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A STUDY REGARDING THE USAGE OF RECOVERY BOILERS FOR OPTIMIZATION OF ENERGY FLOWS IN MARINE ENGINES

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Abstract: In recent years, there has been a large amount of waste heat released into the environment, such as exhaust gases from turbines and internal combustion engines, waste heat from industrial facilities, which lead to environmental pollution. In addition, there are also abundant solar and geothermal energy resources. These heat sources are classified as low temperature heat energy. Therefore, more and more attention has been paid to the use of waste heat for its potential in reducing fossil fuel consumption and mitigating environmental problems. The recovery of residual energy is an increasingly intense concern for building developers, especially at the European level, and in this field researchers have intensively studied the development potential of tube heat recoverers. As in the naval field, increasing energy efficiency and the use of renewable energy sources in the building sector is a priority area.
Key words: energy, recovery, boiler, engine, marine, waste, heat.

1. INTRODUCTION

Since conventional steam cycles cannot give a better performance to recover waste heat, the organic Rankine cycle (ORC) is proposed by a number of researchers for low temperature heat recovery. There are several advantages to using an ORC for waste heat recovery, including economical use of energy resources, smaller systems and reduced emissions of CO, CO_2, NO_x and other air pollutants. The main advantage of the ORC cycle is its superior performance in low temperature waste heat recovery.
In addition to ORC, researchers have proposed various thermodynamic cycles, such as the Kalina, Stirling, Ericsson cycle; to convert waste heat into electricity. Although more power is obtained for the same amount of heat with the Kalina cycle than with the ORC cycle, the organic Rankine cycle is much less complex and requires less maintenance. The organic Rankine cycle is an efficient and cost-effective method of converting waste heat into mechanical and/or electrical energy. It offers the opportunity to exploit low-temperature waste heat that would otherwise be wasted. This technology can play an important role in improving the thermal efficiency of internal combustion engines. A heat engine converts 30% of fuel energy into useful mechanical work; the rest of the energy is lost through coolant and exhaust gases, [21]. This waste heat could be recovered in order to improve the thermal efficiency of the engine and reduce the fuel consumption.
Energy consumption, mainly for buildings, today amounts to about 42% of the final energy consumption in the European Union with a high potential for energy recovery of about 22% in the short term. Currently, 35% of buildings in the EU are over 50 years old. By improving the energy efficiency of buildings, the EU's total energy consumption could be reduced by up to 56% and CO_2 emissions by around 5%. [12]
This problem of heat recovery resulting from human activities is seen more and more often from the point of view of environmental protection against "thermal pollution". Saving energy and protecting the environment require taking heat from non-conventional energy sources or thermal waste for use in various technological purposes. Thermal tube recuperators represent the most widespread application of these devices in the industrial, commercial or domestic field. The heat pipe is a heat exchanger with a small temperature gradient that found numerous applications in the most diverse fields of human activity immediately after the first publication, in 1964, of the principle of operation by the researchers of the Los Alamos Laboratory (USA). Since then, new directions of research have opened up and huge technological advances have been made in the research and improvement

