Heat transfer of flooded impact jets

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Abstract. This paper presents the results of experimental studies on providing heat to the inner surface of the shell plating of the ship through the use of impact jets. In this work, we carried out (by taking into account the existing design features) a modeling and a study of the flow and heat transfer of flooded jets that flow over a flat surface and then spread radially. The visual study revealed different patterns of radial flow of the wall jet. Critical Reynolds criteria and relevant summarizing similarity equations describing heat transfer are determined.

Nomenclature

\( d \) – nozzle inner diameter
\( h \) – clearance between the nozzle edge and the surface
\( R \) – radius of the inscribed circle of the heat transfer surface
\( W \) – liquid flow rate

Similarity criteria:

\( \text{Nu} \) – Nusselt criterion
\( \text{Pr} \) – Prandtl criterion
\( \text{Re} \) – Reynolds criterion

Greek characters:

\( \alpha \) - heat transfer coefficient
\( \xi \) - local resistance coefficient

Indices

\( s \) – surface
\( d \) – diameter
\( h \) – clearance

1. Introduction

The problem of creating of highly effective environmentally friendly marine engine cooling systems required the development of an efficient method of heat transfer through the shell plating (Fig. 1). For the case described water is almost still on the inner side of the shell plating. Heat transfer from the cooled water to the inner surface of shell plating is carried out with gravitational convection. It is well known that the values of heat transfer coefficients in this case are rather small. The use of impact jets allows significantly increasing he heat transfer using relatively simple means.
Impact jets are widely used in practice. They simplify surface heating or cooling processes. At the same time, they allow achieving high enough values of heat transfer coefficients. At the nozzle's outlet, an axisymmetric jet is formed which is transformed into a diverging radial wall jet after a collision with the surface. At the same time, the following factors have an impact on these processes: design features of the nozzle-surface system, Reynolds criterion, and other factors [1, 2, 3, 4].

A distinctive feature of the shell plating is the presence of power components on the inner side. These power components are usually positioned perpendicularly to the surface of the shell plating (Fig. 1). The liquid is supplied into the space between the elements of the frame.

![Figure 1. The scheme of heat sink through the shell plating of the ship](image)

1: shell plating of the ship; 2: container with the cooled water; 3: cooled water supply manifold; 4: nozzles; 5: elements of the frame; 6: outside seawater.

It is clear that such a structure will affect the impact jet flow pattern and the heat transfer process. Therefore, it is required to determine the characteristics of the processes and the corresponding dependencies describing heat transfer.

To obtain this information experimental studies have been carried out.

2. Experiment

For more complete understanding of the underlying processes we carried out combined visual and thermal engineering studies. For this purpose corresponding experimental plants have been created.

The used model is a rectangular container with a transparent wall with dimensions 0.25x0.25m. This wall simulated the shell plating of a ship. On the inner side, perpendicularly to the wall, in the center, the inlet nozzle was installed. There was a possibility to vary the distance from the nozzle to the wall. The container has been completely filled with water, which created conditions for a formation of a submerged jet. The model provided control of the fluid flow through the nozzle. It allowed achieving the maximum speed of water in the nozzle of 1.8m/sec, which corresponded to the value of the Reynolds criterion of \(2.4 \times 10^5\). For the flow visualization purposes we added water based darkening liquid, aluminum paint, and a surface-active agent. We carried out observations from outside the transparent wall, which was additionally illuminated with a direct light.

Thermo technical studies were carried out on a specially designed experimental model (Fig. 2). It consisted of two parts: space 1 (hot cooled water) and space 2 (cold cooling water). The dimensions of the former were \(0.5m \times 0.5m \times 0.5m\), and those of the latter were \(0.5m \times 0.5m \times 0.2m\). Both spaces had a common wall 3 with dimensions of \(0.5m \times 0.5m\) and thickness of \(5mm\), through which the transfer of heat was carried out. The supply of the hot water to the center of wall 3 was carried out through an insulated pipe 4 with a screwed end. To this screwed end 4, various attachments (such as a nozzle) could be fixed. There also existed an opportunity to change the distance \(h\) from the end of the nozzle.
to wall 3. The space 1 has been completely filled with water. All lateral surfaces of the experimental model have been carefully insulated.

