Досліджено використання сонячної енергії як потенційного альтернативного джерела для забезпечення теплопостачанням енергоефективних будинків. Проведено попередній теоретичний аналіз енергетичних показників комбінованої системи теплопостачання за використання геліовікна як частини огородження енергоефективного будинку. З метою підвищення ефективності роботи досліджуваної установки відносно існуючих сонячних колекторів та покращення її в конструктивному аспекті, було розраховано стратифікацію теплоносія в баку-акумуляторі комбінованої системи теплопостачання із геліовікном.

Дослідження ефективності роботи експериментальної установки в системі для використання сонячної енергії проводили в режимі руху циркуляції та гравітації теплоносія за інтенсивності випромінювання імітатора сонячної енергії на систему 600 Вт/м² та 900 Вт/м². Як теплоносій використовували воду.

Проаналізовано зміну температури теплоносія в сонячному колекторі та в баку-акумуляторі запропонованої комбінованої системи сонячного теплопостачання із геліовікном як частини зовнішнього захисту енергоефективного будинку.

Встановлено, що температура теплоносія в режимі циркуляції досягала 26,5 °С. Крім цього, наведено порівняльні натурні, лабораторні та теоретичні розрахунки усередненої температури теплоносія в баку-акумуляторі в режимі руху гравітації теплоносія за різних умов.

Було розраховано коефіцієнт корисної дії експериментальної установки. Описано динаміку зміни коефіцієнта корисної дії системи сонячного теплопостачання із геліовікном. У режимі руху циркуляції теплоносія за накопиченням теплової енергії в баку-акумуляторі, залежно від часу нагріву, ККД становив ≈55 %. В режимі руху гравітації теплоносія було обраховано коефіцієнт корисної дії лише конструкції геліовікна, що становив 53 %

Ключові слова: геліовікно, сонячна енергетика, енергетичний баланс, режим гравітації/циркуляції, інтенсивність випромінювання

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THEORETICAL AND EXPERIMENTAL ANALYSIS OF SOLAR ENCLOSURE AS PART OF ENERGY-EFFICIENT HOUSE

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1. Introduction

In the twenty-first century, society is given a choice to install a wide variety of heat supply and ventilation systems that ensure optimal micro-climatic conditions in the premises. However, their design and security must meet the standards, which are the guarantee of a comfortable and environmentally friendly existence of the future.

Catastrophic climatic changes require rational and well-grounded solutions of a human when it comes to the environment and everyday life. For example, the constructed houses for a long operation period require improvement and modernization in terms of design and power. Or, the improvement of the use of energy-efficient solutions at the initial stages of construction or reconstruction in order to save the climate in the future and the life quality of future generations.

To simplify the calculation of the amount of energy consumed by a house in Europe, the classification of buildings relative to their power dependence was developed:

– in the structures/buildings, which were built before the 1970s, the energy dependence was determined to be ≈300 kW·h/(m²·year) for heating by various kinds of fuels;

– in the structures/buildings, which were built over the period of 1970–2000, the energy dependence was supposed to be up to 150 kW·h/(m²·year) for heating by different fuels;

– in the structures /buildings of low energy dependence – not more than 60 kW·h/(m²·year).

Construction of lower-standard houses for people has not been allowed in Europe since 2002.

A recognized quality brand for new and reconstructed buildings of low energy consumption is «Minergie» [1]. This brand is supported by the Swiss Confederation, the Swiss cantons, and the Principality of Liechtenstein. «Minergie» brand is registered in Switzerland and around the world and, therefore, is protected from the unlicensed use. Brand «Minergie» can only be used for buildings, services and components that actually meet the standard of «Minergie». The quality system proposed by them is intended for new and renovated buildings with low power consumption.

Buildings with low power consumption are classified into 4 types:

1. A passive house – power consumption by such buildings should not be not more than 15 kW·h/(m^2·year).

2. A zero energy house – power consumption by such buildings should be not more than 0 kW·h/(m^2·year). It is worth noting that the architecture of such buildings can comply with the same standard as a passive house, however, in terms of engineering, they must be equipped so that they could consume exclusively only the energy that they will generate [2].

3. A house of low energy consumption in practice has to use active and passive solar structures, construction techniques and the components to decrease power consumption [3].

4. A house plus energy is a building of such type that with the use of the energy-efficient equipment (solar panels, collectors, heat pumps, heat exchangers, recuperators, soil heat exchangers, etc.) would produce more power than it would consume [2].

Thus, the house plus energy is a less energy-dependent structure than a zero-energy house and passive houses [4].

Different information sources offer several definitions of the term «a passive house». Passive or energy efficient house is the building, the main feature of which is low power consumption ≈10 % of specific power that the house consumes per unit of volume, consumed by most modern buildings.

Another, but no less important definition of a passive house is that it is a building standard. That is, this house has to be marginally energy efficient, to create comfortable conditions for residence and have the characteristics of an economically attractive building [5].

It is noted that a passive house is the building, in which it is possible to achieve a comfortable microclimate: in winter – without a separate heating system (or using the low-power compact heating system), in summer – without air conditioning systems [6, 7].

