Application of the environmentally safe operating schemes to the cooling systems of ship power plant

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Abstract. The possibility of using the ecologically safe operation on a closed-loop circuit instead of an open-loop cooling system is considered. For this purpose, it was proposed to use the ballast tanks on the ship. To ensure effective heat removal, it is proposed to apply a jet inlet of coolant to the hull shell of a ship. The flow regimes of the impact fluid jet were visualized. The critical Reynolds number is determined. Heat transfer coefficients and generalizing calculation dependences are determined by experiments.

1. Introduction
The currently widely used open-loop cooling systems of ship power plants involve the consumption of outboard cooling water. However, such system schemes have several disadvantages. The intake of seawater which contains plankton, fish eggs and fish leads to the death of most of these organisms [1-3]. As a result, the basis of the food chain of the reservoir is destroyed, which inevitably affects fish productivity [4-6]. Moreover, it is not always possible to ensure the reliability of the cooling systems operation. For instance, it is impossible when the ship is in a heavily contaminated water area, in ice slush, etc. Water consumption under such conditions can lead to clogging of the system and termination of its operation, which will probably result in an emergency situation.

There are ships that are specially designed to work in these conditions, and their cooling systems operate on a closed-loop circuit. However, such systems require the use of special heat exchangers, the design of which may be different [7, 8]. As a rule, such devices are part of the hull, that is why it is possible to manufacture them only during constructing a ship.

Nevertheless, the possibility of modernization of the cooling system to closed loop scheme exists. This can be achieved by using the seawater from the ship’s ballast tanks. This water is taken from the tank, is pumped through the cooled equipment and is discharged back into the tank (figure 1). The water heated in the tank transfers heat through the hull shell of the ship to the outboard seawater.

However, the operation time is limited and is determined by the time the water temperature in the tank reaches the maximum permissible level at the inlet to the cooled power equipment. In general, this is determined by the capacity of the cooled power equipment, the initial temperature of the water in the tank, the seawater temperature and the heat removal achieved through the ship hull shell to the outboard seawater.

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Figure 1. Scheme of using water in a wing ballast tank to cool the power equipment.

Under normal conditions, the intensity of heat removal through the ship hull shell is relatively low, since the removal occurs almost in free convection. It is well known that the values of the heat transfer coefficients achieved in this case are much lower than the values for forced convection. Therefore, there is a need to envisage additional technical measures to ensure intensive circulation of water in the tank along the ship hull shell. It can make it possible to extend the closed loop operating time using the ballast tank.

The aim can be achieved through the use of an impact jet supply system from the inner side of the ship hull shell (figure 2).

Figure 2. Scheme of water jet supply in the tank: 1 – hull shell; 2 – ballast tank; 3 – chilled water supply manifold; 4 – nozzles for supplying water to the ship hull shell; 5 – elements of the ship's hull structure; 6 – outboard seawater.

The water heated in the ship power plant is not just discharged into the tank, but with a special manifold 3 is distributed along the hull shell 1, which is in contact with the seawater. The manifold is equipped with a system of nozzles 4, at the outlet of which the jets flowing onto the surface are formed 1. After hitting the surface, the jet turns 90 degrees, forming a radially spreading wall jet. Such jets are called impact jets. It is known that impact jets provide sufficiently high values of the heat transfer coefficients, and, hence, the heat transfer coefficient through the ship hull shell.

2. Visual investigation
To determine the calculated dependences, special experimental studies were carried out. These included visualization of the ongoing processes, as well as heat engineering studies, in order to determine the values of the heat transfer coefficients on the inner surface of the ship hull shell.

The experimental model of the examination was a rectangular container with a transparent wall measuring 0.25 - 0.25 m. This wall imitated the hull shell. On the inside, a supply nozzle was installed perpendicular to the wall in the center. It was possible to change the distance (h) from the nozzle to the wall. The container was completely filled with water, which created conditions for the formation of a submerged jet. The experimental plant provided regulation of the water flow through the nozzle. Thus,
the maximum value of the water velocity in the nozzle (1.8 m/s) was reached. This value corresponded to the value of the Reynolds number \( (Re_d = 2.4 \cdot 10^5) \). To visualize the flow, ink, silver and detergent were added to the water. Observations were carried out from the outside of the transparent wall, which was additionally illuminated by a directional light source.

