УДК 536.621.57

Draganov B.

ANALYSIS AND OPTIMIZATION OF UNDERGROUND HEAT ACCUMULATORS IN HEAT PUMP SYSTEMS

Abstract

Application of the heat accumulators in general, and the underground ones in particular, allow the raising of the efficiency of heat pump systems by means of balancing the rates of production and consumption of heat. The system also eliminates operation of the heat pump of low loald. It is most efficient to use water-carrying ground layers as natural underground heat accumulators, also developed a mathematical model and carried out experimental investigation on the proposed system.

Accumulator charging is made from heliosystems. The authors analyse the problems optimization of energy accumulation.

Keywords: – Heat accumulator, Heat pump systems, Mathematical Model, Optimization.

NOMENCLATURE

C_n — exergy costs;	L — graph contour;
e_k — annual exergy charge;	Q — heat flow (kWatt)
F — heat exchange surface (m^2)	T — temperature (°C)
G — discharge ($kg\ s^{-1}$)	V — volume (m^3)
K_n — annual capital expenses;	Z — efficiency criterion;
P — pressure (bar)	h — specific henthalpy ($kJ\ kg^{-1}$)
Π_n — annual exergy consumption;	s — specific enthropy ($kJ\ K^{-1}\ kg^{-1}$)

Greek letters

α — heat transfer coefficient ($kJm^{-2}K^{-1}$)	ξ — concentration ($kg\ kg'^{1}$)
ρ — dencity ($kg\ m^{-3}$)	η — coefficient of perfomance

Subscripts

A — accomolator	U — user heat
e — exergy	S — source heat
opt — optimal	W — wall of the heat exchanger

INTRODUCTION

The climate conditions of Ukraine, and especially of its southern regions, enables the use of various heliosystems: for direct water heating and solar assisted heat pumps.

Research on the application of heliosystems [1] showed the necessity of using heat accumulators to increase their effectiveness.

For heat pump systems which are being investigated for the conditions of Ukraine, the need for heat accumulators is expressed by:

- balancing graphs of production and utilization of heat;
- elimination of the need to use heat pumps at part load conditions.

Economically based abroad, the night tax for electric power and other taxes for equipment exploitation are not yet actual for Ukraine. Short-rate peak heat loadings seem more reasonable to be covered with direct electric heating, which greatly lowers expenses for the creation of the system.

Heat accumulators differ with the terms of charging-discharging, consequently there appear the demands to their work.

SCHEMATIC SOLUTIONS

As an example of the system (Fig. l), the authors analyze the system of heat and cold supply on the basis of the absorption heat pump, as such a system is the most suitable due to the number and temperature levels of the heat accumulators. The system possesses two heat accumulators one with high and medium potential with 24-hour accumulation, and the other with low potential of seasonal heat accumulation. A special characteristic of the summer condition of the system operation is the charging of the low potential heat accumulator.

