

results of the development of a five-stage centrifugal distiller (CDS with THP) and its testing at the benches of the distiller manufacturer - Thermodistillation, Ukraine, and the Customer - Honeywell, USA. These results, as well as some methods for improving the performance of the CDS system, are given in [12, 13]. A more detailed review of works on a system with a centrifugal distiller is given in the first part of the paper [14], and a description of the research process - in [15]. The published results quite convincingly confirm the uniqueness of the system designed for the life support of manned spacecraft in long space missions and on the ISS.

Using the experimental data of a system with a centrifugal five-stage distiller and a thermoelectric heat pump (in terms of production, heat pump power, revolutions, degree of concentration, etc.), this paper substantiates the method of theoretical calculation of the temperature difference at the inlet to the THP $\Delta T_{in} = (T_{in\ hot\ THP} - T_{in\ cold\ THP})$ and the efficiency of THP (η_{THP}). An algorithm for calculating the temperature difference at the inlet to the THP is presented, the basis of which is the method for calculating the heat transfer coefficients in each stage of the distiller. The influence on the accuracy of calculating the temperature difference at the inlet to the THP and on the efficiency of the entire system of temperature depression and condensate supercooling in the cold circuit is shown. The calculation algorithm $\Delta T_{av} = (T_{in\ hot} + T_{out\ hot})/2 - (T_{in\ cold} + T_{out\ cold})/2$ in THP is analyzed.

Determination of the heat transfer coefficient in the distiller stage

Figure 1 shows a sketch of a rotating stage of a five-stage distiller. The stage consists of five parts: 1st central part, 2, 3 and 4 - disks at different angles with respect to the axis of rotation; Part 5 is a cylindrical ring. The total area of heat exchange is 0.07 m². There are five such steps in the apparatus.

The steam formed in the first stage, passing through the separation partitions, enters the condensation zone on the outer side of the disk, where, in contact with the heat exchange surface of the second stage, it condenses, transferring the heat of the initial liquid in the second stage. As a consequence, part of the liquid on the inner surface in the second stage evaporates, and the resulting steam condenses on the outside of the third stage and further, according to the same scheme, to the fifth stage. The steam in the last, fifth stage condenses on the film of technological distillate flowing down the surface of the rotating disk, which enters the apparatus from the cold circuit of THP.

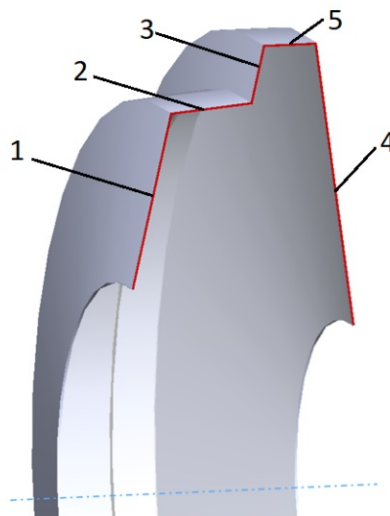


Fig. 1. Profile of heat exchange surface of one stage of a centrifugal distiller

The condensate formed in the condensation zones of each stage, under the action of centrifugal force, flows through the hydraulic shutters into the final condenser, where it is mixed with cooling technological

distillate.

After the system reaches steady-state operation, the concentration of the solution in all stages of the distiller, except the first, remains constant. The concentration of salts in the first stage will change during operation from the initial value to a certain limit, upon reaching which the power supply to the THP ceases.

The efficiency of the system depends on the flow of heat and mass transfer processes in the distiller, the scheme of movement of heat carriers, the efficiency of the battery, etc.

The total heat transfer coefficient of an individual stage is calculated by the formula:

$$k = 1 / (1/\alpha_c + 1/\alpha_e + \delta_w / \lambda_w), \quad (1)$$

where α_c is the coefficient of heat transfer during condensation of steam; α_e is the coefficient of heat transfer during evaporation of the liquid; δ_w and λ_w are the thickness and thermal conductivity of the heat exchange surface, respectively.

However, it is obvious that the value of the heat transfer coefficients depends on the nature of the liquid film flowing on a rotating surface.

Thus, to calculate the heat and mass transfer processes in the distiller requires knowledge of the hydrodynamic characteristics of the film and the coefficients of heat transfer during condensation and evaporation of the liquid on a rotating surface.

Heat exchange at condensation and evaporation in a rotating stage

The main hydrodynamic parameters of a liquid film flowing on a rotating surface are its average thickness and flow rate. A study of the flow of liquid film on a rotating surface was carried out in detail in [16]; as a result, it was possible to single out the characteristic modes of liquid film flow which affect the heat transfer process. The study of condensation on a rotating heat transfer surface was experimentally studied in [17 and 18]. In [19], a review is given of condensation on a rotating surface, and a method for calculating heat transfer is justified.

On surfaces 1–4 (see Fig. 1), when the distiller is operating, the evaporating liquid flows in the form of a thin film under the action of centrifugal acceleration $a \sim \omega^2 R \sin \beta$, where ω is the angular velocity, 1 / sec; R is the radius of the rotating unit of the stage, m; β is the angle between the axis of rotation of the rotor and the heat exchange surface. In the annular (part 5, Fig. 1), evaporation occurs under the influence of forced centrifugal convection.

Condensation on the external surface of a rotating stage

At sites close to disk geometry (sections 1-4, except for part 5), the local coefficient of heat transfer during condensation (α_c) can be calculated by the formula

$$Nu = \alpha_c / \lambda (v^2 / \omega^2 R \cos \beta)^{1/3} = 0.9 Re^{-1/3} \quad (2)$$

where ω are revolutions, 1 / sec; R is the radius of the rotating unit of the stage, m; β is the angle between the axis of rotation and the heat transfer surface. Re is the local Reynolds number, which is calculated as $Re = Gd / 2\pi R \mu$, where Gd is the amount of steam condensed at the local surface section.

The average value of the heat transfer coefficient α_c in each section of the stage is calculated taking into account the change in the area of the rotating surface from the radius of the disk.

Condensation of steam on a liquid film

In the annular condenser, steam condenses on a liquid film entering the condensation zone after cooling in the THP and an additional external heat exchanger [20].

