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В докладах представлены результаты теоретических и экспериментальных исследований, проведенных в Казахстане, Германии, России, Японии и Украине по следующим направлениям: холодильная техника и компрессоростроение, теплохладоснабжение, системы кондиционирования воздуха и жизнеобеспечения, экология в холодильной промышленности, холодильная и пищевая технология. Сборник рассчитан на специалистов и ученых, работающих в областях холодильной, пищевой, химической, нефтеперерабатывающей промышленностей, а также на специалистов по системам кондиционирования воздуха и жизнеобеспечения жилых, коммерческих зданий и спортивных комплексов.

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ENERGY AND EXERGY ANALYSIS OF SOLAR ASSISTED GROUND-SOURCE HEAT PUMP SYSTEM FOR DOMESTIC APPLICATION

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In recent years' heat pumps market is growing rapidly. It is directly linked to increasing energy needs of humanity and the simultaneous depletion of traditional non-renewable energy resources in the world as a whole and in individual regions. Heat pumps are widely used in the chemical and food industries, housing and communal services. Most of the power equipment has high emission rates of energy flows to the atmosphere (refrigeration plants condensation heat, waste water enterprises, the flue gases from internal combustion engines and boilers). The use of heat pumps allows reducing greenhouse gas and carcinogenic substances emissions and, thus reducing human impact on the environment.

The heat pump performance is based with following factors: the temperature of heat source, schematic diagram of the heat pump, climate conditions of the region, working fluids of heat pump (refrigerants and intermediate coolants), heat pump elements (type of compressors, heat exchangers, control systems). There are various types of heat pumps, such as: vapor compression, adsorption, absorption and ejector heat pump systems which are able to use a variety of low-grade heat sources. Due to a simple circuit design vapor compression heat pumps are widely used, especially for domestic applications.

In 2008 more than 25% of European heat pump market share was represented by ground-source heat pumps (Forsén 2008). According to EUObserver (2007) and EHPO (2008) more than 690,000 heat pump units with 7,300 MWt of capacity were installed in Europe in 2006. The total installed capacity has been increasing in recent years and in 2013 reached 24 GW with estimate useful energy production of 13 TWh. Integration of renewable energy sources in heating and cooling applications was approximately 8,26 TWh avoiding 2,12 Mt of CO_2-equivalent emissions. The global installed capacity has reached about 15,400 MWt, and annual energy use is estimated to be 87,500 TJ (Lund 2005). The worldwide GSHP capacity has seen high growth recently. According to Le Feuvre (2008) annual growth rates have exceeded 10% over the last 10 years, mostly in North America and European countries. According to European Heat Pump Market and Statistics Report the number of installed heat pumps was more than 7,5 million units in 2015. Geothermal energy is available everywhere and Ukraine has a potential to effectively use geothermal energy. Nowadays ground- source heat pumps are a small but growing fraction of the global installed base of space- conditioning equipment [1-3].

2. Refrigerant Selection

As working substance refrigerant R410A was selected. When just considering the compressor COP, R410A does not seem to be the best solution. As soon as we extend the scope of the analysis and consider the complete heat pump system immediate advantages are highlighted. Focusing first on the condenser, where the hot water is produced, we can see, that due to the fact that R410A has no glide the dew condensing temperature is lower than with R407C by an average of 2K, leading to higher system efficiency. The sub-cooling effect plays an important role. R410A with zero glide is able to maintain a constant sub-cooling of 3.5K without a liquid receiver. R407C systems need a liquid receiver to keep a stable sub-cooling of 1.5K. In the air coil the same evaporating temperature is used for R410A and R407C. One additional effect of the absence of glide is that the unit will need less defrost cycles but this effect has not been taken into account in this paper because it strongly depends on the fin and air flow design [4].

3. Solar assisted heat pump design

Approximately 1/3 of the total energy consumption comes from heating buildings in Ukraine. Energy-saving building technologies, as well as the cost of the heating system can significantly reduce energy consumption, thereby helping to save natural resources and protect the earth's atmosphere. Considerable savings potential lies in the system of hot water. Thus, the solar collectors in combination with the ground-source heat pump in our latitudes, it is in the summer months represent the most interesting alternative to the use of the conventional heating systems.

Following system was proposed for a house about 200 m^2 floor areas. Home has two heating system: conventional natural gas heater and a ground-source heat pump. The study period was considered 170 days (average heating period). A schematic diagram of the ground source heat pump system is given on fig. 1.