The results of the examination at various Reynolds numbers are shown in figure 3. The liquid flowing out of the nozzle hits the surface and then spreads in radial directions, forming a radial wall jet.

It is quite obvious that, as the Reynolds number changed, the type of the fluid flow along the surface changed accordingly. Thus, when \( Re_d \) equals \( 1.65 \cdot 10^3 \), an almost laminar flow of liquid is observed. In this case, inactive ring structures are clearly visible. These ring structures are the result of the interaction of the impact jet with the surface. As the Reynolds number increases, the size of the ring structures decreases and these structures began to move outside. Turbulent fluctuation appears on the outer part of the rings, which then tend to attenuate. When \( Re_d \) equals \( 2.4 \cdot 10^5 \), developed turbulent vortices are found over the entire area. The intensity of these vortices reaches maximum in the central part and decreases as it moves away from the center.

The visualization carried out made it possible to determine that the lower critical Reynolds number is approximately \( 2 \cdot 10^4 \), and the higher one is about \( 1 \cdot 10^5 \).

When observing the processes occurring in depth, at a distance from the wall, secondary flows directed towards the nozzle were revealed. This was explained by the seepage of fluid from the surrounding space, which confirms the existing ideas about the movement of fluid in the area of the trailing edge of the nozzle [9].

3. Heat-engineering studies
Heat-engineering studies were carried out on a specially created experimental plant, the heat-transfer surface of which was \( 0.5 \times 0.5 \text{ m} \) in size. The nozzle for supplying heat coolant water was directed to

\[
Re_d = 1.65 \cdot 10^3
\]

\[
Re_d = 4.0 \cdot 10^3
\]

\[
Re_d = 1.8 \cdot 10^3
\]

\[
Re_d = 2.35 \cdot 10^5
\]

Figure 3. The flow of a near-wall jet with a nozzle supply of liquid at different Reynolds numbers (\( Re_d \)).
the center of this surface. The heat transfer surface was cooled by water circulating from the outside of this surface.

Moreover, it was possible to change the distance (h) from the nozzle end to the surface. The temperature of the heat-transferring wall from the side of the nozzle was measured with seven chromel-copel thermocouples with the thermoelectrodes 0.15 mm in diameter. The thermal electromotive force was measured with a potentiometer. The temperatures of inlet and outlet of hot and cold water were also measured.

Heat engineering studies were carried out on nozzles with an inner diameter of 18 and 38 mm. The distance (h) from the nozzle to the surface was set equal to 5 mm, 12 mm, and 20 mm. The dependence of the surface-averaged heat sink coefficient (\( \alpha \)) on the liquid flow rate (W) through the nozzle for different diameters is shown in figure 4.

![Figure 4](image)

**Figure 4.** Dependence of the heat sink coefficient (\( \alpha \)) on the flow rate of the liquid (W) supplied through the nozzle with its various diameters (d) when h = 0.012 m.

When the flow rate of the liquid (W) increases, the heat sink coefficient (\( \alpha \)) increases accordingly. However, the dependence of the heat transfer coefficient and the nozzle diameter (d) has not been revealed. Figure 5 shows the dependence of heat transfer at different flow rates (W) and distances (h). Within the range \( h/d = 0.13 \ldots 0.53 \), the values of \( \alpha \) are independent of h.

![Figure 5](image)

**Figure 5.** Dependence of the heat sink coefficient \( \alpha \) at the nozzle supply from liquid flow rate W when d = 0.038 m.

It was previously shown that as a result of visualization, the critical Reynolds numbers were determined. These numbers correspond to different flow regimes of the impact jet (figure 3). The performed heat engineering studies made it possible to determine the similarity equations for various flow regimes:

when \( Re_d = 5 \cdot 10^3 \ldots 2 \cdot 10^4 \)

\[
Nu = 5.7 \cdot Re_d^{0.45} Pr^{0.33} \left( \frac{Pr}{Pr_e} \right)^{0.25}
\]  

(1)

when \( Re_d = 2 \cdot 10^4 \ldots 10^5 \)

\[
Nu = 0.15 \cdot Re_d^{0.83} Pr^{0.33} \left( \frac{Pr}{Pr_e} \right)^{0.25}
\]  

(2)

when \( Re_d = 10^5 \ldots 4 \cdot 10^5 \)
\[ Nu = 26 \cdot R_d^{0.4} \cdot Pr^{0.33} \left( \frac{Pr}{Pr_s} \right)^{0.25} \]

where \( Nu \) is the Nusselt number, \( Pr \) and \( Pr_s \) are, respectively, the Prandtl for the temperature of the liquid and the temperature of the surface of the heat transfer wall.

