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Possibilities for Improving the Cooling Systems of IC Engines of Marine Power Plants

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ABSTRACT

Modern cooling systems for large ships are quite complex. As a rule, such systems are common (combined) for the main and auxiliaries engines. With the auxiliary engines running constantly, even when parked, this system design allows to keep it warm and ready for a quick start of the main engines at any time. Currently, various schemes of such systems are used, including those that are irrational from our point of view. At the same time, there are systems whose schemes are quite consistent with our idea of the rational forms of such structures. It is important to note, and it is saying about it in an article, that such schemes may have a number of significant differences, but at the same time they will comply with the rationality principle if certain rules for the formation of such systems are followed. These schemes will have close compactness. It is also important that there is the possibility of further improvement of such schemes based on certain rules. This improvement is possible due to the introduction of additional heat dissipaters and the organization of appropriate chains of heat sources and heat dissipaters. The article discusses various options for rational schemes of the cooling system for the same ship power plant, as well as the possibility of further improvement of this scheme. It is shown that an increase in the number of coolants of the internal circuit coolant from one to three can reduce the total mass of the heat exchanger cores by 18 %.

Keywords: central cooler; charge air cooler; cooling system; heat transfer surface; marine diesel engine; oil cooler.

INTRODUCTION

Modern cooling systems for large ships are quite complex. As a rule, such systems are common (combined) for the main and auxiliary engines. Because the auxiliary engines are constantly running, even when parked, this system design keeps system warm and ready to quickly start the main engine at any time. The noted feature of ship cooling systems is important, but not the only one.

For cooling systems of any type and purpose, the main task is to maintain the specified temperatures of coolants and charge air in engines with minimum weight, maximum compactness and minimum operating costs. It is from this point of view that rational and irrational cooling systems for any type of power plants can be distinguished.

Currently, various schemes of such systems with natural air or water cooling are used for marine power plants [1, 2], for cars [3–8] and locomotives [9]. In addition, there are works [10, 11], which consider engine cooling systems using refrigeration machines. The systems must have effective regulation [12–15], which will allow maintaining the set temperatures at any ambient temperature and at any load of the power plant.

For all possible power plants, various schemes of such systems are used, including those that are irrational from our point of view [16]. At the same time, there are systems whose schemes are in complete agreement with the authors' idea of the rational forms of such structures [17]. The scheme presented in the last source may have several analogs that are different from it, corresponding to the principle of rationality of formation [18], and at the same time, all of them will have similar parameters of compactness and cost.

A rational design of a ship's cooling system should consist of two types of joined circulation circuits. The first type of circuit should include the main engine. The coolant flow rate of this internal circuit coolant (ICC) in it must be set in advance by the engine manufacturer. In the second circuit, the ICC flow rates should be determined in such a way that their values ensure the maximum compactness of the system at a given charge air temperature in the engine receiver or the maximum possible decrease in the charge air temperature, if such a goal is set. The contours must be merged [17]. The number of pumps in such schemes can be minimal (one for each engine), and can be increased considering possible redundancy and the length of pipelines for ship systems. Schemes formed according to the formulated rules can be considered modern and rational. It should be noted that schemes organized according to a different principle will not meet the conditions of rational formation.

It should be emphasized that the method of forming such circuits is not distinguished in the sources known to us, and, accordingly, there is a controversial possibility of creating cooling circuits on a different basis. The authors of the schemes do not indicate the expediency and even the need to rationalize the costs of HEC in the circulation circuits of the second type. At the same time, the need to use heat exchangers of a special design in such schemes (for certain coolant flow rates) is not considered. In the sources known to us, there is also no information about the possibility of further improvement of such schemes by increasing the number of heat dissipaters (water coolers or radiators).

This article discusses the possibility of using the rules for the formation of efficient and compact cooling systems for internal combustion engines when creating ship systems, as a rule, more complex than other transport systems. In addition, the possibility of further improvement of such cooling systems is being considered. This is possible and expedient, although the issue of compactness is not paramount here.

SYSTEM DESCRIPTION AND ASSUMPTIONS

Consider analogues of rational schemes of cooling systems for the same power plant of a large ship. On Figure 1 shows a scheme of the cooling system, close to that presented in [17]. It was made by the authors as one of the possible options for the cooling system of the power plant of a large vessel and meets the conditions of rational construction.

