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Effects of the electronic expansion valve and variable velocity compressor on the performance of a refrigeration system

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Abstract. Energy consumption of air-conditioning and refrigeration systems is responsible for about 25 to 30% of the energy demand especially in hot seasons. This equipment is mostly electricity dependent and their use in principle affects negatively the environment. Enhancing the energy efficiency of the existing equipment is important as one of the measures to reduce environment impacts. This paper reports the results of an experimental study to evaluate the impacts of the use electronic expansion valve and variable velocity compressor on the performance of vapor compression refrigeration system. The experimental rig is composed of two independent circuits one for the vapor compression system and the other is the secondary fluid system. The vapor compression system is composed of a forced air condenser unit, evaporator, hermetic compressor and expansion elements, while the secondary system has a pump for circulating the secondary fluid, and an air conditioning heat exchanger. The manufacturer's data was used to determine the optimal points of operation of the system and consequently tests were done to evaluate the influence of variation of the compressor velocity and the opening of the expansion device on the performance of the refrigeration system. A fuzzy logic model was developed to control the rotational velocity of the compressor and the thermal load. Fuzzy control model was made in LabVIEW software with the objective of improving the system performance, stability and energy saving. The results showed that the use of fuzzy logic as a form of control strategy resulted in a better energy efficiency.

Keywords: variable velocity compressor; electronic expansion valve; fuzzy control; coefficient of performance; efficiency

1. Introduction

Energy consumption and demand substantially increased in recent years, as a result of the increase of the world population, improvements in living standards, and the increased use of electricity dependent equipment such as refrigeration and air-conditioning systems for food

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preservation and thermal comfort. The substantial increase in energy generation and consumption led to nearly exhausting the ecosystem capacity to handle the ever increasing loads of contaminants, offensive emissions and the continuous degradation. This serious alert forced the world population to search for new sustainable energy resources, and increase the energy utilization efficiency among other measures. Since conventional refrigeration systems are big energy consumers responsible for a substantial share of energy consumption on the domestic and commercial levels, there is a need for continuous research to improve the performance of existing systems and develop new economic and environment friendly concepts and machines.

Recent improvements in conventional vapor compression systems include new environment friendly refrigerants, compressors with variable velocity and adjustable electronic expansion valves among others. The adjustment of the opening of the expansion element directly affects the flow of the refrigerant in the circuit that influences the degree of superheat, and provides a pressure difference between the condenser and evaporator.

Electronic expansion valves (EEV) operate through a control system, usually PID, which increases or decreases the flow of the refrigerant to the evaporator according to the thermal load. Normally, the control system is embedded in a microcontroller, and has as its inputs the signals coming from a temperature sensor (thermistor) and a pressure transducer. Thus, the valves are able to control the flow of refrigerant and, consequently, the thermal load of the system (Choi and Kim 2003, Shanwei *et al.* 2005, Ataş *et al.* 2017). In comparison with the (TEV), the EEV produces more energy economy (Aprea and Mastrullo 2002, Lazzarin and Noro 2008).

The capillary tubes and coiled capillary tubes are other expansion devices widely used in conventional refrigeration systems because of their simplicity and low cost, however, they are not adequate for a system that requires precise control (Zhang 2005, Park *et al.* 2007).

Recent ongoing research follows new routes for simulation and control of EEVs, including Neuro-fuzzy and artificial neural controls. These studies are based on dynamic mathematical models to achieve good operating stability (Xia and Deng 2016, Shang *et al.* 2016, Dantas *et al.* 2017, Tesfay 2018).

The variation of the compressor rotation velocity is also an efficient way to control the system cooling capacity, which is normally adjusted according to the load (Tassou and Qureshi 1998). Thus, for a low load condition, the refrigeration apparatus can decrease the compressor rotation velocity, hence reduce losses and power consumption. The use of compressor velocity control to adjust the cooling capacity of the system has been extensively studied in the last years (Aprea *et al.* 2004, Aprea and Renno 2004, Koury *et al.* 2001, Kizilkan 2011).

Important aspects such as using new strategies (PID, Fuzzy, ANN) and technologies to improve the performance through monitoring and control of the system are key tools to automate and optimize refrigeration and air conditioning systems with EEV and VVC. These strategies improve energy utilization efficiency and reduce greenhouse gas emissions (Filho *et al.* 2011, Lago 2016, Ekren *et al.* 2010, Gill and Singh 2017).

The present study is conducted to investigate experimentally the interaction between the variable velocity compressor and the electronic expansion valve in a refrigeration system and identify the optimum points of operation for maximum efficiency. Experimental tests were conducted with the EEV operating in controlled form and with variation of velocity of the compressor. A fuzzy logic model was developed to control the velocity of the compressor and the thermal load.

The contribution of the present study to the research area of the conventional refrigeration systems includes comparative experimental study and results of different expansion devices and the development of a velocity control model for the compressor to enhance the energy efficiency.

The manuscript is divided into the following sections: introduction including literature revision, description of the experimental rig with experimental conditions and test procedure, error analysis and propagation, system analysis through thermodynamics parameters and fuzzy logic model, results and discussion, conclusions, references and nomenclature.

2. Experimental rig

The general description of the experimental rig, equipment and instrumentation, experimental procedure and uncertainty and propagation of error are presented in this section.

2.1 Description of the experimental rig

Fig. 1 shows the test rig composed of a conventional refrigeration vapor compression system that uses R-134a as a refrigerant and a secondary fluid system for air conditioning using ethanol as working fluid. The compressor is hermetic variable capacity model of EMBRACO. A frequency inverter controlled by an external frequency signal provokes the variation of compressor rotational velocity. The frequency range is 53-150 Hz, and the velocity range is 1600-4500 rpm.

The condenser, manufactured by ELGIN, is a forced air heat exchanger fitted with copper tubes and fins to ensure heat rejection efficiency. Its thermal capacity is 3/4 HP (559.3 W).