It is worth paying attention that the deterrent factor in the construction of «energy efficient houses» is the high cost of their construction in Ukraine. «Energy efficient houses» are by about 8–10 % more expensive than conventional houses. It is also necessary to take not only the cost of a house, but also the cost of its further operation. Operation costs of an «energy efficient house» are by times less than the operation costs of a conventional house, in addition, it is worth considering the constant increase in gas, water and electricity prices, and as a result the costs of housing and communal services as a whole [8].

As a result, alternative (renewable) power engineering gains increasing popularity annually against the background of an acute problem of reducing the level of certain natural resources.

The application/scientific foundations of solar energy in construction were described in paper [9]. In addition, the paper contains practical advice/recommendations to use solar energy in energy-efficient buildings.

A rather effective method for heating premises of a passive house is the use of wind solar heating systems with the phase transition heat battery based on Glauber's salt [10]. Such systems can be used for both additional and fully-fledged heating and heat battery makes it possible to maintain the required air temperature in the premises during cloudiness or at night.

The fundamental question of using such systems remains the ecological and economic aspects. A promising solution is to use a combined solar heat supply system with a solar window as part of the enclosure of an energy-efficient house. This system can be used as a secondary source of heat for heat and gas supply and ventilation in the house.

The range of themes of the above literary sources proves the relevance and versatility of this direction of research as «energy efficient/passive building».

2. Literature review and problem statement

Solar energy is widely recognized as one of the most important renewable energy sources thanks to the uniform distribution, security and serving as an energy source for others. Over the past decade, the global solar heat power has been growing rapidly, and now it is widely used around the world to ensure the microclimate of premises and comfortable conditions of human existence [11].

In paper [12], a comparative analysis of the operation efficiency of solar power engineering in the leading countries of the world was performed. The information was presented that the highest indicators of the use of solar energy were recorded in Europe, specifically, in Spain, owing to warm climate and is 31.4 %, whereas for Ukraine this value is equal to 12.3 %. In addition, the expediency of creation of large solar power plants in the regions with high solar activity and of small sites for the temperate climate was substantiated.

One of the most popular devices for converting solar energy into thermal energy is a flat solar collector, which has a number of advantages, such as reliability, simplicity, relatively low cost of design and long service life. However, during their designing, there arises the challenge, specifically, an increase in operation efficiency in the periods with the low potential of solar radiation and in the cold period season [13, 14].

According to authors of [15], the efficiency of the proposed air system of heating, in particular, the share of solar heating, is 63 %. For heating, it is possible to achieve 34 kWh/m^2 when the area of solar collectors makes up 1/6 of the area of a house and at the latent heat of thermal energy storage with the change of the phase of materials that have the weight of 25 kg/m^2 of the total area of the house. However, these studies are aimed at the system of solar heat supply with the air heat carrier, rather than liquid one.

The authors of paper [16] presented the circuit of the regulation of operation and the strategy of management of a solar heating system with the seasonal storage of heat for the swimming pool (SHS-SWPHS). The research methodology is aimed at the compliance of the heating source with the consumer load. Compared with the traditional heating system, heat storage temperature is constantly decreasing during the heating period. First of all, the research proposed the additional modes of heating a heat pump of indirect heating, direct heating and a water source depending on different temperature levels. In addition to the facts described above, the possibility of applying the useful area of the swimming pool for heating from solar energy was described. That is, the authors offer an improved circuit of operation regulation and the application of the useful area, in this case, of the swimming pool, but not of a house.

Paper [17] considered a new solution for the device that uses solar energy in heating systems – solar collector – energy efficient enclosure. This device is intended to increase the coefficient of utilization of solar energy and an increase in heat resistance of the basic enclosure of a building. The main purpose of this enclosure is a decrease in heat with the help of the device, at the same time increasing technological efficiency of its production and simplicity of mounting.

Each proposed design has its advantages and disadvantages, depending on the hydraulic characteristic of the system; however, the studies do not present either theoretical or experimental data regarding the state of the heat carrier (temperature, etc.) in the accumulating capacity. Due to this, to increase the effectiveness of existing solar plants, it is necessary to analyze possible problematic areas of the proposed setup. In particular, a number of solar power plants have problems related to effective storage and using the obtained heat from a solar collector.

3. The aim and objectives of the study

The aim of this research is to substantiate the possibilities of using solar power as an alternative source without transmitting power over a considerable distance, that is, in close proximity to the consumer.

To accomplish the aim, the following tasks have been set:

– to simulate the stratification of the heat carrier in the tank-accumulator of the proposed experimental solar heat supply system with a solar window;

– to analyze the main operation parameters of the experimental setup of solar enclosure as a part of an energy efficient house;

– to compare the change in temperature of the heat carrier in theoretical calculations in relation to data from laboratory and field studies of solar window.

4. Materials and methods of research

4. 1. Methods for determining energy indicators for the combined heat supply system using a solar window

The procedure of identifying the problematic areas and the reserves for improving energy efficiency of the setup consists of three stages:

– collection of the source data;

– taking the readings of power accounting devices;

– data processing [18].

According to the standard DSTU B EN 15251:2011, there are two main types of the methods for calculation of energy consumption for heating and cooling [19]:

– quasi-stationary methods, using which the thermal balance is calculated for quite a long time (usually one month or a whole season), which makes it possible to take into consideration the dynamic effects by the empirically determined coefficient of using the gains and/or losses;

– dynamic methods, by using which a thermal balance is calculated for short periods of time (typically one hour). However, it takes into account the amount of heat that is accumulated or released depending on the dimensions of a house.

