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A solar and electrical solid sorption refrigerator

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Abstract — A new solar and electrical refrigerator based on solid sorption phenomena was designed and tested. This refrigerator has very short (15 min) non-intermittent two adsorber heat recovery cycles and uses an active carbon fibre as a sorbent bed and ammonia as a working fluid. The system management involves only the actuation of valves of special type to change the direction of the heat pipe heating circuit and the operation of water valves to change the water cooling circuit direction. A new vapour-dynamic thermosyphon with one evaporator and using alternately two coaxial condensers was designed and tested. The evaporator was built as a compact boiler. This evaporator was placed in the focus of a parabolic solar concentrator as a source of energy. As an alternative an electric cartridge heater was used inside the porous heat loaded tube immersed into the liquid of this boiler. The heat power output of such a thermosyphon is 1 000 W, with $R_{ts} = 0.03$ K·W^{-1}. A new thermal control system for the refrigerator chambers was developed. Loop bendable SS-ammonia heat pipes arranged as panels were used in combination with a compact cold store (solid sorption machine evaporator, dry-ice box). The cold output of this panel is 300 W at the temperature $t_c = -5$ °C. © Elsevier, Paris.

solar-electrical refrigerator / adsorption / heat recovery / heat pipes

Résumé — **Réfrigérateur à sorption solide couplant énergie solaire et énergie électrique**. Un nouveau réfrigérateur solaire-électrique basé sur les phénomènes de sorption solide a été conçu, puis testé. Ce réfrigérateur, basé sur deux cycles à adsorption et récupération de chaleur non intermittents, très courts (15 min), fonctionne avec un lit d'adsorbant, composé de fibres de charbon actif, et de l'ammoniac comme fluide thermodynamique. La conduite du système consiste simplement à changer la direction du circuit d'eau de refroidissement par un jeu de vannes de type spécial. Un nouveau thermosiphon dynamique avec un évaporateur et deux condenseurs coaxiaux, utilisés alternativement, a été réalisé et testé. L'évaporateur est conçu comme un bouilleur compact. Cet évaporateur, placé au foyer d'un concentrateur solaire parabolique, joue le rôle de source d'énergie. Une résistance chauffante électrique est insérée à l'intérieur d'un tube poreux immergé dans le liquide de ce bouilleur. La puissance thermique délivrée par ce thermosiphon est de 1 000 W avec $R_{ts} = 0{,}03$ K·W^{-1}. Un nouveau système de régulation thermique pour les enceintes de réfrigérateur a été développée. Des panneaux de caloducs, montés en parallèle, ont été utilisés en combinaison avec un stock compact de froid. La puissance frigorifique obtenue est de 300 W, avec une température de source froide à -5 °C. © Elsevier, Paris.

réfrigérateur solaire-électrique / adsorption / récupération de chaleur / caloducs

Nomenclature

a	adsorbed amount	kg·kg^{-1}
C	specific heat capacity	J·kg^{-1}·K^{-1}
C_{pg}	gas specific heat capacity	J·kg^{-1}·K^{-1}
C_{ps}	dry solid sorbent specific heat capacity	J·kg^{-1}·K^{-1}
E, n	phenomenological constants of the Dubinin-Astakov equation	
P	pressure	Pa
Q	heat flux	W
q_{st}	latent heat of adsorption	J·kg^{-1}
R	universal gas constant	J·mol^{-1}·K^{-1}
R	curvature radius	m
R_{ts}	thermal resistance	K·W^{-1}
r, z	co-ordinates	
S	area	m^2
S_f	fin halfstep	m
T, t	temperature	K or °C

Greek symbols

α	heat transfer coefficient	W·m^{-2}·K^{-1}
λ	conductivity	W·m^{-1}·K^{-1}
ρ	density	kg·m^{-3}
σ	surface tension	m·s^{-2}
τ	time	s

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Subscripts

a adsorbate adsorption
c condenser
d desorption
e evaporator
eff effective
f fin
HP heat pipe
m metal
s sorbent

1. INTRODUCTION

The goal of this work is R&D on solid sorption short cycles refrigerators and heat pumps using physical adsorption. The concept of solar-powered refrigeration has been analysed by Cohen and Kosar [1] and several machines operating on this principal are now commercially available. However, there has been little research into the integration of solar power with electricity, or natural gas.

Such a system could remain operational when solar insolation is low, continuing its all-year-round operation and use in areas where solar energy alone is impractical. Many solar-powered domestic water-heating systems have provision for an electrical one-person heater as a back up. Use of a gas/electricity system would be more economical and utilising gas/electricity and solar power simultaneously would reduce the cost and size of solar collectors employed in solar-driven adsorption systems.

