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Transient behaviour of Working Refrigerants

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ABSTRACT
Transient behaviour of working refrigerant was studied and a computer model to predict these behaviours was written, which also was a base for comparison of the behaviour of different working refrigerants. The study involved developing equations which predict the movement of refrigerant in to a system and the change of state of refrigerant in the system components over time. The continuous physical process occurring in the system components has been represented by the lumped model. Computer models were used to compare the performance of working refrigerants R12, R134a, R413a, R-22 and its replacement R-407c. The new drop-in replacement, R413a, showed a comparable performance compared with R-12 and R-134a at lower evaporator, lower condenser temperatures and higher compressor speeds. The comparative analysis showed that R-407c had an acceptable thermodynamic performance compared with that of R-22.

Key words: Refrigeration, refrigerant, transient.

1. INTRODUCTION
Knowledge of the time and stability characteristics of the individual components and the effects of coupling the components into a system, especially when operating away from design conditions, is becoming increasingly important in the design and control of refrigeration systems. The classical approach to the mathematical analysis of a refrigeration system describes the physical events taking place with equations derived by applying a steady flow balance to each section in turn. These steady state values are used to determine the size and thermal capacities of components which make up the system but equipment that operates efficiently under design conditions may perform poorly at transient.

2. SIMULATION MODEL
Looking at the results of transient modelling [3], [8] shows that, the most problems concerning the dynamics are located in the evaporator and expansion valve rather than compressor and condenser. This concept was adopted in the present research, in which the model for the condenser and compressor were simpler than those of the evaporator and expansion valve. The state of the refrigerant in the heat exchangers was established by writing the conservation equations of mass and energy, together with the state equations which provide the various refrigerant properties needed.

2.1. Evaporator model
The evaporator was divided into two regions, evaporation region and superheating. The approach to analysis is based on the lump model for two regions. Mass and energy equations, along with the mass flow equations for the expansion valve and compressor, describe the temporal behaviour of the complete system. The main idea in the evaporator model is to predict the dynamic behaviour of the evaporator temperature T_e, and the superheat temperature T_{sh} (outlet refrigerant temperature).

2.2. Evaporation region
The evaporation region is considered as one lumped model for coolant, tubes and refrigerant, having variable evaporation length and consisting of three zones, the refrigerant, the coolant and the tube.

Refrigerant Model:

The energy balance equation for refrigerant in the evaporation region can be written:

$$\frac{d}{dt}(M_{ve}h_{ve} + M_{le}h_{le}) = \dot{m}_v h_{ei} - \dot{m}_{com} h_{ve} + \alpha_{re} A_{ee}(T_{pe} - T_e) \tag{1}$$

The inside area of the tubes for the evaporation region is as follows:

$$A_{ee} = n_e \pi \ d_i \ L_e$$

Tube model:

$$M_{pe}(c_p)_{pe}\frac{dT_{pe}}{dt} = \alpha_{ce} A_{eo} \ (T_{cea} - T_{pe}) - \alpha_{re} A_{ee}(T_{pe} - T_e) \tag{2}$$

Where;

$M_{pe} = \rho_{pe}\, A_{pe}\, L_e$, $A_{pe} = n_e\, \pi\, d_o\, t$, $A_{eo} = n_e\, \pi\, d_o\, L_e$
and $T_{cea} = \dfrac{T_{cei} + T_{ceo}}{2}$

Coolant model:

$$M_{cea} (c_p)_{cea} \frac{dT_{cea}}{d\,t} = \dot{m}_{ce} (c_p)_{cea} (T_{cei} - T_{ceo}) \frac{L_e}{L_t} - \alpha_{ce}\, A_{eo} (T_{cea} - T_{pe}) \tag{3}$$

$$M_{cea} = \rho_{cea}\, A_{cea}\, L_e$$

$$A_{cea} = \pi \frac{d_e^2}{4} - n_e\, \pi \frac{d_o^2}{4}$$

Continuity equation:

$$\frac{d}{dt}(M_{ve} + M_{le}) = \dot{m}_v - \dot{m}_{com} \tag{4}$$

where; $M_{ve} = \alpha_{vfe}\, A_e\, \rho_v\, L_e$

and

$$M_{le} = (1 - \alpha_{vfe}) A_e\, \rho_l\, L_e$$

Where $A_e = n_e \pi \dfrac{d_i^2}{4}$

Hughmark [4] proposed an equation that describes the mean void fraction in the evaporator as a function of refrigeration mass flow rate, evaporation temperature and inlet quality as follow:

$$\alpha_{vfe} = ((0.7772471 - 0.019571\, T_e) + (0.0008815 - 0.000000\, T_e)\dot{m}_r$$
$$+ (0.3425608 + 0.0045591 T_e) x + (-0.0004751 - 0.0000216\, T_e)\, \dot{m}_r\, x$$
$$+ (-0.6198063 - 0.0092092\, T_e) x^2 + (0.0016218 + 0.000555 T_e) \dot{m}_r\, x^2$$

