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NUMERICAL STUDY OF A SINGLE STAGE VAPOR COMPRESSION REFRIGERANT UNIT USING NON-CONTAMINANT REFRIGERANTS

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ABSTRACT

A numerical simulation of the thermal and fluid-dynamic behaviour of a single stage vapor compression refrigerant unit has been developed and improved. The modelization consists of a main program that calls sequentially different subroutines until the convergence is reached. These subroutines modelize the physical flow phenomena produced inside a double-pipe condenser and evaporator, a capillary tube expansion device, a reciprocating compressor and the different connecting tubes between these four main elements. The software makes the analysis possible in transient or steady state of a wide range of situations, taking into account different working fluids, geometries and boundary conditions. All the flow variables (temperature, pressure, mass vapor fraction, void fraction, velocity, heat flux, etc...) are evaluated at each point of the grid in which the domain is discretized. An experimental unit has been designed and built to analyze single stage vapor compression refrigerating equipment. The objective of this experimental unit is to validate the numerical simulation of each component and the equipment as a whole. In addition to this validation, some illustrative results are shown using non-contaminant refrigerants.

INTRODUCTION

General and flexible methods to simulate the thermal and fluid-dynamic behaviour of vapor compression refrigerating systems are required for design and optimization purposes of these units in order to take into account different aspects such as: the specific geometric characteristics of each component of the system, the use of new non-contaminant refrigerants, consideration of thermal loadings, etc.

Several models have been presented in the literature. Among them the models developed by Chi and Didion [1], Murphy and Goldsmith [2], and Rajendran and Pate [3] are based on global balances between the inlet and outlet sections of the different elements of the system. McArthur [4] or Yuan and O'Neal [5] solved the condenser and the evaporator using finite difference methods and considering a one dimensional formulation of the governing equations.

The numerical simulation of a single stage refrigerating unit presented in this paper is an improvement of a modelization previously developed [6]. The unit studied consists of a double-pipe condenser and evaporator, a capillary expansion tube device, a reciprocating compressor, and the different connecting tubes. 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 (cf. [7][8]). This formulation requires the use of empirical information for the evaluation of the shear stress, convective heat transfer and void fraction. 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 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 compressors [9][10].

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 the main circuit and in the secondary circuits using sensors and control valves. The mass flow rate and inlet temperature are independently fixed in the secondary condenser and evaporator 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 seven different subroutines. These subroutines have been created to solve: i) the two-phase flow evaporation inside a tube; ii) the two-phase flow condensation inside a tube; iii) the fluid flow inside a capillary tube; iv) the single-phase fluid flow inside an annulus; v) the conduction heat transfer in a tube wall; vi) the heat transfer in a tube with insulator, and vii) the compression process. 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 each subroutine and the global algorithm to solve the complete refrigerating cycle is indicated below.

Two-phase flow evaporation and condensation inside a tube. The one-dimensional and transient governing equations of the fluid flow are integrated numerically using an implicit step-by-step numerical scheme. The empirical information needed is the convective heat transfer, the shear stresses, and the void fraction. Inflow conditions and wall temperature are taken as boundary conditions. From this subroutine the outflow conditions are calculated (output data). This numerical simulation is explained in detail in [7].

Fluid flow inside a capillary tube. There are no differences between this subroutine and the subroutine explained in [8]. 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.

Single-phase fluid flow inside an annulus. This subroutine can be considered as a particular case of the two-phase flow subroutines described above. The differences are that the heating or cooling fluid inside the annulus is water. The inlet flow conditions and the wall temperatures (both the inner and outer tubes) are taken as boundary conditions.

Conduction heat transfer in a tube wall. The one-dimensional heat conduction equation in the tube wall has been discretized on the basis of a central-difference numerical scheme and has been solved using a TDMA (Three-Diagonal Matrix Algorithm). This analysis allows the consideration of longitudinal conduction effects along the heat exchangers and the connecting tubes. The convection heat transfer exchanges with the fluid flow and the wall temperature in the tube ends are taken as boundary conditions.

Heat transfer in a tube with insulator. The heat conduction equation in the insulator has been discretized considering two-dimensional phenomena (in the longitudinal and radial directions), and using a central-difference numerical scheme. 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, heat losses and the power transfer to the system. This empirical information has been evaluated from an advanced simulation model [9] [10], which solves the one-dimensional and transient governing equations of the flow in the whole domain. The equations are discretized using an implicit control-volume formulation and a SIMPLE-like algorithm.

