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The use of solar energy for air-conditioning in the cold season by employing a heating pump on lithium bromide - water solution

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Abstract: *The authors of the present paper discuss the results following the monitoring of a refrigerating plant's performances with absorption of lithium bromide -water solution, where the evaporator, representing the cold source, is fed with hot water prepared in a system of solar panels. Furthermore, they include a working scheme which can be used in both hot and cold seasons, resulting in a substantial reduction of the conventional energy use, with a direct effect on the protection of the surrounding environment.*

Keywords: renewable energy source, environment protection

Nomenclature

c_p	specific heat capacity, [kJ/kgK]
COP	performance coefficient, [-]
f	circulation factor, [-]
h	specific enthalpy, [kJ/kg]
Q_{m0}	refrigerant masic flow rate, [kg/s]
Q_m	masic flow rate, [kg/s]

Greek letters

Φ	heat flux, [kW]
ξ	LiBr concentration

Subscripts

A	absorber
C	condenser
G	heat generator
SC	strong LiBr solution
SD	weak LiBr solution
w	water

1. Introduction

The use of renewable energy sources has become a standard in the industry of new constructions. The plants which work on solar energy are designed foremost for preparing household hot water and for feeding the generator of refrigerating machines with absorption during the hot season. Such a case was studied by the present authors in article [1].

A contemporary challenge is adapting the above mentioned plants so that they can use solar energy also in the cold season, by working in heat-pump regimes. As a result of this, especially in winter days with clear skies, a significant quantity of energy can be captured. Marcos and Izquierdo [2] recorded in the case of a 42 sq. meters surface of solar panels which provides for an 80 sq. meters laboratory, with temperatures up to 70 °C for the thermal agent in the cold season.

Clausse and Alam [3] used solar energy for air heating, recording in the case of 16 solar panels positioned on a surface of 38,7 sq. meters the delivered warm air's temperature, the value of which was 2 °C below the comfort temperature inside the rooms. Although solar energy does not fully cover the requirements of air heating, it becomes noticeable that conventional sources are significantly less used; this fact has implications on the protection of the surrounding environment.

Yamankaradeniz and Horuz [4] developed a similar study, by monitoring the characteristics of a heating pump driven by a solar circuit in clear summer days for Istanbul's area. The analysed characteristics included the thermal power of the condenser and the plant's performance coefficient.

In the sections below the authors will present alterations made to the summer time feeding system of the refrigerating plant and will include a calculus of the plant's performance.

2. Experimental stand

The experimental stand is represented by the following composing elements:

- a system of 30 solar panels with a total surface of 80 sq. meters, which can provide a maximum thermal power of 40 kW in the hot season;
- a system for capturing and storing the panels delivered energy consisting, first, of a plate heat exchanger - here is where takes place the heat exchange between the etilenglicol – water solution belonging to the solar system and the softened water belonging to the rest of the plant, as well as, second, of a storage tank with a volume of 4000 l;
- the water consumer, where the water was prepared in the solar system and used for driving the plant's boiler in the hot season or for ensuring the necessary heat for boiling the refrigerating agent in the evaporator;
- a classic system for warm water preparation composed by a boiler with a thermal power of 50 kW, driving on gas fuel. This boiler fully covers the heat requirements for the cold season.

The refrigerating plant with a lithium-water bromide solution absorption is of a reversible type and thus can be used for cooling the air in the warm season, as well as for warming the air in the cold season, while in the warm season takes on the role of filling in the heat requirements for this equipment during days with low insolation.

The refrigerating agent used in this type of plants is water with vaporization temperature between 3 -5 °C. The advantage here is the notable difference between the components' boiling points, thus obtaining increasingly pure vapours of refrigerating agent, as a result of the desorption process without the need of their ulterior rectification.

Logistics wise, several sensors were used, as follows:

- type K (*NiCr- Ni)* thermocouples of ± 0.25 K accuracy, for temperature monitoring for the measurement sections, as well as for the ambient air;

- flowmeters with ± 3 % accuracy, with ultrasounds for water debit monitoring on the system's different circuits;
- piranometers for measuring the total solar radiation intensity formed by direct and diffuse radiations alike;
- anemometers for measuring the air's speed;

3. Methodology

The aim of the present study was, on one hand, to determine the thermal energy prepared in the cold season and, on the other hand, to establish the boiler's supplementary input in order to ensure the temperature level for the plant's evaporator.