(5)

where; $x = \dfrac{h_v - h_i}{h_{lv}}$

Method of Solution:

Substituting for M_{ve} and M_{le} into Equation (4) gives:

$$\frac{d}{d\,t}(\alpha_{vfe}\, A_e\, \rho_v\, L_e + (1 - \alpha_{vfe}) A_e\, \rho_l\, L_e) = \dot{m}_v - \dot{m}_{com} \tag{6}$$

Substituting for $\alpha_{v\,fe}$ and using the following relations:

$$\frac{d\rho_v}{dt} = \frac{d\rho_v}{dT_e} \frac{dT_e}{dt} \tag{7}$$

and

$$\frac{d\rho_l}{dt} = \frac{d\rho_l}{dT_e} \frac{dT_e}{dt} \tag{8}$$

Equation (6) is converted to an ordinary differential equation as follows:

$$A_e \begin{bmatrix} \left[\alpha_{v\,fe}\, \rho_v + (1 - \alpha_{v\,fe}) \rho_l \right] \dfrac{dL_e}{dt} + \\ \left[\alpha_{v\,fe}\, L_e \dfrac{d\rho_v}{dT_e} + (1 - \alpha_{v\,fe})\, L_e \dfrac{d\rho_l}{dT_e} \right] \end{bmatrix} = \dot{m}_v - \dot{m}_{com} \tag{9}$$

Substituting for M_{ve} and M_{le} into equation (1) gives:

$$\begin{aligned} &(\alpha_{v\,fe}\, h_{ve} \frac{d\rho_v}{dT_e} + (1 - \alpha_{v\,fe})\, h_{le} \frac{d\rho_l}{dT_e}) A_{ee}\, L_e + \\ &(\alpha_{v\,fe}\, \rho_v\, c_v + (1 - \alpha_{v\,fe}) \rho_l\, c_l) A_{ee} L_e \frac{dT_e}{dt} = \\ &\dot{m}_{ve}\, h_{ie} - \dot{m}_{com}\, h_{ve} + \alpha_e \pi\, n_e\, d_i (T_{pe} - T_e) L_e - \\ &(\alpha_{v\,fe}\, \rho_v h_{ve} + (1 - \alpha_{v\,fe}) \rho_l h_{le}) A_{ee} \frac{dL_e}{dt} \end{aligned} \tag{10}$$

Equations (2), (3) (9) and (10) are then a set of equations describing the dynamic behaviour of the evaporation region. To keep the numerical solution simple for the dynamic part, a single step method, "Euler's method", has been used for solving the differential equations. It uses a truncated Taylor series expansion and may be expressed as follows:

$$f_{(f+dt)} = f_t + \frac{df_t}{dt} dt \tag{11}$$

2.3. Evaporator superheat region model

The superheat region connects the evaporation region to a single pipe in which the feeler bulb is attached. The length of the superheat region may be approximated by:

$$L_s = L_t - L_e \tag{12}$$

Like the evaporation region, the evaporator superheat region is considered as one lumped model consisting of three zones, the refrigerant, and the coolant and tube wall

Refrigerant model:

$$M_{se}\, c_{se} \frac{d\ T_s}{d\ t} = \alpha_{es}\, A_{es}\, L_s\, (T_{pes} - T_s) - \dot{m}_{com}\ c_s (T_{sh} - T_e) \tag{13}$$

Where; $M_{se} = \rho_s\ n_e\ \pi\ d_i\ L_s$ and $A_{es} = n_e\, \pi\ d_i\ L_s$

Tube model:

Energy balance equation for the tube walls:

$$\rho_{pes} (c_p)_{pes} A_{pe} \frac{d\,T_{pes}}{d\,t} = \alpha_{ce} \pi n_e d_o (T_{ceas} - T_{pes}) - \alpha_{es} \pi\, n d_i (T_{pes} - T_s) \tag{14}$$

