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Assessment of the thermal accumulating solar stations efficiency

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Abstract. The impact of heat-generating facilities on the environmental condition of adjacent territories is assessed. The possibility of using solar energy to reduce the anthropogenic impact on the environment is substantiated. It is proposed to create cascade solar installations to provide hot water supply to residential buildings and industrial facilities. A scheme of a solar power station with a seasonal solar energy accumulator based on depleted quarries to ensure year-round supply of hot water is proposed. A mathematical model is proposed for calculating the efficiency of solar collectors with transparent and rear insulation, which make up the third stage of a three-stage solar DHW system. The relative error of mathematical models in comparison with test data is presented. The average monthly efficiency of the third cascade of calculated mathematical models for climatic conditions of the Crimean peninsula is given. A refined algorithm for calculating the efficiency of solar collectors of the third stage has been developed.

1 Introduction

The use of unconventional energy sources in recent decades has received significant development due to the use of new technologies and materials. In many countries, installations are being created to receive energy from the sun, wind, and ocean tides. At the same time, the main role in choosing an energy source is played by the climatic features of the region in which the energy conversion station is being created.

2 Materials and Methods

The Crimean peninsula is a unique resort area, where a large variety of flora and fauna is concentrated, and its coastal zone includes a significant number of health complexes. All this attracts tourists from all over the country to Crimea, as well as the active development of tourist infrastructure. With this in mind, it is necessary to preserve the ecosystem of the region and minimize its pollution as a result of anthropogenic impact, including during the operation of heat-generating facilities that supply residential buildings and enterprises with heat and

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hot water [1-3]. In the process of hydrocarbon fuel combustion used in boiler houses, the environmental condition of adjacent territories is influenced by aeropollutants and terrapollutants (Figure 1) [4]. With an increase in the population, there is a need to increase the capacity of heat generating facilities, which in turn leads to an increase in fuel consumption and, as a consequence, to an increase in the amount of solid impurities, toxic and carcinogenic substances emitted into the atmosphere.

The solution to this problem is to replace traditional heat generation sources with alternative ones. The Crimean Peninsula, due to its geographical location and climatic features (Table 1), is one of the promising areas for the use of solar radiation, which is an inexhaustible source of environmentally friendly energy [5,6].

