnage and above must have specified energy efficiency design index, which is a measure for CO_2 emissions of the ship and its impact on the environment. Therefore, the design of waste heat recovery systems requires special attention. The use of these systems is one of the basic ways to reduce CO_2 emissions and to improve the ship's energy efficiency. The paper describes the ship's heating systems designed for the use of waste heat contained in the exhaust gas of self-ignition engines, in which the heat carriers are respectively water vapor, water or thermal oil. Selected results of comparative exergy analysis of simplified steam, water and oil heating systems have been presented. The results indicate that the oil heating system is comparable to the water system in terms of internal exergy losses. However, larger losses of exergy occur in the case of a steam system. In the steam system, a significant loss is caused by the need to cool the condensate to avoid cavitation in boiler feed pumps. This loss can in many cases cause the negative heat balance of ship during sea voyage while using only the exhaust gas boilers.

Keywords: Ship's power plant; Heating systems; Heat carriers; Exergy analysis

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Nomenclature

b_f	–	specific physical exergy, kJ/kg
$\dot{B}$	–	exergy flux, kW
c	–	mean specific heat capacity of the substance being considered, kJ/kgK
i	–	specific enthalpy in the considered working condition, kJ/kg
$\dot{m}$	–	mass flow rate, kg/s
N	–	pump power, kW
p	–	pressure, kPa
$\dot{Q}$	–	heat flux, kW
S	–	entropy, kJ/K
$\dot{S}$	–	entropy flux, kW/K
s	–	specific entropy in the considered working condition, kJ/kgK
T	–	temperature of the substance being considered, K
V	–	specific volume, m^3/kg

Greek symbols

δ	–	exergy loss
$\triangle$	–	difference
ρ	–	density
η	–	efficiency

Subscripts and superscripts

b	–	beginning of the process
c	–	circulation
cd	–	condensate
D	–	driving exergy
e	–	end of the process
eg	–	exhaust gas
f	–	physical
fe	–	feed
fl	–	fuel
fp	–	pressure part of physical exergy
fT	–	temperature part of physical exergy
in	–	inlet
m	–	mean
ol	–	oil
out	–	outlet
p	–	pump
s	–	steam
sw	–	sea water
t	–	total
u	–	useful effect
w	–	fresh water
0	–	reference state
1,2	–	upper and lower heat sources

1 Introduction

Overall efficiency of nowadays manufactured self-ignition two-stroke marine engines oscillates around 50% [1]. This means that only at most half of the chemical energy contained in the fuel is converted into useful work, while the rest as losses is dissipated in the form of heat. These losses are minimized as far as possible by using this waste heat [2].

The issue associated with the improvement of the energy efficiency of a ship through the use of waste heat is currently of special importance due to the requirements to reduce CO_2 emissions from ships [3]. One of the visible trends is the use of waste heat recovery system (WHRS) based on organic Rankine cycle (ORC), inspired by the low temperature of exhaust gas from the two-stroke low-speed engines. For example, Zheshu *et al.* [4] analyzed 3 WHRS solutions, including those based on the ORC, in order to obtain the most significant economic benefits.

The topics related to using the ORC in WHRS were also the subject of research works carried out by Mondejar *et al.* [5]. The authors presented the results of simulation of WHRS based on ORC on a passenger ship. The effects demonstrate several benefits ensuing from the application of ORC in the ship's power plants, particularly in respect of the new and restrictive exhaust gas emission regulations. The authors conclude also, that selection of design assumptions is very important for the obtained benefits.

Reis *et al.* [6] have examined the effects of using ORC in different configurations, i.e., simple one and with regeneration on the floating production storage and offloading platform (FPSO), however in combination with a gas turbine. The results indicate that the application of the ORC in WHRS, allows an average of 22.5% reduction in fuel consumption and CO_2 emissions over the lifetime of FPSO, whereas Khaled *et al.* [7] presented WHRS on liquefied natural gas (LNG) carrier with a dual-fuel engine in the aspect of the possibility of reducing fuel costs and emissions.

