1998

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J. Rigola
*Universitat Politecnica de Catalunya*

C. D. Perez-Segarra
*Universitat Politecnica de Catalunya*

O. Garcia-Valladares
*Universitat Politecnica de Catalunya*

J. M. Serra
*Electrolux Compressors Companies*

M. Escriba
*Electrolux Compressors Companies*

***See next page for additional authors***

**Authors**
J. Rigola, C. D. Perez-Segarra, O. Garcia-Valladares, J. M. Serra, M. Escriba, and J. Pons

# NUMERICAL STUDY AND EXPERIMENTAL VALIDATION OF A COMPLETE VAPOR COMPRESSION REFRIGERATING CYCLE

J.Rigola, C.D.Pérez-Segarra, O.García-Valladares

Laboratori de Termotècnia i Energètica
Dept. Màquines i Motors Tèrmics. Universitat Politècnica de Catalunya
Colom 11, 08222 Terrassa (Barcelona), Spain
FAX: +34-3-739.81.01; Tel. +34-3-739.81.92
E-mail: labtie@termo1.mmt.upc.es

J.M.Serra, M.Escribà, J.Pons

Electrolux Compressors Companies-Spain
08192 Sant Quirze del Vallès (Barcelona), Spain
FAX: +34-3-712.28.10; Tel. +34-3-710.60.08

## ABSTRACT

A detailed numerical simulation of the thermal and fluid-dynamic behaviour of a single stage vapor compression refrigerating unit has been carried out. Different subroutines modelize the physical flow phenomena produced inside all the different elements that make up the compression cycle. The whole modelization is based on a main program that calls the subroutines mentioned above sequentially until the convergence is reached. The analysis is possible in transient or steady state of a wide range of situations, fluids, geometries and boundary conditions. The software allows the knowledge of the flow variables distribution, which are evaluated at each point of the discretized domain.

The latest improvements of the software have been focussed on modelizing the physical flow phenomena of a complete vapor compression considering pure refrigerants plus mixtures. Thus, it has been necessary to improve and adapt all the empirical information and all the thermodynamic and transport properties to the mixtures using local conditions. All the additional information necessary in the compressor model is obtained using an advanced hermetic compressor numerical simulation model. Finally, the numerical solution of the two-phase flow inside ducts can be evaluated by means of a step-by-step method or by means of a pressure-based method (SIMPLEC).

The experimental unit designed and built to analyse single stage vapor compression refrigerating equipment has been numerically contrasted for each element and for all the whole refrigerating system using R-134a. In addition to this validation, some illustrative results are shown using non-chlorinated hydrocarbon refrigerants (HFC134a), hydrocarbons (HC600a) and Near Azeotropic Refrigerant Mixtures NEARMs (R404A).

## INTRODUCTION

Several years ago, the complexity to solve the governing equations of the flow and the heat transfer phenomena were a handicap to design and optimize thermal equipment, which were essentially based on continuous experimentation. On the other side, some environmental problems caused by refrigerants OPD (Ozone Depletion Potential) and GWP (Global Warming Potential) need a solution [1]. Together with this, the use of inefficient equipment produces an excessive power consumption.

For these reasons, the obtaining of general and flexible design methods is very important in the applications and the optimization of these units in order to take into account different aspects such as: the specific geometric characteristics of each component of the system, the consideration of thermal loads, the use of new and non-contaminant refrigerants, etc.