A solid sorption refrigerator with day/night cycle to produce ice using solar energy was demonstrated by Guilleminot et al. [2]. Performance limitations of adsorption cycles for solar cooling were formulated by Critoph [3]. Grenier analysed a cold store charged by solar guided solid sorption refrigerator [4]. Bougard and Veronikis used an ammonia/active carbon solid sorption machine as solar refrigerator [5]. Passos realised the simulation of an intermittent adsorptive solar cooling system [6]. A solar refrigerator with 4 m^2 collection surface was studied by Mhiri and El Golli [7]. This last was an adsorption machine with an intermittent daily cycle that used the active carbon AC35–methanol/air.

Vasiliev tested thermal control system for the solar solid sorption machines [8]. A prototype of a solar powered cooling unit with ammonia and different salts was discussed by Speidel [9].

A combined solar refrigeration and power system offers several advantages over a pure refrigeration system alone. Excess cooling capacity can be reduced and power generation increased, thereby allowing the system to run continuously at maximum efficiency. A combined refrigeration and heating system could be operated to provide refrigeration and a salt-water desalination, refrigeration and drying, refrigeration and cooking, or refrigeration and heating. Lastly, there is a possibility of combining refrigeration, heating and electricity production.

Photovoltaic electricity generation can be also joined together with refrigeration production and be used to power other building services, such as lighting or ventilation, or it may be used to power-up an energy storage to supplement the natural gas supply.

It is interesting to consider a combined refrigeration and power-generation system with short cycles: this which would allow the operation of systems requiring power for valve operation. A combined system would allow unencumbered operation.

Thermosyphons and heat pipes are one of the most convenient heat transfer devices for the solid and liquid sorption machines due to its flexibility, high thermal efficiency, cost-effectiveness, and reliability [10]. Vapour-dynamic thermosyphons are capable of transporting heat up to 10 kW and more for the distance 50–100 m, which is difficult to achieve using conventional thermosyphons, horizontally disposed [11]. In order to avoid the flooding limit and increase the maximum performance, the vapour-dynamic thermosyphons have a tube separator inside used as a vapour conduct and a two-phase coaxial annular channel around this separator where the vapour condensation is produced with high efficiency. For the vapour-dynamic thermosyphons, it is important to know the heat transfer coefficient within the compact evaporator when transient liquid is boiling in the pool, or the liquid is boiling on the porous tubes used as heating elements inside the pool. Another important heat transfer component is a budgeting coaxial condenser with the vapour channel inside and the two-phase coaxial channel around the vapour channel.

Two-phase thermal devices which could be used as a heater and cooler alternatively for one or another supplier, heated in the evaporation zone by a constant energy source (solar, electric) and cyclically cooled in the condenser zone, are convenient for cyclic systems like solid sorption refrigerators. They are new and need to be analysed. A new vapour-dynamic thermosyphon of this type, thermally connected with a solar concentrator, was designed and tested. The solar energy generated by this thermosyphon was alternatively supplied to one or another adsorber [8]. While one adsorber was heating, the second was being cooled by the liquid heat exchanger inside the thermosyphon condenser. The thermal-hydraulic characteristics of this thermosyphon have been theoretically investigated in [11]. The characteristics for one thermosyphon with some condensers and one evaporator operating in the transitional regimes have not been theoretically studied. The first goal of this research programme is the experimental and theoretical analysis of a new solar and electrical solid sorption refrigerator. The second goal of this investigation is the experimental analysis of thermal-hydraulic characteristics of the vapour-dynamic thermosyphon with one evaporator and two condensers used to heat and to cool a solid sorption machine. The third goal is the experimental investigation of new type loop heat pipes used as cooling panels inside the refrigerator or electronic cabinet.

2. MATHEMATICAL MODEL OF TWO HEAT PIPE SORBENT BED REFRIGERATOR WITH HEAT RECOVERY

In 1987 Guilleminot, Meunier, and Pakleza [13] developed a two- dimensional model of adsorbers with solid sorption bed. In this paper this model is used with two extra assumptions: (1) the heat transfer coefficient on the inner heat pipe surface is high and uniform (10^3–10^4 W·m^{-2}·K^{-1}), and (2) this coefficient is definitely higher than the thermal resistance on the interface heat pipe-sorbent bed. The second assumption is related to the non-adiabatic conditions on the sorbent bed canister external surface.

The following assumptions were made for the mathematical model of the heat and mass transfer in a cylindrical adsorber heated and cooled by a finned vapour–dynamic thermosyphon (HP):

- the pressure in adsorber is uniform over its volume;
- the resistance of mass diffusion is insignificant;
- there is a local thermodynamic equilibrium at each moment of time and in each point of adsorber; the particles of a sorbent bed are considered as heat sources, distributed over the whole adsorber with volumetric density:

$$Q = q_{\mathrm{st}} \rho_{\mathrm{s}} \frac{\partial a}{\partial \tau} \tag{1}$$

where q_{st} is the latent heat of adsorption [J·kg^{-1}].