We measured the temperature of wall 3 on nozzle’s side using seven chromel-copel thermocouples with the diameter of thermo electrodes of 0.15 mm. We measured the thermo EMF using a potentiometer. In our model, we also measured the temperature of inlet and outlet of hot and cold water.

The experimental model was connected to the thermal hydraulic stand. There was a possibility to control the flow rate of hot water. Maximum value of flow rate reached 11 m$^3$/h. There also was a coolant flow measuring system. System electric heater allowed setting different power values. In the course of experiments, water temperature reached 80…85°C, which corresponds to the maximum possible temperature level in real conditions.

The pressure drop on the test device was determined using a model pressure gauge. At low liquid flow rates (low pressure drops) the measurements were carried out using a hydro-differential pressure gauge.

![Figure 2. Experimental model scheme of thermo technical studies. 1: hot (cooled) water space; 2: cooling water space; 3: heat transfer wall; 4 and 5: hot water inlet and outlet pipes, respectively; 6 and 7: cooling water inlet and outlet pipes, respectively.](image)

3. Results and discussion
The results of the accomplished visual studies at different values of Reynolds critera are shown in Fig.3. The liquid that leaves the nozzle hits the surface and then spreads in radial directions, forming a radial wall jet.

It is clearly seen that as Re changes, the pattern of the flow along the surface changes accordingly. For instance, when Re$_d$ = 1.65 $\times$ 10$^3$ the laminar fluid flow is observed. At the same time slow-moving annular structures resulting from interaction of an impact jet with the surface were clearly observed. As the value of the Reynolds criterion increases, the size of the annular structures decreases and their more intensive outward movement is observed. Turbulent pulsations arise on the outer part of the rings; these pulsations then tend to decay. When Re$_d$ equals 2.4 $\times$ 10$^5$, multiple developed turbulent swirls are observed throughout the area. The intensity of these turbulent swirls is maximal in the center and decreases as the distance from the center increases.
Figure 3. Pattern of wall jet flow with nozzle-based liquid supply.

The visual studies allowed determining that the lower critical Reynolds criterion is approximately in the range of values of \((1...2) \times 10^4\), and the upper one - in a range of values near \(1 \times 10^5\).

In the course of direct observation of the processes taking place in the depth, at a distance from the wall, secondary flows moving toward the nozzle have been detected. This was caused by suction of fluid from the surrounding space, which confirms the existing understanding of fluid movement in the area of the outlet nozzle edge [4, 5, 6].

Thermo technical studies were conducted using nozzles with an inside diameter of 18 mm and 38 mm. The distance \(h\) from the nozzle to the surface was set to 5 mm, 12 mm, and 20 mm. The dependence of the average (across the surface) heat transfer coefficient \(\alpha\) on the rate of fluid flow \(W\) through the nozzle at different diameters is shown in Fig. 4.

Figure 4. The dependence of the heat transfer coefficient \(\alpha\) on the rate of fluid flow \(W\) supplied through the nozzle at different values of diameter \(d\) \((R = 0.25 m; h = 0.012 m)\).

Increasing \(W\) leads to increasing of \(\alpha\) values. However, no interaction between heat transfer coefficient and diameter \(d\) of the nozzle has been revealed. Fig. 5 shows the dependence of heat transfer at different rates \(W\) and distances \(h\). In the range \(h/d = 0.13...0.53\) the values of \(\alpha\) do not depend on \(h\).
Figure 5. Dependence of heat transfer coefficient $\alpha$ on the fluid flow rate $W$ at the nozzle-based supply ($d = 0.038\,m; R = 0.25\,m$).

The results of experimental data summarizing are shown in Figure 6.

