Dynamic Behavior Model for Cooling System Based on Vapor Compression

Experimental Analysis and Simulation Validation Grounded on a Reduced Order Differential Equation with Few Degrees of Freedom

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***Abstract*—This article presents the dynamic analysis of a single-stage vapor-compression refrigeration system. The model is based on the development of submodels for each component of the refrigeration system, applying the space state approach. The developing and verification of dynamic models for different configurations of thermal systems is critical in order to design control algorithms with optimal energy consumption. This analysis culminates with the comparison between direct performance of the refrigerant R134a as working fluid and the behavior temperature of the compartment in a refrigerator and the results obtained are similar in the evaporator temperature and pressure.**

Keywords-Heat exchangers; Refrigerants; Dynamic Model; Household refrigeration.

I. INTRODUCTION

Refrigeration and air conditioning is an active, rapidly developing technology. It is closely related to the living standard of the people and to the outdoor environment, such as through ozone depletion and global warming.

Mathematical modeling is the most practical way of studying the basic behavior of cycle performance, the relative losses in various components and interactions of their performance characteristics. Standard science and engineering formulations are applied to describe mathematically the basics processes occurring in the Vapor Compressor Refrigerating (VCR) systems. Mathematical modeling is not an end in itself but is a step towards simulation optimization.

The dynamic modeling of VCR system has been subject on interest since the late 1970s, where first principle models were used to describe the heat exchangers. Lumped parameters, moving boundary models [1]-[3], and then McArthur initiated a series of works focus on a distributed parameters formulation [4]. Besides this main modeling stream, other authors have designed more complex model for analyzing specific phenomena but we refer to review of [5] and to subsequent works and the references therein [6][7]. Later on in 1990s traditional feedback control has also been investigated, as in [8] and, more recently, multivariable control strategy have been developed, [6][7][9][10]; in other studies, Jensen and Skogestad 2007a, b [11][12], strategies are developed to select among the degrees of freedom the controlled and the control variables so that an optimal operation is nearly obtained.

Given that thermal dynamics of VCR systems are typically slower than the mechanical dynamics, the bulk of the model complexity generally resides in the heat exchangers. Previous literature reviews [13][14] indicated that the bulk of the research efforts is focused on capturing two-phase flow dynamics in the heat exchangers while seeking a balance between simplicity and fidelity. For the purpose of this article, the four elements of the system will be classified into lumped parameters models. Lumped parameter models refer to models that apply lumped parameter assumptions to the entire heat exchanger or to particular fluid phases within the heat exchanger (i.e. individual lumped models for superheated vapor, two-phase fluid, and subcooled liquid), in this article, the term “lumped parameter model” includes approaches that model the heat exchanger as single-control volume or multiple (time-invariant) control volumes for each fluid phase. Most of the publications in the lumped parameter classification are early efforts, where computational simplicity is paramount to ensure feasible computation times, motivating modeling efforts with few dynamics equations and few (lumped) parameters [15].

Many analytical models are used to simulate steady-state performance of refrigeration systems, but do not predict performance during transient operation. This paper presents a method for predicting the cooling performance of VCR systems during transient and various ambient conditions based on established steady-state performance. In this work,

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we adopt a dynamic model of refrigeration system similar [16] but this is simplest with only four control volume.

This research is carried out in CIDESI Queretaro, for the National Laboratory for Cooling Technologies Research (LaNITeF).

The rest of the paper is structured as follow. In Section II, we review refrigeration cycle principles. Section III presents an explanation of model linearized on VCR systems. Section IV describes the balance of mass and energy of each element of the system, then the four ordinary differentials equations are modeling in Simulink of Matlab. Section V describes the validation of the modeling. Conclusions are found in Section VI.

II. Refrigerating Systems

Vapor-compressor refrigerating systems used with modern refrigerators vary considerably in capacity and complexity, depending on the refrigerating application. They are hermetically sealed and normally require no replenishment of refrigerant or oil during the appliance's useful life. Systems components must provide optimum overall performance and reliability at minimum cost. In addition, all safety requirements of the appropriate safety standard (e.g., IEC Standard 60335-2-24; UL Standard 250) must be met. The fully halogenated refrigerant R-12 was used in household refrigerators for many years. However, because of its strong ozone depletion property, appliance manufacturers have replaced R-12 with environmentally acceptable R134a or isobutene.

