CLIMA 2016 - proceedings of the 12th REHVA World Congress

volume 3

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Publication date:
2016

Document Version
Publisher's PDF, also known as Version of record

Link to publication from Aalborg University

Citation for published version (APA):
Heiselberg, P. K. (Ed.) (2016). *CLIMA 2016 - proceedings of the 12th REHVA World Congress: volume 3*. Department of Civil Engineering, Aalborg University.

Dynamic Thermal Modeling for a System That uses a Compression Heat Pump

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Abstract

The objective of this paper is to illustrate a mathematical model for an unconventional source of energy. This source contains a solar surface that stocks thermal solar energy into a tank that is also connected to a heat pump and the consumer supply line.

The functionality of the system has two steps: charging and consuming the thermal energy (stocked inside the tank). The tank receives energy from the solar surface and the heat pump and transfers that energy to the consumer equipment.

The relations used in this paper determined a method that can be used to simulate the dynamic behavior of the system.

Key words: solar panels, heat pumps, energy recovery

1. Introduction

Unconventional energy systems are becoming more popular with the extent of oil based fuels and are becoming a good alternative to the old systems in the XXI century. Among those, solar based systems and heat pumps are one of the most known systems because of their easy to obtain general requirements [3].

In current paper those two types of unconventional energy systems are being combined in order to build a system that can supply hot water and heating for the residential consumer during the cold season.

The objective of this paper is to create a mathematical model for the thermal transfer of the system. This includes the modelling of the solar surface, the tank and the heat pump processes in order to supply the consumer with the thermal energy needed. This model will solve both charging and discharging steps of the tank.

2. The description of the source system. Operating mode

The system, as presented above, has three components [2], [4]:

- The solar component, including receiving the solar energy on the surface, transforming it into thermal energy and transporting it to the tank
- The storage represented by the tank
- The energy delivery component for the consumer represented by the heat pump

Fig. 1 presents the layout of the system with the three components detailed above. It is considered on this stage that there are two steps for a complete cycle of the system, one in which the tank is charging and another where the tank is discharging the thermal energy. In the first stage the components used are: the collector surface, the heat exchanger (at the lower side) inside the tank and of course the tank. In this stage the heat pump does not work. On the second stage the heat pump starts and the other components excepting the tank stop working. The evaporator of the heat pump is actually a heat exchanger at the upper side of the tank and the condenser is a heat exchanger that transmits the thermic power to the heat delivery system or the hot water delivery system. The thermic charging and discharging of the tank can work together and this situation is also studied, when both the solar collector and the heat pump are working.

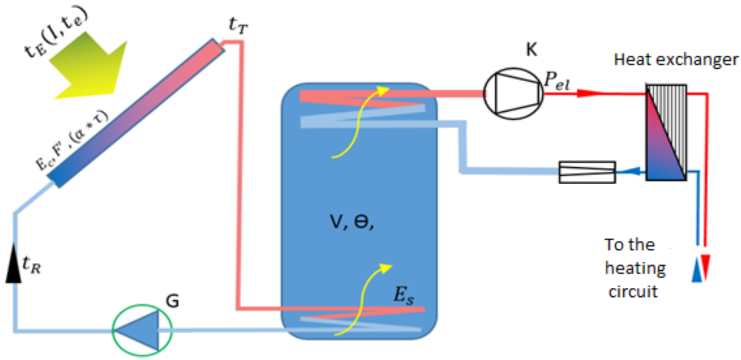

Fig. 1- Scheme of the system used

3. Thermal Process modeling for the source system

The objective that we follow on this paper is to make a mathematical model for both the charging and discharging steps of the tank that will simulate the dynamic behavior of the system in 24 hours. By dynamic behavior of the system we consider the values of the temperature inside the tank, the thermic power accumulated and extracted from the tank at any given value of the time window.

Fig. 2 presents a system made by the collector surface and a tank with heat exchanger at the bottom side. The mathematical model that allows the simulation of the dynamic behavior of this system is represented by the equations for the solar collector and heat exchanger (stationary stand) and the tank (dynamic stand) [2].

