# Dynamic Behaviour of a Vapor Compression Refrigerator: A Theoretical and Experimental Analysis

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# DYNAMIC BEHAVIOUR OF A VAPOR COMPRESSION REFRIGERATOR:

# A THEORETICAL AND EXPERIMENTAL ANALYSIS

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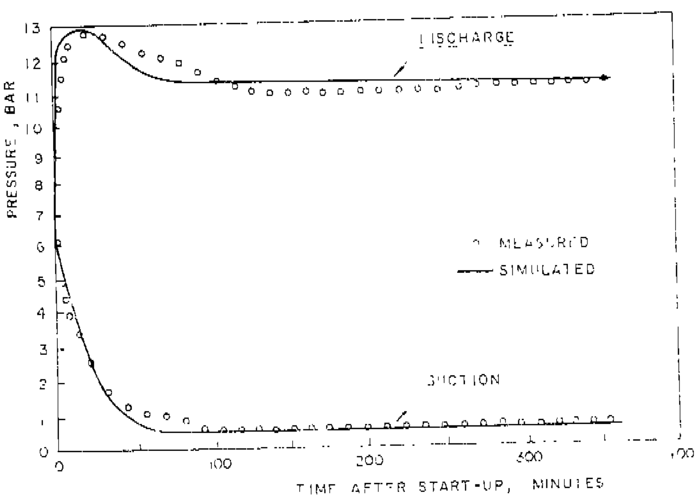
Abstract

This work presents the mathematical model employed to simulate the dynamic behaviour of a domestic refrigerator. The component models include a hermetic reciprocating compressor, a condenser, an evaporator, an accumulator and a capillary tube.

All the components were modeled, and basic equations for them were derived from the physical laws of mass and energy conservation. The simultaneous solution of all the differential and algebraic equations allows the transient analysis of the refrigeration system.

The experimental setup employed to validate the computer predictions is also presented. Results of the simulation were compared with experimental results (see figure below), and the validity of the model was confirmed.




Refrigerant pressures after start-up

COMPORTEMENT DYNAMIQUE D'UN REFRIGERATEUR A COMPRESSION DE VAPEUR. ANALYSE THEORIQUE ET EXPERIMENTALE.

RESUME : Cette communication présente un modèle mathématique pour simuler le comportement dynamique d'un réfrigérateur ménager. Les modèles des composants comprennent un compresseur hermétique à piston, un condenseur, un évaporateur, un accumulateur et un tube capillaire.

Tous les composants ont été modélisés et des équations fondamentales se rapportant à eux découlent des lois physiques de la conservation de la masse et de l'énergie. La résolution simultanée de l'ensemble des équations différentielles et algébriques permet l'analyse en régime transitoire du système frigorifique.

Le montage expérimental employé pour valider les prévisions de l'ordinateur est également présenté. Les résultats de la simulation ont été comparés aux résultats expérimentaux et la validité du modèle a été confirmée.

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## NOMENCLATURE

| | |
|---|---|
| A | area, $m^2$ |
| C | thermal capacity, kJ/K |
| D | capillary tube diameter, m |
| f | friction factor |
| h | specific enthalpy, kJ/kg |
| L | capillary tube length, m |
| M | mass, kg |
| $\dot{m}$ | mass flow rate of refrigerant, kg/s |
| p | pressure, Pa |
| Q | heat transfer rate, kW |
| R | thermal resistance, K/kW |
| Re | Reynolds number |
| S | quality of the refrigerant at the capillary tube inlet |
| T | temperature, °C |
| t | time, s |
| V | velocity, m/s |
| $\rho$ | density, $kg/m^3$ |

Subscripts

| | |
|---|---|
| a | accumulator |
| b | suction |
| c | condenser |
| d | discharge |
| e | evaporator |
| f | freezer |
| g | cabinet |
| $\ell$ | liquid refrigerant |
| m | supply air |
| o | outside |
| p | refrigerant leaving compressor |
| r | refrigerant |
| s | superheated |
| t | refrigerant leaving condenser |
| v | vapor refrigerant |
| w | wall |
| y | refrigerant leaving capillary tube |

## 1. INTRODUCTION

A common procedure in most of the compressor and/or refrigerator manufacturing companies is to perform experimental tests in order to ascertain the refrigerator performance as a function of time. These tests are performed according to reference |1| and normally require a long period of time. One way to speed up such procedure is to employ computational techniques to numerically simulate the refrigerator performance.