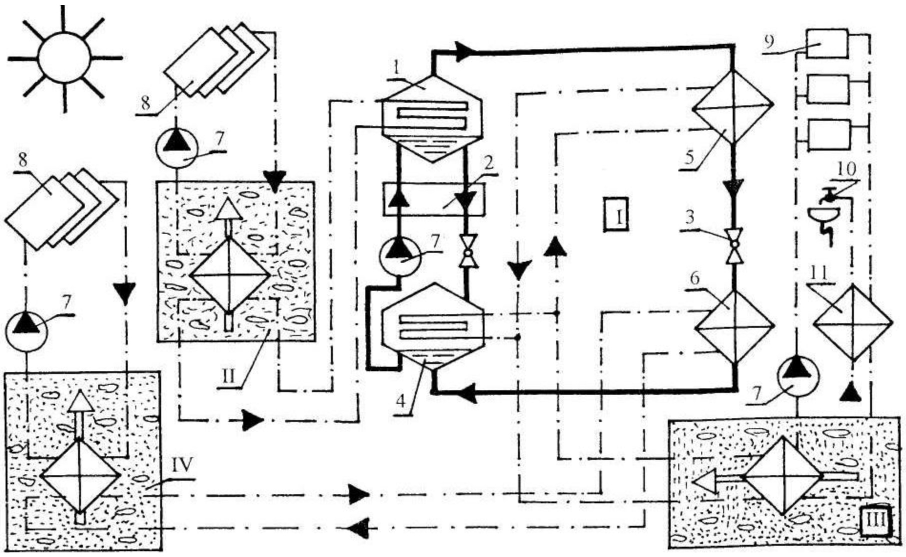

Figure 1. Schematic solution of the heat-and-cold supply system on the base of the absorption heat pump with the heliosystem:
I — outline of the heat pump; **II** — accumulator of the high potential heat; **III** — accumulater of the medium potential heat; **IV** — accumulator of the low potential heat; 1 — generator; 2 — heat exchanger of solutions; 3 — trottle valve; 4 — absorber; 5 — condenser; 6 — evaporator; 7 — pump; 8 — solar collectors; 9 — heating system; 10 — hot water supply system; 11 — boiler of the hot water supply.
—— — refrigerant of the heat pump; — • — • — — heat -and — cold carriers between the heat pump and peripheral equipment, water.

A schematic solution of the seasonal heat accumulator is presented in Fig. 2. From the viewpoint of calculation and design, this accumulator is formed by a series of two-flow heat exchangers. The system of equations which describe its operation, is as follows:
– at the charge
$Q_S = Q_A$
– at the discharge
$Q_A = Q_U$.

High potential and medium heat accumulators present a complex system of accumulation, where the heat flow from the heat source (solar energy, or condensation and absorption heat) is delivered to the consumer through the accumulating level. The principal scheme of it is presented in Fig. 3.

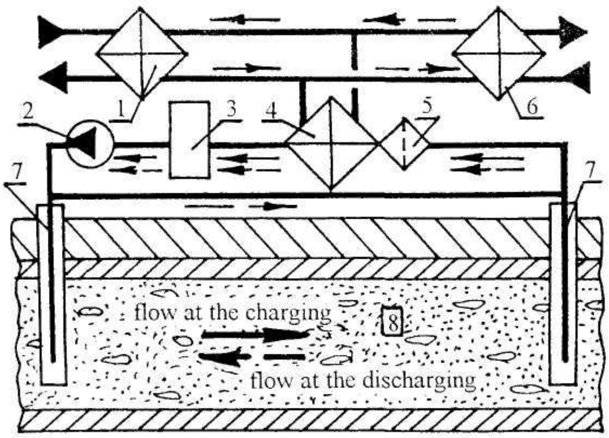

Figure 2. Seasonal Heat Accumulator in the Underground Accumulating Layer: 1 — heat source (heliosystem of low potential); 2 — pump; 3 — tank; 4 — heat exchanger; 5 — filter; 6 — user (evaporator); 7 — borchde; 8 — underground accumulating layer.

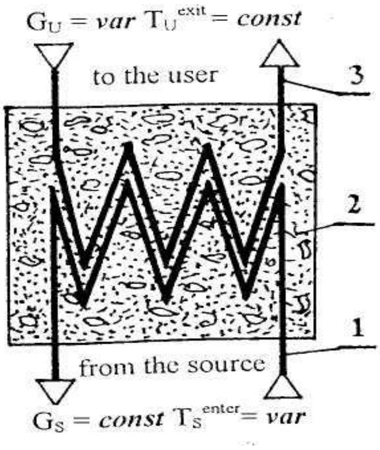

Figure 3. 24-hours Heat Accumulator (principal scheme): 1 — heat source; 2 — accumulating layer; 3 — heat user.

Phases of work of the 24-hour heat accumulator are:
– charge with the consumer switched off
$$Q_S = Q_A \quad (3)$$
– charge with the consumer switched on
$$Q_S - Q_A + Q_U \quad (4)$$
– work with the full charge of the accumulator
$$Q_S = Q_A = Q_U \quad (5)$$
– discharge at "peak" loads
$$Q_S + Q_A = Q_U \quad (6)$$
– discharge with the source switched off
$$Q_A = Q_U. \quad (7)$$

From the system of equations (3)-(7) it can be seen thet there are thre coexisting heat flows in the 24-hour heat accumulator. Calculation of the three-flow heat accumulators may be conducted in two ways:
- balancing the heat flows in the two-flow heat

exchangers;

- creating a method of calculation of three-flow heat exchangers to account for the heat accumulating ability of one of the flows.