Where; $T_{ceas} = \dfrac{(T_{cei} + T_{ceso})}{2}$

Coolant model:

$$\begin{aligned} &\rho_{ceas} (c_p)_{ceas} A_{cea} \frac{d\,T_{ceas}}{d\,t} = \frac{(c_p)_{ceas} \dot{m}_{ce}}{L_t} (T_{cei} - T_{ceso}) - \\ &\alpha_{ce} \pi n d_o (T_{ceas} - T_{pes}) \end{aligned} \tag{15}$$

Method of Solution:

Equations (13), (14) and (15) describe dynamic behaviour of the evaporator superheat region. The Euler's method again was used to solve these equations for $\dfrac{dT_s}{dt}$, $\dfrac{dT_{ps}}{dt}$ and $\dfrac{dT_{ceas}}{dt}$.

Superheat temperature at the end of the evaporator, T_{sh} , may be approximated by:

$$T_{sh} = 2\, T_s - T_e$$

Heat transfer coefficients:

The refrigerant inside the evaporator may exist in the state of superheated vapour or a mixture of liquid and vapour. Most investigations of heat transfer coefficients are related to steady flow and have ignored influences such as the presence of lubricating oil or pressure drop. The inside surface heat transfer coefficient was calculated by equation

$$Nu = \frac{\alpha_{re}\, d_e}{k} = 0.0082\, \mathrm{Re}^{0.8} \left(\frac{\Delta\, x\, h_{lv}}{gL} \right)^{0.4} \tag{16}$$

where $\Delta\, x = 1 - x$

The heat transfer coefficient for the superheat region was calculated using the Dittus and Boelter [5] correlation for one phase flow inside a pipe which is often used for such situations.

$$Nu = 0.023\, \mathrm{Re}^{0.8}\, \mathrm{Pr}^{0.4} \tag{17}$$

For a given evaporator, when the coolant flows in the shell outside the tubes, the heat transfer coefficient for water, α_{ce} is a function of the coolant mass flow rate [7]

$$\alpha_{ce} = C_2\,(\dot{m}_{ce})^{0.6} \qquad (18)$$

C_2 depends on the geometry of the heat exchanger.

For staggered tube banks:

$$C_2 = 0.33(\mathrm{Pr})^{0.33}\left(\frac{D}{A\mu}\right)^{0.6}$$

For an in line arrangement:

$$C_2 = 0.26(\mathrm{Pr})^{0.33}\left(\frac{D}{A\mu}\right)^{0.6}$$

2.4. Condenser

The condenser is described by one single control volume (condensation region), and the other regions (de-superheating and subcooling) are concentrated in the saturation region. The same general assumptions made for the evaporator model were also applied to the condenser. The approach taken in modelling the condenser is to consider it as one lumped region (condensation region). The time step of the dynamic model of the condenser is assumed to be the same as the time step of the evaporator, to avoid problems when combining the models.

Since the pressure drop of the refrigerant in the condenser shell was ignored, only the continuity and energy equations should be found as basic equations to model the condenser.

Refrigerant Mode:

By applying the law of conversation of energy to the refrigerant in the condenser, the energy balance equation can be written:

$$\frac{d}{dt}(M_{vc}\,h_{vc} + M_{lc}\,h_{lc}) = \dot{m}_{com}\,h_{ci} - \dot{m}_v\,h_{co} - \alpha_{rc}\,A_{rw}\,(T_c - T_{pc}) \qquad (19)$$

Where; $A_{rw} = n_c\,L_c\,\pi\,d_o$

Tube Model:

$$\rho_{pc}A_{pc}\,L_c\,c_{pc}\frac{dT_{pc}}{dt} = \alpha_{rc}A_{rw}(T_c - T_{pc}) - \alpha_{cc}A_{cc}(T_{pc} - T_{cca}) \qquad (20)$$

Where; $A_{cc} = n_c\,L_c\,\pi\,d_i$, $A_{pc} = n_c\,\pi\,d_o\,t$ and

$$T_{cca} = \frac{T_{cci} + T_{cco}}{2}$$

Coolant Model:

$$M_{cc}\,c_{cc}\frac{dT_{cca}}{dt} = \alpha_{cc}\,A_{cc}(T_{pc} - T_{cca}) - \dot{m}_{cc}\,c_{cc}\,(T_{cca} - T_{cci}) \qquad (21)$$

where; $M_{cc} = \rho_{cc}n_c L_c \pi \dfrac{d_i^2}{4}$

Continuity equation:

$$\frac{d}{dt}(M_{vc} + M_{lc}) = \dot{m}_r - \dot{m}_l \qquad (22)$$

where;

