CÁLCULO DE DESEMPENHO DA UNIDADE DE AQUECIMENTO DE ÁGUA SOLAR EM UM POSTO DE GASOLINA

PERFORMANCE CALCULATION OF SOLAR WATER HEATING UNIT AT A PETROL FILLING STATION

РАСЧЕТ ПРОИЗВОДИТЕЛЬНОСТИ СОЛНЕЧНОЙ ВОДОНАГРЕВАТЕЛЬНОЙ УСТАНОВКИ НА АВТОМОБИЛЬНО-ЗАПРАВОЧНОЙ СТАНЦИИ

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RESUMO

Fontes alternativas de fornecimento de energia são frequentemente mais implementadas. Devido à renovabilidade e inesgotabilidade desta fonte energia. Este artigo trata do desempenho de uma unidade de aquecimento solar de água. Utilizando-se os resultados teóricos e experimentais, bem como o valor de projeto das temperaturas do transferidor de calor no coletor solar, foram realizados os cálculos de previsão do desempenho térmico diário admissível da unidade nas condições das zonas climáticas da República do Cazaquistão, dependendo do mês e da área do coletor solar. Afirma-se que a área maior dos coletores solares influencia no aumento da produtividade, devido à redução da profundidade de resfriamento e o aumento correspondente no coeficiente de conversão. Tanto o projeto quanto o desempenho experimental diário têm uma precisão razoável de 7%, o que mostra a confiabilidade das fórmulas de projeto. A dependência do desempenho térmico sazonal sobre a área de coletores solares também foi calculada. Foi determinado que a diferença de temperatura e a intensidade da radiação solar praticamente não afetam o modo de operação de instalação.

Palavras-chave: *energia solar, coletor solar, unidade de aquecimento solar de água*.

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ABSTRACT

Alternative resources of energy supply are implemented more often. Due to the renewability and inexhaustibility of this energy source. This paper deals with the performance of a solar water heating unit. By using theory and experimental results, as well as an obtained design value of the heat carrier temperatures in the solar collector, the prediction calculations of the admissible daily thermal performance of the unit in the conditions of the climatic zones of the Republic of Kazakhstan were conducted, depending on the month and the area of the solar collector. It is stated, that the bigger area of solar collectors influences the increase of productivity, due to the reducing of the cooling depth and the corresponding increase in the conversion coefficient. Both design and experimental daily performance has reasonable precision at 7%, which shows the reliability of the design formulas. The dependency of seasonal thermal performance on the solar collectors' area has also been calculated. It was determined that the temperature difference and the intensity of solar radiation practically do not affect the operating mode of the installation.

Keywords: *solar energy, solar collector, solar water heating unit.*

АННОТАЦИЯ

Альтернативные источники энергоснабжения применяются все чаще, поскольку эта энергия возобновляемая и практически неисчерпаемая. Авторами, в данной работе, была рассчитана производительность солнечной водонагревательной системы на автомобильно-заправочной станции. С использованием теоретических и экспериментальных результатов, а также полученных расчётных значений температур теплоносителя в гелиоколлекторе, выполнены прогнозные расчёты возможной суточной теплопроизводительности установки в условиях климатических зон Республики Казахстан, в зависимости от месяца и площади гелиоколлектора. Установлено, что большая площадь гелиоколлекторов влияет на увеличение производительности за счет уменьшения глубины охлаждения теплоносителя и соответственного возрастания коэффициента преобразования. Установлено, что расчетное и экспериментальное суточное КПД имеют удовлетворительную сходимость в пределах 7%, что свидетельствует о достоверности расчетных формул. Так же рассчитана зависимость сезонной теплопроизводительности от площади гелиоколлекторов. Определено, что разность температур и интенсивность солнечной радиации практически не влияют на рабочий режим установки.

Ключевые слова: *солнечная энергия, гелиоколлектор, солнечная водонагревательная установка.*

INTRODUCTION

Flat collector consists of metal absorbent in flat rectangular case. The glass cap at the upper surface and the insulation on the bottom and on each side decreases heat losses. Atmospheric air presents in the space between the metal absorbent and the transparent cap. The flat metal plate serves as a heat exchanger, which absorbs solar radiation, transforming it into heat and transferring the heat to the fluid liquid (Kalogirou, 2004).