Many techniques have been recently proposed in order to improve the cycle performance, more details are given by Wang, 2000, Chap.9 (Wang, 2000). In the current work, a heat exchanger has been added between the suction line and liquid line[5].

The model and assessment was performed using research methodology applied earlier by Rosen and Dincer (2007), Ozgener, O., Hepbasli, A. (2005) to a solar assisted GSHP system. The analysis of ground-source heat pump unit is partly described in this paper [6-7].

Energy, exergy and mass flows balances are provided for the system and all its components.

A general mass balance can be expressed in rate form as:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{1}$$

where $\dot{m}$ is the mass flow rate, and the subscript *in* stands for inlet and *out* for outlet.

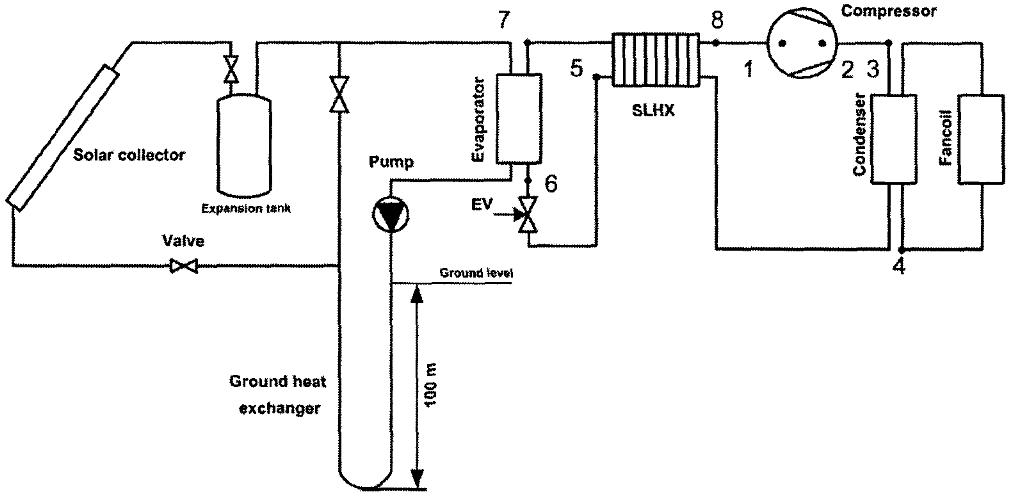


Figure 1 – Schematic diagram of the heat pump system components

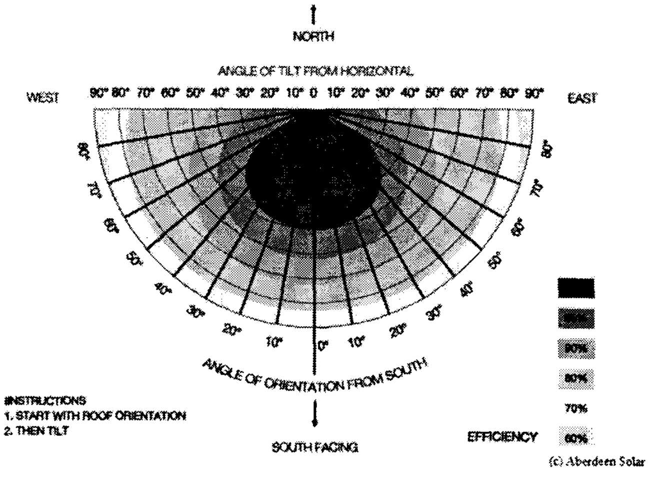
Energy and exergy balances, equating total energy (exergy) inputs to total energy (exergy) outputs, can be written as follows:

$$\sum \dot{E}_{in} = \sum \dot{E}_{out} \tag{2}$$

$$\sum \dot{E}x_{in} - \sum \dot{E}x_{out} - \sum \dot{E}x_{d} = 0 \tag{3}$$

4. Solar collector energy model

Oriented to the south and set at an angle of 30 ° to 65 ° to the horizon, solar collector is able to receive the yearly maximum amount of radiant energy. But even noticeable deviations from the above-mentioned conditions (the orientation of the south-west to south-east, the angle of inclination from 25 to 70 degrees), the solar system will run at peak efficiency (Fig. 2).