Based on the measured values it was determined the heat flow delivered by the solar panels $\dot{Q}_{CS}$ that is equals with the heat flow received by water $\dot{Q}_a$ in the plate heat exchanger.

The heat flow, $\dot{Q}_{CS}$, was calculated by using the following relation:

$$\dot{Q}_{CS} = \dot{m} \cdot c_{p,EG} \cdot \Delta\theta, \text{ [kW]} \qquad [1]$$

where:

- $\dot{m}$ - the masic flow rate for thermal agent in the solar panels system, [kg/s];
- $c_{p,EG}$ - the specific heat at a constant pressure of the water - etilenglicol solution in the solar plants system's circuit, for an average work temperature in the system, [kJ/(kg K)];
- $\Delta\theta$ - the temperature difference between the entrance and exit points in the plate heat exchanger, [°C];

The thermic agent's volume debit in the solar sensors system, as it was determined by using the ultrasounds flowmeters, has a value of 1,2 m^3/h. Similarly was determined also the hot water flow rate in the circuit plate heat exchanger – tank, resulting in a value of 1,8 m^3/h.

The necessary flow to be delivered by the auxiliary source, the boiler, was determined in such a way so that the thermic agent obtained would have a temperature of 15 ºC, considering the flow required by the refrigerating machine's evaporator.

4. Results

The monitoring process took place in the period between late December 2011 and late February 2012.

From within this period, present here is the values variation of the daily average heat flows as delivered by the solar plants system and by the boiler for the 16.01.2012-23.01.2012 time interval, considered to be representative for all possible situations. The heat rates are dependent on the meteorological conditions in the corresponding days, especially external temperature and solar radiation, but also on the heat accumulation resulted in the storing tank throughout the previous days.

