Comparative analysis of energy efficiency of ballast tank anti-freezing systems

D V Bagaev, M N Syraleva and I V Kudinovich
Krylov State Research Centre, Saint-Petersburg, Russia
44 Moskovskoe shosse, St. Petersburg 196158, Russian Federation
E-mail: syraleva_masha@mail.ru

Abstract. Choosing the best way to deal with the internal icing of ballast tanks is the important task of the designing Arctic vessels. To prevent excessive ice formation in the ballast tank, various methods are used, the main of which are: heating the water and increasing of the water circulation intensity inside the tank (air and steam bubbling, forced water circulation). The report presents qualitative assessment of the heating method and air bubbling system and investigation of energy efficiency of anti-freezing systems.

1. Introduction
According to the guidelines for polar ship [1], the typical accident scenarios of the ship operating in low air temperature include: expansion of ballast water, fresh water, cargo due to freezing results in damage of the structure and falling down of the ice blocks when the ballast water is discharged while icing on the upper part of the ballast water occurs results in damage of the structure/system.
In the world the existing anti-icing techniques of ballast tank are:
- hull heating;
- air bubbling system;
- steam bubbling system;
- ultrasonic system;
- changing the physicochemical properties of ballast water;
- water circulation;
- increasing the thermal resistance of the side plating of tanks.
However, the Russian Maritime Register of Shipping [2] includes only rules for heating anti-freezing systems.
In work [3] it is shown that the application of the standard procedure [4] does not allow to correctly calculate the thermal regime of ballast tanks of complex geometric shape. For accurate thermal analysis of such tanks it is necessary to perform the full-scale CFD (computation fluid dynamics) modeling of free-convective flow near cold walls, taking into account tank design features.
Thus, the purpose of this article is comparative analysis of energy efficiency of ballast tank anti-freezing systems.
The objectives of presented research are:
- review of the principles of operation of anti-freezing methods for controlling the internal icing of ballast tanks;
- a qualitative assessment of the heating method and air bubbling system;
- development of calculating procedure anti-freezing systems;
- calculating study of anti-freezing systems energy efficiency.
2. Review of the principles of operation of anti-freezing methods for controlling the internal icing of ballast tanks

The changing in the physicochemical properties of sea water, due to the artificial increase in its salinity, in ballast tanks allows the temperature of ice formation to be reduced. However, the use of this method is justified only for small ballast tanks.

One of the energy saving method may be the setting of an additional light wall near the sides of the tanks. The effect of increasing the thermal resistance of the side is achieved by artificially creating an inactive wall layer along the side that reduces the heat transfer from the ballast water.

According to foreign rules for classification of ships [1], several anti-freezing system, such as heating coil system, ballast water circulation system and air bubbling system, have been applied to the water ballast tanks for the vessels, operating in cold sea (see Figure 1).

![Figure 1. Anti-freezing system in tanks](image)

The advantages of installing a heating system are the ability to regulate temperature conditions, widespread use and proven technologies. Obviously, the heating method is the most energy intensive.

Circulating the ballast water is considered as an efficient alternative in a short period of time because of the natural differences in temperatures between the bottom and surface of ballast tank. This system requires significant structural changes in ballast tanks (installation of a complex of pumps providing the necessary circulation).

Air bubbling system achieves the anti-freezing effect through continuous turbulence of the ballast water to circulate internally. However, the low temperature in tank leads to freezing in local regions within tank, the piping may be damaged by ice falling and coating and structural members in tanks were also damaged. It also has the disadvantage of possible under-cooling of the water and the formation of ice crystals by the cold air. Ballast water containing ice crystals makes the water have a consistence like porridge which is difficult to pump. Air bubbling system is not fit to use in an extreme cold climate.

Through the results of numerical simulations [5], it became clear that the air bubbling system equipped in each frame of water ballast tanks was effective to prevent freezing along the outer surface by promoting convective heat transfer.

At the moment, there are no comprehensive studies of the effectiveness of anti-freezing methods of water in the ballast tanks of arctic vessels. The limits of eligibility for the considered methods are not indicated. From a review of anti-freezing methods for controlling the internal icing in ballast tanks it can be concluded that the most common, reliable and not requiring significant structural changes to existing ballast tanks methods are regenerative heating, water circulation and air injection.

3. A qualitative assessment of the heating method and air bubbling system

For a qualitative assessment of anti-freezing methods, the following assumptions were used (substantiated in previous articles [3, 6]:

...
1. layer of ice exists on the entire wall bordering the outside air;
2. the thickness of the ice is uneven along the height of the wall;
3. the limit ice thickness is realized at the top of the wall and is provided with a given heat flux on the surface \( (q_{ice}) \).