The evaporator is a concentric tube countercurrent flow heat exchanger. The internal copper tube is of 1300 mm length and 6 mm diameter, into which R-134a circulates. The secondary fluid (ethanol) flows through the annular section (external tube) of 17 mm diameter.

The thermostatic expansion valve is TN 2 DANFOOS model with orifice code number 00. The operation of the thermostatic valve depends on the pressure and the temperature of thermostatic bulb installed at the evaporator outlet to continuously monitor the temperature of the refrigerant flowing out of the evaporator, promoting refrigerant passage or restriction thereof.

The electronic expansion valve, E2V03 model is donated by CAREL Company. The temperature and pressure sensors at the evaporator outlet, allow the determination of the actual superheat by the controller which feeds the actuator of the electronic expansion valve. The driver of the electronic valve (EVD evolution), provides the stator with a low-voltage signal to rotate the rotor clockwise or counterclockwise while the internal mechanism converts the rotary motion into axial displacement of the rod.

The secondary fluid system is composed of centrifugal pump and fan coil. The pump has a flow capacity varying from 5 to 40 L/min and variable velocity control from 0 to 3450 rpm. The secondary refrigerant circulates through the fan coil to remove the heat load. The capacity of the heat exchanger of the air conditioning system is 600 W.

Monitoring the operating conditions of the refrigeration system is important to control its operation. The test rig is equipped with calibrated instruments to measure temperature, pressure, flow rate, current, voltage and the data is collected by the data acquisition system from Novus and National Instruments. To determine the overall efficiency of the compressor a calibrated power meter is installed. The power meter used is from Analyzer Power Company of INSTRUTHERM, model MPR-40. The instrument is connected to the compressor and the consumed power is measured (W) in real time.

There are two frequency inverters. A frequency inverter for the motor of the pump in the secondary refrigerant circuit, which has an analog input 0 to 10 V (DC) and controls the motor rotation velocity in the range 0 rpm to 3600 rpm. The second frequency inverter controls the velocity of the motor of the fan coil.

To evaluate the performance of the compressor a waveform generator is used to generate a square wave signal with frequency from 53.4 to 150 Hz, to be fed to the frequency converter (VVC inverter).

The drive used for the valve control is the EVD Evolution with PID control, model EVD 0000E50. To control the super heating, the drive is adjusted according to the refrigerant and capacity of valve refrigeration, with the temperature and pressure sensors connected. For data acquisition the Field Logger from NOVUS Automation is used to monitor all temperature sensors and pressure transducers in the system. A signal conditioning connection block (NI SC-2345) and a module (NI SCC FT-01) are used to send analog signals to the actuator.

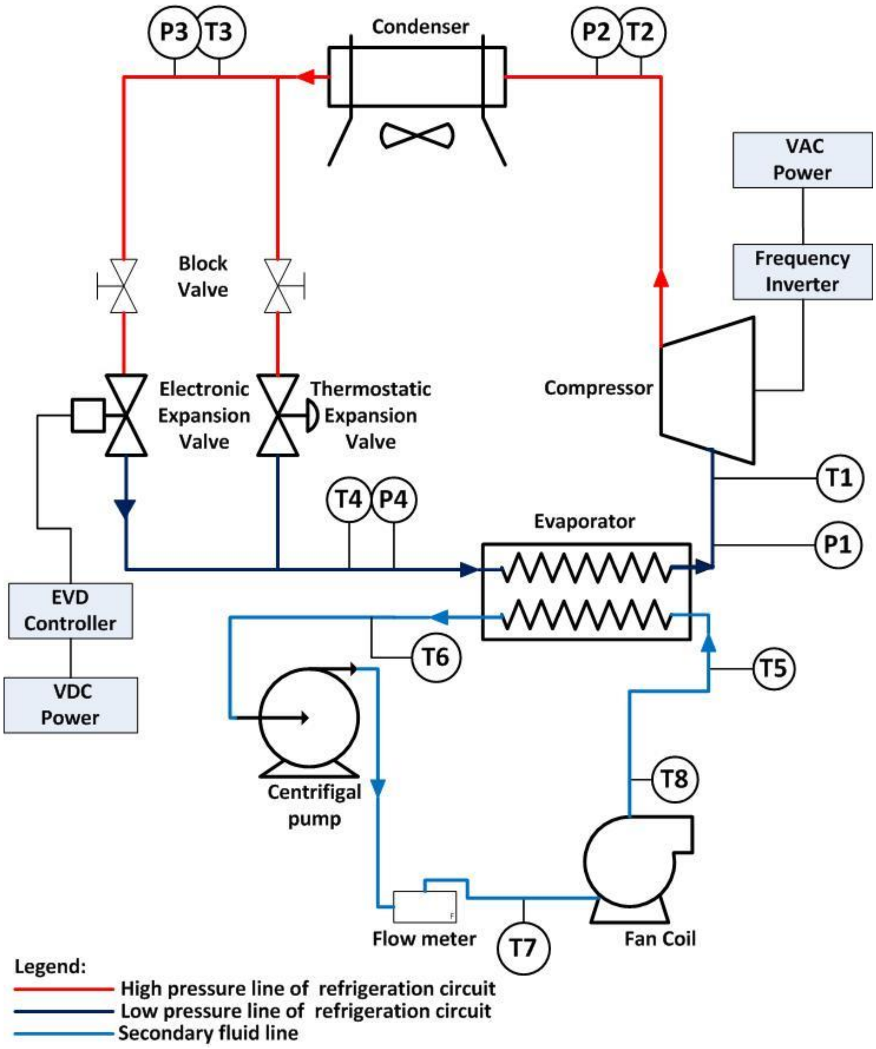


Fig. 1 Simplified scheme of the experimental configuration

Table 1 Uncertainty propagation of variables

Variable	Uncertainty
Temperature of evaporation and condensation	± 0.51 °C
Superheat and subcooling	± 0.51 °C
Pressure	± 0.0042 a ±0.0202 bar
Flowrate R-134a	1.40 x10^{-4} a 3.4x10^{-4} kg/s
COP	± 0.05 a 0.08
Thermal load	± 11.76 a 27.46 W
Efficiency	± 1.02 a 1.88 %
Compression Power	± 0.88 a 2.94 W

2.2 Experimental conditions and test procedure rig

Two types of tests were performed: 1) Tests with variation of compressor rotational velocity and control the electronic expansion valve; 2) Tests with the developed model of the compressor velocity control. These tests were carried out to assess the impact of the variable velocity compressor and the electronic expansion valve on the performance of the cooling system, to monitor the operating points for higher thermal efficiency, more thermal stability and energy saving.

