Improving the compact heat exchangers with profiled tubes for marine power plants

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Abstract. The design features of heat exchangers with flat tubes are considered. The stages of creating a new flat tube and its prototypes are analyzed. An oil refrigerator is chosen as a prototype of the heat exchanger because of the previous use of flat tubes with wells, now replaced with new flat profiled tubes. Thermal and hydraulic tests of a refrigerator made of such tubes are carried out at the stand. During the testing, the hydraulic resistance of the cavities of the cooled and cooling media and thermal parameters are determined: heat power, heat transfer coefficient and heat transfer coefficient from the cooled medium in the inter-tube cavity with the transverse flow of tubes to the tube wall. A satisfactory correspondence of the actual power values determined for both working environments has been established. The discrepancy does not exceed (-7.6%)-(+)5% with an average value of +0.2%. A satisfactory correspondence of the actual and calculated values of the refrigerator power has been obtained. The discrepancy does not exceed (-15%)-(+)9% with an average value of -2.8%. The calculation of the capacity of the refrigerator during its operation in the design mode of oil cooling is carried out. The oil flow is considered both through the pipe and through the inter-pipe space. A good convergence of the calculations with the experimental results has been revealed.

1. Introduction
The sustainable combat capability of the ship and its survivability are associated with the rational use of areas and volumes to accommodate modern weapons systems and ship equipment. One of the ways of such rationalization is to reduce the size of the ship's power plant and its auxiliary equipment. The main elements of the auxiliary equipment of the ship's power plant are heat exchangers. Reduction of their masses and dimensions while maintaining or increasing operation efficiency remains an urgent task. This is ensured by intensifying heat transfer processes by turbulizing the flow of working fluid and increasing the effective areas of heat exchange surfaces. To combine the efficiency of plate and shell-and-tube heat exchangers, a new profile of a flat tube of complex cross-section is proposed. Due
to this, the weight and size characteristics of the device are minimized while maintaining the required operating modes.

The issues of heat exchangers compactness were approached by such specialists as V.M. Kees, A.L. Longlon [1], as well as several Russian scientists. Since compactness is achieved by intensifying heat transfer, its various methods were considered in the works of G.A. Dreitser and B.V. Dzyubenko [2, 3]. The heat transfer of various surfaces was investigated by such specialists as S.S. Kutateladze [4, 5]. Nevertheless, the issue of compactness of heat exchangers in the SEU requires further study.

The object of research is the technology of manufacturing ship heat exchangers with flat profiled tubes. The subject of research is the efficiency and weight and size characteristics of ship heat exchangers. Purpose of the work: substantiation of the design and manufacturing technology of compact heat exchangers with profiled tubes for marine technology.

To achieve the set research goal, it is necessary to solve the following tasks:

- to establish the state-of-the-art in the design and manufacturing technology of compact ship-based heat exchangers;
- to develop a model of compact heat exchangers with flat tubes;
- to develop an algorithm and a program for calculating heat exchangers with flat profiled tubes;
- to develop technologies to manufacture heat exchangers in general and a flat tube in particular;
- to establish the effectiveness of flat profiled tubes in ship heat exchangers experimentally;
- to carry out a comparative analysis of the efficiency of heat exchangers with flat and round tubes.

We provide an overview of the main heat exchanger structures, classified according to the method of heat transfer and by purpose. Various heat exchangers are considered by types of structures. The design of heat exchangers with flat tubes is discussed as well. The history of the creation of a new flat tube and its prototypes is briefly described. An MHD-4 oil cooler is chosen as a prototype of the heat exchangers (Figure 1). It previously used flat tubes with dimples, which have been replaced by new flat profiled tubes.

An algorithm and a program for calculating the heat exchangers of the proposed design have been developed. They are based on the thermal calculation of heat exchangers: the outlet temperature of one medium is preset (in this case, the temperature of the medium to be cooled); the balance value of the heat exchanger power is calculated; the outlet temperature of the second medium is determined; the average temperature of both media and the average temperature head are set. Then the values of the heat transfer coefficients of the outer and inner surfaces of the heat transfer pipes and the heat transfer coefficient between the heat exchanging media and the values of the transmitted power are calculated.