$M_{vc} = V_c\,\alpha_{vfc}\,\rho_v$ and $M_{lc} = V_c(1-\alpha_{vfc})\rho_l$

Where the condenser volume V_c;

$$V_c = \pi\,L_c\frac{d_c^2}{4} - n_c\,\pi\,L_c\frac{d_o^2}{4}$$

Method of Solution:

Substituting for M_{vc} and M_{lc} into equation (19 and using equation (7) and (8) yields:

$$(\rho_v - \rho_l)\frac{d\alpha_{vfc}}{dt} = \frac{1}{V_c}(\dot{m}_r - \dot{m}_l) - (\alpha_{vfc}\frac{d\rho_v}{dT_c} + (1-\alpha_{vfc})\frac{d\rho_l}{dT_c})\frac{dT_c}{dt} \qquad (23)$$

Using equations (7) and (8) into equation (23yields:

$$\begin{aligned}&(\alpha_{vfc}\,h_{vc}\frac{d\rho_v}{dT_c} + (1-\alpha_{vfc})h_{lc}\frac{d\rho_l}{dT_c})\frac{dT_c}{dt} + \\ &(\alpha_{vfc}\,\rho_v c_v + (1-\alpha_{vfc})\rho_l c_l) = \frac{1}{V_c}[(\dot{m}_{com}\,h_{ci} - \dot{m}_v\,h_{oo}) - \\ &(\alpha_{rc}A_{rw}(T_c - T_{pc})] - (\rho_v h_{vc} - \rho_l h_{lc})\frac{d\alpha_{vfc}}{dt}\end{aligned} \qquad (24)$$

Equations (20), (21), (23) and (24) are first order differential equation; which represent expressions for the rate of change of tube wall temperature, coolant mean temperature, refrigerant quality and condensing temperature.

Heat transfer coefficient:

The average heat transfer coefficient associated with a vapour refrigerant condensing on N horizontal tubes is estimated by:

$$\alpha_{rc} = 0.725\left(\frac{k_r^3(\rho_f - \rho_g)g\,h_{fv}}{Nd_o\mu_r(T_c - T_{pc})}\right)^{0.25}$$

Coolant side heat transfer coefficient:

Sieder and Tate [5] presents the following relation for fully laminar flow inside tubes:

$$Nu_d = 1.86\left(\mathrm{Re}_d\,\mathrm{Pr}\right)^{\frac{1}{3}}\left(\frac{d}{L_c}\right)^{\frac{1}{3}}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

2.5. Mass flow rate through the expansion valve

The mass flow rate through the expansion valve is assumed to be depend on the orifice throat area, condensing pressure and evaporator pressure [9].

$$\dot{m}_l = C_d\,A_v\sqrt{\rho_l(P_c - P_e)} \qquad (25)$$

A_v is a linear function of the degree of superheating.

Using a proportional rule:

$$\frac{A_v}{A_t} = \frac{DSH - SSH}{MSH} \qquad (26)$$

Where;

$DSH = T_{sh} - T_e$

Finally the valve area;

$$A_v = \frac{T_{sh} - T_e - SSH}{MSH}A_t \qquad (27)$$

Thus the mass flow rate through the expansion valve is:

$$\dot{m}_v = C_d\left(\frac{DSH - SSH}{MSH}A_t\right)\sqrt{\rho_l(P_c - P_e)} \qquad (28)$$

2.6. Bulb model

The model here is a simplified version of Yasuda et al. [8], where feeler bulb was described by three lumped dynamic models making use of separate control volumes, refrigerant line, bulb wall and bulb contents. It was assumed that the refrigerant temperature is equal to the evaporator superheat temperature.