The analysis of these data makes it possible to use the method proposed in [21] for the calculation of heat transfer for the case of condensation of steam on a liquid film under conditions of gravity. In this case, the heat transfer process is calculated by dependence

$$Nu = 0.018 Re^{0.5} Pr^{0.7} \quad (3)$$

In this equation, the average film temperature is taken as the determining factor, and Re is determined from the initial water flow in the film.

The maximum steam temperature in the last stage is 30 ° C and the coolant temperature is 20 ° C. The calculation, taking into account the actual size of the disk on which steam condenses in the final capacitor, shows that Re can have a value in the range 60... 120, and Pr = 3... 6. Theoretically, the solution of heat transfer in a given region of change in the numbers Re and Pr for the determination of Nu was performed in [22, 23]. These calculations, as shown in [24, 25], give a good coincidence with the experiment, which allows us to use them in the future.

Evaporation on the internal surface of a rotating stage (sections 1-4, Fig.1)

The heat transfer coefficient during the evaporation of a liquid film on a rotating disk was first measured in [18]. In [19], an analysis of studies by other authors was made and an improved method for calculating α_{evap} was described taking into account the effect of turbulence in the film flow.

According to [19] at $Re < 25$, the heat transfer coefficient must be calculated from the dependence:

$$Nu = 1.47 Re^{-1/3} \quad (4)$$

If $Re > 25$, then it will be more accurate to use the graphs obtained in [22, 23] and which take into account the turbulization of the liquid film and the deviation of the heat transfer process from the Nusselt laminar theory.

Evaporation of liquid in the annular channel (section 5, Fig. 1)

In the annular channel (part 5, Fig. 1) there is a heat exchange under the influence on the process of evaporation of free convection, which occurs during microgravity due to the presence of centrifugal acceleration. Experimental studies of evaporation during free convection in the presence of centrifugal acceleration were performed in the 70s of the 20th century in [26–28]. In [26], the heat transfer coefficients were measured for water evaporation at $\omega 2R/g$ from 1 to 5250 and a heat flux $q = (2 \cdot 104 \dots 2 \cdot 105) \text{ W / m}^2$. At low heat fluxes ($q < 2 \cdot 104 \text{ W/m}^2$) that occur in CDS, there is no influence of heat flux on heat transfer even at $\omega 2R/g > 102$ and the heat transfer coefficient can be determined from the dependence [27]:

$$Nu = 0.75(Gr \cdot Pr)^{0.25}. \quad (5)$$

where $Nu = \alpha \delta_u / \lambda$, $Gr = \omega^2 R \delta_u^4 / \nu^2$; δ_u is the thickness of liquid layer in the annular channel, m; R is radius on which the annular channel is located, m.

Calculation of the total temperature difference of steam in the distiller and the temperature difference at the inlet to THP

Tables 1 and 2 show the experimental data as a function of time τ : local temperatures, flow rate of the product (distillate) G_d , flow rate of the solution in the hot circuit, distillate flow in the cold circuit. Based on these data, the Reynolds number is calculated for the case of the flow of a liquid film on the inner and outer surfaces of the CDS working surface and, according to the corresponding formulas (1-5), heat

transfer coefficients for each part of the stage during evaporation and condensation (see Fig. 1).

Total heat transfer coefficient is calculated by formula (1).

Also, heat transfer coefficient is determined for each stage from the dependence:

$$k = Q_{cr}/F_{cr} \cdot \Delta T_{cr.}, \quad (6)$$

where F_{cr} is heat exchange surface, m^2 ; $\Delta T_{cr.}$ is temperature difference in the stage.

The total heat flow in the stage $Q_{st} = G \cdot r$, where G is the amount of steam condensed in the stage, r is phase transition heat of steam.

Knowing from the experiment the distillate production G_d of CDS, we determine $Q = \Sigma G_d \cdot r$ and then for water for the entire distiller $\Sigma \Delta T = (T_{in \text{ hot THP}} - T_{in \text{ cold THP}}) = Q/k \cdot F$

Following this, we compare the calculated $\Sigma \Delta T$ and $\Sigma \Delta T = (T_{in \text{ hot THP}} - T_{in \text{ cold THP}})$ from the experimental data (Table 1).

For the experiments with urine one should take into account the presence of physicochemical temperature depression $\Delta T_{depr.}$ Its value is determined from the plot $\Delta T_{depr.} = f(C)$, where urine concentration depends on the degree of water recovery. The concentration, in turn, is determined by the density of solution in the hot circuit (see Fig. 2).

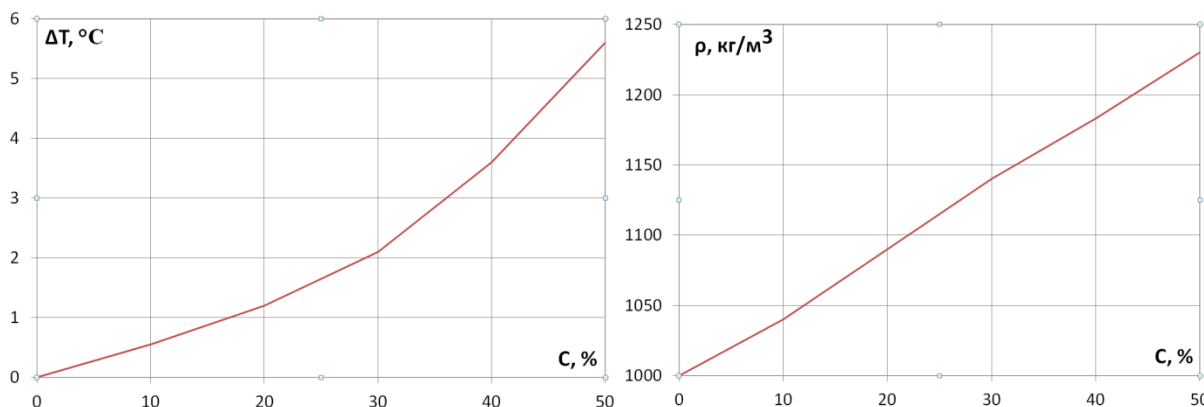


Fig.2 Dependence of physico-chemical temperature depression and solution density on the urine concentration (according to the research institute "KHIMMASH", Moscow)

Taking into account $\Delta T_{depr.}$, the temperature difference in the first stage $\Delta T_1 = T_{in} - \Delta T_{depr.1}$. Further, according to the experimental data, we determine the thermal power supplied to the hot circuit $Q_h = c_p \cdot G_h \cdot (T_{in \text{ THP}} - T_{out \text{ THP}})$. From the heat balance of THP one can determine the thermal power from the cold side $Q_c = Q_h - W_{THP}$.