Several works presenting different models can be found in scientific literature, focussing their attention on modelizing typical vapor compression systems, their components, the overall refrigeration cycle and their experimental comparison. Among them the models developed by Chi and Didion [2], Murphy and Goldsmith [3][4], and Rajendran and Pate [5] are based on global balances between the inlet and outlet sections of the different elements of the system. MacArthur [6], Jung and Radermacher [7] and Yuan and O'Neal [8] solve the condenser and the evaporator using finite difference methods and considering a one dimensional formulation of the governing equations. There is no comparison with experimental data. MacArthur [6] solves the compressor and the expansion valve using parametric models based on non-dimensional correlations. Jung and Radermacher [7] work with pure and mixed refrigerants considering a non-ideal isentropic compressor and solving the simulation system using two possible methods: the successive substitution and the Newton-Raphson method. Yuan and O'Neal [8] consider the compressor as a polytropic process, and the suction and discharge valves adiabatic and isoenthalpic, while the pressure drop in the condenser and evaporator is not considered. In this paper a numerical simulation of a single stage refrigerating unit previously developed [9][10] and recently improved is presented, and an experimental validation is showed together with different illustrative results.

The unit studied consists of a double-pipe condenser and evaporator, a capillary expansion tube device, a reciprocating compressor, the different connecting tubes and extra elements. The heat exchangers and the capillary tube are solved on the basis of a control volume formulation of the governing equations (continuity, momentum and energy), considering one-dimensional flow [11][12][13]. The creation entropy is considered in the capillary tube in order to detect the limitation of the physical process produced under critical flow conditions. This formulation requires the use of empirical information for the evaluation of the shear stress, convective heat transfer and void fraction. All the empirical information is commented in the references indicated above. The empirical information required to permit the use of mixtures has been obtained in [14]. The modelization also takes into account the conduction heat transfer through the external solid elements (tubes and insulator) considering transient and two-dimensional phenomena. The longitudinal conduction in heat exchangers and connecting tubes has also been considered. The compressor has been modelized by means of global balances between its inlet and outlet cross-section. This compressor model also needs some empirical information: the volumetric efficiency, heat losses and input power transferred to the gas. This empirical information has been obtained using an advanced numerical simulation model of hermetic reciprocating compressor [15][16].

Several interesting new features are presented. All the empirical information has been adapted or changed to have the possibility to use mixed refrigerants. The heat exchangers numerical solution allows the use of a pressure-based method. The compressor model which feeds the additional information needed in the single stage studied has been improved. Finally, a new thermodynamic and transport properties database has been implemented to work with pure and mixed refrigerants. The idea to implement a pressure-based method in the condenser and evaporator is focussed on the possibility of working using complex geometries (parallel paths), complex phenomena (reversal flow) and, as a future purpose, extend this kind of analysis to the complete cycle.

An experimental unit has been developed to compare the results obtained with the numerical simulation. The experimental unit registers temperature, mass flow rate and pressure in both the main circuit and the secondary circuits. The mass flow rate and the inlet temperature are independently fixed in the condenser and evaporator secondary circuits. Once this has been carried out, the rest of the results are a consequence (in the experimental unit and in the numerical simulation).

## NUMERICAL SIMULATION

The numerical resolution consists of a main program composed of different subroutines. The mathematical formulation of these subroutines has been carried out to solve the two-phase flow inside a characteristic control volume of a duct and the two-dimensional conduction heat transfer through a characteristic control volume of an external solid element. The single-phase flow liquid or gas and the one-dimensional heat conduction through the tube wall represent particular cases. The compressor process is also formulated. The different elements of the equipment (evaporator, compressor, condenser, expansion device and connecting tubes) are solved calling some of the mentioned subroutines in a convenient way. A brief description of the mathematical formulation and the global algorithm to solve the complete refrigerating cycle is indicated below.

### *Two-phase flow inside tubes*

This numerical simulation is explained in detail in [11][12]. The one-dimensional and transient governing equations of the fluid flow are integrated numerically using a fully implicit step-by-step numerical scheme or by means of a fully implicit segregated pressure-based method of SIMPLE-like [12]. The empirical information needed is the convective heat transfer, the shear stresses, and the void fraction for pure refrigerant and mixtures. Both inflow and outflow conditions depending on the case, and wall temperatures are taken as boundary conditions. The governing equations of the flow have been integrated for pressure, velocity and enthalpy instead of temperature to be used by fluid mixtures.