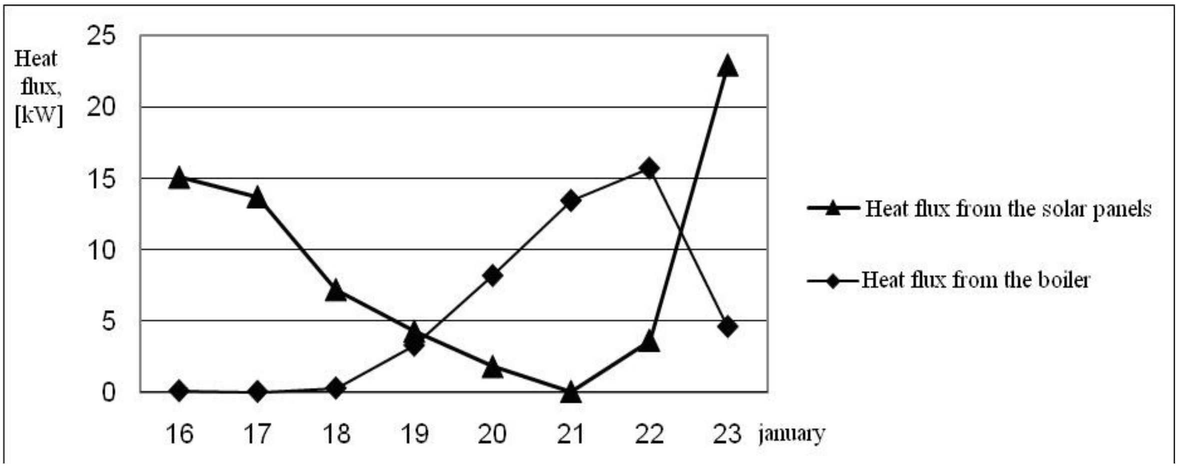

Figure 1. Variation of the yielded flows in the 16.01.2012- 23.01.2012 period

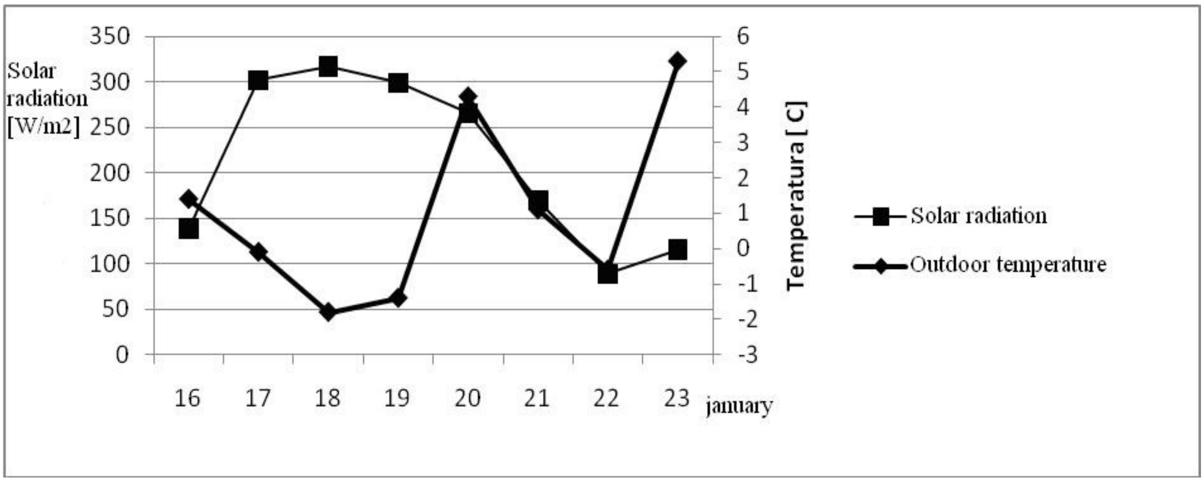

Figure 2. Variations of the ambient temperature and the solar radiation in the 16.01.2012- 23.01.2012 period

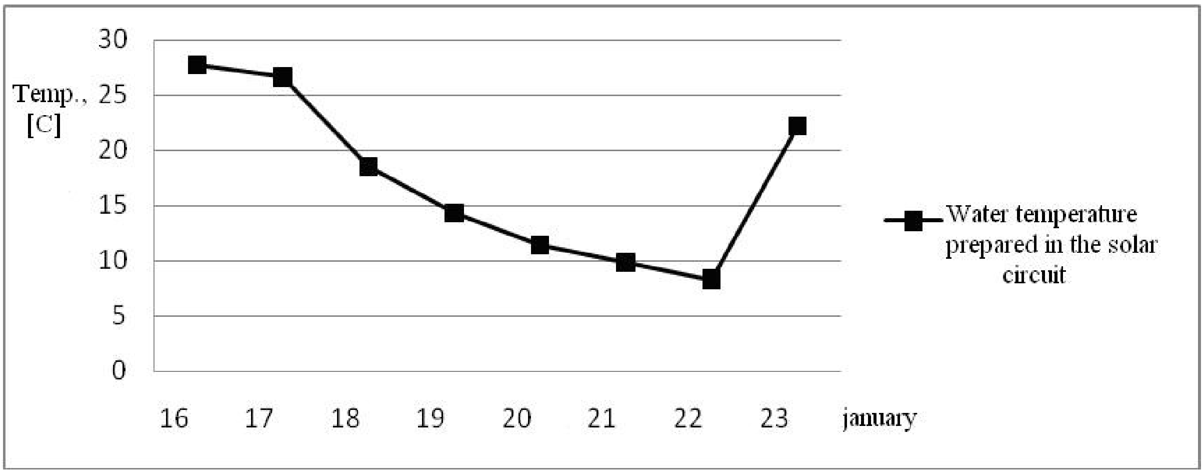

Figure 3. Variation of the warm water prepared through solar processes in the 16.01.2012- 23.01.2012 period

By analysing the profile of the heat flow yielded by the solar sensors – Figure 1 – together with the profile of the solar radiation's intensity – Figure 2 – a correlation between the two becomes noticeable, namely that in the time period 16th – 19th of January, corresponding to a maximum intensity for the period studied, the solar panels covered fully the heat required by the evaporator.

On the other hand, on the days of 20th – 22nd of January the sky was grey, which is why the flow yielded by the solar sensors reached values close to zero.

The water's average temperature when leaving the heat exchanger on the solar panels side within the time lapse of 08.00 – 16.00 o'clock, had a value of 15-20 °C.

Figure 2 illustrates that the solar radiation is not influenced directly by the external temperature, but depends on the degree of sunlight.