Global algorithm considering transient or steady state. The algorithm is organized in such a way that, 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.

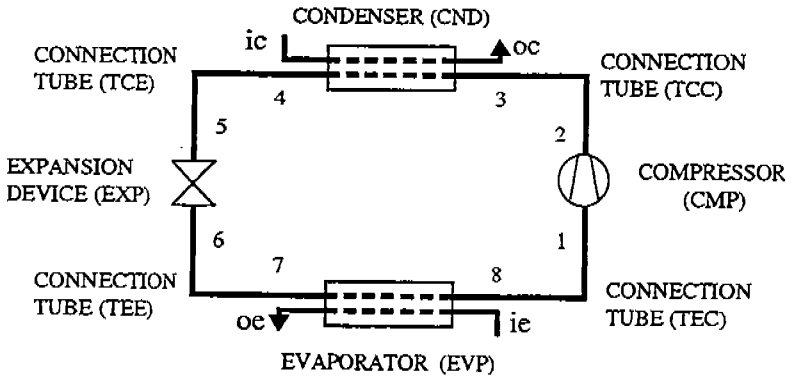

Figure 1 Single stage vapor compression refrigerant unit scheme.

In transient state, the information transfer scheme implemented is illustrated in Table 1. Using this scheme, 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.

Element	CMP	TCC	CND	TCE	EXP	TEE	EVP	TEC
Input data	$p_2, T_1, \dot{m}_1$	$p_3, T_2, \dot{m}_2$	$p_4, T_3, \dot{m}_3$	$p_5, T_4, \dot{m}_4$	$p_6, T_5, \dot{m}_5$	$p_7, x_{g6}, \dot{m}_6$	$p_8, x_{g7}, \dot{m}_7$	$p_1, T_8, \dot{m}_8$
Output data	$p_1, T_2, \dot{m}_2$	$p_2, T_3, \dot{m}_3$	$p_3, T_4, \dot{m}_4$	$p_4, T_5, \dot{m}_5$	$p_5, x_{g6}, \dot{m}_6$	$p_6, x_{g7}, \dot{m}_7$	$p_7, T_8, \dot{m}_8$	$p_8, T_1, \dot{m}_1$

Table 1. Information transfer scheme for the transient state algorithm.

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). The information transfer scheme implemented for steady state situations is illustrated in Table 2.

Element	CMP	TCC	CND	TCE	EXP	TEE	EVP	TEC
Input data	$p_2, T_1, \dot{m}$	$p_2, T_2, \dot{m}$	$p_3, T_3, \dot{m}$	$p_4, T_4, \dot{m}$	p_5, T_5, p_6	$p_7, x_{g6}, \dot{m}$	$p_8, x_{g7}, \dot{m}$	$p_1, T_8, \dot{m}$
Output data	p_1, T_2	p_3, T_3	p_4, T_4	p_5, T_5	$x_{g6}, \dot{m}$	p_6, x_{g7}	p_7, T_8	p_8, T_1

Table 2. Information transfer scheme for the steady state algorithm.

EXPERIMENTAL UNIT DESCRIPTION

An experimental unit has been built to study single stage vapor compression refrigerating systems. This unit has been designed to validate the mathematical model used in the numerical simulation. A schematic diagram of the experimental unit is shown in Figure 2. The geometric parameters of the equipment are commented on in the next section (RESULTS). The thermostatic cooling unit has a power ranging from 400 to 700 W and the thermostatic heating unit has a power of 3 kW. The fluid flow temperatures inside tubes and annuli are measured with calibrated platinum resistance thermometer sensors Pt-100. These sensors are located at the inlet and outlet sections of each element of the main circuit and of the secondary circuits. Condenser and evaporator pressures are measured by transducers, the accuracy is within ±0.1%. The mass flow rate inside the main circuit is measured with a coriolis type mass flowmeter, the accuracy is within ±0.2% (i.e. ±0.0009Kg/min). The mass flow rate in the secondary circuits is measured with magnetic flowmeters, the accuracy is within ±0.0125% (when mass flow rate ranges from 0 to 2.5 l/min) and within ±0.5% (when mass flowrate ranges from 2.5 to 25 l/min). The temperature in the auxiliary circuits is controlled by the thermostatic units and two modulating solenoid valves control the mass flow rate in these circuits. A Data Acquisition and Control Unit and a personal computer process all the information.

