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Complex Modeling of Solar Water Heating Systems

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Abstract

The energy balance equations describe each part of the observed solar water heating system (SWHS) with the differential equation method. The theoretical model consists of a number of such equations and compile into a system. Because all parts of the system interact, the Runge-Kutta method helps to solve it and obtain the close-to-real output parameters. Any kind of embedded modifications cause the reaction of the rest of the parameters, and can also be modeled with the referred method. Experimental data verified the reliability of the represented method, enabling modeled system optimization.

Keywords: *differential equation, heat transfer, modeling, solar collector*

1. Introduction

Experimental data verifies the average monthly production of about 0.8 kWh / m^2 in winter and about 1.85 kWh / m^2 in summer by solar water heating systems (SWHS) on territory of Far East of Russia. The experimental SWHS shown on Figure 1 gained the peak generation of about 2.8 kWh / m^2.

Fig. 1 - General view of the experimental SWHS, its solar collectors (roof of the Far Eastern Federal University, campus building)

This solar water heater is an experimental installation with medium thermal power output, up to 70 kW. It is also equipped with the remote trackable solar activity sensors, thermometers and transducers (Table 1).

Tab. 1: Main specifications of the experienced SWHS

Parameter	Value
The area of solar collectors	150 m^2
Type of solar collectors	vacuum with heat pipes
Total volume of heat storage tanks	10 m^3
Type of coolant circulation	forced

Initially, the system had a number of specific drawbacks: limited average thermal power of about 35 kW per day, insufficiently small storage tank, highly uneven supply of heat to the hot water system of the building.

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Further optimizations were considered for the initial SWHS design to make it more energy efficient and gain the best performance:

- Increase in the number of installed solar collectors to enlarge the capacity of SWHS;
- Increase the storage tank to increase the duration of the hot water supply during the peak loads;
- Apply the heat pump (HP) that takes energy from a low temperature source and supplies additional heat to the main storage tank within the period of maximum load.

The developed mathematical model enables optimization of technical characteristics of the given system taking into account all methods of improvement.

The represented model shows reliable accuracy due to the matching of transient processes curves with the obtained from the experienced SWHS curves. The model can identify the types of transition functions for the main circulation circuits required to configure the control loops of the experimental SWHS.

2. Schematic diagram of the modeled SWHS

The considered peak heating load for the designed SWHS (Figure 2) was set at 150 kW. The project of the model implies an addition of the vapor compression heat pump (VCHP) with maximum generation capacity of 55 kW and 15 kW electric drive compressor load. The temperature of filtered showers dump water is about 25°C and therefore it is acceptable to use it as the source of heat.

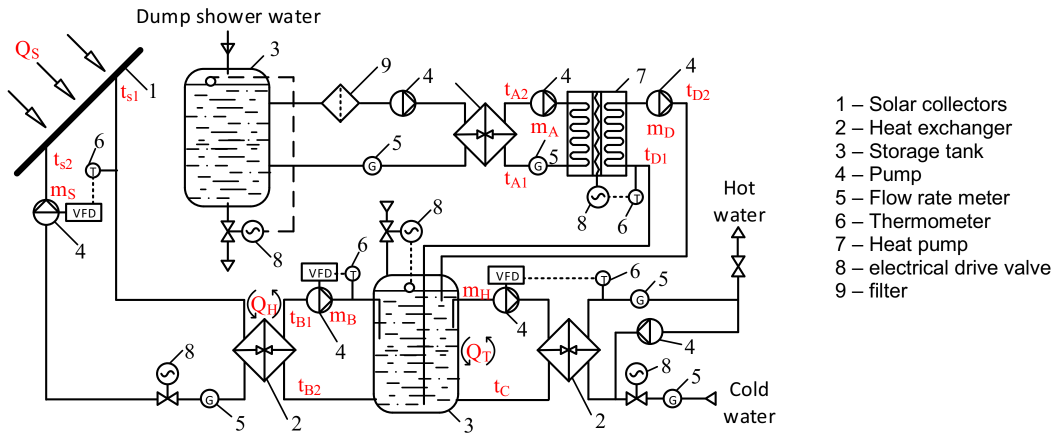


Fig. 2 – Schematic diagram of the modeled SWHS, with VCHP and dumb water heat source

Such complex solution gives the best cost effectiveness within the system lifetime. In addition, it is rather ecologically friendly because the backup source of heat is non-generated wasted energy, which supports the solar system. In addition, the proposed scheme allows the application of ventilation system in order to regenerate even more heat for the storage tank. Both the down hot water and the ventilation emissions from the facility can be the low-temperature heat sources for the HP. Inclusion of an additional buffer storage tank is highly recommended to redistribute load between SWHS and HP.