Figure 3 is a representation of the variation of the water's daily average temperature, where the water receives heat from the etilenglicol-water solution throughout the studied period. It can be noted that this temperature is variable, but nevertheless close to the value required by the authors for the heat pump's evaporator. In order for this variation's influence to be diminished, the plant's scheme includes a blending tank provided for driving the evaporator.

5. Discussions

In the section below the authors present the scheme of the plant which operates on hot water prepared by the solar panels and the auxiliary source in the cold season.

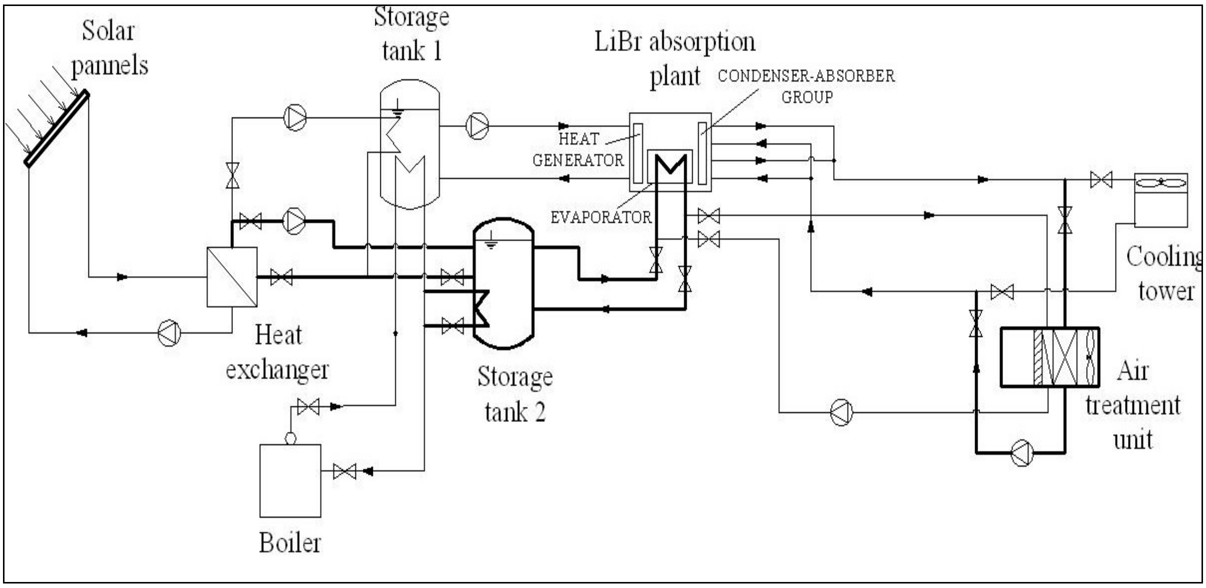

Figure 4. The plant's scheme, with focus on the circuit in winter conditions

The driving principle is as follows:

In the solar panels hot water is prepared, reaching an average temperature of 15°C, and when such a temperature cannot be reached, supplementary heat is used in the evaporator's driving tank - Storage tank 2, with the water prepared in the boiler. This water is used as energy source which enters the absorption refrigerating machine's evaporator. The absorption refrigerating machine operates as a heating pump (PC), with the generator driven by 80- 85 °C hot water, and yields condensation and absorption heat in the heating battery belonging to the air treatment plant, where the air is heated up to a temperature of 27 – 30 °C.

The thermal calculus

The absorption refrigerating plant with a lithium-water bromide solution is presented in Figure 5.