In the northern climatic conditions, the average annual performance of well-designed solar water heating units is usually equal to about 35-40% (The German Solar, 2007). Moderate climate is characterized by the absence of extreme temperatures and precipitation (rain and snow), and contrast summer and winter. Both in summer and in winter moderate weather may be very changeable. One day it may be rainy, and then sunny. These climatic zones are located in the range of 40-60/70° of northern latitudes (Ayompe *et al.*, 2011).

Solar collectors are the main component, so the estimate of their lifecycle heating characteristics is vital. According to EN 12975-2 and ASHRAE 93-86 standards a series of researches were conducted on performance in stationary and quasi-dynamic experimental conditions (CSN EN 12975-2, 2015; ASHRAE-93, 2003). In Padua, Italy Zambolin and Del Col conducted a comparative performance analysis of thermal characteristics of flat plates and evacuated trickle collectors (Zambolin and Del Col, 2010). They presented a new set of data, collected both for flat plate and evacuated trickle collectors, which were tested simultaneously by the stationary and quasi-dynamic performance tests according to standard EN 12975-2.

In India Tiwari *et al.* (1991) analyzed the performance of solar water heating units with the values in the range from 5.399 to 7.024. Amer *et* *al.* (1998) developed a transition method for characteristics of the dynamic behavior of solar FPC and confirmed their result compared to the other results, obtained during the tests at the set level, based on standard ASHRAE 93-86. They also studied the influence of temperature at the inlet and incident angle on the collector's parameters. At different velocities of the flow, they conducted the tests to estimate the performance of two solar water heating units (Chen *et al.*, 2012). In Jordan, they conducted an experimental research, calculation of the performance of four types of solar collectors and an evacuated trickle collector (Sakhrieh and Al-Ghandoor, 2013).

In Madrid, Italy they conducted the performance calculation of 50 m2 of the solar system for use of hot water and hot water supply (Rodríguez-Hidalgo *et al.*, 2012). In Cyprus the experiments with a 68 l hot water tank were conducted, using the data collected during 2 years (Michaelides and Eleftheriou, 2011). In Dublin, Ireland they compared year-round energetic and economic indicators of two solar water heating units with flat plates and evacuated trickle collectors working in the same weather conditions (Ayompe *et al.*, 2011). In Cambridge shire, Great Britain the research institution evaluated the performance of solar water heating unit with the panel of solar panels with flat display in it (Building Research Establishment, 2009). The testing unit included the automated system with the effects of the supplementary heating system (boiler or immersion heater) and daily use of hot water in an average European household, described in the standard ideal withdrawal cycle of the EU (EU M324EN), equal to 100 liters at 60 °C. Their results have shown that during a year a collector with the area of 3 m^2 generated 5266 mJ of heat, which was 57% of the need for hot water. In (Duffie and Beckman, 2006) the authors have developed a hybrid system of using renewable energy sources for local consumers in agriculture, which consists of 3 kW trickle solar collectors. In (Keshuov *et al.*, 2017; Omarov *et al.*, 2017) they have developed the thermal capacity of the combined system with solar collectors and thermal pump. In the article, the system was studied using the example of a cattle-breeding farm (Omarov *et al.*, 2017). The new system (Kunelbayev *et al.*, 2017) performs these functions; he recycles heat, organizes their movement and accumulation and smoothest uneven SE. The main components of the system are Solar power station (SPP), milk cooler, climate unit, heat pump (HP), battery, automatic control system and heating device and hot water. The authors consider heat absorption by a heat-transfer agent in a flat plate solar collector as well as the construction of such solar collectors.

METHODS

A Petrol Filling Station, building roof area is 60 m^2. Amount – 4. In order to install a solar system of hot water supply let us calculate the area for solar collectors, which will be set at the roof of a building (Figure 1, Figure 2).

