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Abstract

This article presents the mathematical model used for transient analysis of domestic refrigerators. It is assumed that the refrigerator cabinet thermal loads control the transient process and that the heat pump works under a quasi-steady process. The results obtained with this approach shows temperature time evolution in agreement with more complex and more time consuming mathematical methods.

INTRODUCTION

This paper describes the transient analysis of domestic refrigerators. It is assumed that in this analysis the transient process is controlled by the thermal loads inside the refrigerator cabinet and that the heat pump works in a sequence of quasi-steady states. The objective of this work is to evaluate the effects of different refrigerants in identical hardware and hardware modifications changing the refrigerator thermal performance. One could, for example, analyze the time to cool/freeze food (useful thermal load), the effect of door opening causing temperature fluctuations, variations caused by on/off cyclic operation, energy consumption reduction due to thickness of insulation, energy consumption reduction due to condenser and evaporator (UA) increase, pull-down test simulation, etc. Some of these design options are discussed in Turiel & Heydari (1988), Turiel & Levine (1989) and in Abramson *et al* (1990). The detailed simulation of the transient behavior of refrigerant and components are not investigated in the present work. Here, a simplified model is used in order to reduce the necessary computing time required for transient analysis.

The steady-state heat pump analysis is discussed in a companion paper by Lemos and Zaparoli (1996). The approach therein is to mathematically model each component (*compressor, capillary-tube, condenser and evaporator*) and combine these individual models. This computer program evaluates refrigerant thermodynamic and transport properties needed in the detailed models describing each component of the heat pump.

The refrigerator transient behavior was calculated by a lumped approach. In this method, an equivalent thermal circuit links the thermal loads inside the refrigerator, cabinet walls and the evaporator. This approach was used by Davis *et al* (1972) to mathematically express the coupling of thermal nodes in different regions of passenger cars and to evaluate the performance of automotive air conditioning systems. Melo *et al* (1987) presents an analogous model used in numerical simulation of the transient behavior of domestic refrigerators.

The results obtained in this work shows a time evolution in agreement with results of mathematically more complex and more time consuming methods. The approach here described is a good option to simulate transients due to thermal loads inside the freezing and cooling compartments, and to economically evaluate performance parameters of domestic refrigerators. All computations shown below were run on a personal computer.

REFRIGERATOR TRANSIENT MODEL

The approach used to calculate the cabinet thermal load behavior is a lumped parameter model and an equivalent thermal circuit coupling the several nodes is shown in Figure 1.

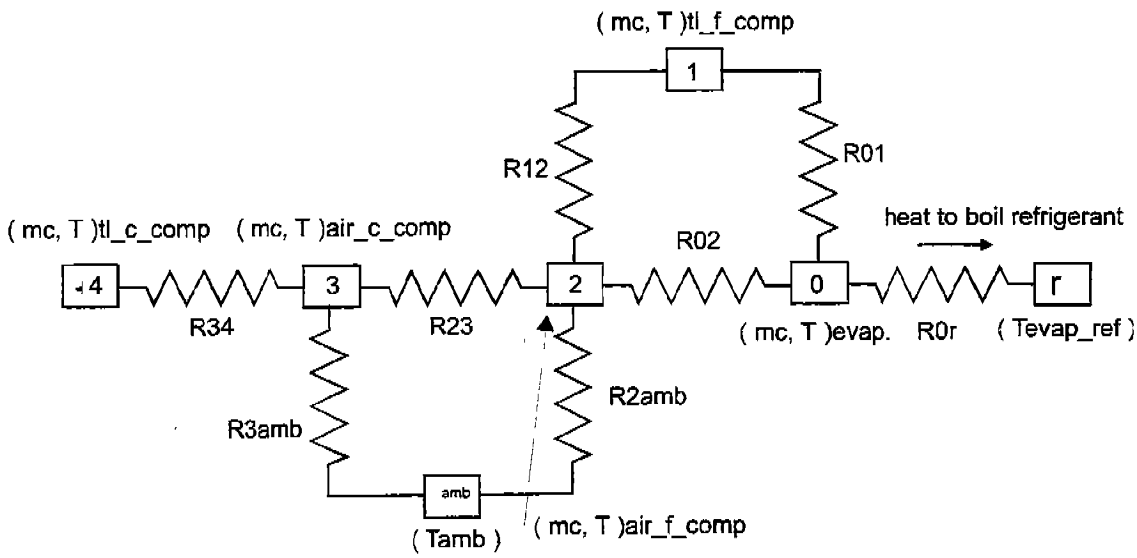

Figure 1 - Equivalent thermal circuit linking cabinet nodes. Node **r**: refrigerant; node **0**: evaporator plate; node **1**: freezing compartment thermal load; node **2**: air inside freezing compartment; node **3**: air inside cooling compartment; node **4**: cooling compartment thermal load; node **amb**: external ambient.

It was assumed that the thermal resistance between the refrigerant and the evaporator was equal to zero due to the comparatively high boiling heat transfer coefficient inside the tube. This fact implies that the refrigerant evaporation temperature is equal to the evaporator wall temperature which is calculated by the method described bellow.