Design of refrigerating systems for refrigerators and freezers has improved because of new refrigerants and oils, wider use of aluminum, and smaller and more efficient motors, fans, and compressors. These refinements have kept the vapor-compression system in the best competitive position for household application.

A. Refrigerating circuit

Figure 1 shows the refrigeration cycle on p-h diagrams. The refrigerant evaporates entirely in the evaporator and produces the refrigerating effect. It is then extracted by the compressor at state point 1, compressor suction, and is compressed isentropically from state point 1 to 2. It is next condensed to liquid in the condenser, and the latent heat of condensation is rejected to the heat sink.

The liquid refrigerant, at state point 3, flows through and expansion valve, which reduces it to the evaporating pressure. In the ideal vapor compressor cycle, the throttling process at the expansion valve is the only irreversible process, usually indicated by a dotted line. Some of the liquid flashes into vapor and enters the evaporators at state point 4. The remaining liquid portion evaporates at the evaporating temperature, thus completing the cycle [17].

Note that energy enters the systems through the evaporator (heat load) and through the compressor (electrical input). Thermal energy is rejected to the ambient by the condenser and compressor shell. A portion of the capillarity tube is usually soldered to the suction line to form a heat exchanger. Cooling refrigerant in the capillarity tube with the suction gas increases capacity and efficiency.

A strainer-drier is usually placed ahead of the capillarity tube to remove foreign material and moisture. Refrigerant charges of 150 g or less are common.

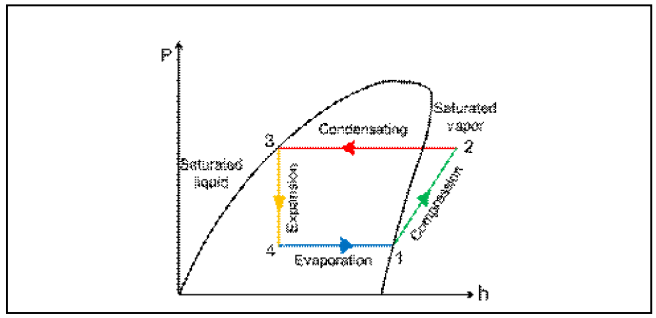

Figure 1. Diagram enthalpy vs Pressure.

A thermostat (or cold control) cycles the compressor to provide the desired temperatures in the refrigerator. During the off cycle, the capillarity tube allows pressure to equalize throughout the system [17][18].

III. Linearized Model

Considering the dynamic response of commercial coolers based on vapor compression, various specialists have chosen to simplify the dynamic model on the basis of its ability to retain heat and the thermal resistance of its barriers [19][20]. These models consider the evaporator as a source of heat, which can inject or extract energy from the thermodynamic system to the inside of the cooler. Thus, C is considered as the thermal capacity of the volume inside the cooler and R heat resistance of its barriers, the linearized mathematical model is established as:

$$Q(t) = C\frac{dT_i}{dt} + [T_i - T_e]/R \quad (1)$$

Where $Q(t)$ is the heat injected or absorbed by the source in this case the evaporator, T_i and T_e are considered internal and external temperatures respectively. The model is easily recognizable as a first-order dynamic system and the internal temperature depends on the motor compressor angular speed, as shown in Figure 2.

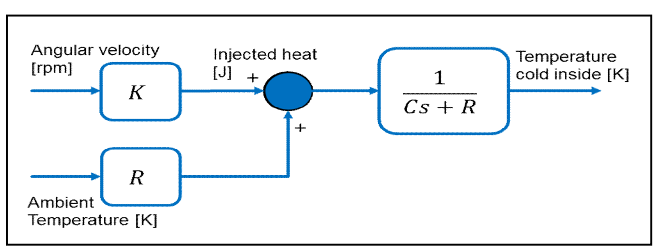

Figure 2. The Dynamic Model Analized Scheme.