### *Evaporating flow through capillary tubes*

A similar subroutine to the one described in [13] has been used extending its validity range to mixtures. The flow has been considered adiabatic. The one-dimensional and transient governing equations of the fluid flow are also integrated numerically using an implicit step-by-step numerical scheme. The empirical information needed is the shear stress and the void fraction. Since the critical mass flow rate is fixed for a given capillary tube, all the inflow conditions cannot be simultaneously input data. Hence, the inlet mass flow rate or pressure has to be considered as output data, and it is evaluated by means of a Newton-Raphson algorithm.

### *Conduction heat transfer through the solid elements (tubes and insulators)*

The two-dimensional heat conduction equation in the solid element has been discretized considering two-dimensional phenomena (in the longitudinal and radial directions), on the basis of a central-difference numerical scheme [11]. The set of discretized equations has been solved using a line by line algorithm (i.e. the discretized equations are iteratively solved by lines in the axial and radial directions). The fluid temperature and local heat transfer coefficient distribution (inside the tube), the ambient temperature and the temperatures of the solid at the extremes of the tube and insulator are taken as boundary conditions.

*Compressor process*
The modelization has been carried out on the basis of global balances of mass and energy between the inlet and outlet cross-sections of the compressor. This formulation requires additional empirical information for the evaluation of the volumetric efficiency, power consumption and heat transfer losses. This empirical information has been evaluated in detail from an advanced simulation model [15][16], which solves the thermal and fluid-dynamic behaviour of hermetic reciprocating compressors in the whole domain. The one-dimensional and transient governing equations of the fluid flow are discretized using an implicit control-volume formulation and a SIMPLE-like algorithm. The solid thermal behaviour is based on heat global balances at each solid component.

*Global algorithm considering transient or steady state*
The algorithm solves the global equations system using the successive substitution method. Thus, at each time step, the subroutines that solve all the different elements are called sequentially, transferring adequate information to each other, until the convergence is reached. Figure 1 shows the refrigerant unit scheme that has been simulated. Transferred information depends on whether transient or steady state is considered. The boundary conditions for the simulation of the whole system are the inlet temperature, pressure and mass flow rate of the secondary flow in the condenser and evaporator, the compressor speed and the ambient temperature and pressure. The value of the dependent variables for the initial conditions has been evaluated from the solution of the system in steady state using the boundary conditions of the initial time. The global algorithm has been explained in detail in [9][10].

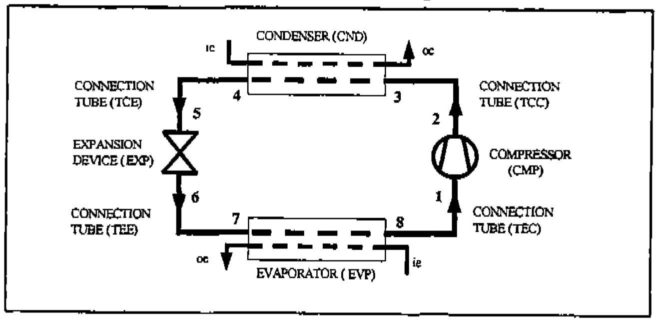

Figure 1. Single stage vapor compression refrigerant unit scheme.

In transient state, the pressure input data in the subroutines of the condenser and the evaporator are the outlet pressure of these elements. Thus, the solution algorithm for these elements requires knowledge of the pressure drop in them, which is iteratively calculated from the preceding iteration. In steady state, the mass flow rate is constant in the whole domain. So the continuity equation applied to each control volume gives n-1 linearly independent equations (n is the total number of control volumes). Therefore, the set of discretized equations is not determined and an additional equation is needed. Even though the total mass of fluid refrigerant can be used as an additional equation, the easiest way is to fix any flow variable at any point of the domain (in this case the outlet compressor pressure has been chosen).

## EXPERIMENTAL UNIT DESCRIPTION

An experimental unit has been presented and explained in detail in [10]. The geometric parameters of the equipment and the measurement instruments has also been explained in the reference commented above. This unit has been built to study single stage vapor compression refrigerating systems and to validate the mathematical model used in the numerical simulation. A schematic diagram of the experimental unit is shown in Figure 2.

