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GENERALIZED MODEL OF HEAT TRANSFER BETWEEN SOLID WALL AND SUBLIMATING OR MELTING RFRIGERENT

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Abstract

Heat exchange between a solid surface and the surface of a melting or sublimating refrigerant has long been used in low-temperature engineering. With such a rich history of sublimation chillers, it's surprising that no universal and reliable methods have been developed for the design of heat exchangers that remove heat by the sublimation or melting of a solid refrigerant. A simple and universal mathematical model for the process of heat transfer from the wall of the heat exchanger apparatus to the sublimating or melting refrigerant has been proposed. The analysis of this model led to the development of a system of dimensionless criteria and a criterion equation that can describe a wide range of processes involving melting or sublimating refrigerant on the working surface. Comparison of present experimental data and calculated data shows a good agreement between the results. The equations obtained can be used to improve the design of heat exchangers with melting or sublimating refrigerant.
Key words: heat transfer; melting; sublimation; criterion equation.

УЗАГАЛЬНЕНА МОДЕЛЬ ТЕПЛООБМІНУ МІЖ ТВЕРДОЮ СТІНКОЮ І СУБЛІМУЮЧИМ АБО ТАНУЧИМ ХОЛОДОАГЕНТОМ

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Анотація

Теплообмінні апарати, в яких присутні холодоагенти, що сублімують чи плавляться, давно використовуються в низькотемпературній техніці. Незважаючи на довгу історію використання подібних холодоагентів, досі не існувало зручних і надійних методів проектування теплообмінних апаратів в яких тепло підводиться до холодоагенту, що сублімує чи плавляться. В статті запропоновано просту й універсальну математичну модель процесу теплопередачі від стінки теплообмінного апарату до холодоагенту, який сублімує чи плавиться. Аналіз цієї моделі дозволив розробити систему безрозмірних критеріїв і запропонувати відповідні критеріальні рівняння, які можуть описувати широке коло процесів, пов'язаних із плавленням або сублімацією холодоагенту на робочій поверхні. Порівняння експериментальних і розрахункових даних показує хороший збіг результатів. Отримані рівняння можуть бути використані для вдосконалення конструкції теплообмінних апаратів в яких використовуються холодоагенти, що сублімують чи плавляться.
Ключові слова: теплообмін; плавлення; сублімація; критеріальне рівняння.

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Journal of Chemistry and Technologies, 2024, 32(2), 489-497

Introduction

Heat exchange between a solid surface and the surface of a melting or sublimating refrigerant has long been used in low-temperature engineering [1; 2]. The history of refrigeration itself began with the use of water ice to cool and preserve food. Obviously, the operation of such cooling system is based on heat transfer between the melting ice and a solid surface.

The installation of Karol Olszewski and Zygmunt Wroblewski, in which liquid oxygen was first obtained in 1883, already used gas cooling by sublimation of solid carbon dioxide on the wall of the heat exchanger. The history of cryogenics can be traced back to this installation [3].

The first cooling systems in space were also sublimation coolers based on solid carbon dioxide, launched into orbit in 1972 [4].

Today, sublimation cooling is used in food processing, cooling of infrared receivers, medical cooling, drying and engineering cooling, thermal suits for mine rescuers, and in other fields of technology [5; 6].

With such a rich history of sublimation chillers, it's surprising that no universal and reliable methods have been developed for the design of heat exchangers that remove heat by the sublimation or melting of a solid refrigerant. Reputable heat transfer monographs simply do not have sections devoted to heat transfer during melting or sublimation.

Perhaps the first attempt at a strict theoretical description of the heat exchange process between the wall and the sublimating refrigerant was described in the author's article [7].

Similarly, a theoretical description of the heat exchange between a dry ice pellet placed on a hot sapphire surface has been attempted on paper [8]. In this work, the Leidenfrost effect was studied for solid disks of carbon dioxide sublimating on a solid wall. However, the theoretical results obtained in this work were not generalized in the form of criterion relations, as is usually done in fluid dynamics and heat and mass transfer.