In the same approach of the open-loop model, there are several control strategies implemented to regulate the internal temperature by applying a control law to manipulate the compressor speed.

As is the case with the thermodynamic model, there are several efforts where basic control strategies are implemented to regulate the internal temperature of the chiller based on the speed of the motor of the compressor control, such strategies are tuned considering the parameters of the linearized model [21][22]. A similar analysis is employed to define the conditions of the drivers hysteresis, [23][24]. A different approach found in literature, with respect to the modeling of the dynamics of the vapor compressor system implemented using heuristic algorithms, the most typical tool in Neural Networks [25], fuzzy logic, [26] and genetic algorithms [27]. The disadvantage of these heuristic models is that their internal variables do not have any physical interpretation, although very well solve the problem of the internal temperature regulation whereas the non-linearity which affects the performance of the model linearized described above.

IV. DYNAMIC MODEL OF THE VCR SYSTEM

A. Energy Conservation Law

Because the refrigeration cycle of a reciprocating refrigerating system is a closed cycle, if the system is operated in continuous and steady state (i.e., in an equilibrium state), according to the principle of continuity of mass and energy balance, the mass flow rates of refrigerant flowing through the evaporator, compressor, condenser and expansion or float valve must all be equal. Also, the total amount of energy supplied to the refrigeration system must be approximately equal to the total energy rejected from the system.

A continuous and steady state means that the flow is continuous, and the properties of the refrigerant at any point in the refrigeration system do not vary over time. Therefore, during the design of a refrigeration system, the system components selected should have equal or approximately equal mass flow rates of refrigerant at stable conditions [17][28].

A balance of energy (first law of thermodynamics), combined with mass conservation in the control volume provides the following equation;

$$\dot{Q} + \dot{W} = \dot{m}\left[\left(h_o - h_i\right) + \frac{v_o^2 - v_i^2}{2} + g\left(z_o - z_i\right)\right] \quad (2)$$

The equation (2) $\dot{Q}$ is transferred to the system by the surrounding heat flow, $\dot{W}$ is the work power performed by the electric motor of the compressor, h is enthalpy function that is associated with the sum of the internal energy and the work flow, $u + Pv$, the linear kinetic energy $v^2/2$ and potential energy gravity gz of the fluid stream. The subscripts i and o represent the conditions of entry and exit, respectively. In the absence of appreciable variations of kinetic and potential energy the equation above reduces to:

$$\dot{Q} + \dot{W} + \dot{Q}_{Pi} = \dot{m}_r c_p \left(T_o - T_i\right) \quad (3)$$

This relationship is grounded on the consideration that all variables distributions along in the finite volume (control volume) are homogeneous, and it is going to be applied for each one of the control volume from the four stage of the refrigeration cycle. In this equation, $\dot{m}_r$ is the mass of refrigerant in circulation, c_{pi} is the specific heat of the refrigerant at room temperature, which is considered constant, (Table 1). Q_{Pi} represents the heat loss or heat generated by the system and corresponds to the Newton's law of cooling, dT/dt is the speed with which the object cools for this analysis represents the change in temperature of the refrigerant, [29].

$$Q_{Pi} = C_{Ti}\frac{dT_i}{dt} + \frac{\left(T_i - T_a\right)}{R_{Ti}} \quad (4)$$

In the volume control 1, C_{Ti} is the thermal capacitance to the interior of the space confined to the evaporator element and for other elements is the capacitance of the material, which can be also expressed $C_{Ti} = mc_p$, thus, m is the mass of the elements and $c_{p,m}$ is considered the heat capacity of the material with which is manufactured elements, T_i is the temperature inside the system, and T_a is the ambient temperature. This model enthalpy is considered like ideal gas;

$$\Delta h = c_p (T_o - T_i) \quad (5)$$

Figure 3 shows the physical quantities involving to the system analyzed.