The thermodynamic cycle, consisting of two isobars and two isosters, was modelled by the assumption that the vapour temperature in HP in the first two stages coincides with the temperature of the external source of energy, and in the second two stages – with the temperature of a sorbent bed

$$T_{\mathrm{HP}} = T_{\mathrm{d}} \quad \text{if} \quad \tau_0 < \tau \leqslant \tau_2 \tag{2}$$

$$T_{\mathrm{HP}} = T_{\mathrm{a}} \quad \text{if} \quad \tau_2 < \tau \leqslant \tau_4 \tag{3}$$

where τ_0, τ_2 is the time of desorption and τ_2, τ_4 the time of adsorption.

From the mathematical point of view, the problem is described by a two-dimensional parabolic partial differential equation representing the law of energy conservation:

$$\rho_{\mathrm{s}}(C_{\mathrm{ps}}+aC_{\mathrm{pa}})\frac{\partial T_{\mathrm{s}}}{\partial \tau} = \frac{1}{r}\frac{\partial}{\partial r}\left(r\lambda_{\mathrm{eff}}\frac{\partial T_{\mathrm{s}}}{\partial r}\right)+\frac{\partial}{\partial z}\left(\lambda_{\mathrm{eff}}\frac{\partial T_{\mathrm{s}}}{\partial z}\right)+Q \tag{4}$$

with the initial conditions:

$$T(r,z,\tau = 0) = T_0 \tag{5}$$

and the boundary conditions:

$$\left(\lambda_{\mathrm{eff}}\frac{\partial T}{\partial r}\right)_{r=r_0} = \alpha_{\mathrm{HP}}(T_{\mathrm{HP}} - T_{r=r_0}) \tag{6}$$

where r_0 is the heat pipe/sorbent bed interface and α_{HP} is a heat transfer coefficient between inner surface of cylindrical sorbent layer and the two-phase fluid in a heat pipe with temperature T_{HP}:

$$-\left(\lambda_{\mathrm{eff}}\frac{\partial T}{\partial r}\right)_{r=r_1} = \alpha_{\mathrm{env}}(T_{r=r_1} - T_{\mathrm{env}}) \tag{7}$$

where r_1 is the outer surface of the sorbent bed canister and α_{env} is a heat transfer coefficient on this surface:

$$\frac{\partial T}{\partial n}_{z=0} = 0,\ \frac{\partial T}{\partial n}_{z=S_{\mathrm{f}}} = 0 \tag{8}$$

For the finned HP equation (4) takes the form:

$$\rho_{\mathrm{m}} C_{\mathrm{m}} \frac{\partial T}{\partial \tau} = \lambda_{\mathrm{m}} \Delta T \tag{9}$$

The equation of heat balance (4) is connected with the mass balance equation of Dubinin-Asthakov form, the equilibrium state equation [12] and the Clapeyron equation:

$$\mathrm{d}a = n\left(\frac{RT}{E}\right)^n a \ln\left(\frac{P_{\mathrm{s}}(T)}{P}\right)^{n-1}\left(\mathrm{dln}P - \frac{q_{\mathrm{st}}}{RT^2}\mathrm{d}T\right) \tag{10}$$

and the saturation curve for the coolant:

$$P = P_{\mathrm{s}}(T) \tag{11}$$

with the initial conditions:

$$P(r, z, \tau = 0) = P_0 \tag{12}$$

The boundary conditions for equation (10) during isosteric periods of adsorber operation have the form:

$$\frac{\mathrm{d}}{\mathrm{d}\tau}\iint a\,(T(r,z), P)\mathrm{d}r\mathrm{d}z = 0 \tag{13}$$

which means total adsorbent mass conservation. For isobaric periods of desorption and adsorption, the pressure is determined by constant pressures of condensation and evaporation:

$$P = P_{\mathrm{c}} \quad \text{or} \quad P = P_{\mathrm{e}} \tag{14}$$

The numerical analysis of the problem was conducted using an implicit multidimensional finite-difference scheme.