For the bulb wall:

$$C_{bw}\frac{dT_{bw}}{dt} = H_{bws}(T_{sw} - T_{bw}) - H_{bwc}(T_{bw} - T_{bc}) \qquad (29)$$

Where;

$H_{bws} = \alpha_{bws}\,A_{bws}$

$H_{bwc} = \alpha_{bwc}\,A_{bwc}$

For the refrigerant line wall:

$$M_{rbw}\, c_{rbw}\, \frac{d\,T_{rbw}}{dt} = \alpha_{es}\, \pi\, d_i\, L_b\, (T_{sh} - T_{rbw}) - \alpha_{bws} A_{bws} (T_{rbw} - T_{bw}) \tag{30}$$

Where;

$M_{rbw} = \rho_{rbw}\, A_{rbw}\, L_b$

and

$A_{rbw} = \pi\, d_o\, t$

For the bulb contents:

$$C_{bc}\, \frac{dT_{bc}}{dt} = H_{bwc}\, (T_{bw} - T_{bc}) \tag{31}$$

Where;

$H_{bwc} = \alpha_{bwc}\, A_{bwc}$

Characteristic values for the bulb section such as C_{bc}, C_{bw}, H_{bwc} and H_{bws} are confidential to manufacturers. A report by Yasuda et. al [8] estimated the values of C_{bw}, H_{bwc}, C_{bc} and H_{bws} as follows:

$C_{bc} = 0.003726 \quad (\frac{kJ}{K})$ and $H_{bws} = 1.057e-3 \quad (\frac{kJ}{s})$

$C_{bw} = 0.007284 \quad (\frac{kJ}{K})$ and $H_{bwc} = 1.86 \times 10^{-4} \quad (\frac{kJ}{s})$

Equations (29), (30) and (31) are solved for $\frac{dT_{rbw}}{dt}$, $\frac{dT_{bw}}{dt}$ and $\frac{dT_{bc}}{dt}$. The Euler approximation is used to solve these equations for T_{rbw}, T_{bw} and T_{bc}.

2.7. Comperessor model

The compressor model formulation is based on the following simplifying assumptions:

1-The compressor process is a polytropic process.

2-Motor speed is constant.

3-Pressure drop due to refrigerant flow through the valves is ignored.

4- Heat losses are neglected.

5-Interaction of the working refrigerant and lubricant are neglected.

Based on the above assumptions, the mass flow rate through the compressor and the work done by the compressor can be easily derived as functions of compressor geometry, suction and discharge pressure ratio, compressor speed, suction temperature and specific heat ratio of the refrigerant. The mass flow through the compressor can be calculated by:

$$\dot{m}_r = \rho\, V\, \eta_{vol}\, N \tag{32}$$

3. SIMULATION SCHEME

The simulation model was run and results obtained from the developed simulation model compared with experimental data obtained in Sheffield University, indicating that the model in its present state of development can predict the dynamic behaviour of a system.

The input data to the program are physical specification of condenser, evaporator, expansion valve, bulb and compressor plus the input data of coolant and assuming $DSH = T_{bc} - T_e$. Time step and maximum time of operation are also necessary.

The properties of some of the material from which the system components were constructed and some of the fluid properties are assumed to be constant. The following information in Table 1 was used for all system components.

Table 1 Material information

Metal	Density $\frac{kg}{m^3}$	Specific heat $\frac{kJ}{kg\ K}$	Thermal conductivity $\frac{kW}{mK}$
Copper	8930	0.38	0.385
Steel	7860	0.42	0.075

The component models are combined to configure a dynamic simulation model for the entire refrigeration system. The specification and operating conditions of all components and initial values for all the independent variables are given as input values in the simulation program. At the start of the simulation program, it is assumed that the compressor has been shut down for a sufficiently long time so that the temperature of the refrigerant in the various components of the refrigeration system is constant. The vapour pressure in the system at this stage will also be the same throughout. Once the initial values at $t = 0$ are known, the program evaluates the various variable at the time $t + \Delta t$. The program starts with some preliminary calculations for system pressure of the evaporator and condenser. Once the integration time step and maximum operation time are determined, a series of subroutines are used to calculate the mass and energy transfer in the condenser, evaporator, compressor and expansion valve. In each iteration the local properties of the refrigerant in the system are determined. This is repeated for each time step until the end of the transient simulation is reached. Each component has its own inputs which have no direct influence on the other components, such as coolant inlet conditions for heat exchangers, compressor characteristics such as compressor speed. The final equilibrium conditions of the refrigeration system after start-up are a function of the refrigerant mass, physical characteristics of each component, refrigerant and heat transfer coefficients.