The area of the solar collectors F, is defined for the "coldest" month (February) according to the Equations 1, 2, 3 (Duffie and Beckman, 2006):

$$F = Q/q_i, \quad \text{(Eq. 1)}$$

$$Q = 1.163 \cdot N \cdot m(55 - t_x) \quad \text{(Eq. 2)}$$

$$q_i = q' \cdot \dot{\eta}_k \cdot \dot{\eta} \cdot \dot{\eta} \quad \text{(Eq. 3)}$$

where Q – daily need of thermal energy, W · h, for hot water supply; q_i– heat flow rate in *i*-month, W/m^2; N – standard 55 ºC hot water consumption per capita daily, accepted according to the standards of design of internal water pipelines of buildings (Duffie and Beckman, 2006). For average standards of hot water it is recommended accepting minimum standard of 65 l for living houses/(per capita daily); m = 190 – number of residents; t_X– water temperature ºC, in cold water pipeline in summer period (t_X=15 ºC); q' – specific heat flow rate, W, falling on 1m^2 of inclined surface of solar collector (Duffie and Beckman, 2006); $\dot{\eta}_k$– performance of solar collector; $\dot{\eta}$ – coefficient considering the level of the atmosphere transparency, the value of which varies from 0.8 in industrial districts to 1 in resort areas; $\dot{\eta}_2$ – coefficient considering the losses of heat from solar collector to consumer, the size of which varies from 0.85 for large centralized systems of hot water supply to 0.98 for local hot-water heaters.

Based on correlation (2) we define daily need of thermal energy

$$Q = 1.163 \cdot 65 \cdot 190(55 - 15) = 574522 \text{ W·h/day}.$$

Using the values of Table 1 it is calculate the specific heat flow and the amount of the generated thermal energy.

Table 1 contains the following values: $\mathbf{t_0}$ **–** average atmospheric air temperature in an atmosphere in the daylight hours of the month; $\dot{\eta}_0$ – coefficient considering real cloud conditions; **φ –** latitude of the territory.

Performance of solar collector $\dot{\eta}_k$ is defined according to the Equation 4

$$\dot{\eta}_k = 0.82 - 0.007(t_k - t_0), \quad (4)$$

t_k = 50 °C – the heating temperature of the collector (Duffie and Beckman, 2006).

Let us define parameter qi for every month for Almaty in accordance with correlations (3), (4). Let us summarize the obtained data in Table 2. According to Equation 1 let us calculate the area of solar collectors:

$$F = 574522/3722 = 152\ m^2$$

Amount of W_m, kW-hours, of the thermal energy generated in the solar water heater, is calculated for each month as follows Equation 5:

$$W_m = 10^{-3} \cdot z \cdot q_i \cdot F \cdot \dot{\eta}_0 \cdot \dot{\eta}_3 \quad \text{(Eq. 5)}$$

where z – the number of days in a month; $\dot{\eta}_3$ – coefficient considering the losses, caused by non-stationary heat transfer at variable cloudiness ($\dot{\eta}_3$ =0,9). The design values of the thermal energy amount are summarized in Table 1.

RESULTS AND DISCUSSION:

In October, in the Republic of Kazakhstan, Almaty region at a solar intensity of about *600 W/m²* and environment temperature of about plus *12 ºC*, the sought temperature of heat carrier during 8…10 min. went down to plus *2 ºC*. Maximum gap between theoretical and experimental results was equal to *2.1 ºC*, which evidences adequacy of the theoretical model.

In January, at a solar intensity of about *400 W/m²* and environment temperature of about minus *4 ºC*, the heat carrier temperature during 8-10 min. went down to minus *7 ºC*. It has been stated that here the adequacy of the theoretical and experimental regularity is observed, where maximum divergence was equal to *2.2 ºC*.

In March, solar intensity $e_m \approx 730\ W/m^2$ started increasing the absorbing capacity of the solar collector, that's why the heat carrier temperature regarding the environment, did not decrease but increased from *3.7* to *9.6* ºC. At the same time, the design value of the temperature amounted to *11.3* ºC, so the divergence was equal to *1.7* ºC.

Thus, the experimental research confirmed the adequacy of the theoretical models. Using the obtained results and reference intensity values of the summarized solar radiation, environment temperature, and the calculations of the heat carrier dynamics in solar collector were performed for HMO "Almaty", at the specific capacity of solar collector absorption of 500 W/m².