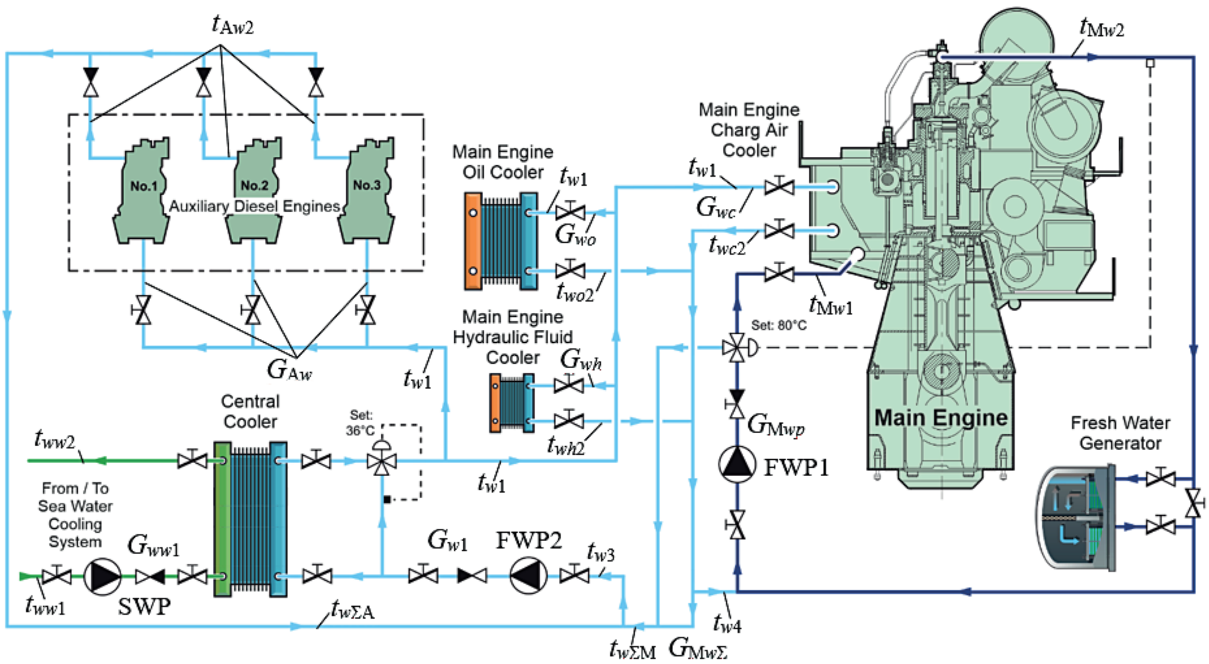

Fig. 1. Scheme of the cooling system of a ship power plant with one central cooler: FWP – fresh water pump; SWP – sea water pump

The scheme differs from the original, with the exception of a number of elements that are essential for the manufacturer and consumer, but not important from the point of view of the main purpose of such systems cooling the elements of the power plant. In particular, there is no expansion tank, parallel pumps, preheating of ICC in front of the main engine, etc. At the same time, cooling of the hydraulic fluid is possible in the above scheme. In addition, in the original, the ICC enters the first circuit (the main engine circuit) from the second circulation circuit after mixing all the flows after the coolers and auxiliary engines. In the above diagram, the ICC after the auxiliary engines goes directly to the inlet of the central cooler (CC). Analogues, different from each other, but designed to perform the same functions and operating under the same conditions, were created and studied by the authors to establish the possibility of improving cooling systems with a single the internal circuit cooler (CC).

In the primary circulation circuit (Figure 1), after passing through the cooled cavities of the main engine, most of the ICC flow enters the inlet of the main engine pump. At the same time, a smaller part of the ICC flow, having absorbed heat from the cooled engine cavities, is discharged from the primary circuit to the second circuit, with the ICC cooler (CC), heat exchangers and auxiliary engines, where it mixes with a part of the ICC flow from the main engine and cools. Part of the ICC flow, after three heat exchangers of the main engine, partially enters the inlet of the main engine pump, where it mixes with the flow after the main engine, as a result of which the required temperature of the ICC is set in front of the engine. The second part of the ICC flow after the heat exchangers, go to the central cooler, where it mixes with the flows from the main and auxiliary engines and cools down. The coolant of the internal circuit, cooled in the central cooler, partly goes to the auxiliary engines, and partly to the heat exchangers of the main engine.

In principle, the cooling systems of auxiliary engines should be organized rationally, as recommended by the rules for the formation of rational cooling systems. In this article, for simplicity, the problem of rationalizing auxiliary engine systems is not considered. At the same time, they are all connected to a common cooling system, as required by the design of the cooling system on the ship, and emit a certain amount of heat into this system. The release of a certain amount of heat is the main function of these engines in this consideration.

The fresh water generator was not considered in the calculations of the scheme, since it is able to remove a significant amount of heat during operation, facilitating the operation of the central cooler. It was assumed that the central cooler should be able to cope with the cooling of the ICC in the system without a working fresh water generator, providing the proper temperatures. Thus, the most difficult (but possible) operating conditions were created for the system. Table 1 show the main data of the cooling system of this ship in accordance with [17], which were used in the calculations.

