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Universitatea Maritimă din Constanţa

Facultatea de Electromecanică Navală

Departamentul de Inginerie Mecanică în Domeniu Mecanic şi Mediu

STUDII ŞI CERCETĂRI PRIVIND SISTEMELE DE RECUPERARE A ENERGIEI REZIDUALE LA NAVELE MARITIME

STUDIES AND RESEARCH ON WASTE ENERGY RECOVERY SYSTEMS FOR SHIPS

REZUMAT în limba engleză

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Content

Chapter 1. Introduction

The paper entitled ***"Studies and research on waste energy recovery systems for ships"*** is structured in seven main chapters and conclusions, as follows: in the first chapter, will present the overview of the importance and objectives of the thesis, in the second chapter will present issues on the current state of implementation of energy technologies in ships, the third chapter will deal with the notions of energy efficiency for the oil tanker taken as a reference, and the fourth chapter will deal with supercharging as a basic process of main engine operation; the fifth chapter will deal with the production of technical water on board the ship as a final element regarding the use of the energy flow of the cooling water of the main engine, and in chapter six it refers to the mathematical model of the energy flow obtained in the economizer with the help of exhaust gases and will with captures the study of its behavior with the help of the CFD module (Computational fluid dynamics) within Ansys Fluent, as well as the experimental validation by performing measurements on the actual installation on board the reference ship in chapter seven. Chapter eight will present the final conclusions of the thesis, personal contributions, development directions and dissemination of research results.

The thesis thus elaborated is **important**, because it studies a way to improve the efficiency of the entire ship by optimizing the on-board installations and obtains an experimental validation on the efficiency for an oil tanker of 300000 dwt.

The thesis is also **opportune** because, in the current context, when international standards impose a series of restrictions in terms of energy efficiency that must be respected, the paper opens new perspectives on how to streamline ships.

This paper will focus on the following major objectives:

1) The current state of international norms on ship efficiency;
2) The current state of efficiency methods regarding merchant ships;
3) Energy analysis of residual energy flow recovery systems for the 300,000 dwt oil tanker;
4) Analysis of the impact of supercharging of the main engine on the efficiency of the ship;
5) Analysis of the efficiency of the ship by studying the energy flow of cooling water;
6) Modeling and studying the energy transfer using the CFD (Computational Fluid Dynamics) module within Ansys Fluent in the recovery boiler;
7) Validation of the results by comparison with the results obtained by a 1: 1 scale experiment.

Chapter 2. Current state of energy efficiency optimization in shipping

A ship needs fuel for both travel and on-board operations. In the most general case, the fuel is converted on board the ship into energy in the form required for its end use: mechanical power for propulsion, electrical power for auxiliary board systems and thermal power for thermal needs.

A ship is built and operated for a specific purpose, which varies from ship to ship (for example, for freight, passenger or military transport etc.). To achieve its intended purpose, a ship must be able to perform several functions in addition to propulsion. These can range from carrying out on-board activities in a safe environment to providing accommodation facilities for on-board crew.

In general, the energy required on board the ship is divided into three categories:

- propulsion power: the movement of the ship is influenced to a large extent by the resistance resulting from friction with water and to a small extent with air; this resistance is directly influenced primarily by the speed of the ship, but also by the characteristics of the ship's hull (shape of the ship's hull, its condition, wet surface etc.); external factors, such as the deposition of various species of organisms on the living work of the ship, as well as unfavorable weather conditions, directly influence the need for greater propulsion power;
- auxiliary electrical power: many on-board units require electricity to operate. Some of them are found on all types of ships as basic elements in the correct operation, such as navigation equipment, cooling and lubrication pumps, compressors in air conditioning systems, ballast pumps, lights etc; we also have specific equipment depending on the type of each ship, such as the inert gas installation for oil tanks, refrigerated containers for container vessels, cargo loading pumps (also for oil tanks);
- auxiliary thermal power: heat is generally required in three important directions: crew comfort, fuel heating, and technical water generation; in a similar way to auxiliary electric power, various types of ships require a certain amount of heat, as in the case of oil tanks (for heating low-viscosity cargo) or the case of passenger ships, for the comfort of those on board.