In the ship's power plant the main streams of waste heat are in the engine exhaust gases, in jacket cooling water, in the charge air cooling water as well as in the lubricating oil. The mentioned waste heat sources differ substantially in terms of the possibility of their use mainly due to the temperature, specific heat capacity or heat transfer coefficient from this heat carrier. Another important aspect that decides about the possibility of using this heat is the type of medium in the heating system, to which this heat is transferred [8]. A good indicator for the choice of heat source as well as the medium is exergy, which is a measure of the quality of energy associated with a given form of matter. In every real process, including process of heat transfer at finite temperature difference, there are irrecoverable exergy losses. Each loss of exergy furthermore contributes to increasing investment outlays or operating costs. Irreversible heat flow in the boiler between the flue gas and the heat carrier is, for example, one of the main reasons for the relatively low efficiency of the steam power plant [9]. Thus the exergy determines the practical heating medium energy utilization and the exergy analysis provides information on the possibility of improving the energy processes and is therefore often applied. For example, Kowalczyk *et. al* [10] used exergy analysis to evaluate the Szewalski binary vapor cycle in connection with a WHRS. Exergy analysis allowed them to evaluate the efficiency of energy conversion processes at various stages. A good example of an exergy analysis is the analysis of a low-temperature heating system using electric and absorption heat pumps, in which Sekret *et al.* [11] for the purpose of analysis divided the system into five subsystems. The authors have displayed that the exergy loses are higher in the heating system with absorption heat pump.

One of the major objectives within the ship's power plant design process is to ensure the effective utilization of the waste energy recovery in the power plants. The correct and appropriate evaluation of the waste energy sources and the heat carriers allows to complete the adequate design of ship's recovery system. In the design practice of the shipyard design offices it is common to consider the energetic analysis while leaving aside the exergetic analysis which should be taken into consideration obligatorily. For this reason, for the comparative evaluation of ship heating systems operated with different heat carriers what consists the topic of this article, the use of exergy analysis has been considered as particularly justified. The article is intended as having practical value and likely to form the guidelines for the designers of the marine power plants and contribute to the improvement of the ship's power plant design process.

2 General characteristics of ship heating systems

The heat is used on the ship to heat up mediums such as fuel, lubricating oil or water, as well as to heat rooms and possibly cargo. The main source of heat on seagoing motor ships is the waste heat in the exhaust gases from the main engines as well as in the engine and charge air cooling water. It also comes from the combustion of fuels in auxiliary boilers during shortage or lack of waste heat, e.g., when on the roadstead or in winter conditions.

The basic heat carrier used in ship heating systems is saturated steam (water vapor), which main advantage is the high specific heat capacity and constant temperature throughout the condensation process. Water or special thermal oils are less used [11]. It appears necessary to apply high pressure in order to obtain appropriately high heating temperature using such heat carriers as water and steam. The high pressure of the heating medium increases the risk of leakage occurrence and in consequence, the seepage of heating medium to other media or cargo heated, all of which in certain circumstances can cause extremely dangerous situations. Considering the hazards mentioned above, recently the thermal oils have become more and more commonly used as heating mediums since they keep their physical properties, in remarkably wide temperature range [11,12].

One of the most significant properties of thermal oil, and therefore making it more suitable medium than the steam, is the possibility to use it at low working pressure values. The thermal oil heating system can be the open type installation and then the value of these pressures depends almost exclusively on flow resistance or alternatively the closed type, under low pressure, with pressure values not exceeding $0.1-0.3$ MPa.

Due to thermal oil much higher temperatures in comparison to steam, the use of waste heat in thermal oil heating systems requires special design solutions, other than in the steam heating systems. For example, it is practically impossible to use the heat from the engine cooling water due to its relatively low temperature, which does not exceed the temperature of the oil returning from the installation to the oil heater. Subject to the thermal oil heating manner the heating installations can be divided into the installations with independent heater (oil-fired boiler) where the heat for oil heating comes from fuel burning in the burner and the installations with exhaust gas heater (exhaust gas boiler) where the heat used for oil heating comes from exhaust gas either from main or auxiliary engines.

The overall power plant efficiency is increased by the application of the combined thermal oil heating system, i.e., in the oil-fired and exhaust gas boilers. Both oil heaters can operate either separately or jointly, in the serial or parallel operating order. In the serial operation mode the low temperature exhaust gas heater comes as the first. The parallel connection is used in case when large heat amounts are necessary for cargo heating in the ship's cargo tanks. The independent heater is automatically started, if and when the heat demand exceeds the exhaust gas heater output capacity.