INSTRUCTIONS
1. START WITH ROOF ORIENTATION
2. THEN TILT

(c) Aberdeen Solar

Figure 2 – Influence of orientation, inclination and shade on the received radiant energy

A smaller angle is optimal in the event that the surface of the collector cannot be oriented to the south. In this case, the solar collector, located at an angle of 30 degrees, even if the orientation is south-west, with an azimuth of 45 ° will still provide up to 95% of the optimal solar power. And even when the orientation is east or west, you can still receive up to 85% of energy, with a roof inclination angle of 25 ° to 40 °. If the angle of the collector is greater, higher amount of incoming energy will be received through the year. Therefore, collectors used for the maintenance of heat pump system, installed at a large angle. This reduces the amount of excess heat in the summer, while the efficiency of the collectors in winter sunlight falling under a smaller angle optimized. Vacuum tube collectors with heat pipes should not be set at an angle less than 20 °, since in this case device will not work effectively due to lack of effect of natural convection.

When installing on an inclined roof angle of the collector is determined by the steepness of the roof. The absorber reservoir may take the maximum amount of energy if the collector plane perpendicular to the direction of the sun (Fig. 3).

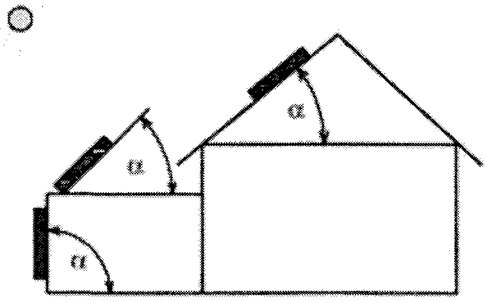


Figure 3 – Solar collector installation angle

Since the insolation in the middle of the day is the most intense, the collector plane should be oriented as far as possible to the south. However, good results can be achieved in case of deviations from the south to 45 ° south-east or south-east. Larger deviations can be compensated by a slight increase in the surface area of the reservoir. Solar collectors can be mounted and fixed on the surface of any buildings, structures or directly on the ground. Therefore, the minimum allowable angle at which they should be installed, is 25 °.

Insolation is a flow of energy, the sun emitted uniformly in all directions. On the outer surface of the Earth's atmosphere is constantly affected part of the flow of radiant energy output of 1.36 kW / m^2. This value is called the solar constant. When passing through the earth's atmosphere due to solar radiation is attenuated reflection effects, absorption and scattering of dust particles and gas molecules. Part of the radiation which passes unimpeded through the atmosphere, falls directly onto the surface of the Earth; This so-called direct radiation. Part of the solar radiation that is reflected or absorbed by dust particles and gas molecules, hits the Earth's surface non-directionally; this so-called scattered radiation. The sum of direct and diffused solar radiation is called the total radiation Unit. Under optimal conditions (cloudless sky, the middle of the day) it reaches the max. 1000 W / m^2. Solar collectors can, depending on their type and dimensions of the installation, use about 75% of total solar radiation.

The collected useful energy is transferred to the hot liquid storage tank and the evaporator of the heat pump is supplied with input thermal energy. Necessary equations for calculating the solar radiation, I_T, are taken from [8-9].

The total irradiance on a tilted surface under clear sky conditions is calculated by the equations depending upon R_b which is the ratio of the instantaneous direct solar radiation on a tilted surface to the instantaneous direct solar radiation on a horizontal surface:

$$R_b = \frac{\cos(\varphi-\beta)\cos\delta\cdot\cos\omega+\sin(\varphi-\beta)\sin\delta}{\cos\varphi\cdot\cos\delta\cdot\cos\omega+\sin\varphi\cdot\sin\delta} \qquad (4)$$

Table 1 – The average annual amount of solar energy falling on a daily basis on a horizontal surface in different regions of Ukraine.