Chapter 3. Notions regarding the analysis of the energy systems of the ship. Elements of calculation

To correctly understand the requirements of a system, we must first study its schematics. In the case of energy systems, they require a detailed diagram of energy performance. In addition to the standard data analyzed, energy and exergetic analysis is required. When dealing with energy flows of any kind, the singular analysis of energy can lead to inconclusive results, not taking into account the quality of energy. This problem can be partially solved by exergy analysis. In the ship, exergy is defined as the maximum load of an aggregate coming from the mechanical system in a specific reference environment.

The exergy content of a system depends on the quality of the energy contained. Moreover, unlike energy, exergy is not conserved and can be destroyed, representing the deterioration of energy quality.

The exergic content of a material flow is generally divided into four parts: physical, chemical, kinetic and potential. Potential and kinetic exergy flows coincide with their energy counterparts. In the case of chemical exergy, substantial differences can be found when analyzing systems involving more advanced chemistry; in this case, combustion is the only chemical reaction considered.

Heat losses can be classified into high - temperature losses, medium - temperature and low - temperature losses. On board ships with low-speed propulsion engines, heat recovery systems operate in the range of 100–400 °C. Many residual energy recovery systems from internal combustion engines are under development: for example the MAN WHR development program for Tier III ternologies.

The chapter dealt with the energy analysis of the VLCC tanker, based on real measurements on board the ship. The energy analysis was used to calculate the ship's potential to recover the lost heat energy.

Fig. 3.1 *The main engine of the VLCC*

Recovery of residual heat is welcome on board the ship, but its potential may vary depending mainly on size, number, use and efficiency on board. Moreover, these sizes are usually irrelevant for the adaptation of equipment to old installations, due to high costs, redesign efforts, difficult activities such as welding, excess weight etc.

The energy balance for the VLCC tanker showed that the residual thermal energy has a range between 1.01 - 4.08% of the total energy flow of the engine at load between 50 - 100%.

$$Q_w = Q_{intr} - (Q_u + Q_{coolpist} + Q_{cooloil} + Q_{coolinj} + Q_{ex}) \quad (3.1)$$

Tab. 3.5 *Parameters regarding the energy balance*

Engine speed [rpm]	**Engine loading [%]**	**Engine power [kW]**	Q_{intr} **[KJ/h]**	Q_u **[KJ/h]**	Q_{cool} **[KJ/h]]**	Q_{ex} **[KJ/h]**	Q_w **[KJ/h]**	Q_w **[%]**
76	100	27,020	$2.548\cdot10^8$	$9.727\cdot10^7$	$5.793\cdot10^7$	$8.920\cdot10^7$	$1.40\cdot10^7$	4.08
72	85.0	22,967	$2.165\cdot10^8$	$8.268\cdot10^7$	$5.178\cdot10^7$	$7.583\cdot10^7$	$1.04\cdot10^7$	2.90
65	72.2	19,776	$1.864\cdot10^8$	$7.119\cdot10^7$	$4.574\cdot10^7$	$6.530\cdot10^7$	$4.23\cdot10^6$	2.27
60	54.2	14,642	$1.381\cdot10^8$	$5.271\cdot10^7$	$3.493\cdot10^7$	$4.830\cdot10^7$	$2.00\cdot10^6$	1.45
50	30.8	8,328	$7.85\cdot10^7$	$2.990\cdot10^7$	$2.022\cdot10^7$	$2.750\cdot10^7$	$7.99\cdot10^5$	1.01

The analyzes performed proved that the propulsion installation is the major consumer (67.2%), but we were shown that neither the need for electricity (11.8%) and the heat fluxes (21%) are not negligible. A large amount of energy is wasted into the environment by cooling the engine and exhaust gases. Using energy analysis, the potential to implement a heat recovery system on board can be estimated.

Chapter 4. Study of the functional parameters of the power supply system as an integral part of the waste heat recovery system

In this chapter, was treated both theoretical calculation elements and real elements of the operation of the turbocharging system, for the main reference engine.

I chose to treat this system because it is a main subassembly for the engine, in terms of operation and thus its efficiency.

In the theoretical calculation I will start from the initial calculation data and I will expose the geometric and functional parameters of the turbocharger.

The supercharger is defined by the joint operation of the turbine and the compressor. The principle of operation consists in entraining the exhaust gases from the engine cylinders, in the turbine blades. The turbine transmits the rotational movement of the compressor, and it drives the working fluid (air) into the engine cylinders.