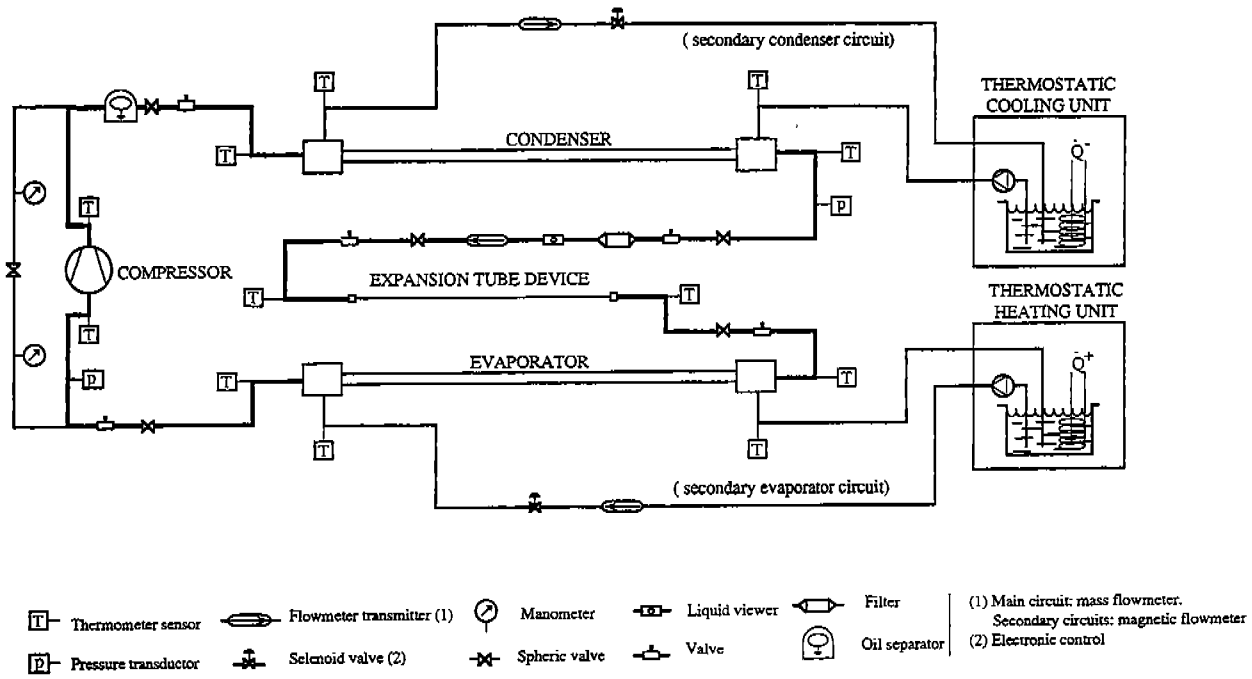

Figure 2. Schematic representation of the experimental unit.

RESULTS

In this section a comparison between numerical results and experimental data obtained in the unit described above (see Figure 2) is presented. 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 CFC-12 and two non-contaminant refrigerants: the HFC-134a and the R-290 (propane).

The situation analyzed for comparison between numerical and experimental results corresponds to the following values:

Fluid	R-12.
Compressor	GP12EB (Electrolux Compressor Companies); geometry parameters are given in [6].
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 = 34.94 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: $3.8 \cdot 10^{-3}$.
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, 58mm; length: 6m; smooth tube; counterflow.
	ii) annulus: water at Ti = 15.19 C; mass flow rate: 4 l/min.
Tube connection	i) geometry: diameters: 8, 9.6, 47.6 mm; length: 1.1 m; smooth tube.
Cycle	i) condensation pressure: 13.10 bar; steady state.