The average figure for the last 22 years (According to NASA)	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Year avg.
Odessa, Latitude 46.30 N, Longitude 30.46 E	1.08	1.78	2.68	3.87	5.4	5.7	6.39	5.63	3.96	2.45	1.06	0.87	3.41

$$I_T = I_b R_b + I_d \frac{1+\cos\beta}{2} + (I_b + I_d)\rho \frac{1-\cos\beta}{2} \quad (5)$$

Solar collectors are modeled with the formulation as suggested by Duffie and Beckman [8-9]:

$$\mathbf{\dot{Q}_u = A_c I_T \eta_c} \quad (6)$$

where Q_u is the useful energy collected in system collectors, A_c – the collector area, and the collector efficiency, η_c, is:

$$\boldsymbol{\eta_c = F_R\left[(\tau\alpha) - U_L \frac{T_{in}-T_0}{I_T}\right]} \quad (7)$$

In this equation, T_{in} is the collector inlet temperature and T_o – the environment temperature. A water storage tank is placed after the solar collectors as shown in fig. 1. Perfect mixing within the tank is assumed. If the rate of heat addition and removal for a reasonable time period of Δt are assumed to be constant, equations can be written for each time interval as suggested by Duffie and Beckman [8-9]:

$$T_{st,new} = T_{st,old} + \frac{\Delta t}{(mc_{vw})_s}\left[\dot{Q}_u - \dot{Q}_e - (UA)_s(T_{st} - T_0)\right] \quad (8)$$

In eq. (8) Q evap is the extracted energy from the storage tank in the evaporator, Ts – the main storage temperature for the period, and m – the storage tank mass. $(UA)_s$ is taken as 11.1 W/K [8-9].

5. Energy and exergy analysis.

For analysis purposes, the following assumptions were made:

1. The storage tank temperature, which is connected to storage tank mass, collector type and area, is calculated for hourly input from 08:00 to 18:00.
2. All processes are steady-state and steady-flow with negligible potential and kinetic energy effects and no chemical or nuclear reactions.
3. Heat transfer to the system and work transfer from the system are positive.
4. In this study, the reference state is taken to be the state of environment at which the temperature and the atmospheric pressure on 15 July 2016 were 30°C and 101.3 kPa, respectively. The thermodynamic properties of water, air and R410A are found using the REFPROP software package.

The mass balances of the solar collector can be calculated as:

$$\dot{m}_1 = \dot{m}_7 = \dot{m}_{w,col} \quad (9)$$

Energy balances can be determined as follows.

Heat balance of the storage tank:

$$\dot{Q}_t = \dot{Q}_e = \dot{m}_{w,st} \cdot c_{p,w}\left(T_{w,o} - T_{w,i}\right) \quad (10)$$

Heat balance of the solar collector:

$$\dot{Q}_{col} = \dot{m}_{w,st} \cdot c_{p,w}\left(T_{w,o} - T_{w,i}\right) \quad (11)$$

The conventional parameter that has been used to describe the heat pump performance is COP (Coefficient of Performance), which is the ratio of the quality of the useful heat output to the quantity of work driving the compressor:

$$COP_{GSHP} = \frac{\dot{Q}_c}{\dot{W}_{comp}} \quad (12)$$

in terms of electrical input

$$COP_{GSHP} = \frac{\dot{Q}_c}{\dot{W}_{comp,e}} \quad (13)$$

for the whole system

$$COP_{sys} = \frac{\dot{Q}_c}{\dot{W}_{comp} + \dot{W}_{pumps} + \dot{W}_{fans}} \quad (14)$$

for the whole system in terms of electrical input,

$$COP_{sys} = \frac{\dot{Q}_c}{\dot{W}_{comp,e} + \dot{W}_{pumps,e} + \dot{W}_{fans,e}} \quad (15)$$

The specific flow exergy of refrigerant, air or water is evaluated as

$$ex = (h - h_0) - T_0(s - s_0) \quad (16)$$

where h is enthalpy, s is entropy, and the subscript zero indicates properties at the dead (reference) state (i.e., at P0 and T0).

The exergy rate can be determined as:

$$\dot{E}x = \dot{m} \cdot ex \quad (17)$$

The exergy efficiency is expressed as the ratio of total exergy output to total exergy input:

$$\varepsilon = \frac{\dot{E}x_{output}}{\dot{E}x_{input}} \quad (18)$$

where “output” refers to “net output” or “product” or “benefit” or “desired value”, and “input” refers to “driving input” or “fuel”.

