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The Development and Calculation of an Energy-saving Plant for Obtaining Water from Atmospheric Air

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Abstract. The article shows the calculation of characteristics of energy-efficient water generator from atmospheric air. This installation or the atmospheric water generator is the unique mechanism which produces safe drinking water by extraction it from air. The existing atmospheric generators allow to receive safe drinking water by means of process of condensation at air humidity at least equal to 35% and are capable to give to 25 liters of water in per day, and work from electricity. Authors offer to use instead of the condenser in the scheme of installation for increase volume of produced water by generator in per day, the following refrigerating machines: the vapor compression refrigerating machines (VCRM), the thermoelectric refrigerating machines (TRM) and the Stirling-cycle refrigerating machines (SRM). The paper describes calculation methods for each of refrigerating systems. Calculation of technical-and-economic indexes for the atmospheric water generator was carried out and the optimum system with the maximum volume of received water in per day was picked up. The atmospheric water generator which is considered in article will work from autonomous solar power station.

1. Introduction

At present, there are problems of energy saving in various areas of human life and in power complexes of industrial plants [1, 2, 3].

According to the World U.N. report about development of water resources, many countries already achieved limit opportunities of water use: consumption of fresh water for the last half a century increased triply. In extensive regions of developing world unequal access to safe drinking water, water purification for production of provisions and processing of drainage water remains. Nearly five billion people will be left without drinking water to 2030 if people will not undertake nothing, and it is about 67% of the population of the planet. Problems with drinking water and water resources have led to creation of atmospheric water generators. The atmospheric water generator is the unique mechanism which produces safe drinking water by extraction it from air. For the first time the atmospheric water generators were developed in the 1990th. They were similar to system which is used for dehydration air in refrigerators.

In this article the analysis is made of the energy-efficient installation working from the photo-electric converter (installation power W = 200V) with use of refrigeration installations, such as the VCRM, the TRM and the SRM. The analysis allowed to estimate fully overall operational efficiency of the generator and quality of water, and also promoted adoption of the design solutions increasing the volume of received water by installation in per day.

2. The atmospheric water generator working at VCRM basis

Achievable temperatures in installations, refrigerating capacity and expenses of mechanical work significantly depend of kind and properties of refrigerant coolants. Refrigerant coolants need to have ability to absorb a large amount of warmth at evaporation, to have small specific volumes of steam, low critical temperatures, viscosity and density, high coefficient of heat dissipation and heat transfer, to be harmless, fireproof, available and inexpensive. Therefore, for work two refrigerant coolants were chosen: R134A and R502.

In this work for calculation and analysis of a cycle of the VCRM the CoolPack [4] program was used. It is the program for design, calculation, analysis and optimization of refrigerating systems.

We will consider the return Rankine cycle for each refrigerant coolant in autumn, spring, summer and winter seasons. Example is shown on the figure 1. Values of average temperatures and humidity in the Samara region for every season, and also the consumed power by installation from the photo-electric converter are presented in table 1.

Table 1. Initial data for calculation of the VCRM

	winter	spring	*summer*	autumn
Temperature(°C)	-15	+8	+24	+10
Moisture content(%)	85	65	50	75
Power consumption plant N (W)			200	

Such levels of temperature of evaporation at calculation are caused by need of choice of the optimum mode which corresponds to the maximum quantity of the received water. All values received in the program are tabulated for each refrigerant coolant and season.

For definition of moisture content at each mode we will use I-d diagram of humid air [5].

Schedules of dependences of moisture content d, mass and volume consumption (m and V) of refrigerant coolant, refrigerating coefficient COP, heat input Qe and heat output Qc from temperature T for each refrigerant coolant were constructed. We will consider dependences for refrigerant coolant R134A on the figures 1-4.