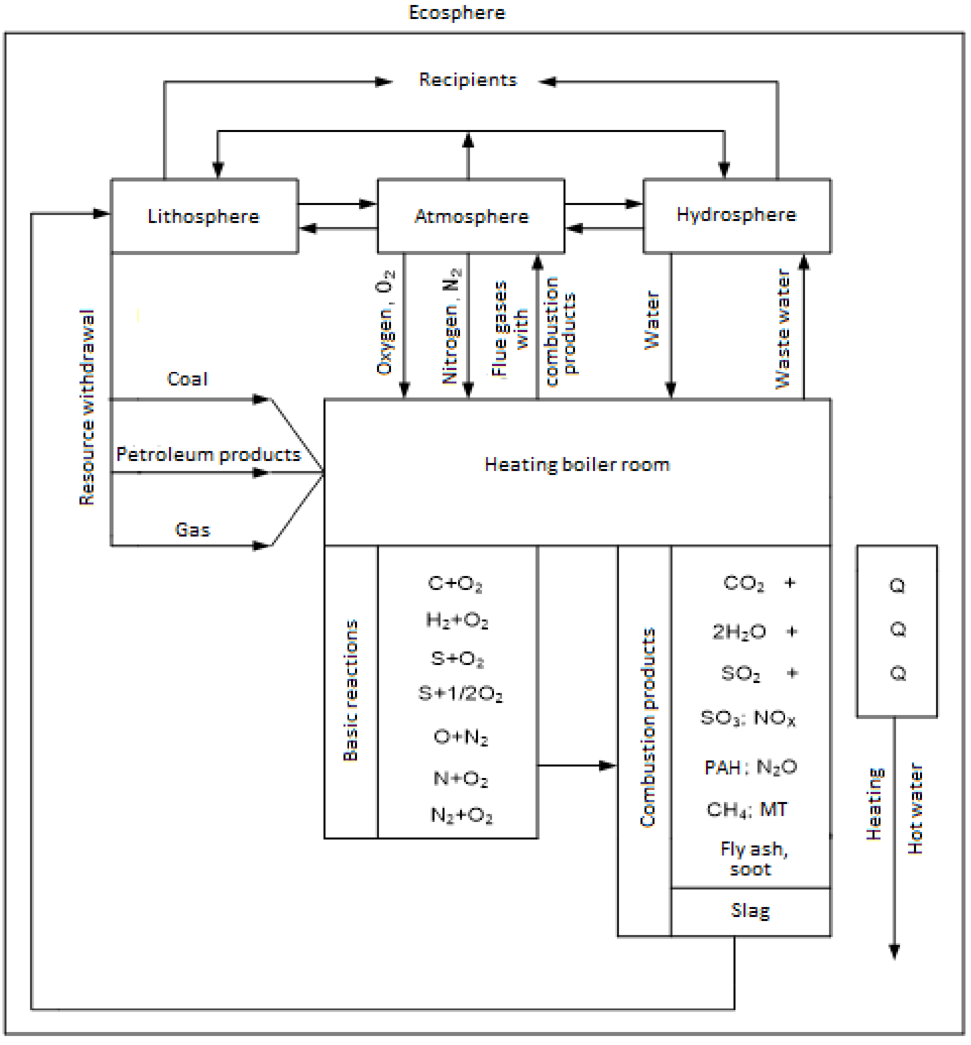

Fig. 1. The scheme of relations of the technological heat generation process in traditional heat supply with the biosphere components.

In [7], a scheme of a solar station with a thermal accumulator (Figure 2) according to the solar (salt) pond type created on the basis of a depleted quarry is proposed. This system allows not only to convert solar energy into thermal energy, but also to accumulate it for use during those periods of the year when the received solar radiation is not enough to generate the necessary amount of heat. The use of depleted quarries as a thermal accumulator also makes it possible to solve the issue of their reclamation [8].

Table 1. Duration of the day (light part of the day) in Crimea.

Month	Number of days	Number of hours	Possible number of sunshine hours		Actual number of sunshine hours		Average length of the day	Midday height of the sun on the 15th of the month
			Hours	% of total hours	Hours	% of total hours		
January	31	744	286	38.5	56	22	9 h 14 min	23°53'
February	28	672	291	43.3	74	29	10 h 23 min	32°20'
March	31	744	369	49.6	132	41	11 h 54 min	42°52'
April	30	720	405	56.2	194	52	13 h 30 min	54°46'
May	31	744	460	61.8	275	64	14 h 49 min	63°52'
June	30	720	467	64.8	308	69	15 h 34 min	68°19'
July	31	744	474	63.7	348	78	15 h 17 min	66°32'
August	31	744	436	58.6	327	80	14 h 04 min	59°03'
September	30	720	370	51.4	244	72	12 h 20 min	48°02'
October	31	744	340	47.2	173	58	10 h 55 min	36°29'
November	30	720	288	40.0	88	36	9 h 36 min	26°31'
December	31	744	274	36.8	63	23	8 h 50 min	21°44'
Year	365	8760	446	51.0	2272	51	12 h 13 min	-
Maximal	31	744	474	64.8	348	80	15 h 37 min	68°30'
Minimal	30	672	274	36.8	56	22	8 h 45 min.	21°30'

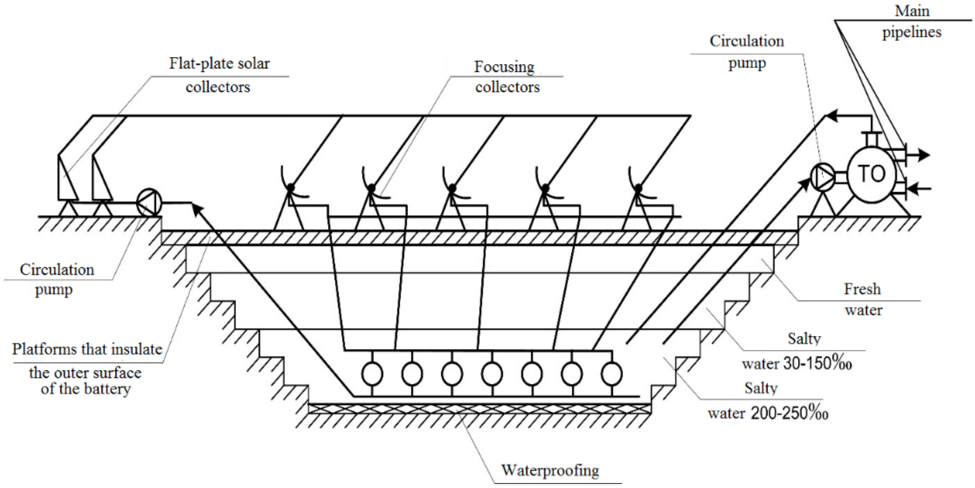

Fig. 2. Diagram of a solar station with a seasonal heat accumulator.

One of the main tasks in a solar heat supply system designing is to ensure minimum capital costs during its construction and operation, considering the guaranteed (specified by the customer) thermal characteristics. The main costs are determined by the cost and the size of the sun-absorbing surface area, which make up 40-50% of the capital costs. The basis of the calculation is based on solving the problems of heat balance and heat transfer, in determining the flow characteristics of the coolant and hydraulic losses in the system [9,10].