of the device. The first researches were carried out in the national laboratories specialized in the development of cutting-edge technologies in the space and nuclear fields in the states with a scientific and economic potential: USA, Great Britain, France, Germany, Italy. Among the Eastern European countries where the thermal tube technology has found a favorable echo is Romania.
In a heat recovery system, the thermal tubes are placed in a beam maintained by a separation plate that divides the device into two areas, - the Evaporator - through which the primary heat-yielding agent circulates, vaporizing the working agent, and - the Condenser - in which the agent secondary receives heat from the condensing working agent. The heat input from the hot fluid, taken in the evaporation zone, is given to the cold fluid, in the condensation zone. In the adiabatic area of the thermal tubes having a relatively short length, the two heat-carrying agents are separated.
The advantages of systems of this type are:
- The thermal tube is a static heat exchange device, without moving parts, which gives it high reliability;
- The thermal tubes transform inhomogeneous heat flow distributions into homogeneous ones and behave well in variable mode;
- From a constructive point of view, they can be made in compact solutions;
- Thermal tube heat recovery systems have a high efficiency;
- Works in a wide range of temperatures;
- Can flatten temperature variations;
- Does not require additional energy sources for operation;
- A recovery system consisting of a number of thermal tubes does not interrupt its operation when a component thermal tube fails;
- The thermal flow transmitted is proportional to the number of thermal tubes in operation;
- Thermal tubes do not require internal maintenance. It is cleaned on the outside, like any other heat recovery unit.
The main disadvantages would be:
- The transferred thermal flow decreases with the increase of the transport length;
- The functional parameters of the thermal tubes can deteriorate over time due to corrosion effects inside the thermal tubes or changes in the wetting properties of the capillary structure, etc. [11]
In the field of functional installations, the use of heat recovery systems with thermal tubes has led to new concepts for the realization of installations for the preparation of domestic hot water or hot air, using recovered energy or renewable forms of energy.
The compactness and the possibility of adjusting the transferred flow recommend heat exchangers with thermal tubes for applications in the field of air conditioning and mechanical ventilation. Despite the slightly higher prices of these recuperators, it is estimated that the advantages they present will ensure a promising development.
The property of reversibility gives this type of device the quality of being able to be used both for cooling and for heating various thermal agents. Currently, various Western companies produce heat pipe batteries for air conditioning. They are competitive from an economic point of view, with heat exchangers "consecrated" in this field. At the Technical Physics Center - Iaşi, an air conditioning installation with thermal tube batteries has been operating for a long time, with an experimental title.

2. THEORETICAL ANALISYS OF THERMAL BALANCE IN MARINE DIESEL ENGINES OPERATION

The thermal balance of an engine expresses the distribution of thermal energy (available heat), between the energy equivalent to the effective mechanical work and the different losses that occur:

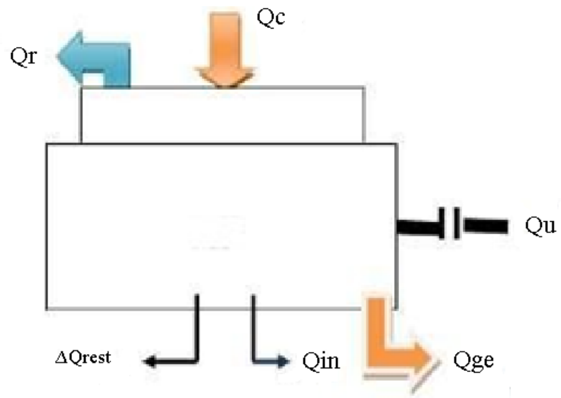

Fig. 1. Diagram of heat flows entering and leaving a piston internal combustion engine, [3]

From the analysis of the terms of the thermal balance, it results, if the thermal energy is used efficiently, corresponding to the economic operation of the engine, contributing to finding and removing the causes that lead to the uneconomical use of thermal energy. The thermal balance is useful in the calculation of the dimensions of the cooling surfaces, the calculation of the installation for the recovery of the evacuated gases, etc.
The heat balance equation is written based on the energy conservation equation (1), in the form, [3]:

$$Q = Q_u + Q_r + Q_{ga} + Q_{in} + Q_{rest} \tag{1}$$

where:
- Q_u - thermal energy transformed into effective mechanical energy (effective mechanical work), in kW;
- Q_r - the heat flow given to the cooling fluid, in kW;
- Q_{ge}- the thermal flow lost through the combustion gas, through evacuation, in kW;
- Q_{in} - the heat flux lost due to incomplete chemical combustion, in kW;
- Q_{rest} - the rest of the thermal balance, in kW.

The thermal flow, developed by the combustion of fuel in the engine cylinders, is determined by the relation:

$$Q_c = Q_i \cdot C_h / 3600 \tag{2}$$

The heat flow lost through the cooling fluid, in liquid-cooled engines, is determined by measuring the flow of cooling water in kg/s, and its temperature at the entrance and exit of the engine, in ºC. The relation for calculating the heat flow lost by cooling the engine is:

$$Q_{rac} = m_{ap} \cdot c_p \qquad (t_i - t_e) \tag{3}$$

where:
- m_{ap} - is the cooling water flow, in kg/s;
- c_p - average specific heat of water, in kJ/(kgK);
- $(t_i - t_e)$- water temperatures at the entrance, respectively at the exit from the engine, in ºC.