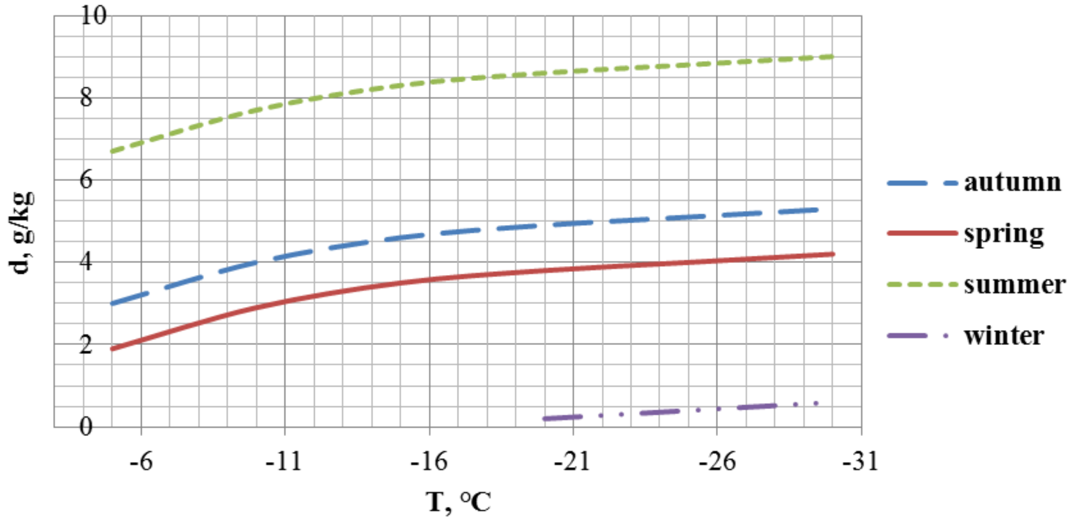

Figure 1. Addiction of moisture content d from temperature T for refrigerant coolant R134a

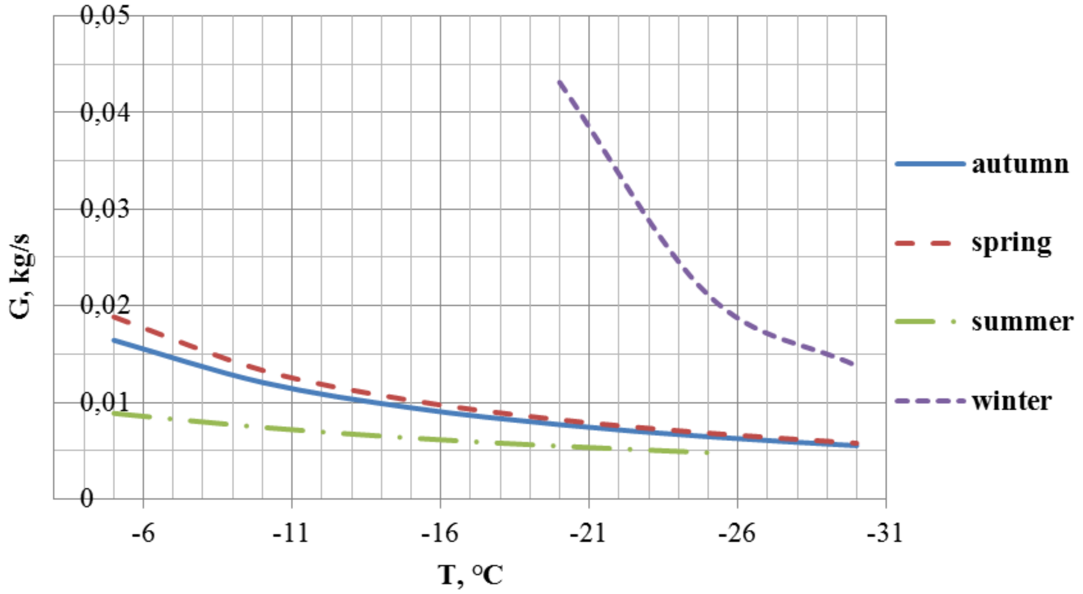

Figure 2. Addiction of mass consumption from temperature T for refrigerant coolant R134a