The authors of article [20] believe that the most acceptable for Ukraine is the recommendation on the use of the quasi-stationary monthly method with the simplified approach of the calculation of energy efficiency for all types of houses.

In DSTU B EN ISO 13790, three other types of the method were proposed:

– completely determined monthly quasi-stationary calculation method (where the seasonal method is taken into consideration);

– completely determined simplified dynamic hourly calculation method;

– calculation procedures for detailed (hourly) dynamic methods of simulation.

Due to this, the implementation of effective and structurally simple systems of solar heat supply to energy efficient houses remains a relevant and not completely studied problem.

Energy balance of the system of solar heat supply can be written down in the form of formula (1):

$$Q_T - L + E = \Delta U, \tag{1}$$

where Q_T is the monthly thermal productivity of a solar plant; L is the sum of monthly loads of heating and hot water supply; E is the total amount of electricity obtained within a month from a doubling source; ΔU is the change in the amount of energy in the accumulating plant.

Given the dimensions of batteries that are commonly used in the systems of solar heat supply systems, where difference ΔU is small compared with Q_T, L and E and can be accepted to be equal to zero, equation (1) can be written down in the form:

$$f = (L - E)/L = Q_T/L, \tag{2}$$

where f is the part of total monthly load ensured due to solar energy.

Equation (2) cannot be used by itself to calculate f, since magnitude Q_T is a heavy function of descending radiation, ambient temperature, and thermal loads. However, consideration of the parameters, on which Q_T depends, suggests that the replacement coefficient f can be empirically associated with two dimensionless complexes (3) and (4):

$$X = AF'_R U_L (T_{ref} - \bar{T}_\alpha) \Delta t / L, \tag{3}$$

$$Y = AF'_R (\overline{\tau\alpha}) \bar{H}_T N / L, \tag{4}$$

where A is the area of a solar collector, m^2; F'_R is the effective coefficient of heat withdrawal, that takes into account the heat exchanger; U_L is the complete coefficient of thermal losses of the collector, W/(m^2·°C); Δt is the number of seconds per month; T_{ref} is the basic temperature, accepted by 100 °C; $\bar{T}_\alpha$ is the average monthly ambient temperature, °C; L is the complete monthly thermal load, J; $\bar{H}_T$ is the average daily arrival of total solar radiation at the sloping surface of the collector within a month, J/M^2; N the number of says in a month; $\overline{\tau\alpha}$ is the average reduced absorbing capacity per month.

Dimensionless complexes X and Y have physical orientation:

X – monthly thermal losses of the collector under conditions of the basic temperature in relation to the complete monthly thermal load;

Y – amount of energy that is absorbed by the plate of the collector within a month relative to the total monthly load.

For convenience of calculation, expressions X and Y can be rewritten in a modified form (5) and (6).

$$X = F_R U_L (F_R'/F_R)(T_{ref} - \bar{T}_\alpha)\Delta t (A/L), \quad (5)$$

$$Y = F_R (\tau\alpha)_n (F_R'/F_R)[(\overline{\tau\alpha})/(\tau\alpha)_n]\bar{H}_T N (A/L). \quad (6)$$

Magnitude $F_R U_L$ and $F_R(\tau\alpha)_n$ are determined by the results of testing of the collector. Values $\underline{T}_\alpha$ are known for different geographical points. Magnitude $\bar{H}_T$ is determined by values $\bar{H}$ and $\bar{R}$ using ratios and tables. The area of collector A is assigned in the process of calculations. Thus, all magnitudes in the last two formulas can be determined if the source data are known.

4. 2. Theoretical analysis of stratification of a heat carrier in the tank-accumulator as part of the system of solar heat supply with a solar window

Water tanks-accumulators can operate at a significant degree of stratification under conditions of unequal temperature of the heat carrier along the height of the tank. In this case, it is possible to describe the mathematical model of the real situation that occurs under conditions when water that enters the tank has low velocity and occupies its own level taking into consideration its density.

For the two-section tanks, energy balances can be written down as follows:

– for the upper (layer 1) (7):

$$\frac{dt_{\tan k_1}}{d\tau} = \frac{1}{M_1 \cdot c_w} \cdot \begin{bmatrix} G_{last} \cdot c_w \cdot (t_{last}(\tau) - t_{\tan k_1}(\tau)) + \\ + G_{con} \cdot c_w \cdot (t_{\tan k_2}(\tau) - t_{\tan k_1}(\tau)) - \\ - 3.6 \cdot F_{\tan k_1} \cdot K_{\tan k_1} \cdot (t_{\tan k_1}(\tau) - t_{amb}) \end{bmatrix}; \quad (7)$$

– for the lower (layer 2) (8):

$$\frac{dt_{\tan k_2}}{d\tau} = \frac{1}{M_2 \cdot c_w} \cdot \begin{bmatrix} G_{last} \cdot c_w \cdot (t_{\tan k_1}(\tau) - t_{\tan k_2}(\tau)) + \\ + G_{con} \cdot c_w \cdot (t_A(\tau) - t_{\tan k_2}(\tau)) - \\ - 3.6 \cdot F_{\tan k_2} \cdot K_{\tan k_2} \cdot (t_{\tan k_2}(\tau) - t_{amb}) \end{bmatrix}. \quad (8)$$

As a result of numerical modeling, we obtained the distribution of temperatures, shown in Fig. 1. In particular, the temperature that was achieved in the upper layer of the two-section tank of the battery amounted to 45 °C.