Water heating systems can also be of open or closed type. In the open heating system, water temperature does not exceed 373 K. Thermal expansion of water requires the use of an expansion tank, connected to the atmosphere. Such systems can be used on ships where the required heating temperatures of the working media does not exceed 333−343 K. This practically limits the possibility to use such arrangements only on the vessels whose engines are powered by marine diesel oil (MDO) or marine gas oil (MGO) which does not require heating. As a rule, these are some small inland, coastal or harbour vessels. If hot water is only used for room heating, then an open heating system can operate on the principle of natural circulation without the use of circulation pump. In a closed heating system, the expansion tank is not connected to the atmosphere and it is then a pressure system. Owing to this, it is possible to obtain temperatures exceeding 373 K. The forced circulation is always used in this type of system. As already mentioned, the disadvantage of water heating system, just like steam ones, is the need to use high pressures to obtain the temperatures necessary for heating of heavy fuel oil (HFO) to about 423 K [14]. Thermal oil heating systems with independent oil fired thermal oil heater and waste heat oil heater can completely replace steam or water systems on most vessels.

3 Exergy analysis

3.1 Basis of calculation

The exergy loss can be determined by comparing any actual mechanical machine with a corresponding ideal machine working in a reversible manner. The kind of process in question has no effect whatsoever on the result of the investigation. To both processes taking place in the devices under consideration there should be applied the second law of thermodynamics and then the sum of entropy increments of all bodies participating in the process determined. The sum of entropy increments of all bodies participating in the actual machine processes is bigger than zero whereas in the ideal machine it is equal to zero.

Also using the energy balance equations recorded for both devices the loss of work caused by irreversible operation of the actual unit can be determined. From the relations referred to hereinabove, after transformations the equation is obtained also known as the Gouy-Stodola equation [10,15]:

$$\delta B = T_0 \sum \Delta S \ . \tag{1}$$

This equation determines the exergy loss caused by the irreversible nature of the phenomena investigated. Formula (1) allows to isolate and define these installation elements which mainly affect the efficiency of energy transformation therein.

Internal and external exergy losses can be distinguished. The internal losses result from the irreversibility of the processes occurring inside the investigated device or machine whereas the external losses arise from the discharge to the surroundings of the process waste products which have a positive exergy. The external exergy in numbers equals to the exergy of the waste product discharged to the surroundings.

The most important component of the exergy in the processes under consideration is the physical exergy resulting from the difference in pressure and temperature of the medium in relation to the values occurring at a given moment in the surroundings. The physical exergy can be divided into a temperature part and a pressure part. It can be assumed that the pressure part is determined at the ambient temperature and the temperature part is calculated at the real pressure of the substance [9]. The value of specific physical exergy, b_f, is determined by the formula [15]

$$b_f = i - i_0 - T_0 \left(s - s_0 \right) \ . \tag{2}$$

When considering substance with constant specific heat capacity, c_p, the temperature part is determined from the formula [9,15]

$$b_{fT} = c_p \left(T - T_0 - T_0 \ln \frac{T}{T_0} \right) \ . \tag{3}$$

Pressure exergy term can be expressed by the formula [16]

$$b_{fp} = v_0 \left(p - p_0 \right) \ . \tag{4}$$

The changes of exergy resulting from the change in pressure values of mediums in the heat exchange processes analysed in this paper in heating systems are negligible. Therefore, in the exergy analysis of these systems, only the temperature part of exergy, b_{fT}, of mediums has been taken into account.

The sum of increment of entropy streams in the heat transfer process, $\dot{Q}$, with a finite temperature drop is defined as

$$\sum \Delta \dot{S} = \frac{\dot{Q}}{T_2} - \frac{\dot{Q}}{T_1}\,, \tag{5}$$

where T_1, T_2 are the temperature of the upper and lower heat sources respectively.

The loss of the exergy can be presented as follows:

$$\delta \dot{B} = \dot{Q}\left(\frac{1}{T_2} - \frac{1}{T_1}\right) T_0\,. \tag{6}$$

If the heat exchange occurs at changing temperature of the medium, the mean thermodynamic temperature should be used T_m

$$T_m = \frac{\Delta i}{\Delta s}\,, \tag{7}$$

where: Δi – increment of specific enthalpy of the substance when receiving or giving away heat, Δs – increment of specific entropy of the substance when receiving or giving away heat.