Theoretical value of temperature difference in the cold stage

$$\Delta T_c = Q_c / c_p \cdot G_c. \quad (7)$$

Calculation of the dependence of THP conversion coefficient on the temperature difference of heat carriers

The requirement is obvious that the design parameters and the supply current of the THP meet the condition for the maximum heating coefficient K defined by the formula

$$K = \frac{Q_h}{W} = \frac{Q_h}{Q_h - Q_c}, \quad (1)$$

where $W = Q_h - Q_c$ is electric power consumption; Q_c , Q_h are external heat flows on the cold and hot

surfaces of thermoelectric module, respectively. This will provide for heating of heat carrier in the working circuit of heat pump with minimum power consumption.

To optimize thermoelectric modules, the theory of optimal control is successfully used [28 – 33]. This method is also easily generalized to calculate the maximum heating coefficient, which characterizes the heat pump mode.

According to optimal control methods, the operating efficiency of thermoelectric module in heating mode can be assessed by the functional

$$J = \ln \frac{Q_h}{Q_c} = \ln \frac{q_h}{q_c} = \ln q_h - \ln q_c, \quad (8)$$

where

$$q_h = \frac{Q_h}{n I}, \quad q_c = \frac{Q_c}{n I} \quad (9)$$

- are specific (related to current strength I) heat flows on the hot and cold thermocouple junctions, respectively, n is the number of thermocouples in a thermopile. The minimum of functional J corresponds to maximum value of heating coefficient K .

To calculate the densities of heat flows q_c, q_h , we use the system of equations of non-equilibrium thermodynamics, which for the n - and p -type thermoelement legs is of the form

$$\left. \begin{aligned} \frac{dT}{dx} &= -\frac{\alpha j}{\kappa} T - \frac{q}{\kappa} \\ \frac{dq}{dx} &= \frac{\alpha^2 j^2}{\kappa} T + \frac{\alpha j}{\kappa} q + \frac{j^2}{\sigma} \end{aligned} \right\}_{n,p}, \quad (10)$$

where $j = \frac{I}{S}$ is specific current density in the legs, S is cross-section of legs, I is the value of supply current. The Seebeck coefficient, the electrical conductivity and thermal conductivity of leg materials are functions of temperature: $n,p = n,p(T)$ and can be assigned on the basis of approximation of the experimental data of measuring characteristics of thermoelectric materials.

Solution of system (10) for boundary conditions

$$T_n(0) = T_p(0) = T_h, \quad T_n(l) = T_p(l) = T_c \quad (11)$$

will make it possible to calculate heat flows q_c, q_h using the relations

$$\begin{aligned} q_c &= -\sum_{n,p} [q(l) + j^2 r_0], \\ q_h &= -\sum_{n,p} [q(0) - j^2 r_0], \end{aligned} \quad (12)$$

where l is the height of thermoelement legs, r_0 is the value of contact resistance on thermoelement junctions.

From relations (12) it follows that q_c, q_h depend on the parameters of the current density in thermoelement legs j and on the magnitude of the contact resistance. In accordance with the optimal control theory, the values of j_n, j_p , which ensure the minimum of the functional J (8), must satisfy the following optimum conditions

$$-\frac{\partial J}{\partial j} + \int_0^l \frac{\partial H(\psi, T, q, j)}{\partial j} dx = 0, \quad (13)$$

where the Hamiltonian function H is given by

$$H = \sum_{n,p} (\psi_1 f_1 + \psi_2 f_2), \quad (14)$$

$(f_1, f_2)_{n,p}$ are right-hand sides of equations (4); $\psi = (\psi_1, \psi_2)_{n,p}$ is pulse vector the method of determination of which is described in [1, 2].

Relations (8) - (14) are the basis for computer design of optimal structures and calculation of optimal parameters of thermoelectric modules in heating mode for heat pumps.

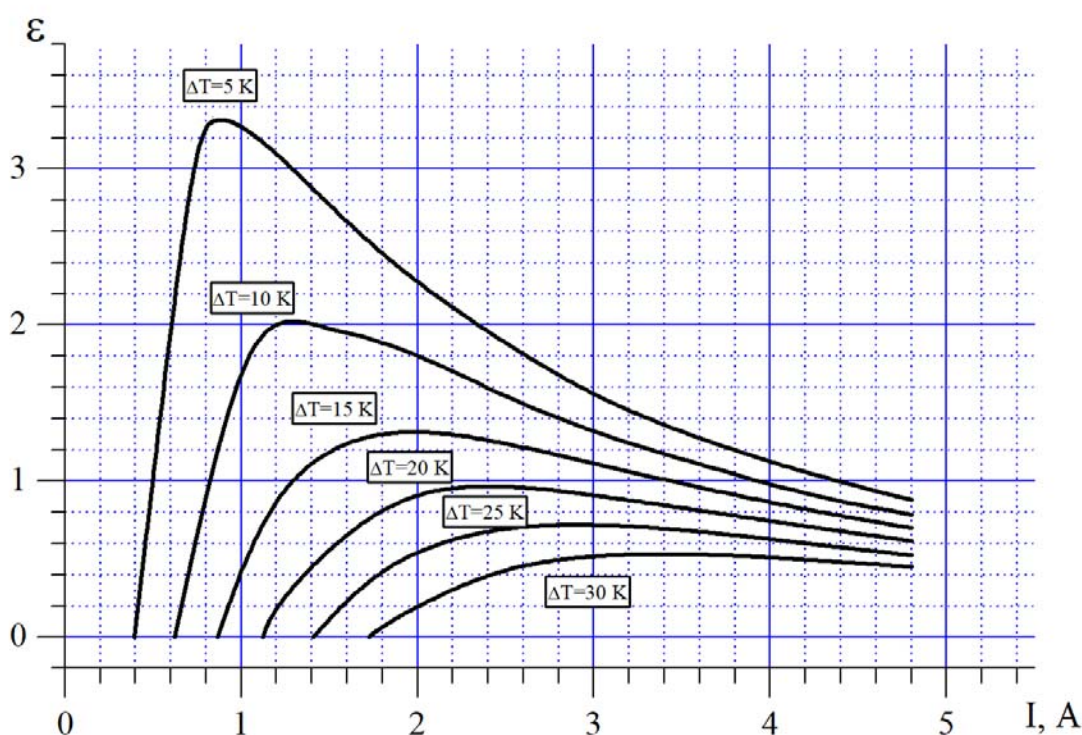


Fig.3 Dependence of the conversion coefficient of THP on supply current for different values of temperature difference between heat carriers