Selection of the most optimal version of scheme for the heat generating complex with reversible HP is the key issue of the described analysis. Several designs of the combined SWHS were carried out and compared with the practically modeled data. Analytical approach to the dynamic SWHS model has a number of practical advantages. The described further steps explain the performed elements analysis and ensure the necessary links between the basic units. Reflected communication between the physical parameters of different contours describe the potential problems for practical implementation. An absence of such problems in any analyzed model would therefore mean the most optimal performance within the modeled parameters.

3. System modeling

3.1 Solar collector

Solar heat absorption Q_S is changing during daylight hours, and also depending on the time of year. The equation $Q_S = f(\tau)$ represents an integral function varies with the special characteristic of the area, taking into account the specification and location of the solar collectors. The energy balance equation for the solar collector is:

$$Q_S + \dot{m}_S \cdot C_{pW} \cdot (t_S - t_S) = \dot{m}_S \cdot C_{pW} \frac{dt_{S2}}{d\tau} \qquad \text{(eq. 1)}$$

where $\dot{m}_1$ is the mass flow rate over the solar collectors circuit.

3.2 Recuperative heat exchanger

The following equations represent the changing of temperature of the heat transfer agent, neglecting the heat loss from the surface of the heat exchanger into the environment:

$$\dot{m}_S C_{pW}(t_{S1} - t_{S2}) - Q_H = \dot{m}_S \cdot C_{pW} \cdot \frac{dt_{A2}}{d\tau} \;; \qquad \text{(eq. 2)}$$

$$\dot{m}_B C_{pW}(t_{B1} - t_{B2}) + Q_H = m_B \cdot C_{pW} \cdot \frac{dt_{B2}}{d\tau}, \qquad \text{(eq. 3)}$$

where Q_H is the heat transfer coefficient of the heat exchanger.

3.3 Storage tank

Water-filled tank is the heat storage system. The cold medium draws in at the bottom and accumulated in the upper zone hot water supplies the domestic demand. Overall, the model of the tank consists of two or more containers that exchange the heat through conventional stepped septum membrane.

The following energy balance equations represent the top and the bottom of the tank respectively:

$$\dot{m}_{T01} t_{T01} C_{pW} + \dot{m}_B t_{B1} C_{pW} - \dot{m}_H \cdot t_H C_{pW} - Q_T = \dot{m}_{TH} C_{pW} \frac{dt_{TH}}{d\tau} \;; \qquad \text{(eq. 4)}$$

$$\dot{m}_{T02} t_{T02} C_{pW} - \dot{m}_B t_{B2} C_{pW} + \dot{m}_C \cdot t_C C_{pW} + Q_T = \dot{m}_{TC} C_{pW} \frac{dt_{TC}}{d\tau}. \qquad \text{(eq. 5)}$$

Heat transfer within the water volume is fixed by $Q_T = V_T \cdot K_{HM}(t_{TH} - t_{TC})$, where V_T- volume of the tank, K_{HM} – heat and mass transfer coefficient. The coefficient depends on the temperature range and on the geometric characteristics of the tank.

3.4 Heat pump

The HP is the plate heat exchanger with two sections separated by an intermediate wall with simultaneous heating and cooling inside. The heat transferred from the cold section Q_C and heat transferred to the heated agent Q_{HP} is the function of the power drive of the compressor P_{HP} : $Q_{HP} = Q_C + P_{HP}$. The energy balance equations for the heat transfer inside of the HP are:

$$\dot{m}_A \cdot C_{pW}(t_{A2} - t_{A1}) - Q_{HP} = \dot{m}_A \cdot C_{pW} \cdot \frac{dt_{A1}}{d\tau} \;; \qquad \text{(eq. 6)}$$

$$Q_{HP} - \dot{m}_D C_{pW}(t_{D2} - t_{D1}) = \dot{m}_D \cdot C_{pW} \cdot \frac{dt_{D2}}{d\tau} \qquad \text{(eq. 7)}$$

Power to drive of the compressor is the variable parameter. The COP of the HP therefore varies due to changes in temperature of the condenser and evaporator.