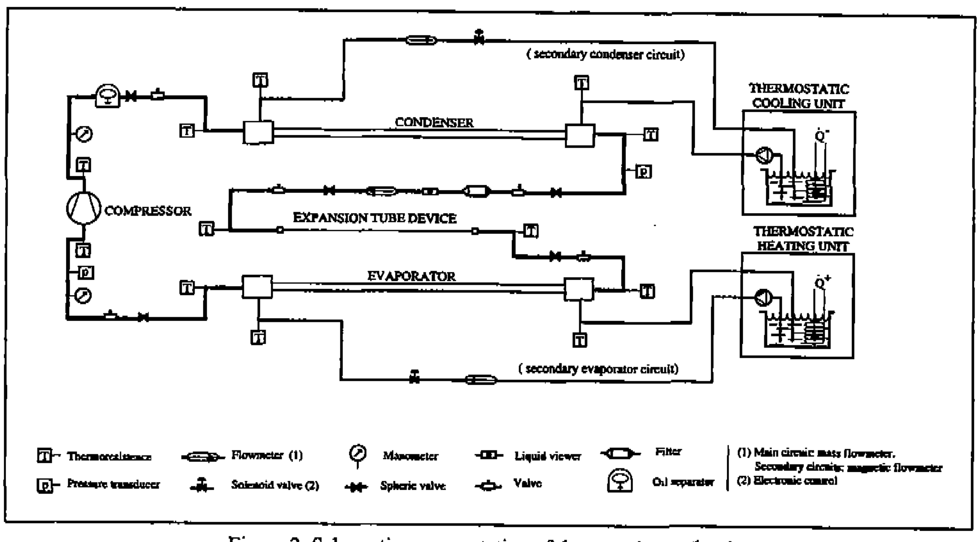

Figure 2. Schematic representation of the experimental unit.

# RESULTS

In this section a comparison between numerical results and experimental data obtained is presented using the unit described above (see Fig. 2) and working with R-134a. The comparison is made for each element of the system (compressor, double tube condenser, expansion device and double tube evaporator) and for the whole system. Furthermore, some illustrative results are presented using HFC134a, hydrocarbon HC600a (isobutane) and a mixed refrigerant R404A working in the same conditions. All the thermodynamic and transport properties have been evaluated by means of a properties database program also used and referenced in [12][16]. The situation analyzed for comparison between numerical and experimental results corresponds to the following values:

| | | |
|---|---|---|
| *Fluid* | R-134a. | |
| *Compressor* | Electrolux Compressor Companies-Spain | |
| *Tube connection* | i) geometry: | diameters: 4.9, 6.4, 44.4 mm; length: 2.1 m; smooth tube. |
| *Condenser* | i) geometry: | diameters: 6, 8, 16, 20, 58 mm; length: 2 m; smooth tube; counterflow. |
| | ii) annulus: | water at Ti = 23.07 C; mass flow rate: 4 l/min. |
| *Tube connection* | i) geometry: | diameters: 11.2, 13, 51 mm; length: 5.1 m; smooth tube. |
| *Capillary tube* | i) geometry: | diameter: 0.65 mm; length: 1.06 m; relative rugosity: $8.8\cdot10^{-3}$. |
| *Tube connection* | i) geometry: | diameters: 8, 9.6, 47.6 mm; length: 0.8 m; smooth tube. |
| *Evaporator* | i) geometry: | diameters: 8, 9.6, 16, 20, 58 mm; length: 6m; smooth tube; counterflow. |
| | ii) annulus: | water at Ti = 19.24 C; mass flow rate: 3 l/min. |
| *Tube connection* | i) geometry: | diameters: 8, 9.6, 47.6 mm; length: 1.1 m; smooth tube. |
| *Cycle* | i) condensation pressure: 13.87 bar; steady state. | |

The two-phase flow empirical information is referenced in [12]. All the additional compressor information for all the numerical results has been evaluated by means of the advanced compressor model [15][16]. The results obtained for the studied cases are presented in Figures 3,4 and 5.