The thermal flow lost through the evacuated gases is determined by making the difference between the enthalpy of the evacuated burned gases and the enthalpy of the fresh load entering the engine; this is because when filling the cylinder, the fresh load has an enthalpy that does not enter into the first member of the heat balance equation:

$$Q_{ge} = m_{ge} \cdot c_{pg} (t_{rae} - t_a) \text{ [Kw]} \tag{4}$$

where:
- m_{ge} - is the flow rate of evacuated gases in kg/s;
- c_{pg} - specific heat of exhaust gases and air, in kJ/(kg.K);
- d_{raw} - the temperature of the gases in the exhaust connection, in ºC;
- t_a - inlet air temperature, in ºC.

The gas flow is determined from the relation:

$$m_{ge} = w_{ge} \cdot A \cdot \rho \tag{5}$$

where:
- A - is the area of the exhaust pipe section, in m^2;
- w_{ge} - speed of gases in the exhaust system, in m/s;
- ρ - the density of the evacuated gases, in kg/m^3.

The combustion gas flow rate can also be determined on the basis of the fuel combustion calculation, taking into account the elemental composition of the fuel and the excess air coefficient of the engine. The thermal flux lost due to incomplete chemical combustion is determined by analyzing the evacuated gases and determining in this way the components that still contain chemical energy. Incomplete combustion can occur not only for an air excess coefficient $\lambda > 1$, but also in the case when fractions of the fuel burn incompletely due to the imperfection of the mixture and combustion, [3]. The calculation of this lost energy is done with the relation:

$$Q_{in} = \Sigma\, V_{ai} \cdot Q_{inf} \text{ [Kw]} \tag{6}$$

where:
- V_{ai} - is the amount of unburned components in the exhaust gas, in m^3N/s;
- Q_{inf} - the lower calorific value of the unburned gaseous components of the exhaust gas, in kJ/m^3N.

The rest of the heat balance is determined as follows:

$$Q_{rest} = Q_c - (Q_u + Q_r + Q_{ge} + Q_{in}) \tag{7}$$

The heat lost by cooling the engine presents several aspects, which depending on the type of engine and mode of operation can be calculated or estimated based on the experience of operating the engines. Thus, for engines used in this type of industrial applications, the heat lost by cooling the engine must include:
- The heat transmitted to the cylinder cooling water, which is calculated with the formula:

$$Q_{r.\,cyl.} = m_{ap} \cdot c_p \,(t_i - t_e) \tag{8}$$

- The heat transmitted to the water (oil) for cooling the pistons, which is calculated with the relation:

$$Q_{r.pis.} = m_{ap} \cdot c_p \,(t_i - t_e) \tag{9}$$

- The heat evacuated with the lubricating oil, which is calculated with the formula:

$$Q_{ul.} = m_{ul.} \cdot c_{p.u}l. \,(t_i - t_e) \tag{10}$$

- The heat transmitted by convection from the hot surfaces of the engine, which is calculated with the relation:

$$Q_{conv.} = \Sigma\, S_{conv}\, \alpha_i \,(t_p - t_a) \tag{11}$$

where:
- S_{conv}. - the sum of the hot surfaces of the engine of the same temperature;
- α_i - the convection coefficient from the warm surface to the external environment;
- t_p - temperature of the warm surface, in ºC;
- t_a - the air temperature in the car compartment.

The necessary size of the thermal balance for a certain engine operating mode is obtained from direct measurements, having the engine mounted on a stand, and for large propulsion engines, some measurements can be made for some engine values (maximum 50%) load. Measurements for the balance of compression power generating engines can be made with equipment and apparatus corresponding to the equipment in operation.