MATHEMATICAL MODEL

Problems of optimization at the present stage of devdopment pressuppose computer based calculations. Consequently, when analysing the complex system of heat and cold supply, the problems of optimization and calculation are also of complex character and consider all the energy transformations both inside the system and in its interaction with the peripheral equipment.

The three-flow heat exchanger has found a wide range of applications in cryogenic installations for the separation of air [2], and the existing methods of its calculation take into consideration the following :

- all the three flows are in the gaseous state;
- the flows have no output and heat direction of their own;
- a big thermal head occurs in the process of heat exchange.

It is clear that none of the above named conditions is observed in the three-flow heat accumulator. Due to this, for the basis there can be taken only the principle of composing conditions of heat transfer in the three-flow heat exchanger of cryogenic installations for the specific case of simultaneous heat contact of three flows (the heat exchanger with the conditionally soldered pipes on their whole length).

In general, the heat exchange conditions are written in the following way (8)-(10):

$dQ_s = \alpha_s(T_s - T_w)\, dF_s$

$dQ_u = \alpha_u(T_w - T_u)\, dF_u$

$dQ_A = \alpha_A(T_w - T_A) dF_A$

and one of the equations of the system (3)-(7) depending on the mode of operation.

The work of he heat pump absorption system is based on two graphs:

- solar radiation (conditions of work of the highly potential heat accumulator);
- utilization of hot water (conditions of work of the medium potential heat accumulator).

The system of equations (8)-(10) for the first case should be completed with the statistic information on the solar radiation intensity in the given area $dQ_s = f(T_s)$; for the second case it should be completed with the statistic information on water consumption for object analysed $dQ_u = f(G_u)$, on condition that the temperature of the hot water 0ieat carrying medium) at the outlet from the heat accumulator is always constant.

On the basis of the generalized mathematical model of any two-flow heat exchanger which is included into the absorption heat pump (theory, methods, testing in [3]), the authors offer the next mathematical model of the three-flow heat accumulator.

Mathematical model of the system as a whole is presented by models of each separate element (i-th element) in the form of series of functional operators:

$$Y_i = f_{Yi}\,(X_i, U_i, K_i, \Gamma_i)$$

$$\Phi_i = f_{\Phi i}\,(X_i, U_i, K_i, \Gamma_i) \qquad (11)$$

$\psi = (P, T, h, \rho, s, \xi)$

where Y_i — outcoming parametres of the i-th element; Φ_i — functional characteristics of the i-th element; f_{Yi}, $f_{\Phi i}$ — non-linear functions of the i-th element; X_i — incoming internal parametres of the i-th element; U_i — incoming external parametres of the i-th element; Γ_I — topology of joining of the i-th element; ψ — look of the state equation; K_i — construction parametres of the i-th element.

The system (11) for the three-flow heat accumulator may be shown as:

$X_A = \{P_A, h_A, G_A, \xi_A, P_s, h_s. G_s, \xi_s, P_U, h_U, G_U, \xi_U\}$

$Y_A = \{T_A, s_A, \rho_A, \xi_A\}$

$U_A = \{\varnothing\}$ — with the full charging of the accumulator

$U_A = \{T_A\}$ — for all the other cases (12)

$\Phi_A = \{Q_A, \eta_A\}$

$K_A = \{F_s, V_A, F_U\}$.

In the case of analysing the seasonal heat accumulator (two-flow one) in the system of equations (12), the absent flows in the analysed operational phase are taken as equal to the zero.

Concrete definition of the system connections (12) is fulfilled on the dependencies like those presented in [3]. Solution of the concretely defined connections is finished by the group of balanced equations of the i-th elements: wastes, energy, hydraulic heads, enthalpy changes.

The theory nd calculation of the heat accumulating properties of underground heat accumulators for combined exploitation with the heliosystem are presented in [1].

EXPERIMENTAL INVESTIGATION

In order to study experementally the system of heat-and-cold supply with heat accumulators, a building with 60 apartments situated on the southern shore of the Crimea peninsula was chosen. Calculated refrigerated capacity of this system for air-canditioning purposes in the summer time is 0.1 MWt. Temperature level of the higly potential energy obtained in the heliosystem has not given the possibility to use this energy as a heating source for the generator of the absorption heat pump. It seemed reasonable to substitute the absorption heat pump by a vapour compression one.