The components of flue gas turbochargers are subject to extreme operating conditions.

The exhaust gases up to a temperature of 650 °C pass continuously through the turbine and heat its components, without its system of contraction cooling. In particular, the shaft bearing must withstand high operating temperatures without ever breaking the lubrication device.

On the compressor side, the air is heated to over 200 °C. High temperatures lead to extreme thermal loads of the material in many locations. Speeds are extremely high: METxxSE turbochargers run at speeds between 10,000 and 35,000 rpm, depending on size. In this respect, the tangential speeds of 560 m / s and even more are reached at the compressor turbine, which rises to 1.7 times the speed of sound or 2,000 km / h.

Centrifugal forces are extremely high: forces of several hundred kN can be easily applied to the base of the turbine blade.

The complete engine gas exchange is performed by the flue gas turbocharger. For this machine, the combustion air flow can reach 24 m^3 / s. Simplified, it can be said that approx. 1/3 of the power produced by the engine is transferred to the turbocharger.

Starting from the fact that, out of the total amount of energy contained in the burned fuel in a diesel engine, only 20% is used for the effective propulsion of the ship, shipbuilders act in all ways to reduce energy losses and increase propulsion efficiency. Their attention is focused on both the engine and the engine itself and the ship.

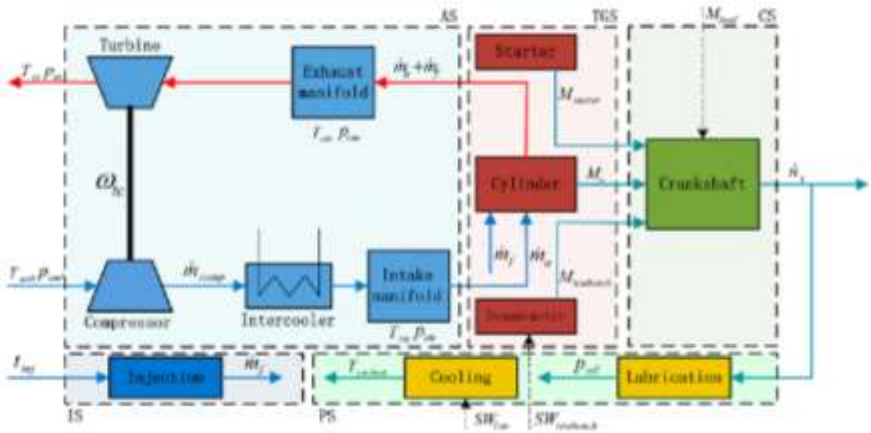

Fig. 4.1 *Energy model*

The turbocharger can be modeled by isentropic thermal efficiency. Compressor outlet temperature is

$$T_{out,comp} = T_{amb}\left\{\frac{1}{\eta_{comp}}\left[\left(\frac{p_{out,comp}}{p_{amb}}\right)^{\frac{\gamma_a-1}{\gamma_a}}-1\right]\right\} \tag{4.1}$$

The power of the compressor can be modeled by

$$P_{comp} = \dot{m}_{comp}\, c_{pa}\, T_{amb}\, \frac{1}{\eta_{comp}}\left[\left(\frac{p_{out,comp}}{p_{amb}}\right)^{\frac{\gamma_a-1}{\gamma_a}}-1\right] \tag{4.2}$$

The power of the turbine can be modeled by

$$P_{turb} = \dot{m}_{turb}\, C_{pe}\, T_{in,turb}\, \eta_{turb}\left[1-\left(\frac{p_{out,comp}}{p_{in,turb}}\right)^{\frac{\gamma-1}{\gamma}}\right] \tag{4.3}$$

The component that aims to enrich the mixture that will be consumed by the engine, is the centrifugal compressor driven by the gas turbine.

The turbocharging process ensures the recovery of a fraction of the energy contained in the flue gases emitted from the engine, which represents approximately 30-40% of the energy released in the combustion process.

Air flow for gas exchange:

$$d_{hsg} = d_{asg}\cdot P_e\left[\frac{Kgaer}{h}\right] \tag{4.4}$$

Obviously, the internal combustion engine is a machine characterized by the presence of stationary flow of the working fluid, as a result of which the turbine of the turbocharger group must be able to operate in such conditions.