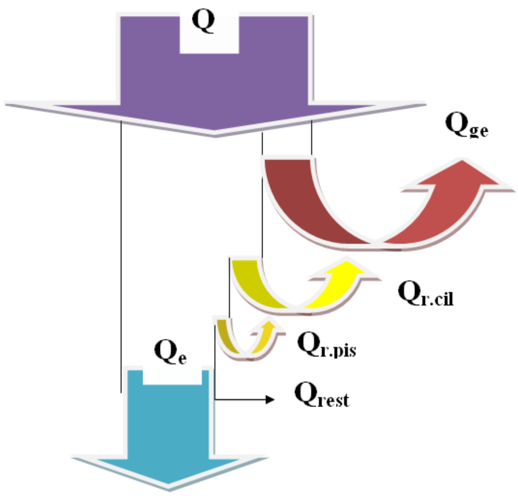

Fig. 2. Diagram of the heat flows out of the heat balance contour of the engine, [2]

3. RECOVERY OF THERMAL ENERGY OF THE EXHAUST GASES

3.1. Usage of gas turbine turbogenerator engine

The simplest and cheapest system consists of a turbine that works with exhaust gas (so-called power turbine) installed on a bypass of the gas route, and connected in turn to an electric generator that provides electricity. For more solutions for the power turbine, the engine manifold will be equipped with 2 exhaust gas connections, one for the engine exhaust gas bypass (EGB) and one for the power turbine.[14] The connection for the power turbine must be of a larger diameter so that the power turbine unit is positioned a few meters behind the main engine in the engine room.

The bypass of the evacuation route together with the control valve come from the manufacturer together with the engine and are an integral part of it. The power turbine and the generator are positioned on the same stand. TCS - PTG compound system (turbo compound system - power turbine generator, turbine - generator system) are

produced by MAN Diesel. The power turbine is driven by a part of the exhaust gases that bypasses the turbocharger.
The turbine produces the mechanical work necessary for the production of electrical energy, which depends on the volume of gas bypassed. The solution to recover the lost thermal energy can work only with the recovery boiler or in parallel through the production of electrical energy. The bypass valve will be closed at a power of the main engine lower than 40% of the nominal power. Using this TCS-PTG solution, increase your recovery rate by 3-5%.

3.2. Steam turbine and gas turbine engine equipment

This system is also built on the by-passed principle of exhaust gases. If the electrical energy demand of the compression system is very high, the gas turbine and the steam turbine can be built together in a combined system. The gas turbine and the steam turbine are placed on a common pedestal and are connected to a generator through a reducer. The mechanical work performed by the gas turbine drives the generator through a reducer with a special coupling system, [13]. The steam turbine will start at 30-35% prime engine power, then it is followed by the gas turbine at 40-50% prime engine power. This system also reduces fuel costs and is able to cover the demand for electrical power. Also a shaft motor/generator connected to the motor can be an option for electricity production.
In the operation of the recovery boiler, the following specific aspects are worth noting:
a) due to the moderate temperature of the gases, the heat transfer is realized only by convection;
b) in order not to unnecessarily increase the heat exchange surface, the temperature difference between the gas inlet and outlet of the recovery boiler must not exceed 30...40^oC;
c) the temperature of the burnt gases, at the exit from the boiler, must not be lower than 160...170 ^oC, in order to avoid the condensation of water vapors and therefore the formation of sulfuric acid (especially in the case of using heavy fuel);
d) the gas-dynamic resistance of the boiler must not exceed 250mmcol H_2O for engines in 2^o and, respectively, 400 mmcol H_2O, [7] for engines in 4^o;
e) the operation of the recovery boiler is efficient only in the case of normal engine operation modes;
f) the minimum temperature of the burnt gases at the entrance to the boiler must not be lower than 200...250 ^oC.

3.3. Steam turbine, power turbine and generator (ST-PT) system

If the need/consumption of electrical energy is very high, the compressor plant, the power turbine and the steam turbine can be built together to form a combined energy system. The power turbine and the steam turbine are mounted on a common base plate, being connected to each other through a reduction-type drive system, which are connected to the same generator.
The amount of energy generated through the power turbine can be added to the amount of energy generated by the generator through a reducer or through a special transmission box. However, under the system, the steam turbine will start first when the main engine is operating at a rated load of 30 – 35% of SMCR, then the power turbine will start at a rated main engine load of 40 – 50% of SMCR.
The combined WHRS ST and PT scheme can be applied, and this leads to the creation of a system that, under normal operating conditions, reduces fuel consumption considerably, being able to cover the total electrical energy requirements of the state.[10-12] Using a complete WHRS system – which combines both power turbines and steam turbines – leads to the recovery of 8 – 11% of the total power, and this depends on the size of the main engine and the environmental conditions. As a general rule, it is recommended to choose a waste heat recovery system, as follows:
- Main engine power > 25000 kW – a combined ST and PT system;
- Main engine power < 25000 kW – a ST or STG type system (with superheater);
- Main engine power < 15000 kW – a PTG or ORC (organic Rankine cycle) system,[5].