Initially, the compressor rotation velocity was changed by varying the frequency in steps of 10 Hz of the compressor maximum velocity, 150 Hz (4500 rpm) down to 70 Hz (2100 rpm). The changes of the frequency were made to coincide with the operation points in the manufacturer's manual. The final set of tests was realized after the implementation of fuzzy logic in the software LabVIEW for the control of the compressor rotational velocity.

The tests were conducted according to the (ISO 917 1989) standard, which indicates the necessary requirements for testing refrigeration compressors. Monitoring the pressures of the system over time was used to characterize the system's permanent regime, since the suction and discharge pressures directly influence all other dependent variables, such as the degree of superheat and the refrigeration capacity, etc. Hence, when the steady-state criterion is satisfied for the suction and discharge pressures, all other dependent variables are considered stable in the chosen time interval.

2.3 Propagation of experimental errors

The measuring instruments were calibrated and installed at the required points of the circuits. The experimental uncertainties of the calculated variables of this study were determined in accordance with the standard statistical procedures with 95% confidence interval. Table 1 presents the propagation of uncertainty of the calculated variables.

3. System analysis

An experimental planning matrix associated with permanent regime thermodynamic calculations was elaborated in order to assess the behavior of the variables of the system. The

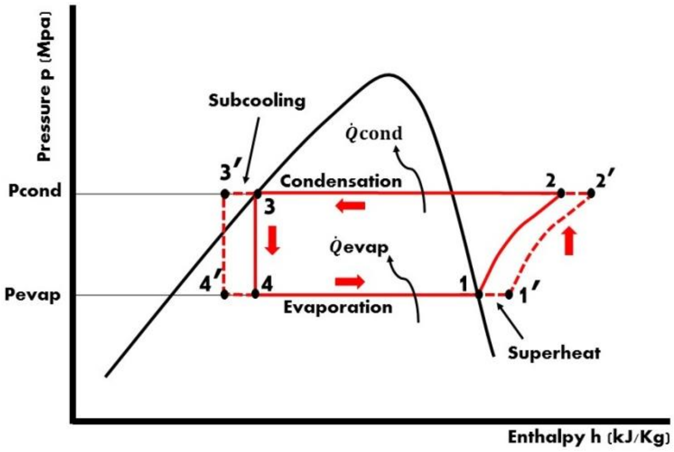


Fig. 2 P-h diagram of vapor compression refrigeration real cycle

performance parameters of the refrigeration cycle, such as the COP and compressor work were investigated for various evaporation and condensation temperatures. A control model with fuzzy logic was created to control the rotational velocity of the compressor.

3.1 Thermodynamic parameters

By using the first Law of Thermodynamics it is possible to enclose each component of the refrigeration system into a control volume and establish, in permanent regime, the balance of mass and energy involved. Fig. 2 shows the P-h diagram of the refrigeration cycle corresponding to the experimental set-up shown in Fig. 1.

From the theoretical analysis of a reversible adiabatic compressor it is possible to determine the consumed compression power, based on the enthalpy values of the refrigerant mass at entry and exit of the control volume:

$$\dot{W}_c = \dot{m}(h_2 - h_1) \tag{1}$$

where, h_1 and h_2 are the refrigerant enthalpies at inlet and outlet of the compressor. The amount of heat per unit time, discarded in the condenser, from the superheated vapor is given by:

$$\dot{Q}_c = \dot{m}(h_2 - h_3) \tag{2}$$

where, h_2 and h_3 are the refrigerant enthalpies at inlet and outlet of the condenser. Considering the simplified control volume for the evaporator, where the primary and secondary fluids flow under steady state conditions, the thermal load for the refrigerant and for the secondary fluid are given by Eqs. (3) and (4), respectively.

$$\dot{Q}_{R134a} = \dot{m}_{R134a}(h_1 - h_4) \tag{3}$$

$$\dot{Q}_{fsec} = \dot{m}_{fsec}(h_{fs,out} - h_{fs,in}) \tag{4}$$

where, h_1 and h_4 are the enthalpies of the refrigerant at the evaporator outlet and inlet, and $h_{fs,in}$ and $h_{fs,out}$ are the enthalpies of the secondary fluid at the evaporator inlet and outlet. Applying the law of energy conservation:

$$\dot{Q}_{R134a} = \dot{Q}_{fsec} = \dot{Q}_{evap} \tag{5}$$

Knowing the pressures and temperatures of the refrigerant at the evaporator inlet and outlet, as well as the compressor inlet and outlet temperatures, the performance coefficient, COP, for the ideal cycle can be defined in terms of the enthalpies:

$$COP_{theoretical} = \frac{h_1 - h_4}{h_2 - h_1} \tag{6}$$

In order to determine the actual COP, is determined from the ratio of the evaporator cooling capacity and the power consumed by the compressor as in Eq. (7),

$$COP_{actual} = \frac{\dot{m}_{R134a}\ \Delta h_{evap}}{\dot{W}_{c,real}} \tag{7}$$

where, $m_{R134a}\Delta h_{evap}$ and $W_{c,real}$ are the evaporator cooling capacity and the power consumed by the compressor, respectively.