The algorithm for calculating the maximum heating coefficient is implemented numerically using computer modeling tools. In this case, the optimal current densities in the thermocouple legs j and their corresponding heat flows q_c , q_h are calculated, and the maximum value of the heating coefficient is determined.

Based on the above theory, the dependences of the THP conversion coefficient on the supply current were calculated for different values of the temperature difference between the heat carriers (Fig. 3).

As follows from Fig. 3, the conversion factor of THP strongly depends on the temperature difference between the heat carriers of the water distillation system. In addition, when designing a distiller, the THP mode of operation is important, namely, ensuring the optimal supply mode of thermoelectric power converters.

Results

Water evaporation

Table 1 shows the experimental data of studying the distiller operation on distilled water, the duration of operation was 50 minutes, the average power of the heat pump was 100 W, and the average production was 2.45 kg/h.

Table 1

Experimental data (water, n = 1100 rpm)

Time	Drive			THP	Weight		TDS	Flow		Product- ion	SPC	Temperature			
	<i>U</i>	<i>I</i>	<i>W</i>		<i>W</i>	In		Out	hot			cold	Hot in THP	Hot out THP	Cold in THP
min	V	A	W	W	Γ		mg/l	l/h		l/h	W-h/kg	°C			
10	20.6	3.3	68.0	101	224	152	5	90	88	2.18	77.5	25.5	29.6	22.6	19.5
30	20.6	3.3	68.0	99	1000	902	5	90	88	2.42	69.0	26.3	30.1	22.8	20.0
40	20.6	3.3	68.0	100	1438	1326	4	90	88	2.58	65.1	25.5	29.0	22.0	19.5
50	20.6	3.3	68.0	100	1838	1710	5	90	88	2.60	64.6	25.5	28.7	22.0	19.6
Average	20.6	3.3	68.0	100			5			2.45	69.1				
Total					1838	1710									

Here, production stands for distiller capacity, TDS is total number of dissolved solids.

In the calculation process, only the correction of the steam temperature in the last stage was performed:

1st step: according to the known inlet and outlet temperatures of the thermoelectric heat pump in the hot circuit and the flow rate, we determine the real thermal power, which is supplied in the hot circuit:

$$Q_h = c_p G_h (T_{in \text{ hot THP}} - T_{out \text{ hot THP}});$$

2nd step: from the heat pump balance we determine the thermal power on the cold side of the heat pump: $Q_c = Q_h - W_{THP}$;

3d step: we determine the calculated temperature difference in the cold circuit: $\Delta T_c = (T_{in \text{ cold THP}} - T_{out \text{ cold THP}}) = Q_c / c_p \cdot G_c$.

As a result of the calculation, the obtained deviation of ΔT_c from the experimental data does not exceed 0.1 °C.

In some individual cases, the theoretical calculation of the total temperature difference at the distiller $\Sigma \Delta T$ and $\Delta T_{in} = (T_{in \text{ hot THP}} - T_{in \text{ cold THP}})$ shows that there is a deviation of the value ΔT_{in} , which is several degrees higher than the calculated one. This was due to supercooling of the distillate in the cold circuit of the heat pump, which led to a significant increase in the total temperature difference $\Sigma \Delta T$ on the heat pump and, as a result, to a decrease in its efficiency. An additional experiment, where the degree of condensate cooling in the final condenser was specially controlled, showed that it is possible to significantly reduce the temperature difference at the inlet of the heat pump from the cold side in the calculation and in the experiment. At the same time, the efficiency of the heat pump η grows, which becomes the same as in the

theoretical calculation. This, in some cases, gives a reduction in specific energy consumption of SPC by 10-15 %.

Urine evaporation

Table 2 shows the experimental data of studying the distiller operation on urine, the duration of operation was 107 minutes, the average power of the heat pump was 203 W, and the average production was 3.35 kg / h.

Table 2

Experimental data (urine, n = 1100 rpm)

Time	Drive			THP	Weight		TDS	Flow		Production	SPC	Temperature				
	<i>U</i>	<i>I</i>	<i>W</i>		<i>W</i>	in		Out	hot			Cold	Hot in THP	Hot out THP	Cold in THP	Cold out THP
min	V	A	W	W	G		mg/l	l/h		kg/h	W-h/kg	°C				
5	20.8	3.35	69.7	202	-	-	-	-	-	-	-	-	-	-	-	-
13	20.8	3.35	69.7	202	680	710	37	89	90	3.50	77.6	33.1	38.2	25.5	22.5	
25	20.8	3.40	70.7	202	1366	1416	60	89	89	353	77.3	32.9	37.9	25.1	22.0	
47	20.8	3.43	71.3	201	2024	2098	79	90	89	3.41	79.9	33.4	38.5	25.4	22.3	
59	20.8	3.43	71.3	199	2674	2770	84	89	89	3.36	80.4	34.3	39.6	25.8	22.8	
71	20.9	3.45	72.1	211	3324	3444	90	89	89	3.37	84.0	34.9	40.2	26.0	23.1	
83	20.9	3.47	72.5	205	3360	4106	94	90	89	3.31	83.8	35.6	41.1	26.4	23.6	
95	20.9	3.50	73.1	205	4572	4734	96	89	89	3.14	88.6	36.6	42.0	27.0	24.2	
107	20.9	3.50	73.1	201	5170	5362	97	89	89	3.14	87.2	36.5	41.7	26.0	23.4	
Average	20.8	3.44	71.7	203						3.35	115.1					
Total					5170	5362										

Table 2 shows that, over time, the temperature difference on the heat pump increases and, as a consequence, the system production decreases and the specific power consumption increases.