The dependences are correct when \( Pr=2.54\ldots4.87, \ Pr/Pr_s=0.66\ldots0.96, \ h/d=0.13\ldots0.67, \ R/d=6.6\ldots13.9 \).

4. Discussion

However, on some ships, especially with a high power-to-weight ratio, the use of a nozzle supply of cooled water cannot ensure the operation of the system in a closed loop for an unlimited time. It is possible to ensure cyclical operation of such a system, which is typical for ships of the technical fleet [7]. Thus, for the floating cranes (figure 6) the crane mode is predominant in duration and is carried out, as a rule, in the coastal area with a great number of plankton, fish eggs and fish in the water. In most cases, the ship power plant load is 10% of the rated power, and sometimes can increase up to 35% [2-4, 10-13].

Figure 6. Relative load of power plant of floating cranes: 1 – “Chernomorets”; 2 – "Vityaz"; 3 – "Bogatyr".

Let us consider the possibilities of taking into account the load of the ship power plant on the example of self-propelled floating cranes of the projected design No. 15201 with a lifting capacity of 500 tons. The diesel generating set, which ensures the operation of the crane's lifting mechanisms, has a completely closed cooling system with a plating heat exchanger. In such gear, located on the inner side of the ship hull shell, the movement of the inner coolant along the labyrinth channel is provided [7]. The average heat removal of the cooling system during crane operation is about 250-300 kW and it is fully provided by a plating heat exchanger. During periods of direct lifting and movement of large cargoes, it is necessary to turn on one main diesel generator, which requires the additional heat removal of about 900 kW. We assume that the cyclicity of such connections corresponds to the data in figure 7, and the open-loop cooling system is transferred to closed-loop operation due to the temporary connection to the ballast tank filled with water (see figure 2). The rest of the time, circular pumping of water is carried out in order to cool water through the walls of the tank. The calculation results are shown in figure 7.
Figure 7. Dependence of changes in water temperature $t_2$ in the system (tank) over time $\tau$ at different values of the mass of water in the tank $M_2$, seawater temperature $t_3$ and initial temperature in the tank $t_{2i}$:

1. $M_2 = 350$ tons, $t_3 = 15^\circ C$, $t_{2i} = 16^\circ C$; 2. $M_2 = 350$ tons, $t_3 = 25^\circ C$, $t_{2i} = 26^\circ C$; 3. $M_2 = 100$ tons, $t_3 = 15^\circ C$, $t_{2i} = 16^\circ C$; 4. $M_2 = 100$ tons, $t_3 = 25^\circ C$, $t_{2i} = 26^\circ C$.

When the value of $M_2$ is low, a faster heating of water in the tank is observed. However, when the main diesel generator is turned off, the cooling process is also fast. Thus, when $M_2 = 100$ t and seawater temperature is $25^\circ C$, $t_{3} = 25^\circ C$, the system will be able to operate for about 4 hours until the temperature reaches $32^\circ C$, $t_{2} = 32^\circ C$. When filling the tank up to 350 tons $M_2 = 350$ tons, the operation time is 9...10 hours. With a decrease in the seawater temperature, working conditions become more favorable, and the system operation time increases significantly. When the temperature reaches $32^\circ C$, a longer break in operation or change of water in the tank when entering a clean outboard water area can be performed.

The impact jet system for supplying water to the ship hull shell in the area of ballast tanks was used on self-propelled floating cranes of the projected design No. 16491 with a lifting capacity of 140 tons (figure 8).

Figure 8. Self-propelled floating crane of the projected design No. 16491.

5. Conclusion

Thus, the use of seawater ballast tanks available on the ship and the proper arrangement of the water flow in these tanks along the ship hull shell make it possible to extend the operation time of the standard open-loop cooling system in a closed circuit. As a result, the intake of seawater is reduced or even completely excluded. Therefore, the negative anthropogenic impact of cooling systems decreases.
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