# Cyclical reverse thermosiphon

**YURIY DOBRIANSKY**[1*]
**YIGZAW G. YOHANIS**[2]

[1] *Department of Engineering and Technical Sciences, University of Warmia and Mazury in Olsztyn, Oczapowski St. 11, 10-736 Olsztyn, Poland*

[2] *Centre for Sustainable Technologies, School of the Built Environment, University of Ulster, Shore Road, Jordanstown, Newtownabbey, Northern Ireland, UK*

**Abstract**    We describe the development of a new type of heat exchanger. This heat exchanger operates using reverse thermosiphon action and consists of a self-acting and self-controlled liquid circulation loop with heat transfer in a downward direction, opposite to the direction of natural convection. This process moves a heat-carrying hot liquid downwards with the help of local heat transferred through the loop. This flow loop is partly filled with liquid and the upper part of the loop contains vapour from the liquid heat-carrier. The pressure difference in the saturated vapour is used to move the heated liquid downwards. The principles of action and the possibility of developing such a device using laboratory experimental methods are presented.

**Keywords:** Reverse thermosiphon; Liquid circulation loop; Passive heat transfer downward

## Nomenclature

$a$    –    acceleration, m/s$^2$
$B$    –    exergy, J
$c$    –    heat capacity of water, J/(kg K)

---

$^*$Corresponding author. E-mail address: dobr@uwm.edu.pl

$D, d$     –    diameter, m
$F, f$     –    area, m$^2$
$g$        –    acceleration of gravity, m/s$^2$
$L$        –    length, m
$m$        –    mass, kg
$N$        –    force, N
$p$        –    pressure, Pa
$Q$        –    amount of heat, J
$r$        –    latent heat of vaporization, J/kg
$t$        –    temperature, $^\circ$C
$V$        –    volume, m$^3$
$W$        –    capacity, W
$w$        –    velocity, m/s

**Greek symbols**

$\eta_{rec}$   –    degree of heat recirculation
$\lambda$      –    hydraulic resistance coefficient
$\rho$         –    density, kg/m$^3$
$\tau$         –    time, s

**Superscripts**

$'$        –    boiling
$''$       –    saturated

**Subscripts**

$A$        –    hot lowering pipe
$B$        –    cold lifting pipe
$col$      –    cooling
$hyd$      –    hydraulic
$m$        –    average
$pour$     –    poured
$push$     –    pushed
$sour$     –    source
$vap$      –    vapour

# 1   Introduction

Circulation loops that use the natural convection of liquid heat-carriers are
often used in devices for heat transfer. Self-acting heat transfer occurs
due to the action of gravitational forces and the expansion of the heat-
carrier, causing a decrease in density as the temperature rises. The hotter
and therefore lighter part of the liquid rises in the gravitational force field,
resulting in natural convection.

If the heating causes evaporation, the difference between the density of the liquid and that of the vapour is significantly higher and natural convection is strongly intensified. Devices that use this phenomenon are known as two-phase thermosiphons. The main advantage of liquid circulation loops with natural convection and thermosiphon action is that of self-action; however, these devices only transfer heat upwards.

Some technical proposals have shown downward convective heat transfer, i.e., transfer in the direction opposite to that of natural convection. Roberts [11] has reviewed ten different technical solutions but only one is widely used in practice. The system consists of a mechanical liquid pump that operates using external energy sources. The main drawback of this system is that it does not act autonomously.

Other known devices that operate autonomously and transfer heat downwards use vapour as a heat carrier and saturated vapour pressure to create lift for the condensed fluid. These devices are applied in solar installations and for the pumping of water with the help of solar collectors [3,12-14]. The heat carriers used in such devices are refrigerants like Freon R-114 or pentane. In this case, the heat transfer height is relatively low (about few meters) since the difference between the density of the liquid and vapour is too large, requiring high vapour pressure difference and high temperature difference between the hot and cold branches. The device has similar shortcomings, since the vapour pressure initially compresses the inert gas and then this gas pushes the condensed fluid upwards to the heater for boiling. This device has been tested under laboratory conditions at a temperature difference of 10-25 °C and has lifted liquid to a height of 1.7 m [1]. Methanol was used as the heat carrier for this device.

The use of a large quantity of refrigerants, hydrocarbons or methanol is undesirable because these materials pollute the atmosphere and some are inflammable. Other proposed passive downward heat transfer systems also have significant drawbacks and demonstration of such systems has not yet been published.

Based on the above information, the aims of this paper are:

- to develop a simple self-acting device that will transfer heat downward to a depth of 10 m with a temperature difference between the hot and cold branches of only a few degrees;

- to test the performance of the device under laboratory conditions and after integration in a solar installation.

## 2  Physical grounds of operation

In devices using vapour as a heat carrier, the main disadvantage is the high pressure difference (around 10 kPa), which results in the requirement of a high temperature difference (around 10 degrees) for transportation of the liquid heat carrier to a height of a few meters.

If the liquid is used as a heat carrier, a high pressure difference is not required for liquid motion. The reason for this is that when both the hot and cold branches are filled with liquid, the difference in liquid density is small in the hot and cold branches. For example, when the temperature difference is 5 K, the height of the loop is 10 m, the temperature is within 20 °C to 100 °C, and the heat carrier is water, then the operating pressure of natural convection (which depends on temperature difference) and the height of circulation loop is only equal to 0.1–0.4 kPa and 12–38 mm $H_2O$, respectively. In contrast, the saturated water vapour pressure difference ranges from 0.85–20 kPa for the same temperature difference and same temperature range. Therefore, the saturated vapour pressure difference between the hot and cold branches for liquid pumping overcomes the natural convection forces and moves the hot liquid in a direction opposite to the direction of natural convection.

### 2.1  Principles of operation and scheme of the device

In order to use vapour pressure to drive the action, space is required for the vapour. Therefore, the upper part of the circulation loop must not be filled with liquid. For such a loop, one branch (A) is referred to as the hot branch and the other (B) as the cold branch; the liquid level in the hot branch is maintained as slightly higher than the level in the cold branch (see Fig. 1a). This difference in levels depends on the temperature difference and the height of the loop and can be calculated from the pressure equilibrium at the common points in both branches at the bottom of the loop. In this case, the upper part of the loop is open and pressures of the volatile phase above the liquid in the hot and cold branches are the same ($p_A = p_B = p_{AB)}$):

$$p_A^{botom} = p_B^{botom}, \tag{1}$$

$$p_A^{botom} = p_{AB} + \rho(t_A)gh_A, \tag{2}$$

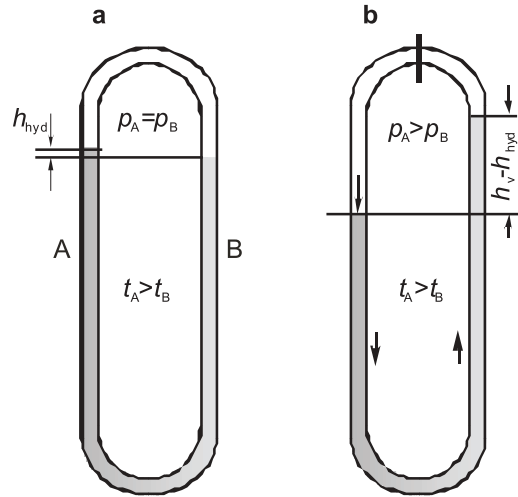$$p_B^{botom} = p_{AB} + \rho(t_B)gh_B. \tag{3}$$

Figure 1. Liquid levels in the loop filled partly by liquid: a – action of liquid density difference when upper part of loop is open, b – action of saturated vapour pressure difference and liquid density difference when upper part of loop is blocked (e.g., by control valve).