It was assumed that the process of heat regeneration is carried out only during isobaric heating and cooling periods of two adsorbers. At the beginning of the second stage for the first adsorber (the stage of isobaric heating), the second adsorber is at the beginning of the fourth stage (the stage of isobaric cooling). Thus both adsorbers are connected by a two- phase loop of low thermal resistance. Basically, this two-phase loop joined to the ends of the heat pipes, on which adsorbers were installed. The heat balance equation was used to determine t_{HP}. For a part of the heat pipe, on which 1-th adsorber is mounted, the heat balance equation assumes the following form:

$$C\frac{dt_1}{d\tau} = -q_1 + \alpha_{HP}S(t_{HP} - t_1) \quad (15)$$

Similarly, for the second HP, we can write:

$$C\frac{dt_2}{d\tau} = -q_2 + \alpha_{HP}S(t_{HP} - t_2) \quad (16)$$

where t_1 and t_2 are the temperatures of the heat pipe wall, C is the heat capacity of this part of the pipe, q_1 and q_2 are heat inputs from the wall of HP to the layer of sorbent, and S is the heating surface. The heat balance equation for the external parts of HP, and for the two-phase heat exchange loop we consider, assuming that the temperature of all elements in this equation is equal to a HP vapour temperature, looks like:

$$C_{12}\frac{dt_2}{d\tau} = -\alpha_{HP}S(t_{HP} - t_1) - \alpha_{HP}S(t_{HP} - t_2) \quad (17)$$

where C_{12} is the total heat capacity of all elements. Heat fluxes q_1 and q_2 in (15) and (16) were calculated by solving the two-dimensional equations for the one-adsorber case. Taking into account a large value of internal heat transfer coefficient inside heat pipes $\alpha_{HP} = (10^3 - 10^4\ \mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}})$, it is possible to simplify the system (15)–(17), assuming that $t_1 = t_2 = t_{HP}$. In this case we have to solve only one differential equation:

$$(2C + C_{12})\frac{dt_{HP}}{d\tau} = -q_1 - q_2 \quad (18)$$

The initial value of temperature t_{HP} is taken equal to a half-sum of temperatures t_{a1} and t_{d1}. The differential equation (18) was solved using an explicit scheme with a rather rough Euler's method on each step, enabling one to determine t_{HP} before the beginning of the next time step for the partial differential equations. The detailed comparison of the numerical analysis data and a set of experimental results will be analysed in the next paper.

3. DESCRIPTION OF THE REFRIGERATOR OPERATION

The solid sorption refrigerator utilises a solar collector (mirror), two sorbent bed canisters, connected by the heat recovery loop, a two-phase heat transfer system (vapour–dynamic thermosyphon), one condenser, two evaporators, and two cold panels (loop heat pipes) positioned inside the refrigerator cabinet.

The energy required comes mainly from the heat supplied to the solar receiver. However, a small amount of work is required by the valve system to switch on and off the vapour–dynamic thermosyphon and to heat or to cool one sorbent bed after another (two-step heat machine). This is in contrast to conventional vapour–compression systems, which require shaft work for the compression process. The main parameters of the refrigerator are included in the *table*. Devices of this type can be used as cooler and electricity generators, cooler and desalination systems, and cooler and air conditioning systems.

TABLE
The main parameters of the refrigerator.

Sorbent bed canisters dimensions	0.05 × 1.2 m^2
Active carbon fibre 'Busofit' mass in one sorbent bed	0.75 kg
Ammonia mass adsorbed in one sorbent bed	0.3 kg
Water mass in the vapour-dynamic thermosyphon	0.6 kg
Refrigerator cabinet volume	0.15 m^3
Temperature interval inside the refrigerator cabinet	0 °C–10 °C
Surface of one loop heat pipe inside the refrigerator cabinet	0.6 m^2
Ammonia mass inside the one loop heat pipe	0.05 kg
Cold output from the surface of the two loop heat pipes in the refrigerator	150–200 W
Heat output from the condenser and sorbent bed canisters	300 W
The time of the adsorption–desorption cycle	15 min

The dynamics of the solid sorption refrigerator are as follows (*figure 1*). During the heating of adsorbers, the heat is supplied to the sorbent bed by the two-phase heat transfer device-thermosyphon guided by alternate valve operation. Two special valves make it possible to switch on and off one or other adsorber (1, 2). The temperature of the carbon fibre inside the adsorber increases up to 110 °C, and the generation of ammonia at high pressure occurs in pores, since the process is endothermal. This superheated ammonia enters the condenser 5, cools, and condenses on the cold surface of the water heat exchanger 22. The liquid ammonia enters the porous evaporator 6 or 7, and is stored in the liquid reservoir, positioned inside the evaporator. This completes the first half of a cycle. During the second half of a cycle, the active water cooling of adsorber 6 or 7 starts, the sorbent bed temperature falls down to