The efficiency of the operation of the turbocharger system related to the main propulsion engine depends on the internal processes that take place in its operation, and also on the environmental conditions. In this way, the direction of processing on the installation is dictated by the engine load, taking into account the extreme situations, corresponding to the operation of the ship in special conditions (overload, tropical tem perature, arctic temperature etc).

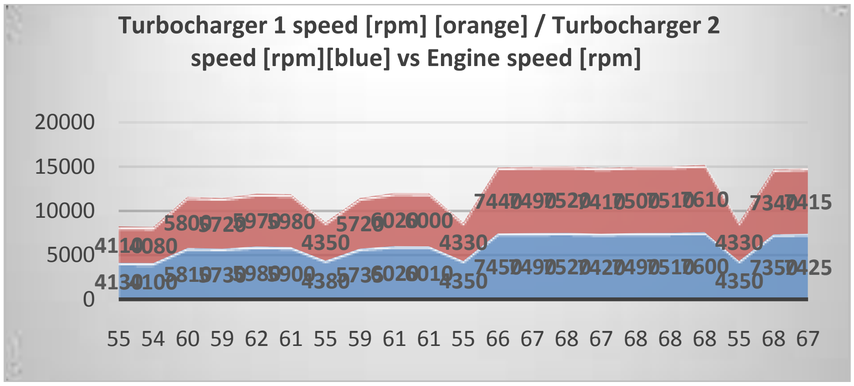

Fig. 4.11 *Turbocharger speed vs engine speed*

Chapter 5. Study of the functional parameters of freshwater generation system as an integral part of the waste heat recovery system

In this chapter I propose to deal with both theoretical and real elements of operation of the freshwater generation system from the main engine of the ship.

I chose to treat this system because it is a main subassembly of the residual heat recovery system from the main engine through cooling water.

Fig. 5.1 *Freshwater generator SASAKURA KM50*

In the theoretical calculation I will start from the initial calculation data and I will expose the geometric and functional parameters of the freshwater generator.

In the final part of the chapter I will refer to the real parameters of the freshwater generation system where I will draw operation diagrams to highlight as concretely as possible the operation of this system on board the ship.

Cooling water energy

$$\dot{Q}_{racire} = \dot{V}_{racire} \cdot \rho_{apamare} \cdot c_{apamare} \cdot (t_{ies} - t_{in}) = 1137{,}12 \text{ kW} \quad (5.1)$$

Fresh water energy

$$\dot{Q}_{apamotor} = \dot{V}_{apatehnica} \cdot \rho_{apatehnica} \cdot c_{apatehnica} \cdot (t_{ies} - t_{in}) = 1140{,}47 \text{ kW} \quad (5.2)$$

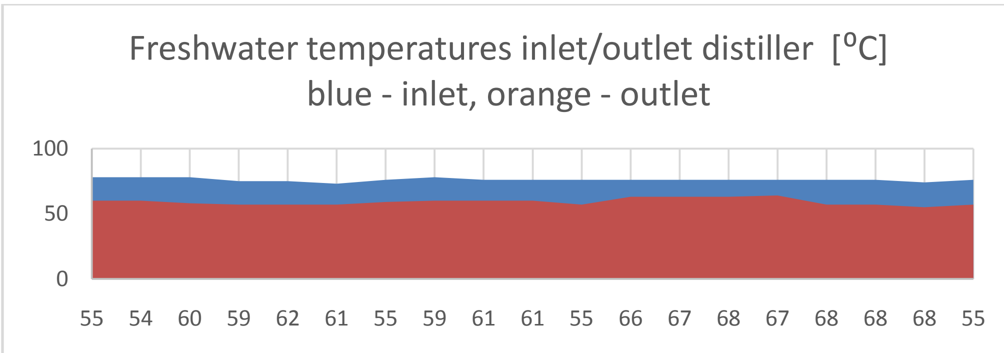

Fig. 5.10 *Freshwater temperatures inlet/outlet distiller*

Chapter 6. Study of the installation using exhaust gases from the main engine as an integral part of the waste heat recovery system

Chapter 6 is the most important section of this paper. In the subchapter technical characteristics of the ship and of the recovery boiler proposed for study, are presented the data of the Horaisan ship whose boiler is proposed for study including data on the parameters of the main engine and especially the geometric and operating characteristics of the economizer. The economizer, with a height of almost 10 m and a side of 7 m, is a massive equipment and such equipment is impossible to simulate in their entirety on common computing equipment. A strategy for simplifying the model was chosen for the numerical simulation. Considering that the three sections of the economizer, namely the low and high pressure stages and the superheater, are placed vertically and taking into account that each stage contains 18 rows of tubes, a vertical section (like a slice of cake) was practiced through the boiler. , containing only one row of tubes. This modeling decision is the most important in the whole simulation.