The exergetic coefficients of performance of the GSHP unit and whole system are as follows:

$$COP_{ex,GSHP} = \frac{\dot{Q}_c(1-\frac{T_0}{T_c})}{\dot{W}_{comp,e}} \tag{19}$$

$$COP_{ex,sys} = \frac{\dot{Q}_c(1-\frac{T_0}{T_c})}{\dot{W}_{comp,e}+\dot{W}_{pumps,e}+\dot{W}_{fans,e}} \tag{20}$$

The exergy efficiency of the heat exchanger (condenser or evaporator) is determined as the increase in the exergy of the cold stream divided by the decrease in the exergy of the hot stream, on a rate basis, as follows:

$$\varepsilon_{HE} = \frac{\dot{E}x_{cold,out}-\dot{E}x_{cold,in}}{\dot{E}x_{hot,in}-\dot{E}x_{hot,out}} = \frac{\dot{m}_{cold}(ex_{cold,out}-ex_{cold,in})}{\dot{m}_{hot}(ex_{hot,in}-ex_{hot,out})} \tag{21}$$

Exergy destruction in solar collector:

$$Ex_{d,col} = \dot{m}_{w,col}(ex_9 - ex_{10}) - A_c I_T(1 - \frac{T_0}{T_p}) \tag{22}$$

Exergy efficiency of solar collector:

$$\varepsilon_{col} = \frac{\dot{E}x_{out}-\dot{E}x_{in}}{A_c I_T\left(1-\frac{T_0}{T_p}\right)} = \frac{\dot{m}_{w,col}(ex_{out}-ex_{in})}{A_c I_T\left(1-\frac{T_0}{T_p}\right)} \tag{23}$$

6. Results and discussion

Ground-source heat pump was calculated according to European Standard EN 14511-2:2013. Heat pump operation modes were selected B0/W35 and B0/W55. The European Standard EN 14825 was used for the domestic hot water production in the seasonal efficiency calculation. A strength of standard prEN14825 is that it includes all kinds of heat pumps (except exhaust air heat pumps). The model treats heat pumps both in heating and cooling operation. The fact that the heat pump is tested in exactly part load should result in more sufficient results compared to degradation coefficient etc. The model is foreseeable and quite easy to follow, though it is not completely clear with its definitions of part loads.

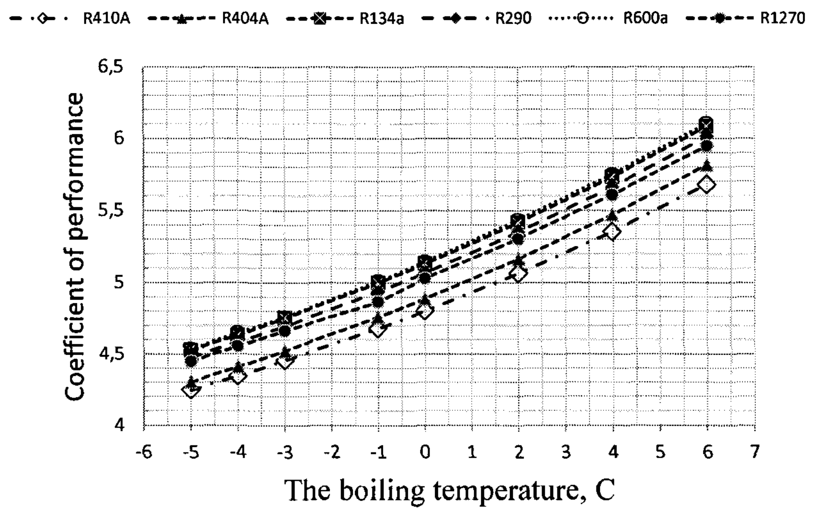


Figure 4 – Coefficient of performance from refrigerants boiling point dependence COP=f(t_0)

For analysis purposes different refrigerants were considered such as: R134a, R404a, R410A, R290, R600a, R1270. Coefficient of performance from refrigerants boiling point dependence is shown on figure 4. Results show that the maximum thermodynamic efficiency of the heat pump corresponds to R600a and R134a as a working fluid, the minimum - R410a. In the case of R600a and R134a heat pump COP is 7.6% higher in comparison with refrigerant R410a. These results are also connected with molecular weight of different refrigerants [10-11]. Large values of evaporation enthalpy are found for substances with light molecule weight and the energy losses across a compressor’s valves are high when the molecule weight is high. The thermal transformation coefficient for R290 and R1270 are practically the same range. Analysis of the of R134a and R600a performance shows that in terms of ease of use and maintenance R600a is more preferred. Refrigerant R600a is a natural working substance, in the case of depressurization of the system can be easily refueled in contrast to the blend refrigerant R134a.