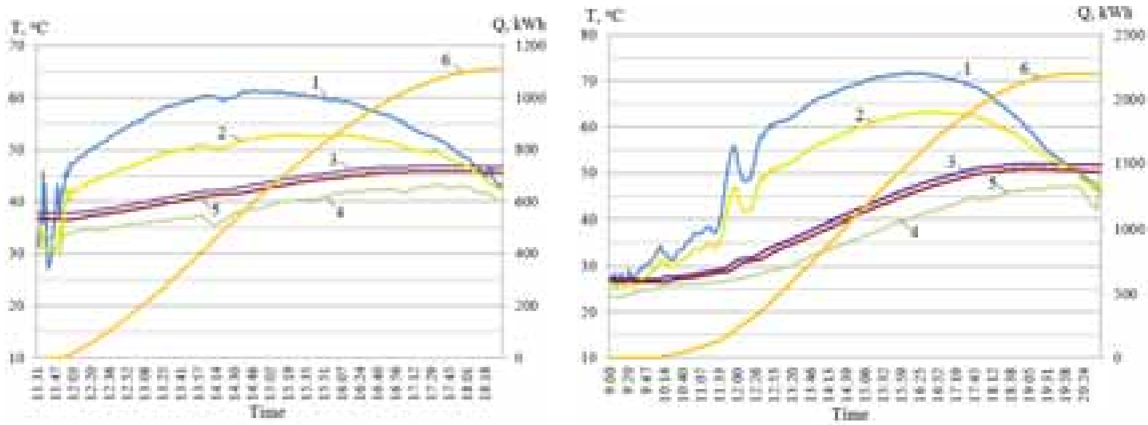
3.5 Important notice for the system of equations

The simplified model can consist of only main equations mentioned above, interacting with each other. For the additional analysis, further input elements must have sufficient number of known parameters to satisfy both mathematical solution logics and thermodynamics laws.

The unified Runge-Kutta method was applied in MathLab software to conduct the approximated modeling of the observed system of equations.

4. Verification of the results

The modeled parameters were confirmed by comparison with the experimental data. The gathered data were obtained from the field tests of the combined SWHS, installed on the roof of campus building of the Far Eastern Federal University, Far East of Russia. The variation of main parameters of the observed SWHS with the given thermal generation is represented on Figure 3.



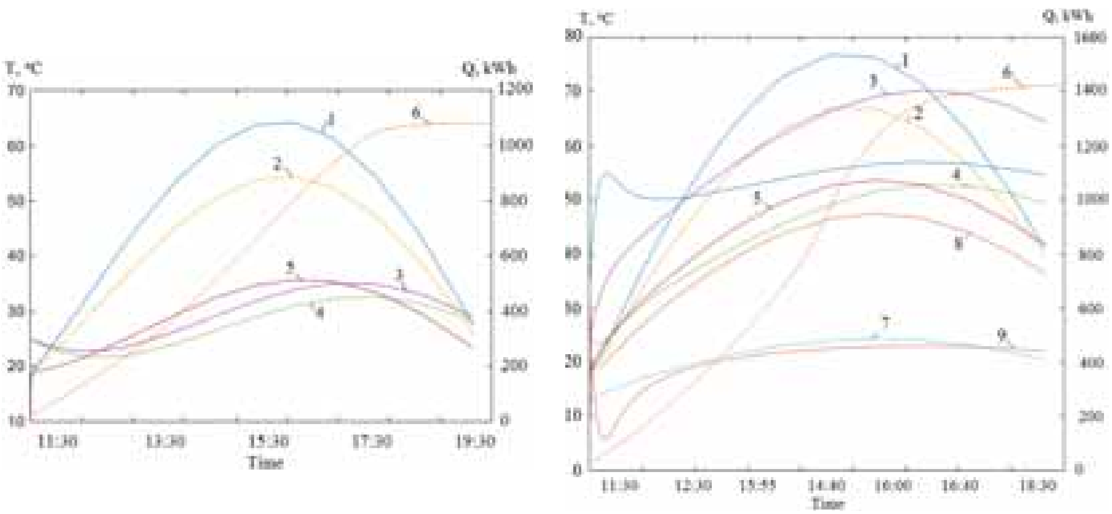
a b

Fig. 3 - Experimental curves of the experienced SWHS from data gathered in January(a) and in July (b): 1 - temperature of the collector; 2 – temperatures of the first heat exchanger, collector agent circuit; 3 - average temperature of the tank; 4 – temperatures of the supply hot water; 5 – temperature at the bottom of the tank; 6 – cumulative solar heat gain.