For the case of urine processing, when calculating the total temperature difference on a thermoelectric heat pump $\Sigma\Delta T = (T_{\text{in hot THP}} - T_{\text{in cold THP}})$, it is also necessary to take into account the effect of temperature depression, which increases with increasing urine concentration, i.e. with increasing operating time of the distiller.

During the experiments, the temperature of the water supplied to the additional heat exchanger to cool the cold circuit distillate of the heat pump was not monitored separately.

As a result, the main indicators of the system (production and power consumption) when working on a distillate, all other things being equal, are significantly better than when processing urine.

Conclusions

1. This paper shows that a multistage centrifugal distiller is a complex structure in which the individual stages of the distiller have a heat exchange surface assembled from elements whose angle of inclination with respect to the centrifugal acceleration vector varies from 00 to 900.
2. To perform an adequate thermal calculation of such an apparatus, dependences were selected and

substantiated for calculating heat transfer during evaporation and condensation, as well as the heat transfer coefficient on the individual components of the heat transfer surface.

3. The effect of physicochemical temperature depression and supercooling of a technical distillate in a cold circuit in a thermoelectric heat pump on the energy efficiency of a distillation system is shown.
4. The experimental and theoretical results obtained in this paper and in the previous parts [14-15] will be used to create a mathematical model of the water regeneration (distillation) system. A mathematical model is necessary to optimize the operating and geometric parameters of the system as applied to specific operating conditions.

References

1. Rifert V., Barabash P., Goliad N. (1990). Methods and processes of thermal distillation of water solutions for closed water supply systems. *SAE Paper 901249, 20th Intersociety Conference on Environmental Systems (Williamsburg, July 1990)*.
2. Samsonov N, Bobe L., Novikov V., Rifert V., et al. (1994). Systems for water reclamation from humidity condensate and urine for space station. *SAE Paper 941536, 24th International society Conference on Environmental Systems (June, 1994)*.
3. Samsonov N.M., Bobe L.S., Novikov V., Rifert V.G., Barabash P.A, et al.. (1995). Development of urine processor distillation hardware for space stations. (1995). *SAE Paper 951605, 25th International Conference on Environmental Systems (San Diego, July 1995)*.
4. Samsonov N.M., Bobe L.S, Novikov V., Rifert V.G., et al.(1997). Updated systems for water recovery from humidity condensate and urine for the International space station. *SAE Paper 972559, 27th International Conference on Environmental Systems (Nevada, July 1997)*.
5. Samsonov N.M., Bobe L.S, Novikov V., Rifert V.G., et al. (1999). Development and testing of a vacuum distillation subsystem for water reclamation from urine. *SAE Paper 1999-01-1993, 29th International Conference on Environmental Systems, 1999*.
6. Rifert V., Usenko V., Zolotukhin I., MacKnight A., Lubman A. (1999). Comparison of secondary water processors using distillation for space applications. *SAE Paper 99-70466, 29th International Conference on Environmental Systems (Denver, July 1999)*.
7. Rifert V., Strikun A., Usenko V. (2000). Study of dynamic and extreme performances of multistage centrifugal distiller with the thermoelectric heat pump. *SAE Technical Papers 2000. 30th International Conference on Environmental Systems (Toulouse; France; 10-13 July 2000)*.
8. Rifert V., Usenko V., Zolotukhin I., MacKnight A. and Lubman A. (2001). Design optimisation of cascade rotary distiller with the heat pump for water reclamation from urine. *SAE Paper 2001-01-2248, 31st International Conference on Environmental Systems (Orlando, July 2001)*.
9. Rifert V. G., Usenko V.I., Zolotukhin I.V., MacKnight A.and Lubman A. (2003). Cascaded distillation technology for water processing in space. *SAE Paper 2003-01-2625. 34st International Conference on Environmental Systems (Orlando, July 2003)*.
10. Lubman A., MacKnight A., Rifert V., Zolotukhin I. and Pickering K. (2006). Wastewater processing cascade distillation subsystem. design and evaluation. *SAE International, 2006-01-2273. July 2006*.
11. Lubman A., MacKnight A., Rifert V., and Barabash P. (2007). Cascade distillation subsystem hardware development for verification testing. *SAE International, 2007-01-3177, July 2007*.
12. Rifert V.G., Anatyshuk L.I., Barabash P.A., Usenko V.I., Strikun A.P., Prybyla A.V. (2017). Improvement of the distillation methods by using centrifugal forces for water recovery in space flight applications. *J. Thermoelectricity*, 1, 71-83.
13. Rifert Vladimir G., Barabash Petr A., Usenko Vladimir, Solomakha Andrii S., Anatyshuk Lukyan I., Prybyla.A.V. (2017). Improvement the cascade distillation system for long-term space flights. *68th*

International Astronautical Congress (IAC) (Adelaide, Australia, 25-29 September 2017). IAC-17-A1.IP.25.