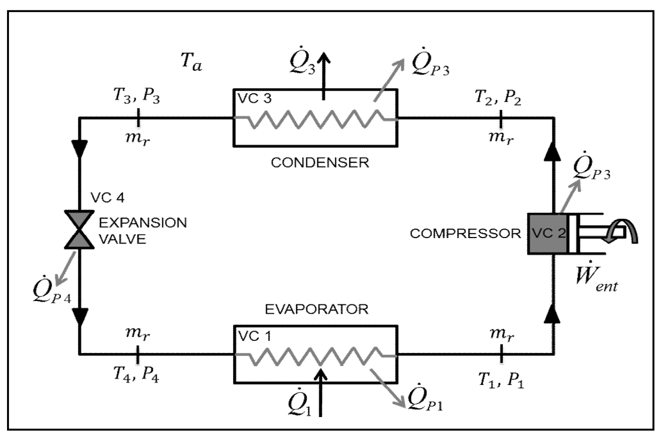

Figure 3. The Dynamic Model Analyzed Scheme.

The following simplifications are considered: (i) the physical properties related to the refrigerant is considered uniform in the heat exchanger transversal section, (ii) the refrigerant liquid and vapor phases are in thermodynamic equilibrium, (iii) the heat exchangers have a perfect thermal insulation, (iv) the axial heat conduction in the pipes is not taken into account.

B. Evaporator Mathematical Model

To obtain this model $\dot{Q} = 0$ as the heat removed by the refrigerant from the confined space which is also known as thermal load, the work $\dot{W} = 0$, equation 3 and 4, arises:

$$\dot{Q}_1 + \dot{Q}_{Pi} = \dot{m}_r c_p (T_1 - T_4) \tag{6}$$

Substituting into (4), Newton's law of cooling and ordering terms is obtained;

$$\frac{dT_{R1}}{dt} = T_1\left(\frac{c_p \dot{m}_r}{C_{T1}} - \frac{1}{C_{T1}R_{T1}}\right) - T_4 \frac{c_p \dot{m}_r}{C_{T1}} + \frac{T_a}{C_{T1}R_{T1}} - \frac{\dot{Q}_1}{C_{T1}} \tag{7}$$

C. Mathematical Model of Compressor

Motor shaft dynamics are modeled from an angular momentum balance between the driving and braking torques. The torque-speed characteristics of the motor itself are obtained from manufacturer´s data. From these, the driving torque and speed to the compressor are known.

For this model, the following hypothesis is established; there is no friction on the compressor, is an adiabatic process $\dot{Q}_1 = 0$. The other variables set for analysis.

$$\dot{W}_C = \frac{\mathrm{n}}{(n-1)} \eta_p V_c \omega_c p_1 \left[1 - \left(\frac{p_2}{p_3}\right)^{\frac{n-1}{n}}\right] \tag{8}$$

The torque provided by the motor of the compressor $\dot{W} = \dot{T}_R$ and ω is the angular velocity of the shaft of the electric motor of the compressor. Equation (3) and (4), arises;

$$\dot{W}_R + \dot{Q}_{P2} = \dot{m}_r c_p (T_2 - T_1) \tag{9}$$

Substituting into (4) to (8) and ordering terms, gets;

$$\frac{dT_2}{dt} = -T_1 \frac{c_p \dot{m}_r}{C_{T2}} + T_2\left(\frac{c_p \dot{m}_r}{C_{T2}} - \frac{1}{C_{T2}R_{T2}}\right) + \frac{T_a}{C_{T2}R_{T2}} - \frac{\dot{W}_1}{C_{T2}} \tag{10}$$

D. Mathematical Model of Condenser

In this element, only the superheat vapor is considered and the vapor phase is considered to be in the thermal equilibrium and moving at the same velocity. Work is not performed in this element $\dot{W} = 0$, $\dot{Q} = 0$ it is the heat released into the environment. It arises from (3) and (4);

$$\dot{Q}_3 + \dot{Q}_{P3} = \dot{m}_r c_p (T_3 - T_2) \tag{11}$$

Substituting (4) into (10) and ordering terms, gets;

$$\frac{dT_3}{dt} = -T_2\left(\frac{c_p \dot{m}_r}{C_{T3}}\right) + T_3\left(\frac{c_p \dot{m}_r}{C_{T3}} - \frac{1}{C_{T3}R_{T3}}\right) + \frac{T_a}{C_{T3}R_{T3}} - \frac{\dot{Q}_3}{C_{T3}} \tag{12}$$