For the three-section tank, energy balances can be written down as follows:

– for the upper (layer 1) (9):

$$\frac{dt_{\tan k_1}}{d\tau} = \frac{1}{M_1 \cdot c_w} \cdot \begin{bmatrix} G_{last} \cdot c_w \cdot (t_{last}(\tau) - t_{\tan k_1}(\tau)) + \\ + G_{con} \cdot c_w \cdot (t_{\tan k_2}(\tau) - t_{\tan k_1}(\tau)) - \\ - 3.6 \cdot F_{\tan k_1} \cdot K_{\tan k_1} \cdot (t_{\tan k_1}(\tau) - t_{amb}) \end{bmatrix}; \quad (9)$$

– for layer 2(10):

$$\frac{dt_{\tan k_2}}{d\tau} = \frac{1}{M_2 \cdot c_w} \cdot \begin{bmatrix} G_{last} \cdot c_w \cdot (t_{\tan k_2}(\tau) - t_{\tan k_3}(\tau)) + \\ + G_{con} \cdot c_w \cdot (t_{\tan k_3}(\tau) - t_{\tan k_2}(\tau)) - \\ - 3.6 \cdot F_{\tan k_2} \cdot K_{\tan k_2} \cdot (t_{\tan k_2}(\tau) - t_{amb}) \end{bmatrix}; \quad (10)$$

– for the lower layer (layer 3) (11):

$$\frac{dt_{\tan k_3}}{d\tau} = \frac{1}{M_3 \cdot c_w} \cdot \begin{bmatrix} G_{last} \cdot c_w \cdot (t_{\tan k_2}(\tau) - t_{\tan k_3}(\tau)) + \\ + G_{con} \cdot c_w \cdot (t_c(\tau) - t_{\tan k_3}(\tau)) - \\ - 3.6 \cdot F_{\tan k_3} \cdot K_{\tan k_3} \cdot (t_{\tan k_3}(\tau) - t_{amb}) \end{bmatrix}. \quad (11)$$

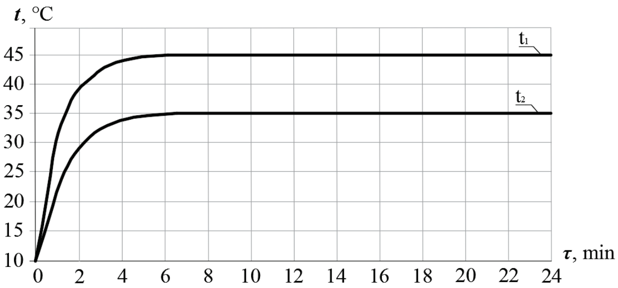

Fig. 1. Distribution of temperatures in the two-section tank-accumulator over 24 hours

As a result of numerical modeling, we obtained the temperature distribution shown in Fig. 2.

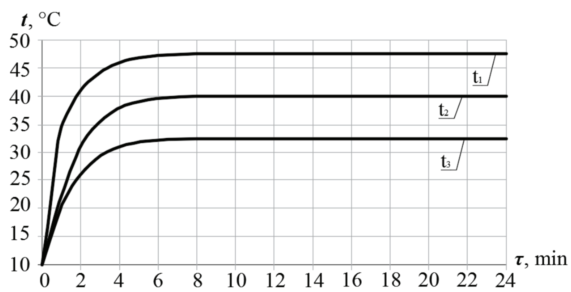

Fig. 2. Distribution of temperatures in the three-section tank-accumulator over 24 hours

For the nine-section tank, energy balances can be written down as follows:

– for the upper (layer 1) (12):

$$\frac{dt_{\tan k_1}}{d\tau} = \frac{1}{M_1 \cdot c_w} \cdot \begin{bmatrix} G_{last} \cdot c_w \cdot (t_{last}(\tau) - t_{\tan k_1}(\tau)) + \\ + G_{con} \cdot c_w \cdot (t_{\tan k_2}(\tau) - t_{\tan k_1}(\tau)) - \\ - 3.6 \cdot F_{\tan k_1} \cdot K_{\tan k_1} \cdot (t_{\tan k_1}(\tau) - t_{amb}) \end{bmatrix}; \quad (12)$$

– for layer 2 (13):

$$\frac{dt_{\tan k_2}}{d\tau} = \frac{1}{M_2 \cdot c_w} \cdot \begin{bmatrix} G_{last} \cdot c_w \cdot (t_{\tan k_2}(\tau) - t_{\tan k_3}(\tau)) + \\ + G_{con} \cdot c_w \cdot (t_{\tan k_3}(\tau) - t_{\tan k_2}(\tau)) - \\ - 3.6 \cdot F_{\tan k_2} \cdot K_{\tan k_2} \cdot (t_{\tan k_2}(\tau) - t_{amb}) \end{bmatrix}; \quad (13)$$