14. Rifert V.G., Anatyshuk L.I., Barabash P.O., Usenko V.I., Strikun A.P., Solomakha A.S., Petrenko V.G., Prybyla A.V. (2019). Evolution of centrifugal distillation system with a thermoelectric heat pump for space missions. Part 1. Review of publications on centrifugal distillation in the period of 1990 – 2017. *J. Thermoelectricity*, 1, **crop.**
15. Rifert V.G., Anatyshuk L.I., Barabash P.O., Usenko V.I., Strikun A.P., Solomakha A.S., Petrenko V.G., Prybyla A.V. (2019). Evolution of centrifugal distillation system with a thermoelectric heat pump for space missions. Part 2. . Study of the variable characteristics of a multi-stage distillation system with a thermoelectric heat pump. *J. Thermoelectricity*, 2.
16. Muzhilko A.A., Rifert V.G., Barabash P.A. (1985) Flow of liquid film over the surface of a rotating disk. *Heat transfer. Soviet research.*
17. Butuzov A.I., Rifert V.G. (1972). An experimental study of heat transfer during condensation of steam at a rotating disk. *Heat Transfer-Soviet Research*, 4(6).
18. Butuzov A.I. and Rifert V.G. (1973). Heat transfer in evaporation of liquid from a film on a rotating disk. *Heat Transfer-Soviet Research*, 5(1).
19. Rifert V.G., Barabash P.A., Solomakha A.S., Usenko V., Sereda V.V., Petrenko V.G.(2018). Hydrodynamics and heat transfer in centrifugal film evaporator. *Bulgarian Chemical Communications*, 50, *Special Issue K*, 49-57.
20. Rifert V.G., Barabash P.A., Goliyad N.N. Condensation of steam on a water film falling down a rotating surface. *Heat transfer. Soviet research*, 16 (3), 132-137.
21. Gimbutis G. (1988). *Heat exchange with gravitational flow of liquid film*. Monograph. Vilnius.
22. Bae S., Maulbetsch J.S., Rohsenow W.M. (1969). *Refrigerant forced-convection condensation inside horizontal tubes*. Report No. DSR-79760-64. Massachusetts Institute of Technology (Cambridge, MA).
23. Travis D.P., Baron A.N., Rohsenow W.M.(1971). Forced-convection condensation inside tubes. Report No. DSR-72591-74. Massachusetts Institute of Technology.(Cambridge, MA)
24. Rifert V.G., Sereda V.V., Gorin V.V., Barabash P.A., Solomakha A.S. (2018). Restoration of correctness and improvement of a model for film condensation inside tubes. *Bulgarian Chemical Communications*, 50, *Special Issue K*, 58-69.
25. Rifert V., Sereda V., Solomakha A. (2019). Heat transfer during film condensation inside plain tubes. Review of theoretical research. *Heat and Mass Transfer/Waerme- und Stoffuebertragung*, 2019.
26. Usenko V.I., Fainzilberg S.N. (1974). Effect of acceleration on the critical heat load with the boiling of freons on elements having small transverse dimensions. *High Temperature*.
27. Usenko V.I., Podgoretskiy V.M., Kudelya P.P. (1985). Nucleate boiling heat transfer from freon-11 and 12 at very high «g». *Heat transfer. Soviet research*.
28. Anatyshuk L.I., Prybyla A.V. (2015). Optimization of thermal connections in thermoelectric liquid-liquid heat pumps for water purification systems of space application. *J. Thermoelectricity*, 4, 45 – 51.
29. Anatyshuk L.I., Prybyla A.V. (2015). Optimization of power supply system of thermoelectric liquid-liquid heat pump. *J. Thermoelectricity*, 6, 53 – 58.
30. Anatyshuk L.I., Rozver Yu.Yu., Prybyla A.V. (2017). Experimental study of thermoelectric liquid-liquid heat pump. *J. Thermoelectricity*, 3, C. 33 – 39.
31. Anatyshuk L.I., Prybyla A.V. (2017). Limiting possibilities of thermoelectric liquid-liquid heat pump. *J. Thermoelectricity*, 4, 33 – 39.
32. Anatyshuk L.I., Prybyla A.V. (2017). The influence of quality of heat exchangers on the properties of thermoelectric liquid-liquid heat pumps. *J. Thermoelectricity*, 5, 33 – 39.

33. Anatyshuk L.I., Prybyla A.V. (2017). On the coefficient of performance of thermoelectric liquid-liquid heat pumps with regard to energy loss for heat carrier transfer. *J. Thermoelectricity*, 6, 33 – 39.

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Риферт В.Г., док. техн. наук¹
Анатичук Л.І., акад. НАН України²
Барабаш П.О., канд. техн. наук¹
Усенко В.І., док. техн. наук¹
Стрикун А.П.¹
Соломаха А.С., канд. техн. наук^{1,2}
Петренко В.Г., канд. техн. наук¹
Середа В.В. канд. техн. наук¹
Прибула А.В., канд. фіз.– мат. наук¹

¹НТУ «КПІ», вул. Політехнічна, 6, Київ, 03056, Україна;

²Інститут термоелектрики, вул. Науки, 1,

Чернівці, 58029, Україна; e-mail: anatyshuk@gmail.com;

³Чернівецький національний університет

ім. Юрія Федьковича, вул. Коцюбинського 2,

Чернівці, 58000, Україна

ЕВОЛЮЦІЯ СИСТЕМИ ВІДЦЕНТРОВОЇ ДИСТИЛЯЦІЇ З ТЕРМОЕЛЕКТРИЧНИМ ТЕПЛОВИМ НАСОСОМ ДЛЯ КОСМІЧНИХ МІСІЙ

Частина 3. Аналіз локальних і інтегральних характеристик системи відцентрової дистиляції з термоелектричним тепловим насосом

У роботі описана методика розрахунків коефіцієнта теплопередачі й ефективності відцентрового дистилятора, а також визначення ефективності термоелектричного теплового насоса. Для цього використані результати теоретичних і експериментальних досліджень процесів конденсації й випару на обертовій поверхні й дослідження процесів у рідинно-рідинному термоелектричному тепловому насосі (ТНР). Зроблене порівняння розрахункових значень середнього температурного перепаду ΔT_{cp} у ТНР і різниці температур на вході в ТНР ΔT_{in} з експериментальними даними, отриманими при випробуванні відцентрового дистилятора на воді й урині. Бібл. 33, рис. 3, табл. 2.

Ключові слова: термоелектрика, тепловий насос, дистилятор

Риферт В.Г., док. техн. наук¹
Анатычук Л.И., акад. НАН Украины^{2,3}
Барабаш П.О. канд. техн. наук¹
Усенко В.И. док. техн. наук¹
Стрикун А.П.¹
Соломаха А. С., канд. техн. наук^{1,2}
Петренко В. Г., канд. техн. наук¹
Середа В.В. канд. техн. наук¹
Прибила А. В. канд. физ.-мат. наук^{2,3}

¹НТУ «КПИ», вул. Политехническая, 6,
Киев, 03056, Украина;

²Институт термоэлектричества НАН и МОН Украины,
ул. Науки, 1, Черновцы, 58029, Украина,
e-mail: anatyuch@gmail.com

³Черновицкий национальный университет
им. Юрия Федьковича, ул. Коцюбинского, 2,
Черновцы, 58012, Украина