The main element in the solar heat supply system is a solar collector (SC), which converts solar energy into thermal energy. The sun-absorbing surface area of the cascades is determined by the formula [11-13]:

$$A_{1,2,3} = \frac{1.16 \cdot G \cdot \Delta t_i}{\eta_{1,2,3} \cdot \sum g_i}, \tag{1}$$

where $\eta_{1,2,3}$ is the efficiency of the calculated month of the first, second, and third cascade, respectively; $\sum g_i$ – the average daily intensity of the incident solar radiation of the calculated month in the collector plane, W·h/m^2; G is the required amount of hot water per day, kg/day; Δt_i is the temperature difference of the calculated month between the inlet and outlet of the collector of the first, second, and third cascades, respectively.

It follows from (1) that the A value depends on the SC efficiency, which is a measure of its technical perfection and affects the overall system efficiency. The SC efficiency is a complex function and depends both on the technical perfection of the collector and on the environmental parameters (solar radiation flux density, ambient air temperature, wind speed, etc.). Therefore, a detailed study of the SC efficiency is a very difficult task. Let's consider the efficiency calculation for SC with transparent insulation (third cascade). At the same time, a number of certain assumptions are made that do not distort the overall picture of heat balance and heat transfer. The main assumptions include the following:

1. The heat transfer process is stationary.
2. Hydraulic manifolds ensure uniform coolant distribution through the pipes.
3. The heat flow through the transparent coating and through the thermal insulation of the lower wall of the housing is one-dimensional.
4. Losses through the transparent insulation and the lower wall of the collector occur in an environment having a constant temperature.

In [12], the Hottel-Wheeler-Blass equation is essentially used to determine the average monthly efficiency of the third cascade system (with transparent insulation):

$$\eta_3 = F''' \left[(\alpha\beta) - \frac{\sum_{\tau}^{n} U_L''' \cdot (\Delta t)'''}{I_i} \right], \tag{2}$$

where F'''(αß) is the constructive efficiency of the third cascade solar collectors, (αß)≡θ; U_L''' is the average daily heat loss of the billing month for the third cascade of collectors; Δt''' is the average daily temperature difference of the billing month between the heated medium and the air temperature for the third cascade; I is the average monthly level of solar radiation, W·h/m^2.

The average daily efficiency of the SC is recommended to be calculated by the following equation:

$$\eta = \frac{Q_P}{Q_{PAD}} = F_R \left[\theta - \frac{U_L(t_1 - t_a)}{\Sigma g_i} \right], \tag{3}$$

where Q_p is the useful thermal energy transferred to the heat conductor moving in the SC; Q_{PAD} is the incident solar energy into the SC plane; F_R is the coefficient of heat removal from the collector, calculated by the following formula [2]:

$$F_R = \frac{G \cdot c_p}{U_L} \left(1 - e^{-U_L F'/G c_p} \right), \tag{4}$$

where G is the heat conductor flow rate, kg/(m^2·s); c_r is the specific isobaric heat capacity of the heat conductor, J/(kg·K); U_L is the coefficient of thermal losses of SC to the environment, W/(m^2·K).

θ is the reduced optical coefficient; t_1 is the liquid temperature at the collector inlet, °C; t_a is the average daily air temperature, °C; g_i is the average daily intensity of incident solar radiation in the collector plane, W/m^2.

In the regulatory document [5], the efficiency is determined by the following formula:

$$\eta = 0{,}8\left\{\theta - \frac{U_L\left[t_j - t_a\right]}{\sum g_i}\right\}, \tag{5}$$

where t_j is the temperature of the water in the collector, °C, $t_j = 0.5\ (t_1 + t_2)$. The coefficient 0.8 adopted in formula (5) is nothing else than F' [4]. In this equation, F'=const=0.8, does not consider the design features of the SC and its operating mode. Nevertheless, it is a function of a number of variables. For SC of the "sheet" or "plate" type, F' is determined by the following equation [4]:

$$F' = \frac{1/U_L}{\frac{1}{\alpha}+\frac{\delta}{\lambda}+\frac{1}{U_L}}, \tag{6}$$

where *1/α* is the coefficient of thermal resistance from the plate wall to the liquid; λ is the coefficient of thermal conductivity of the plate (sheet); δ is the plate (sheet) thickness, m.