Since the dynamic modelling of the refrigerator requires each element of the refrigeration system to be considered in depth, the model allows not only the simulation of the experimental tests but also helps to develop a deep understanding of the fundamental processes occuring.

Most of the models, published in the open literature, are directed towards heat pump simulation and, only very few of them |2,3,4|, are experimentally validated. The particular model, now considered, is limited to the reciprocating compressor, forced-draft condenser, accumulator, capillary tube and forced-draft evaporator system. This model combines many of the features presented in earlier models |2-8| while introducing several new features. The space being cooled, for example, is a top-mount domestic refrigerator, which makes, both the theoretical and experimental analysis, more complex.

## 2. MODELING OF SYSTEM COMPONENTS

Each of the refrigeration system components is modeled by one or more control volumes. Thus, three control volumes are employed for the condenser, in order to accomodate the region of superheated vapor, mixture of saturated vapor and saturated liquid, and subcooled liquid. The evaporator model employs two control volumes, one for the liquid region and the other for the vapor region. The compressor and the compressor shell are modeled as two separate control volumes. The accumulator is modeled by two control volumes, one for the saturated condition and the other for the superheated condition. The capillary tube model contains three control volumes, in order to take into account the following situations: capillary tube containing only superheated vapor, containing only saturated vapor and containing only subcooled liquid.

Conservation equations for mass and energy are written, in differential form, for each control volume. The combination of these equations with refrigerant property relations and with some empirical parameters yields a set of algebraic and differential equations that, when properly solved, allows the transient analysis of the refrigeration system.

Due to space limitations only the following control volumes are presented in this paper: condenser (superheated flow region), capillary tube and space being cooled.

### 2.1. Condenser

The purpose of the condenser in a refrigeration system is to remove heat from superheated vapor discharged from the compressor. Depending upon the rate of heat transfer between the refrigerant and the condenser wall, the condenser contains superheated vapor, a mixture of saturated vapor and saturated liquid, and subcooled liquid.

The condenser model, being used in this work, employs, as previously mentioned, three control volumes. Thus, at the begining of the operation cycle, only the superheated vapor control volume is employed. As soon as the condensation process starts the model neglects the superheated flow region and employs only the saturated vapor control volume. The remaining control volume is employed when some subcooled liquid is formed in the condenser.

The assumption of disregarding the influence of the superheated flow region, when some liquid is formed in the condenser, is due to the fact that this region occupies only 5-10% of the total coil volume |7|. Obviously this assumption is not valid for natural-draft condensers.

Following an approach similar to Dhar |5|, the control volume is treated as a stirred tank, in which the conditions existing at the outlet of the tank are the same as the bulk conditions within the tank. Figures 1 and 2 show, respectively, a sketch and the heat flow diagram of the superheated vapor model.

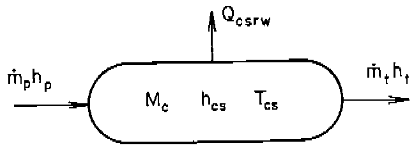


Fig. 1 - Superheated vapor model

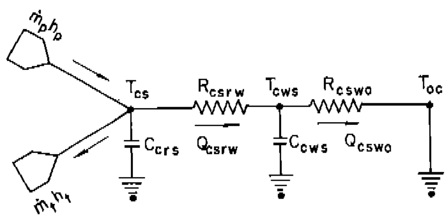


Fig. 2 - Heat flow diagram for the superheated flow model

Applying the continuity equation to the control volume gives

$$d(M_c)/dt = \dot{m}_p - \dot{m}_t \tag{1}$$

Applying the energy equation to the node points of Figure 2, yields

$$d(M_c\ h_{cs})/dt = \dot{m}_p h_p - \dot{m}_t\ h_{cs} - Q_{csrw} \tag{2}$$

and

$$d(C_{cws}\ T_{cws})/dt = Q_{csrw} - Q_{cswo} \tag{3}$$

In equation (2) the refrigerant internal energy was replaced by the refrigerant enthalpy. In doing so, the pressure variation with time was disregarded. This assumption, however, has only a minor effect on the simulation results |6|.

Combining equations (1) and (2), gives:

$$d(h_{cs})/dt = \left[\dot{m}_p\ (h_p - h_{cs}) - Q_{csrw}\right]/M_c \tag{4}$$

Numerical integration of equations (1) and (4) gives the mass of refrigerant in the condenser, $M_c$ , and the bulk enthalpy of refrigerant vapor, $h_{cs}$, respectively.