From Equations (1) – (3), we obtain:

$$h_B - h_A = 2h_m \frac{\rho(t_A) - \rho(t_B)}{\rho(t_A) + \rho(t_B)}, \tag{4}$$

where $h_m = (h_A + h_B)/2$.

These values are shown in Tab. 1 and Fig. 2 with water used as a heat carrier.

When the upper part between the branches is blocked, the saturated vapour pressure within each branch is established according to the temperature. The difference in the liquid levels between the cold and hot branches can be calculated from the same principle of pressure equilibrium at common points at the bottom of the loop, as shown below:

$$p_A^{botom} = p_A'' + \rho(t_A)gh_A, \tag{5}$$

$$p_B^{botom} = p_B'' + \rho(t_B)gh_B. \tag{6}$$

From Equations (1), (5) and (6), we obtain:

$$h_B - h_A = 2h_m \frac{\rho(t_A) - \rho(t_B)}{\rho(t_A) + \rho(t_B)} + 2 \frac{p_a'' - p_B''}{[\rho(t_A) + \rho(t_B)]g}. \tag{7}$$

Table 1.  Pressure difference between hot and cold branches and difference in liquid levels
        between cold and hot branches when temperature difference and height of loop
        are 5 K and 10 m, respectively.

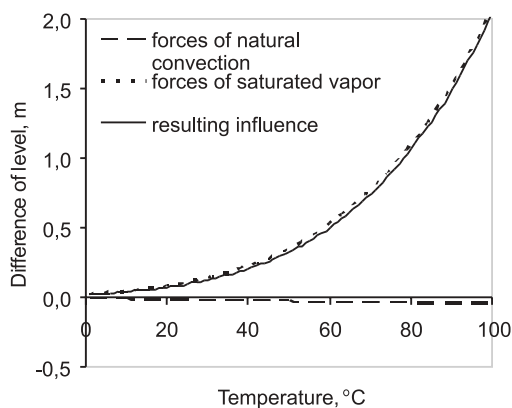| Temperature in the cold branch | Difference in saturated vapour pressure between hot and cold branches | Difference in liquid levels, which can cause forces of natural convection | Difference in liquid levels, which can cause difference in saturated vapour pressure without contraction of natural convection | Resultant difference in liquid levels between cold and hot branches, when upper part of loop is blocked |
|---|---|---|---|---|
| [°C] | [kPa] | [m] | [m] | [m] |
| 20 | 0.83 | -0.0116 | 0.085 | 0.073 |
| 40 | 2.21 | -0.0203 | 0.227 | 0.207 |
| 60 | 5.10 | -0.0269 | 0.529 | 0.502 |
| 80 | 10.45 | -0.0328 | 1.098 | 1.065 |
| 100 | 19.48 | -0.0382 | 2.076 | 2.038 |



Figure 2.  Pressure difference between hot and cold branches and difference of liquid levels
        between cold and hot branches when temperature difference is 5 K and loop
        height is 10 m.

The pressure difference and difference in liquid levels between the hot and
cold branches are shown in Tab. 1 when water is used as a heat carrier and
the temperature difference is 5 K.

When the liquid level in the cold branch rises, there is a possibility that the excess cold liquid pours off from the cold branch to the top of the hot branch. This phenomenon takes place when an additional intermediate canal is located between the cold and hot branches (see Fig. 3b) and the upper part of circulation loop is unblocked for the same vapour pressure in both branches.
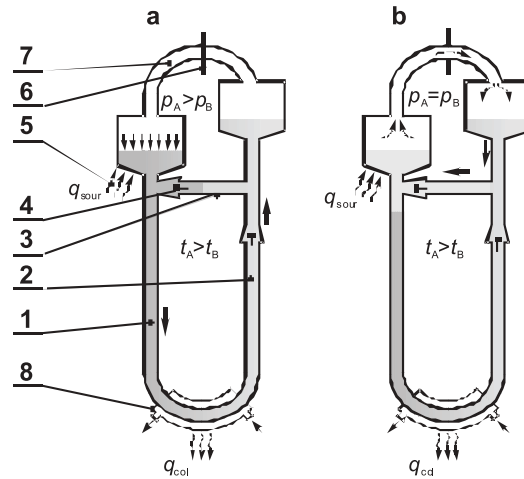


Figure 3. Scheme and stages of action of the device: 1 – hot lowering pipe, 2 – cold lifting pipe, 3 – intermediate canal, 4 – non-return valve, 5 –source of heat, 6 – control valve (liquid seal), 7 – upper part of the loop, 8 – cooler, a – push stage (hot liquid moves downward), b – pouring off stage (cold liquid moves to hot branch).

Therefore, the device must operate in a cycle with two stages:

- heating the upper part of the hot branch and pushing the hot liquid downward results in a rise in the liquid level in the cold branch (see Fig. 3a),

- opening the upper part of the circulation loop results in the same vapour pressure in both branches and the pouring off of excess cold liquid through the intermediate canal to the upper part of the hot branch via gravity (see Fig. 3b).

The upper parts of each branch must be widened to allow a greater portion of liquid to be pumped during each cycle. The directions of liquid flow in

the main part of the loop and intermediate canal are maintained by placing
non-return valves. A special control valve should be used for periodical
opening and closing of the upper part of the circulation loop. A liquid seal
can be used as the control valve (Fig. 4). The liquid seal is filled with liquid
through siphon action (3) and vapour flow will be blocked when the liquid
level reaches the upper bend of the siphon (3). The passage between the hot
(1) and cold (2) branches opens through the bypass (5) when the pressure
in the hot vessel becomes greater than is needed to push liquid out from
the bypass (5).



Figure 4. Control valve in a form of liquid seal: 1 – hot vessel, 2 – cold vessel, 3 – siphon
filling, 4 – vessel for liquid reserve, 5 – vapour bypass tube.

The device can be named according to these principles as:

- self-acting circulation loop with convection opposite to the direction
  of natural convection,

- self-acting and self-controlled pump for passive heat transfer down-
  ward, powered by local low-potential heat,

- cyclical reverse thermosiphon (according to the classification of heat
  pipes presented in [2]).

## 2.2   Some bases and explanations

### 2.2.1   Preliminary confirmation of possibility of circulation

It can be seen from Tab. 1 and Figs. 2 and 3 that natural convective forces
lift the hot liquid up to a very small level. However, the motion of natural

convection occurs when this small rise of liquid level is poured off to the cold branch. This phenomenon is often observed in nature and is used in many technical installations. The difference in forces caused by saturated vapour pressure is much stronger (by a factor of hundreds) than the difference in forces of natural convection. Therefore, the circulation loop within the reverse thermosiphon will operate under hydraulic conditions for which natural convection operates, even under adverse conditions.