For the sheet-pipe SC, F' is determined by the following equation [2]:

$$F' = \frac{1/U_L}{W\left[\frac{1}{U_L[D+(W-D)F]}+\frac{1}{C_b}+\frac{1}{\pi D_i \alpha}\right]} \tag{7}$$

where *W* is the distance between the centers of the pipes, m; *D* is the outer diameter of the pipe, m; *F* is the efficiency of the rib; C_b is the conductivity of the connection of the sheet with the pipe; D_i is the inner diameter of the pipe, m; $1/\pi D_i\alpha$ is the coefficient of thermal resistance from the pipe wall to the liquid;

In [14-16], it is recommended to present the SC efficiency in its technical documentation by the manufacturer. In the absence of this value, the following dependency is proposed:

$$\eta = 0{,}82 - 0{,}007(t_2 - t_a), \tag{8}$$

where t_2 is the water temperature at the collector outlet, limited, no more than 50°C.

To assess the equivalence of the equations (3,5,8), the test results of the experimental sheet-pipe SC (Figure 2) with the parameters of the absorber were used: *D*=0.014 m; D_i=0.012 m; *W*=0.065 m.

Processing and determination of the efficiency of the experimental SC was carried out according to the following formulas:

$$Q_k = \frac{G \cdot c_p}{A} \cdot \Delta t, \tag{9}$$

$$\Delta t = t_2 - t_1, \quad (10), \quad \eta = \frac{Q_k}{E} = f\left(\frac{t_j - t_a}{E}\right), \quad (11) \quad t_j = \frac{t_2 + t_1}{2}, \quad (12)$$

where E is the density of the total radiation flux in the collector plane, W/m^2; A is the area of the heat-receiving surface, m^2.

The test results of the experimental SC and their processing are presented in Table 2.

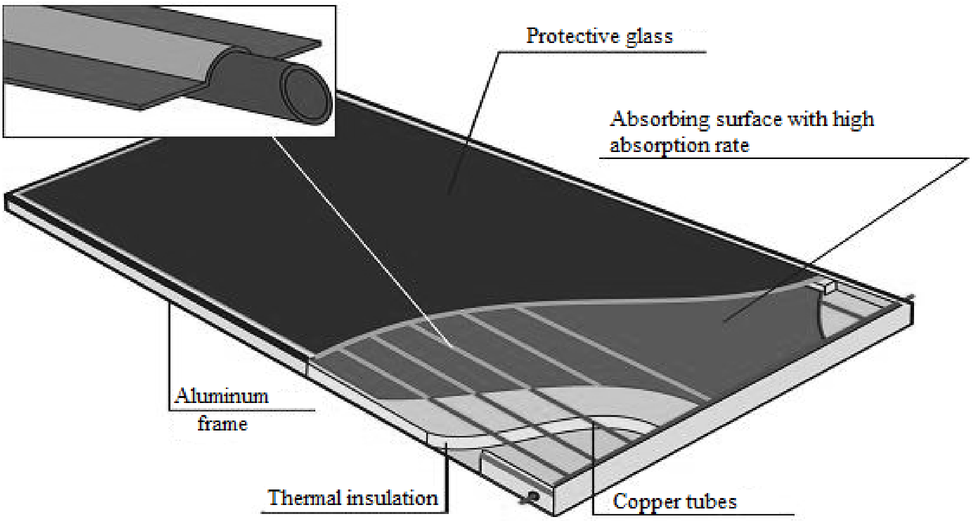

Fig. 3. "sheet-pipe" SC.

Table 2. Test results of the experimental SC and their processing.

	A, m^2		t_2, °C	Δt, °C	t_j, °C	t_a, °C	G, kg/h	E, W/m^2	Q_k, W/m^2	η
1		40.8	54.1	13.3	47.5	24.1	30.2	727.3	396.7	0.55
2		41.7	54.9	13.2	48.3	24.5	31.0	727.3	405.4	0.56
3		41.4	55.0	13.7	48.2	24.5	30.8	727.3	416.6	0.57
4		55.1	66.0	10.9	60.6	24.0	31.6	735.3	341.5	0.46
5		18.4	38.0	19.6	28.2	21.2	25.1	709.7	487.9	0.69
6		17.6	38.0	20.4	27.8	22.0	25.4	704.1	511.9	0.73
7		31.7	47.6	15.9	39.7	23.2	28.4	707.3	446.9	0.63
8	1.17	32.1	47.3	15.3	39.7	23.5	28.4	699.3	428.6	0.61
9		46.8	59.2	12.4	53.0	24.2	29.6	702.5	363.0	0.52
10		48.4	61.4	12.6	54.7	24.6	28.8	732.9	359.1	0.49
11		64.0	74.1	10.1	69.1	24.6	28.4	711.3	283.0	0.40
12		64.0	74.1	10.1	69.1	23.5	27.9	711.3	279.4	0.39
13		70.4	77.9	7.5	74.2	24.6	34.2	711.3	254.3	0.34
14		69.6	78.4	8.8	74.4	24.6	27.3	711.3	238.2	0.33
15		18.5	37.2	18.7	27.8	22.0	29.1	704.1	538.6	0.76

Tests (9) allowed us to clarify θ. Then, according to the parameters of 15 test modes from Table 3, the calculated efficiency was determined according to the equations (3,5,8) in the output temperature range from 37.8°C to 78.4°C, as well as their relative errors in relation to these tests (Table 3) [17].