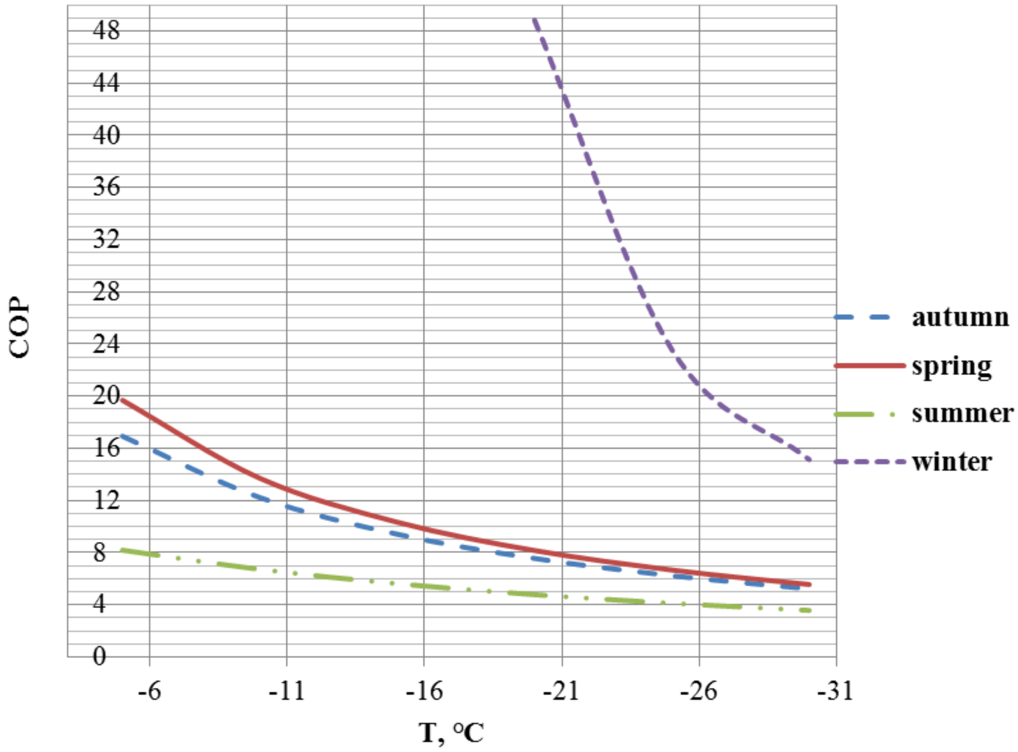

Figure 3. Addiction of refrigerating coefficient COP from temperature T for refrigerant coolant R134a

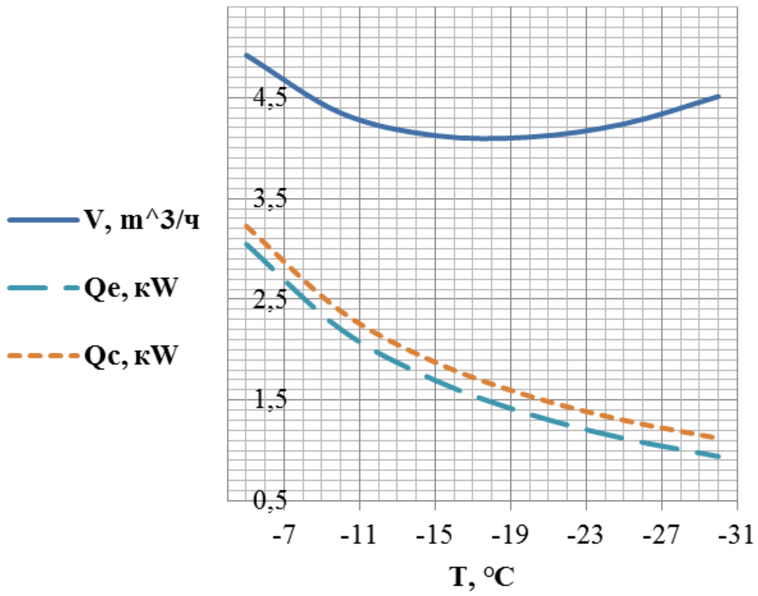

Figure 4. Addiction of volume consumptionV, heat input Qe and heat output Qc from temperature T for refrigerant coolant R134a

The analysis of received values showed that during the summer period at temperature of evaporation -30 °C in the VCRM on refrigerant coolant R134a the maximum quantity of moisture content is developed. But for further calculation we choose the autumn period which corresponds to average value of received moisture content.

Further by known methods [6], main elements of the VCRM were calculated: compressor, condenser and evaporator. Values of coefficients of heat transfer are defined k_{ev}=39,1 W/m^2*K and k_{cond}=23 W/m^2*K, heat dissipation α=41,31 W/m^2*K and α=23 W/m^2*K, effective refrigerating coefficients ε=2,24, Qe=0,942 kW, Qc=1,122 kW and geometrical parameters of evaporator F=0,27 m^2, condenser F=0,18 m^2 and compressor Ne=0,309 kW.