In the paper [9], the freezing due to solid-liquid direct contact heat transfer associated with sublimation was studied experimentally and theoretically using dry ice in water.

Despite the variety of sublimation cooler designs, the basic principle remains the same. The sublimation cooler consists of a thermally insulated container with a supply of solid refrigerant, a heat exchanger, and a solid heat-conductor on which the object to be cooled is placed. To enhance the heat transfer between the solid refrigerant and the heat exchanger wall, mechanical pressure is applied to the refrigerant against the wall of the heat exchanger, or/and the container with the sublimating refrigerant is filled with a liquid coolant. In the K. Olszewski and Z. Wroblewski installation, ether was used to filling the container with sublimating refrigerant as a liquid coolant [3].

This article focuses on sublimation coolers and the development of a universal and reliable method for calculating the heat exchange intensity between a sublimating or melting refrigerant and the heat exchanger wall.

Results and discussion

Analytical study of heat transfer between the heat exchanger's wall and a sublimating or melting refrigerant.

For clarity, we will use the term refrigerant sublimation only. This means that the process of refrigerant sublimation is similar to the process of refrigerant melting.

The design of the generalized cooling system with dry ice is shown in Figure 1.

In order to intensify the heat transfer in the capsule, mechanical pressure of the sublimating substance (1) on the heat transfer surface (2) is used. This can be done by means of a spring (3) or any other method.

Qualitatively, the heat transfer process in a sublimation cooler can be described as follows.

Heat dissipation from the heat transfer surface causes sublimation of the solid refrigerant upon contact with the heat transfer surface. The resulting vapor is forced out through the gap between the frozen substance and the heat transfer surface of the cooler. The thermal resistance of this gap is the main factor that determines the intensity of heat transfer.

The described heat transfer pattern between the sublimating refrigerant and the solid wall is similar to the Leidenfrost effect which occurs when a liquid comes into contact with a hot solid surface. In this case, a heat-insulating vapor layer is also created between the heated surface and the liquid, which slows down the rapid boiling of the liquid [9–14].