Table 3. Relative errors of calculated efficiency relative to test results, %.

Test modes	Test η	Equation (3)		Equation (5)		Equation (8)	
		η	rel. er.	η	rel. er.	η	rel. er.
1	0.55	0.571	±3.7 %	0.475	±14.6 %	0.61	±10.3 %
2	0.56	0.57	±1.8%	0.474	±16.6%	0.61	±8.5%
3	0.57	0.572	±0.4%	0.475	±18.18%	0.61	±6.8%
4	0.46	0.455	±1.1%	0.375	±20.4%	0.526	±13.4%
5	0.69	0.716	±3.7%	0.597	±14.5%	0.7	±1.4%
6	0.73	0.729	±0.1%	0.605	±18.7%	0.708	±3.1%
7	0.63	0.634	±0.6%	0.526	±18%	0.649	±3%
8	0.61	0.632	±3.5%	0.526	±14.8%	0.653	±6.8%
9	0.52	0.518	±0.4%	0.429	±19.2%	0.575	±10%
10	0.49	0.515	±5%	0.427	±13.7%	0.562	±13.7%
11	0.40	0.376	±6.2%	0.309	±18.9%	0.474	±17%
12	0.39	0.375	±3.9%	0.308	±23.5%	0.473	±19.2%
13	0.34	0.327	±3.9%	0.268	±23.7%	0.447	±27.2%
14	0.33	0.326	±1.2%	0.269	±20.4%	0.443	±29.2%
15	0.76	0.733	±3.6%	0.604	±22.9%	0.714	±6.2%

It follows from the table that in equation (5) the relative error of the calculated efficiency, in comparison with the efficiency of the tests, is from ±13.7% to ±23.7%. According to equation (8), the relative error of the calculated efficiency, compared with these tests at t_2 ≤50°C, does not exceed ±3%. For other modes in which t_2 >50°C, from ±6.8% to ±29.2%. According to equation (3), the relative error is from ±0.1% to ±3.9% and one mode is ±6.2%. Thus, equation (3) can be taken as a "reference" when calculating the local efficiency of the SC.

Considering that in equation (5) F'=const=0.8, and as is known, F' and U_L are interrelated. In equation (5'), instead of the coefficient 0.8, the parameter F' is introduced, i.e.

$$\eta = F'\left[\theta - \frac{U_L(t_j - t_a)}{\sum g_i}\right] \tag{5'}$$

U_L was calculated by the equation:

$$U_L = U_t + U_b, \tag{13}$$

where U_b is the coefficient of losses through the lower surface of the collector (W/m²·K) is determined by the expression:

$$U_b = \frac{k_{in}}{h_{in}} \tag{14}$$

where k_{in} is the thermal conductivity coefficient of insulation, W/(m·K); h_{in} is the insulation thickness, m.

U_t – the loss coefficient through the upper surface of the third cascade collector is determined by the following equation [3]:

$$U_t = \left(\frac{1}{h_{p-c}+h_{r,p-c}} + \frac{1}{h_\omega+h_{r,c-s}}\right)^{-1},$$

(15)

where h_{p-c} is the coefficient of convective heat transfer between the plate and the glass, is determined by the following equation [3]:

$$h_{p-c} = [1 - 0.0018(\bar{T} - 10)] \cdot \frac{1{,}14\Delta T^{0.31}}{l^{0.07}},$$

(16)

where $\bar{T}$ - the average temperature between the plates, °C; ΔT is the temperature difference between the average temperature of the plate and the glass, °C; l is the distance between the plates, cm;

$h_{r,p-c}$ is the coefficient of heat transfer by radiation from the plate to the glass, determined in accordance with the following equation:

$$h_{r,p-c} = \frac{\sigma\left(T_p{}^2+T_c{}^2\right)\left(T_p+T_c\right)}{\left(1/\varepsilon_p\right)+(1/\tau)-1},$$

(17)

where σ=5.67·10^{-8} W/m^2·K; T_s, T_p is the temperature of the glass and plate, respectively, K; ε_p is the degree of plate blackness; τ is the glass transmittance.

h_ω is the convective heat transfer coefficient in the OS:

$$h_\omega=5.7+3.8U_0, \tag{18}$$

where U_0 is the wind speed, m/s. The calculation was performed at U_0=0 m/s.