The density of the refrigerant vapor is found knowing the mass of vapor and the condenser volume. The temperature and the pressure in the condenser is calculated from property relations. The standard Dittus-Boelter equation is used to calculated the heat transfer coefficient on the inside of the tubes. The heat transfer from the outside of the tubes to the air is obtained from reference |9|.

## 2.2. Capillary tube

The main task of the capillary tube expansion device is to maintain the minimum pressure in the condenser at which all the refrigerant can condensate.

The capillary tube, although physically simple, is behaviourly complex. Some items of complex mathematical analysis are: the friction factor for the two-phase flow, the thermal contact with the suction line, the metastable flow, the oil circulation with the refrigerant, etc. Many researchers have investigated the flow through a capillary tube and a review on the subject can be found in reference |10|.

A capillary tube model taking into account all the factors that influence the flow rate of refrigerant along the tube, would require an enormous CPU time. Therefore, it was decided to develop a first-stage model, based on the following assumptions: i) the capillary tube is a straight, horizontal, constant inner diameter tube, ii) flow in the capillary is one-dimensional, homogeneous, and adiabatic, iii) the refrigerant is free of oil, and iv) the choked and metastable flow phenomenas are neglected.

Applying the momentum equation for an element of fluid yields,

$$\rho A^2 \, dp/\dot{m}_y^{\,2} + dV/V + 2f \, dx/D = 0 \tag{5}$$

Assuming an average refrigerant density and an average friction factor along the tube, the integration of equation (5), gives,

$$\dot{m}_y = A \, \{\bar{\rho}(P_d - P_b)/(\ln(\rho_d / \rho_b) + 2 \, \bar{f} \, L/D)\}^{1/2} \tag{6}$$

Two relationships are employed to evaluate the friction factor. Equation (7), derived from Moody's chart for a relative roughness of 0.0006, is used for the single phase flow.

$$\bar{f} = \{\exp(0.75 - 0.68 \ln \overline{Re} + 0.024(\ln \overline{Re})^2)\}/4 \tag{7}$$

Erth's equation |11| is employed for the two-phase flow

$$\bar{f} = \{ 0.775 \exp[(1 - S^{0.25})/2.4] \, /(Re_\ell)^{1/2}\} \tag{8}$$

## 2.3. Space being cooled

The space being cooled is a 0.42 m³(14.8 ft³) domestic top mount refrigerator, whose sketch and heat flow diagram are indicated in Figures 3 and 4, respectively.

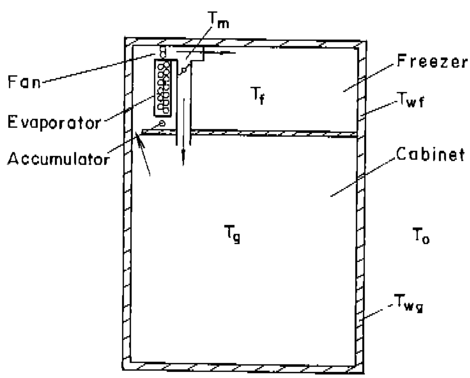


Fig. 3 - Top-mount refrigerator

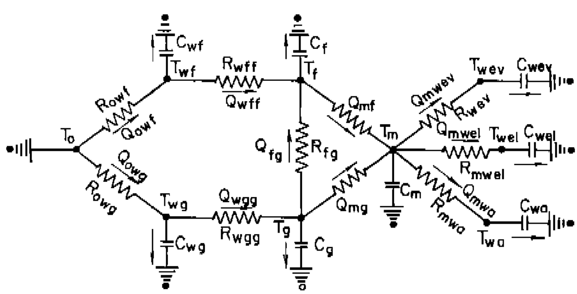


Fig. 4 - Heat flow diagram for the top-mount refrigerator

Applying the energy equation to the node points of Figure 4, gives

$$d(T_{wf})/dt = \{Q_{owf} - Q_{wff}\}/C_{wf} \tag{9}$$

$$d(T_f)/dt = \{Q_{wff} + Q_{fg} - Q_{mf}\}/C_f \tag{10}$$

$$d(T_m)/dt = \{Q_{mf} + Q_{mg} - Q_{mwev} - Q_{mwe\ell} - Q_{mwa}\}/C_m \tag{11}$$

$$d(T_g)/dt = \{Q_{wgg} - Q_{fg} - Q_{mg}\}/C_g \tag{12}$$

$$d(T_{wg})/dt = \{Q_{owg} - Q_{wgg}\}/C_{wg} \tag{13}$$

Numerical integration of equations (9-13) gives the bulk temperature of the points indicated in Figure 3.