### 2.2.2 Maximum vertical distance of heat transfer

The difference in saturated vapour pressure that forces hot liquid into motion depends only on the difference in temperature. At the same time, the opposing forces of natural convection also depend on the difference of temperature, but increase further when the height of the loop increases. At a certain height, the forces are equal and the natural convective forces will be higher than the forces caused by saturated vapour pressure. The maximum vertical distance of heat transfer can be calculated when the difference of levels in the loop defined by Eq. (7) is equal to zero. From Eq. (7) we obtain:

$$h_{max} = \frac{p''_A - p''_B}{[\rho(t_b) - \rho(t_A)]g}.$$ (8)

The maximum vertical distances of heat transfer calculated for different work agents are shown in Fig. 5.
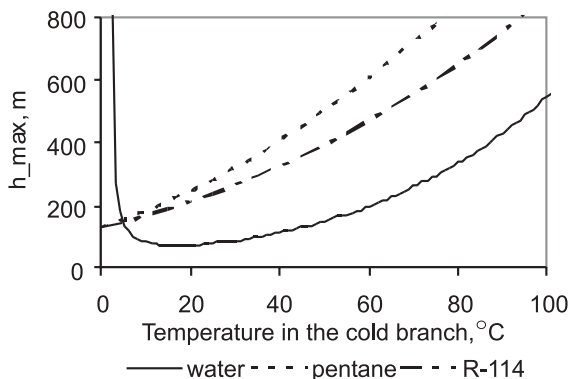


Figure 5. Maximum vertical distance of heat transfer calculated at temperature difference of 5 K for different work agents.

### 2.2.3   Estimation of thermosiphon characteristics

The following data set has been used for the proposed device:

- portion of pumping $\Delta V = 0.001$ m$^3$

- duration of pushing of hot liquid downward $\tau_{push} = 15$ s

- duration of pouring off of surplus cold liquid $\tau_{pour} = 5$ s

- duration of a cycle $\tau_c = \tau_{push} + \tau_{pour} = 20$ s

- temperature difference between the branches $\Delta t = 5$ K

- heat-carrier fluid water

- internal diameter of the circulation loop pipes $d = 0.02$ m

- cross-sectional area of the pipes $f = 0.000314$ m$^2$

- internal diameter of wide part of the hot branch $D = 0.1$ m

- cross-section area of wide part of the hot branch $F = 0.0079$ m$^2$

The amount of heat transfer during one cycle is:

$$Q = \Delta V \rho c \Delta t = 0.001 \times 1000 \times 4187 \times 5 = 20935 \text{ J},$$

where $\rho$ – density of water and $c$ – specific heat of water.
The average heat flux is:

$$W = Q/\tau_c = 20935/20 = 1047 \text{ W} = 1.05 \text{ kW}.$$

If the temperature difference increases to 20 K, the heat flux increases to 4 kW. The additional increase in the portion of pumped liquid of up to 0.0025 m$^3$ (2.5 l) in one cycle causes the increase in heat capacity of up to 10 kW. The sizes of the device will be small enough for corresponding electrical circulation pump with its control system.

### 2.2.4  Inertia forces

The mass of the heat-carrier needs to be accelerated and to be put into motion at the beginning of every cycle. For example, when such a device is used in solar installation and the height, $L$, is 15 m, the volume of the tubes are:

$$V_0 = 2Lf = 2 \times 15 \times 0.000314 = 0.0094 \text{ m}^3 \ .$$

The mass of liquid in the tubes is:

$$m_{tubes} = V_{tubes}\rho = 0.0094 \times 1000 = 9.4 \text{ kg.}$$

If a solar installation has 3–4 collectors and each collector contains 0.001 m$^3$ of liquid, the total mass including the liquid mass in other elements will be equal about 15 kg.

The average liquid velocity in tubes will be equal to:

$$w = \Delta V/(f_{push}) = 0.001/(0.000314 \times 15) = 0.212 \text{ m/s.}$$

If the duration of acceleration ($\tau_{acc}$) is 2 s, then the acceleration is:

$$a = (w - 0)/\tau_{acc} = (0.212 - 0)/2 = 0.106 \text{ m/s}^2$$

and the mean acceleration force is:

$$N = m_{inst}a = 15 \times 0.106 = 1.59 \text{ N.}$$

The additional pressure acting on the liquid surface in the wide part of hot branch is:

$$p_{acc} = N/F = 1.59/0.0079 = 202 \text{ Pa} = 20 \text{ mm H}_2\text{O.}$$

When the heat flux is equal to 4 kW, the portion of pumped liquid during one cycle was equal to 0.001 m$^3$ and the temperature difference was equal to 20 K, the additional pressure is very low.

It can be seen that the inertia force due to the cycle behaviour of the device operation does not cause any serious problems.

### 2.2.5  Hydraulic resistance

Liquid flow and heat transfer are strongly influenced by hydraulic resistance. The hydraulic resistance depends on flow velocity and the properties of the canal and liquid, as shown below [10]:

$$\Delta p = \frac{\lambda L w^2 \rho}{2d}, \tag{9}$$

where $\lambda$ is the hydraulic resistance coefficient, which depends on Reynolds number:

$$\mathrm{Re} = \frac{wd}{\nu} = \frac{0.212 \times 0.02}{1 \times 10^{-6}} = 4246$$

and $\nu$ represents kinematic viscosity which, for water at a temperature of 20 ºC, is equal to $1 \cdot 10^{-6}$ m$^2$/s.

In the case of turbulent flow, the critical Reynolds number is Re > 2300. The hydraulic resistance coefficient in a smooth pipe is calculated as:

$$\lambda_L = 0.3164 \times \mathrm{Re}^{-025} = 0.0392. \tag{10}$$

Therefore, the hydraulic resistance is:

$$\Delta p = \frac{\lambda L w^2 \rho}{2d} = \frac{0.0392 \times (15 \times 2) \times 0.212^2 \times 1000}{2 \cdot 0.02} = 1320 \text{ Pa (13 cm H}_2\text{O)}. \tag{11}$$

The above value is significantly lower than the saturated vapour pressure difference between the branches.

### 2.2.6  Work necessary for lifting of liquid

A certain amount of energy is needed to lift the liquid heat-carrier into the cold branch. This work depends on the mass of the lifted liquid and the lift height. If the lift height ($\Delta h$) is equal to 0.2 m, the work done during one cycle for 1 kg of water is:

$$L_{lift} = mg\Delta h = 1 \times 9.81 \times 0.2 = 2 \text{ J} \tag{12}$$

The power is:

$$W_{lift} = L_{lift/\tau_c} = 2/20 = 0.1 \text{ W}. \tag{13}$$

The heat flux at during this same time period is above 1 kW. Therefore, the work required for lifting of liquid is insignificant (0.01%) compared to the power of the heat transfer and thus can be neglected. The analysis of thermal-to-mechanical energy conversion efficiency is not presented.

### 2.2.7  Heat recirculation

When the upper part circulation loop is unblocked, excess cold liquid pours off to the hot branch. Hot vapour then flows into this free space through the upper part of the loop, and thus some of the heat contained within the

vapour will not be directed to the cooler but to the cold branch directly. The vapour will then be condensed, releasing its latent heat to the cold liquid, which enters the cold vessel from the cooler. Generally, this energy will not be lost because it returns to the hot vessel at the next stage through the intermediate canal. Therefore, less heat will be needed for heating of liquid in the hot vessel. Nevertheless, this situation is undesirable because the temperature of the hot vapour will decrease during cooling and condensation in the cold vessel, and only the quality of heat (exergy) will be lost.