2. Heat exchangers with flat tubes

MHD-4 oil cooler is chosen as a prototype of the heat exchanger.
Working fluid in MHD-4 are oil as a cooled fluid and outboard (sea) water as a cooling fluid. The cooled fluid flows in the tubes, and the cooling fluid flows in the annular cavity.

Due to the complexity of the tube shape (Figure 2), the flow in it is uneven, and different velocities lead to different heat transfer coefficients for different parts of the tube.

To determine the values of the flow rate of the liquid inside, each tube is conventionally divided into 2 elliptical channels and 4 round ones, the calculation of heat transfer for them is carried out separately.

The proposed ratio:

$$\Delta P_l = \left[ \left( \frac{\zeta_{in} + \zeta_{tab}}{d_g} + \zeta_{out} \right) \frac{G^2}{2F^2\rho} \right]_{l} \text{ = idem,}$$

where

$\Delta P$ - hydraulic resistance;

$\zeta$ - the coefficient of local resistance;

$L$ – channel length;

$dg$ - hydraulic diameter;

$G$ - the flow rate of the working medium;

$F$ - the area of the passage of the working medium in the channel;
\( \rho \) - the density of the working medium;

\( i \) - channel number.

In parallel connected channels of two different standard sizes, but of the same length, the flow rates in different channels, the area of the media passage and the coefficients of hydraulic resistance are related by the ratio

\[
[K1G/(dg)^2F]_i = \text{idem}
\]

Then

\[
G1/G2=K2(dg1)^2F1/K1(dg2)^2F2,
\]

and with a known value of the total flow rate of the fluid through all channels:

\[
G1/GΣ=1/(1+K2(dg1)^2F1/K1(dg2)^2F2),
\]

and

\[
G1= GΣ-G2.
\]

It is possible to find the flow rate of the working fluid in each channel, and, subsequently, the speed of the working fluid, the Reynolds number and the heat transfer coefficient. The final heat transfer coefficient was calculated as an average, considering the perimeters of the channels.

The transverse flow around the tube bundle is considered by analogy with the flow around the in-line bundle of membrane tubes.

To calculate the flow around a bundle of ribbed tubes in general form, the heat transfer coefficient from the flow to the tube wall is first determined when flowing around a similar bundle of smooth tubes. Then, to consider the effect of ribbing, the heat transfer coefficient for a bundle of smooth pipes is multiplied by a correction coefficient.

Thermal-hydraulic tests of the oil cooler MHD-4 were carried out at the complex stand of NPO CKTI (St. Petersburg). To simplify the testing, the TA working media were replaced with fresh water in both circuits. The features of the bench equipment regulated the supply of the cooled fluid into the inter-tubular cavity of the heat exchangers, and the cooling one directly into the tubes.

In the process of testing the MHD-4 refrigerator, the hydraulic resistance of the cavities of the cooled and cooling fluid and the thermal parameters (thermal power, heat transfer coefficient and the coefficient of heat transfer from the cooled medium in the annular cavity with transverse flow around the tubes to the tube wall) were determined (Table 1). Conclusions have been drawn on

- satisfactory correspondence of the actual power values, determined for both working environments (Mi; M2). The discrepancy does not exceed (-7.6%) - (+ 5%) with an average value + 0.2%;
- satisfactory correspondence between actual and calculated power values MHD-4 (No. and M). The discrepancy does not exceed (-15%) - (+ 9%) with an average of -2.8%.