Both anti-freezing systems must provide this heat flux on the surface, so the following equation is obtained:

\[
\frac{\alpha_{conv}}{\alpha_{bubbling}} = \frac{\Delta T_{conv}}{\Delta T_{bubbling}}
\]

where \( \alpha_{conv} \), \( \alpha_{bubbling} \) - heat transfer coefficients on the ice surface during free convection for heating systems and forced convection for air bubbling system, respectively; temperature difference with free convection \( \Delta T_{conv} = T_{water} - T_{ice} \), \( T_{water} \) - heated water temperature, \( T_{ice} \) - freezing temperature of sea water; \( \Delta T_{bubbling} = T_{sea} - T_{ice} \), \( T_{sea} \) - sea temperature in the navigation region.

In work [7] it is seen that the heat transfer coefficient during bubbling asymptotically does not exceed \( \alpha_{bubbling, max} = 5000 \text{ W/(m}^2\text{C)} \).

![Figure 2. Minimum water temperatures in Barents sea](image)

As can be seen from Figure 2, the minimum temperature difference in the Barents Sea is \( \Delta T_{bubbling, min} = 0.2 \text{ °C} \). The dependence of the heat flux on the ice surface, providing an ice thickness of no more than 0.1 m, on the outdoor air temperature is shown in Figure 3. As follows from Figure 3, for each ice thickness, the use of an air bubbling system has a limit on the outdoor temperature.

Thus, on the basis of a qualitative assessment, the following conclusion can be made: heating and air bubbling systems are competitively capable in the range of outdoor air temperatures up to -30 °C, however, under extreme conditions, heating systems are more reliable.
4. Calculating procedures of anti-freezing systems
The following anti-freezing systems are considered: heating system, ballast water circulation system and air bubbling system. Ice formation on cooled surfaces enclosing liquid volumes is determined by the processes of convective heat transfer (a) and forced convection (b, c) (see Figure 4).

![Figure 4. Calculation model for anti-freezing systems (a – heating system, b – ballast water circulation system, c – air bubbling system)]](image)

The main regulatory parameters of anti-freezing systems defining the ice thickness are heating power $N_{heating}$ in case of heating system; the circulate pump flow rate $G_{wp}$ in case of ballast water circulation system; the air flow rate $G_{air}$ in case of air bubbling system.

The energy efficiency of the anti-freezing systems is determined by the following characteristics: $N_{heating}$ for heating system; the power required for water circulation $N_{friction}$ for ballast water circulation system; the fan (compressor) power $N_{comp}$ for air bubbling system.

Calculating procedures anti-freezing systems are based on the following assumptions.

1. Uses a one-dimensional model (height dependent). The steady state is considered.
2. It is assumed that there is ice on the entire inner surface of the outer side.
3. Heat transfer downward waterline (WL) is negligible due to bottom plating.
4. The vertical wall is installed in the tank to organize directional movement in the gap. The thermal resistance of the partition is not taken into account. The calculations are performed for a typical ballast tank (b=0.2 m, L=7 m, H =0.7m). The ice thickness is determined from condition:

\[ q_{\text{ice}} = q_{\text{out}}. \]

The heat flux from the water to the surface of the ice is

\[ q_{\text{ice}} = \alpha(T - T_{\text{ice}}). \]

where \( \alpha \) – heat transfer coefficient, \( T \) – water temperature in the gap.

The heat flux from the ice surface to outside is:

\[ q_{\text{out}} = \frac{T_{\text{ice}} - T_{\text{out}}}{R}, \]

where \( R = \frac{1}{a_{\text{out}}} + \frac{\delta_{\text{ice}}}{\lambda_{\text{ice}}} + \frac{\delta_{\text{wall}}}{\lambda_{\text{steel}}} \) is the thermal resistance, \( \delta_{\text{ice}} \) is the ice thickness, \( \delta_{\text{wall}} \) is the wall thickness, \( \lambda_{\text{ice}} \) is the thermal conductivity of sea ice, and \( \lambda_{\text{steel}} \) is the thermal conductivity of steel.

Thermal balance in the gap bounded by a partition:

\[ -G_{\text{w}c_{\text{p}}}[T(H) - T(0)] = Q_{\text{tank}} + Q_{\text{ice}}, \]

where \( H \) – height above WL, \( G_{\text{w}} \) – water mass flow in the circuit; \( c_{\text{p}} \) – heat capacity of water in a ballast tank; \( T(0) \) – water temperature at \( z=0 \), \( T(H) \) – water temperature in the gap at \( z=H \).