The analysis shows that the difference between the temperature of environment and heat carrier (*Δt*) in the solar collector, as well as solar intensity, do not almost influence the velocity of the unit starting operation. The main factor, defining the process dynamics is the value of the specific capacity of the heat pump.

Using the theoretical and experimental results, as well as the obtained design values of the heat carries temperatures in the solar collector, the prediction calculations of possible daily thermal performance of the unit in conditions of the climate area of the republic were conducted, depending on the month and the area of solar collector, which is summarized at Figure 3.

The analysis shows that along with all the republican territory during the heating season the system is effective because useful energy exceeds the energy wasted on the compressor starting.

The regularity is of parabolic nature where minimum thermal performance is shown in the period of December – January.

In October and in March thermal performance is the highest, because the atmospheric air temperature increases together with solar intensity, while the corresponding increase in the heat carrier temperature in the adsorption tank also leads to fast growth of heat pump performance coefficient.

The most favorable conditions for the system work are provided in the southern region (Almaty HMO), where thermal performance at the same conditions is by 30-32% higher compared with the north (Kostanay).

Thermal performance increases with the increase in the area of solar collectors due to the decrease in the depth of the cooled heat carrier and a corresponding increase in the transformation coefficient HT (φ_i). However, at the same time, the exploitation costs proportionally increase. The optimal area may be set through comparative technical and economic calculations.

For comparative analysis and objective substantiation of the unit parameters, it is important to define the structure of the usefully generated energy flow, which varies depending on the month.

The calculation results of seasonal thermal performance structural characteristics, conducted for the conditions of the Almaty and Kostanay regions, are presented at Figure 4, where the diagrams show the content of the shares in percentage of solar energy plus heat from the atmospheric air (Q_{se}+Q_{TAV}) and the heat from the compressor operation (Q_{ht}).

The analysis shows that the total share of the absorbed solar energy and heat from the atmospheric air in the structure of thermal performance during the entire heating season at the area of the adsorbers equal to $S_{AD}>1m^2$ exceeds 50%, which evidences total energy efficiency of the system.

The unfavorable months are December-January, when the share of the absorbed energy in the northern zone goes down to 52-54%, while in the south it goes down to 62-64%. The favorable months are October and March, when in the northern zone the share of the absorbed energy increases up to 82-86%, while in the south it increases up to 86-87%.

The regularity of the heat share from the atmospheric air during the entire heating season both for the northern and southern zone has an explicit maximum at the adsorbers' area from 3 to 4 m^2.

The share of solar energy in the heating season is significantly lower compared to the heat from the atmospheric air. In December and January, the difference is up to 2.0 times, in October it is up to 5.0 times, while in March due to the increase in solar intensity it goes down to 1.5 times.

The authors have calculated and built the dependence of seasonal thermal performance on the area of solar collectors (Figure 4). The obtained regularities show that thermal performance grows up to 4m^2. Further increase in the area does not provide the expected results.

The results of seasonal thermal performance served the source data at technical and economic calculations of energy-saving technology, where the solar unit is a component.

The results of calculating the structure of seasonal thermal performance of the unit, conducted using the example of Almaty and Kostanay regions, are presented in Figure 4. The diagrams show that in the seasonal thermal performance of the unit, over 80% of the energy gets from the sun and heat of the atmospheric air, while up to 20% of the energy gets from the heat generated at the compressor operation.

One of the main characteristics of the solar unit as any other energy unit is its performance, which may be calculated according to the known methods or determined experimentally by the correlation of usefully generated heat energy and total solar intensity. However, the performance calculation formula requires being checked, because the developed collector contains a new constructive element (Figure 5).

The theoretical calculation of the average daily performance of the TC shows that, Equation 6:

$$\eta_{theor.} = \frac{6655}{7579 \cdot 1{,}68} = 0{,}49 \qquad \text{(Eq. 6)}$$

CONCLUSIONS:

This paper deals with the performance of solar water heating unit at petrol filling station. In the paper, the comparison of the effectiveness of water heating installations in different regions, taking into account climatic difference, is performed. In north regions, the effectiveness of water-heating systems constitutes about 40%. In Great Britain, this system provided the need in the hot water on 57%. The object of the research performed is the gas station total area of 60m2. It is stated, that the usage of such an installation is effective on the whole territory of the republic, in the heating period. It is characterized by that the useful energy exceed the energy, which was necessary for the compressor drive.