4. RESULTS AND DISCUSSIONS

The model was employed to predict and compare the dynamic behaviour of the working refrigerants R12, R134a, R22, R413a, and R407c. It was assumed the system has not been operated for a long time, and all temperatures were the same as ambient temperature which was assumed to be 15 °C. Figures 1 through 6 are samples of the transient behaviour (start-up). These figures show the variation of system pressure and temperature, evaporator cooling capacity, volumetric refrigeration effect (VRE) and coefficient of performance against time. Numerical results depicted in these figures indicate that the system stabilised after periods of 300 seconds from start-up. At time equal to zero, evaporating and condensing temperatures are equal. Immediately after the compressor is started, the condensing pressures and temperatures rise rapidly and then gradually approach their steady state value (Figures 1 and 2). .

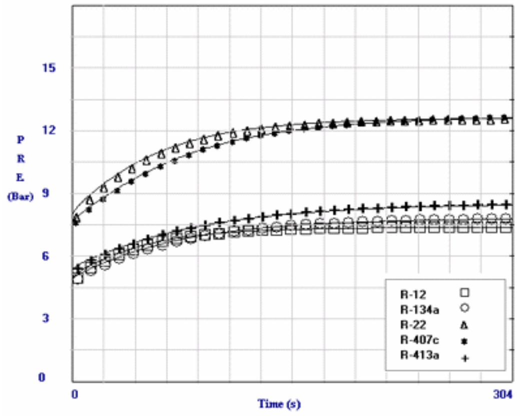

Figure 1 Variation of condenser pressure

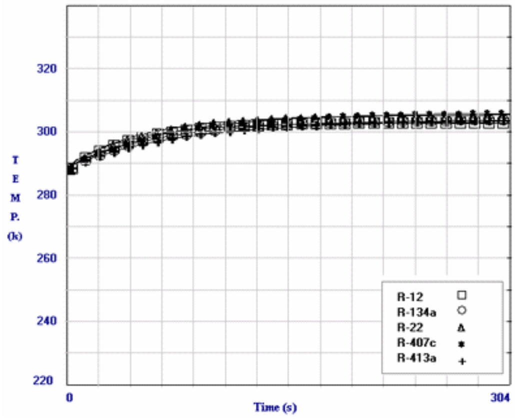

Figure 2 Variation of condenser temperature

The evaporating pressures and temperatures drop rapidly at first and then gradually approach a steady state value (Figures 3 and 4). The sudden drop in evaporating pressure and following that in the evaporator temperature causes the refrigerant in the evaporator to vaporise rapidly. The evaporator pressure and temperature increase after about 20 seconds. That represents the instant when the feeler bulb pressure at the outlet of the evaporator reaches a point where it forces the expansion valve to open, thus increasing the evaporating pressure and temperature to steady values. After the first time step, the compressor delivers refrigerant to the condenser against a rapidly increasing pressure differential. As the evaporator pressure decreases the density of the refrigerant entering the compressor decreases. On the other hand as the compressor pressure ratio increases, the volumetric efficiency of the compressor decreases.

Figure 6 shows the cooling capacity of the evaporator plotted against time for working refrigerants. During the first 150 sec the cooling capacity of working refrigerants decrease rapidly and level off to a steady value of 2.1, 2.2 and 2.25 kW for R12, R134a and R413a respectively. R407c shows a similar behaviour with that of R22 and both refrigerants reached the steady value at the same time. A 2 % increase in the cooling capacity with R407c over that of R22 was seen at the steady state. Figure 5 shows the coefficient of performance of the system plotted against time. During the first 10 sec, the coefficient of performance rises very rapidly and then falls and levels off to a steady value of 1.8 for R-12 and 1.6 for R413A. As soon as the liquid refrigerant starts flowing through the expansion valve, the cooling process starts and the coefficient of performance jumps to large value. Simultaneously, the pressure differential across the compressor starts increasing, with the result that the value of compressor power increases and the coefficient of performance is reduced