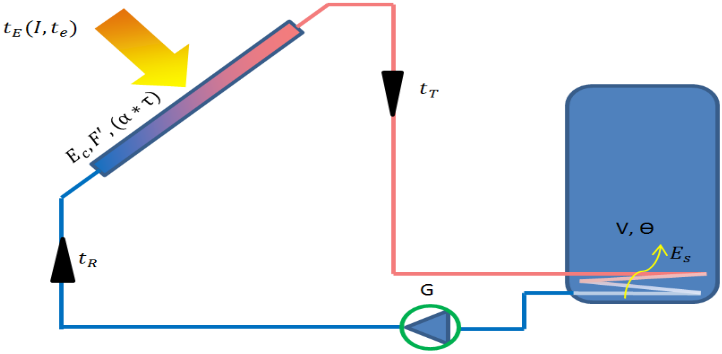

Fig. 2- Solar Collector circuit scheme

As presented in Fig. 1 and 2 the thermic equation in dynamic stand for this step of charging the tank can be written as:

$$P_{CS} = G \cdot (\rho c) \cdot (t_T - t_R) = V \cdot (\rho c) \cdot \frac{d\theta}{d\tau} \qquad (1)$$

Writing the thermic equations for the collector surface and the heat exchanger in the tank, equation (1) becomes:

$$E_{CS} \cdot G \cdot (\rho c) \cdot (t_E - \theta) = V \cdot (\rho c) \cdot \frac{d\theta}{d\tau} \qquad (2)$$

Where:

$$t_E = \frac{\alpha \tau}{k_C} \cdot I + t_e$$
$$E_{CS} = \frac{(1 - E_C) \cdot (1 - E_S)}{1 - E_C \cdot E_S} \qquad (3)$$

Relation (2) becomes in a canonical form:

$$\frac{d\theta}{d\tau} = -\frac{1}{C_T} \cdot \theta + \frac{1}{C_T} \cdot t_E \qquad (4)$$

Where C_T is the time constant with the expression:

$$C_T = \frac{V \cdot (\rho c)}{E_{CS} \cdot G \cdot (\rho c)} \qquad (5)$$

Considering limited time steps ($\Delta\tau$) on which the equivalent temperature $t_E(\tau)$ is constant, we obtain the values for the temperature inside the tank $\theta(\tau)$:

$$\begin{aligned} \theta_1 &= E^{\Delta\tau} \cdot \theta_0 + \left(1 - E^{\Delta\tau}\right) \cdot t^*_{01} \\ \theta_2 &= E^{\Delta\tau} \cdot \theta_1 + \left(1 - E^{\Delta\tau}\right) \cdot t^*_{12} \\ &\dots\dots\dots\dots\dots\dots\dots\dots \\ \theta_n &= E^{\Delta\tau} \cdot \theta_{n-1} + \left(1 - E^{\Delta\tau}\right) \cdot t^*_{n-1,n} \end{aligned} \qquad (6)$$

Where:

$$\begin{aligned} E &= \exp\left(-\frac{1}{C_T}\right) \\ t_{E(n-1,n)} &= \frac{1}{2} \cdot \left(t_{E(n-1)} + t_{E(n)}\right) \end{aligned} \qquad (7)$$

The relations allow to determine the quantity of the heat stored inside the tank for the value of time τ:

$$Q = V \cdot \rho c \cdot (\theta - \theta_0) \qquad (8)$$

In what regards the thermic system that works on discharging step, the components are: the tank, the evaporator and the condenser of the heat pump. The power of the condenser needed for the heating system will be considered constant or variable for the second stage.

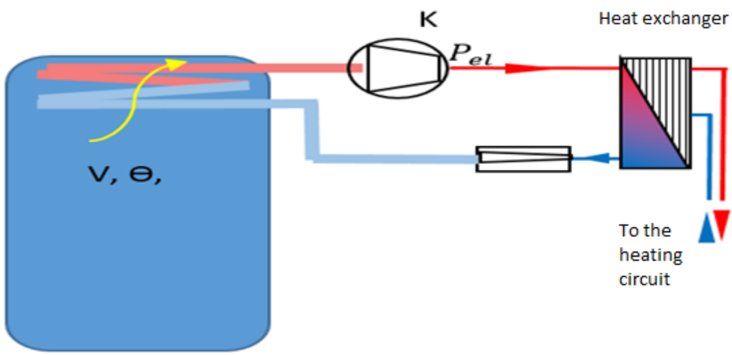

Fig. 3 – Supply system containing the heat pump

The heat pumps with mechanic compression, with electric modulator, can set the electric power of the compressor very precise (fig. 3). We can rely on changing the thermal power for the condenser conforming to what the consumer needs, by changing the electric power for the compressor. This concludes that the evaporator power depends on the variation of the condenser power and the ambient temperature [1].