The authors calculated the experimental

and theoretical performance of a solar water heating unit, which shows reasonable precision within 7%, evidencing reliability of the computing formulas. It is explained by successful constructive solution, application of the mirror reflectors and cellular structure of transparent coating, which confirms the initial hypothesis. The results of the calculation of the seasonal thermal performance unit are the following: in seasonal thermal performance unit, over 80% of energy gets from the solar energy and the heat of the atmospheric air, while up to 20% of energy gets from the heat generated during the compressor operation.

It was analyzed, that unfavorable month for installation work considered to be December and January. The best productivity of water heating installation is observed in March and October. Since in this period, the rate of the swept up energy on the northern territory of the country, increases to 82-86%, and in the south up to 86-87%. Results of seasonable heating capacity are considered as an original data with the technical-economical counting of energy-efficient technology, in which two loops of solar systems is the constitute part. The authors presented the dependence of the hour-calculation ECF of the solar collector on the time of heating. ECF is the one of the main characteristics of the solar system.

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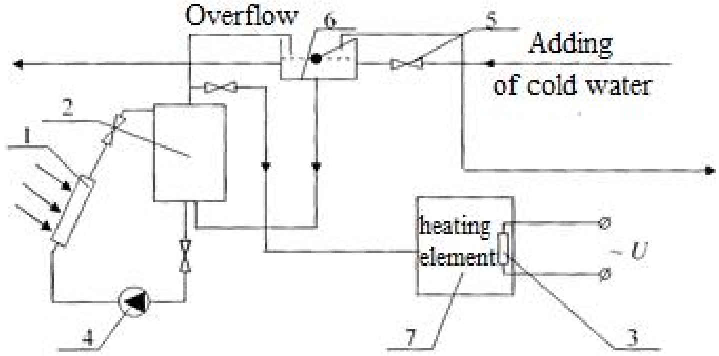

Figure 1. Scheme of Solar Water Heating System: 1 – solar collector; 2 – storage tank; 3 – heating element; 4 – circulating water pump; 5 – globe valve; 6 – expansion tank; 7 – reserve tank

Figure 2. General View of Solar Water Heating Unit

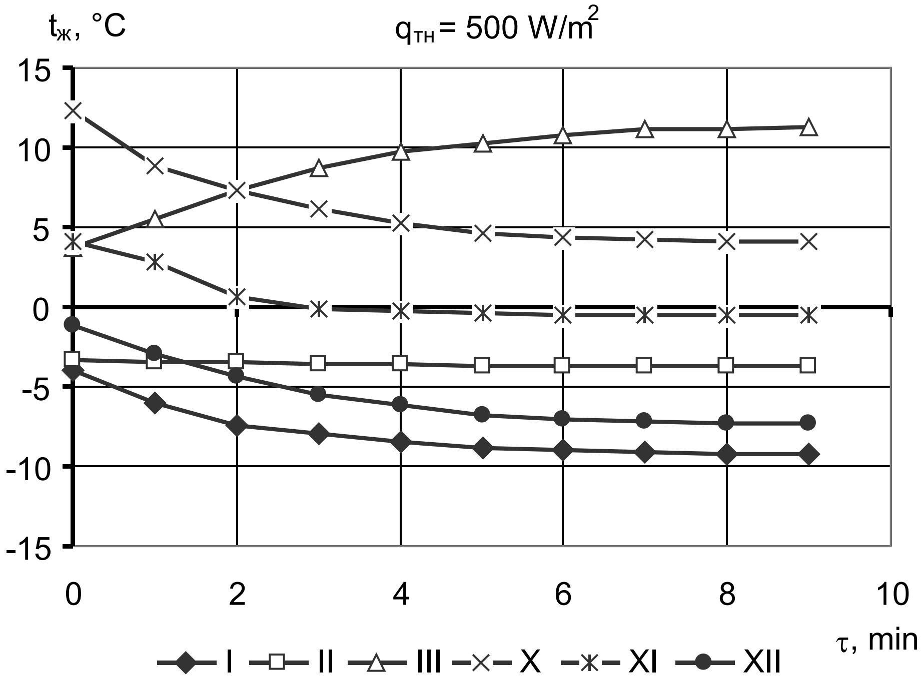

Figure 3. Dynamics of Heat Carrier Temperatures of Solar Collector at the Compressor Capacity of 500 W/m^2