The ratio of the ideal compression work to the real compression work is defined as the total compression efficiency, and is used to determine the performance of the compressor by means of Eq. (8).

$$\dot{\eta}_c = \frac{\dot{m}_{R134a}\ \Delta h_{compr}}{\dot{W}_{c,real}} \tag{8}$$

where, $\dot{\eta}_c$, $m_{R134a}\Delta h_{compr}$ and $W_{c,real}$ are the total compression efficiency, the isentropic compressor work, the real compressor work, respectively.

Two other important factors that make the cycles differ from one another are superheat at the suction of the compressor and subcooling of the refrigerant at the outlet of the condenser and are formulated as in Eqs. (9) and (10):

$$SH = Tem.of\ the\ Suction\ Line - Temp.of\ Saturation \tag{9}$$

$$SB = Tem.of\ Condensation - Temp.of\ the\ Liquid\ Line \tag{10}$$

It is important to observe that the enthalpy of the refrigerant is obtained indirectly by measuring the temperature and pressure. Thus, it was possible to determine the enthalpies by inserting the temperature and pressure data obtained by the Field Logger in CATT3 program (Computer Aided Thermodynamic Table 3).

3.2 Fuzzy logic model

In order to improve the energy efficiency of the refrigeration system, a fuzzy logic based control was developed to control the rotational velocity of the compressor and hence adjust the air

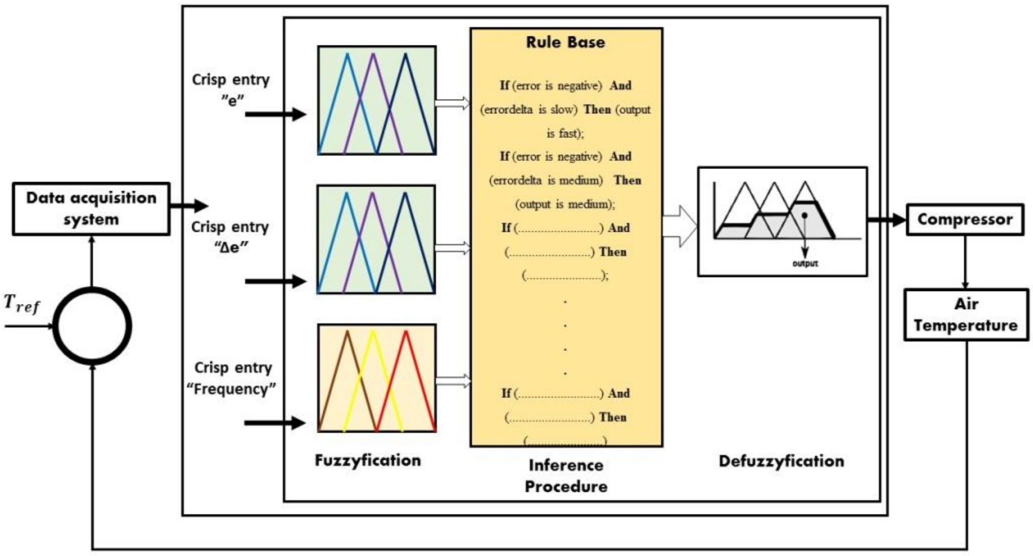


Fig. 3 Block diagram of the Fuzzy control system

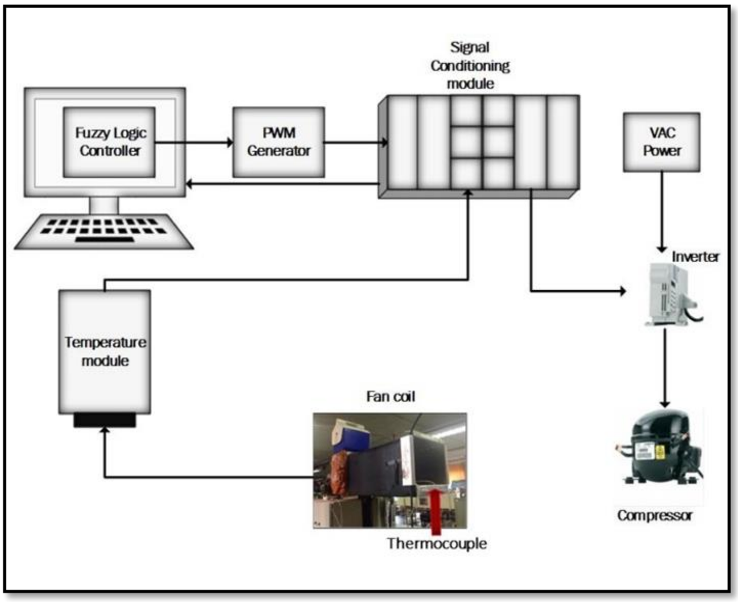


Fig. 4 Basic configuration of the developed model

temperature at exit of the fan coil to the desired value. Fig. 3 shows the fuzzy block diagram of the refrigeration control system as a closed loop system. The controller is designed to provide the correct value of the AC voltage frequency so that the compressor runs at the rotation velocity, which maintains the temperature at the desired set point.

The typical input variables in a fuzzy control are evaluated by the temperature error, (e), and the error change, Δe, defined by the following equations:

$$e = T_{setpoint} - T_{air} \tag{11}$$

$$\Delta e = \frac{d\left(T_{setpoint} - T_{air}\right)}{dt} \tag{12}$$

The control of the compressor rotation velocity is done by using a fuzzy controller developed by the Fuzzy System Designer tool inside the LabVIEW platform. The control inputs are the fan coil air outlet temperature measured in real time and the set point of the same temperature (whose values are obtained from the experimental data of the variation of the velocity of the compressor in consonance with the variation of the thermal load). The perturbations in the dynamics of the system were introduced by the variation of the thermal load (changing the rotation of the fan coil and the pump promoting the change of the set point of the temperature of the air at exit of the fan coil).