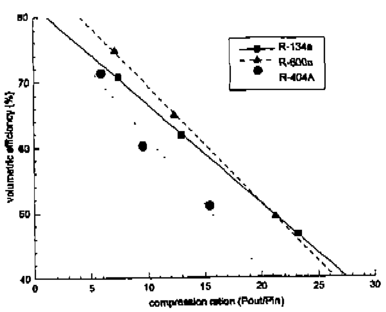


Fig. 3 Volumetric efficiency/compression ratio

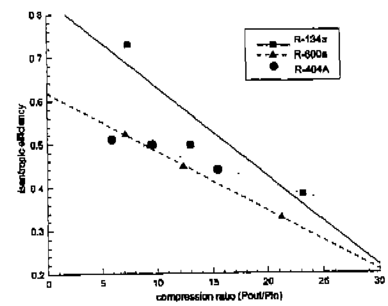


Fig. 4 Isentropic efficiency/compression ratio

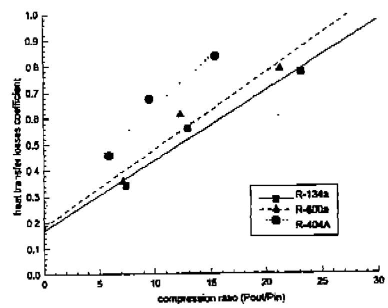


Fig. 5 Heat transfer losses/compression ratio.

The results presented for the ***compressor*** have been carried out using the simple compressor model. However, the additional information for this case has been obtained using the numerical results from Figures 3, 4 and 5. Table 1 shows the comparative results. The differences in the mass flow rate are 4%.

Table 1. Compressor: experimental vs. numerical results (boundary conditions in brackets).

| results | $T_1$(C) | $T_2$(C) | $P_1$(bar) | $P_2$(bar) | $\dot{m}$ (Kg/h) | [illegible] | [illegible] | [illegible] |
|---|---|---|---|---|---|---|---|---|
| *experimental* | *20.11* | *98.45* | *1.357* | *13.87* | *6.33* | - | - | - |
| *numerical* | *(20.11)* | *98.13* | *(1.357)* | *(13.87)* | *6.29* | *60.5* | *78.5* | *78.0* |

Tables 2, 3 and 4 show the comparative results for the ***condenser***, the ***capillary tube*** and the ***evaporator***. As can be seen, good agreement is obtained (eg. mass flow rate prediction in the capillary tube is within 4%). The values $T_{ic}$, $T_{oc}$, $T_{ie}$, $T_{oe}$ are the inlet and outlet temperatures in the auxiliary condenser and evaporator circuits, and the mass flow rates in these circuits are $\dot{m}_{auxc}$ and $\dot{m}_{auxe}$ (see Figure 2).

Table 2. Condenser: experimental vs. numerical results (boundary conditions in brackets).

| results | $T_3$(C) | $T_4$(C) | $\dot{m}$ (Kg/h) | $\dot{m}_{auxc}$ (Kg/h) | $T_{ic}$(C) | $T_{oc}$(C) | $P_{cond}$(bar) |
|---|---|---|---|---|---|---|---|
| *experimental* | *74.74* | *24.75* | *6.33* | *4.00* | *23.07* | *24.72* | *13.87* |
| *numerical* | *(74.74)* | *24.63* | *(6.33)* | *(4.00)* | *(23.07)* | *24.45* | *(13.87)* |

Table 3. Capillary tube: experimental vs. numerical results (boundary conditions in brackets).

| results | $T_5$(C) | $T_6$(C) | $\dot{m}$ (Kg/h) | $P_{cond}$(bar) | $P_{evap}$(bar) | [illegible] |
|---|---|---|---|---|---|---|
| *experimental* | *23.45* | *-19.48* | *6.33* | *1.357* | *13.87* | - |
| *numerical* | *(23.45)* | *-19.48* | *6.37* | *(1.357)* | *(13.87)* | *0.272* |