For all schemes, it was assumed that the heat exchange surfaces and designs of all heat exchangers are the same and correspond to the typical capabilities of conventional production of such products. All heat exchangers of cooling system, designed to operate with, as a rule, reduced (compared to conventional systems) liquid flow rates, are distinguished by increased numbers of strokes along the ICC and relatively low coolant velocities in their channels. In other respects, their designs mirror those of traditional heat exchangers.

The scheme of the cooling system, in which all the rules for the formation of rational schemes will be fulfilled, can be made with several differences. Figure 2 shows parts of such circuits that differ from the circuit in Figure 1 but have the same functions.

Not shown parts of the circuits are similar to Figure 1. Schemes in Figure 2 were calculated and optimized. The difference between scheme *a* and that shown in Figure 1 consists in the fact that the supply of ICC to the main engine pump occurs not after all the heat exchangers, but from the channel in front of them. The difference in scheme b is that the main engine pump has excess capacity. To reduce it to the calculated value, a part of the flow rate is transferred to the central cooler. In addition, the entire ICC flow behind all heat exchangers enters the main engine pump inlet, mixing with the ICC flow behind the engine. In scheme *b*, the ICC after the heat exchangers does not go to the central cooler, as in scheme *a*. Scheme *c* differs in that it introduces a bypass of a part of the ICC after the main engine directly to the input of the central cooler. The difference between scheme *d* and scheme *c* is that here the ICC flow behind the heat exchangers is bifurcated.

Table 1. Data for calculations of cooling systems

Items	Designation	Values
Heat flow in ICC from cylinders and parts of the main engine (kW)	Q_{Mw}	1450
Heat flow into oil from main engine (kW)	Q_{Mo}	790
Heat flow from main engine charge air cooler (kW)	Q_{Mc}	4501.9
Heat flow in the ICC of the auxiliary engine (kW)	Q_{Aw}	239*
Heat flow into the oil from auxiliary engine (kW)	Q_{Ao}	117*
Heat flow from charge air cooler auxiliary engine (kW)	Q_{Ac}	369*
Heat flow from the hydraulic system of the main engine (kW)	Q_{Mh}	40
Charge air consumption of the main engine (kg s^{-1})	G_{Mc}	25.9
Oil consumption of the main engine (kg s^{-1})	G_{Mo}	57
Fluid flow in the hydraulic system (kg s^{-1})	G_{Mh}	5
ICC flow through the cooling cavities of the main engine (kg s^{-1})	G_{Mwp}	23
Outboard water consumption through the ICC cooler (CC) (kg s^{-1})	G_{ww}	178
Number of auxiliary engines	Z_{Ae}	3
Number of main engine charge air cooler (CAC)	Z_{Mc}	1
Sea water temperature (°C)	t_{ww1}	36
Ambient temperature (°C)	t_0	45
Air temperature behind the compressor (°C)	t_1 (t_k)	214
Charge air temperature in the main engine receiver (°C)	t_s	51
ICC temperature after auxiliary engine (°C)	t_{Mw2}	95
ICC temperature after main engine (°C)	t_{Aw2}	95
Oil temperature at the exit from the main engine and the entrance to the oil cooler OC (°C)	t_{o1}	51
Hydraulic fluid temperature before hydraulic fluid cooler (°C)	t_{h1}	51
Ambient air pressure (bar)	p_0	1
Air pressure after compressor (bar)	p_1	3.5

*For one auxiliary engine

Part of the flow goes to mixing with ICC after the engine, as in scheme *c*, and part goes directly to the central cooler.

All presented schemes of systems are built based on of rules that allow to create rational cooling systems. In all these systems, one heat dissipater central cooler is used. If the number of heat dissipaters is increased (theoretically, up to the number of heat sources), it is possible to improve the system parameters (Figure 3).

An improvement can be achieved only if the chains of heat sources and heat dissipaters are properly organized and rational ICC flow rates for these chains are chosen. In Figure 3, the first chain consists of central cooler 1 and charge air cooler of main engine, the second of central cooler 2, oil cooler and hydraulic fluid cooler, the third of central cooler 3 and auxiliary engines.

METHODS

The calculations of the systems were performed using the calculation complex, described in sufficient detail in [18]. When calculating systems and heat exchangers, programs were used that were repeatedly verified by various full-scale tests. For all compared options, the same designs of heat exchangers and the same heat exchange surfaces were assumed. All compared systems were calculated under the same initial conditions. Based on the calculations, dependencies similar to those shown in Figures 4–8 were obtained for all systems, where they are given for a system with three central coolers (Figure 3).