4. PRESENTATION OF ENERGY ECONOMY THROUGH THE DIMENSIONAL CALCULATION OF THE RECOVERY BOILER

The data obtained through this calculation is essential to establish the component elements within the system framework. Further, on the basis of the same calculation, the technical water generator that serves this installation and for which the operation optimization study will be carried out will be established.
As mentioned in the paragraphs above, by using a residual energy recovery system, the following amounts of electrical energy can be obtained, depending on the type of WHRS system used:
- TCS-PTG systems: 3 – 5%;
- STG systems – simple pressure system: 4 – 7%;
- STG systems – double pressure system: 5 – 8%;

- Complete WHRS system (ST – PT): 8 – 11%, [3].

4.1. Thermal calculation for the boiler
Theoretical gas enthalpy at the boiler entrance, [4]:

$$V_{R02} = V_{CO_2} + V_{SO_2} \tag{12}$$

$$V_{R02} = 1.56\ [m^3N]$$

for $t_{gi} = 397^oC$**:**

$$i_{R01}(t_{g1}) = 157.524\ [KJ/Kg]$$
$$i_{H_2O}(t_{g1}) = 621.098\ [KJ/Kg]$$
$$i_{N_2}(t_{g1}) = 254.206[\ KJ/Kg]$$

for $t_{ge} = \mathbf{82^oC}$:

$$i_{R02}(t_{ge}) = 660.850\ [KJ/Kg]$$
$$i_{H_2O}(t_{ge}) = 276.196\ [KJ/Kg]$$
$$i_{N_2}(t_{ge}) = 19.545[\ KJ/Kg]$$

Calculation of the flow of heat taken from water from the boiler route (available heat)

$$Q_d = Q_i - Q_{ge} = 3569300.844[\ KJ/h] \tag{13}$$

Calculation of useful heat flow:

$$Q_u = \eta_c - Q_d = 2054301.481[\ KJ/h] \tag{14}$$

Calculation of the flow rate of steam that circulates through the boiler:

$$i_{a_{si}} = 3052.2[KJ\ /\ Kg]$$
$$i_a = 185.02[KJ/Kg]$$

$$D_{ab} = \frac{Q_u}{i_{a_{si}} - i_a} = 709.583[Kg/h] \tag{15}$$

- a_{si} **-** is the enthalpy of the steam at the exit from the superheater;
- i_a **-** is the enthalpy of the water entering the economizer.
Gas flow through the boiler
- $\rho_{gm} = 0{,}903\ [kg/m^3N]$ – exhaust gas density;
- $v_g = 5\ [m/s]$ – average speed for exhaust gases.

$$i_{gi} = \frac{ig_i}{\rho_{gm} \cdot V_g} = 2672.379\ [Kj/Kg] \tag{16}$$

$$i_{gi} = \frac{ig_i}{\rho_{gm} \cdot V_g} = 2672.379\ [Kj/Kg] \tag{17}$$

$$i_{ge} = \frac{ig_e}{\rho_{gm} \cdot V_g} = 2018.822\ [Kj/Kg] \tag{18}$$

4.2. Thermal calculation of the superheater
Preliminary calculation of the heat exchange surface:
The global thermal transfer coefficient,

$$k = 35\ [W/m^2K]$$

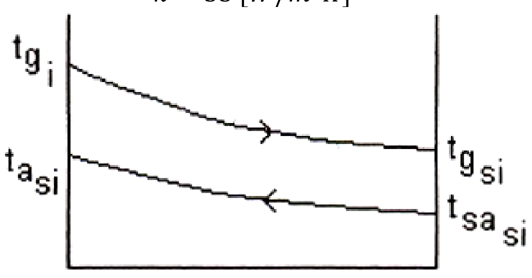

Fig. 3. Temperature variation diagram

- tg_i - gas inlet temperature in the superheater;
- tg_{si} - temperature after superheater;
- ta_{si} - the temperature of the steam at the exit from the superheater;
- tsa_{si} - the temperature of the steam at the superheater entrance, determined as a function of P_{steam} (saturation temperature).