The exergetic efficiency of each system under consideration has been determined as the ratio of the process usable effect to the driving exergy consumption [16,17]

$$\eta_B = \frac{\dot{B}_u}{\dot{B}_D}\,. \tag{8}$$

3.2 Case study

3.2.1 Assumptions and input data

To obtain comparable results, the analysis was carried out on the example of three systems, which can be separate parts of the heating system, namely steam, water and oil respectively. In each case, the internal exergy losses in the system during the preheating of the IFO 380 fuel in the storage tank from the temperature of 315 K to 415 K have been estimated. It was assumed that there is no increase in the exergy of media in the pumps. No exergy losses caused by hydraulic friction have been included in the analysis of exergy losses either, because this would necessitate the availability of data referring to the installation length, pipe diameters, number of elbows, flow rates, etc., thus it would have been obligatory to have a technical project/design of each installation available.

The engine exhaust gas exergy stream and the electric power supplied to the pump drive have been the driving exergy in all systems. In case of steam system also the condensate cooling water exergy stream has been taken into consideration in the process driving exergy. In this system the influence of condensate throttling in the cooler and the increase of water pressure in the feed pump were omitted. It was assumed for the simplicity sake that the electric power of the pump drive is entirely transformed to the heat released into the surroundings. Therefore the pump power was handled as a loss of exergy [15]. It was assumed that the reference temperature, T_0, is equal to ambient temperature $T_0 = 300.15$ K [18]. The pump head has been assumed on the basis of the similar projects and the design practice.

Other data taken for calculations concerning the tank regardless of the type of heating system:

heat flux transferred between a heating medium and fuel	$\dot{Q} = 195.4$ kW,
fuel temperature at which heating process begins	$T_b = 315.15$ K,
fuel temperature at which heating process ends	$T_e = 332.15$ K,
fuel mean specific heat capacity	$c_{flm} = 1.9$ kJ/kgK,
fuel specific enthalpy before heating process	$i_{flb} = 74.9$ kJ/kg,
fuel specific enthalpy after heating process	$i_{fle} = 106.9$ kJ/kg.

The heat source in the steam, water or thermal oil boiler was the exhaust gas from the four stroke propulsion engine of one of the world's leading manufacturer. In each case there were the same parameters, i.e.:

exhaust gas mass flow rate	$\dot{m}_{eg} = 19.6$ kg/s,
average specific heat capacity of exhaust gas	$c_{egm} = 1.06$ kJ/kgK,
exhaust gas temperature at the inlet to the waste heat boiler	$T'_{eg} = 631.15$ K.

The specific exergy of exhaust gas on the inlet to the waste heat boiler was calculated assuming that it is handled as an ideal gas with a constant value of specific heat capacity and at ambient pressure based on formula (3).

3.2.2 Thermal oil heating systems

A simplified diagram of the thermal oil heating system is shown in Fig. 1. Data assumed for the calculation of the oil heating system or determined

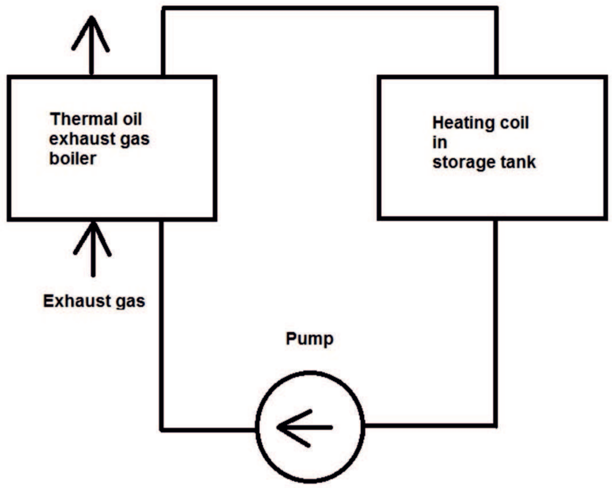

Figure 1: A simplified diagram of the thermal oil heating system.

on the basis of the exchanged heat flow between the mediums in the fuel storage tank:

thermal oil mean specific heat capacity	$c_{olm} = 2.6$ kJ/kgK,
thermal oil mean density	$\rho_{ol} = 790$ kg/m^3,
thermal oil temperature at inlet to heating coil in fuel oil storage tank	$T_{olin} = 463.15$ K,
thermal oil temperature at outlet from heating coil in fuel oil storage tank	$T_{olout} = 438.15$ K,
thermal oil mass flow rate	$\dot{m}_{ol} = 3.01$ kg/s.