– for layer 3 (14):

$$\frac{dt_{\tan k_3}}{d\tau} = \frac{1}{M_3 \cdot c_w} \cdot \begin{bmatrix} G_{last} \cdot c_w \cdot (t_{\tan k_2}(\tau) - t_{\tan k_3}(\tau)) + \\ + G_{con} \cdot c_w \cdot (t_{\tan k_4}(\tau) - t_{\tan k_3}(\tau)) - \\ - 3.6 \cdot F_{\tan k_3} \cdot K_{\tan k_3} \cdot (t_{\tan k_3}(\tau) - t_{amb}) \end{bmatrix}; \quad (14)$$

– for layer 4 (15):

$$\frac{dt_{\tan k_4}}{d\tau}=\frac{1}{M_4\cdot c_w}\cdot\begin{bmatrix}G_{last}\cdot c_w\cdot(t_{\tan k_3}(\tau)-t_{\tan k_4}(\tau))+\\+G_{con}\cdot c_w\cdot(t_{\tan k_5}(\tau)-t_{\tan k_4}(\tau))-\\-3.6\cdot F_{\tan k_4}\cdot K_{\tan k_4}\cdot(t_{\tan k_4}(\tau)-t_{amb})\end{bmatrix};\ (15)$$

– for layer 5 (16):

$$\frac{dt_{\tan k_5}}{d\tau}=\frac{1}{M_5\cdot c_w}\cdot\begin{bmatrix}G_{last}\cdot c_w\cdot(t_{\tan k_4}(\tau)-t_{\tan k_5}(\tau))+\\+G_{con}\cdot c_w\cdot(t_{\tan k_6}(\tau)-t_{\tan k_5}(\tau))-\\-3.6\cdot F_{\tan k_5}\cdot K_{\tan k_5}\cdot(t_{\tan k_5}(\tau)-t_{amb})\end{bmatrix};\ (16)$$

– for layer 6 (17):

$$\frac{dt_{\tan k_6}}{d\tau}=\frac{1}{M_6\cdot c_w}\cdot\begin{bmatrix}G_{last}\cdot c_w\cdot(t_{\tan k_5}(\tau)-t_{\tan k_6}(\tau))+\\+G_{con}\cdot c_w\cdot(t_{\tan k_7}(\tau)-t_{\tan k_6}(\tau))-\\-3.6\cdot F_{\tan k_6}\cdot K_{\tan k_6}\cdot(t_{\tan k_6}(\tau)-t_{amb})\end{bmatrix};\ (17)$$

– for layer 7 (18):

$$\frac{dt_{\tan k_7}}{d\tau}=\frac{1}{M_7\cdot c_w}\cdot\begin{bmatrix}G_{last}\cdot c_w\cdot(t_{\tan k_6}(\tau)-t_{\tan k_7}(\tau))+\\+G_{con}\cdot c_w\cdot(t_{\tan k_8}(\tau)-t_{\tan k_7}(\tau))-\\-3.6\cdot F_{\tan k_7}\cdot K_{\tan k_7}\cdot(t_{\tan k_7}(\tau)-t_{amb})\end{bmatrix};\ (18)$$

– for layer 8 (19):

$$\frac{dt_{\tan k_8}}{d\tau}=\frac{1}{M_8\cdot c_w}\cdot\begin{bmatrix}G_{last}\cdot c_w\cdot(t_{\tan k_7}(\tau)-t_{\tan k_8}(\tau))+\\+G_{con}\cdot c_w\cdot(t_{\tan k_9}(\tau)-t_{\tan k_8}(\tau))-\\-3.6\cdot F_{\tan k_8}\cdot K_{\tan k_8}\cdot(t_{\tan k_8}(\tau)-t_{amb})\end{bmatrix};\ (19)$$

– for the lower layer (layer 9) (20):

$$\frac{dt_{\tan k_9}}{d\tau}=\frac{1}{M_9\cdot c_w}\cdot\begin{bmatrix}G_{last}\cdot c_w\cdot(t_{\tan k_8}(\tau)-t_{\tan k_9}(\tau))+\\+G_{con}\cdot c_w\cdot(t_c(\tau)-t_{\tan k_9}(\tau))-\\-3.6\cdot F_{\tan k_9}\cdot K_{\tan k_9}\cdot(t_{\tan k_9}(\tau)-t_{amb})\end{bmatrix}.\ (20)$$

As a result of numerical modeling, we obtained the distribution of temperatures in the tank-accumulator (Fig. 3). The temperature in the upper layer of the tank-accumulator reaches the value of more than 50 °C, while in the lower layer, it is 27 °C. That makes it possible to determine the importance of taking into consideration the stratification of the heat carrier in the proposed studied setup for further research.

It is worth noting that while analyzing the results of the modeled calculations of temperature, it was found that in the first layer of the tank-accumulator, the temperature reaches the value of more than 50 °C, that is, the sufficient temperature for a low-potential heat source. This also makes it possible to argue about the need for better insulation of the storing capacity in the layer, which is closer to the pipeline with the supplying (hot) heat carrier. This information should be taken into consideration in the studied experimental setup with the energy saving aim.