3. Calculation of the Stirling-cycle refrigerating machine

Calculation was carried out by Schmidt's method [7].

For calculation the following main equations were used:

- Equations of motion

$$V_h = 0,5 \cdot V_{OX} \cdot (1+\cos\varphi) \quad (1)$$

$$V_c = 0,5 \cdot K \cdot V_{OX} \cdot (1+\sin\varphi) \quad (2)$$

$$0,5 \cdot K \cdot V_{OX} \cdot (1+\sin\varphi) + 0,5 \cdot V_{OX} \cdot (1+\cos\varphi) + X \cdot V_{OX} \quad (3)$$

$$V_P = X \cdot V_{OX} \quad (4)$$

- Energy balance equations

$$L_c = Q_1 - Q_2 \quad (5)$$

$$Q_1 = Q_2 \cdot \tau \quad (6)$$

$$\varepsilon = \frac{Q_2}{L_c} \quad (7)$$

- Material balance equations

$$m_z = m_h + m_c + m_M \quad (8)$$

$$\tau = \frac{T_h}{T_c} \quad (9)$$

$$K = \frac{V_h}{V_c} \quad (10)$$

$$S = \frac{2 \cdot X \cdot \tau}{\tau + 1} \quad (11)$$

$$\delta = \frac{\sqrt{\tau^2 + k^2 + 2 \cdot \tau \cdot k \cdot \cos\alpha}}{\tau + k + 2 \cdot S} \quad (12)$$

- Gas equation

$$m_h = \frac{P_h \cdot V_h}{R \cdot T_h} \quad (13)$$

$$m_c = \frac{P_c \cdot V_c}{R \cdot T_c} \quad (14)$$

$$m_M = \frac{P_M \cdot V_M}{R \cdot T_M} \quad (15)$$

Calculation of the regenerative heat exchanger was carried out. Parameter found

$$\eta_P = \frac{N_1}{N_1 + 1}\left[1 - \frac{1}{9\left(\frac{C_H m_H}{C_P m_3}\right)^2}\right] \quad (16)$$

and losses for approach, the hydraulic resistance and total losses from ratio l/d, hot and cold heat exchangers, external loading.

4. Calculation of the thermoelectric refrigerating machines

Calculation of parameters of the thermoelectric refrigerating machines was carried out by method [8]. Calculation of the refrigerating thermobattery in mode of the maximum refrigerating coefficient.

Further the software product «KRYOTHERM» [9] was used for calculation and choice of standard modules. Initial data for calculation in program are Tr = 283 K, Tx= 243 K, Qo = 25 W. The module TB–127–2,0–1,65 was picked up for initial data and characteristics of chosen module were constructed (figure 5, 6).