Applying the energy balance in each node of the thermal circuit and considering the lumped formulation results in the following first-order ordinary differential equations system:

Evaporator plate:

$$(mc)_{evap}\frac{dT_{evap}}{dt} = -\dot{Q}_{evap} - \frac{1}{R01}(T_{evap} - T_{tl\_f\_comp}) - \frac{1}{R02}(T_{evap} - T_{air\_f\_comp}) \qquad (1)$$

Thermal load in the freezing compartment:

$$(mc)_{tl\_f\_comp}\frac{dT_{tl\_f\_comp}}{dt} = -\frac{1}{R01}(T_{tl\_f\_comp} - T_{evap}) - \frac{1}{R12}(T_{tl\_f\_comp} - T_{air\_f\_comp}) \qquad (2)$$

Air in the freezing compartment:

$$(mc)_{air\_f\_comp}\frac{dT_{air\_f\_comp}}{dt} = -\frac{1}{R02}(T_{air\_f\_comp} - T_{evap}) - \frac{1}{R23}(T_{air\_f\_comp} - T_{air\_c\_comp}) - \frac{1}{R12}(T_{air\_f\_comp} - T_{tl\_f\_comp}) - \frac{1}{R2amb}(T_{air\_f\_comp} - T_{amb}) \qquad (3)$$

Air in the cooling compartment:

$$(mc)_{air\_c\_comp}\frac{dT_{air\_c\_comp}}{dt}=-\frac{1}{R23}(T_{air\_c\_comp}-T_{air\_f\_comp})-\frac{1}{R34}(T_{air\_c\_comp}-T_{tl\_c\_comp})-\frac{1}{R3amb}(T_{air\_c\_comp}-T_{amb}) \tag{4}$$

Thermal load in the cooling compartment:

$$(mc)_{tl\_c\_comp}\frac{dT_{tl\_c\_comp}}{dt}=-\frac{1}{R34}(T_{tl\_c\_comp}-T_{air\_c\_comp}) \tag{5}$$

where:

$\dot{Q}_{evap}$ = heat transfer rate to refrigerant in the evaporator,

Rmn = thermal resistance between node m and n of thermal circuit,

T = temperature

(mc) = mass specific heat product,

$_{evap}$ = evaporator plate,

$_{tl\_f\_comp}$ = thermal load in the freezing compartment,

$_{tl\_c\_comp}$ = thermal load in the cooling compartment,

$_{air\_f\_comp}$ = air in the freezing compartment,

$_{air\_c\_comp}$ = air in the cooling compartment,

$_{amb}$ = external ambient.

The heat transfer between the thermal circuit nodes occurs by natural convection resulting in low coefficients of heat transfer. In many applications, these low heat transfer coefficients imply in low Biot numbers. Due to this fact, as discussed in Ozisik (1985), the lumped parameter approach is applicable, the temperature distribution during transients within the solids at any instant is uniform and that simplifies the calculation of the variation of temperature with time

Due to the combination of accuracy, relative low cost, and ease of programming, a fourth-order Runge-Kutta subroutine, given in Ferziger (1981), was used to solve the above system of five ordinary differential equations.

THERMAL RESISTANCES EVALUATION

In the cabinet analysis it was assumed that the rate of change in the boundary conditions for the free convection problems was low enough that these cases could be considered as quasi-steady processes. That is, the boundary layer equations without the time derivative would be an accurate approximation describing the problem at any instant of time. (Eckert and Drake, 1972). If the above condition is true, the Nusselt number correlations for steady-state free convection can then be used to evaluate the heat transfer coefficients.

The following thermal resistance equations were obtained assuming unidimensional, steady-state heat transfer.

1. **R0r** - thermal resistance between evaporator plate and refrigerant inside evaporator tube - this resistance was set to zero because of the relatively high boiling heat transfer coefficient.
2. **R01** - thermal resistance between freezing compartment thermal load and evaporator plate - there are two resistances in series: a contact resistance and an ice slab resistance

$$R01=\frac{1}{h_{contact}*A_{contact}}+\frac{L_{ice}}{k_{ice}*A_{contact}} \tag{6}$$

where:

$A_{contact}$ = thermal contact area,

$h_{contact}$ = heat transfer coefficient in thermal contact resistance

L_{ice} = ice slab width,

k_{ice} = thermal conductivity of ice slab.

3. **R02** - thermal resistance between evaporator plate and air in the freezing compartment

$$R02 = \frac{1}{h_{evap} * A_{evap}} \quad (7)$$

where:

h_{evap} = weighted area mean free convection heat transfer coefficient (free convection in vertical and horizontal plates),

A_{evap} = evaporator plate heat transfer area.

4. **R12** = thermal resistance between air in the freezing compartment and the thermal load in the freezing compartment

$$R12 = \frac{1}{h_{tl\_f\_comp} * A_{tl\_f\_comp}} \quad (8)$$

where:

$h_{tl\_f\_comp}$ = free convection heat transfer coefficient,

$A_{tl\_f\_comp}$ = heat transfer area.