Dependencies should be obtained and built repeatedly, considering the parameters obtained at local minima M_Σ for each of the dependencies (obtained at constant other parameters). With this approach, there is a gradual approach to the possible absolute minimum M_Σ, depending on all the varied values. Here we present (for ease of presentation and understanding) a fundamental approach to finding the optimum by the method of descent along the coordinate. With a machine account, the process can be formalized and automated, including based on of other methods of finding the optimum. These dependencies are the basis for obtaining the most rational parameters of cooling systems, including the rational

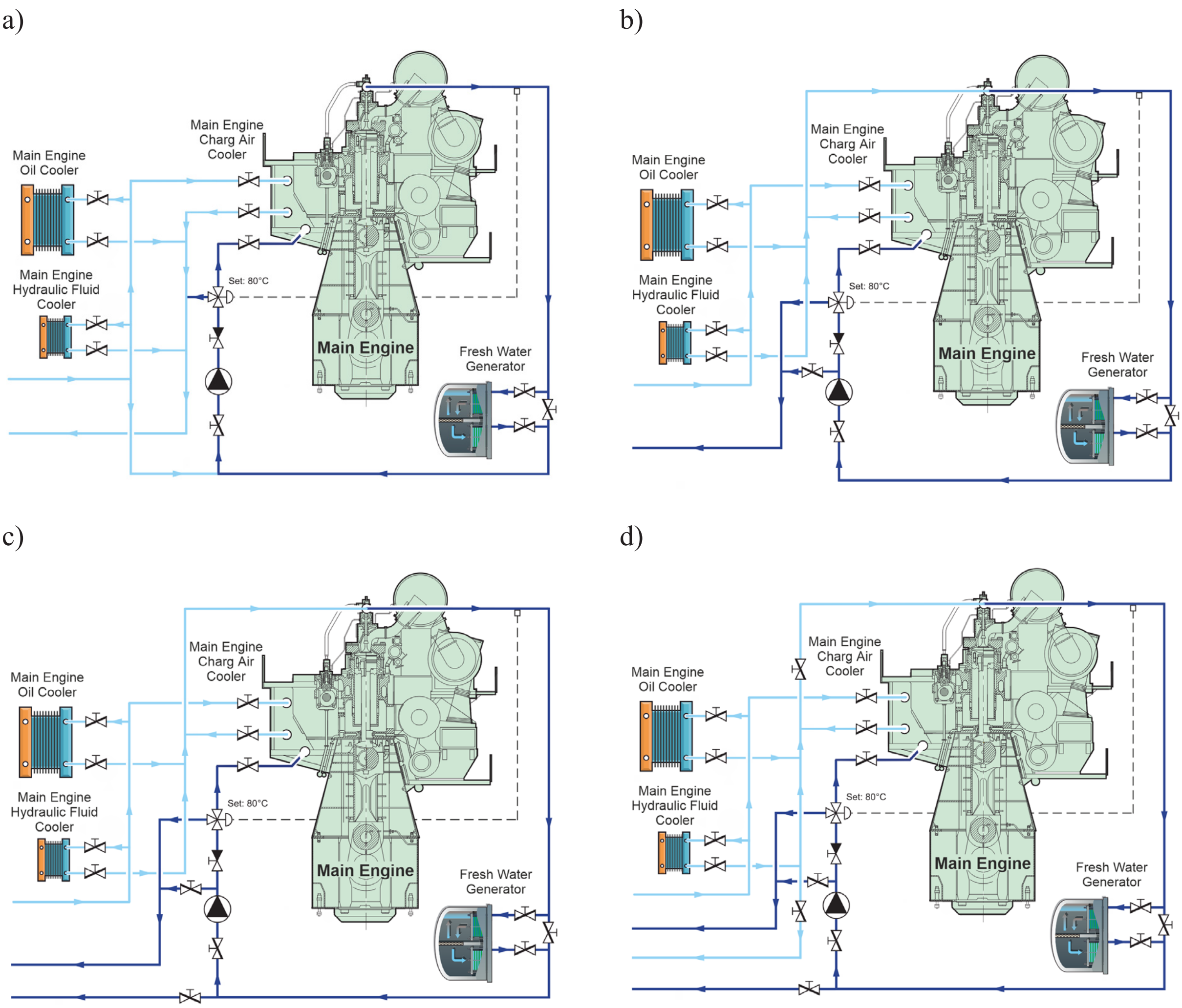

Fig. 2. Alternative schemes of the cooling system with one central cooler

distribution of coolant flow rates among the system branches. The parameters at which the minimum sum of the masses of the cores has been achieved are used to determine all design data of the heat exchangers. The final results of the study are presented in Tables 2 and 3.

1. The water flow of the FWP1 (main engine pump) may be higher than the allowable flow through the main engine; excess is removed by bypass.
2. For one auxiliary engine.
3. ICC consumption for a scheme with one central cooler is equal to the consumption of the ICC pump FWP2.

