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IDENTIFICATION OF THE DYNAMIC MODEL OF THE WATER OUTLET TEMPERATURE OF A HEAT PUMP OPERATING WITH CO_2 AND A SOLAR EVAPORATOR

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Abstract. *The growing global concern about the environmental impact has led to an increase in the interest of renewable and more efficient energy sources. In this context, the use of heat pumps for domestic water heating has attracted attention. This work presents the dynamic model identification of the water outlet temperature, of a heat pump with a direct expansion solar evaporator. This system was designed for residential water heating and uses* CO_2 *as refrigerant fluid. The developed model was based on experimental tests and the system dynamic response was obtained through step disturbances in the water flow rate. The model system identification was performed employing six different methods suggested by different authors. The results were evaluated by quantitative techniques that allowed selecting the best identification method. By using the methods presented in this paper, it was possible to obtain a model with a similarity percentage, when compared to the experimental data, of over 80%.*

Keywords: *Heat pump, system identification, water temperature.*

1. INTRODUCTION

The growing global concern about the environmental impact has led to an increase in the interest of renewable and more efficient energy sources. In this context, there has been a great interest in the use of heat pumps for domestic water heating due to the energy savings these systems provide, since their coefficient of performance (COP) is better when compared to electric and gas heating systems, which are commonly used in Brazilian residencies to this day (Kong et al., 2011; Starke, 2013).

Sporn and Ambrose (1955) were the pioneers to present the concept and the benefits of using a solar evaporator together with a heat pump. From their study, several researches have contributed for the evolution and viability of the system. Chaturvedi et al. (1982) presented strategies for lowering the implementation cost for this system. Their study showed that the direct expansion of the refrigerant fluid minimizes the costs with materials and improves the equipment's performance. Besides that, the authors state that, for regions located in low latitudes, the use of solar collectors with no glass covering does not imply in great heat losses. This type of collector, however, increases the variability of the thermal load and, together with a constant speed compressor, causes a decrease in the equipment's performance. This behavior was demonstrated by Chaturvedi and Shen (1984). The problem was solved by Chaturvedi et al. (1998), who studied the heat pump response to variations in the compressor frequency and, through a control algorithm, optimized the compressor-collector relation for each heat input. Kuang et al. (2003) simulated the operation of a direct expansion heat pump assisted by a solar evaporator through an entire year in order to predict the equipment's performance on the long run and study which parameters influence the most on the system's behavior.

The strategy for increasing the COP behind the use of the solar evaporator is raising the evaporation temperature (TE). The solar evaporator uses forced or natural convection alongside solar radiation in order to raise the evaporation temperature, thus improving the system's COP. Kong et al. (2011) developed a mathematical model for a heat pump with a solar evaporator, which was experimentally validated, and demonstrated the variations in the equipment's performance caused by the variations in the radiation, the ambient temperature and the wind speed.

Even though a solar evaporator does increase the performance of the heat pump significantly, this kind of heat exchanger is subject to the climatic variations that happen throughout the day. Changes in the cloud positions lead to radical variations in the incident solar radiation on the evaporator and, therefore, create different heat inputs and variations to the water outlet temperature (WOT). In the domestic usage, it is important that this temperature is kept under comfortable values for bathing (approximately 40 ºC). Because of that, the heat pump must be equipped with a device capable of controlling the WOT value. Choi (2013) showed that the WOT can be effectively controlled through the variation of the water flow in the gas cooler.

The previously mentioned studies with heat pumps were conducted with conventional refrigerants (such as R22), but the conclusions they presented can all be applied to the CO_2, which has been widely explored as a refrigerant in heat pumps and refrigeration systems as of recently (Sarkar et al., 2009; Yokoyama et al., 2007; Yu et al., 2014). This article aims on presenting a methodology for the determination of a transfer function and the dynamic behavior of the temperature in the gas cooler outlet of a heat pump for domestic water heating. Afterwards, this study will be used on the implementation of the control system on the equipment.

2. PROTOTYPE DESCRIPTION

The heat pump used in this study was designed to heat water to temperatures suited for bathing and to work with a transcritical cycle with CO_2 as its refrigerant. Fig. 1 presents a photograph of the heat pump used in this study. This heat pump is made of several devices, as specified in Faria. (2013) and Oliveira. (2013).

Figure 1. Heat pump used in this study.