E. Mathematical Model of the Expansion Valve

In this device there is no interaction of work and heat $\dot{W} = 0$ and $\dot{Q} = 0$, the expansion is isenthalpic. Then from (3) and (4), arises;

$$\dot{Q}_{P4} = \dot{m}_r c_p (T_3 - T_4) \tag{13}$$

Substituting (4) into (12) and ordering terms;

$$\frac{dT_4}{dt} = -T_3\left(\frac{c_p \dot{m}_r}{C_{T4}}\right) + T_4\left(\frac{c_p \dot{m}_r}{C_{T4}} - \frac{1}{C_{T4}R_{T4}}\right) + \frac{T_a}{C_{T4}R_{T4}} \tag{14}$$

F. State-space Representation of VCR System

In previous sections equations (7), (10), (12) and (14) has been presented as first-order linear ordinary differential equations, with the temperature of the refrigerant as the system output.

Whereas the following expressions in the matrix form, [30]:

$$\begin{aligned} \dot{T}(t) &= Ax(t) + Bu(t) \\ y(t) &= Cx(t) + Du(t) \end{aligned} \tag{15}$$

Equation (7), (10), (12) and (14) shall be replaced in (15):

$$\begin{bmatrix} \dot{T}_1 \\ \dot{T}_2 \\ \dot{T}_3 \\ \dot{T}_4 \end{bmatrix} = \begin{bmatrix} \left(\frac{c_p \dot{m}_r}{C_{T1}} - \frac{1}{C_{T1}R_{T1}}\right) & 0 & 0 & -\frac{c_p \dot{m}_r}{C_{T1}} \\ -\left(\frac{c_p \dot{m}_r}{C_{T2}}\right) & \left(\frac{c_p \dot{m}_r}{C_{T2}} - \frac{1}{C_{T2}R_{T2}}\right) & 0 & 0 \\ 0 & -\left(\frac{c_p \dot{m}_r}{C_{T3}}\right) & \left(\frac{c_p \dot{m}_r}{C_{T3}} - \frac{1}{C_{T3}R_{T3}}\right) & 0 \\ 0 & 0 & -\frac{c_p \dot{m}_r}{C_{T4}} & \left(\frac{c_p \dot{m}_r}{C_{T4}} - \frac{1}{C_{T4}R_{T4}}\right) \end{bmatrix} \begin{bmatrix} T_1 \\ T_2 \\ T_3 \\ T_4 \end{bmatrix}$$

$$+ \begin{bmatrix} \frac{1}{C_{T1}R_{T1}} & -\frac{1}{C_{T1}} & 0 & 0 \\ \frac{1}{C_{T2}R_{T2}} & 0 & -\frac{1}{C_{T2}} & 0 \\ \frac{1}{C_{T3}R_{T3}} & 0 & 0 & -\frac{1}{C_{T3}} \\ \frac{1}{C_{T4}R_{T4}} & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} T_a \\ \dot{Q}_1 \\ \dot{T}_R \\ \dot{Q}_3 \end{bmatrix}$$

$$y(t) = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} T_1 \\ T_2 \\ T_3 \\ T_4 \end{bmatrix} + 0$$

V. VALIDATION OF THE MODELING

The mathematical model obtained in the previous section for the prediction of dynamic behavior was coded in Matlab Simulink. The typical values of input parameters have been presented in Table 1 the input parameters include refrigerant

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type, ambient temperature and the initial temperature of the compartment, etc. The values of c_{pi} of the refrigerant were defined according the average of the phase in regard on p vs h diagram.

Considering the thermodynamic cycle starts-up when the system is powered by the angular speed of the electric motor of the compressor, the flow of the refrigerant quickly tends to a permanent state, it can be noted that the refrigerant tends to decrease its temperature at the evaporator element, trying to keep the relationship of equilibrium of Pressure and Temperature [31].