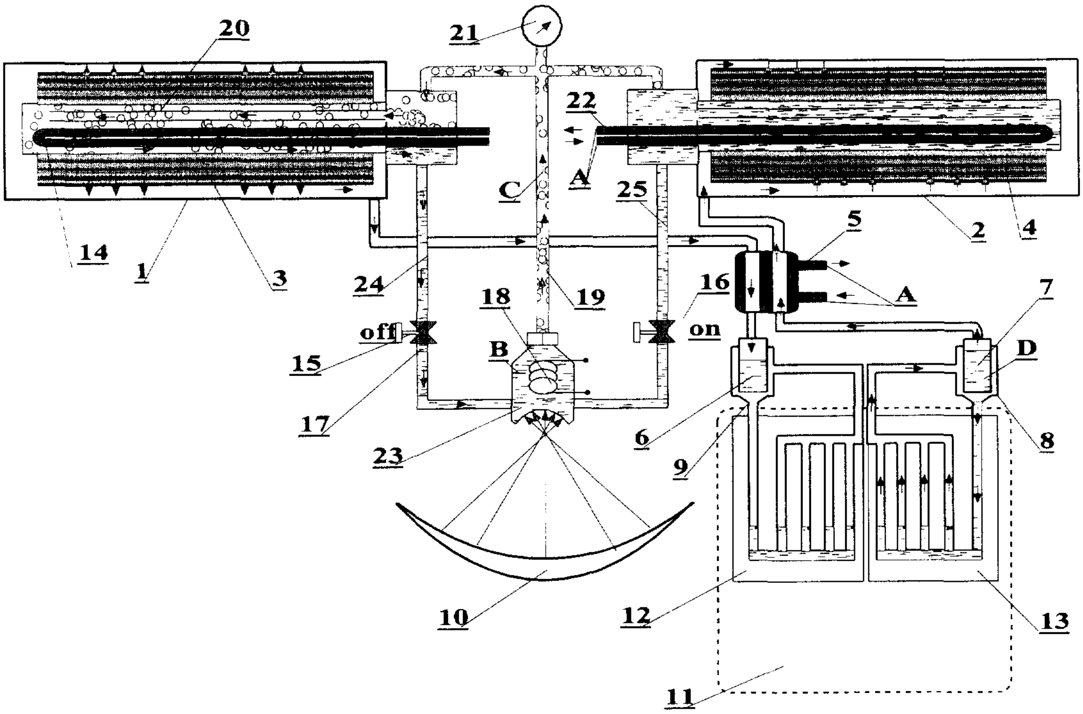

Figure 1. Schematic diagram of the solar sorption refrigerator. **1**, **2**: adsorbers; **3**, **4**: sorption beds; **5**: condenser; **6**, **7**: porous evaporators; **8**, **9**: condensers of the *spaghetti* heat pipes; **10**: parabolic solar concentrator; **11**: refrigerator chamber; **12**, **13**: *spaghetti* heat pipes; **14**, **22**: elongated cylindrical condensers; **15**, **16**: electrical valves; **17**, **24**, **25**: flexible liquid pipes; **18**: electric cartridge heater; **19**: vapour pipe; **20**: vapour channel inside the condenser of the two-phase heat transfer device; **21**: pressure gauge; **23**: boiler–evaporator of the two-phase heat transfer device. **A**: water cooling loops; **B**: water; **C**: vapour; **D**: ammonia.

20–30 °C, and the ammonia pressure falls to 0.5 bar or less. When this pressure in the adsorber becomes lower than the ammonia pressure in the evaporator, the process of the liquid ammonia evaporation inside the porous structure begins with intense evaporator wall cooling down to −30 °C.

The evaporator 6, 7 is thermally connected with the refrigerator cabinet through the loop heat pipe. The heat pipe condenser is positioned on the outer surface of the evaporator. The multi-bent heat pipe evaporating part is inserted into the refrigerator cabinet along the walls. These two heat pipes are used as a second ammonia circuit thermally connected with the first ammonia circuit (evaporators 6 and 7). When the temperature of the evaporators 6 and 7 decreases and becomes lower than the air temperature inside the refrigerator, the ammonia evaporation in the heat pipe starts, with its condensation moreover on the outer surface of the evaporator 6, 7. Heat transfer between the air inside the refrigerator and cold heat pipe panels is realised by natural convection, the temperature of heat pipe being lower −3 °C.

Periodical switching on and off of the loop heat pipe is done automatically following the adsorption/desorption cycles of the sorbent bed (15 min).

3.1. Solar concentrator

The solar concentrator (10) is made from an aluminium plate as tray (TV parabolic antenna) of diameter 1.2–1.8 m; the inner surface is covered by metallic polymer film with a high degree of reflection 0.68 (mirror).