ЭВОЛЮЦИЯ СИСТЕМЫ ЦЕНТРОБЕЖНОЙ ДИСТИЛЛЯЦИИ С ТЕРМОЭЛЕКТРИЧЕСКИМ ТЕПЛОВЫМ НАСОСОМ ДЛЯ КОСМИЧЕСКИХ МИССИЙ

Часть 3. Анализ локальных и интегральных характеристик системы центробежной
дистилляции с термоэлектрическим тепловым насосом

В работе описана методика расчета коэффициента теплопередачи и эффективности центробежного дистиллятора, а также определение эффективности термоэлектрического теплового насоса. Для этого использованы результаты теоретических и экспериментальных исследований процессов конденсации и испарения на вращающейся поверхности и исследование процессов в жидкостно-жидкостном термоэлектрическом тепловом насосе (ТНР). Сделано сравнение расчетных значений среднего температурного перепада ΔT_{cp} в ТНР и разницы температур на входе в ТНР ΔT_{in} с экспериментальными данными, полученными при испытании центробежного дистиллятора на воде и урине. Библ. 33, рис. 3, табл. 2.

Ключевые слова: термоэлектричество, тепловой насос, дистиллятор.

References

1. Rifert V., Barabash P., Goliad N. (1990). Methods and processes of thermal distillation of water solutions for closed water supply systems. *SAE Paper 901249, 20th Intersociety Conference on Environmental Systems (Williamsburg, July 1990).*
2. Samsonov N, Bobe L., Novikov V., Rifert V., et al. (1994). Systems for water reclamation from humidity condensate and urine for space station. *SAE Paper 941536, 24th International society*

- Conference on Environmental Systems (June, 1994).*
3. Samsonov N.M., Bobe L.S., Novikov V., Rifert V.G., Barabash P.A, et al. (1995). Development of urine processor distillation hardware for space stations. (1995). *SAE Paper 951605, 25th International Conference on Environmental Systems (San Diego, July 1995).*
 4. Samsonov N.M., Bobe L.S, Novikov V., Rifert V.G., et al.(1997). Updated systems for water recovery from humidity condensate and urine for the International space station. *SAE Paper 972559, 27th International Conference on Environmental Systems (Nevada, July 1997).*
 5. Samsonov N.M., Bobe L.S, Novikov V., Rifert V.G., et al. (1999). Development and testing of a vacuum distillation subsystem for water reclamation from urine. *SAE Paper 1999-01-1993, 29th International Conference on Environmental Systems, 1999.*
 6. Rifert V., Usenko V., Zolotukhin I., MacKnight A., Lubman A. (1999). Comparison of secondary water processors using distillation for space applications. *SAE Paper 99-70466, 29th International Conference on Environmental Systems (Denver, July 1999).*
 7. Rifert V., Strikun A., Usenko V. (2000). Study of dynamic and extreme performances of multistage centrifugal distiller with the thermoelectric heat pump. *SAE Technical Papers 2000. 30th International Conference on Environmental Systems (Toulouse; France; 10-13 July 2000).*
 8. Rifert V., Usenko V., Zolotukhin I., MacKnight A. and Lubman A. (2001). Design optimisation of cascade rotary distiller with the heat pump for water reclamation from urine. *SAE Paper 2001-01-2248, 31st International Conference on Environmental Systems (Orlando, July 2001).*
 9. Rifert V. G., Usenko V.I., Zolotukhin I.V., MacKnight A.and Lubman A. (2003). Cascaded distillation technology for water processing in space. *SAE Paper 2003-01-2625. 34th International Conference on Environmental Systems (Orlando, July 2003).*
 10. Lubman A., MacKnight A., Rifert V., Zolotukhin I. and Pickering K. (2006). Wastewater processing cascade distillation subsystem. design and evaluation. *SAE International, 2006-01-2273. July 2006.*
 11. Lubman A., MacKnight A., Rifert V., and Barabash P. (2007). Cascade distillation subsystem hardware development for verification testing. *SAE International, 2007-01-3177, July 2007.*
 12. Rifert V.G., Anatyshuk L.I., Barabash P.A., Usenko V.I., Strikun A.P., Prybyla A.V. (2017). Improvement of the distillation methods by using centrifugal forces for water recovery in space flight applications. *J. Thermoelectricity, 1, 71-83.*
 13. Rifert Vladimir G., Barabash Petr A., Usenko Vladimir, Solomakha Andrii S., Anatyshuk Lukyan I., Prybyla.A.V. (2017). Improvement the cascade distillation system for long-term space flights. *68th International Astronautical Congress (IAC) (Adelaide, Australia, 25-29 September 2017). IAC-17-AI.IP.25.*
 14. Rifert V.G., Anatyshuk L.I., Barabash P.O., Usenko V.I., Strikun A.P, Solomakha A.S., Petrenko V.G., Prybyla A.V. (2019). Evolution of centrifugal distillation system with a thermoelectric heat pump for space missions. Part 1. Review of publications on centrifugal distillation in the period of 1990 – 2017. *J. Thermoelectricity, 1.*
 15. Rifert V.G., Anatyshuk L.I., Barabash P.O., Usenko V.I., Strikun A.P, Solomakha A.S., Petrenko V.G., Prybyla A.V. (2019). Evolution of centrifugal distillation system with a thermoelectric heat pump for space missions. Part 2. . Study of the variable characteristics of a multi-stage distillation system with a thermoelectric heat pump. *J. Thermoelectricity, 2.*
 16. Muzhilko A.A., Rifert V.G., Barabash P.A. (1985) Flow of liquid film over the surface of a rotating disk. *Heat transfer. Soviet research.*
 17. Butuzov A.I., Rifert V.G. (1972). An experimental study of heat transfer during condensation of steam at a rotating disk. *Heat Transfer-Soviet Research, 4(6).*
 18. Butuzov A.I. and Rifert V.G. (1973). Heat transfer in evaporation of liquid from a film on a rotating disk. *Heat Transfer-Soviet Research, 5(1).*