The heat pump has a solar evaporator that is responsible for the heat exchange between the environment and the work fluid (CO_2), taking it from a liquid/gas mixture to the state of a superheated gas. The evaporator is made of a copper tube with a diameter of 6 mm and a length of 16.3 m, shaped as a coil on top of an aluminum plate with an area of 1.57 m².

There is also a gas cooler that is made of a countercurrent concentric tube heat exchanger coiled in the outer part of the water reservoir, which has a capacity of 0.27 m³. The tube in which the CO_2 flows is a copper tube with an internal diameter of 6 mm and an external diameter of 7.58 mm. Water flows in the countercurrent through the annular section, and the outer tube has an internal diameter of 12 mm and an external diameter of 12.70 mm. The total length of the tubing is 24.3 m.

Its compressor is a hermetic compressor, with a low power and high starting torque made by EMBRACO (model EK6210CD), designed to work exclusively with CO_2 as its fluid.

It also has a needle expansion valve with a diameter of 1.6 mm, which provides enough fluid flow to keep the superheating level in an adequate range at the evaporator outlet. As of now, this control is done manually. In the future, an automatic system will be installed to control this superheating level. The needle vale is of the model SS-31RS4 from manufacturer SWAGELOK.

A manual needle valve is used for controlling the flow of water entering the system.

An electromagnetic flow measurer from INCONTROL, model VMS PRO, is used for determining the mass flow of water.

The heat pump has 7 thermocouples of the T type (copper-constantan), with a diameter of 1.5 mm and a 10 cm long sheath, from manufacturer ECIL. The thermocouples were obtained along with a calibration certificate and installed in the inlet and outlet of each component of the heat pump, as depicted in Fig. 2.

In the prototype there are also two solar radiation sensors. One of them is responsible for measuring the incident radiation on the evaporator plane, which has an inclination of 30º from the horizontal, and the other one measures it in the horizontal plane. These sensors are manufactured by BLACK & WHITE PYRANOMETER, model 8-48.

All of the data generated by the different sensors here presented are received and treated by a data acquisition system. This system is made of basically a module for the conditioning of signals from the thermocouples, the flow measurer and the radiation sensors. For the thermocouples an acquisition board of the model USB-9162 (24 bits) is used; for the water flow and solar radiation, a USB-6211 model board is used, both manufactured by National Instruments and installed on a PC computer. The LabVIEW® software was used in order to process the collected data.