The degree of heat recirculation caused by vapour flow from the hot to cold vessel is estimated using the coefficient of recirculation. The degree of heat recirculation is defined as the ratio of the heat of vapour transferred from the hot vessel to the cold vessel to the total heat from the hot vessel:

$$\eta_{rec} = \frac{Q_{vap}}{Q + Q_{vap}}. \tag{14}$$

To simplify estimation, it was assumed that the vapour volume that passed from the hot to the cold vessels was equal the volume of liquid pumped during each cycle plus a small volume of condensate produced from vapour. This additional volume is very small, since the specific volume of liquid is a thousand times less than vapour; thus, this additional volume can be ignored. The hot liquid leaving the hot vessel during a pushing stage and the vapour leaving a hot vessel during a pouring off stage have the same temperature, equal to $t_A$. The cold liquid temperature coming out from the cooler to the cold vessel is $t_B$. The cold liquid becomes slightly hotter due to vapour condensation and the temperature is $t'_B$. At this temperature, liquid will pour off from the cold to the hot vessel. A schematic of the heat and mass transfer is shown in Fig. 6.

The amount of heat that the liquid heat-carrier receives in the hot vessel is:

$$Q = mc(t_A - t'_B). \tag{15}$$

The heat contained in the hot vapour is used to heat liquid from temperature $t'_B$ to $t_A$ and the heat for vaporization is:

$$Q_{vap} = m_{vap}c(t_A - t'_B) + m_{vap}r. \tag{16}$$

The amount of heat that the cold liquid gains from vapour for a temperature rise from $t_B$ to $t'_B$ is:
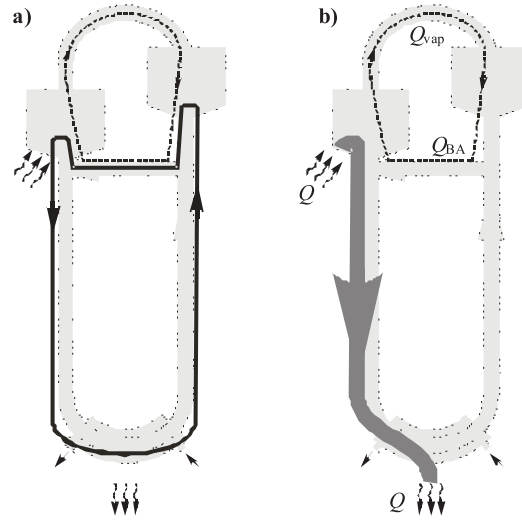
$$Q_{BA} = mc(t'_B - t_B). \tag{17}$$

Figure 6. Schematic diagram of the mass a) and heat b) transfer used to determine the
heat recirculation coefficient: — flow of liquid heat carrier, - - - flow of vapour
and its condensate.

By equating $Q_{vap} = Q_{BA}$, it is possible to calculate the value of intermediate
temperature $t'_B$:

$$t'_B = \frac{m_{vap}ct_A + m_{vap}r + mct_B}{m_{vap}c + mc}. \tag{18}$$

The calculation shows that when the temperature is between 20 and 100 °C,
and the temperature difference $(t_A - t_B)$ between the branches is equal to
5 K, the amount of heat carried by vapour from the hot vessel to the cold
vessel is equal to 0.2–7.2% of the total heat that was introduced into the
hot vessel (see Fig. 7 and Tab. 2).

The degree of exergy losses in vapour is noteworthy (0.32) at low tem-
peratures; however, the total degree of exergy losses are insignificant (less
than 0.003) because the value of the recirculation vapour is very small, par-
ticularly at low temperatures, as shown in Tab. 3. The values of the exergy
$B$ of heat were calculated by [9]:

$$B = Q\left(1 - \frac{T_0}{T}\right), \tag{19}$$

where $T_0$ is the absolute temperature of the environment $(T_0 = 10 + 273.15)$
and $T$ is the absolute temperature of the investigated medium.

Table 2. Heat flows during a single cycle and heat recirculation coefficient with dependence on temperature.

| Temperature of hot vessel, $t_A$ | Temperature of cooler, $t_B$ | Temperature of pouring cold heat carrier, $t'_B$ | Mass of vapour, $m_{vap}$, | Heat carried downward, $Q$ | Heat of vapour, $Q_{vap}$ | Recycled heat, $Q_{AB}$ | Coefficient of recirculation, $\eta_{recyr}$ |
|---|---|---|---|---|---|---|---|
| [°C] | [°]C | [°C] | [kg] | [J] | [J] | [J] | [%] |
| 25.51 | 20 | 20.01 | 1.73E-05 | 20891 | 44 | 44 | 0.21 |
| 45.53 | 40 | 40.03 | 5.12E-05 | 20806 | 129 | 129 | 0.62 |
| 65.58 | 60 | 60.08 | 1.30E-04 | 20606 | 329 | 329 | 1.57 |
| 85.18 | 80 | 80.18 | 2.94E-04 | 20195 | 740 | 740 | 3.54 |
| 105.36 | 100 | 100.36 | 5.98E-04 | 19427 | 1508 | 1508 | 7.20 |



Figure 7. Coefficient of heat recirculation with dependence on temperature.

## 3  Experimental testing

The aims and objectives of the tests were as follows:

- to confirm workability of the proposed devices by:

  - testing performance under laboratory conditions,
  - testing performance when the device is integrated with a solar installation,

- to investigate the possibility of using water as a work-agent,

Table 3. Change of exergy with dependence on temperature.

| Exergy carried by liquid, $B$ | Exergy of vapour, $B_{vap}$ | Exergy of poured off liquid, $B_{BA}$ | Degree of loses of the vapour exergy, $(B_{vap}-B_{BA})/B_{vap}$ | Total degree of loses of the exergy $(B_{vap}-B_{BA})/(B+B_{vap})$ |
|---|---|---|---|---|
| [J] | [J] | [J] | [J] | [J] |
| 1052 | 2.2 | 1.5 | 0.32 | 0.0007 |
| 2291 | 14.2 | 12.4 | 0.13 | 0.0008 |
| 3356 | 53.6 | 49.4 | 0.08 | 0.0012 |
| 4237 | 155.3 | 147.0 | 0.05 | 0.0019 |
| 4894 | 379.9 | 364.8 | 0.04 | 0.0029 |

- to develop a transparent experimental setup for visual observation of the action of internal elements, particularly the control valve (one of the most important elements of the device),

- to develop an experimental setup that is simple, low cost, and consists of only minimum measurement equipment due to lack of financial support,

- to develop a model that works regardless of good efficiency or detailed analyses of the quantitative results.

## 3.1   Laboratory model

If water is used as a heat agent, the pressure inside the installation must be kept below the atmospheric pressure. It was decided that an electrical vacuum pump would not be used and the proposed device would not include an electrical circulation pump. A vacuum was created by employing the gravity method in the model. Therefore, the laboratory model contained two building blocks:

- pumping of hot heat carrier downward, and

- achieving and maintaining vacuum.