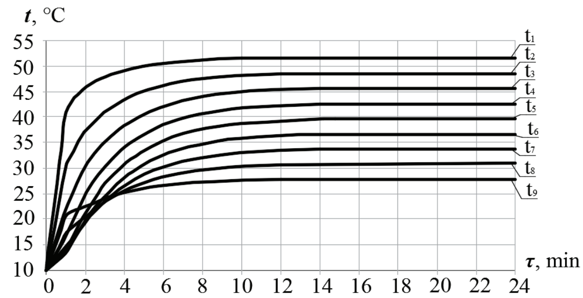

Fig. 3. Distribution of temperatures in the nine-section tank-accumulator over 24 hours

4. 3. Structural diagram of the experimental setup of solar enclosure

Structural diagram of the system of the solar heat supply with a solar window as solar enclosure of the energy-efficient house is shown in Fig. 4. In the experimental setup of the system of solar heat supply with a solar window, all pipelines and the tank-accumulator were thermally insulated to reduce heat losses.

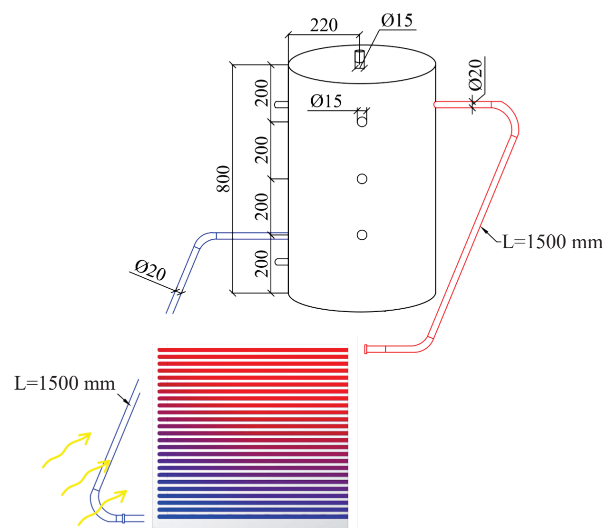

Fig. 4. Schematic of experimental setup of the system of solar heat supply with a solar window

According to the methodology and the research plan, the operation to register the obtained indicators was started. The heat carrier flow was measured by flow meters and the rotameter of RM-0.016Zh type, which were mounted on the reverse line of the motion of the cooled heat carrier before entering the solar collector. Before conducting the experiments, the rotameter was calibrated by the volumetric method. Experimental research was conducted by applying the hypothetical assumptions and simplifications.

The following conditions were chosen: density of heat flow was accepted to be equal around the entire area of the solar window plane; the solar window structure was not shaded; the influence of solar energy reflected from the surrounding objects was not taken into account; confident probability of the results of the experiment $\alpha=0.95$ was accepted.

Additional working point was to control that the experiment should not be influenced by other factors (solar energy through the window, smooth surfaces, shading of solar collector, etc.).

The system of solar heat supply with the solar window can operate under the mode of circulation and gravitational motion of the heat carrier.

The operation principle is the following. The water from the tank-accumulator arrives in the pipe of the heated circulation circuit. By the principle of natural convection, the water returns to the tank-accumulator under the mode of gravitational motion of the heat carrier and as a result of heating by the solar energy simulator. Then it is possible to take the heat carrier through the pipe to the consumer.

The intensity of the flow of energy emitted by the solar energy simulator was measured by the actinometer AT-50 (Russia).

The temperature of the heat carrier at the inlet and at the outlet of the solar window and in the tank-accumulator was measured by resistance transducers 50M (Ukraine), operating with the regulator-meter of RT-0102 type (Ukraine). The temperature of the ambient air and its velocity was measured with the thermoelectroanemometer TESTO 405-V1 (Germany).

5. Results of experimental research into the system of solar heat supply with a solar window as part of solar enclosure at an energy-efficient house

The research findings were based on the experimental data of research into the heat carrier of the proposed system of solar heat supply with a solar window.

The temperature of the heat carrier in the proposed system of solar heat supply with the solar window at the intensity of the solar power of 600 W/m² at the outlet of the solar window reached 26.5 °C (Fig. 5).

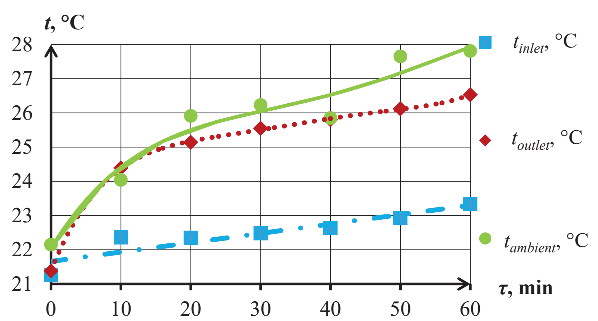

Fig. 5. Temperature of the heat carrier at the inlet t_{inlet}, °C and at the outlet t_{outlet}, °C of the solar window and the ambient temperature $t_{ambient}$, °C throughout the experiment under the mode of heat carrier circulation under conditions of radiation of the solar energy simulator on the setup at the intensity of 600 W/m²

The temperature in the tank-accumulator at the intensity of radiation of the solar energy simulator of 600 W/m² reached 25.5 °C (Fig. 6).

It is necessary to pay attention that in to determine the efficiency of the system of solar heat supply with a solar window in general (Fig. 7), first, it was necessary to determine the thermal energy that was accumulated in the tank-accumulator from formula (21):

$$Q_{\tan k} = m \cdot c \cdot (t_b - t_e), \quad (21)$$

where m is the weight of the heat carrier in the tank-accumulator, kg; c is the average specific thermal capacity of the heat carrier (at constant pressure) at arithmetic mean temperature of heat carriers, J/(kg·K); t_b, t_e are the temperature of the heat carrier in the tank-accumulator at the beginning and at the end of the experiment, respectively, K.