The shown graphs describe the typical operaton cycle of the practical SWHS in cloudy day of summer (07/05/2012) and in snowy day of winter (01/17/2013).

In simplified view, the daily solar radiation function is an approximated sinusoidal dependence: $Q_S = Q_{S\ max} \cdot \sin(\tau)$. Also, it might be set by the parabolic function $Q_S = Q_{S\ max} \cdot (a\tau 2 + b\tau)$, where $Q_{S\ max}$ is the maximum density of solar radiation during the day.

Comparing with the modeled case, Figure 4 shows the output curves for the simulated 70 kW system with the same input data, and for the optimized system with the increased number o fsolar collectors, doubled volume of storage tank, and implemented 40 kW HP.



a b

Fig. 4 - Transition functions of temperature changes in the solar collectors, heat exchangers and in main storage tank of the modeled SWHS (a – without HP, b – with the HP): 1 - the temperature of the collector; 2 – temperatures of the first heat exchanger, collector agent circuit; 3 - average temperature of the tank; 4 – temperatures of the supply hot water; 5 – temperature at the bottom of the tank; 6 – cumulative solar heat gain.

The represented model shows reliable accuracy due to the matching of transient processes curves with the obtained from the experienced SWHS curves. The model can identify the types of transition functions for the main circulation circuits required to configure the control loops of the experimental SWHS.

5. Description of the automatic control system

The modeled parameters of management actions are:

- mass flow rate regulation of the liquid in the solar circuit,
- HP power control,
- supply water flow rate control,
- water level control at the storage tank.

The operational disturbing influencers are:

- solar radiation changes at the collector surface,
- hot water demand changes,
- temperature changes of showers dumb water.

Figure 5 shows the block diagram of the control and measuring complex, which carried out the registration and recorded parameters on the main computer and automatically controlled the experimental SWHS during the tests.

Monitoring and recording environmental parameters and capacity of solar radiation incident on a horizontal and inclined surfaces produces automatic weather station Weather Station. In addition, accurate recording of direct solar radiation was carried out digital display solar radiation SOLRAD.

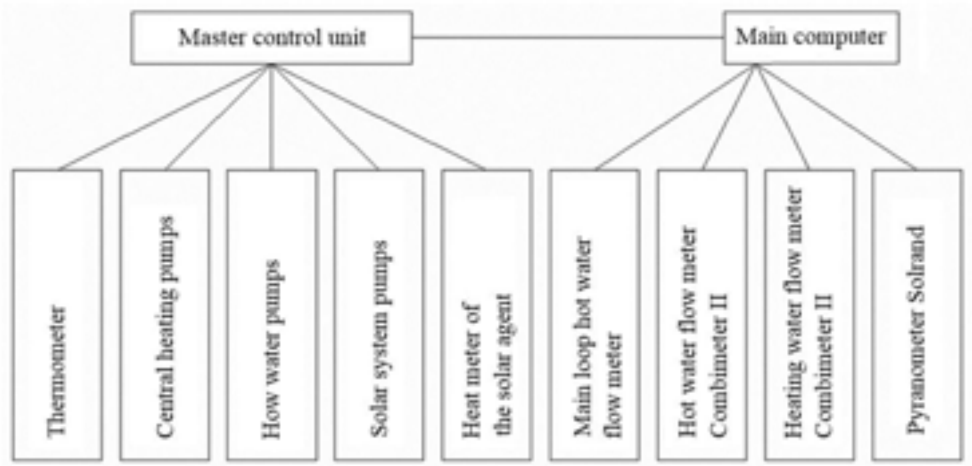


Fig. 5 - Measuring complex of the experimental SWHS

All electrical equipment for the experimental SWHS was tested and processed into the MS Excel spreadsheets.

6. Findings

1. One of the most effective and optimal scheme of SWHS is the combined solar thermal system with vapor compression heat pump (connected to two different low-temperature heat source).

2. Mathematical model improves the practical research findings. The tested method describes thermal processes of production, storage and transport of heat in the system, which can involve both series and parallel connection types, various sources of heat, and fluctuations of solar radiation intensity during the day. The model can also take into account the daily and seasonal changes in the thermal load of the consumer.

3. Comparative analysis of the experimental and calculated output characteristics data has shown the accuracy of the developed method.

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