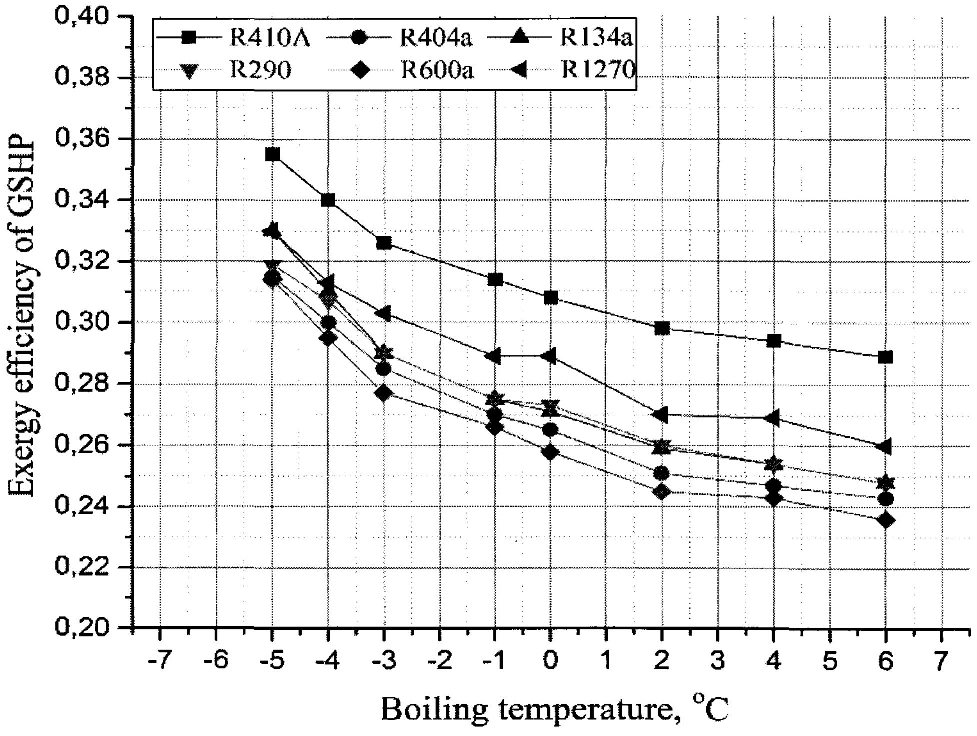


Figure 5 – Exergy efficiency of ground source heat pump unit.

Figure 5 presents exergy efficiency data for GSHP unit at different boiling temperature for all considered in Table 1 refrigerants. From this graph, the lowest value of exergy efficiency for the GSHP unit are for R600a in the range from 23.4% to 31.7% and highest for R410A in the range 29% to 35.6% respectively at the boiling point -5 °C and reference state temperature 27 °C.

7. Conclusions

"Energy Strategy of Ukraine for the period until 2035" provides radical changes in the structure of heat sources. The main factor that causes these changes is a sharp increase in world prices for natural gas, oil and petroleum products. Therefore, it is forecasted the gradual replacement of natural gas boilers and CHP plants majority that currently provide the vast proportion of the thermal energy production with new technologies, including the use of environmental heat, renewable energy sources (solar, wind, water, geothermal heat) and especially heat pumps. This research of the solar assisted heat pump allows us to analyze efficiency of the solar assisted ground-source heat pump system and determine efficiency of solar collector. In this paper, solar assisted ground source heat pump system with indirect expansion for space heating using a working fluid R410A was investigated. The discharge temperature of R410 is perfect for high temperature applications (domestic hot water production). In this case, it can be said that this simulation is reliable for determining the sizing and selection of system requirements such as collector type and area and storage tank mass for domestic application in south Ukrainian region. Insolation data was provided for last 22 years' period.

The exergy analysis allows to evaluate the amount of exergy destroyed for each component of the system and determine which component weights more on the overall system inefficiency. Depending on the selected collector type and area, storage tank mass and condenser capacity, exergetic analysis can be applied on the system and exergy destruction rate, improvement potential and exergy efficiencies of the system components can be estimated. Before installation of the system, the equipment which have the high irreversibility values may be improved and so overall performance of the system can be increased. As a result, the simulation and results is a helpful guide to engineers for design of solar assisted heat pump systems.

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