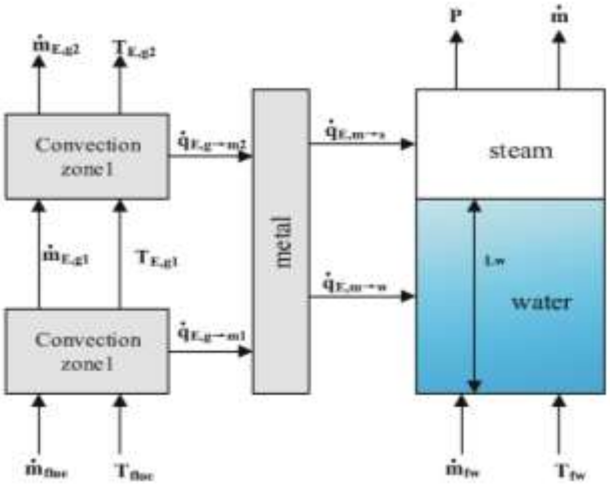

Fig. 6.1 *Economizer energy model*

The mathematical model for the first covection zone

Calculation of gas energy flow in the first convective zone

$$U_{E,g1} = M_{E,g1}h_{E,g1} = V_{E,g1}q_{E,g1}c_{p,E,g1}(\bar{T}_{E,g1} - T_{ref}) \qquad (6.1)$$

Calculation of heat transfer between the gas and the boiler wall

$$\dot{q}_{E,g-m1} = A_{E,g1}\alpha_{E,g1}(\bar{T}_{E,g1} - T_{ref}) \qquad (6.2)$$

The mathematical model for the second convection zone

Calculation of gas energy flow in the second convective zone

$$U_{E,g2} = M_{E,g2}h_{E,g2} = V_{E,g2}q_{E,g2}c_{p,E,g2}(\bar{T}_{E,g2} - T_{ref}) \qquad (6.3)$$

Calculation of the energy flow at the outlet from the convective zone no.2

$$\dot{q}_{E,g2} = \dot{m}_{E,g2}h_{E,g2} = \dot{m}_{E,g2}c_{p,E,g2}(T_{E,g2} - T_{ref}) \qquad (6.4)$$

As the optimization mode in ANSYS, the method of response surfaces was chosen. The results of the model run in CFX will then be retrieved and processed in the Response Surface Optimization module.

Important for optimization analysis are the definition of input and output parameters (input and output from the model). These comprise the input parameters:

- GasVelocity is the speed of the flue gas that enters the boiler and has as central value the value of the real recovery boiler of 55 m / s. To determine the design space, this parameter will be allowed to vary between 49,5 and 60,5 m / s.
- GasTemp is the temperature of the flue gas that enters the boiler and has as central value the value of the real recovery boiler of 536,5^{o}K. To determine the

design space, this parameter will be allowed to vary between 482,54^{o}K and 589,77^{o}K.

- LP1Velocity is the speed of the water that enters the low pressure stage LP and has as central value the value of the real economizer of 0,28 m / s. To determine the design space, this parameter will be allowed to vary between 0,252 and 0,308 m / s.
- LP1Temp is the temperature of the water that enters the low pressure stage LP and has as central value the value of the real economizer of 403,15^{o}K. To determine the design space, this parameter will be allowed to vary between 362,84^{o}K and 443,47^{o}K.