Journal of Chemistry and Technologies, 2024, *32*(2), 489-497

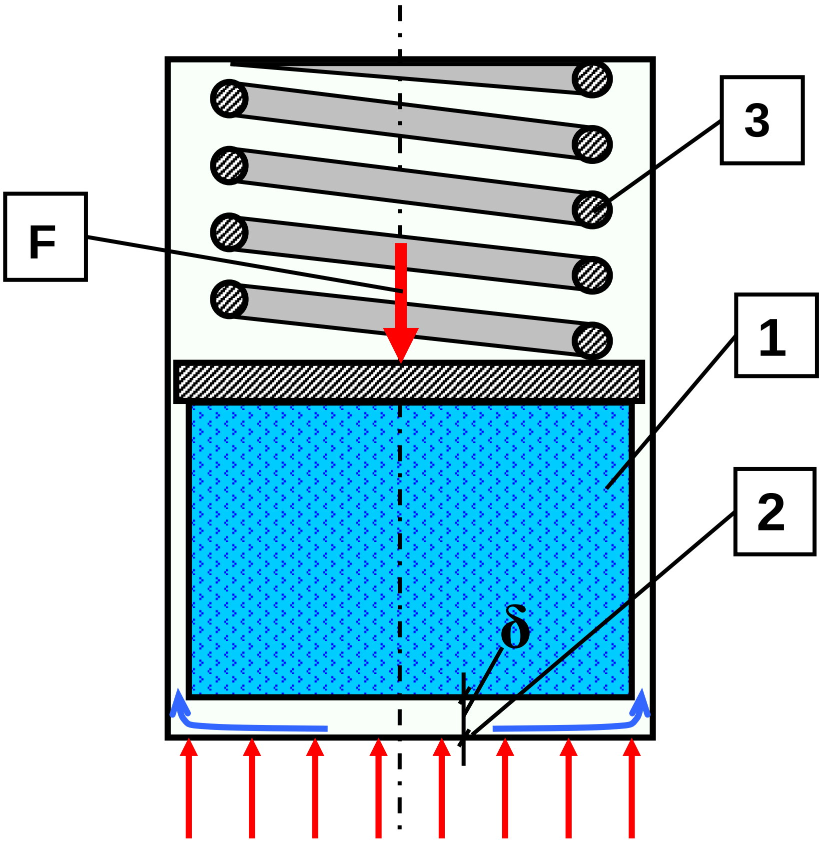

Fig. 1. Design of the generalized cooling system with dry ice.
1 is sublimating substance; 2 is the heat transfer surface; 3– spring

The size of the gap between the heat exchanger surface and the sublimating refrigerant depends on the mechanical pressure force and the vapor flow in the gap. Vapor generation in the sublimation cooler is proportional to the thermal load on the cooler. Obviously, the greater the pressure, the smaller the gap between the solid phase and the heat exchange surface, and the greater the total heat flow in the apparatus.

On the other hand, the greater the thermal load of the apparatus, the greater the vapor flow in the gap and the greater the vapor pressure under the sublimating refrigerant. An increase in vapor pressure in the gap between the refrigerant and the heat-transfer surface repels solid coolant and causes the gap to increase.

This pattern of heat exchange between the sublimating substance and the heat dissipating surface is qualitatively confirmed by experiments. In particular, it allows us to explain the experimentally established fact that concentric fins on the heat exchanging surface not only do not increase the heat transfer from this surface, but, on the contrary, reduce it. The explanation for this phenomenon is that the concentric fins impede the outflow of vapor on the heat exchange surface. This leads to an increase in the gap between the surface and the sublimating substance and an increase in the thermal resistance of this gap.

To construct a mathematical model of the process of heat transfer from the solid wall to the sublimating substance, we accept the following assumptions

- ✓ all processes and flows of vapor and energy in the apparatus are symmetrical with respect to its axis;
- ✓ the sublimation temperature of the solid refrigerant is the same at all points on the phase boundary;
- ✓ the vapor flow in the gap between the heat releasing surface and the sublimating substance is laminar;
- ✓ there is no convective heat transfer in the gap.

We will construct a mathematical model for the steady state of heat transfer in which the following parameters remain unchanged: the temperature distribution on the heat exchange surface, the solid refrigerant pressure on the heat exchange surface, and the thermal load of the device.

The constancy of the temperature field at the heat releasing surface means that in steady state the shape of the phase boundary remains unchanged even though the block of solid refrigerant is continuously moving toward the heat releasing surface. In other words, the phase boundary moves at a constant velocity relative to the solid refrigerant and is stationary relative to the heat exchange surface.

Since the speed of movement of the sublimation front relative to the solid refrigerant is determined by the heat flux density at each point of this front, this speed can be found from the heat balance equation at the phase boundary:

$$V = \frac{q}{\sigma_v} = \frac{\lambda \cdot grad(T)}{\sigma_v} = const\ , \qquad (1)$$

where V is the velocity vector of movement of a point on the phase interface; q is the vector of heat flux density at this point; λ is the thermal conductivity of the solid refrigerant vapor; T is the current vapor temperature; σ_v is the volumetric heat of sublimation of a solid refrigerant,

$$\sigma_v = \sigma \cdot \rho\ ,$$

where σ is the specific heat of sublimation of the solid refrigerant; ρ is the density of the solid phase of the refrigerant.

This leads to a very important conclusion. Since for a given solid refrigerant the volumetric heat of sublimation can be considered constant, then in a steady state the heat flux density on the heat releasing surface will be the same at all its points.

Due to the previously accepted assumption of the constancy of the sublimation temperature of the substance at the phase boundary, this phase boundary coincides with the isotherm. Therefore, the vectors of heat flux density and velocity of motion of any point on the melting front are perpendicular to the phase boundary.