$$\Delta t_{min} = t_i + ta_{si} = 95\ [°C] \tag{19}$$

$$\Delta t_{max} = tg_{si} + tsa_{si} = 95\ [°C] \tag{20}$$

The logarithmic mean temperature is:

$$\Delta t_m = \frac{\Delta t_{max} - \Delta t_{min}}{\ln\left(\frac{\Delta t_{max}}{\Delta t_{min}}\right)} = 84.955\ [°C] \tag{21}$$

Heat flow calculation:

$$Q_i = \frac{D_b}{\eta_c} \cdot (ia_{sie} - ia_{sii}) = 432621.401\ [KJ/h] \tag{22}$$

- ia_{sie} - the enthalpy of the steam at the exit from the superheater;
- ia_{sii} - enthalpy of steam at the superheater entrance.

Calculation of thermal transfer surface

$$A_{si} = \frac{Q_{si} \cdot 10^3}{k \cdot \Delta t_m \cdot 3600} = 47.548\ [m^2] \tag{23}$$

Calculation of the heat transfer surface for the bundle of four pipes of the coil:

Calculation of the number of nerves:

$$n = L_{si}/p_n + 1 = 254.333 \tag{24}$$

The value chosen is $n = 255\ nerves$.
- $L_{si} = 3{,}8\ [m]$ - the length of the coil;
- $p_n = 0{,}015\ [m]$ - the step between the nerves.

Calculation of the A_n surface:

The total heat exchange surface consists of the ribbed surface and the surface of the free part. The calculation of the surface of the nerves A_n will be done adopting the following conditions, [7]

$$A_n = n\left(a \cdot b - \frac{\pi d_e^2}{4}\right) = 5.75[m^2] \tag{25}$$

Tube total number calculation:

$$n_t = \frac{A_{si}}{A_f} \cdot 4 = 13.317 \tag{26}$$

Thus: $n_t = 14\ tube\ bundles$

Calculation of the real thermal transfer coefficient

Steam circulation speed:

$$wa_{sio} = \frac{D_{ab}}{A_{si} \cdot \rho g_{sim} \cdot 3600} = 22.374\ [m/s] \tag{27}$$

Gas speed circulation calculation:

$$wg_{si} = \frac{D_g}{Ag_{si} \cdot \rho g_{sim} \cdot 3600} = 0.536\ [m/s] \tag{28}$$

$$Ag_{si} = L_i \cdot 3 - 6{,}83 = 4.58\ [m^3] \tag{29}$$

Determination of the real heat transfer coefficient:

$$\alpha_2 = 0.023 \frac{\lambda a_{sim}^{m}}{d_e \cdot 0.00065} \cdot \left(wa_{si} \frac{d_e}{va_{sim}} \right) \cdot P_{ra}^{0.4} \tag{30}$$

4.3 Thermal calculation of the economizer

The water preheater is built from a pipe system similar to that of the superheater in which the feed water is heated due to the combustion gases that are recovered. To simplify the calculation, it is considered that the water in the economizer is brought to the boiling temperature. The contrary provision will be used. Preliminary thermal calculation of the heat exchange surface depends on the global heat transfer coefficient – $k = 35 \left[\frac{W}{m^2 K}\right]$

4.4 Calculation of the difference of the logarithmic mean of the temperature

$$\Delta t_m = \frac{\Delta t_{max} - \Delta t_{min}}{\ln\left(\frac{\Delta t_{max}}{\Delta t_{min}}\right)} \tag{31}$$

Where:

- $\boldsymbol{tg_e}$ - gas temperature after the economizer;
- $\boldsymbol{t_a}$ - water temperature at the economizer inlet.

$$P = \frac{ta_{si} - tsa_{si}}{tg_i - tsa_{si}} = 0.571 \tag{32}$$

$$R = \frac{tg_i - tsa_{si}}{ta_{si} - tsa_{si}} = 0.341 \tag{33}$$

Thus: $\psi = 1{,}33$.