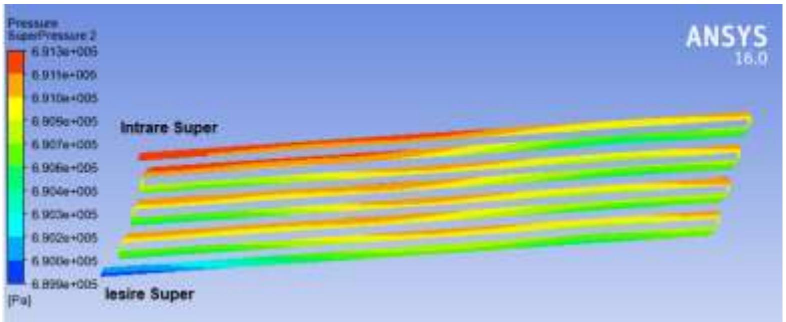

Fig. 6.1 *Superheater pressure*

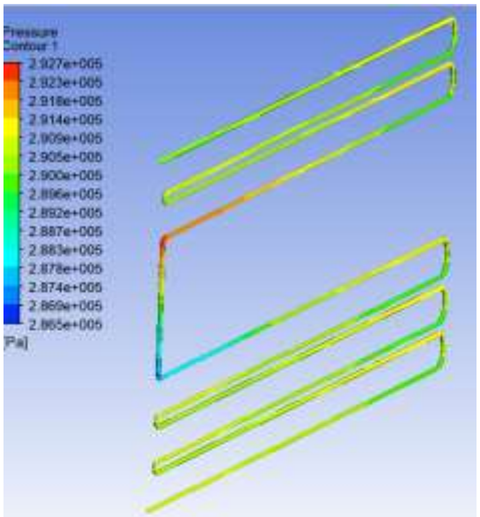

Fig. 6.2 *Pressure variation*

Chapter 7. Experimental results validation

Experimental validation for the turbocharger system

Fig. 7.1 *Turbocharging system measuring points*

Tab. 7.1 *Calculated errors for turbocharging system*

Theoretical air flow at maximum main engine capacity [m^3/s]	**Real air flow at maximum main engine capacity [m^3/s]**	**Error [%]**
28	29,6	5,4
Turbocharger speed 1[rpm]	**Turbocharger speed 2[rpm]**	
6178	6173	0,08
Turbocharger lubricating oil temperature difference 1 [^oC]	**Turbocharger lubricating oil temperature difference 2 [^oC]**	
11,8	11,19	5,16
Air cooler temperature difference 1 [^oC]	**Air cooler temperature difference 2 [^oC]**	
69,52	69,19	0,47
Water cooler temperature difference 1 [^oC]	**Water cooler temperature difference 2 [^oC]**	
12,52	9,85	21,3
Cooling air pressure drop no. 1 [mmH2O]	**Cooling air pressure drop no. 2 [mmH2O]**	
89,9	105,9	15,1

Experimental validation for the freshwater generation system

Fig. 7.2 *Freshwater generation system measuring points*

Tab. 7.2 *Calculated errors for freshwater system*

Fluxul energetic teoretic al apei de mare de răcire (medie) [kW]	**Fluxul energetic real al apei sărate de intrare de răcire [kW]**	**Eroare [%]**	**Fluxul energetic teoretic al apei tehnice provenite de la motorul principal [kW]**	**Fluxul energetic real al apei tehnice [kW]**	**Eroare [%]**
1191,4	1137,1	4,6	1124,2	1140,5	1,44

Experimental validation for the economizer

Fig. 7.3 *Economizer measuring points*

Tabel 7.3 *Calculated errors for economizer*

Parameter	LP Stage	HP Stage	Superheater Stage
Measured temperature	215^{o}C	209^{o}C	297^{o}C/
Calculated temperature	256^{o}C	256^{o}C	256^{o}C
Error %	**+16%**	**+18%**	**-16% sau +4%**
Steam type obtained	Saturated	Saturated	Saturated
Steam type measured	Saturated	Saturated	Saturated
Error %	**0%**	**0%**	**0%**
Steam pressure measured	0,32 MPa	0,8 MPa	0,72 MPa
Steam pressure calculated	0,29 MPa	0,74 MPa	0,69 MPa
Error %	**-10%**	**-8%**	**-4%**

Chapter 8. Final conclusions, personal contributions and recommendations for future work

This paper, entitled ***"Studies and research on waste energy recovery systems for ships"***, demonstrates that contributions can be made to improving the operation of the entire system of recovery of residual energy flow on board ships, through adequate redesign and optimization of the main components of the system.