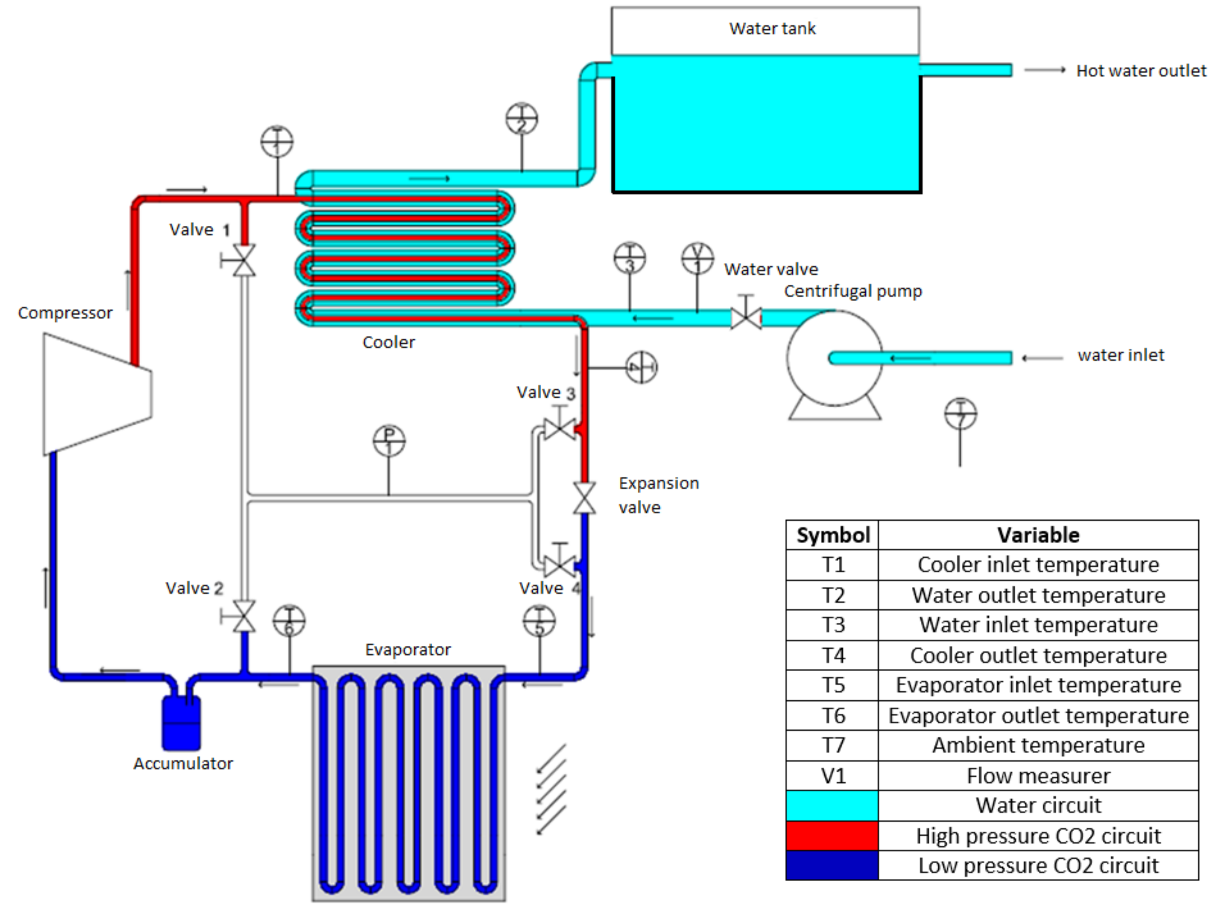

Figure 2. Schematic drawing of the heat pump.

3. METHODOLOGY

Two tests with different evaporation temperatures were made. In each of them, step disturbances were made in the water flow through a needle valve located in the cooler entrance. The tests were conducted with no water accumulation in the reservoir.

The first test was made with the evaporator in the shadow (zero solar radiation) and the water flow was set to 1.05 L/min. In these conditions, the water temperature in the cooler outlet took approximately 20 minutes to settle. In this operation point, the evaporation temperature was equal to 3.3 °C. Then, a sudden increase in the water flow from 1.05 L/min to 1.48 L/min was made (an increase of approximately 40% in the water flow). Once again, the temperature of the water in the cooler outlet took some time to stabilize. After reaching the steady state, a new change in the water flow from 1.48 L/min to 0.95 L/min, was made and the steady state regime reached. Lastly, the heat pump was turned off. During the whole test, the acquisition system collected all the information from the sensors and the variables were saved in an EXCEL® file.

The second test was made with the heat pump exposed to a cloudy sky, with an average incident radiation of 103.4 W/m², with five turns in the expansion valve and an initial water flow of 0.99 L/min. As in the first test, the steady state regime was reached after a while, with the evaporation temperature kept at 13.12 °C. Afterwards, the water flow was suddenly increased from 0.99 to 1.42 L/min and, after waiting for the system to reach stability, the flow was suddenly reduced to 0.99 L/min.

4. RESULTS

4.1 Experimental Results

In Figs. 3 and 4 the conditions of the refrigeration cycle during the tests are presented, in which EIT, EOT, CIT, VIT, SD, WOT, FLOW are: Evaporator Inlet Temperature, Evaporator Outlet Temperature, Cooler Inlet Temperature, Expansion Valve Inlet Temperature, Superheating Degree, Water Outlet Temperature and the Water Flow, respectively.