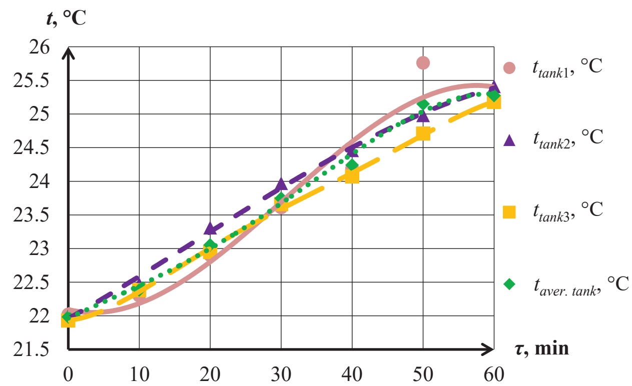

Fig. 6. A change of the temperature of the heat carrier in three layers of tank-accumulator $t_{tank1...3}$, °C and averaged temperature of the tank-accumulator $t_{aver.tank.}$, °C depending on heating duration under the mode of motion of heat carrier circulation at radiation of the solar energy simulator onto the setup at intensity of 600 W/m²

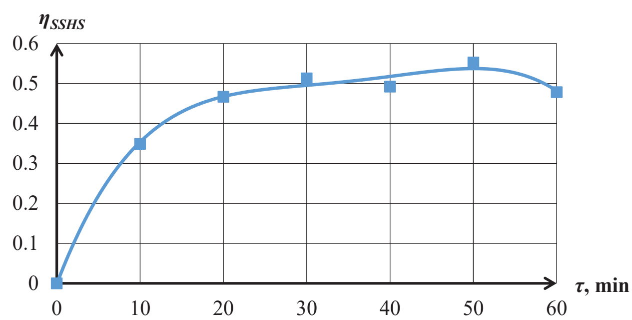

Fig. 7. Dynamics of change in efficiency of the system of solar heat supply with the solar window η_{SSHS} in general (at accumulation of thermal energy in the tank-accumulator) depending on duration of heating under the mode of heat carrier circulation motion at the radiation of the solar energy simulator onto the setup at the intensity of 600 W/m²

A change in the heat carrier temperature in the tank-accumulator under the mode of gravity motion of the heat carrier is shown in Fig. 8.

An important aspect in studying the system of solar heat supply is to analyze the changes in efficiency for mounting the solar window as a solar collector under the mode of gravity motion of the heat carrier (Fig. 9).

Fig. 10 shows data from the field, laboratory, and theoretical calculations of change in the average temperature of the heat carrier in the tank-accumulator under different experimental conditions under the mode of gravity motion of heat carrier.

An appropriate problem under the mode of gravity motion of the heat carrier was to examine the instantaneous values for specific thermal energy for the solar heat supply systems, which are shown in Fig. 11.

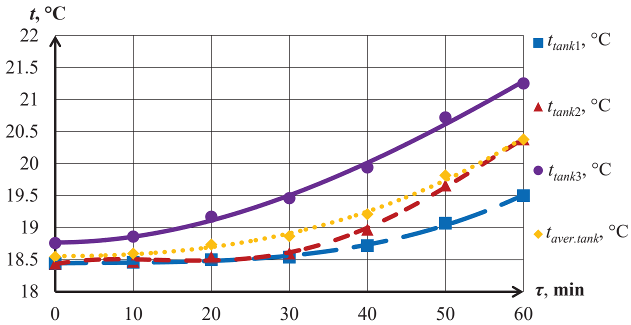

Fig. 8. A change in the heat carrier temperature in three layers of tank-accumulator $t_{tank1...3}$, °C and the averaged temperature of tank-accumulator $t_{aver.tank.}$, °C depending on the duration of heating under the mode of gravitation motion of heat carrier at the radiation of solar energy simulator onto the setup with intensity 600 W/m^2

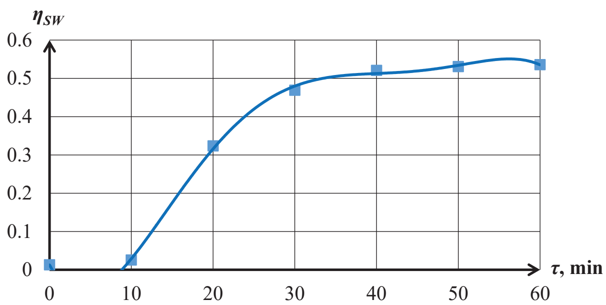

Fig. 9. A change in efficiency η_{SW} of the solar window throughout the experiment under the mode of gravitation motion of the heat carrier at radiation of solar energy simulator onto the setup at intensity 600 W/m^2