Heat flow calculation:

$$ia_{ec.e} = 719.2 [KJ/Kg]$$
$$ia_{ec.i} = 185.02 [KJ/Kg]$$

- $ia_{ec.e}$ - the enthalpy of the steam at the economizer exit;

$ia_{ec.i}$ - the enthalpy of the steam at the economizer entrance.

$$Q_{ec} = D_{ah} (ia_{ec.e} - ia_{ec.i}) = 379044.916 [KJ/Kg] \tag{34}$$

Calculation of the heat transfer surface for a beam:

$$A_{ec} = \frac{Q_{ec} \cdot 10^3}{k \cdot \Delta t_m \cdot 3600} = 52.185 [m^2] \tag{35}$$

- $p_n = 0{,}015 [m]$ - the step between the pipes;
- $L_{ec} = 4$ [m] - the length of the coil.

$$n = \frac{L_{ec}}{p_n} + 1 = 267.667 \tag{36}$$

Thus n = 270 [m].

Calculation of the number of nerves for a bundle of pipes:

$$n_{ec} = \frac{A_{ec}}{A_f} \cdot 4 = 14.616 \tag{37}$$

4.5. Calculation of the thermal transfer coefficient

Physical parameters of steam from the vaporizer:

- $ta_m = 109.677 [°C]$;
- $pa_{ecm} = 943.574 [Kg/m^3]$;
- $ca_{ecm} = 4.229 [KJ/KgK]$;
- $\lambda a_{ecm} = 68.485 \cdot 10^{-2} [W/mK]$;
- $va_{ec.m} = 0.274 \cdot 10^{-6}$;
- $Pr_{aec} = 1606$;
- $V'' a_{vm} = 0.208 [m^3/Kg]$

Physical parameters of combustion gases:

$$tg_m = \frac{tg_v + tg_e}{2} = 192 [°C] \tag{38}$$

For this value the following parameters will result:
- $\rho g_{ecm} = 0.762 [Kg/m^3]$;
- $cg_{ecm} = 1.1904 [KJ/KgK]$;
- $\lambda g_{ecm} = 3.950 \cdot 10^{-2} [W/mK]$;
- $vg_{ecm} = 32.001 \cdot 10^{-6}$;
- $p_{rgec} = 0.0671$;
- $V''g_{ecm} = 0.935 [m^3/Kg]$

Steam circulation calculus:

$$wa_{ec} = \frac{D_b}{A_{ec} \cdot \rho a_{ecm} \cdot 3600} = 1.498 \ [m/s] \tag{39}$$

Gas circulation calculus:

$$A_{ec} = L_v - B - (A_1 - A_2) = 2.129 [m^2] \tag{40}$$

$$wg_v = \frac{D_g}{A_v \cdot \rho g_{ec} \cdot 3600} = 1.49 \ [m/s] \tag{41}$$

Real thermal coefficient calculation:

$$m = 0.01, \qquad \omega = 0.95$$

$$k_r = \frac{1}{\frac{1}{\omega \cdot \alpha_c} + m + \frac{1}{\alpha_2}} \tag{42}$$

$$\alpha_c = \alpha_{cn} \cdot c_f \cdot c_2 \tag{43}$$

$$\alpha_c = 28.643 [W/m^2K]$$

$$\alpha_{cn} = 0.2 \cdot c_f \cdot C_2 \frac{\lambda g_{ecm}^m}{d_e} \cdot \left(wg_{ec} \frac{d_e}{vg_{ecm}} \right)^{0.65} \cdot P_{rg}^{0.4} \tag{44}$$

$$\alpha_{cn} = 71.035 [W/m^2K]; \ \alpha_2 = \alpha_{2n} \cdot C_d; \ \alpha_2 = 117.063 [W/m^2K]$$

- α_{cn} - can be chosen from the nomogram depending on Wg;
- m - the dirt coefficient;
- α_c - thermal convection coefficient from the gas to the wall;

$$k_r = \frac{1}{\frac{1}{\omega \cdot \alpha_c} + m + \frac{1}{\alpha_2}} = 34.543 \ [W/m^2K] \tag{45}$$

The actual calculation of the total transfer surface:

$$A_u = n_{cv} \left(a \cdot b - \frac{\pi \cdot d_i^2}{4} \right) = 9.543 \ [m^2] \tag{46}$$