The specific exergy of exhaust gas on the inlet to the waste heat boiler has been determined using formula (3) [9]. In the simplified system, considered herein, the oil temperature at the inlet to the exhaust gas boiler is equal to the oil temperature at the heating coil outlet. Thus the oil temperature at the outlet of the exhaust gas boiler is equal to the oil temperature at the inlet to the heating coil. It is analogous with the oil exergy. The average thermodynamic temperature of oil in the boiler was determined based on the formula (7), i.e., $T_m = 450.5$ K, which is the same value as in the heating coil.

The heat flux, $\dot{Q}$, is transferred from the oil to the fuel in the thermal oil system. Other losses of heat flux associated with the heating medium do not occur and therefore the same heat flux is transferred in the boiler from the exhaust gas to the thermal oil. On this basis, the exhaust gas temperature at the boiler outlet has been determined, i.e., $T''_{eg} = 621.15$ K. The power absorbed by the circulation pump in the thermal oil system was determined from the formula

$$N_{pc} = \frac{\Delta p_c}{\rho_{ol}\eta_p}\dot{m}_{ol} \ , \tag{9}$$

where: $\Delta p_c = 200$ kPa – total head, $\eta_p = 0.7$ – overall efficiency of the pump.

Calculated losses of exergy in each elements of this system are presented in Tab. 1.

Table 1: Internal exergy losses in thermal oil heating system.

Thermal heating system		
Element of the system	Internal exergy losses, kW	Relative internal exergy loses, %.
Thermal oil exhaust gas boiler	36.57	40.6
Heating coil in fuel oil storage tank	52.46	58.2
Circulation pump	1.09	1.2
Total	90.12	100.0

3.2.3 Steam heating system

A simplified diagram of the steam heating system which produces saturated steam under pressure of 0.8 MPa is shown in Fig. 2. Data assumed for the calculation of the steam heating system and determined on the basis of the *i-s* chart or on the basis of the heat flux exchanged between the mediums in the fuel oil storage tank:

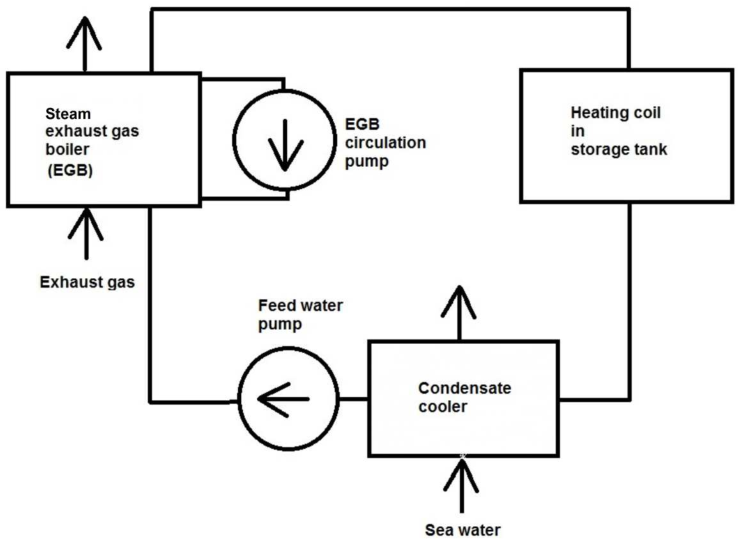

Figure 2: A simplified diagram of the steam heating system.

steam temperature at inlet to heating coil in fuel oil storage tank	$T'_s = 442.15$ K,
condensate temperature at inlet to the cooler	$T_{cdin} = 442.15$ K,
condensate temperature at outlet of the cooler	$T_{cdout} = 343.15$ K,
sea water temperature at inlet to the condensate cooler	$T_{swin} = 305.15$ K,
sea water temperature at outlet from the condensate cooler	$T_{swout} = 321.15$ K,
steam or condensate mass flow rate	$\dot{m}_s = 0.09$ kg/s,
condensate density at outlet from condensate cooler	$\rho_{cd} = 977.7$ kg/m^3,
circulation rate of the steam-water mixing in the boiler	$k = 5$.