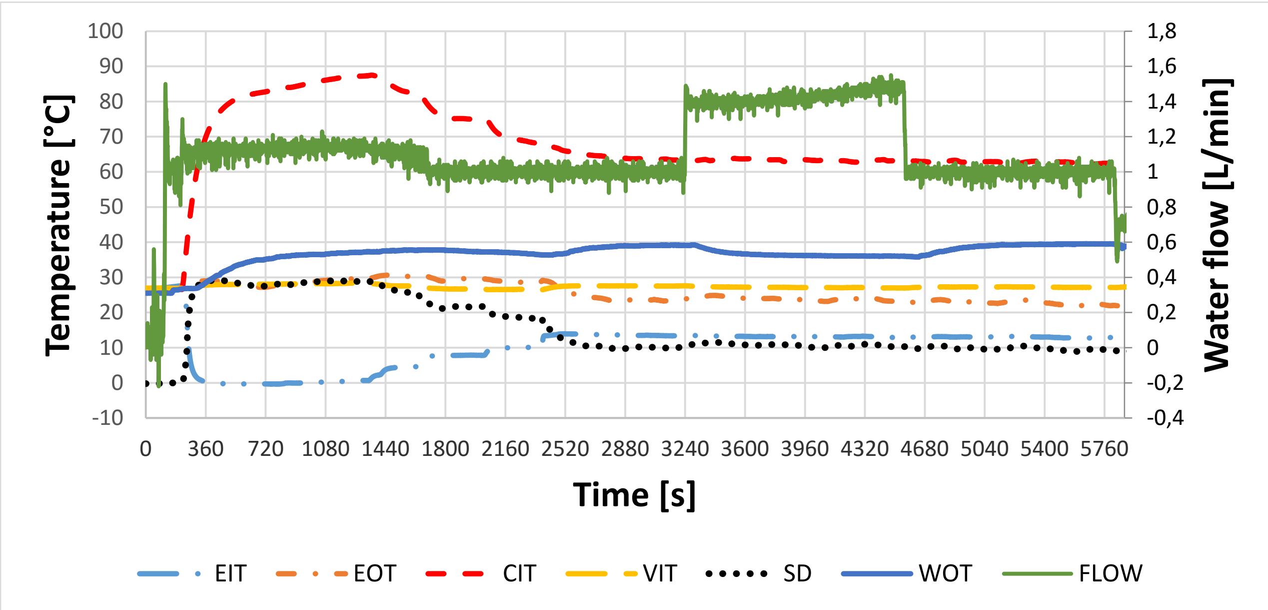

Figure 3. Behavior of the temperatures in the heat pump during the first test

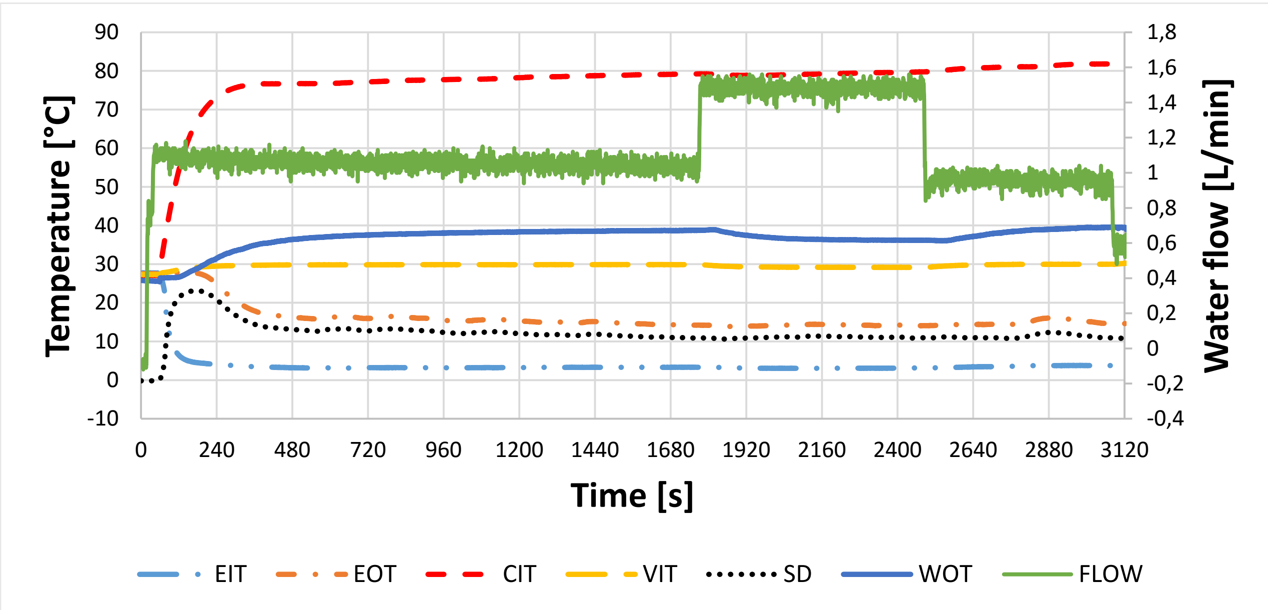

Figure 4. Behavior of the temperatures in the heat pump during the second test

The system will be represented by a First Order Plus Dead-Time system (FOPDT), which is used in a lot of representations of thermal systems and has solid literature available (Eq. 1).

$$G_{(s)} = \frac{K_p}{\tau\, s+1} e^{-\theta\, s} \tag{1}$$

Where the static gain (K_p) – determines the sensitivity of the process to a change in the entrance. The time constant (τ) – measures the time in which the monitored variable will reach its final value. The dead time (θ) – is the required time for the monitored variable to begin responding to a change in the manipulated variable.