RESULTS AND DISCUSSION

Before comparing the original system and its analogues, each of these objects underwent an improvement procedure based on the methods developed by the authors. When performing calculations, it was assumed that the initial conditions are the same in all cases, the designs of the heat exchangers are of the same type, and the heat exchange surfaces are the same. At the same time, the task was not to accurately reproduce the parameters of the original system (for this, the authors did not have a number of initial data). Inaccuracies of this kind do not matter here, since for the correctness of the conclusions it is enough that the systems are calculated under the same conditions.

As it turned out, in terms of compactness, all possible schemes of rational single-radiator schemes are almost the same. The total mass of heat exchanger cores in them can be brought up to approximately 9500 kg (the figure is conditional, it corresponds to the accepted parameters of heat exchange surfaces and heat exchanger designs). It is possible that when manufacturing systems according to the

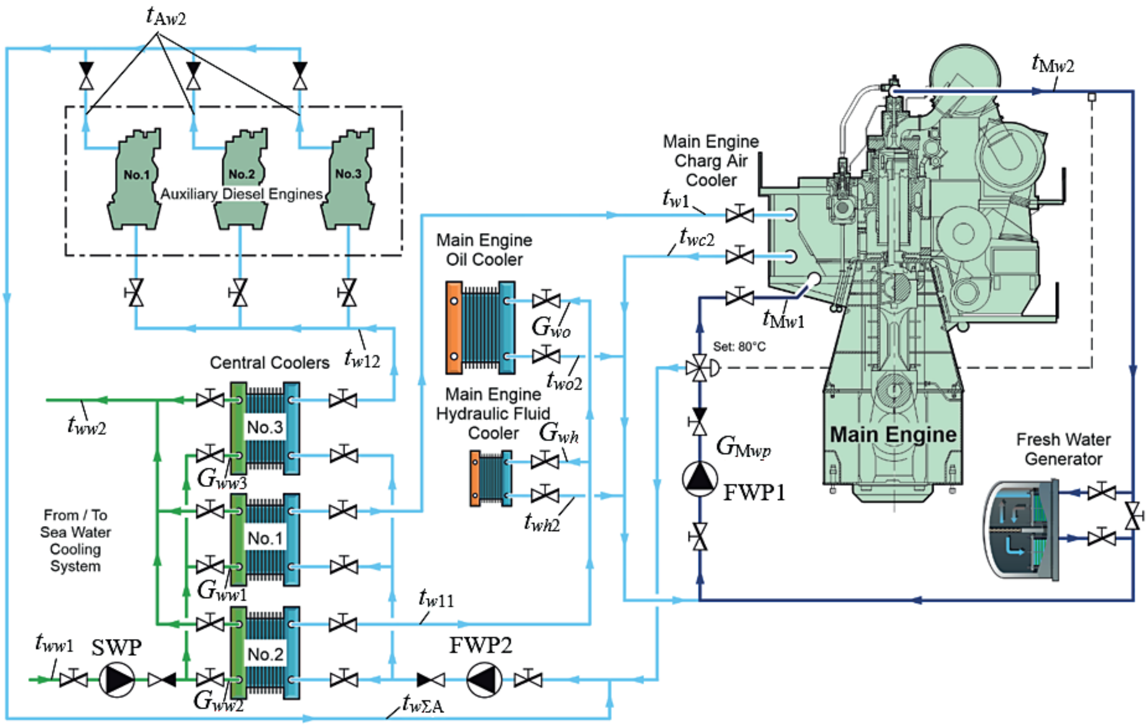

Fig. 3. Cooling system with three central coolers

presented schemes, some of them will be more convenient for the manufacturer, and some less. There is no particular difference in the systems, they are rational, have approximately the same mass with the same efficiency. The result obtained confirms that rational, practically equivalent to each other and at the same time essentially different systems can be built according to universal rules.

When switching to multi-radiator circuits, it is possible to reduce the total mass of the cores of the heat exchangers of the system. This is ensured by the formation of more independent chains of heat sources and heat dissipaters than in a single-radiator system, and the selection of rational ICC flow rates through them. To illustrate this situation, in addition to one radiator, schemes of two-radiator and three-radiator systems were studied.