TABLE I. VALUES OF MATLAB ALGORITHM OF SIMULIK PROGRAM

C_{p1}=1330	J/kg-K	C_{T1}=4500	J/K	R_{T1}=0.090	K/W
C_{p2}=1400	J/kg-K	C_{T2}=2500	J/K	R_{T2}=0.025	K/W
C_{p3}=1138	J/kg-K	C_{T3}=1250	J/K	R_{T3}=0.048	K/W
C_{p4}=1318	J/kg-K	C_{T4}=500	J/K	R_{T4}=3.20	K/W
Q_1=195	W	T_a=298	K	m_r=0.000035	kg/s
Q_3=-200	W	T_R=5	W		

The following figures show the dynamic evolution of the refrigeration cycle of two important variables: temperature and pressure starting from ambient temperature; then, it approaches the steady state.

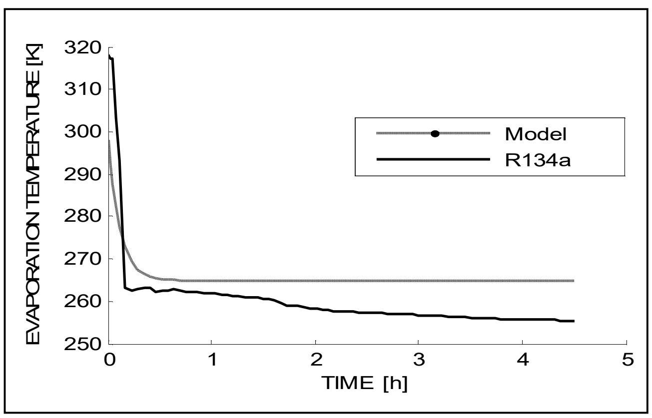

Figure 4. Behavor of the temperature of the evaporation system.

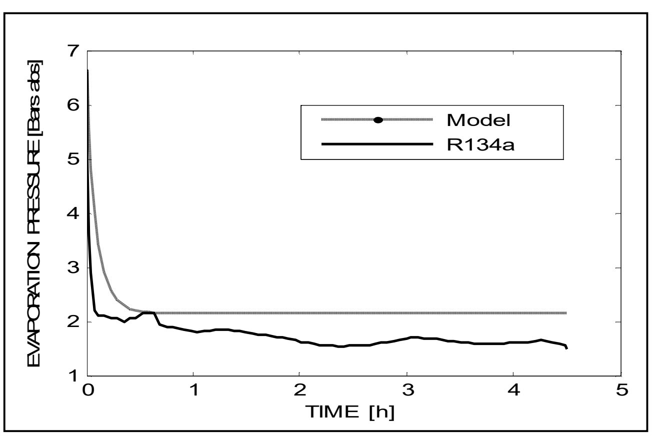

Figure 5. Transient temperature profile from development model.

Figure 4 shows the evolution of temperatures behaves in the compartment of the refrigerator, the model is compared with Embraco Data [31], using R-143a, it can be observed takes about 10 minutes in the transient state, then the room temperature reaches the steady state operation.

Figure 5 shows the evolution the behave pressure of the refrigerant in the evaporator when it flow thought the length tube, we can appreciates the same approach considering it begin from external temperature. Figure 5 shows the results when the thermodynamic cycle has been developed on the diagram pressure versus enthalpy, explained in section II.

A reasonable agreement between model and measure values from Embraco Data [31] was observed for the whole start-up period, during the first 5 minutes, the pressure below from 6.5 bars to 2.2 bars, and then it tends to decrease slowly until to reach steady state operation.

VI. CONCLUSION

In the present work, a methodology has been developed to model transient behavior of the refrigerator. This would act as foundation to transient model of domestic refrigerator, using lumped transient model. This methodology, based on lumped parameters model will reduce the overall cycle time in predicting the transient compartment temperature and will also decrease the experimental effort.

The results presented in this paper agree qualitatively with temperature time evolution shown other papers. Calculation for transient analysis of domestic refrigerators, obtained with the computer program here presented, will further validate with experimental data in near future.

Therefore, it is considered that it is a good starting point for the study of cooling systems applying state variables. The first strategy will be the vector by the feedback of states since becoming a linear approximation; however, you can see that there are nonlinearities, so it is contemplated in the near future to implement an algorithm control parameters identification and adaptive control, which will be validated with experimentation in the facilities of LaNITeF.

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