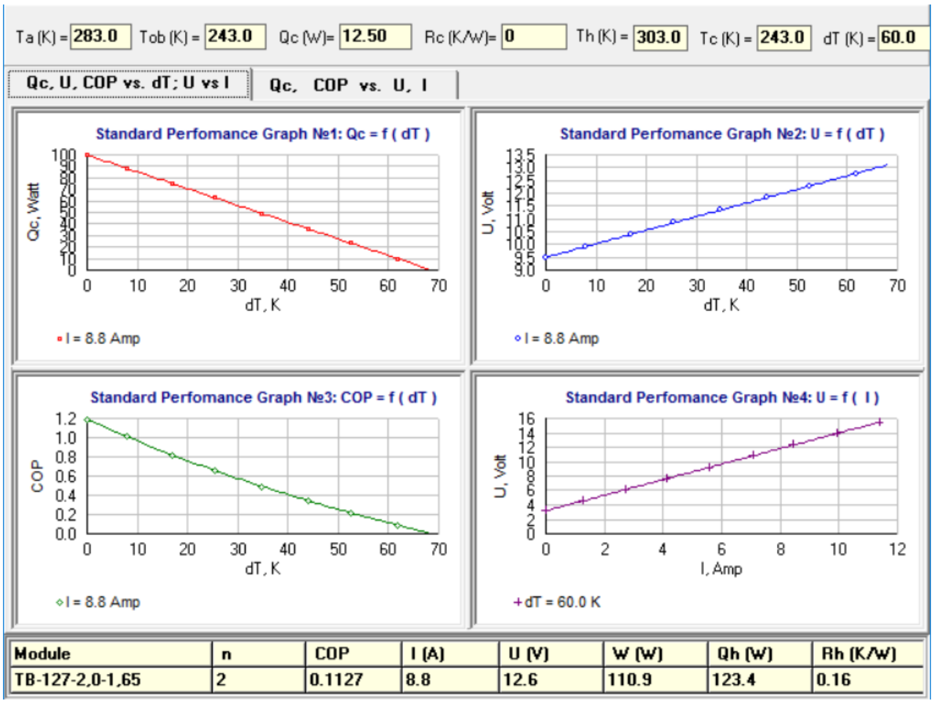

Figure 5. Standard performance Qc = f(dT),U = f(dT), COP = f(dT), U = f(I) module TВ – 127–2,0–1,65

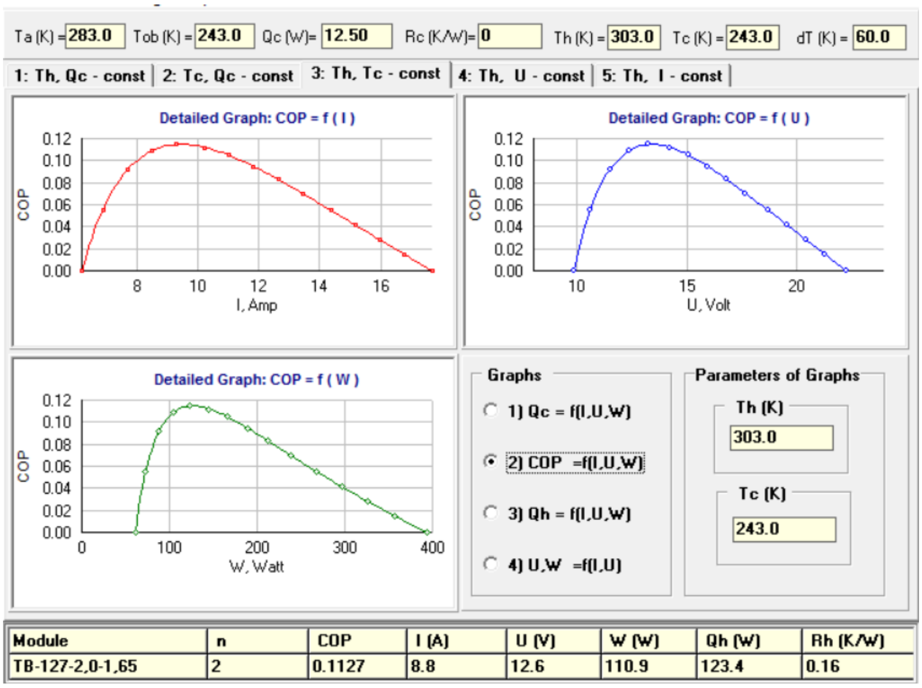

Figure 6. Detailed performance Qc = f(dT),U = f(dT),COP = f(dT), U = f(I) module TB – 127–2,0–1,65

5. Determination of efficiency of installation

The produced quantity of water in per day by the atmospheric generator with various refrigerating machines was determined by the following method:

$$m_w = \frac{G_a \cdot d \cdot 24}{1000} \qquad (17)$$

where G_a - air-mass flow through automatic ventilating machine; d - humidity.

Air-mass flow through automatic ventilating machine is equal:

$$G_a = V_{fan} \cdot \rho_a \cdot n \qquad (18)$$

where V_{fan} - air-quantity flow through automatic ventilating machine; ρ_a - density of air; n - quantity of automatic ventilating machines.

We will define the necessary quantity of automatic ventilating machines: $n = \frac{F_{out}}{F_{fan}}$ and air-quantity flow through automatic ventilating machine V_{fan}, knowing F_{out} and F_{fan} .

6. Results

The produced quantity of water in per day by the atmospheric generator with refrigerating systems table 2.

Table 2. The produced quantity of water in per day by the atmospheric generator with refrigerating systems

	N, W	*n, pcs.*	*Nn, W*	*d, g/kg*	*mw, l/d*
VCRM	180	12	10,08	5,3	84
SRM	180	7	5,88	5,3	48
TRM	180	1	0,84	4,8	2

Number of technical-and-economic indexes was entered for determination of efficiency, total power consumption, energy and monetary cost 1 l/days of water table 3.