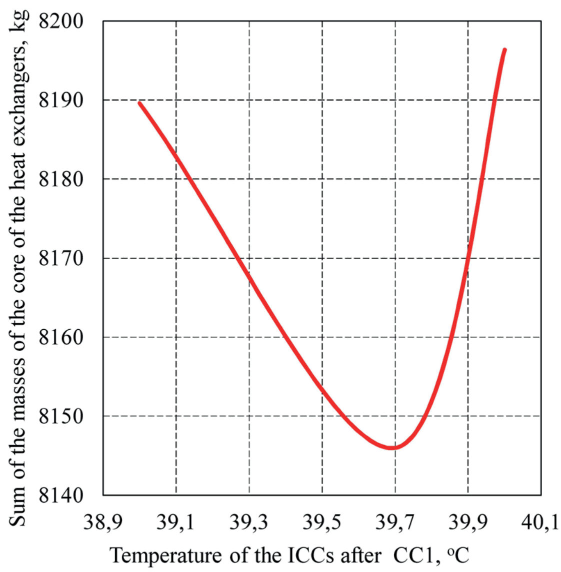

Fig. 4. Dependence of M_Σ on the ICC temperatures after central cooler 1, t_{w1}, at t_{w11} = 37.3 °C and t_{w12} = 55.2 °C; G_{ww1}/G_{ww} = 0.34; G_{ww2}/G_{ww} = 0.52

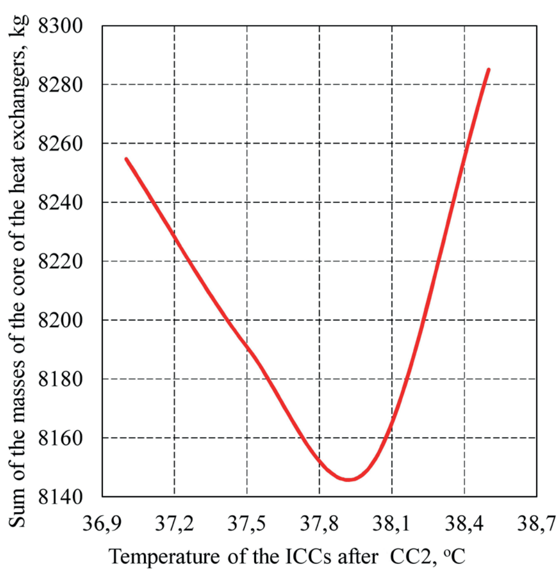

Fig. 5. Dependence of M_Σ on the temperature of the ICC after central cooler 2, t_{w11}, at t_{w1} = 39.7 °C; t_{w12} = 55.2 °C, G_{ww1}/G_{ww} = 0.34; G_{ww2}/G_{ww} = 0.52

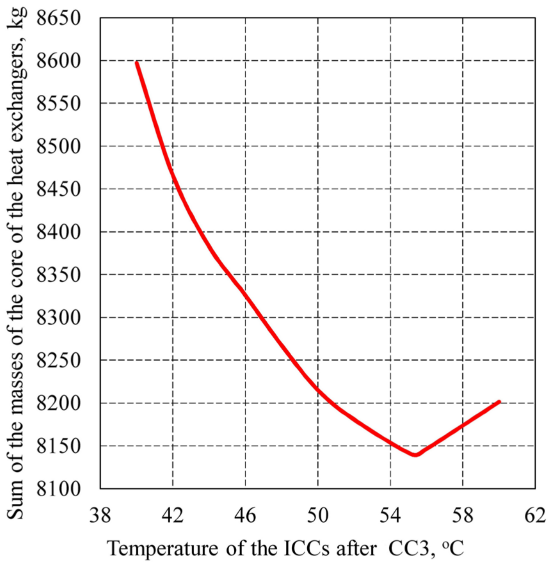

Fig. 6. Dependence of M_{Σ} on the temperature of the ICC after central cooler 3, t_{w12}, at t_{w1} = 39.7 °C; t_{w12} = 37.3 °C, G_{ww1}/G_{ww} = 0.34; G_{ww2}/G_{ww} = 0.52

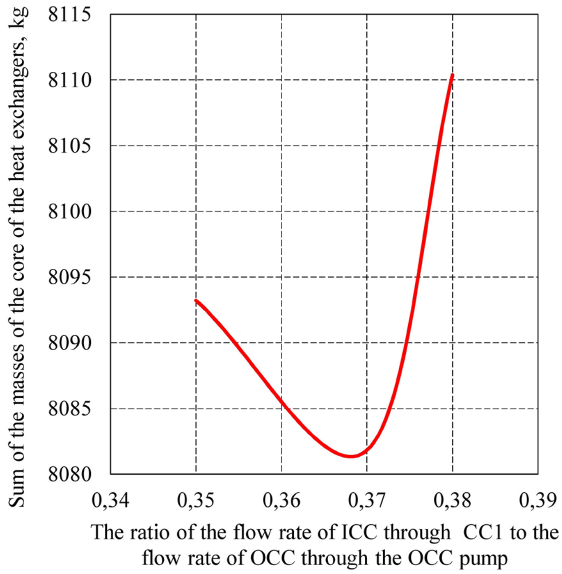

Fig. 7. Dependence of M_{Σ} on the ratio G_{ww1}/G_{ww} (outboard water flow through central cooler 1 to sea water flow through the pump); at: t_{w1} = 39.7 °C; t_{w11} = 38 °C; t_{w12} = 55.2 °C, G_{ww2}/G_{ww} = 0.52

The two-radiator system is made according to the scheme similar to Figure 3. At this system, central cooler 1 was used to work with charge air cooler, and central cooler 2 – with others heat sources. The three-radiator system was made in accordance with Figure 3. The results of the study of all schemes are presented in Table 2.