| №  | G1  | P1  | Cooled fluid | Cooling fluid | Nd1 | dP1 | G2  | P2  | Nd2 | dP2 | Np  |
|----|-----|-----|--------------|--------------|-----|-----|-----|-----|-----|-----|-----|
|    | kg/s| MPa | T1(C)        | T1(C)        | kPa | kg/s| kPa | MPa | kWt | kPa | kWt |
| 1  | 2.39| 0.19| 28.5         | 25.5         | 29  | 3.71| 0.8 | 0.37| 2.9 | 11.1| 27  | 2.00| 33  |
| 2  | 2.05| 0.20| 31.3         | 27.6         | 32  | 3.43| 0.79| 0.37| 2.9 | 12  | 30  | 2.02| 35  |
| 3  | 2.21| 0.19| 42.6         | 37.6         | 47  | 3.5 | 0.78| 0.36| 2.8 | 16.7| 46  | 1.97| 50  |
| 4  | 2.52| 0.17| 45.4         | 40.2         | 50  | 3.84| 0.77| 0.36| 2.8 | 18  | 49  | 1.84| 54  |
| 5  | 2.18| 0.19| 49.4         | 43.3         | 55  | 3.6 | 0.78| 0.36| 2.7 | 19.3| 54  | 1.92| 59  |
| 6  | 2.47| 0.17| 50.8         | 54.3         | 57  | 3.97| 0.77| 0.36| 2.7 | 20.3| 57  | 1.94| 62  |
| 7  | 2.69| 0.17| 52.6         | 47.3         | 60  | 4.18| 0.77| 0.36| 2.8 | 21.2| 60  | 1.9  | 65  |
| 8  | 2.48| 0.18| 54.4         | 47           | 76  | 3.94| 1.43| 0.33| 2.7 | 15.4| 76  | 6.25| 82  |
| 9  | 2.18| 0.20| 60.7         | 53           | 70  | 3.85| 0.77| 0.35| 2.8 | 24.5| 70  | 1.91| 73  |
3. Heat exchangers with flat tubes of other designs
Replacing round tubes with flat ones in shell-and-tube coolers of the OKP series allowed reducing the weight and size characteristics of existing coolers 2-2.5 times.

The oil is the cooling fluid flowing in the vessel, and the cooling medium (seawater) flows in the pipes.

The OKP 29-42 cooler is selected as a prototype (Figure 3).
Figure 3. General view of OKP 29-420.

The purpose of the tests was to establish the agreement between the measured thermohydraulic characteristics of the modified heat exchanger and the calculated data, to compare them with the characteristics of a shell-and-tube cooler of a classical design with round pipes.

The algorithm and program for the thermal calculation of the OKP 29-420 cooler are built on the same principle as for the MHD-4 oil cooler, but they have several differences. So, to calculate heat transfer at the outer surface of pipes in the ideal case (no leaks through the baffles, no additional baffles, only the transverse flow around the tube bundle is considered), recommendations for the calculation of heat transfer on the surface of the located corridor membrane bundles were used. For the design operating mode of the OKP 29-420 cooler (Tout = 35° C and N = 702 kW), the temperature of the oil at the outlet of the cooler (T1out = 87° C) and its power (N = 313 kW) were determined. The calculation results are shown in table 2:

| Initial data | Required values for the project | Calculation results |
|--------------|---------------------------------|--------------------|
| G1           | T1in                            | G2                 |
| kg/s         | °C                              | kg/s               |
| T2in         | °C                              | T1out              |
| °C           | °C                              | N                  |
| °C           | kWt                             | N1out              |
| °C           | °C                              | T2out              |
| °C           | kWt                             | N                  |
Due to the revealed inconsistency between the design and calculated data, the formula for calculating the thermal conductivity coefficient is used:

\[ \alpha = 10.5 \lambda P_r^{0.33} \left( \frac{G}{\mu} \right)^{0.59} \]

The actual value of the heat transfer coefficient of the cooled fluid \( \alpha \) is three times less than the calculated value of \( \alpha_{\text{theor}} \). This is due to the presence of medium flows through the gaps in the partitions between the channels and the stagnation of the fluid in the cavities between the additional baffles.

To reduce the area of the gaps between the intermediate baffles, the design of the profiled tube is changed. In this case, the pipe sections passing through the baffles are not profiled (see Figure 4).

In the old design, the tubes were profiled along the entire length, with the exception of the end areas of connection to tube sheets (see Figure 5).

![Figure 4. New tube design.](image)

![Figure 5. Old tube design.](image)

To ensure the counterflow of working fluid inside the heating unit, a longitudinal baffle is introduced (Figure 6). The proposed design has no seal assembly, but the technological gap between the longitudinal baffle of the tube bundle and the shell is minimized. To ensure a uniform flow of the working fluid in the annular cavity, the tube bundle is rotated 90 degrees.