As presented in fig. 1, the thermic equations for the system in the discharge step is:

$$V \cdot (\rho c) \cdot \frac{d\theta}{d\tau} = -P_{VP} = -a \cdot \theta - b \qquad (9)$$

Where:

$$a = \frac{P_{CD}}{\theta_{CD} + \Delta_{CD} + 273.15}$$
$$b = \frac{(273.15 - \Delta_{VP}) \cdot P_{CD}}{\theta_{CD} + \Delta_{CD} + 273.15} \qquad (10)$$

Relation (9) becomes in the canonical form:

$$\frac{d\theta}{d\tau} = -\frac{1}{C_T^*} \cdot \theta - \frac{1}{C_T^*} \cdot t^* \qquad (11)$$

Where:

$$C_T^* = \frac{V \cdot (\rho c)}{a}$$
$$t^* = 273.15 - \Delta_{VP} \qquad (12)$$

”a” constant represents the capacity of thermal transfer for the tank in the discharging stage.

Making the hypothesis that the thermal power needed by the consumer (the condenser power) is constant on different time periods ($\Delta\tau$), the temperature inside the tank can be found similar to the charging stage. Considering the $t^*(\tau)$ constant (in time), the values of the water temperature inside the tank- $\theta(\tau)$ are:

$$\begin{aligned}
&\theta_1 = E^{\Delta\tau} \cdot \theta_0 + \left(1 - E^{\Delta\tau}\right) \cdot t_{01}^* \\
&\theta_2 = E^{\Delta\tau} \cdot \theta_1 + \left(1 - E^{\Delta\tau}\right) \cdot t_{12}^* \\
&\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots \\
&\theta_n = E^{\Delta\tau} \cdot \theta_{n-1} + \left(1 - E^{\Delta\tau}\right) \cdot t_{n-1,n}^*
\end{aligned} \tag{13}$$

Where:

$$\begin{aligned}
&E = \exp\left(-\frac{1}{C_T^*}\right) \\
&t_{n-1,n}^* = \frac{1}{2} \cdot \left(t_{n-1}^* + t_n^*\right)
\end{aligned} \tag{14}$$

Therefore we can calculate the $P_{VP}(\tau)$, $P_{EL}(\tau)$, $COP_{VP}(\tau)$ and $COP_{CD}(\tau)$.

The third stage studied combines both the accumulation and the discharge for the tank. The thermic equation for the tank in dynamic stage is:

$$\begin{aligned}
P_{CS} - P_{VP} &= E_{CS} \cdot G \cdot (\rho c) \cdot (t_E - \theta) - (a \cdot \theta + b) = \\
&= V \cdot (\rho c) \cdot \frac{d\theta}{d\tau}
\end{aligned} \tag{15}$$

Relation (9) becomes in the canonical form a differential equation of first order:

$$\frac{d\theta}{d\tau} = -\frac{1}{C_T^{**}} \cdot \theta + \frac{1}{C_T^{**}} \cdot t^{**} \tag{16}$$

Where:

$$
C_T^{**} = \frac{V \cdot (\rho c)}{E_{CS} \cdot G \cdot (\rho c) + a}
$$
$$
t^{**} = \frac{E_{CS} \cdot G \cdot (\rho c) \cdot t_E - b}{E_{CS} \cdot G \cdot (\rho c) + a} \tag{17}
$$