$h_{r,c-s}$ is the coefficient of heat transfer by radiation from the glass to the OS, determined by the following equation:

$$h_{r,c-s} = \varepsilon_c\sigma\left(T_c + T_s{}^2\right)(T_c + T_s),$$

(19)

where T_s is the air temperature, K.

T_s is determined by the iteration method using the following expression [2]:

$$T_S = T_p - \frac{U_t\left(T_p + T_a\right)}{h_{p-c} + h_{r,p-c}},$$

(20)

Calculated local efficiencies according to equations (3) and (5'), as well as their relative errors in relation to test data are presented in Table 4. At the same time, for modes in which t_{out} >50°C and t_{in}>t_v, considering the logarithmic change in the liquid temperature, it was determined by the following equation:

$$\Delta t = (t_2 - t_a) \cdot \left(exp^{-\frac{F' U_L}{G \cdot c_p}} \right) \tag{21}$$

Table 4. Relative errors of the calculated efficiency relative to the test results according to equations (3) and (5').

Test modes	Test η	Equation (3)		Equation (5')	
		η	rel. er.	η	rel. er.
1	0.55	0.571	±3.7 %	0.565	±2.7 %
2	0.56	0.57	±1.8%	0.565	±0.8%
3	0.57	0.572	±0.4%	0.565	±0.9%
4	0.46	0.455	±1.1%	0.447	±2.9%
5	0.69	0.716	±3.7%	0.709	±2.7%
6	0.73	0.729	±0.1%	0.719	±1.5%
7	0.63	0.634	±0.6%	0.626	±0.6%
8	0.61	0.632	±3.5%	0.627	±2.7%
9	0.52	0.518	±0.4%	0.511	±1.7%
10	0.49	0.515	±5%	0.511	±3.6%
11	0.40	0.376	±6.2%	0.368	±0.5%
12	0.39	0.375	±3.9%	0.367	±2.5%
13	0.34	0.327	±3.9%	0.32	±3.7%
14	0.33	0.326	±1.2%	0.317	±1.2%
15	0.76	0.733	±3.6%	0.721	±5.3%

When calculating the solar DHW system [18-20] in the non-heating period, the following are used: the average daily intensity of the incident solar radiation of the calculated month in the collector plane, the average daily air temperature of the calculated month, average daily heat losses.

Based on the analysis of the results of the calculated efficiency relative error with the data determined during the tests, the formula (5') was adopted as the "reference" equation for calculating the DHW system.

4 Results

With regard to the climatic conditions of the Crimean Peninsula, according to the equations (2,3,8), the daily efficiency for the third cascade was calculated by months for the SC "plate" with one layer of transparent insulation, a surface of $1 m^2$ and parameters: U_L=2 W/($m^2 \cdot K$), U_L=4.3 W/($m^2 \cdot K$) and U_L=8 W/($m^2 \cdot K$); θ – the reduced optical characteristic is adopted for a real collector as 0.809. Preliminary studies have shown that the optimal temperature at the inlet to the third cascade in the period from April to October is 31°C [6]. Outlet temperature t_2=50°C. The coolant flow rate is assumed to be 10 kg/($m^2 \cdot h$).

The obtained efficiency values are presented graphically (Figure 4).

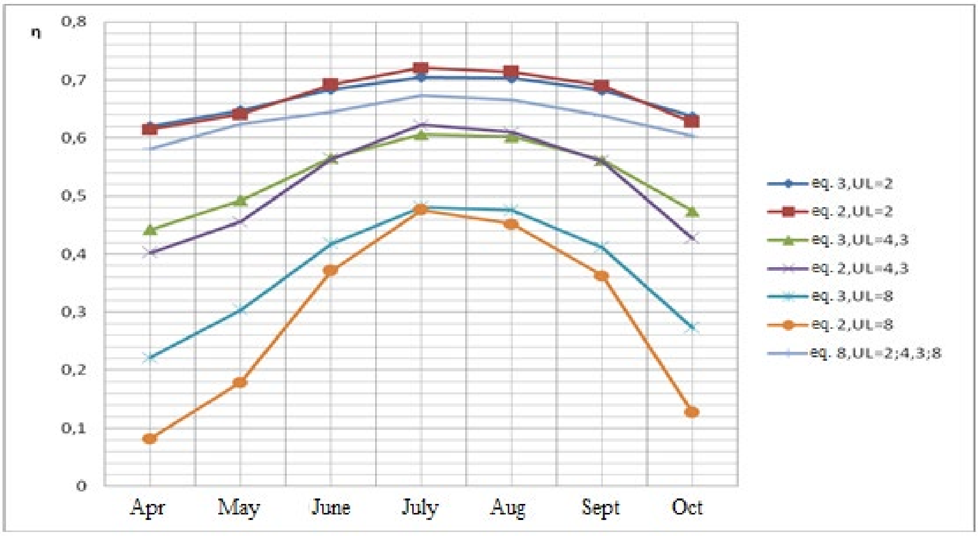

Fig. 4. Calculated values of the SC "plate" (sheet) efficiency for U_L = 2 W/(m^2·K), U_L=4.3 W/(m^2·K), and U_L=8 W/(m^2·K).