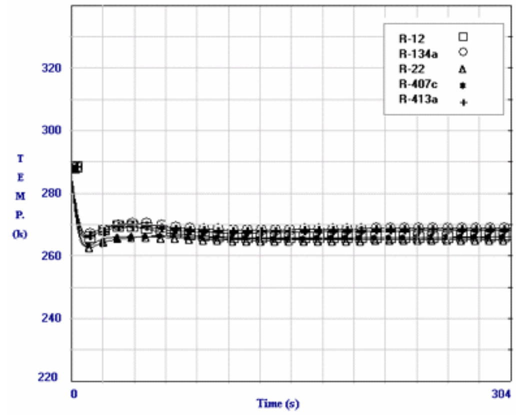

Figure 3 Variation of evaporator temperature

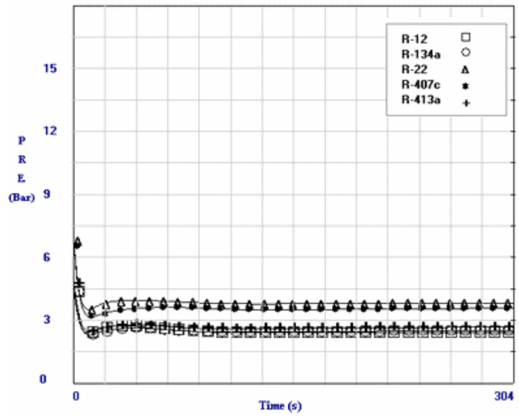

Figure 4 Variation of evaporator pressure

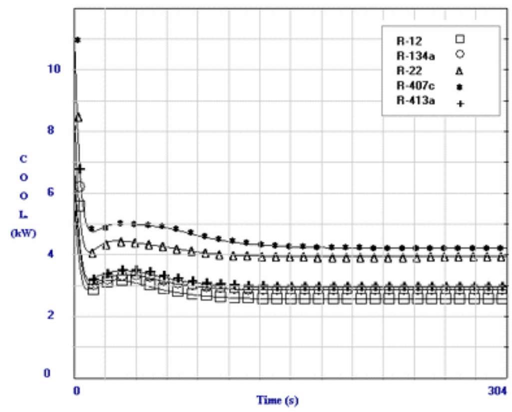

Figure 5 Variation of cooling capacity

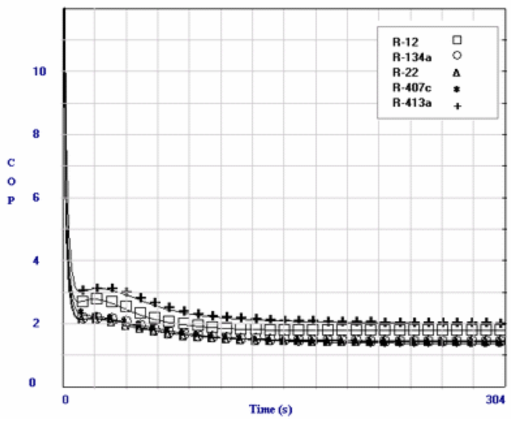

Figure 6 Variation in coefficient of performance

5. CONCLUSIONS

A description of the mathematical models and sub-models to simulate the dynamic behaviour of a vapour compression cycle and its thermal functioning has been presented. The mathematical model involves developing a set of equations which predict the movement of refrigerant in the system and its change of state within each component as a function time. Since pressure drops in the evaporator and condenser were ignored, the mass and energy equations, along with the mass flow rate for the expansion valve and compressor describes the temporal behaviour of the complete system. Property relations and correlation to evaluate heat transfer coefficients complete the set of equations.

The model was derived with an emphasis on the evaporator and expansion valve. The governing differential equations were described and continuity and energy equations have been solved. Once the integration time step was determined, the program was used to calculate the mass and energy transfer in the condenser, evaporator, expansion valve and compressor. During each time step, temperature and pressure for both condenser and evaporator, and the mass flow rate through the compressor and expansion valve were determined. This process was repeated for each time step until the end of the transient simulation was reached. The model was used to compare the transient behaviour of refrigerant R-12, R22, R134a, R407c and R413a. The results showed that new alternatives refrigerants (R407c and R413a) had acceptable behaviour relative to that of R-12 and R22.

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7. NOMENCLATURE

Symbol	Meaning	Unit
M	mass of refrigerant	*kg*
C_p	specific heat	*kJ/kgK*
L	length	*m*
T	temperature	^oK
$\dot{m}$	mass flow rate	*Kg/s*
ρ	density	kg/m^3
α_{vfe}	void fraction	——
α	heat transfer coefficient	kW/m^2K
Subscripts		
s,sh	super heat	
e	evaporator	
Com	compressor	
r	refrigerant	
c	coolant	
l	liquid	
w	tube	
v	vapour	
pes	Pipe in super heat region	