3.2. Two-phase heat transfer device

The two-phase heat transfer device (*figure 1*) is constructed as a vapour–dynamic thermosyphon, which has one small boiler–evaporator, two elongated cylindrical condensers 14, 22 inside the sorbent canisters, vapour chamber 23 and two flexible liquid pipes 24 and 25, and one vapour pipe 19. There are two valves on the liquid pipes to regulate the boiler water feed.

The basic particularity of this two-phase temperature control system is the periodical valve-directed on-off switching of the condensers with the constant rate of the boiler heat load (*figure 2*).

The boiler (*figure 3*) has vapour generating horizontal SS porous tubes containing an electric cartridge heater inside. The experimental data acquisition system includes the temperature sensors, vapour pressure gauge and computer. The heat transfer coefficient with boiling was 3–4 times higher compared with water boiling on the smooth tubes.

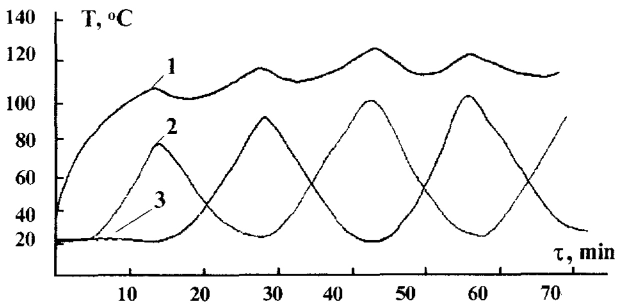

Figure 2. Temperature evolution of the boiler and adsorbers surface as a function of time. **1**: vapour-dynamic thermosyphon evaporator; **2**: adsorber No. 1; **3**: adsorber No. 2.

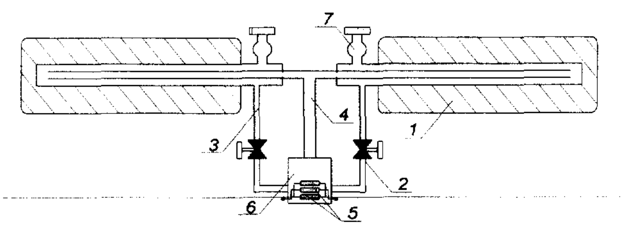

Figure 3. Schematic diagram of the vapour-dynamic thermosyphon with two condensers. **1**: adsorber; **2**: valve; **3**: liquid channel; **4**: vapour channel; **5**: SS porous tube with an electric cartridge heater; **6**: boiler; **7**: small volume and gap for non-condensed gas.

3.3. Evaporators

Two special type of ammonia porous evaporators 6 and 7 (*figure 1*) are used to ensure the temperature fall during adsorption cycle down to −30 °C. These evaporators 6 and 7 are thermally connected with loop heat pipes 8, 9. The heat pipe condensers are made as coaxial tubes on the outer surfaces of the evaporators to ensure heat pipe working-fluid (ammonia) condensation on the outer evaporator surface.

The capillary porous wick for the evaporators was performed by the gas-thermal aluminium powder sintering in the water vapour atmosphere. The metal-oxide Al_2O_3 powder wick was performed with particle diameters $(10–40)\cdot10^{-6}$ m thermally sintered together with porosity 40–50 % and pore diameter $(2–10)\cdot10^{-6}$ m (Al_2O_3 ceramic).

The total pore volume in this ceramic is:
$(0.4–0.5)\cdot10^3$ mm$^3\cdot$g.

The macro pore volume is:
$(0{,}35–0{,}38)\cdot10^3$ mm$^3\cdot$g.

The micro pore volume is:
$(0.05–0.07)\cdot10^3$ mm$^3\cdot$g^{-1}.

The permeability of such a system due to the macro pores equals (2–3) 10^{13} m^2 with the specific macro pore surface $(25–35)\cdot10^{-6}$ m$^2\cdot$g^{-1}.

The compression mechanical strength of such Al_2O_3 ceramic is not less than 30–40 mPa. Ceramic of this kind is a good adsorbent for the ammonia vapour and the capillary pressure $\Delta P_{\text{cap}} = 2(\sigma_{\text{sg}} - \sigma_{\text{sl}})/R$ is near 0.5 bar.

3.4. Solid sorbent canisters

The two solid sorbent canisters 3 and 4 are filled with active carbon fibre "Busofit" wrapped on the surface of condensers 14 and 22 between the aluminium fins. The length of the canister is 1.2 m with the outer diameter 50 mm.

3.5. Condenser

The ammonia condenser 5 is constructed as a stainless steel tube- in-tube heat exchanger cooled by water to ensure complete vapour condensation and liquid ammonia cooling down to the room temperature.