Figure 5 shows the SC "sheet-pipe" efficiency made of aluminum material according to the formulas (2,3,8) in relation to the climatic conditions of the city of Sevastopol. At the same time, the optical parameters remained the same [21]. For the SC manufactured as "sheet-pipe", the parameters of a real collector are introduced: W is the distance between the pipes (W= 0.1 m); the efficiency of the rib F = 0.989; $D_{internal}$ = 0.012 m; D_{outer}=0.014 m; rib thickness is 0.001 m. For these conditions, when calculating the efficiency according to (3), the parameters F' and F_R were specified according to equations (4,7).

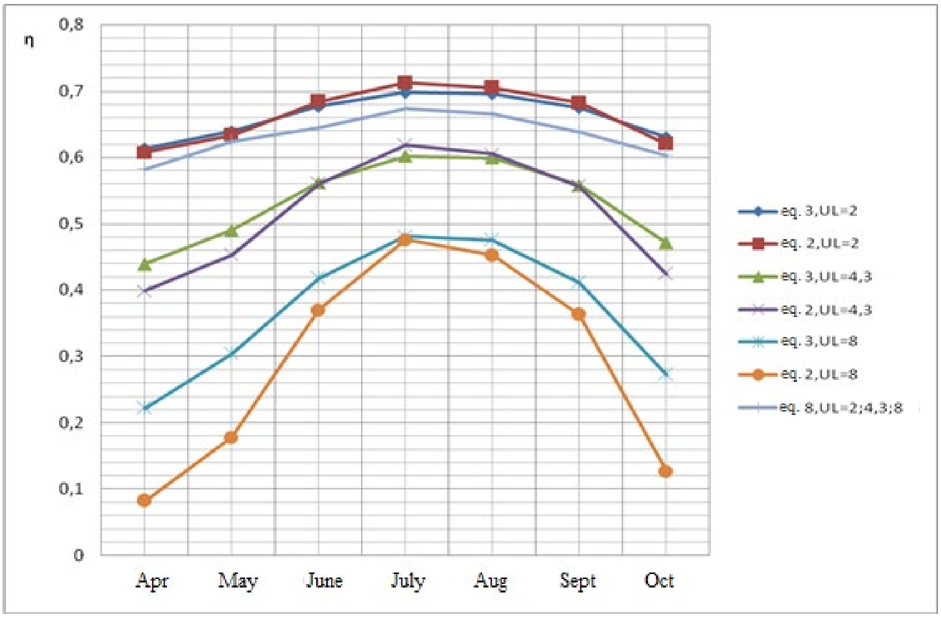

Fig. 5. Calculated values of the sheet-pipe SC efficiency for U_L=2 W/(m^2·K), U_L=4.3 W/(m^2·K), and U_L=8 W/(m^2·K).

The calculated values of the relative error of equations (3) and (8) relative to equation (2) at U_L>2 W/(m^2·K) exceed the five percent barrier. Calculations have shown that at t_2>50°C the relative error of these equations increases significantly.

It follows from Figures 3 and 4 that the calculated values of the average daily efficiency for months in (8) do not depend on the design of the SC (F') and the quantitative values of environmental losses (U_L). Formula (8) cannot be accepted for calculations of solar DHW systems based on SC of various designs. The efficiency values under (3) can be calculated at U_L≤2 W/(m^2·K). The graphs in Figures 3 and 4 show that SC with parameters U_L=8 W/(m^2·K) are not acceptable for the third stage. To calculate the efficiency of the third cascade in the solar DHW system during the non-heating period, equation (2) should be used, the calculation algorithm of which is shown in Figure 6.