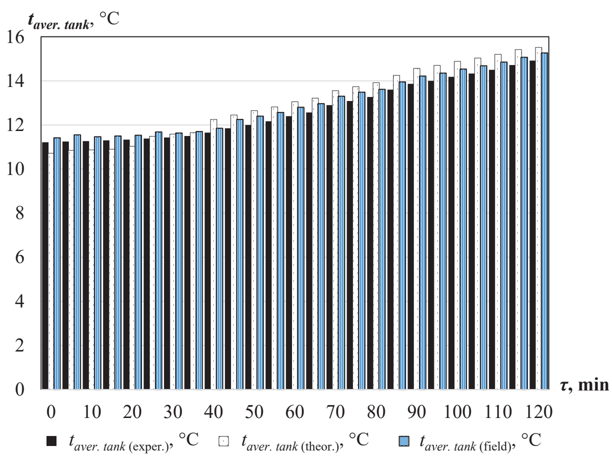

Fig. 10. Comparison of theoretical, experimental, and field data on temperature in the tank-accumulator of the solar window under the mode of gravity motion of heat carrier at radiation intensity 900 W/m^2

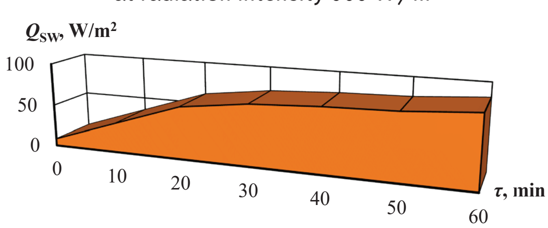

Fig. 11. Instantaneous power of solar window as solar collector Q_{SW}, W/m^2 under the mode of gravity motion of heat carrier at the radiation of solar energy simulator onto the setup with intensity 900 W/m^2

An analysis of the results reveals that the instantaneous power of a solar window at radiation of the solar energy simulator onto the setup reaches the values of 96 W/m^2 at intensity 900 W/m^2 (Fig. 11).

6. Discussion of the experimental and theoretical studies into a solar heat supply system with a solar window

The obtained results from the conducted theoretical, field, and experimental studies into the system of solar heat supply with a solar window confirm the possibility to implement the proposed schematic solution for houses at the stage of design and renovation.

The present research is continuation of the earlier theoretical-experimental study into the solar enclosure, study of the thermo-technical characteristics of solar collectors, panels, as well as research into solar radiation [10, 13, 14]. Compared with earlier studies, in this work, the design of a solar enclosure in the system of solar heat supply was examined based on the radiation of the solar energy simulator onto the setup at intensity 600 and 900 W/m^2 and under the mode of motion of gravity or circulation of a heat carrier. According to the simulated results of stratification of the heat carrier, it was found that the temperature that can be achieved by the tank-accumulator in the upper layer is 50 °C (Fig. 3). The averaged temperature of the heat carrier in the tank-accumulator at the mode of motion of the heat carrier gravity reached the value that is by 20 % smaller that the value under the mode of the motion of heat carrier circulation (Fig. 6, 8). The maximum achieved efficiency of the solar window under the mode of motion of gravity of the heat carrier as a solar collector was ≈53 % (Fig. 9).

Based on these results, we can state that the theoretical studies of the temperature of the heat carrier in the tank-accumulator were more substantial compared to the field studies, but for the laboratory tests, the opposite is true (Fig. 10).

Instantaneous specific heat capacity at constant volume consumption of 0.25 l/min is gaining growing importance, which can be associated with heating up the system, the instantaneous specific thermal capacity at these parameters changed in the range of 61–96 W/m^2 at constant solar radiation of 900 W/m^2.

Structural solutions with solar windows are simple and inexpensive to implement, which is a prerequisite for their widespread use. Since such design solutions with solar windows make it possible to receive solar energy throughout the entire day and to accumulate it more after noon, then they can be considered inertial. On the other hand, the tank-accumulator enables the additional heating of the heat carrier without a significant removal from the accumulated heat back to the solar window. Night thermal accumulation by the reduced tariff will increase the efficiency of the system in general, because the thermal energy that has not been used can remain on the tank-accumulator.

Such structural circuits of solar enclosure can solve the problems related to effective utilization of useful area in a building and solar energy as early as the stage of designing houses/buildings.

It is planned to focus further research on the comparison of factors that affected the setup during the field research, such as wind velocity and direction, etc.

7. Conclusions

1. Based on theoretical modeling of heat carrier stratification in a two-, three-, and nine-section tank-accumulator, the first and the last layer were the problematic zones in all cases. That is why, in order to enhance the efficiency reserve of the setup, it was proposed to improve the insulation of the storing capacity in the layer that is closest to the pipeline with the supplying (hot) heat carrier in such structures.

2. According to the comprehensive research into the system of solar heat supply with a solar window in the form of a solar enclosure as part of an energy efficient house, it was found that the system of solar heat supply with a solar window demonstrates the following:

– averaged temperature in the tank-accumulator under the mode of circulation motion of the heat carrier reaches 25.5 °C;

– averaged temperature in the tank-accumulator under the mode of gravitation motion of the heat carrier reaches 20.4 °C;

– instantaneous specific thermal power of the heat carrier under the mode of gravitation motion is 96 W/m^2;

– efficiency is 53 % and 55 % under the mode of gravitation and circulation motion of the heat carrier, respectively.

The benefits of the setup are the simplification of the proposed design for mounting and calculations. The utilization of useful area of the enclosure under conditions of using inexpensive building materials in the circuit solution confirms the possibility of its widespread use in the system of solar heat supply with a solar window.

3. It was found that the theoretical calculations of the averaged temperature of the heat carrier in the tank-accumulator under the mode of gravitational motion of the heat carrier are 9.5 % larger than the results obtained under real conditions.

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