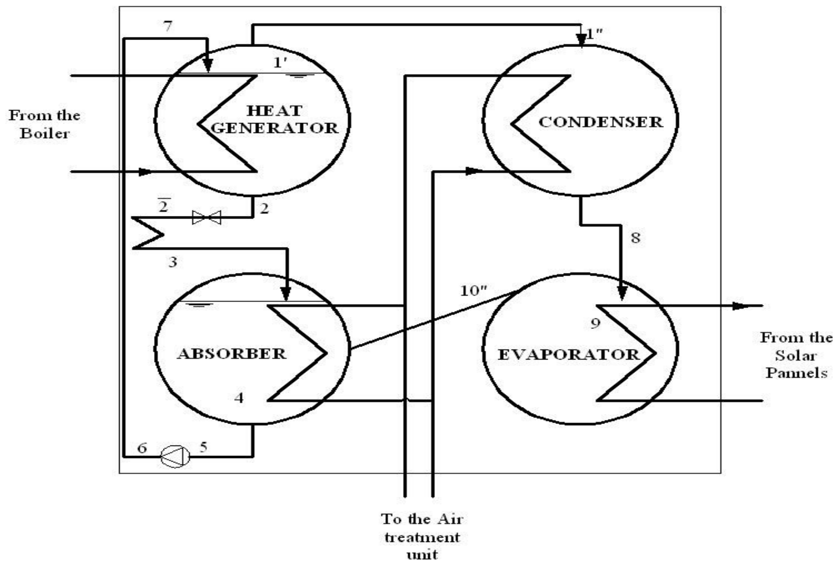

Figure 5. The absorption machine with lithium-water bromide solution

The thermodynamic cycle corresponding to the absorption refrigerating plant with a lithium-water bromide solution is represented in diagram h - ξ (Figure 6).

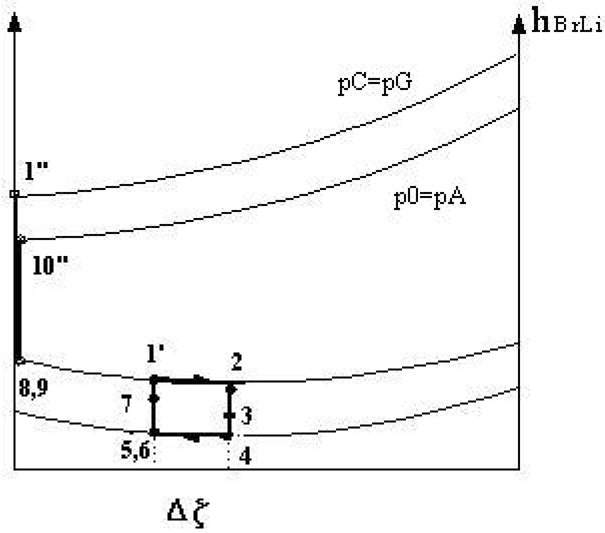

Figure 6. The plant's theoretical thermodynamic cycle

The plant is used for warming the air in habitable spaces during the cold season. The warm water, which represents the thermal agent in the air's heating battery, is waste heat resulted from cooling the condenser and absorber in the absorption refrigerating plant. In order for the plant to operate in the cold season a hot source is required, represented by warm water prepared in the solar panels and by hot water prepared in a boiler with gas fuel.

Known factors:

- refrigerating capacity, Φ_0= 12 kW*;*
- temperature values of the cooled water , $\theta_{w3} / \theta_{w4}$= 12/7 ºC;
- temperature values of the warm water to the condenser, $\theta_{w5} / \theta_{w6}$= 27/31 ºC.

Admitted factors:

- the difference between concentrations, $\Delta\xi = \xi_{SC} - \xi_{SD} > 5\%$;
- pressure loses are insignificant (the vapours' circuit is very short), respectively $p_A = p_0;$ $p_G = p_C$;
- the work pumping of the pumps, *P*, is insignificant.

Factors to be determined:

- the flow rate for the concentrated solution Q_{mSC} and diluted Q_{mSD} as well as for the refrigerating agent Q_m;
- the plant's thermal driving power, Φ_G;
- the thermal powers yielded to the cooling water Φ_A, Φ_C;
- the plant's energetic balance sheet;
- the plant's performance coefficient, COP.

The plant's operating regime (represented by the vaporisation temperature and pressure, as well as by the condensation temperature and pressure) was based on the temperature variations in the case of the working fluids in both evaporator and condenser, as well as on the minimum temperature difference between one medium and the other ($\Delta\theta_0 = \Delta\theta_C = 3$ °C), as follows:

$$\theta_0 = \theta_9 = \theta_{10} = \theta_{w4} - \Delta\theta_0 = 4 \text{ °C} \tag{2}$$

$$\theta_C = \theta_8 = \theta_{w6} + \Delta\theta_C = 34 \text{ °C} \tag{3}$$

Subsequently, based on the water tables, the vaporising and condensing pressures can be determined:

$$p_0 = f(\theta_0) = 864.8 \text{ Pa}$$

$$p_C = f(\theta_C) = 5354 \text{ Pa}$$

The difference between concentrations is determined by the temperature variations of the fluids in the boiler and absorber alike and by the minimum temperature difference between the ($\Delta\theta_G$ = 3 °C, $\Delta\theta_A$ = 5 °C). starting from diagram h-ξ-p, the solution's concentrations are determined:

$$\xi_{SC} = \xi_2 = f(p_G, \theta_2 = 80 \text{ °C}) = 61\ \%$$

$$\xi_{SD} = \xi_5 = f(p_A, \theta_5 = 32 \text{ °C}) = 54\ \%$$

From both specific thermal and energetic balance sheets for the plant's devices, the specific powers are determined (reported to the refrigerating agent's flow rate, Q_m) and consequently, by multiplying them with the above mentioned debit. Table 1 contains a synthetic presentation of mass and heat balance equations for every heat exchanger (H.E.)

Table 1

Energy balance for the heat exchangers

H.E.	Energy balance	Results
Generator	$q_G = h_{1''} + (f-1)\cdot h_{2'} - f\cdot h_7$ $Q_{mSC} = Q_{mSD} - Q_m$ $Q_m = \frac{\Phi_0}{q_0}$ $Q_{mSD}\cdot\xi_{SD} = Q_m\cdot\xi_{1''} + Q_{mSC}\cdot\xi_{SC}$ $f = \frac{\xi_{SC}}{\xi_{SC}-\xi_{SD}}$	$q_G = 3432$ [kJ/kg] $Q_{mSC} = 0.039$ [kg/s] $Q_m = 0.05$ [kg/s] $Q_{mSD} = 0.044$ [kg/s] $f = 8.71$ [-]
Absorber	$q_A = h_{10''} + (f-1)\cdot h_3 - f\cdot h_5$	$q_A = 3401$ [kJ/kg]
Evaporator	$q_0 = h_{10''} - h_9$	$q_0 = 2365$ [kJ/kg]
Condenser	$q_C = h_{1''} - h_9$	$q_C = 2413$ [kJ/kg]

The thermal powers, corresponding to each component device within the plant (generator, absorber, condenser) are determined by multiplying the masic powers by the refrigerating agent's flow rate,Q_m.

The plant's energetic balance sheet was developed according to the relation:

$$\Phi_0 + \Phi_G = \Phi_A + \Phi_C \ \text{[kW]} \tag{4}$$

$$12 + 17.41 = 17.25 + 12.24 \ \Rightarrow \ 29.41 = 29.49$$

The energetic balance is closing with a deviation of 0.2 %.

The plant's performance coefficient, COP, was determined with the relation below:

$$COP = \frac{\Phi_C + \Phi_A}{\Phi_G} = 1.69 \tag{5}$$

The heat flux provided by the condenser and absorber can be used in a air handling unit for space heating with a volume of approximately 600 m^3, in the cold season.

6. Conclusions

Based on the graphs presented by the authors, as well as on the discussions following the performance monitoring for the solar panels set in contact with the lithium-water bromide type plant in the cold season, it can be concluded that significant energy savings can be achieved by performing small constructive alterations of the plant designed for the hot season.

Throughout the monitored period, the water's daily average temperature in the solar circuit was of approximately 15 °C, and on the days with clear sky the solar panels covered totally the heat requirements for the refrigerating agent's vaporization in the cold source.

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UTILIZAREA ENERGIEI SOLARE PENTRU CLIMATIZARE IN SEZONUL RECE CU AJUTORUL UNEI POMPE DE CALDURA CU SOLUTIE BROMURA DE LITIU – APA

Abstract

In lucrarea de fata autorii prezinta rezultatele monitorizarii performantei unei instalatii frigorifice cu absorbtie cu solutie de bromura de litiu – apa la care vaporizatorul, reprezentand sursa rece, este alimentat cu apa preparata intr-un sistem de captatoare solare. Este prezentata o schema de lucru care poate fi folosita atat in sezonul cald, cat si in sezonul rece, avand ca rezultat o reducere substantiala a consumului de energie conventionala, cu efect direct in protectia mediului inconjurator.

Cuvinte cheie: surse de energie regenerabile, protectia mediului