As can be seen from the above data, the condensate temperature at the heating coil outlet is equal to the temperature of the heating steam. It means that the usually occurring small subcooling is omitted. It has been assumed, as in the case of thermal oil heating system, that the steam exergy at the inlet to the heating coil is equal to the exergy of steam at the boiler outlet, and the condensate exergy at the outlet from the cooler is equal to the exergy of feed water at the inlet to the boiler. The mean thermodynamic temperature of water and steam in the boiler has also been determined on the basis of formula (7), i.e., $T_m = 439.5$ K.

The power absorbed by the feed water pump has been determined using the formula (9) and the appropriate values of thermophysical properties for water. The following has been assumed:

total head	$\Delta p_{fe} = 800$ kPa,
overall efficiency of the pump	$\eta_p = 0.7$.

Power of the boiler circulation pump has been similarly determined, with the difference that the total head was assumed as lower, i.e., $\Delta p_c = 300$ kPa, while the mass flow, taking into account the circulation rate in the boiler, was specified as $\dot{m}_c = 0.45$ kg/s.

The calculations determined:

heat flux transferred from condensate in the cooler	$\dot{Q}_{cd} = 37.42$ kW,
and total heat flux which was transferred from the steam in the system, i.e., in the storage tank and condensate cooler	$\dot{Q}_t = 217.9$ kW.

The same stream $\dot{Q}_t$ is transferred in the boiler from exhaust gas to the steam. On this basis, the exhaust gas temperature at the boiler outlet has been determined, i.e., $T''_{eg} = 620.7$ K.

Calculated losses of exergy in each elements of this system are presented in Tab. 2.

3.2.4 Water heating system

The water heating system in terms of structure is analogous to the oil heating system. Differences in the structure of its elements result from slightly different properties of the medium. To ensure proper heating of the fuel supplied to the engine (about 413 K), a solution with a closed type pressure system has been proposed. The simplified diagram of the water heating system is shown in Fig. 3.

In addition to the previously mentioned common data, the following additional data have been assumed for the calculation of the water heating system:

Table 2: Internal exergy losses in steam heating system.

Steam heating system		
Element of the system	Internal exergy losses, kW	Relative internal exergy loses, %.
Steam exhaust gas boiler	44.23	43.55
Heating coil in fuel oil storage tank	50.01	49.24
Condensate cooler	7.03	6.92
Circulation pump	0.20	0.19
Feed water pump	0.11	0.10
Total	101.58	100.00

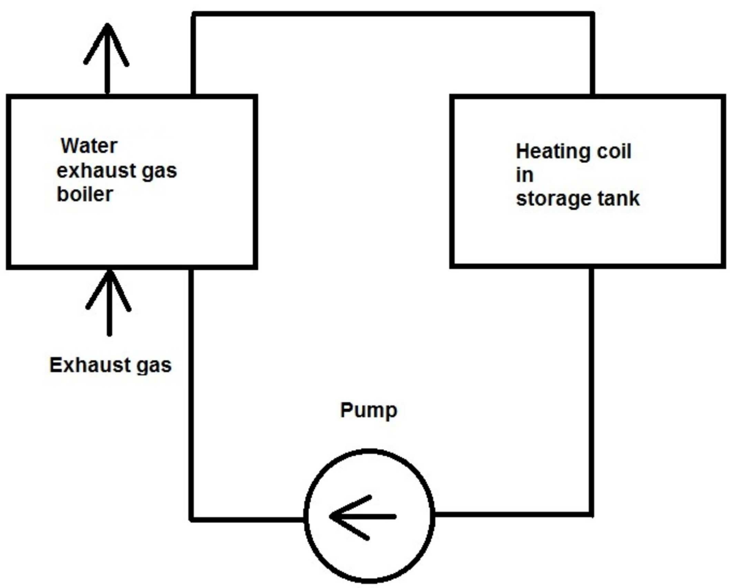

Figure 3: A simplified diagram of the water heating system.

water pressure in the heating system	$p_w = 800$ kPa,
water temperature at inlet to heating coil in fuel oil storage tank	$T_{win} = 443$ K,
water mass flow	$\dot{m}_w = 0.83$ kg/s.

In the analysed system, the water temperature at the inlet to the exhaust gas boiler is equal to water temperature at the outlet of the heating coil, and the water temperature at the outlet from the boiler is equal to the water temperature at the inlet to the heating coil. It is analogous with the water exergy.