Since the thickness of the gap between the solid phase and the working surface of the apparatus is several orders of magnitude smaller than the diameter of the apparatus, we can assume that the phase interface is parallel to the heat transfer surface of the apparatus, and therefore the heat flow vector is normal to the heat transfer surface of the apparatus.

From here,

$$\frac{\lambda \cdot grad(T)}{\sigma_v} = \frac{\lambda \cdot \Delta T}{\sigma_v \cdot \delta} = const, \qquad (2)$$

where ΔT is the temperature difference between the heat exchange surface and the phase boundary; δ is the size of the gap in the section of the apparatus under consideration.

In other words, on the heat exchange surface of the apparatus, the heat flux density at steady state can be considered constant regardless of how heat is applied to the heat exchange surface of the apparatus and regardless of the shape of that surface.

The physical significance of this statement is as follows: if the heat flux density at any point on the working surface is higher than at adjacent points, then sublimation of the solid phase will be intensified at that point. This leads to an increase in the thickness of the vapor layer between the solid phase and the heat transfer surface. As a result, the heat flux density at this point will decrease until it becomes equal to the heat flux density at neighboring points. In this case, the speed of propagation of the sublimation front relative to the sublimating substance becomes constant at all points of this front. The size of the gap between the sublimating substance and the heat-emitting surface is also stabilized. Conversely, if the heat flux density decreases at any point of the heat transfer surface, the gap at that point decreases.

As follows from the last formula, the ratio of the temperature difference to the gap size at all points of the heat-transfer surface remains unchanged:

$$\frac{\Delta T}{\delta} = const. \qquad (3)$$

Consider an elementary annular section of the gap between the solid phase and the working surface of the capsule. The height of this section is equal to the gap size δ, the distance from the capsule axis is equal to R, and the width of the elementary section is equal to dR. In this case, the increase of the vapor flow rate through the annular gap is equal to:

$$dG = \frac{2\pi \cdot R \cdot q}{\sigma} \cdot dR. \qquad (4)$$

Then the absolute value of vapor flow in a circular section with radius R will be equal to:

$$G = \int_0^R dG = \int_0^R \frac{2\pi \cdot r \cdot q}{\sigma} dr. \qquad (5)$$

Since we are considering the case $q = const$, this constant heat flux density can be removed from the integral, along with other constants:

$$G = \frac{2\pi \cdot q}{\sigma} \int_0^R r dr = \frac{2\pi \cdot q}{\sigma} \cdot R^2. \qquad (6)$$

Journal of Chemistry and Technologies, 2024, *32*(2), 489-497

Knowing the vapor flow rate in each circular section of the capsule, one can find the pressure drop in the gap section under consideration. Since the absolute value of the gap is relatively small, we can consider the flow in this gap to be laminar and use the well-known formula to determine the pressure drop in a gap of infinite width.

$$dP = \frac{12 \cdot \nu \cdot g_w \cdot dP}{\delta^3}; \quad (7)$$

where g_w is the vapor consumption per unit slot width,

$$dP = \frac{6 \cdot \nu \cdot G}{\pi \cdot R \cdot \delta^3} \cdot dR. \quad (8)$$

The force with which the mechanical pressure acts on a solid refrigerant is balanced by the vapor pressure generated by the sublimation of the solid refrigerant. In this case, these forces are relatively easy to determine:

$$g_w = \frac{G}{2\pi \cdot R}.$$

where ν is the kinematic viscosity of vapor.