A schematic diagram of the laboratory installation and a photo are shown in Figs. 8 and 9. The hot and cold work vessels were made of steel cylinders and are horizontally mounted. The diameter and the length of the cylinders were 122 mm and 134 mm, respectively. The bottoms of both cylinders were

transparent (made of Plexiglas) for visual observation. These vessels were placed into open rectangular boxes with transparent side walls. The boxes were filled with water to prevent air leakage inside the hot and cold work vessels through areas with imperfect assembly joining. An electrical resistance heater was used as a source of heat. The heater was placed into the rectangular water-filled box under the hot vessel. The upper part of the circulation loop was transparent in order to visualise action of the control valve.



Figure 8. Schematic diagram of the prototype passive heat transfer model: a) circulating block, b) vacuum creator; 1 – hot descending branch, 2 – cold upflowing branch, 3 – intermediate channel, 4 – electrical heater, 5 – non-return valve, 6 – hot work vessel, 7 – liquid seal vessel, 8 – air outlet valve, 9 – hot sealing box, 10 – cold sealing box, 11 – cold work vessel, 12 – air outlet valve, 13 – sealing water, 14 – upper part of loop, 15 – cooler, 16 – outlet vessel, 17 – inlet valve, 18 – inlet tube, 19 – vacuum tube, 20 – outlet valve, $t$ – temperature measurement, $p$ – pressure measurement.

The heat exchanger (cooler) used for cooling the heat-carrier was located 3.5 m below heater. The heat exchanger consisted of a copper coil with a surface area of 0.28 m$^2$ placed into a cylindrical vessel with a capacity of
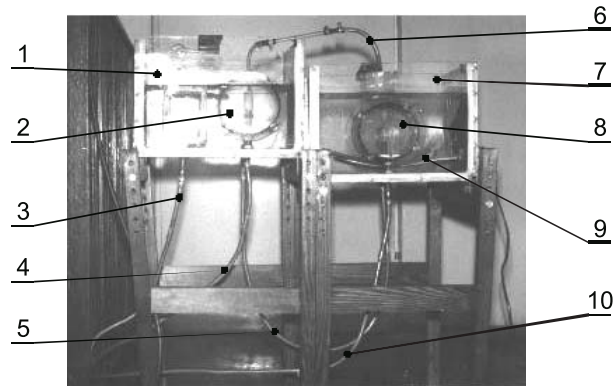
Figure 9. Upper part of laboratory installation: 1 – cold sealing box, 2 – cold work vessel, 3 – vacuum tube, 4 – cold upflowing pipe, 5 – intermediate canal, 6 – upper part of loop, 7 – hot sealing box, 8 – hot work vessel, 9 – heater, 10 – hot descending pipe.

$0.01$ m$^3$, which was filled with water (see Fig. 10). The cooler vessel was filled with water that was not in motion, but only accumulated heat leading to a temperature rise during the experiment.



Figure 10. Cooler – heat exchanger; takes heat from the heat-carrier (without a tank-container).

The outlet vessel (in which the water descended to create vacuum inside the installation) was placed in the basement level, 11.5 m below the top of the loop. The upper part of the vacuum tube had a larger diameter ($\phi$70 mm) for accumulation of air during operation of the system. This

tube was also transparent in order to observe the changing pressure during each of the cycles as well as to monitor the incoming air.

### 3.1.1  Measurement equipment

The main objective of the measurements was to prepare an operational data set from the investigated device for analysis. The analysed parameters included:

- duration of cycle and its two stages: pumping and pouring off,

- temperature characteristics and changes in pressure,

- capacity of heat transfer.

The processes in the model are transient. It was decided that measurement should be taken every 10 s since the duration of each cycles was expected to be 1 min or more.

All measured data was stored in a database for further processing and analysis. The following methodologies were used for measurement:

- output from the sensors was an analog electrical signal;

- analog signal was transformed into a standard current analog signal of $4 - 20$ mA, independent of transmission distance;

- standard analog signal was transformed into a digital signal, which is convenient for transmission and acceptable for computer input;

- digital network was created because two different locations for measurement were planned: inside of the building for the laboratory model and outside of the building for the solar installation.

The installed measurement equipment was capable of taking measurements for only five measurements points. Therefore, the recorded data in the computer consisted of measurement of pressure inside the cold vessel and four temperature measurements, taken at the top and the bottom of the inside hot vessel, at the descending pipe and at the cooler vessel. The rest of the measurements were taken manually, including the temperature of both rectangular boxes containing the hot and cold work vessels, electrical heater capacity and maximum liquid levels in the work vessels. A schematic diagram of the measurement equipment is shown in Fig. 11.
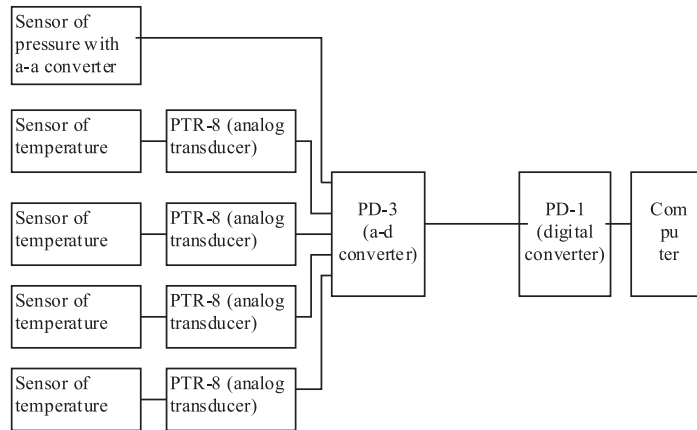
Figure 11. Schematic diagram of measurement equipment.

The pressure in the cold work vessel was measured by an Aplisens PC-50 sensor and the temperature was measured by Cu-100 resistance thermometers.

### 3.1.2   Processing of experimental data

The measured values were recorded in a computer and then copied to Excel spreadsheets for further processing and analysis.

The primary analysis showed that the data set contained many short disturbances caused by the variable influence of the surrounding environment on the installation, turbulent phenomenon inside the installation, and transformation of data sets. Additional small splashes and drips were recorded on diagrams. These impeded identification of real processes. Filtering of short wave interference was done with the help of an appropriate algorithm [5]. Figure 12 shows an example of short wave violations using a diagram and the results of action of the algorithm. After this process, it was possible to define the quantity and frequency of the cycles of real processes.

The amount of heat carried by the liquid heat carrier from the hot vessel during one cycle was calculated by using the average temperature difference, $t_{desc.pipe}$ measured at the lower pipe and the average minimum temperature value, $t_{min.bot}$ measured at the bottom of the hot vessel:

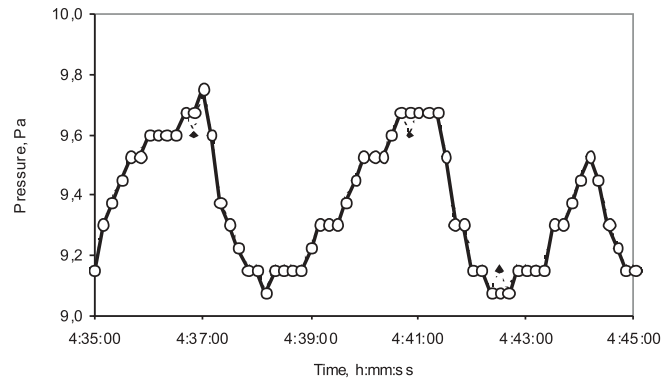$$Q = m_c c(t_{desc.pipe} - t_{min.bot}). \tag{20}$$

Figure 12. Result of algorithm action for filtering of short wave violations: dotted line presents the primary data.