5. **R23** = thermal resistance between air in the freezing compartment and air in the cooling compartment

$$R23 = R02 * (BF) \quad (9)$$

where:

BF = blockage factor ($0 \leq BF \leq 1$).

6. **R2amb** = thermal resistance between air in the freezing compartment and external ambient - there are three resistance's in series: internal free convection, cabinet wall insulation, and external free convection

$$R2amb = \frac{1}{h_{f\_comp\_int} * A_{f\_comp}} + \frac{L_{ins}}{k_{ins} * A_{f\_comp}} + \frac{1}{(h_{f\_comp\_ext} + h_r) * A_{f\_comp}} \quad (10)$$

where:

$h_{f\_comp\_int}$ = weighted area mean internal free convection heat transfer coefficient (free convection in vertical and horizontal plates),

$h_{f\_comp\_ext}$ = weighted area mean external free convection heat transfer coefficient (free convection in vertical and horizontal plates),

$h_r = \sigma\varepsilon(T_{w\_ext}^2 + T_{amb}^2)(T_{w\_ext} + T_{amb})$ = radiation heat transfer coefficient,

T_{w_ext} = external wall temperature,

A_{f_comp} = freezing compartment cabinet wall heat transfer area,

L_{ins} = insulation cabinet wall width,

k_{ins} = insulation cabinet wall thermal conductivity.

The computer subroutine which calculates this thermal resistance iterate several times because of the coupling between internal and external cabinet wall temperatures and the corresponding heat transfer coefficients.

7. **R3amb** = thermal resistance between air in the cooling compartment and external ambient - the same procedure used for **R2amb** was here adopted, except for the temperature of the air inside the cabinet cooling compartment.

8. **R34** = thermal resistance between air in the cooling compartment and thermal load in the cooling compartment

$$R34 = \frac{1}{h_{tl\_c\_comp} * A_{tl\_c\_comp}} \qquad (11)$$

where:

$h_{tl\_c\_comp}$ = free convection heat transfer coefficient,

$A_{tl\_c\_comp}$ = heat transfer area.

The heat transfer coefficients in the equations presented above are calculated with Nusselt number correlations recommend in Kakaç *et al* (1986). The thermal resistance values are updated in every time interval in the integration of the ordinary differential equations system.

RESULTS AND DISCUSSION

The results of the refrigerator cabinet transient, after the insertion of thermal loads in the freezing and cooling compartments (*useful thermal load and air*), are shown in Figure 2. At time equal to 160s an on/off cyclic operation is simulated. The evaporator and condenser temperatures increase quickly first because of the large amount of heat available in the thermal loads. After this first time interval, the evaporator temperature decreases and, at the same time, there is a decrease in the heat pump coefficient of performance due to an increase in the evaporation/condensation temperature difference. Further, the condensation temperature is reduced due to a lower heat transfer rate to the external ambient. The thermal load temperatures decrease until the on/off control cycle begins.

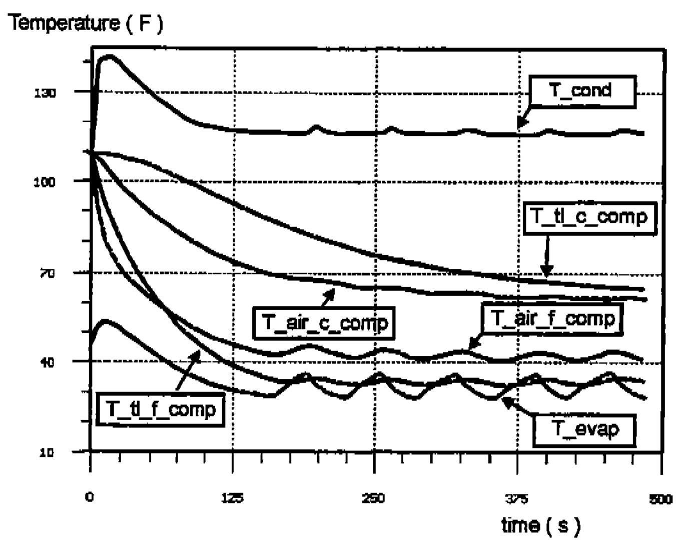

Figure 2 - Time evolution of several temperatures of a refrigerator cabinet following thermal load insertions. T_cond = condensation temperature.

The results for the condensation temperature do not decrease during the off period of the cycle because of the steady-state assumption used in the heat pump model.

CONCLUSIONS

The approach presented is this article is less time consuming than more complex models normally used in calculating the transient behavior of heat pumps. The results here shown qualitatively agree with temperature time evolution shown in other papers. Calculation for transient analysis of domestic refrigerators, obtained with the computer program here presented, will be further validated with experimental data in near future. An advantage of the simplified model here described is the possibility of using personal computers due to the relatively low computational effort required for running the calculations.

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