Considering the function t*(τ) is constant on finite time steps we obtain the values for the water temperature inside the tank - θ(τ):

$$
\begin{aligned}
\theta_1 &= E^{\Delta\tau} \cdot \theta_0 + \left(1 - E^{\Delta\tau}\right) \cdot t_{01}^{**} \\
\theta_2 &= E^{\Delta\tau} \cdot \theta_1 + \left(1 - E^{\Delta\tau}\right) \cdot t_{12}^{**} \\
&\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \\
\theta_n &= E^{\Delta\tau} \cdot \theta_{n-1} + \left(1 - E^{\Delta\tau}\right) \cdot t_{n-1,n}^{**}
\end{aligned} \tag{18}
$$

Where:

$$
E = \exp\left(-\frac{1}{C_T^{**}}\right)
$$
$$
t_{n-1,n}^{**} = \frac{1}{2} \cdot \left(t_{n-1}^{**} + t_n^{**}\right) \tag{19}
$$

The time step in this kind of application can be considered a half an hour or one hour.

4. Case study illustrated

This chapter presents the functionality of this kind of system equipped with solar collector, heat pump and tank that are not actually correlated to the needs of the consumer.

The hypothesis are again mentioned:

- Uniform temperature in the tank;
- Different stages for both accumulation and discharge of the tank.

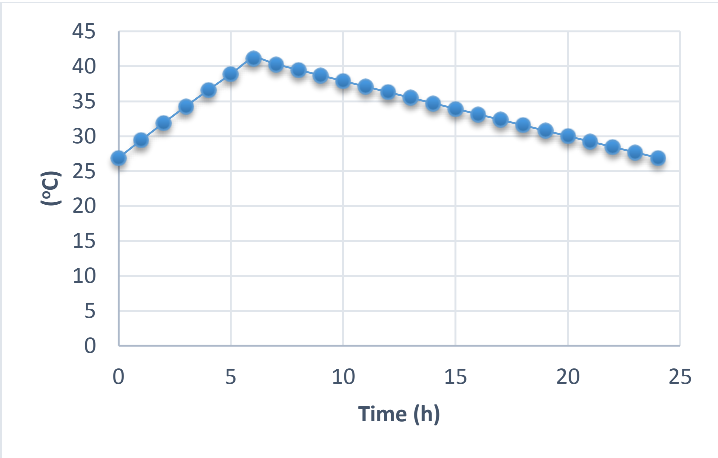

Fig. 4- Temperature of water in the tank

In fig. 4 we can distinguish the two stages for this exercise, one of accumulating power inside the tank and one of discharging the power of the tank.

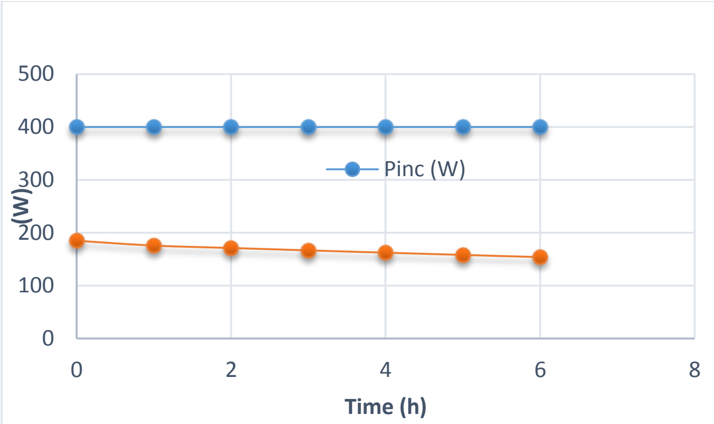

Fig. 5- Incident Power and Received Power (W)

Fig. 5 presents the dynamic of both incident and received power during the charging step of the tank. The efficiency for the solar surface is around 45% and lowers as the water is starting to heat up inside the tank.