Figure 6. The dependence of $\lg \left( \frac{Nu}{Pr^{0.33} \left( \frac{Pr}{Pr_s} \right)^{0.25}} \right)$ from $\lg Re_d$ at

$R = 0.25\,m$; $Re_d = 5 \times 10^3 ... 4 \times 10^5$; $Pr = 2.54 ... 4.87$; $Pr/Pr_s = 0.66 ... 0.96$;

$h/d = 0.13 ... 0.67$; $R/d = 6.6 ... 13.9$;

$a$: $Re_d = 5 \times 10^3 ... 2 \times 10^4$; $b$: $Re_d = 2 \times 10^4 ... 10^5$; $c$: $Re_d = 10^5 ... 4 \times 10^5$

Based on the results of visual studies, the experimental points are approximated by a line with two shoulders. Each section of this line corresponds to a different flow pattern. Moreover, according to the data summarizing, the lower critical Reynolds criterion is the value of $2 \times 10^4$, and the value of $1 \times 10^5$ is the upper critical Reynolds criterion. For the selected sections of the approximating line, the following similarity equations are defined:

For $Re_d = 5 \times 10^3 ... 2 \times 10^4$:

$$Nu = 5.7 \times Re_d^{0.45} Pr^{0.33} \left( \frac{Pr}{Pr_s} \right)^{0.25};$$

(1)

for $Re_d = 2 \times 10^4 ... 10^5$:

$$Nu = 0.15 \times Re_d^{0.83} Pr^{0.33} \left( \frac{Pr}{Pr_s} \right)^{0.25};$$

(2)
for \( \text{Re}_d = 10^5 ... 4 \cdot 10^5 \):

\[
\text{Nu} = 26 \cdot \text{Re}_{d}^{0.4} \text{Pr}^{0.33} \left( \frac{\text{Pr}}{\text{Pr}_{r}} \right)^{0.25} .
\]  \( (3) \)

The dependencies are valid for: \( \text{Pr} = 2.54..4.87 \), \( \text{Pr}/\text{Pr}_{r} = 0.66..0.96 \), \( h/d = 0.13..0.67 \); \( R/d = 6.6..13.9 \). A typical linear size of the Reynolds criterion is \( d \).

The visual studies revealed different patterns of fluid flow along the heat exchange surface in the case of nozzle-based and screen-based ways of the fluid supply.

The joint analysis of the results of visual and thermo technical studies in the case of nozzle-based supply allowed determining the boundaries of the different regimes of fluid flow. In this case, the lower critical number is \( \text{Re}_d = 2 \times 10^3 \), and the upper one is equal to \( 10^5 \). For each of the flow patterns the dependencies describing heat transfer have been determined. Fig. 7 presents a comparison of these dependencies with the known ones. There is a good coordination with operation [8], where the average heat transfer was determined for an impact jet in the range \( h/d = 0.25..1.0 \). The maximum difference between the results (in the range of \( \text{Re}_d = 5 \times 10^3 ... 4 \times 10^4 \)) does not exceed 30...35%. Line 2 acts as an averaging line for the zigzag line 1 describing the results of the present study. This is explained by the fact that in [8] the fact of changing fluid flow patterns has not been taken into account, whereby the obtained experimental data is approximated by a straight line in the logarithmic coordinates.

### Figure 7. Comparison of impact jet heat transfer study results for

- \( h = 0.020 \text{m} \); \( R = 0.25 \text{m} \); \( d = 0.03 \text{m} \)
- 1: a: dependence (1); b: dependence (2); c: dependence (3)
- 2: [7] for \( h/d = 0.25..10 \); 3: [9] for \( h/d = 0.5..6.2 \); 4: [11,12]; 5: [8];
- 6: [10] for \( h/d = 3..8 \); 7: [13] for \( D/d = 2..16 \)

In the dependencies (1)–(3) obtained in this work, the maximum value of power (equal to 0.83) of the number \( \text{Re}_d \), is observed in the case of the transition area (\( \text{Re}_d = 2 \times 10^3 ... 10^5 \)). In laminar and turbulent flows, these values equal to, 0.45 and 0.4 respectively. The results for the laminar flow should be compared with the results of [6, 7, 8]. As the range of the Reynolds criteria offsets towards lower values, the slope of the line decreases. In the case of [9], the slope becomes almost equal to that observed in the present work. A certain exception in this regard is the results of [10], where the slope is large enough, but it does not exceed the value observed for the transition area. In this case, the design of the inlet nozzle or a high turbulence of the supplying flow could impact the results [5].
4. Conclusions

Visual studies conducted in this work revealed different wall jet flow patterns. This allowed determining the heat transfer summarized dependencies that correspond to each pattern. Eventually, using rather simple methods the initial problem of increase of heat transfer through the shell plating of the ship is solved.

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