To implement the control system the triangular membership function was used since there is no discontinuity between the change in function growth and its rapid processing. The three pertinence functions for input variable *Error* (set point temperature minus air outlet temperature) are: negative, zero, positive. For *Deltaerror* variable, the three pertinence functions were used: low, medium, high. For output variable of the *Frequency-comp-controller*, the universe of speech is defined in the interval of [53,150] Hz. The diagram of the basic electrical circuit for rotation velocity control of the compressor is shown in Fig. 4.

The fuzzy rules for the controller are constructed based on the information on the refrigeration system. The inference mechanism used to provide the output of the controller is the Mamdani (Mamdani 1974) method and for the defuzzification the center of mass or centroid is used. Thus, by reading the values of the input variables the output is the value that corresponds to the center of gravity of the output nebulizer. The defuzzification consists of the weighted average of the centroid points of the X's axis and the calculated areas.

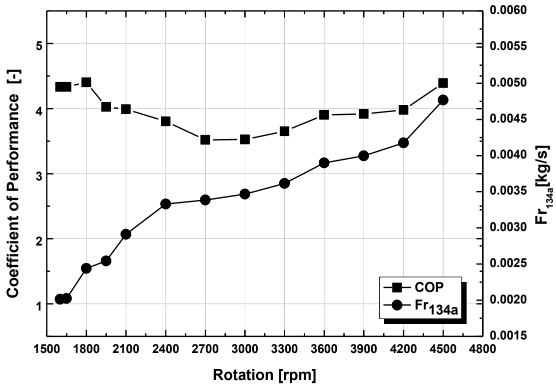
4. Results and discussion

4.1 Effects of the EEV and VVC on the performance of the system

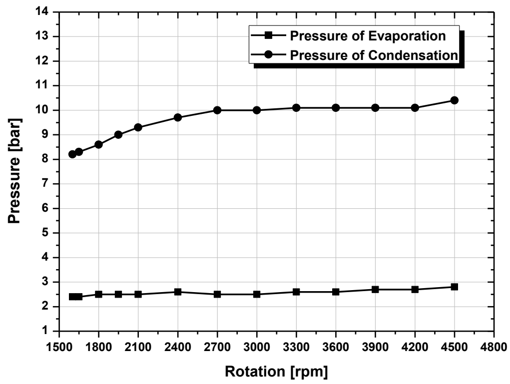
The tests were conducted to evaluate the electronic expansion valve with the compressor rotational velocity. The tests were done for compressor velocity range from 1602 rpm to 4500 rpm at an initial mass flow rate of ethanol of 0.072 kg/s.

The experiment started by operating the system at the maximum velocity of 4500 rpm (150 Hz) and the electronic expansion valve automatically controlled to 8°C of superheat. The frequencies of the inverters of the pump and fan coil were adjusted according to the heat load required for such rotational velocity to achieve steady state and evaluate the COP and the other parameters. In sequence, the compressor rotational velocity is reduced by changing the frequency in steps of 10-10 Hz to rotation velocity 2100 rpm (70 Hz), then variation of 5-5 Hz to rotation velocity 1650 rpm (55 Hz) and a final variation 1.6 Hz to the minimum velocity.

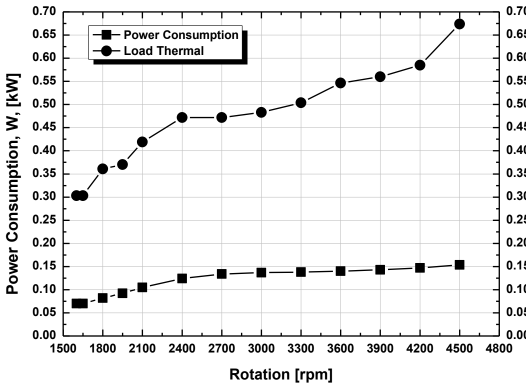
Fig. 5 shows the effects of varying the rotation velocity on the thermal performance parameters of the system. Fig. 5(a), shows the changes in COP and flow rate (Fr), in Fig. 5(b), one can observe the variations in the condensing and evaporation pressures, while Fig. 5(c), shows the



(a) Effects on the COP and the refrigerant mass flow rate (R134a)



(b) Effects on the pressures of evaporation and condensation.



(c) Effects on the thermal load and power consumption

Fig. 5 Variations of the performance parameters with the compressor rotation velocity

variations in the measured power consumption and thermal load of the system.

Table 2 shows that the use of the electronic expansion valve causes a reduction in the condensation temperature within the whole operating range of the compressor in comparison with the thermostatic expansion valve (TEV). This is because the EEV operates with lower values of condensation pressure and also the EEV has fast reaction to changes in capacity, facilitating the rapid stabilization of the system. Hence, for the rotational velocity 1602 rpm and 2100 rpm, the COP is higher, Fig. 5(a), because of the low condensation temperature, lower compression rate and lower compressor power consumption, Fig. 5(b) and Fig. 5(c). With the increase of the compressor rotational velocity from 2400 rpm to 3900 rpm, the condensation pressure increases, Fig. 5(b), there is an increase in the mass flow of the refrigerant and thermal load, however, the power consumption is higher as in Fig. 5(c), and the COP is lower, Fig. 5(a). This occurs because from the rotation velocity of 2700 rpm the power consumed increases considerably until reaching its maximum value at the rotation velocity of 4500 rpm. If the condensation pressure increases, there is a higher compression ratio and higher power consumption values of compressor. The COP is a relation of thermal load with consumed power. Thus at 2700 rpm and 3000 rpm the COP values are low (Fig. 5(a)), because the power consumed has increased almost to its maximum while the