Hence,

$$F = \int_0^{Ro} 2\pi \cdot r \cdot \left(\int_0^r dP \right) dr;$$

$$F = \int_0^{Ro} 2\pi \cdot r \cdot \left(\int_0^r \frac{6 \cdot \nu}{\pi \cdot U \cdot \delta^3} \cdot \frac{\pi \cdot q}{\sigma} \cdot U^2 dU \right) dr;$$

$$F = \frac{12\pi \cdot \nu \cdot q}{\sigma} \cdot \int_0^{Ro} r \cdot \left(\int_0^r \frac{U}{(\delta(U))^3} dU \right) dr;$$

As can be seen from the resulting formula, in order to determine the mechanical pressing force, it is necessary to find the dependence of the gap size on the radius. To do this, we first determine the temperature distribution on the working surface of the capsule and then use the formula:

$$\delta = \frac{\lambda \cdot \Delta T}{q}, \quad (10)$$

Let's find the required dependence of the value of the gap thickness on the radius.

Let us assume that the temperature distribution on the working surface of the capsule is the same at all points of this surface. Then the temperature difference between the working surface and the phase boundary will be the same everywhere, taking into account the assumption about the stability of the sublimation temperature of the solid refrigerant. Consequently, the gap thickness, which can be found from equation (10), will be the same everywhere in this case.Since the gap value is a constant, it can be removed from the integral sign in equation (9). Taking this into account, we can write:

$$F = \frac{12\pi \cdot \nu \cdot q}{\sigma} \cdot \int_0^{Ro} r \cdot \left(\int_0^r \frac{U}{(\delta(U))^3} dU \right) dr = \frac{3\pi \cdot \nu \cdot q}{2\sigma \cdot \delta^3} \cdot Ro^4. \quad (12)$$

Considering that

$$\pi \cdot Ro^2 \cdot q = Qo, \quad (13)$$

where Qo is the total heat load of the capsule, we have:

$$F = \frac{3 \cdot \nu \cdot Qo}{2 \cdot \sigma \cdot \delta^3} \cdot Ro^2. \quad (14)$$

From here, if we replace δ with its value from equation (10), we find the dependence of the thermal load of the capsule on the mechanical pressing force and the temperature difference at the working surface for a given solid refrigerant:

$$Qo = Ro \cdot \sqrt[4]{\frac{2 \cdot \pi^3 \cdot F \cdot \sigma \cdot \Delta T^3 \cdot \lambda}{3 \cdot \nu}}. \quad (15)$$

Dimensional analysis of a problem. Dimensional analysis is a standard method in the theory of heat and mass transfer. However, this powerful method has not been applied to the

Journal of Chemistry and Technologies, 2024, *32*(2), 489-497

problem of heat transfer between a melting or sublimating substance and a solid wall.

To use this method, we first write down all the dimensional variables that affect heat transfer during sublimation of a solid refrigerant on a solid wall, within the framework of the assumptions made above:

$$Qo, Ro, F, \sigma, \Delta T, \lambda, \nu .$$

So we have seven variables describing the process we want to study. Next, we will use Buckingham's π-theorem. The dimensions of these seven variables describing the heat transfer process contain five main (independent) dimensions: joule, meter, second, kilogram, degree Celsius.

Consequently, this process can be described by 7–5=2 dimensionless complexes.

There are a large number of possible ways to select these complexes, but it is best to use the following dimensionless criteria: $\Theta = \dfrac{\Delta T \cdot \lambda}{q \cdot Ro}$ and

$$\Omega = \frac{q \cdot \nu}{f \cdot \sigma \cdot Ro}, \qquad (16)$$

where f is the specific pressure in the contact zone, equal to the ratio of the absolute value of the force to the contact area of the wall and the sublimating refrigerant.

The first dimensionless criterion Θ characterizes the thermal conductivity through the vapor layer, and the second criterion Ω gives the ratio of the rates of vapor production and removal from the contact zone.

Using these notations makes aquation (15) look very simple:

$$\Theta^3 = \frac{3}{2} \cdot \Omega . \qquad (17)$$

Comparison of the obtained results with the results of experiments. Final conclusions about advantages and disadvantages of a mathematical model can be drawn only after comparison with experimental results. Experimental data presented in the work of A.P. Dvornitsyna [17–19] were used for comparison.