The heat carried by vapour from the hot to the cold vessel was calculated from the latent heat of the portion of saturated vapour and the temperature was taken as the mean value of the maximum temperatures at the top of the hot work vessel ($t_{max.top}$):

$$Q_{vap} = \frac{\Delta V}{\nu''(t_{max.top})} r(t_{max.top}).$$   (21)

## 3.2   Results and discussion of testing on the laboratory model

### 3.2.1   Changes of temperature and duration of cycles

The design and quality of the control valves were not satisfactory and therefore required continuous improvement and replacement during testing. Due to modification of the control valves and different heater capacity, the volume of the hot liquid that was pushed downward and the temperature difference between the vessels demonstrated different values.

The effect of the improperly working control valve on the measured operating parameters and on the cycle duration (for a heater capacity of 500 W) can be seen in Figs. 13 and 14, respectively. Figure 14 clearly shows that the cycle duration varies from 1 to 6 min due to mechanical elements in the control valves that were not working properly.

The control valve used during the test with the 750 W capacity heater was of better quality, and therefore, the cycle duration was found to be
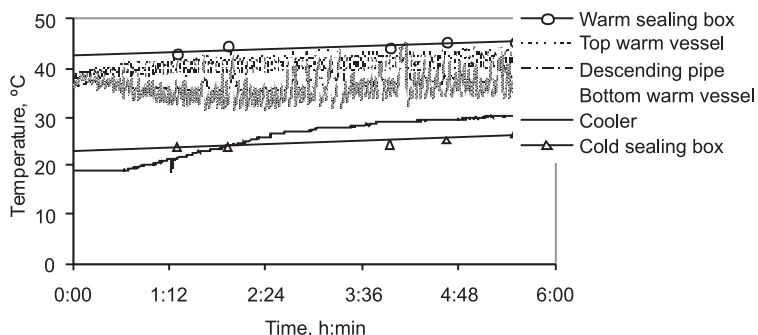
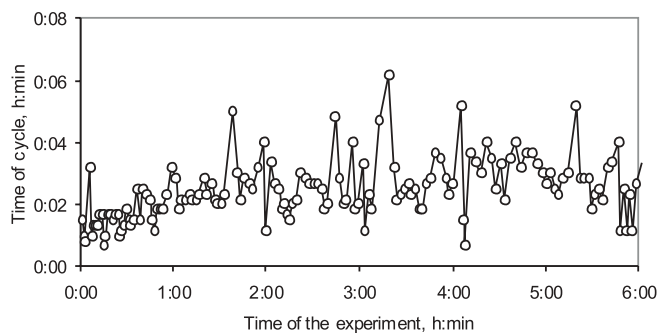Figure 13. Operating temperature of experimental setup for heater power of 500 W.



Figure 14. Variation of cycle duration with respect to time for heater power of 500 W.

within 1.6 to 2.5 min (see Figs. 15 and 16). The best stable cycle duration values close to the sensitivity of measurement were found for the control valve used during the test with a 900 W capacity heater (see Figs. 17 and 18).

### 3.2.2   Air leakages

The pressure in vessels can be measured from the saturated vapour pressure within the cold work vessel. The measured pressure could be used to determine the temperature and other parameters of saturated vapour pressure. The temperature calculated using the measured pressure was found to be higher than expected. This may be due to the fact that the measured pressure is the sum of the partial saturated vapour pressure and partial pressure
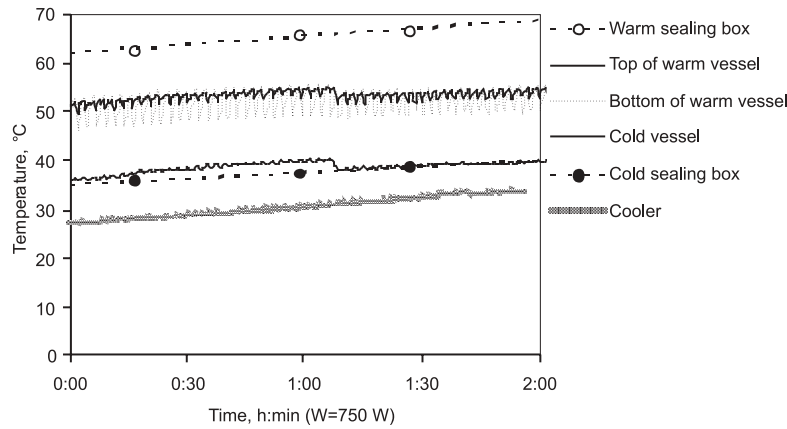
Figure 15. Operating temperature of experimental setup for heater power of 750 W.
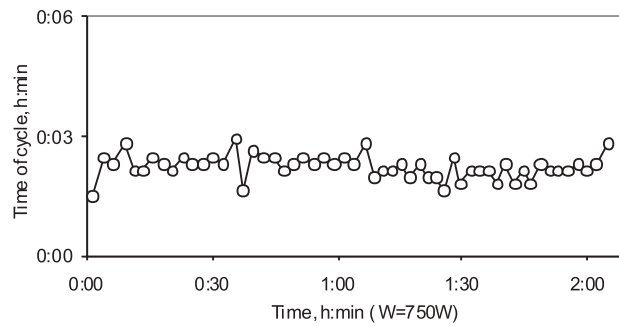


Figure 16. Variation of cycle duration with respect to time for 900 W heater.

of air present inside the installation. The total pressure is therefore significantly influenced by the partial air pressure when the saturated vapour pressure is very low (below 10 kPa). The air enters from leaks through construction imperfections and also occurs as a result of extraction from the water heat carrier containing dissolved air. The following conclusions can be drawn for the air present in the installation:

- Air gradually accumulates in the cooler part of the installation, especially in the cold work vessel.

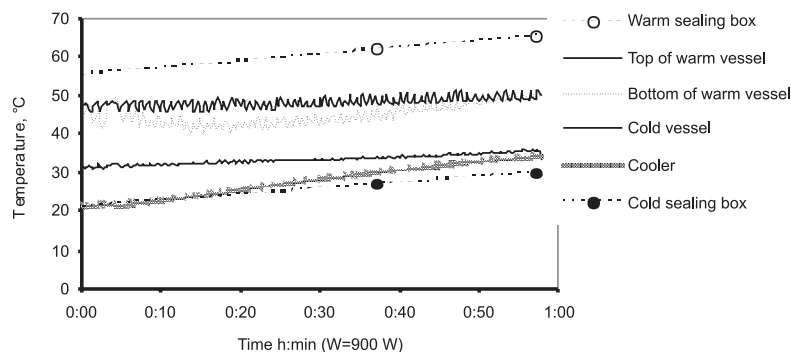- Increase of the total pressure in the cold work vessel will increase

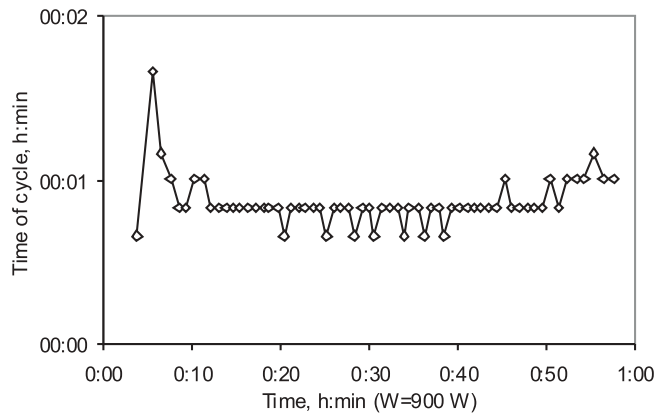Figure 17. Operating temperature of experimental setup for 900 W heater.