## 3. COMPARISON BETWEEN LABORATORY VERSUS COMPUTER PREDICTED RESULTS

In order to validate the model, an experimental test was performed following the recomendations of reference |1|. Since one of the assumptions of the capillary tube model is to consider the flow as adiabatic, it was necessary, before the beginning of the test, to insulate the refrigerator capillary tube. After that, the refrigerator was placed in an environmental test chamber, in which the air temperature is maintained in a constant value of 43°C (109.4 °F).

The temperature and pressure in several locations of the refrigerator, as indicated in Figure 5, are measured respectively by copper constantan thermocouples and strain gage pressure transducers. Both the thermocouples and the pressure transducers are connected to a data acquisition system which registers all variables within a time interval of two minutes. The pressures, are also continuously registered by a two-channel recorder.

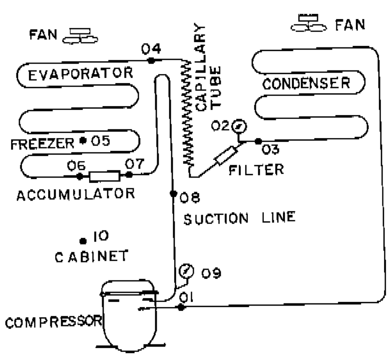


Fig. 5 - Refrigeration system under analysis

The results of such test, when compared with the computational results are indicated in Figures 6 and 7. Figure 6 shows the pressure behaviour with time. The results are shown only for the six first minutes since the pressure variation, after this time, is negligible. As one can see, Figure 6 indicates a good agreement between laboratory versus model data. Figure 7 shows the freezer and cabinet temperatures versus time. As indicated, the steady state value of these variables is reached after ten hours of operation. The agreement between the experimental and computational results is also shown to be good.

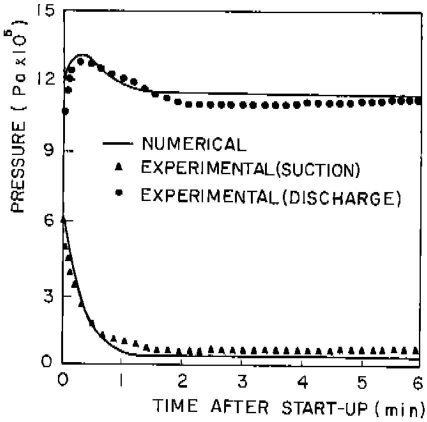


Fig. 6 - Refrigerant pressure versus time

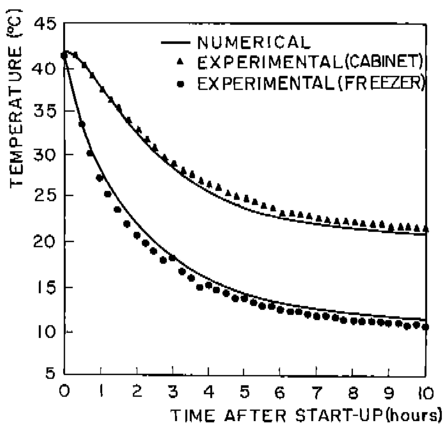


Fig. 7 - Freezer and cabinet temperature versus time

The temperatures, indicated in Figure 7, are higher than would be expected. This is partly due to the absence of any heat exchange in the capillary tube and partly due to the low charge of refrigerant employed in the experimental test (250g(0.55 ℓb)).

## 4. CONCLUSIONS

A dynamic model of a top-mount domestic refrigerator has been developed and validated. To the best of the knowledge of the authors, this is the first work on this subject published in the open literature.

The CPU time, required by an IBM 4341, for simulating ten hours of operation, reaches the value of 3.75 hours. This data, when compared with the normal experimental time of 24 hours, shows one of the advantages of the computational approach. Besides that, in contrast to the experimental approach, any alteration in the refrigeration system can be accomodated quite easily, in the program, by simple alteration of the input data.

The agreement between the model and laboratory data has been shown to be quite good even with the simplifying assumptions used in the model.

In the next stage of this work, most of the assumptions will be, progressively, removed and a comparison will be made between the CPU time required and the improvements on the simulation results.

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