Table 1. Hydrogeological characteristics

Characteristics	Measuring units	Values
Depth of laying	metre	60
Porosity	%	35
Density of the rock	kg m^{-1}	2100
Filtration coefficient	m(24hrs)$^{-1}$	0.5
Heat capacity of the rock	kJ kg^{-1} K^{-1}	0.8
Heating coefficient of the rock	-	0.7

Hydrogeological conditions of the site for creating the underground heat accumulator which assures ecologycal safety, are shown in Table 1 technical task for the calculation of heat' accumulators included into' the heat pump system is shown in Table 2, calculated design characteristics of the underground heat accumulator are shown in Table 3; characteristics of the heat pump are shown in Table 4.

The obtained calculated design data of technical characteristics of the underground heat accumulators included into the heat pump equipment, coincide well with the experimental data of systems operating around tiie world . This underlines the correctness of the theory of designing the underground heat accumulators and the mathematical model for their calculation.

Tabie 2. Technical task

Characteristics	Measuring units	Values
Heat capacity	MWt	0.5
Period of charge-discharge	24 hrs	126
Temperature of the heat carrying medium in the charge mode: - at the entrance to the hot borehole - at the exit from the cold borehole	 °C °C	 60 25
Temperature of the heat carrying medium in the discharge mode: - at the exit from the hot borehole - at the entrance to the cold borehole	 °C °C	 50 35
Characteristics of boreholes: - depth - diametre	 metre metre	 60 0.2

Table 3. Calculated Characteristics of the Heat Accumulator

Characteristics	Measuring units	Values
Heat-carrying medium waste	$m^3 hrs'^{-1}$	18
Volume of the water-carrying horizon	m^3	46650
Area of water-drainage	m^2	3580
Heat accumulating volume	J	2534

Table 4. Calculated Design Characteristics of the Heat Pump

Characteristics	Measuring units	Values
Type of the heat pump	-	compressor
Refrigerant	-	R142b
Boiling temperature	°C	20
Condensation temperature	°C	75
Heat loading to the apparatus: - condenser - evaporator	 kWt kWt	 655 500
Heattransfer surface of the apparatuses: - condenser - evaporator	 m^2 m^2	 82 100
Area of the helium collector for the low potential heat	m^2	1300

The application of thermoeconomic (exergoeconomic) principle of optimization [4-8] is based on estimation of exergy losses in system by money. In this case use the economic characteristics incorporated in exergy estimation of system. Such approach unites energy and economic estimation and does not concede on objectivity of generality technicoeconomic optimization.

Let's consider homogeneous system consisting of various elements where one flow h_1 consistently and unitary cooperates with n flows (Fig. 4) [8].

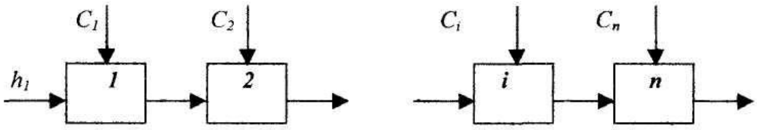

Fig. 4. The linear scheme of power system

In this case of optimum synthesis problem can be formulated as; it is necessary to distribute the set of flows C_1, i = 1,2,..., n along the flow h_j, j = 1 that the flow parameters h_1 after system were in given interval of meanings, and the chosen optimization criterion accepted the minimal value [4].

Let's accept expression of total thermoeconomic expenses in system as optimization criterion

$$\sum_i \sum_j Z_{ij} = Z_\Sigma^{min} , \qquad (13)$$

where Z_{ij} is thermoeconomic expenses in i-th element of system (as = l)

$$Z\left\{Z_{i_p}^{(p)}\right\}, \; p = 1, 2, \ldots, k; \; i_p = 1, 2, \ldots, [n - (p - 1)]. \qquad (14)$$

The $Z\left\{Z_{i_p}^{(p)}\right\}$ can be broken on k subsets

$$Z\left\{Z_{i_p}^{(p)}\right\} = \bigcup_{p=1}^{k} Z_p \left\{Z_{i_p}^{(p)}\right\}.$$

Here subset

$$Z_p\left\{Z_{i_p}^{(p)}\right\} = \left\{Z_1^{(p)}\, Z_2^{(p)}, \ldots, Z_{i_p}^{(p)}, \ldots, Z_{[n-(p-1)]}^{(p)}\right\} \qquad (15)$$

is possible meanings of thermoeconomic expenses on some stage p, $p \le k$.