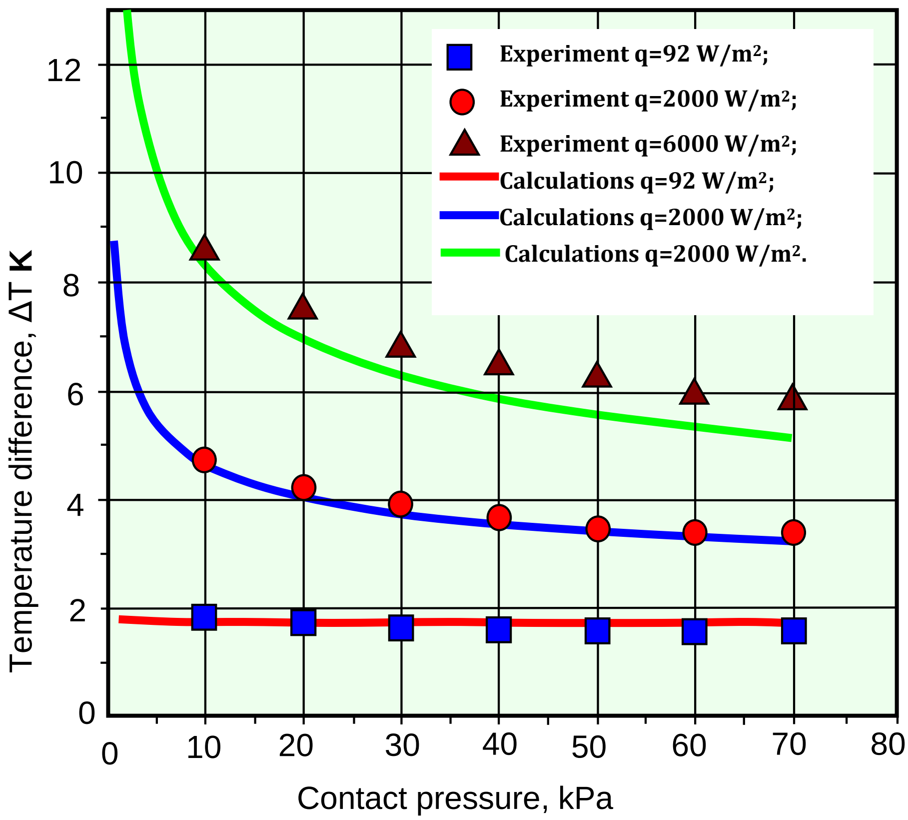

Fig. 2. Dependence of temperature difference in the contact zone on contact pressure

Journal of Chemistry and Technologies, 2024, *32*(2), 489-497

The experimental study was carried out on three capsule samples. The working surface area of the first sample was S_1=1.2·10^{-3} m^2. Since the working surface had the shape of a circle, then, knowing the area, it is not difficult to determine its radius R_1=20 *mm.* Similarly, for the second sample: S_2=2.4·10^{-3}m^2, R_2=28 *mm.* And for the third sample S_3=8.5·10^{-3}m^2, R_2=52 *mm.*

Thermal conductivity of vapor at atmospheric pressure: λ=0.0146 *W/(m·K).*

Dynamic viscosity of vapor at atmospheric pressure: μ=1.37·10 $^{-5}$*Pa·s.*

Vapor density at atmospheric pressure*:* ρ=1.976 *kg/ m*^{3}.