Table 2 Variables of the system operating with expansion valves

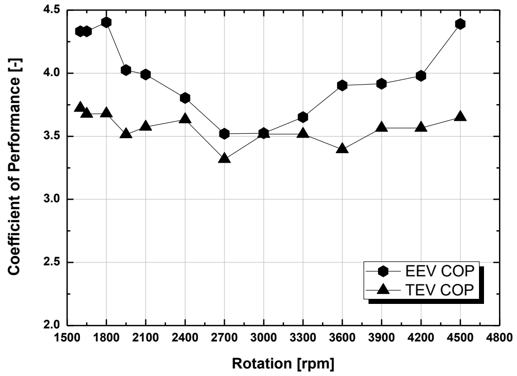
Rotational velocity [rpm]	TEV			EEV		
	Evaporation Temperature [°C]	Condensation Temperature [°C]	Power Consumption [kW]	Evaporation Temperature [°C]	Condensation Temperature [°C]	Power Consumption [kW]
4500	-5.36	35.93	0.148	-2.2	39.01	0.1535
4200	-5.36	36.32	0.147	-2.2	39.76	0.147
3900	-5.36	37.11	0.147	-4.3	39.76	0.143
3600	-5.36	37.88	0.147	-4.3	39.76	0.140
3300	-5.36	38.26	0.146	-4.3	39.76	0.138
3000	-5.36	39.01	0.148	-5.36	39.39	0.137
2700	-5.36	40.49	0.146	-5.36	39.76	0.134
2400	-5.36	42.27	0.127	-6.48	39.39	0.124
2100	-5.36	42.27	0.11	-6.48	39.39	0.105
1950	-5.36	42.27	0.098	-5.36	38.26	0.092
1800	-5.36	42.57	0.087	-5.36	36.72	0.082
1650	-5.36	42.87	0.08	-5.36	35.53	0.07
1602	-5.36	43.01	0.079	-5.36	33.89	0.07

developed thermal load is still at its intermediate range. In the range, 4200 rpm (140 Hz) to 4500 rpm the COP is high because the thermal load and refrigeration flow rate are the highest, and the power consumption is reasonable. Despite the increased power consumed, due to a higher condensing pressure, there was only a small variation of power consumed from rotational velocity of 3000 rpm (100 Hz) to maximum rotational velocity of 4500 rpm (150 Hz).

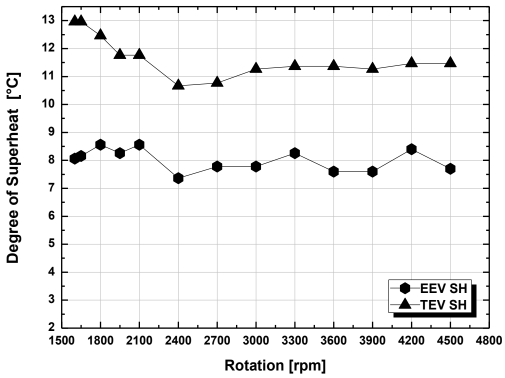
To evaluate the performance of the two expansion devices, TEV and EEV, the COP, the degree of superheat and the efficiency were used as indicators of performance and are presented in Fig. 6. The two expansion valves were set at the same evaporation temperatures, as shown in Table 2.
In the case of the TEV, the COP varied in the range 3.5 to 3.75 while in the case of the EEV the COP varied in the range 3.6 to 4.4 for the rotational velocity range according to the thermal loads, as shown in Fig. 6(a). One can observe that the use of the electronic expansion valve improves the performance coefficient of the vapor compression refrigeration system, since the COP is closely dependent on the evaporation and condensation pressures. The use of the electronic expansion valve reduces the condensation temperature for the whole range of rotational velocity of the compressor, thus reducing the compression work, causing an increase of the cooling efficiency and the COP.

The evaporation temperature of -6.48°C to -2.2°C with the condensing temperatures in the range of 33°C to 39°C and the degree of superheat around 8°C, contributed to the good results of COP of the EEV for the whole range of thermal load, as compared to the temperatures values of the TEV in Table 2.

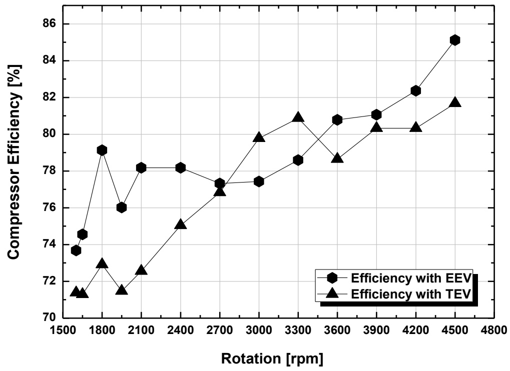
The use of the electronic expansion valve controls the superheat and makes it more stable. The degree of superheat of the electronic expansion valve is around 8°C while that of the thermostatic expansion valve is higher for the whole range of rotational velocity of the compressor and unstable for smaller rotational velocities than 2400 rpm as can be seen in the Fig. 6(b).



(a) Coefficient of performance



(b) Degree of superheat

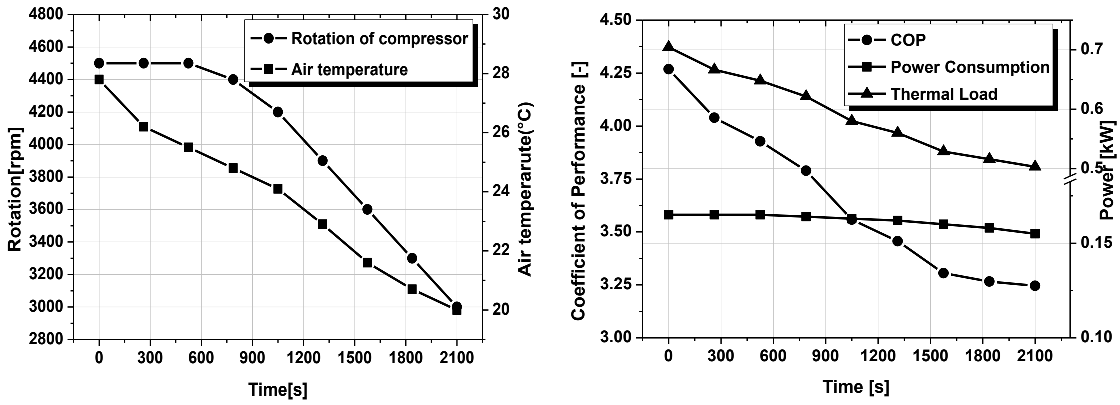


(c) Total efficiency

Fig. 6 Comparison of parameters between TEV and EEV for various compressor rotation velocities