Table 4. Evaporator: experimental vs. numerical results (boundary conditions in brackets).

| results | $T_7$ (C) | $T_8$ (C) | $\dot{m}$ (Kg/h) | $\dot{m}_{oct}$ (Kg/h) | $T_{ie}$ (C) | $T_{oe}$ (C) | $P_{evap}$ (bar) | [illegible] |
|---|---|---|---|---|---|---|---|---|
| experimental | -19.60 | 19.22 | 6.33 | 3.00 | 19.24 | 18.28 | 1.357 | - |
| numerical | (-19.60) | 19.22 | (6.33) | (3.00) | 19.24 | 17.69 | (1.357) | 0.272 |

For the complete refrigerating system a comparison between experimental and numerical results is presented in Table 5 (in this case only the inlet and outlet flow conditions in the secondary circuit and the condenser pressure are fixed as boundary conditions). The accuracy required has been $1.10^{-6}$. A reasonable accordance between experimental and numerical results has also been obtained.

Table 5. Compression cycle: experimental vs. numerical results (boundary conditions in brackets).

| [illegible] | $T_1$ (C) | $T_2$ (C) | $T_3$ (C) | $T_4$ (C) | $T_5$ (C) | $T_6$ (C) | $T_7$ (C) | $T_8$ (C) | [illegible] | [illegible] | (Kg/h) | $P$ (bar) | $P_{cond}$ (bar) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| exp | 20.11 | 98.54 | 74.74 | 24.71 | 23.45 | -19.48 | -19.60 | 19.22 | - | - | 6.33 | 1.357 | 13.87 |
| num | 19.42 | 97.17 | 76.79 | 24.72 | 24.23 | -18.87 | -18.91 | 19.20 | 0.27 | 0.29 | 6.36 | 1.36[illegible] | (13.87) |

| [illegible] | $T_{ic}$ (C) | $T_{oc}$ (C) | (Kg/h) | $T_{ic}$ (C) | $T_{oc}$ (C) | (kg/h) | $T$ (C) |
|---|---|---|---|---|---|---|---|
| exp | 23.07 | 24.72 | 4.00 | 19.24 | 18.28 | 3.00 | 22.63 |
| num | (23.07) | 24.48 | (4.00) | (19.24) | 18.72 | (3.00) | (22.63) |

Finally, a comparison between three refrigerating fluids (HFC-134a, HC-600a and R-404A) is presented. The situation analyzed has the same geometry as the one indicated at the beginning of this section, the empirical information is also the same used above. The difference is the electrical motor, on which additional information is detailed in Figures 3,4 and 5. The results have been obtained considering only the condensation pressure equal to the saturation pressure of 55°C and ambient temperature of 32°C..

Table 6 shows some important flow variables for the different fluids studied, together with the most important comparative global working parameters. Figure 6 and 7 shows some global results (pressure vs. enthalpy diagram and temperature vs. entropy diagram) of the system working with these fluids.

Table 6. Compression cycle comparative numerical results.

| [illegible] | (bar) | (bar) | (C) | (C) | (C) | (C) | (C) | (C) | (C) | (C) | [illegible] | [illegible] | $\dot{m}$ (Kg/h) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| R134a | 1.388 | 14.84 | 31.98 | 112.75 | 89.66 | 33.85 | 33.40 | -18.53 | -18.57 | 31.98 | 0.33 | 0.36 | 6.08 |
| R600a | [illegible] | [illegible] | 31.99 | 101.49 | 79.98 | 33.00 | 32.72 | -11.25 | -11.27 | [illegible] | [illegible] | [illegible] | [illegible] |
| R404A | [illegible] | 26.00 | 31.97 | 142.15 | [illegible] | 35.14 | 34.54 | -29.79 | -29.80 | 31.98 | [illegible] | [illegible] | [illegible] |

| [illegible] | $W$ (W) | $Q$ (W) | $Q$ (W) | $\dot{m}$ (Kg/h) | $\eta$ (%) | COP |
|---|---|---|---|---|---|---|
| R134a | 191.3 | 299.8 | 369.6 | 6.08 | 65.5 | 1.57 |
| R600a | [illegible] | 255.0 | [illegible] | [illegible] | [illegible] | [illegible] |
| R404A | [illegible] | [illegible] | 480.9 | [illegible] | [illegible] | [illegible] |