The amount of heat removed from the board to the environment:

\[ Q_{\text{ice}} = L \int_{0}^{H} \alpha(T(z) - T_{\text{ice}})dz, \]

Amount of heat removed to the tank is:

\[ Q_{\text{tank}} = L \int_{0}^{H} K(T(z) - T_{\text{tank}})dz, \]

where \( L \) – tank length, \( K = \frac{1}{\alpha_{\text{tank}}} \) – heat transfer coefficient, \( \delta_{\text{part}} \) – partition thickness, \( T_{\text{tank}} \) - temperature in the tank, \( \alpha_{\text{tank}} \) – heat transfer coefficient in the tank.

Temperature distribution along the height of the heated wall:

\[ T(z) = \frac{1}{A}((AT(0) + B)e^{-Cz} - B), \]

where \( A = \alpha + K ; B = -(\alpha T_{\text{ice}} + KT_{\text{tank}}), C = \frac{AL}{G_{\text{w}c_{\text{p}}}}. \)

4.1. Heating system

The temperature height distribution, heat transfer rate and hydraulic losses and heating power are determined the following system of equations:

\[
\begin{align*}
T(z) &= \frac{1}{A}((AT(0) + B)e^{-Cz} - B); \\
\alpha &= \max(\alpha_{\text{n.conv}}, \alpha_{f.conv}); \\
(\rho_{\text{tank}} - \bar{\rho})gH &= \lambda_{\text{friction}} \frac{H^{2}}{2\rho (bL)^{2}}; \\
G_{\text{w}c_{\text{p}}}[T(H) - T_{\text{tank}}] + Q_{\text{tank}} = 0; \\
N_{\text{heating}} &= G_{\text{w}c_{\text{p}}}[T(0) - T_{\text{tank}}],
\end{align*}
\]

where \( \rho_{\text{tank}} \) – density at \( T = T_{\text{tank}} ; \bar{\rho} = \frac{1}{H} \int_{0}^{H} \rho_{0}\beta T(z)dz \) – average density, \( \rho_{0} \) - density at \( T = 0^\circ \mathrm{C} \).
\( \lambda_{\text{friction}} \) – coefficient of friction; \( \beta \) - volume expansion coefficient (the condition \( T_{\text{tank}} \geq 2^\circ \text{C} \) is considered); \( \lambda_w \) – water thermal conductivity coefficient; \( d_{\text{eq}} = \frac{2bL}{b+L} \) – equivalent diameter, \( b \) – width tank, \( Re = \frac{2g_{w}}{\mu(b+L)} Pr = \frac{\mu_cp}{\lambda_w \mu^2} Gr = \frac{g\beta H^2 \Delta T \rho^2}{\mu^2} \), \( \mu \) – dynamic viscosity,

\[
\alpha_f = \frac{N_u_{\text{conv}} \lambda_w}{d_{\text{eq}}} \quad \text{and} \quad N_u_{\text{conv}} = \begin{cases} 0.037 Re^{0.75} Pr^{0.4}, & \text{if } Re \leq 2000, \\ 0.021 Re^{0.8} Pr^{0.43}, & \text{if } Re > 2000 \end{cases}
\]

\( \alpha_n = \frac{N_u_{n,\text{conv}} \lambda_w}{H} \), \( N_u_{n,\text{conv}} = 0.63 (Gr \cdot Pr)^{0.25} \) [8].

The results of calculating the maximum ice thickness (in the upper part of the tank) depending on the input power are presented in Figure 5.

Figure 5. The dependence ice thicknesses of heating power

Usually in case of thermal steady state \( Q_{\text{tank}} \ll Q_{\text{ice}} \) due to \( T_{\text{tank}} \equiv \bar{T} \left( \bar{T} = \frac{1}{H} \int_0^H T(z)dz \right) \) - average temperature).

Figure 5 shows that acceptable ice thickness of 0.1 m [3] provided by the heating power \( N_{\text{heating}} = 2...4 \) kW depending on the outside air temperature \( (T_{\text{air}} = -10 ... -30^\circ \text{C}) \). The absence of ice in the tank is ensured at heating power \( N_{\text{heating}} > 7 \) kW \( (T_{\text{air}} = -10^\circ \text{C}) \) and \( N_{\text{heating}} > 25 \) kW \( (T_{\text{air}} = -30^\circ \text{C}) \).

4.2. Ballast water circulation system

The thermal condition of the water in the gap, hydraulic friction losses, friction pump power \( (N_{\text{friction}}) \) are determined from equations:

\[
T(z) = \frac{1}{A} \left( (AT_{\text{sea}} + B)e^{-cz} - B \right);
\]

\[\alpha = \frac{0.021 Re^{0.8} Pr^{0.43} \lambda_w}{d_{\text{eq}}} ;\]

\[\Delta p_{\text{gap}} = \lambda_{\text{friction}} \frac{H}{d_{\text{eq}}} \frac{G_w^2}{2\rho(bL)^2} ;\]

\[N_{\text{friction}} = \frac{G_w}{\rho} \Delta p_{\text{gap}} ;\]

\[N_{\text{pump}} = \frac{G_w}{D} \Delta p_{\text{pump}} \frac{\eta_{\text{pump}}}{\eta_{\text{pump}}} ,\]
where $\lambda_{friction}$ - friction factor, $\eta_{pump}$ - pump efficiency.