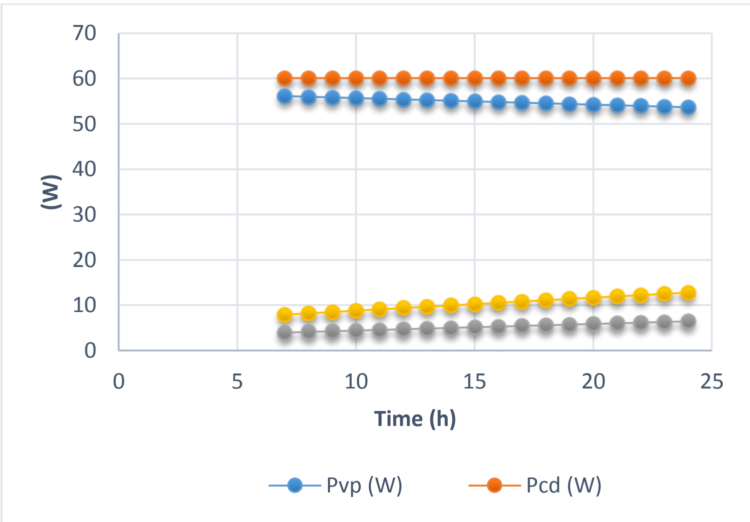

Fig. 6- Condenser, Evaporator and Electric Power

Fig. 6 presents the power dynamic in the second stage of thermic discharge. The condenser power was considered constant. The evaporator power is lower and is reducing as the temperature inside the tank drops. The total electric power consumed is considerably lower than the two powers mentioned above. P_{el1} – is the electric power used that transforms the wasted ambient energy (anergy) into good potential energy P_{VP} (exergy), and the difference until P_{el2} is consumed in order for the machine to function [4].

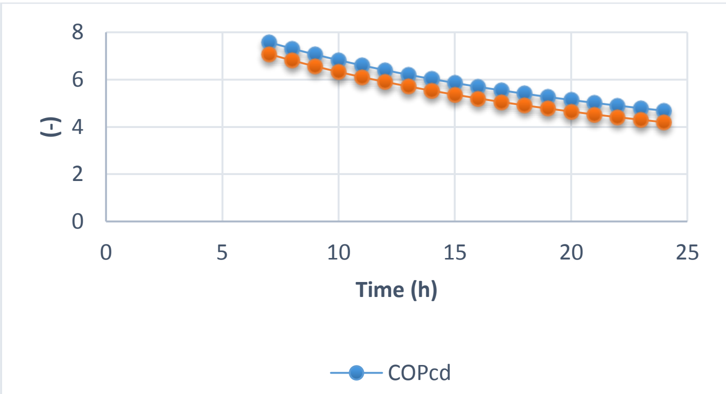

Fig. 7- Performance coefficients for the condenser and evaporator

Fig. 7 presents the dynamic of the performance coefficients at the level of the condenser and evaporator represented by the heat pump. A big performance drop occurs below the value of 8 (down to 4) for the 18 hours of discharge during the second stage.

There was identified two parameters that have a big influence to the dynamic behavior for the charging stage fallowed by the discharging stage. Those two parameters are the specific volume of energy accumulated (V/Sc) and the specific thermic power delivered to the condenser of the heat pump (PCD/Sc). Fig. 8 presents the influence of specific volume to the energy of the tank (Note: in Fig. 8, the COP_{cd} is dimensionless).

Considering the specific volumes for the tank between 20 and 100 l per m^2 collector, the energetic performances of the system remain relatively constant, however, water temperatures in the tank vary between 10 °C and 54 °C if the volume is 20 l/m^2 and only between 29 to 38 °C if the volume is 100 l/m^2.

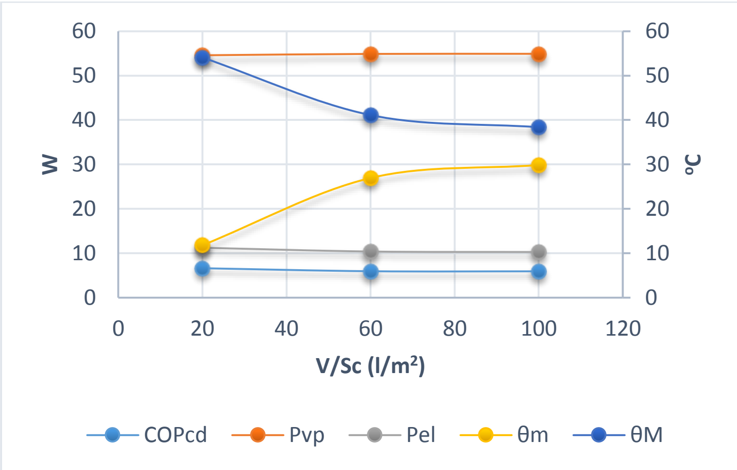

Fig. 8- Study for the behavior of the system using Specific volume of accumulated energy

Fig. 9 presents the influence of the specific power of the condenser used by heat pump to the behavior of the system. The optimum value for the heat pump power is considered to be 60 W/m^2. It can be seen that for low values of heat pump power of 50 W/m2 the performance is low and for values above the 60 W/m^2 the performance coefficient does lock around the value of 5. The temperature fluctuation inside the tank is very low at values of power above 55 W/m^2. The electric power of the compressor increases as the condenser power goes up (Note: in Fig. 9, the COP_{cd} is dimensionless).