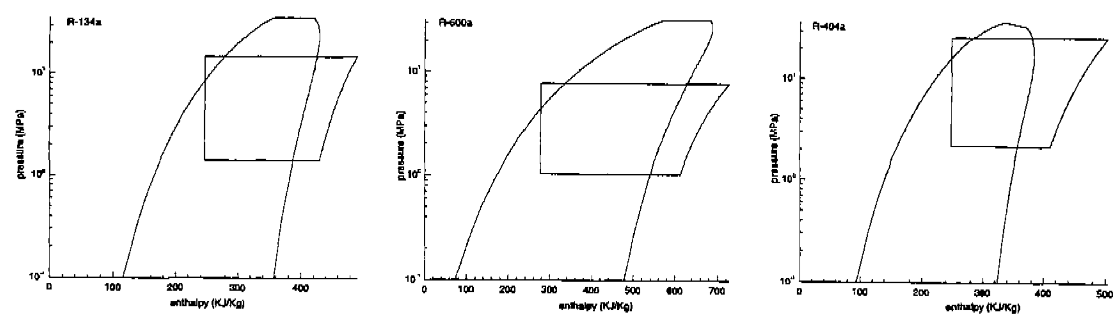


Figure 6. Pressure vs. enthalpy for R134a, R600a, and R404A.

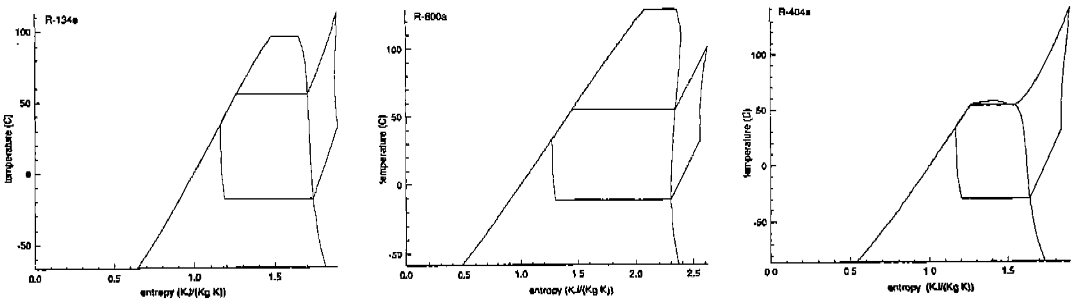


Figure 7. Temperature vs. entropy for R134a, R600a, and R404A.

## CONCLUSIONS

A detailed numerical simulation of the thermal and fluid-dynamic behaviour of a vapor compression refrigerating unit has been developed. The governing equations of the flow (continuity, momentum and energy) have been integrated in transient or steady state using an implicit control-volume formulation and prepared for the use of mixed refrigerants . For the compressor, the additional information needed (volumetric efficiency, heat losses and compressor power consumption) has been obtained from an advanced compressor simulation model. The global simulation solves sequentially the different elements of the system (compressor, condenser, expansion device, evaporator, connecting tubes and extra elements) until the convergence is reached. Good agreement between numerical results and experimental data has been obtained. Finally, some illustrative results comparing three different working fluids have also been presented in order to show the possibilities all this model offers.

## ACKNOWLEDGEMENT

This study has been financially supported by Electrolux Compressor Companies-Spain and by the "Comisión Interministerial de Ciencia y Tecnología", Spain (project number TIC95-0724).

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