3.6. Refrigerator cabinet

The refrigerator cabinet has a volume $V = 150$ L and is used to cool the goods in the temperature interval 0–10 °C. Two mild steel loop heat pipes 8, 9 are made from finned pipe having outer diameter 5 mm and inner diameter 3 mm. The heat pipe evaporators are made as a finned pipe heat exchanger positioned on the two lateral surfaces inside the refrigerator chamber. The total heat transfer surface of two looped heat pipe evaporators is 1.2 m^2. The air temperature evolution in time inside the evaporator cabinet for different temperatures of the cooling water in the condenser 5 is shown in *figures 4–5*. There is no strong dependency of the air temperature inside the refrigeration cabinet for 14 and 22.5 °C on the cooling water temperature.

3.7. Loop heat pipes as cooling panels for the refrigerator cabinet

To maintain good food quality, cold chambers are increasing needed. To fulfil these needs, the loop heat pipe panels were used as uniform temperature sheets inside the cold chamber connected with a cold store unit (solar sorption refrigerator, cold accumulator). Two loop heat pipe panels bent as an L-shape were installed inside the refrigerator cabinet. The solar sorption refrigerator evaporator was sited on the upper

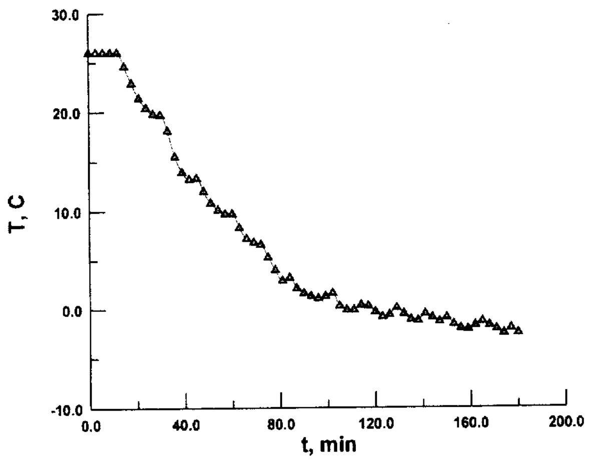

Figure 4. The average temperature inside the refrigerator cabinet as a function of time (temperature of the condenser cooling water 14 °C).

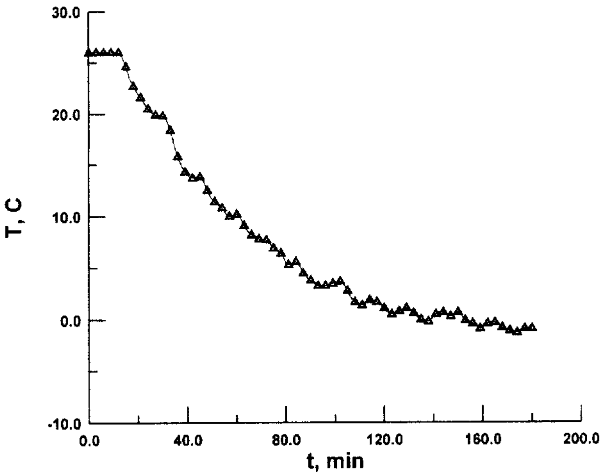

Figure 5. The average temperature inside the refrigerator cabinet as a function of time (temperature of the condenser cooling water 22.5 °C).

part of the cabinet. The heat pipe panels on its upper side (heat pipe condenser) have good thermal contact with this evaporator; the extended lower part of the heat pipe panels is used as heat pipe evaporator. This type of heat pipes with diameter 3 mm and the length near 1 m has no capillary structure inside, and functions under the oscillating motion of the two-phase ammonia because there is a big difference between the liquid and vapour densities under the heat load. The driving force of the loop heat pipes is the pressure force generated by the liquid boiling in the high temperature zones (lower part of the refrigerator cabinet), the non-equilibrium state between vapour and liquid, where vapour bubbles collapse in the upper cold part of the panel. Vapour plugs (bubbles) push the liquid plugs to the cold part of the unit where vapour bubbles collapse with the increasing pressure difference between vapour and liquid. Due to the interconnections between the two-phase channels, the motion of vapour bubbles and liquid plugs is influenced by the motion of fluid in the adjacent sections on the loop heat pipe panel. Two different kinds of loop heat pipe panels were used. The first one is shown in *figure 6*, the second in *figure 7*.