For a single-radiator system, the minimum value M_{Σ} was given earlier, for a two-radiator system, the minimum value $M_{\Sigma} \approx$ 8500 kg, for three radiators $M_{\Sigma} \approx$ 7800 kg. As can be seen, with a change in the amount of central cooler, a significant decrease in the total mass of the heat exchanger cores is possible. The main parameters of the systems are summarized in Table 3. It can be seen from it that in a three-radiator system, masses of the central coolers cores significantly decreases, masses of oil coolers cores decreases to a lesser extent, masses of charge air coolers cores increases slightly, masses of hydraulic fluid coolers cores practically does not change. The gain in the mass of heat exchanger cores, compared with a single-radiator system, is achieved by increasing the number of central coolers and is about 18%.

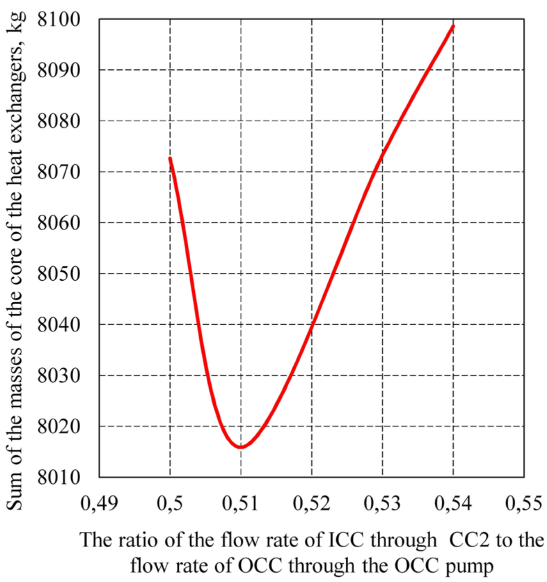

Fig. 8. Dependence of M_{Σ} on the ratio G_{ww2}/G_{ww} (outboard water flow through central cooler 2 to sea water flow through the pump); at: t_{w1} = 39.7 °C; t_{w11} = 38 °C; t_{w12} = 55.2 °C, G_{ww1}/G_{ww} = 0.37

CONCLUSIONS

Thus, the proposed rules are also suitable for designing schemes of cooling systems for marine power plants. The use of rules allows significant differences in schemes, but their observance ensures almost the same compactness. A further reduction in the weight and an increase in the compactness of ship systems can be achieved by increasing the number of heat dissipaters (ICC coolers) to the number of heat sources. The possible number of heat dissipaters is determined depending on the feasibility of improving the system parameters.

Table 2. Calculated values of the main parameters of the compared cooling systems at rational heat carrier flow rates for all elements of these systems