The orifice of the thermostatic expansion valve was chosen according to the capacity of the system based on the evaporation and condensation temperatures. The static superheat value of the valve is as set by the manufacturer, and can also be adjusted manually. The set point values set by the manufacturer is 7℃ with maximum operation pressure (MOP) and 5℃ without MOP, where the MOP value is the evaporation pressure at which the expansion valve will restrict the injection of liquid refrigerant into the evaporator, thereby preventing the evaporation temperature from increasing and consequently the suction pressure. After reaching the MOP point, increasing the temperature of the thermostatic expansion valve will not cause any more opening of the expansion valve.

Fig. 6(b) shows the variation of the degree of superheat of the system for both the electronic expansion valve and the thermostatic expansion valve as function of the rotation velocity of the compressor. The TEV valve shows superheat values in the range from 10.7 to 12.96°C, higher than the manufacturer's set point. These were the best obtained values of superheating with the thermostatic expansion valve, because by setting the degree of superheating manually the MOP point will change, the valve would only respond to a certain thermal load and opening the valve more the value of the superheating is very low, causing blow of liquid in the compressor.



(a) (b)

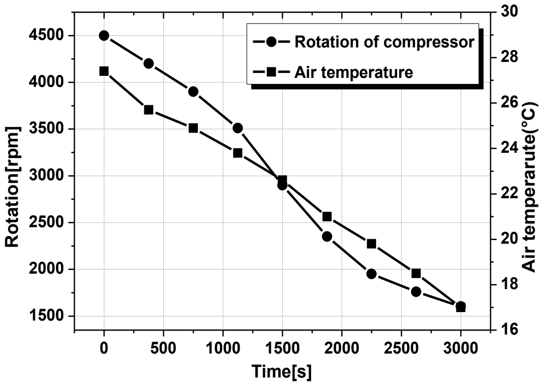
Fig. 7 Behavior of the system for the set point temperature of 20°C: (a) Rotation velocity and air temperature and (b) COP, power consumption and thermal load

The electronic expansion valve with EVD controller action has a pressure transducer and a thermistor at the evaporator outlet to forward the information for opening or closing the valve and maintains the degree of superheat around 8.0°C (the manufacturer's set point for a refrigeration capacity of 1.2 kW). This value of superheat is ideal for the present system, because if the superheat is too low, a return of liquid to the compressor may occur and damage it. However, it cannot be too high to avoid significant reduction of the COP. As already mentioned, the reduction of the condensation temperature increases the cooling efficiency, since it reduces both the compression ratio and the power consumption, which is one of the advantages of EEV. The variation of the total efficiency of the compressor with the rotation velocity of the compressor for both valves is presented in Fig. 6(c). Both valves show good efficiency, but the use of EEV is more advantageous due to the low power consumption. At the low rotation velocity range, the system equipped with EEV showed efficiency of about 6.5% higher than that of the TEV, while at the high rotation velocity range, the system fitted with EEV has global compressor efficiency 4.5% higher than that of the system fitted with TEV. Then, system equipped with EEV has the highest compressor efficiency because it presents lower compression rates.

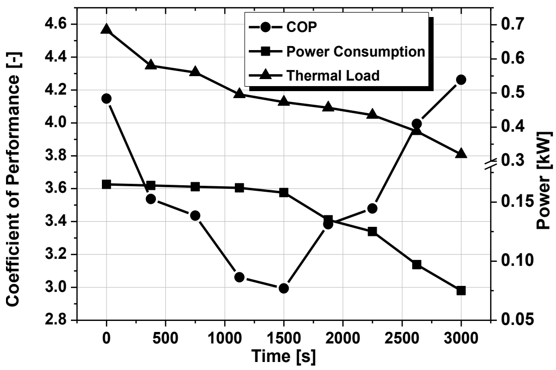
4.2 Effects of the developed compressor velocity control model

To analyze the response of the control of the compressor rotational velocity, several tests were made with the test rig operating in closed loop until achieving the steady state regime. under which simulations of the changes of the evaporator thermal load were applied to the model for two air temperature set points of 20°C and 17°C. This two set points of air temperature were chosen to characterize the air-conditioned environments.

The variation of the temperature as input to the proposed model and the rotational velocity as response are shown in Fig. 7. It can be seen that the fan coil exit air temperature is the room temperature and the set point is fixed at 20°C, hence the controller makes the compressor to work at maximum velocity of 4500 rpm (150 Hz) and maintains it constant for some time. The consumed power is 165 W, and the COP shows high values at the beginning as a consequence of the power consumption and the high thermal load.



(a)



(b)

Fig. 8 Behavior of the system for the set point temperature of 17°C: (a) Rotation velocity and air temperature and (b) COP, power consumption and thermal load

As the temperature approaches the set point temperature, a drop occurs in the rotational velocity of the compressor. This variation of the rotational velocity is due to the fact that as the system shows a decrease in the thermal load and approaches the set point temperature with compressor still in operation, the temperature variation becomes fast and consequently the rotational velocity decreases. There is no significant variation in the consumed power of the compressor (165 W to 155 W) working within the range 4500 to 3000 rpm, which means an abrupt decrease of the COP due to the thermal load (503 W) in the final interval.