4.2 Calculation of the parameters

The parameter K_p of the transfer function can be calculated through (Eq. 2).

$$K_p = \frac{\Delta y}{\Delta U} = \frac{y_f - y_0}{U_f - U_0} \tag{2}$$

Usually, the calculations of τ and θ are based on the response curve using two points, which correspond to certain levels reached by it when responding to a degree disturbance in open loop in relation to the steady state value of the system. These constants are calculated with the help of Eq. 2 and 3.

$$\tau = a_1(t_2 - t_1) \tag{3}$$
$$\theta = b_1 \cdot t_1 + (1 - b_1) \cdot t_2 \tag{4}$$

Table 1 presents the variation percentages in the response required to determine the two times necessary in the identification process and the value of the constants in the equations above for the identification methods suggested by Alfaro (2011), Chen Yang (2000), Ho et al. (1995), Smith (1972), and Vitecková et al. (2000).

Table 1. Values for the constants for the estimation of parameters for FOPDT models

Method	t_1	t_2	a_1	b_1
Alfaro	25,00	75,00	0,910	1,262
Chen and Yang	33,00	67,00	1,400	1,540
Ho et al.	35,00	85,00	0,670	1,300
Smith	28,30	63,20	1,500	1,500
Vitecková et al.	33,00	70,00	1,245	1,498

4.3 Model validation methods

In this work, the 5 methods for obtaining the parameters of the mathematical model were evaluated by three validation methods with the intent of determining the best transfer function. These methods are: the mean square error method (MSE) (Eq. 5); the relative error method (RE) (Eq. 6); and the method of the percentage of the variation in relation to the real value (FIT) (Eq.7):

$$MSE = \frac{1}{n}\sum_{n=1}^{n}\left(\hat{Y}_i - Y_i\right)^2 \tag{5}$$
$$RE = 100 \cdot \frac{1}{n}\sum_{i=1}^{n}\left|\frac{Y_i - \hat{Y}_i}{Y_i}\right| \tag{6}$$
$$FIT = \left(1 - \frac{\sqrt{\sum_{i=1}^{n}(Y_i - \hat{Y}_i)^2}}{\sqrt{\sum_{i=1}^{n}(Y_i - \bar{Y}_i)^2}}\right) \cdot 100 \tag{7}$$

The transfer function parameters for a first order system (Eq.1) can be calculated with the help of the constants suggested by the authors presented in Table 1 for (Eq. 3) and (Eq. 4). These calculations are done by a programmed routine in the software Matlab®, which calculates the parameters from the data obtained in the experimental tests. With the help of the validation methods, it was determined that the transfer function that best represents the system is the one represented by the parameters presented in Table 2.

Table 2. Results of the system identification

Test	K_p	τ	θ
1°	6,8620	160,1600	94,8880
2°	7,6403	196,5600	111,4080

5. CONCLUSIONS

The objective of this paper was to study the thermal behavior of the WOT produced in a heat pump operating with CO_2. With the experimental data on the behavior of the water temperature obtained in this study, it was possible to conclude that the variations in the insolation significantly affect the behavior of this kind of equipment. With the help of experimental tests involving a step variation in the water flow, it was possible to conclude that the response in the temperature of the water presented a high delay time, which can be explained by the existence of elements with the ability to store energy such as the tubings and even the water inside the tubings. By analyzing the response of the five methods used to identify the system, one can conclude that the dynamic of the system is different when an increase or a decrease step in the water flow is made, since the system is faster when an increase step is made in the water flow. This can be explained by the change in the heat transfer coefficient by the water flow in the gas cooler.

After doing a comparative analysis between the five methods to determine the parameters of the transfer function of the heat pump, it is possible to conclude that the best method to identify this system is the Alfaro method, followed by Ho et al., then Vitecková et al., then Chen and Yang and, lastly, the Smith method.

When identifying the parameters of the transfer function that represents each of the tests with both their steps in the flow, the identification quality was always better when the step reducing the flow was used for the identification.

The best tool for selecting the identification method that best describes the system was the FIT parameter, because it allowed for a clear, quantitative and percentage comparison of how good did the response curves for each of the different methods coincided with the experimental data obtained from the prototype.

Also, when the prototype heat pump is working under constant climatic conditions, the refrigeration cycle's temperatures are not significantly affected by the step variations made in the water flow entering the gas cooler.

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7. RESPONSIBILITY NOTICE

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