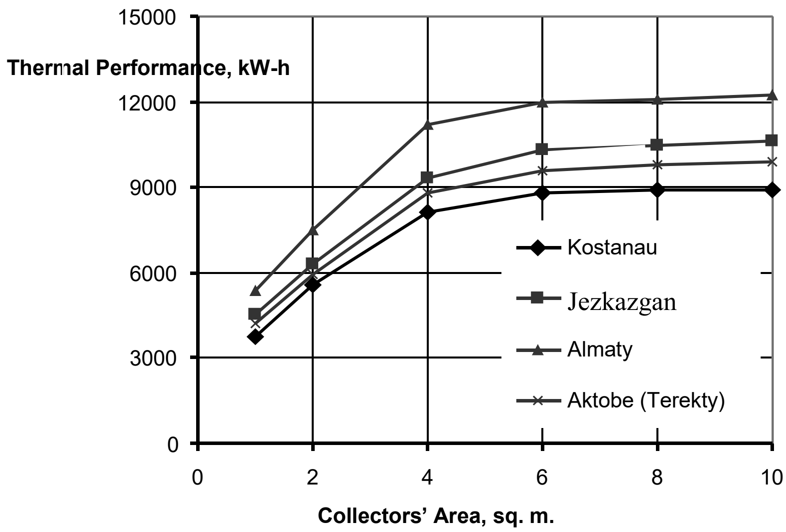

Figure 4. Seasonal Thermal Performance of DGU depending on the Collectors' Area

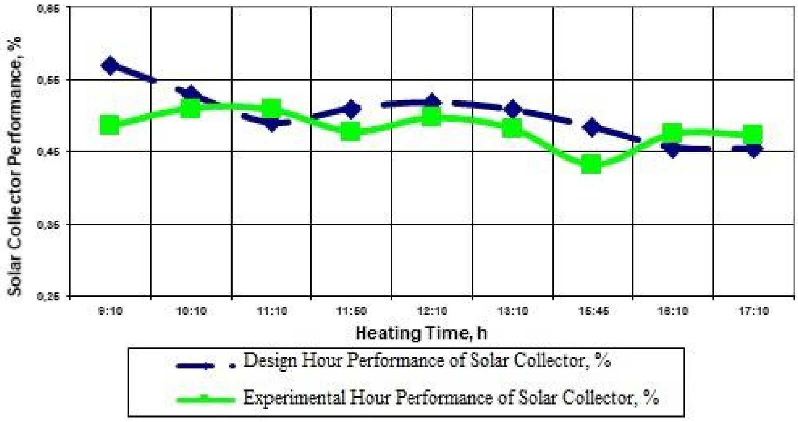

Figure 5. Dependence of Calculation of Solar Collector Hour Performance on the Heating Time

Table 1. *Temperature Data and Values of the Coefficient for the Cities of Kazakhstan*

City		**Months**											
		April		May		June		July		August		September	
		t_0	**η_0**	**t_0**	**η_0**	**t_0**	**η_0**	**t_0**	**η_0**	**t_0**	**η_0**	**t_0**	**η_0**
Kostanay	4	16	0.71	22	0.77	25	0.86	29	0.86	28	0.88	24	0.85
Jezkazgan	4	16	0.69	21	0.71	25	0.78	28	0.86	27	0.86	22	0.81
Aktobe	6	15	0.67	22	0.76	26	0.8	29	0.83	28	0.86	23	0.75
Almaty	6	14	0.72	21	0.77	25	0.82	28	0.86	27	0.86	23	0.78

Table 2. *Design Values*

Odessa	April	May	June	July	August	September
$\dot{\eta}_k$	0.568	0.617	0.645	0.666	0.659	0.631
q_i, W/m^2	3722	4346	4468	4556	4320	4136
W_m ,W-h	11 175	14 400	15 271	16 600	15 980	14 020