Fig. 8 shows the rotational velocity response of the compressor motor for a set point temperature of 17°C. The difference between the set point temperature and the recorded temperature is even higher than the previous case and consequently the set point temperature is achieved after 50 minutes later. The control action involves reducing the rotation velocity of the compressor to match the thermal load being developed, and leading the system to a new stable regime. The fuzzy rule system allows running at full velocity for too long and when the set point temperature is reached the rotational velocity decreases to a minimum value. When the compressor rotational velocity is at its minimum of 1602 rpm, the power consumption is also at the minimum level of 75 W while the COP reaches its maximum value of about 4.26, Fig. 8(b).

Fig. 9 shows a comparison between conventional on/off compressor with CT (capillary tube), TEV and EEV and the proposed model with fuzzy logic. This allows assessing the real benefits based on the average power consumption, working a period of 8 hours daily during 30 days.

The energy consumption of the equipment in kWh / month is estimated from:

$$Consumption\ KWh/Month = \frac{Power \times numb.of\ hours\ used \times numb.of\ day\ of\ use\ in\ the\ month}{1000} \tag{13}$$

For case of set point temperature of 17°C, the conventional system with capillary tube (CT) consumes 144.72 kWh/month, the conventional system with thermostatic expansion valve consumes 82.56 kWh/month while the conventional system with electronic expansion valve (EEV) consumes 75.84 kWh/month and the proposed system consumes 60.96 kWh/month. When the set point temperature is 20°C the conventional system with CT consumes 138.96 kWh/month, the

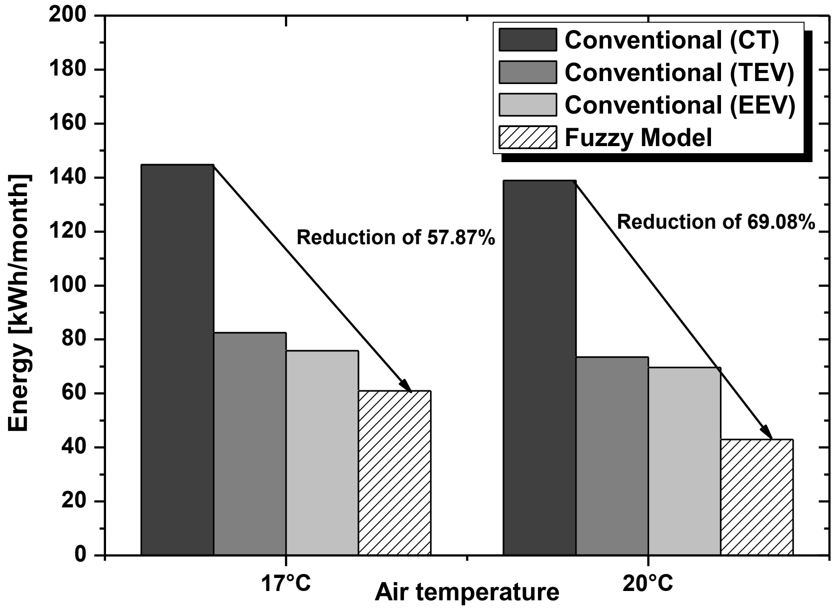


Fig. 9 Comparison of energy consumption between the On-Off conventional systems and fuzzy control system

conventional system with TEV consumes 73.44 kWh/month, the conventional system with EEV consumes 69.6 kWh/month and the proposed system consumes 42.96 kWh/month. Therefore, the energy consumption of the proposed Fuzzy system is much less than that of the conventional systems.

The use of EEV in the conventional system is more advantageous due to the low power consumption to attend the thermal load. For the 17°C temperature range, the system fitted with EEV showed energy consumption 8% lower than the system fitted with TEV and for the 20°C temperature range the consumed energy is 5% lower. Based on the above, it is possible to conclude that systems with EEV have big potential for energy reduction.

Fig. 9 shows that the proposed system with fuzzy logical saves more energy when working at high set point temperature as 20°C. This is due to the fact that higher set point temperature is easier to achieve since it is closer to the environmental temperature when the system is switched on and the compressor has to do less work in comparison to the case when the system is switched on at low set point temperature.

5. Conclusions

The present study is conducted to investigate experimentally the interaction between the variable velocity compressor and the electronic expansion valve in a conventional refrigeration system. The main conclusions are:

1. Systems with electronic expansion valves and variable velocity compressors are more efficient as they operate at a maximum COP point and provide cooling capacity equivalent to or

close to the required heat load demand.

2. The electronic expansion valve (EEV) operates with lower superheat values than the thermostatic expansion valves (TEVs).

3. Use of Fuzzy logic to control the rotational velocity of compressor leads to considerable energy savings. The consumed energy by the proposed system for set point temperatures of 17°C and 20°C is found to be less by 57.87% and 69.08%, than that consumed by the conventional On/Off refrigeration system.

4. For a low set point temperature, the energy consumption is higher, whether it is a conventional On/Off system or with the intelligent control.

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CC

Nomenclature

c_p specific heat capacity [$kJ \cdot kg^{-1} \cdot K^{-1}$]

H enthalpy

I Current

$\dot{m}$ mass flow rate [$kg \cdot s^{-1}$]

P pressure [bar]

$\dot{Q}$ heat [kW]

rpm	rotation per minute
SB	subcooling
SH	superheat
T	temperature [°C]
U_c	uncertainty
V	voltage [W]
$\dot{W}$	power [kW]

Greek symbols

η	compressor efficiency [%]

Subscripts & Superscripts

c	compressor
cond	condenser
evap	evaporator
fsec	secondary fluid
in	inlet
out	outlet
R134a	tetrafluoroethane
1	compressor inlet
2	compressor exit
3	condenser exit
4	evaporator inlet

Abbreviations

ANN	artificial neural network
COP	coefficient of performance

CT	capillary tube
EEV	electronic expansion valve
PID	proportional-integral-derivative
PWM	pulse width modulation
TEV	thermostatic expansion valve
VVC	variable velocity compressor