Kinematic viscosity of vapor at atmospheric pressure: ν=6.933·10^{-6}m^2*/s.*

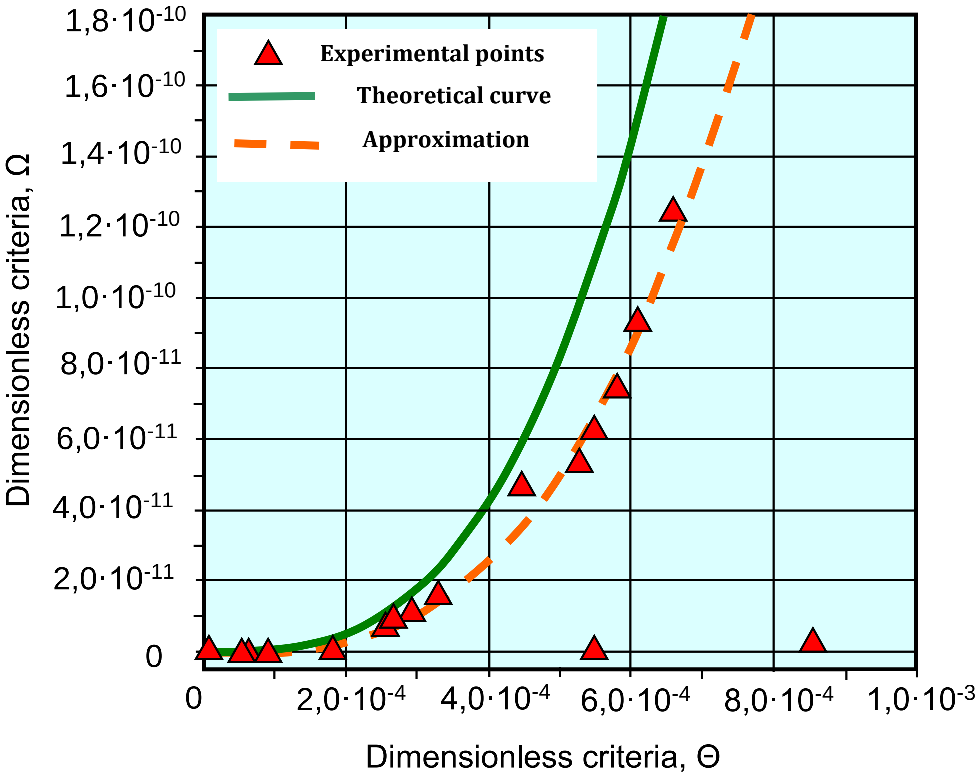

Fig. 3. Comparison of calculated and experimental data in dimensionless form

The calculations using the formulas obtained have been carried out for the atmospheric pressure. In this case, the following values of the constants included in these equations were used. The heat of sublimation of CO_2 at atmospheric pressure is assumed to be σ=570.9 *kJ/kg* [20].

As we can see, there is a very good agreement between the calculated and experimental data at heat flux densities of 92 *W/m*^{2} and 2000 *W/m*^{2}. At higher heat flux densities, the discrepancy between the calculated and experimental results is somewhat larger, but in general it can be argued that even such a simple mathematical model describes the experimental results well.

Figure 3 shows a comparison of the same experimental data with calculations, but in dimensionless form. A comparison of the graphs shows that the discrepancies are much more noticeable in the dimensionless form. For example, in Figure 3 we see two distinct outlier points that are also present in the previous graph, but their deviation is not as noticeable (these points correspond to minimum contact pressures).

Figure 2 shows calculated graphs and experimental values reflecting the dependence of the temperature difference between the surface of the heat exchanger and the sublimation temperature of the refrigerant on the contact pressure in the apparatus.

The dashed line on the graph shows the result of approximation of the available experimental data by the equation

$$\Omega = 0{,}4 \cdot \Theta^3. \qquad (18)$$

Good agreement between experimental results and calculated data, when only one coefficient is adjusted, indicates a successful choice of dimensionless criteria and the form of the criterion equation to describe the problem under consideration.

Journal of Chemistry and Technologies, 2024, *32*(2), 489-497

Figure 4 shows graphs comparing results of experiments with acetone for heat transfer enhancement and for "dry" heat transfer (without acetone). This graph also shows the calculated values for dry heat transfer and the results of approximating the heat transfer data with acetone using equation (17). As can be seen from the graphs, there is very good agreement between the experimental results and the results of the theoretical study. By selecting only one coefficient, it is possible to obtain an equation that generalizes all the experimental data obtained using acetone to enhance heat transfer. The given graphs clearly show the radical enhancement of heat transfer when acetone is used.