Figure 18. Variation of cycle duration with respect to time for 900 W heater.

pressure and temperature in the hot work vessel, which is required for the device operation. This phenomenon can be noticed particularly at low temperature and low saturated vapour pressure.

• The air, which remains in places of devaporation, blocks access of the vapour to cold surfaces, therefore retarding the process of condensation, especially in the end of the drain stage.

### 3.2.3 Heat flow

The investigated model produced high heat losses due to poorly insulated walls and open water surfaces from which water was vaporised. Therefore, only 4–19% of the total heat delivered by the heater was transferred to the heat sink. The higher values were from the case where the heater capacity was high and the experimental test time was short. A schematic diagram of the heat flows in the experimental model is shown in Fig. 19.
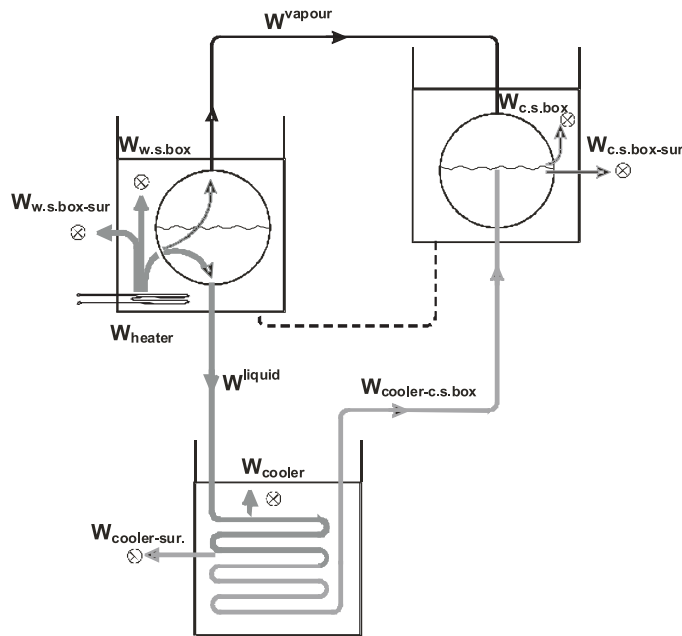


Figure 19. Scheme of heat flows in experimental model.

The heat carried out by the vapour from the hot to cold work vessel through the upper path of the loop was 0.08–0.13% of the heat capacity (see Tab. 4). The heat recirculation coefficient was around 0.4–0.5%.

Not all of the heat was accumulated in the cooler. Some part of the heat travelled through the cooler coil and reached the cold work vessel and cold rectangular vessel.

The average accumulated heat was calculated by:

$$Q_{acc} = m_{vessel}c(t_{end} - t_{start}). \tag{22}$$

Table 4 shows the calculated values of the average accumulated heat. It was

Table 4. Heat flows in experimental model.

| Duration of experiment, s | | 21600 | 6900 | 3180 |
|---|---|---|---|---|
| Power of heater, W | | 500 | 750 | 900 |
| Duration of a cycle, s | | 147 | 140 | 48 |
| Heat flow, which lost hot work vessel ($W^c$) | Starting temperature, °C | 30.54 | 38.86 | 33.36 |
| | Final temperature (in lowering pipe), °C | 36.76 | 51.61 | 45.00 |
| | Mass of pumped portion, kg | 0.45 | 0.36 | 0.278 |
| | Capacity $W^c$, W | 80 | 137 | 282 |
| Heat flow, which was carried to the cold work vessel by vapour ($W^p$) | Temperature of vapour, °C | 41.1 | 53.5 | 48.32 |
| | Volume of the vapour, $m^3$ | 0.00045 | 0.00036 | 0.000278 |
| | Specific volume of the vapour, $m^3/kg$ | 17.8 | 9.7 | 12.4 |
| | Mass of pumped portion of the vapour, kg | 2.5E-05 | 3.7E-05 | 2.2E-05 |
| | $W^p$, W | 0.43 | 0.64 | 1.16 |
| Heat, which was accumulated in the cooler's vessel ($W_{ch}$) | Initial temperature, °C | 19.30 | 27.3 | 21 |
| | Final temperature, °C | 30.30 | 34.2 | 34 |
| | Mass of water in the vessel, kg | 10 | 10 | 10 |
| | $W_{ch}$, W | 21.3 | 41.9 | 171.3 |
| Heat, which was accumulated in the cold rectangular vessel ($W_{nuz}$) | Initial temperature, °C | 23.00 | 35 | 29 |
| | Final temperature, °C | 26.00 | 39 | 33 |
| | Mass of water in the vessel, kg | 11.5 | 11.5 | 11.5 |
| | $W_{nuz}$, W | 7 | 28 | 61 |

found that only 4.3–18.8% of the total heat delivered by the heater is utilised in the cooler.

## 3.3  Work of reverse thermosiphon in a solar installation structure

The reverse thermosiphon model has been modified for installation in a solar structure. To avoid damaging the solar structure, the thermosiphon was connected to the solar installation through an additional heat exchanger.

The block diagram for achieving and maintaining vacuum had two lines. This permitted evacuation of the air from the device without stopping the reverse thermosiphoninc action. The basic scheme of the solar installation

with the reverse thermosiphon and gravitational support vacuum is illustrated in Fig. 20 [7,8].
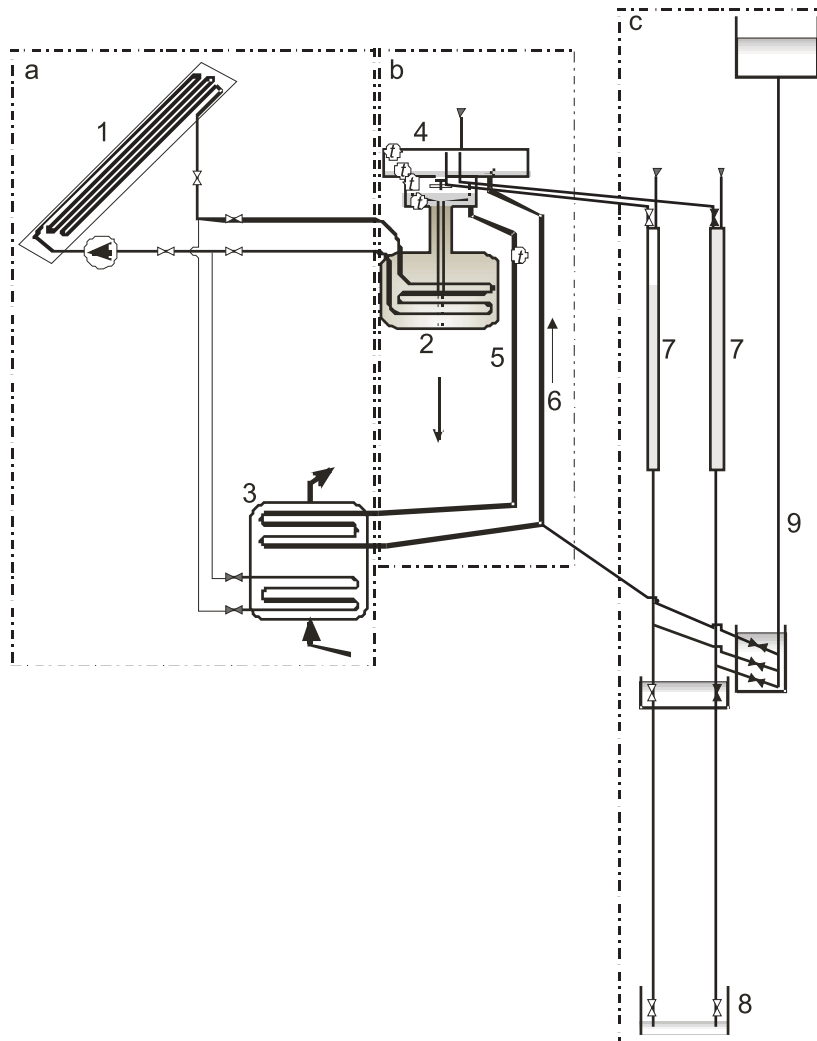