Designation	Items	Column Numbers		
		I	II	III
G_{Mwp}	ICC flow through FWP1 (main engine pump)[1] (kg s^{-1})	23	67.4	67.4
$G_{Mw\Sigma}$	ICC flow from the main engine to the CC (kg s^{-1})	57.1	44.4	44.5
G_{Aw}	ICC flow through auxiliary engine[2] (kg s^{-1})	9.2	9.4	13
G_{wo}	ICC flow through oil cooler (kg s^{-1})	29	24.8	24.8
G_{wh}	ICC flow through hydraulic fluid cooler (kg s^{-1})	1.5	1.25	1.3
G_{wc}	ICC flow through CAC of main engine (kg s^{-1})	18.2	18.4	18.4
G_{w1}	ICC flow through central cooler 1[3] (kg s^{-1})	66.4	18.4	18.4
G_{ww1}	External coolant circuit flow through central cooler 1 (kg s^{-1})	178	62.3	65.86
G_{w2}	ICC flow through central cooler 2 (kg s^{-1})	–	35.2	26.06
G_{ww2}	External coolant circuit flow through central cooler 2 (kg s^{-1})	–	115.7	90.78
G_{w3}	ICC flow through central cooler 3 (kg s^{-1})	–	–	13.0
G_{ww3}	External coolant circuit flow through central cooler 3 (kg s^{-1})	–	–	21.36
G_{ww1}/G_{ww}	The ratio of the flow rate of external coolant circuit through CC1 to the flow rate of external coolant circuit through the CCs pump (SWP)	–	0.35	0.37
G_{ww2}/G_{ww}	The ratio of the flow rate of ICC through CC2 to the flow rate of External coolant circuit through the CCs pump (SWP)	–	–	0.51
t_{ww2}	External coolant circuit for central cooler 1 (°C)	57.7	47.9	48.7
t_{ww22}	External coolant circuit for central cooler 2 (°C)	–	48.7	93.4
t_{Mw1}	ICC before main engine (°C)	80	80	80
t_{w1}	ICC for CC1 and before CAC of main engine (°C)	39	39.5	39.7
t_{w11}	ICC for CC2 and before OC, HFC (°C)	–	39.5	38
t_{w12}	ICC after CC3 and before auxiliary engines (°C)	–	–	55.2
t_{wc2}	ICC for CAC of main engine (°C)	98	98	98
t_{wh2}	ICC for hydraulic fluid cooler (HFC) (°C)	45.5	45.6	45.6
t_{wo2}	ICC after oil cooler (°C)	45.5	45.6	45.6
t_{w4}	ICC after all heat exchangers (°C)	66.7	68.7	68.8
t_{w3}	ICC before all central coolers (°C)	71.2	78.4	83.4
$t_{w\Sigma M}$	ICC in the branch of the main engine before all central coolers (°C)	67.3	75.0	80.0
$t_{w\Sigma A}$	ICC in the auxiliary engines branch before all central coolers (°C)	95	95	95
η_o	Efficiency of cooling system by charge air cooling	0.972	0.972	0.972
CAC	Charge air cooler			
η_c	Efficiency of CAC	0.988	0.991	0.993
Δp_c	Air resistance (mm H_2O)	122.7	134.9	128.6
Δp_{wc}	Water resistance (kPa)	42.9	46.4	48.4
M_c	Weight of CAC core (kg)	882.8	943.6	1022.7
CC1	Central cooler 1			
η_{w1}	Central cooler efficiency	0.915	0.927	0.941
Δp_{icc}	Internal coolant circuit resistance (kPa)	37.0	43.6	56.4
Δp_{ww}	External coolant circuit resistance (kPa)	54.2	117.8	51.1
M_{w1}	Tube bundle weight (kg)	5571.6	1300.1	1335.8
CC2	Central cooler 2			
η_{w2}	Central cooler 2 efficiency	–	0.91	0.958
Δp_{icc}	Internal coolant circuit resistance (kPa)	–	42.4	46.17
Δp_{ww}	External coolant circuit resistance (kPa)	–	43.8	60.16
M_{w2}	Tube core weight (kg)	–	3477.4	2576.8
CC3	Central cooler 3			
η_{w3}	Central cooler 3 efficiency	–	–	0.958
Δp_{icc}	Internal coolant circuit resistance (kPa)	–	–	47.8
Δp_{ww}	External coolant circuit resistance (kPa)	–	–	51.1
M_{w3}	Tube core weight (kg)	–	–	328.0

Table 2. Cont. Calculated values of the main parameters of the compared cooling systems at rational heat carrier flow rates for all elements of these systems

Designation	Items	Column Numbers		
		I	II	III
OC	Oil cooler			
η_o	Oil cooler efficiency	0.599	0.585	0.58
Δp_o	Total oil resistance of all units (kPa)	227.2	210.3	203.3
Δp_{icc}	Total ICC resistance of all units (kPa)	47.1	32.0	57.3
M_o	Mass of all oil cooler cores (kg)	2951.8	2727.4	2461.7
HFC	Hydraulic fluid cooler			
η_h	Efficiency hydraulic fluid cooler	0.543	0.586	0.59
Δp_o	Total hydraulic fluid resistance of all cores (kPa)	172.4	150.1	138
Δp_{icc}	Total ICC resistance of all cores (kPa)	40.9	26.5	27
M_h	Mass of all cores hydraulic fluid coolers (kg)	119.3	106	106.5
M_Σ	The sum of the masses of the cores of all heat exchangers in the system (kg)	9525.5	8554.5	7831.5

Notes: Column Numbers: I – system with one central cooler; II – with two central coolers; III – with three central coolers.

Table 3. Changes in the masses of the cores various heat exchangers

Heat exchanger	Cooling system with		
	One central cooler	Two central coolers	Three central coolers
CAC	882.8	943.6	1022.7
CC	5571.6	4777.5	4240.6
OC	2951.8	2727.4	2461.7
HFC	119.3	106	106.5
The sum of the masses of the cores of all heat exchangers in the system (kg)	9525.5	8554.5	7831.5
The gain in the mass of heat exchanger cores (%)	0	10.2	17.8

Finding the optimum is done iteratively. The first dependencies involve the arbitrary assignment of a number of parameters, which are then refined in the process of searching for optimums. The rule of comparison of the last approximation with the previous one applies.

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