Negative factors are minimized. The heat exchange surface area has increased by 20% without increasing the body diameter.

| Table 3. Calculated data for various cooler designs. |
|----------------|----------------|----------------|----------------|
| \( T_{1\text{in}}, \, ^\circ C \) | \( T_{1\text{out}}, \, ^\circ C \) | \( N, \text{kWt} \) |
|----------------|----------------|----------------|
| Without longitudinal baffle | Without longitudinal baffle | Without longitudinal baffle |
| 120 | 45.4 | 635 |
| 80 | 37.4 | 314 |
| With longitudinal baffle without seal unit | With longitudinal baffle without seal unit | With longitudinal baffle without seal unit |
| 120 | 42 | 657 |
| 80 | 35.2 | 328 |
| With longitudinal baffle and seal assembly | With longitudinal baffle and seal assembly | With longitudinal baffle and seal assembly |
| 120 | 35.5 | 691 |
| 80 | 30.8 | 349 |
Figure 6. General view of OKP 29-420.
The OKP 29-420 cooler of a new design with flat tubes and a cooler with round tubes have passed the heat engineering tests organized by Vineta LLC with Azrogaz LLC on the territory of the CIAM Research Center (Lytkarino, Moscow region). It has been established that the characteristics of flat-tube and circular-tube coolers are fully consistent when operating in accordance with the specification requirements (Table 4 and 5). According to the test results, flat-tube coolers of the OKP type are recommended for use on various ships.

Table 4. Test results of circular tube cooler OKP 29-420 at project operating modes.

| Temperature | ≈50 | ≈80  | ≈120 |
|-------------|-----|------|------|
| **Heating** |     |      |      |
| Oil inlet/outlet | 50/31.2 | 50.8/31.9 | 80.8/37.6 | 82.3/39.2 | 117/48.7 | 120/50.3 |
| Water inlet/outlet | 26/27.5 | 26/28 | 26/31 | 27/32 | 28/35 | 28/36 |
| **Cooling** |     |      |      |
| Oil inlet/outlet | 50/35.2 | 51.6/36 | 80.8/42.4 | 83.2/43.1 | 108/49.5 | 111.1/50.3 |
| Water inlet/outlet | 30/32 | 30/32 | 30/33 | 29/33 | 28/35 | 28/35 |

Table 5. Test results of flat-tube cooler OKP 29-420 at project operating modes.

| Temperature | ≈50 | ≈80  | ≈120 |
|-------------|-----|------|------|
| **Heating** |     |      |      |
| Oil inlet/outlet | 50/28.7 | 50.8/29.5 | 80/38.4 | 81.5/39.2 | 120/51.1 | 120.5/51.8 |
| Water inlet/outlet | 16/18 | 17/19 | 17/22 | 17/22 | 17/25 | 17/25 |
Cooling Oil
inlet/outlet 50.8/31.2 53.1/31.9 80.8/40.8 81.5/41.6 111/50.3 117/51.8
Water
inlet/outlet 19/21 19/21 19/23 19/23 18/25 18/25

4. Conclusion
A model of a new shell-and-tube heat exchanger with flat profiled tubes is proposed. It differs from shell-and-tube heat exchanger with round tubes by greater compactness while maintaining efficiency.

To eliminate the discrepancy between the calculation and experiment results, the dependence is introduced into the model to determine the heat transfer coefficient when flowing around a bundle of flat tubes.

The study has shown that the sustainable combat capability of the ship and its survivability may be ensured by reducing the size of the ship's power plant and its auxiliary equipment. Intensification of heat transfer in a heat exchanger due to turbulization of the flow of working media and an increase in the effective areas of heat exchange surfaces allows reducing the mass and size characteristics of devices 2-2.5 times while maintaining the design characteristics.

The proposed model, algorithm and program for calculating heat exchange with flat profiled tubes serve to reach a satisfactory correspondence between the actual and calculated values of the power of MHD-4 and OKP 29-420. The discrepancy ranges from (-15%) to (+9%) with an average of -2.8%.

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