The mean thermodynamic temperature of water in the boiler has been determined as in the previous cases and is $T_m = 411.5$ K. The power absorbed by the pump in the water heating system has also been determined using the formula (9) and the appropriate values for water. The following data have been assumed:

heating water density	$\rho_w = 897.7$ kg/m^3,
total head	$\Delta p_c = 300$ kPa,
overall efficiency of the pump	$\eta_p = 0.7$.

The heat flux, $\dot{Q}$, is transferred from water to fuel and the same heat flux is transferred from exhaust gases in the boiler to water in the water heating system. Other losses associated with the heating medium do not occur. Thus, the temperature and exergy of the exhaust gas at the boiler outlet are as in the case of the thermal oil system.

Calculated internal exergy loses in each element of this system are presented in Tab. 3.

Table 3: Internal exergy losses in water heating system.

Water heating system		
Element of the system	Internal exergy losses, kW	Relative internal exergy loses, %
Water exhaust gas boiler	48.69	54.7
Heating coil in fuel oil storage tank	39.87	44.8
Circulation pump	0.39	0.5
Total	88.95	100.0

3.2.5 Results of the analysis

The total internal exergy losses and the exergetic efficiency of the analysed systems are presented in Tab. 4. The analysis has demonstrated that the highest losses of internal exergy occur in the steam heating system mainly

Table 4: Internal exergy losses and exergetic efficiency in each heating system.

Type of heating system	Internal exergy losses, kW	Exergetic efficiency, %
Thermal oil heating system	90.12	2.91
Steam heating system	101.58	2.63
Water heating system	88.95	2.33

due to the presence of a condensate cooler. In the water and thermal oil heating systems, the losses are at a similar level. However it is clearly visible, that in the same devices, for example in waste heat boilers, the internal exergy losses are higher as the temperature difference between the heat exchanging mediums grows. The mean thermodynamic temperature of the exhaust gas in the boiler is practically the same in all systems (slightly lower in the case of a steam system), while the mean thermodynamic temperatures of the heating medium differ. The lowest occurs in the water system and the highest in the oil system.

For the assumed temperature values the highest exergetic efficiency has been achieved for the oil heating system, whereas the lowest – for the water system. The low exergetic efficiency values result from the fact that in the isolated simplified heating installations the utilization of just a very small amount of exhaust gas waste heat available for use has been assumed, thus the waste product exergy discharged to the surroundings has a large value and consists of the internal exergy loss.

4 Summary and conclusions

The exergy analysis of a selected part of the heating system showed some advantage of the thermal oil and water heating systems over the steam system. The water and oil systems themselves proved to be comparable by the assumed operational parameters. Significant positions of losses in all cases are the losses related to the irreversible heat flow, particularly in the waste heat boilers. The influence of temperature difference between the mediums on exergy losses is greater in processes running at lower temperatures. Because exhaust gas heat utilization systems involving the modern marine engines should be considered as low-temperature ones, the design of the same requires special care and attention.

In a steam system, total losses of internal exergy are the greatest which is due to additional elements such as condensate cooler and feed water pump. Apart from operational aspects, it should be noted that in the case of steam heating systems, a significant loss is caused by the need to cool the condensate outside the heater system to avoid cavitation in the boiler feed water pump. In the case of thermal oil and water systems, these types of losses do not occur, which makes them more heat efficient. The internal exergy losses in the condensate cooler are relatively small, being in the considered case about 6.9% of the total internal exergy losses in the steam system. It leads however to a higher heat consumption, which in case of exhaust gas waste heat systems from piston engines can be important. This loss may be in many cases the reason for the lack of balancing of the ship's heating system while at sea and the possibility of operating exclusively with exhaust gas boilers.

The presented simplified exergetic analysis contains a selection of the entire complex of issues related with designing the marine heating installations. A major issue is that concerning, e.g., heat exchange processes, the application of various heating mediums. The length of the heating coils, significantly affecting the engine room space, mass and volume, may largely depend on the medium applied and assumed temperature values. Although some manufacturers of the marine oil heating systems declare that for instance the heat exchange surfaces do not have to be of bigger dimensions than in the steam or water installation, if the oil temperature has been assumed as not less than 533 K [19], all the same, the selection of such temperature value renders the independent operation of oil waste heat boiler impossible. There is only a possibility of the serial mode operation together with oil fired boiler to provide additional heating for oil. However, such an arrangement would fail to provide the same fuel savings in the ship power plant as would have been provided with the parallel setup allowing the independent operation of the exhaust gas boiler during voyage at sea.

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