Total heat exchange surface

$$A_{ec} = 12.85 \ [m^2]$$

5. PRESENTATION OF ENERGY ECONOMY THROUGH THE DIMENSIONAL CALCULATION OF THE RECOVERY BOILER

In order to make the energy flows from the compression station more efficient, a recovery boiler can be installed on the evacuation line of the marine engines. The fuel cost ratio of the operational cost of the ship energy system a threshold close to the value of 50%, in the context of the increase in the price of fuel, but also of the cost of the operational flow in the specific field of operation, [21]. As a response, the reduction of fuel consumption and the efficiency of the operation of the main engine, as well as part of the total operating cost of the specific installations, has become one of the main objectives for the company that owns the ship. For this reason, they appealed to the technical proposal to use a combined system with a turbo blower and recuperative boiler driven by electric motors. Moreover, four-stroke dual-fuel engines tend to have a higher exhaust gas temperature when operating under

reduced load conditions, which can lead to reduced reliability, increased fuel consumption, and more smoke.
In their initial configuration, in view of the efficiency of the four-stroke dual fuel engines, they are equipped with two recovery boilers, driven by electric motors. The suction part of the recovery boiler is connected directly to the exhaust gas exhaust system of the engine that drives the compressor. A one-way valve is mounted between the air cooler and the flue gas receiver, which closes automatically when the recovery boiler is supplied with air or flue gas from the exhaust.
Both recovering boilers normally come into operation before the main engine starts, delivering sufficient working pressure of the burnt gases to ensure safe starting of the four stroke fuel engine.
During the operation of auxiliary engines both recovery boilers will start automatically whenever the engine load falls between 30% - 40% and will continue to operate until the engine operating load returns to 40% and climbs to 40% - 50% of the nominal load value of the main engine, [18].
In general, the starting controls of electrically operated recovery boilers depend on the pressure developed by the burnt gases in the gas receiver of the fuel engine, taking into account the correct unit of measurement. These values must be set by the main engine manufacturer, but must be allowed to be adjusted according to the technical characteristics of the auxiliary fuel engine.
The recovery boiler is used together with a supply point. It is made of carbon steel. The fan is driven by the joint and the telescopic shaft of the dual-fuel engine. A manifold with 8 or 12 inlet connections can be mounted on the fan.
The recovery boiler must be started during the main engine checks. It is recommended to check the current consumption when starting the group and the current consumption when the recovery boiler reaches the maximum operating speed. The operation of the rotation system must be silent, without strong vibrations. The temperature rises in the sealing areas, as well as that of the electric motor, must not exceed 50 degrees Kelvin, and the temperature of the bearings must not exceed 60^0K, these values being a function of both temperatures. [25] A check of the operation of the phase breakers in the event of a power supply interruption on one phase must be done. If the results presented by the recovery boiler are satisfactory, then it will remain in operation.
Distribution boxes and control boxes must be installed in places away from water and other solid materials that can break these boxes, in an area where the degree of protection indicated by the manufacturer is not exceeded. The distribution and control boxes must be mounted in such a way as to ensure easy access for carrying out inspections and maintenance work. These boxes must not be installed in areas close to a strong heat source, as this may generate an exaggerated increase in the temperature at which they are designed to operate.

6. CONCLUSIONS

The waste heat recovery systems present on maritime transport ships are able to recover only a small part of the waste heat, this not being enough to provide the entire amount of thermal and electrical energy required for the ship's auxiliary equipment. They mainly recover only a certain part of the thermal energy contained in the exhaust gases and the cooling water of the main and auxiliary engines.
The recovery of thermal energy from the combustion gases is done through a recovery boiler, which further produces saturated or superheated steam required for the various installations on board the ship.
From the heat contained in the cooling water of the main and auxiliary engines, fresh sea water is obtained in the desalinizer.
Sometimes part of the heat contained in the cylinder cooling water is also recovered by preheating the feed water of the recovery boiler mounted on the flue gas exhaust pipes. All these techniques ensure low recovery performance.
Analyzing both the pressure and temperature values of the steam produced in the heat recovery system, it can be observed that the caloric capacity ratio C_g/C_c can reach values in the range of 0.07...0.25.
If the expansion vaporization technique (water expander) is used, a greater amount of heat can be recovered from the cylinder cooling water.

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