Then on each intermediate stage p it is necessary to choose such flow for which

$$Z_{i_p}^{(p)} \in Z_p\left\{Z_{i_p}^{(p)}\right\}$$

$$Z_{i_p}^{(p)} = Z_{min}^{(p)}, i_p = 1, 2, \ldots, [n-(p-1)] \qquad (16)$$

where $Z_{min}^{(p)}$ is minimal thermoeconomic expense for stage p. Then the chosen flow is excluded from the further consideration. Hence, for numbers of elements p-th and (p - 1)-th subset the ratio is fair

$$Z\left\{Z_{i_p}^{(p)}\right\} = Z_{(p+1)}^{(p)}\left\{Z_{i_{(p+1)}}^{(p+1)}\right\}. \qquad (17)$$

Then from eq. (16) with eq. (17) the number of elements of set $Z\left\{Z_{i_p}^{(p)}\right\}$ is equal to number of possible variants of distribution of cooperating flows

$$Z\left\{Z_{i_p}^{(p)}\right\} = \prod_{p=1}^{k} Z\left\{Z_{i_p}^{(p)}\right\} = \frac{n!}{[n-(k-1)]}. \qquad (18)$$

For achievement given parameters for flow h_j it is necessary $k \le n$ elements, i.e. necessary to find the set of flows $C_k \in C$ for eq. (13) was carried out.

Generally thermoeconomic criterion of optimality is

$$Z_\Sigma = \left(\frac{\sum_n C_n\, \Pi_n + \overline{K}_n}{\sum_k e_k}\right), \qquad (19)$$

where C_n, Π_n — cost and annual exergy consumption from external sources; $\overline{K}_n$ — annual capital and others expenses associated with n-th element; e_k — annual exergy charge for k-th product reception.

Eq. (19) has more simple kind for special cases. For example, for installation with one product (where B is output of product)

$$Z_\Sigma = \min\left(\frac{\sum_n C_n\, \Pi_n + \overline{K}_n}{B}\right), \qquad (20)$$

Thus, the optimization problem can be generally shown to search extremum of function

$$Z_{opt} = \min\, Z_\Sigma \qquad (21)$$

or for parametrical optimization

$$\eta_{opt} = \max \eta_e^\Sigma . \qquad (22)$$

The special interest represents the geometrical device of exergoeconomic optimization. This method is clear consequently it is convenient for decision of optimization problems [4-8].

CONCLUSIONS

The obtained calculated design data of technical characteristics of the underground heat accumulators included into the heat pump equipment coincide well with the experimental data of systems operating around the world. This underlines the correctness of the theory of designing the underground heat accumulators and the mathematical model for their calculation.

REFERENCES

[1] Draganov B., Fara L., Enchancement of heliothermal systems efficiency, Solar Energy for Sustainable Development, Vol. 3, No. 1-2, 63-66, (1994).

[2] Alekseev V., Vajnshtejn G., Gerasimov P. Calculations and modelisations of the apparatus kriogenic systems, Leningrad, Energoatomisdat, p. 13-16, (1987).

[3] Morosuk T. Schematic methods of raising the efficiency of power installations by means of inclusion into their composition the absorption thermotransformers. These of doctorat, Odessa State Academy of Refrigeration, (1994).

[4] Nikulshin V., Andreev L. Exergy Efficiency of Complex Systems. Proceedings of International Conference of Ocean Technology and Energy, OTEC/DOWA, 99, Japan, pp. 161-162,1999.

Bejan A., Tsatsaronis G., Moran M. Thermal Design and Optimization. New York J. Wiley, 1996.

[5] El-Sayed Y. Revealing the cost efficiency trends of the design concepts of energy intensive systems. Energy Convertion and Management, 40, pp. 1599-1615, 1999.

[6] Morosuk T. Exergoeconomics methods in absorption thermotransformers optimization. Industrial heat engineering, 4, 22, 2000, pp. 15-19.

[7] Dragan G., Draganov B. Methods of Power Systems Optimization // Bulletin of the Politechnic Institute of Iassy. Tom XLVIII (LII). 2002. P. 191-198.