Table 3. Technical-and-economic comparison of installation with various refrigerating systems

	N, W	NΣ, W	Nn, W	m, kg	mw, l/d	Q c, W	Kq0, l/d*kW	Km, l/d*kg	P, rub	Kc, l/d*rub	WΣ, kW*h	C, rub/l
VCRM	180	190,1	10,08	13,5	84	942	0,09	6,22	47 000	0,002	4,6	0,13
SRM	180	185,9	5,88	11	48	1134	0,04	4,36	82 000	0,001	4,46	0,23
TRM	180	180,8	0,84	1,3	2	25	0,08	1,54	14 000	0,0001	4,34	5,58

For technical-and-economic comparison we offer to use the following coefficients:

- coefficient of efficiency depending on its refrigerating capacity, mass, price, and also energy and monetary cost 1 l/days of water.

$$K_{q0} = \frac{m_w}{Q_c}, \frac{l}{d \cdot kW} \tag{19}$$

- coefficient of efficiency depending on mass and price of refrigerating system is determined by the following formulas

$$K_m = \frac{m_w}{m_\Sigma}, \frac{l}{d \cdot kg} \tag{20}$$

$$K_c = \frac{m_c}{P}, \frac{l}{d \cdot rub} \tag{21}$$

- the total power consumed by installation

$$W_\Sigma = \frac{N_\Sigma}{1000} \cdot 24, kW \cdot h \tag{22}$$

- We will use the following formulas for determination of energy and monetary cost 1 l/days

$$C_{en} = \frac{W_\Sigma}{m_w}, \frac{kW \cdot h}{l / d} \tag{23}$$

$$C = C_{en} \cdot i, \frac{rub}{l / d} \tag{24}$$

Electric rate is $i = 2,57 \frac{rub}{kW \cdot h}$.

7. Conclusion

This article describes methods of calculation and selection of main elements of refrigerating systems used in the atmospheric generator for increase of volume of received water. As refrigerating system the vapor compression refrigerating machines, the thermoelectric refrigerating machines and the Stirling-cycle refrigerating machines are considered.

The return Rankine cycles were constructed in the program CoolPack for the vapor compression refrigerating machine. Calculation of cycle parameters is carried out for autumn, spring, summer and winter seasons for refrigerant coolants R134a and R502. On the basis of received results the optimum mode - the autumn season was chosen ($t_u = -30°C$, $t_к = 10°C$, $Q_e = 0,942 kW$, $Q_c = 1,122 kW$, $d = 5,3 g / kg$) which is used in further calculations. Calculation was carried out for evaporator
($F = 0,27 m^2$), condenser ($F = 0,18 m^2$) and compressor ($N_e = 0,309 kW$).

In the program Kryotherm the thermoelectric module TB-127-2,0-1,65 (n = 2 pcs) was picked up and was calculated for the thermoelectric refrigerating system. Calculation was carried out for regenerator ($\eta_p = 0,98$, $\Delta Q_\Sigma = 115,63W$), heat exchangers of external loading of compressor and expander cavities, volumes of hot and cold cavities ($Q_1 = 78,3J$, $Q_2 = 67,5J$) for the Stirling-cycle refrigerating machine.

The automatic ventilating machine DEEPCOOL XFAN 120 was picked up ($V = 44m^3 / h$, $N = 0,84W$). The necessary quantity of automatic ventilating machines for the VCRM is n=12 pcs, for the SRM is n=7 pcs, for the TRM is n=1 pcs. The quantity of received water in per day was calculated. So, the atmospheric generator with the VCRM can generate $m_w = 84\ l / d$, with the SRM - $m_w = 48\ l / d$, with the TRM - $m_w = 6\ l / d$

Coefficients of efficiency were calculated for each system depending on its refrigerating capacity K_{q0} , mass K_m , price K_c , and also energy C_{en} and monetary C cost 1 l/days of water. Monetary cost 1 l/days of water received by the atmospheric generator with the VCRM is $C = 0,13\frac{rub}{l / d.}$, by the atmospheric generator with the SRM is $C = 0,23\frac{rub}{l / d.}$, by the atmospheric generator with the TRM is $C = 5,58\frac{rub}{l / d.}$.

The optimum refrigerating machine with the maximum efficiency of water in per day is the vapor compression refrigerating machine for work in fixed conditions.

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