Many solutions have been detailed to improved energy efficiency for ships, all of which have already been implemented and are yielding tangible results; topics were addresed on optimization the shape of the hull, installations and equipment with a low level of energy consume and ways to improve the energy performance of ships in operation, all important in the stratety of reducing operating costs and reducing the impact of ships on the environment.

Performing the energy balance of the main engine and the Sankey diagram of engine power flows was also drawn.

Regarding the supercharging system, in case of comparing the theoretical results with the real ones, we obtained very good results and errors at air flows, speeds, lubrication temperatures, cooling water temperatures. The errors were below 5,5%, so very good results.

The errors for the energy flows that take place in the reference in the freshwater generator have been calculated. For the energy flow from salt water and which takes place in the condenser, we obtained an error of 4,6%. This value is very considering that the salt water comes from the marine environment. For technical water the error was 1,44% so is a good result and more than satisfactory.

Regarding the recovery boiler - the main flow and heat transfer characteristics of the three stages of the boiler, the fluid pressure, its speed, temperatures, volumetric and mass fractions of steam developed in the tubes were calculated, resulting in a complete picture of the equipment operation.

the optimization analysis of the numerical model comprises two phases, namely: determining the optimal candidate, after which, with optimal thermo-hydraulic parameters, the complete thermo-hydrodynamic model was generated to obtain the results necessary to compare the two models, the real and the optimal. It has been seen that the optimized model delivers quantities of steam with a higher quality than the real model. The most important conclusion is that the water temperature at the entrance to the steps of the recovery boiler has the greatest impact on the quality of the steam delivered.

In the case of temperatures recorded and calculated for the low and high pressure stages, the calculated values are higher by 16% and 18%, respectively, indicating worsened heat transfer conditions in these stages. In any case, the calculated and recorded values are comparable.

The types of steam calculated and those obtained in reality are identical, which shows that the numerical model is valid.

The paper was based both on the theoretical study, on the simulation of energy flow processes with the help of specialized electronic programs, but also on the study of real flows with the help of a 1:1 scale study object, respectively the 300,000 dwt oil tank.

The possibility to perform, over a period of time at choice, exercises on a real ship at a scale of 1:1, allowed the validation of the results obtained with the numerical program. The validation was performed with the help of a chain of real installations: main engine - turbocharging system - distiller - recovery boiler.

The results were satisfactory, which confirms that the program created can also be used by students, PhD students, researchers and specialists who specialize in the divisions of special ships, in deepening the understanding of waste energy recovery systems on board the ship.

Personal contributions

At this point we can specify thinks like the summary of the current state of play of international standards on ship efficiency, the energy balance of the main engine on board the ship both by classical calculations and on the basis of Sankey-type diagrams, sizing of the turbocharger unit, drawing operating charts for reference engine turbochargers, based on measurements made on board the ship; energy modeling of the turbocharger; energy modeling of the distiller; the mathematical model of the recovery boiler - the mathematical model of the first convection zone, of the second convection zone and the mathematical models of the metallic zone respectively of steam and water;the following were performed: the simulation strategy with finite volumes of the biphasic flow and boiling through the tubes of the recovery boiler, the geometric model, the boundary conditions etc; the results obtained after the launch of the model (pressures calculated per model, speeds calculated per model, volume fraction of vapors, etc.) - finite element analysis of the fluid-structure interaction for the low pressure stage LP, optimization analyzes of the numerical model.

Recommendation for future work

This paper opens a certain variety of possibilities related to possible studies and research that can be done in the future, starting from this paper.

They are given below without any claim to completeness, as follows:

• one of the first directions, and the most handy, is the improvement and improvement of the calculation program through the variety of simulation conditions that may be required; thus simulations can be made for different working conditions of the ship (ship in motion, ship at anchor, ship in tropical or arctic conditions) when the working conditions of the main engine differ;

• the geometry used is simplified but can be easily extended to a bunch of tubes for a more realistic representation of the phenomena of flow and heat exchange in the economizer;

• you can try different pipe path geometries than the existing ones to increase the heat exchange;

• solutions can be tried at the level of the ribbed tubes to increase the heat exchange.

It should be noted that the above proposals require enormous computational resources and as such will require a significant investment in these resources to perform the simulations. I am currently considering a collaboration with specialized research centers or at those companies that have invested enough in new and modern CFD computing methodologies and software to achieve what I set out to do in the future.

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