The head of the circulation pump ($\Delta P_{pump}$) and consequently its power ($N_{pump}$) defined by circulation system construction ($\Delta P_{pump} \gg \Delta P_{gap}$).

The results of calculating the maximum ice thickness (in the bottom part of the tank) depending on the water flow rate are presented in Figure 6.

![Figure 6. The dependence ice thicknesses of water flow rate](image)

The calculation results show that the permissible ice thickness can be achieved at low water flow rates even at outside air temperature $T_{air} = -30^\circ C$. The absence of ice is ensured at $G_w = 5000 \, kg/s$, in this case the power required to overcome friction loses $N_{friction} \approx 150 \, W$.

4.3. Air bubbling system

Calculating relationships of air bubbling are represented in [7].

The temperature height distribution, water flow and heat transfer depending on air flow are determined from equations:

$$T(z) = \frac{1}{A} \left( (AT(0) + B)e^{-cz} - B \right);$$

$$\alpha_{bubbling} = \frac{N_{bubbling} \lambda_w}{t_{layer}};$$

$$\varphi = 0.26 \left( \frac{w_{air}^2}{g\sigma} \right) \left( \frac{\rho_w}{\rho_w - \rho_{air}} \right)^{0.12};$$

$$\varphi \rho_w g H = \lambda_{friction} \frac{H}{d_{eq}} \frac{\rho_w W_w^2}{2} \left[ 1 + x \left( \frac{\rho_w}{\rho_{air}} - 1 \right) \right],$$

where $N_{bubbling} = 0.146 K_{\delta}^{1/4} P_r^{1/3}$, $K_{\delta} = \frac{w_{air}^2}{(g \sigma)^{1/3}}$, $W_{air} = Q_{air} / b_l$, $t_{layer} = \left( \frac{v^2}{g} \right)^{1/3}$, $x = \frac{1}{1 + \varphi \frac{\rho_w}{\rho_{air}}}$.

The power of air fan is determined:

$$N_{comp} = \frac{1}{\eta_{comp}} \frac{Q_{air}}{p_Z} t_{ad}';$$

$$t_{ad}' = -\frac{k}{k - 1} RT_0 \left( \frac{p_2}{p_0} \right)^{\frac{k}{k - 1}} - 1;$$
where $\eta_{\text{comp}}$ – air fan efficiency, $Q_{\text{air}}$ – volume flow rate, $k = 1.4$; $R = 287 \left[ J \, k g^{-1} K^{-1} \right]$, $T_0$ – air temperature [K]; $p_0$ – atmosphere pressure [Pa], $\rho_2(p_2)$ – air density inside bubble tube, $\rho_w$ – water density, $\xi_{\text{hole}}$ – flow energy loss coefficient ($\xi_{\text{hole}} \approx 1.5$ [7]), $d_{\text{hole}}$ – hole diameter of bubble tube, $n_{\text{hole}}$ – number of bubble tube holes, $\varphi$ – air volume fraction, $w_{\text{hole}} \equiv 10 \, m/s$ – air velocity in the hole of bubble tube.

The results of calculating the maximum ice thickness (in the upper part of the tank) depending on the air flow rate are presented in Figure 7.

![Figure 7. The dependence ice thicknesses of air flow rate](image)

The air bubbling system is able to limit icing not more than $\delta_{\text{ice}} = 0.02 \, m$ and $\delta_{\text{ice}} = 0.09 \, m$ at appropriate air temperature $T_{\text{air}} = -10^\circ C$ and $T_{\text{air}} = -30^\circ C$. The absence of ice may be ensured at air temperature $T_{\text{air}} = -10^\circ C$ by $G_{\text{air}} = 0.3 \, kg/s$, in this case the fan power $N_{\text{comp}} \equiv 250 \, W$. In case of $T_{\text{air}} = -30^\circ C$ total lack of icing cannot be obtained.

5. Conclusions
The results of this study show that the most energy effective anti-freezing systems are ballast water circulation system and air bubbling system. These ones prevent icing by supplying ambient heat from sea water. Its effectiveness depends on the salinity of the initial ballast water. External energy supplies are necessary only to ensure the circulation of water in the tank. The disadvantage of the ballast water circulation system is the difficult design of the circuit with a high capacity pump.

The air bubbling system is convenient in operation but has usage restrictions at extremely low outside air temperatures.

The heating system is not energy efficient; however, it is simple in design and prevents ballast tank icing at any extreme temperatures.

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