Figure 20. Schematic diagram of thermosiphon integrated with a solar installation: a – solar installation, b – reverse thermosiphon, c – subsite for creating and maintaining vacuum; 1 – solar collector, 2 – heat exchanger, 3 – heat accumulator, 4 – reverse thermosiphon, 5 – descending pipe, 6 – upflow pipe, 7 – air accumulators, 8 – discharge tank, 9 – filling pipe, $t$ – temperature measurement.

The working vessels consisted of transparent walls (see Fig. 21). The bottom of the hot vessel was made with transparent lateral walls and the top of the cold vessel had a larger diameter and a transparent cover.



Figure 21. Reverse thermosiphon model integrated with a solar installation.
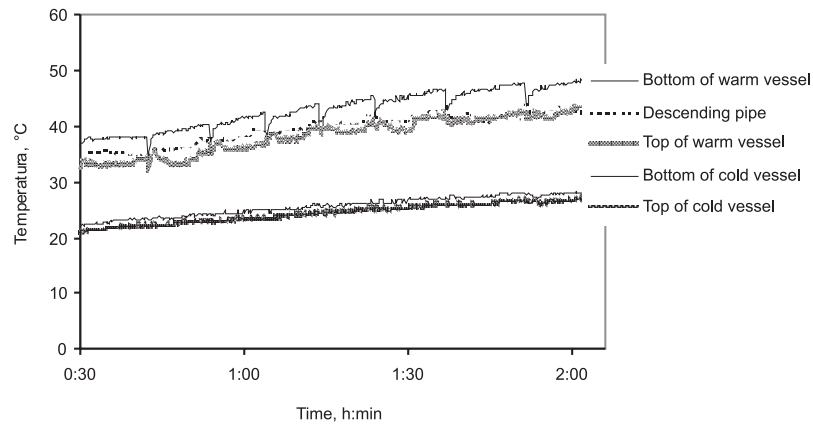


Figure 22. Operating temperature of prototype reverse thermosiphon model integrated into a solar installation with respect to time.

The experimental results (see Fig. 22) show that the influence of air inflow on the thermosiphon operation was significant even when the air inflow was insignificant, resulting in an increase in operating temperature differ-

ence and even halting its operation (in spite of the imperfect mechanical control valve that caused non-uniform operation of the thermosiphon).

## 4    Conclusions

The study showed that the developed novel heat exchanger, a reverse thermosiphon, is a self-acting and self-controlling device that can transfer heat downwards in a liquid circulation loop. It operates using a small quantity (less than 0.1%) of local low-grade heat. The exergy losses are also very low, less than 0.1%. The device can begin operation even with a temperature difference of few degrees between the hot and cold branches of the loop. The device is simple in design. Although the use of water as a heat carrier and the presence of air inside the device created some operating difficulties, the problem was not critical. One model was integrated to a solar installation and tested successfully [4,6]. However, water does not provide adequate operating conditions at temperatures below 100 ºC as its saturated vapour pressure is lower than the atmospheric pressure. A new device with low-boiling point working agents has been designed, developed, and tested at the University of Warmia and Mazury in Olsztyn.

The limitations of the low-boiling working agents are their cost and harmful influence on the surroundings. A promising solution to this problem consists of combining the advantages of water and low-boiling working agents in one device. This will be the subject of future investigations.

*Received 29 June 2009*

## References

[1] Beni G.De, Friesen R., Thoma H. and Veneroni R.: *Device for passive downward heat transport: desing criteria and operational result*, Advances in Heat Pipe Technology, London 1982, 673-684.

[2] Bezrodny, M.K., Pioro, I.L. and Kostyuk,T.O.: *Transfer Processes in Two-Phase Thermosyphon Systems. Theory and Practice*, 2nd edn., Fact Publ. House, Kiev 2005 (in Russian).

[3] Davidson J.H., Walker H.A., Lof G.O.G.: *Experimental study of a self – pumping boiling collector solar hot water system*, Journal of Solar Energy Engineering, **111**(3), 1989, 211-218.

[4] Dobriański J., Fieducik J., Derela E.: *Investigations of the device spontaneously transporting heat in opposite direction to natural convection*, XII Symposium

of Heat and Mass Transfer, Academy of Mining and Metallurgy Publishers, Cracow 2004, 199-202 (in Polish).

[5]  DOBRIANSKY Y., DUDA M.: *Algorithm for processing in Excel experimental data set of cycling processes*, Technical Sciences, **10**(2007), 69-80.

[6]  DOBRYANSKY Y., FIEDUCIK J.: *The first applying of reverse flow circuit for solar collector*, Proceeding of the VII World Renewable Energy Congress, 2002, 396.

[7]  FIEDUCIK H.: *MSc thesis*, University of Warmia and Mazury, Olsztyn 2003 (in Polish).

[8]  FIEDUCIK J.: *PhD thesis*, The Szewalski Institute of Fluid-Flow Machinery Polish Academy of Sciences, Gdansk 2005.

[9]  PERROT P.: *A to Z of Thermodynamics*, Oxford University Press, New York 1998.

[10]  *Heating and Air-conditioning hand book, 1st edn.*, EWFE, Gdansk 1994 (in Polish).

[11]  ROBERTS C.C.: *A Review of Heat Pipe Liquid Delivery Concepts/Advances in heat pipe technology*, Pergamon Press, Oxford 1982, 693-702.

[12]  WALKER H.A., DAVIDSON J.H.:  *Second – law analysis of a two-phase self-pumping solar water heater*, Journal of Solar Energy Engineering, 1992, 188-190.

[13]  WALKER H.A., DAVIDSON J.H.: *Analysis and simulation of a two-phase self-pumping water heater*, Journal of Solar Energy Engineering, **112**(1990), 153-160.

[14]  WONG Y.W. AND SUMATHY K.: *Solar water pump with u-pentane and ethyl ether as working fluids*, Energy Conversion and Management. Exeter, Elsevier Science Ltd., **41**, 2000, 915-927.