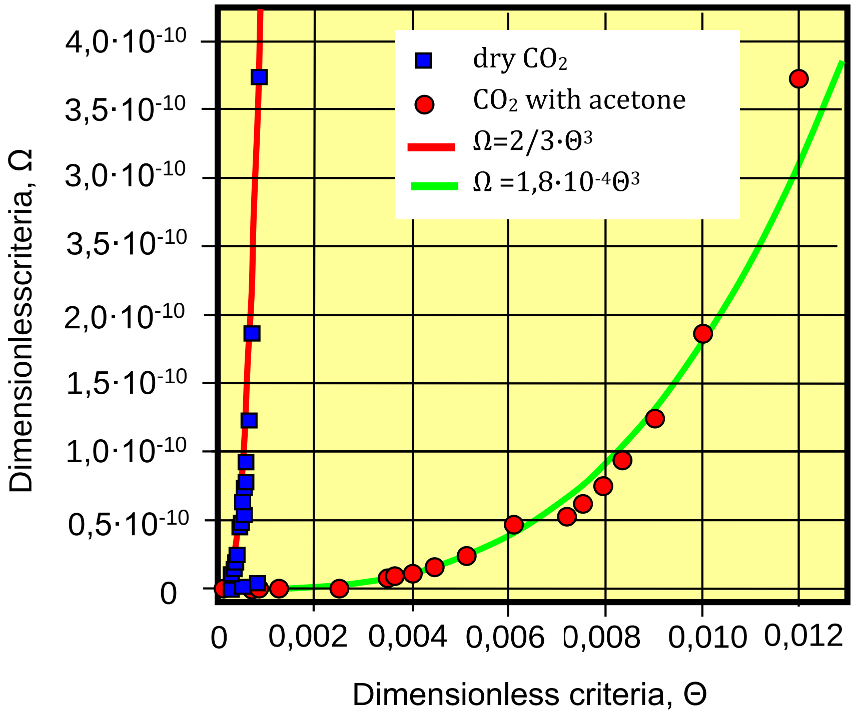

Fig. 4. Comparison of the results of experiments using acetone to intensify heat transfer, and for "dry" heat transfer (without acetone), calculated values for dry heat transfer and the results of approximation of heat transfer with acetone

The good agreement between the experimental and calculated data indicates that the chosen system of assumptions is quite acceptable under the heat transfer between the sublimating refrigerant and the solid surface of the heat exchenger. Particular, the theoretical conclusion is confirmed that on the surface of heat exchange with a melting or sublimating refrigerant, the heat flux density is automatically established to be identical over the entire heat transfer surface. This makes it possible to use the heat exchange during melting or sublimation of a substance to simulate boundary conditions of the second type *q = const*.

When the first type of boundary condition is modeled by heat exchange during boiling or condensation, an electric heater has usually been used to simulate the second type of boundary condition. However, the electric heater allows you to easily set the value of the total heat flux and generally does not guarantee equality of the heat flux density on the heat exchange surface. The use of heat exchange during sublimation or melting, on the other hand, guarantees equality of heat flux density over the entire heat exchange surface.

Conclusion

A simple system of dimensionless criteria and a criterion equation have been obtained. They allow to summarize with high accuracy experimental data on heat transfer during sublimation of refrigerant on the heat exchange surface.

The experimental data on heat transfer during sublimation of the refrigerant on the heat transfer surface in the presence of acetone, which intensifies the heat transfer, could be generalized and described by adjusting only one coefficient.

The heat transfer of the wall under boundary conditions of the second type *q = const* can be modeled using the heat transfer during melting or sublimation of the refrigerant.

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