19. Rifert V.G., Barabash P.A., Solomakha A.S., Usenko V., Sereda V.V., Petrenko V.G.(2018). Hydrodynamics and heat transfer in centrifugal film evaporator. *Bulgarian Chemical Communications*, 50, Special Issue K, 49-57.
20. Rifert V.G., Barabash P.A., Goliyad N.N. Condensation of steam on a water film falling down a rotating surface. *Heat transfer. Soviet research*, 16 (3), 132-137.
21. Gimbutis G. (1988). *Heat exchange with gravitational flow of liquid film*. Monograph. Vilnus.
22. Bae S., Maulbetsch J.S., Rohsenow W.M. (1969). *Refrigerant forced-convection condensation inside horizontal tubes*. Report No. DSR-79760-64. Massachusetts Institute of Technology (Cambridge, MA).
23. Traviss D.P., Baron A.N., Rohsenow W.M.(1971). Forced-convection condensation inside tubes. Report No. DSR-72591-74. Massachusetts Institute of Technology.(Cambridge, MA)
24. Rifert V.G., Sereda V.V., Gorin V.V., Barabash P.A., Solomakha A.S. (2018). Restoration of correctness and improvement of a model for film condensation inside tubes. *Bulgarian Chemical Communications*, 50, Special Issue K, 58-69.
25. Rifert V., Sereda V., Solomakha A. (2019). Heat transfer during film condensation inside plain tubes. Review of theoretical research. *Heat and Mass Transfer/Waerme- und Stoffuebertragung*, 2019.
26. Usenko V.I., Fainzilberg S.N. (1974). Effect of acceleration on the critical heat load with the boiling of freons on elements having small transverse dimensions. *High Temperature*.
27. Usenko V.I., Podgoretskiy V.M., Kudelya P.P. (1985). Nucleate boiling heat transfer from freon-11 and 12 at very high «g». *Heat transfer. Soviet research*.
28. Anatyshuk L.I., Prybyla A.V. (2015). Optimization of thermal connections in thermoelectric liquid-liquid heat pumps for water purification systems of space application. *J.Thermoelectricity*, 4, 45 – 51.
29. Anatyshuk L.I., Prybyla A.V. (2015). Optimization of power supply system of thermoelectric liquid-liquid heat pump. *J.Thermoelectricity*, 6, 53 – 58.
30. Anatyshuk L.I., Rozver Yu.Yu., Prybyla A.V. (2017). Experimental study of thermoelectric liquid-liquid heat pump. *J.Thermoelectricity*, 3, C. 33 – 39.
31. Anatyshuk L.I., Prybyla A.V. (2017). Limiting possibilities of thermoelectric liquid-liquid heat pump. *J.Thermoelectricity*, 4, 33 – 39.
32. Anatyshuk L.I., Prybyla A.V. (2017). The influence of quality of heat exchangers on the properties of thermoelectric liquid-liquid heat pumps. *J.Thermoelectricity*, 5, 33 – 39.
33. Anatyshuk L.I., Prybyla A.V. (2017). On the coefficient of performance of thermoelectric liquid-liquid heat pumps with regard to energy loss for heat carrier transfer. *J.Thermoelectricity*, 6, 33 – 39.

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P.V. Gorskiy

P.V. Gorskiy *doc. phys.– math. science*^{1,2},
Mytskaniuk N.V.^{1,2}



Mytskaniuk N.V.

¹Institute of Thermoelectricity of the NAS and MES of
Ukraine, 1, Nauky str, Chernivtsi, 58029, Ukraine;

e-mail: anatysh@gmail.com

²Yu.Fedkovych Chernivtsi National University,
2, Kotsiubynskiy str., Chernivtsi, 58000, Ukraine

**ON THE TEMPERATURE DEPENDENCES OF THERMOELECTRIC
CHARACTERISTICS OF BISMUTH TELLURIDE-METAL TRANSIENT LAYER
WITH DUE REGARD FOR PERCOLATION EFFECT**

The basic relationships are obtained by calculation, which determine the temperature dependences of thermoelectric characteristics of thermoelectric material-metal transient contact layers with due regard for percolation theory. Specific quantitative results and plots of the temperature dependences of the electrical and thermal contact resistances, the thermoEMF, the power factor, and the dimensionless thermoelectric figure of merit are given for bismuth telluride – nickel contact pair. It has been established that in the temperature range of 200-400 K on retention of uneven distribution of metal particles in transient layer and its thickness in the range of 20-150 μm , the electrical contact resistance varies from $7 \cdot 10^{-7}$ to $1.9 \cdot 10^{-5}$ $\text{Ohm} \cdot \text{cm}^2$, the thermal contact resistance – from 0.052 to 0.98 $\text{K} \cdot \text{cm}^2/\text{W}$, the thermoEMF – from 155 to 235 $\mu\text{V}/\text{K}$, the power factor – from $4.2 \cdot 10^{-5}$ to $6.8 \cdot 10^{-5}$ $\text{W}/(\text{m} \cdot \text{K}^2)$, the dimensionless thermoelectric figure of merit – from 0.35 to 1.08. After levelling the concentration, the electrical contact resistance decreases by a factor of 1.12 – 3.6, the thermal contact resistance decreases by a factor of 1.15 – 2.08, the thermoEMF is practically unvaried, the power factor increases by a factor of 1.19 – 2.79, the dimensionless thermoelectric figure of merit increases maximum 1.2 times. Bibl. 14, Fig. 22.

Key words: thermoelectric material-metal contact, near-contact transient layer, electrical contact resistance, thermal contact resistance, thermoEMF, percolation theory.

Introduction

The efficiency of thermoelectric modules of sufficiently large sizes is mainly determined by the figure of merit of thermoelectric materials of thermoelement legs. However, as the size of thermoelectric legs decreases in the direction of temperature gradient, this efficiency becomes increasingly dependent on the electrical and thermal contact resistances of thermoelectric material – metal layer, in particular, due to the Joule heat release on the contact resistances. These resistances should be essentially lower than those of thermoelectric legs [1-3]. This fact put a limit on the miniaturization of thermoelectric power converters – generators and coolers.

Indeed, let, for instance, r_{ce} – specific, that is related to unit contact area, electrical contact resistance. Then, if ρ_s – specific electrical resistance of TEM, and l_s – the length of thermoelectric leg in the direction of temperature gradient, then the condition of small influence of contact resistance on the efficiency of thermoelectric power converter is given by:

$$l_s \gg r_{ce} / \rho_s, \quad (1)$$