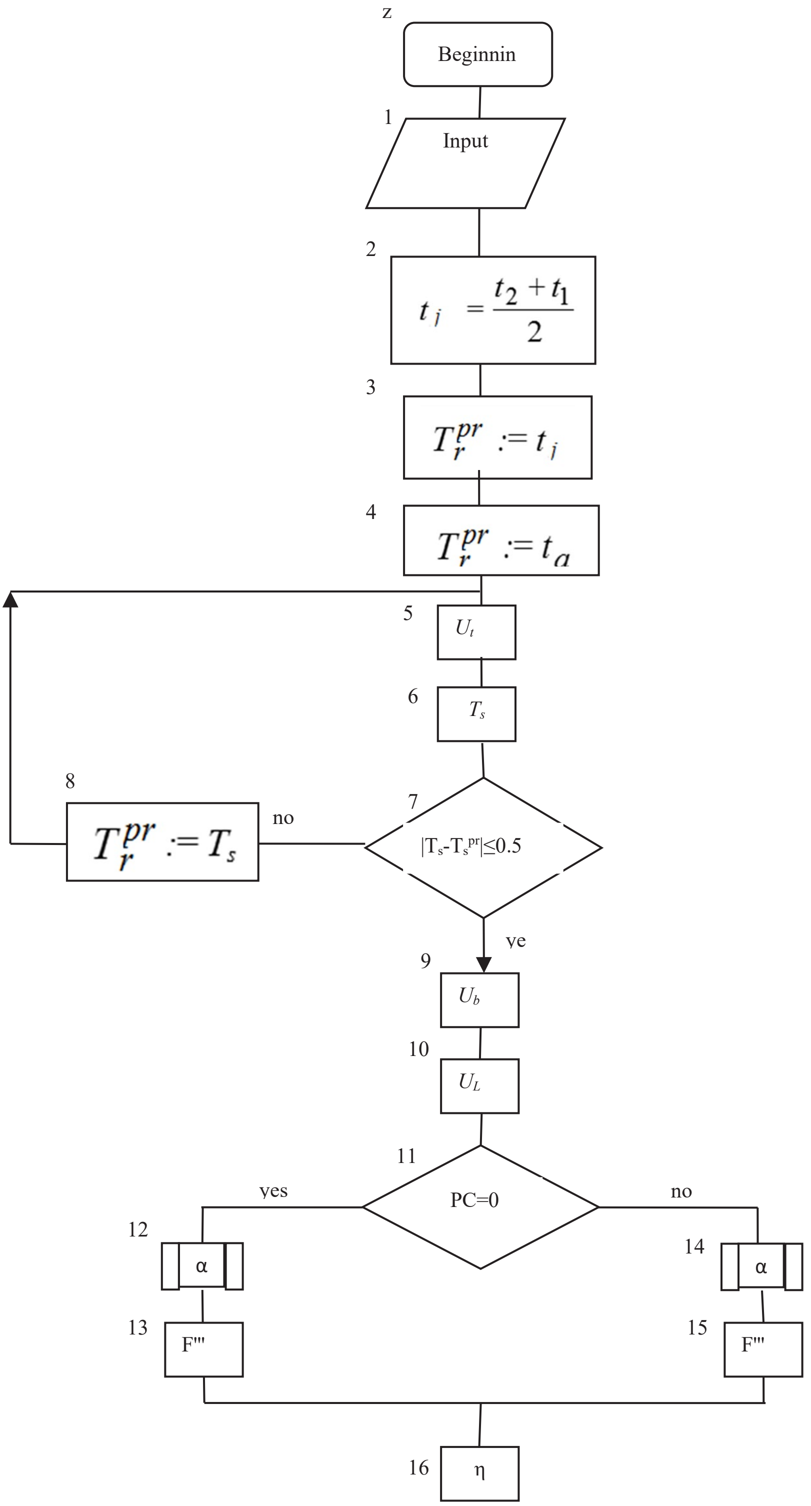

Fig. 6. The algorithm for calculating the efficiency of the third stage in the solar DHW system.

Block 1 "Input": input of structural and optical parameters of the SC of the third cascade; physical properties of the heated liquid; physical parameters of the air; total daily values of solar radiation levels by month according to statistical data of the region, considering the SC location; physical parameters of the SC materials; modes of heat conductor flow through the SC and their temperatures. Block 5: calculation of U_t by formulas (15)-(19). Block 6: calculation of T_s by the formula (20). Block 9: calculation of U_b by formulas (14). Block 10: calculation of U_L by formulas (13). Block 11: PC=0 – SC "plate"; PC≠0 – SC "sheet-pipe". Block 12, 14: calculation of α by the formula: $\alpha = \frac{Nu \quad \lambda}{d_{in}}$; $Nu=0.33 \cdot Re^{0.5} \cdot Pr^{0.33}$. Block 13: calculation of F" according to the formula (6). Block 15: calculation of F" according to the formula (7). Block 16: efficiency calculation according to the formula (2).

To clarify the calculations of efficiency in the solar heat supply system of the second and first cascades, thermal tests of the solar collector without transparent and rear insulation are being conducted.

5 Conclusions

The equations for calculating the efficiency of SC with transparent and rear insulation in a three-cascade solar heat supply system in the non-heating period are clarified. It is established that in the SC of the third cascade, the parameter U_L should not exceed 4.3 W/($m^2 \cdot K$).

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