The total surface of the two loop heat pipe panels inside the refrigerator cabinet was 1.2 m^2; the ammonia mass inside one panel was 50 g. Cold output from the

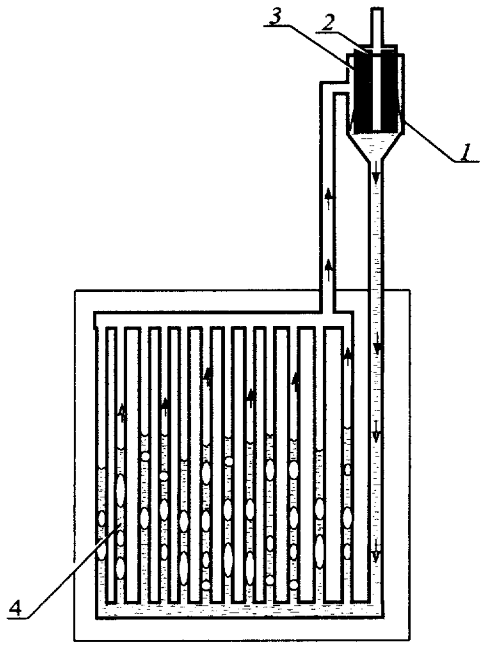

Figure 6. Diagram of first kind of loop heat pipe panel. **1**: condenser of the heat pipe, **2**: evaporator of the adsorption refrigerator, **3**: porous structure, **4**: heat pipe evaporator.

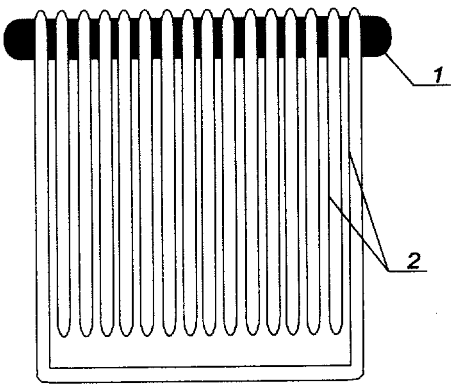

Figure 7. Diagram of the second kind of loop heat pipe panel. **1**: condenser zone of the heat pipe; **2**: evaporator part of the heat pipe.

heat pipe panels was 300 W. Due to the cycling action of the solar refrigerator, the time of the active panel cooling was limited to 15 min for one cycle. The temperature deviation of the two loop heat pipe panels inside the refrigerator cabinet for the time interval 0–200 min is shown in *figure 8*. The ambient (room) temperature was 37 °C.

4. CONCLUSIONS

A solar and electrical solid sorption machine with 1.2 m^2 collection surface was designed and studied to build an industrial refrigerator/dryer in the future. The ratio between solar and electrical energy supply was automatically maintained on the total level of 1 kW. The machine is an adsorption machine with very short (15 min) non-intermittent cycles that uses the active carbon fibre "Busofit" as a sorbent bed and ammonia as a working fluid. Air is used as a medium to cool (inside the cabinet) and water as a medium to heat (inside the adsorbers and the condenser). The system management consists only in actuating the special type valves to change the heating circuit and water valves to change the water cooling circuit. With solar collector efficiency of nearly 0.7, it is possible to obtain a high solar COP of close to 0.3. There is considerable scope for the application of such hybrid electricity/solar-adsorption systems where intermittent or low solar insulation currently restricts their use. Such working fluid and energy sources are attractive on environmental grounds. Calculations for a combined refrigeration and power-generation system operating on electrical/solar or gas/solar energy with heat recovery shows that a COP of 0.75 is achievable. Further increases in performance may be attained using multi-effect systems.

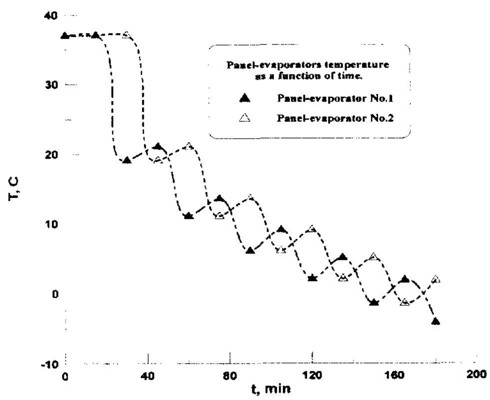

Figure 8. Temperature evolution of the loop heat pipe panels inside the refrigerator cabinet. Panels are cooled alternatively by the solid sorption evaporators 6 and 7.

Research into the economical viability of these systems is required to assess their commercial potential. Any such research should take into account anticipated changes in legislation governing the use and manufacture of conventional refrigerators.

A new SS-water vapour-dynamic thermosyphon with two coaxial condensers was tested as a new thermal control device for the solar sorption refrigerators with heat load 1 kW and thermal resistance $R_{ts} = 0.01$ K·W^{-1}.

A new type loop SS-ammonia heat pipes $d = 3$ mm and $L = 1$ m panel was developed and tested as a possible alternative to the aluminium flat panel made by the Roll-Bond technology for the refrigerator cabinets. The cold output for such panels was near 300 W.

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