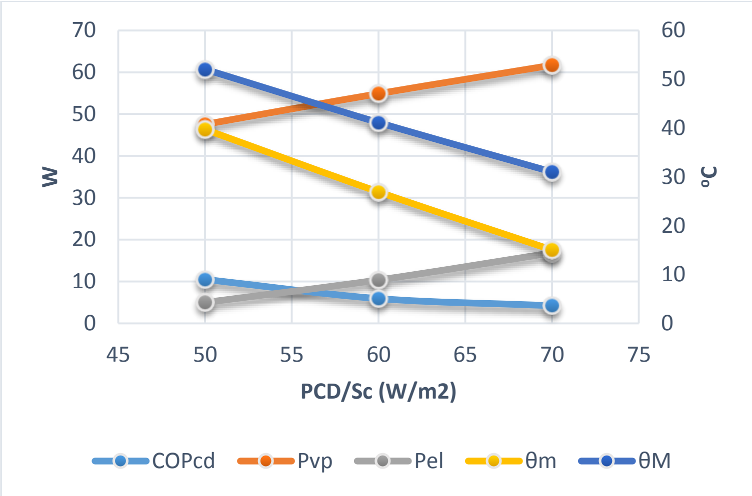

Fig. 9- Study for the behavior of the system using the power for the condenser PC

The best values for those two parameters mentioned are: $V/Sc = 50\ l/m^2$ and $P_{CD}/Sc = 60\ W/m^2$.

5. Conclusions

This paper had as main objective creating a model for the dynamic process of charging and discharging of the tank using an unconventional source of energy.

The equations used for the two stages allow to find any variables needed.

The charging step was chosen in order to take three times less than the discharging one.

The mathematical model allows to simulate the functionality of the system when both of the stages are active during some period of time.

This model also shows the dynamic of the three power used, the tank stoked power, the condenser power and the total electric power.

Symbols

t_e – outside temperature for the solar collectors , °C;

t_E – equivalent temperature factor for the climate, °C;

θ - temperature inside the tank, °C ;

θ_0 – initial temperature of the tank, °C ;

t_T, t_R – entering and exiting temperature from the solar collector, °C ;

I – solar radiation, W/m^2;

V – tank volume, m^3;

ρ - water density of the tank, kg/m^3;

c – specific heat of water, J/kg.K;

G – thermal flow meter inside solar collectors, m^3/s;

k_C - global thermal transfer coefficient for solar collectors, W/m^2.K;

$\alpha\tau$ - absorption and transparency coefficients for the solar collector, -;

τ - time, s ;

$\Delta\tau$ - time step, s ;

E_C – intrinsic feature for solar collector, - ;

E_S – intrinsic feature for the heat exchanger, - ;

E_{CS} – combined intrinsic feature for both collector and heat exchanger, - ;

C_T, C_T^*, C_T^{**} – time constants for the different stages of the tank, s ;

P_{CD} – thermic power of the condenser, W;

P_{VP} – thermic power of the evaporator, W;

P_{EL} – electric power of the compressor, W;

P_{CS} – thermal power collected, W ;

Q – total thermal energy of the tank, J ;

θ_{VP} – water temperature inside the tank ^{o}C;

θ_{CD} – inside medium temperature for the condenser, ^{o}C;

Δ_{VP} – medium difference of temperature for the evaporator, ^oC;
Δ_{CD} – medium difference of temperature for the condenser, ^oC;
θ_m – minimum temperature inside tank during a full cycle of charge-recharge, ^oC;
θ_M – maximum temperature inside tank during a full cycle of charge-recharge, ^oC;

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