The results presented for the *compressor* have been carried out using the simple compressor model. However, the empirical information for this case has been obtained using the advanced simulation model [9][10]. Table 3 shows the comparative results. The differences in the mass flow rate are 8.5%.

results	T_1 (C)	T_2 (C)	p_1 (bar)	p_2 (bar)	$\dot{m}$ (Kg/h)	η_v	w (kJ/Kg)	q (kJ/Kg)
experimental	20.40	138.63	1.32	13.10	6.62	0.454	–	–
numerical	(20.40)	142.68	(1.32)	(13.10)	7.18	0.493	72.23	-0.01

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

Tables 4, 5 and 6 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 0.5%). The values T_{ic},T_{oc},T_{ie},T_{oe} are the inlet/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).

results	T_3 (C)	T_4 (C)	$\dot{m}$ (Kg/h)	$\dot{m}_{aux}$ (l/min)	T_{ic} (C)	T_{oc} (C)	p_{cond} (bar)
experimental	97.89	35.73	6.62	4.00	34.94	36.02	13.10
numerical	(97.89)	36.34	(6.62)	(4.00)	(34.94)	36.10	(13.10)

Table 4. Condenser: 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)	x_{g6}
experimental	28.65	-23.22	6.62	1.32	13.10	–
numerical	(28.65)	-23.40	6.65	(1.32)	(13.10)	0.301

Table 5. Capillary tube: experimental vs. numerical results (boundary conditions in brackets).

results	T_7 (C)	T_8 (C)	$\dot{m}$ (Kg/h)	$\dot{m}_{aux}$ (l/min)	T_{ie} (C)	T_{oe} (C)	p_{evap} (bar)	x_{g7}
experimental	-23.31	15.27	6.62	4.00	15.19	14.26	1.32	–
numerical	-23.40	15.15	(6.62)	(4.00)	(15.19)	14.28	(1.32)	0.301

Table 6. Evaporator: experimental vs. numerical results (boundary conditions in brackets).

For the complete refrigerating system a comparison between experimental and numerical results is presented in Table 7 (in this case only the inlet and outlet flow conditons in the secondary circuit and the condenser pressure are fixed as boundary conditions). The accuracy required has been $1 \cdot 10^{-6}$. A rasonable accordance between experimental and numerical results has also been obtained.

results	T_1 (C)	T_2 (C)	T_3 (C)	T_4 (C)	T_5 (C)	T_6 (C)	T_7 (C)	T_8 (C)	x_{g6}	x_{g7}	$\dot{m}$ (Kg/h)	p_e (bar)	p_c (bar)	T_{ic} (C)	T_{oc} (C)	T_{ie} (C)	T_{oe} (C)	$\dot{m}_{auxc}$ (l/min)	$\dot{m}_{auxe}$
exper.	20.40	138.63	97.89	35.73	28.65	-23.22	-23.31	15.27	–	–	6.62	1.32	13.10	34.94	36.02	15.19	14.26	4.00	4.00
numer.	15.40	138.64	114.06	36.48	31.05	-25.79	-25.81	15.16	0.327	0.357	6.45	1.17	(13.10)	(34.94)	36.14	(15.19)	14.35	(4.00)	(4.00)

Table 7. Cycle: experimental vs. numerical results (boundary conditions in brackets).

Finally, a comparison between three refrigerating fluids (CFC-12, HFC-134a and R-290) is presented. The situation analyzed is the same as the one indicated at the begining of this section with the exceptions of: i) compressor GL99EJ (Electrolux Compressor Companies), a detailed description of this compressor is given in [10]; ii) the length of the condenser is 3 m; iii) in order to obtain a similar evaporation and condensation temperatures (-23.3 C and 55 C respectively) a capillary tube length of 1.06 m, 2.2 m and 1.15 m has been fixed for CFC-12, HCF-134a and R-290 respectively. Table 8 and Figure 3 show some global results of the system working with these fluids (for the R-290 thermodynamical properties as ideal gas have been assumed).

fluid	p_{cond}/p_{evap}	$\dot{W}_{cp}$ (W)	$\dot{Q}_{evap}$ (W)	$\dot{Q}_{cond}$ (W)	$\dot{m}$ (Kg/h)	η_v	COP
R-12	10.158	160.5	-253.0	329.8	7.164	0.581	1.58
R-134a	13.673	128.9	-218.3	362.9	4.835	0.540	1.69
R-134a	8.734	242.9	-377.4	625.9	4.954	0.699	1.55

Table 8. Numerical illustrative results for the three refrigerants (R-12, R-134a and R-290).

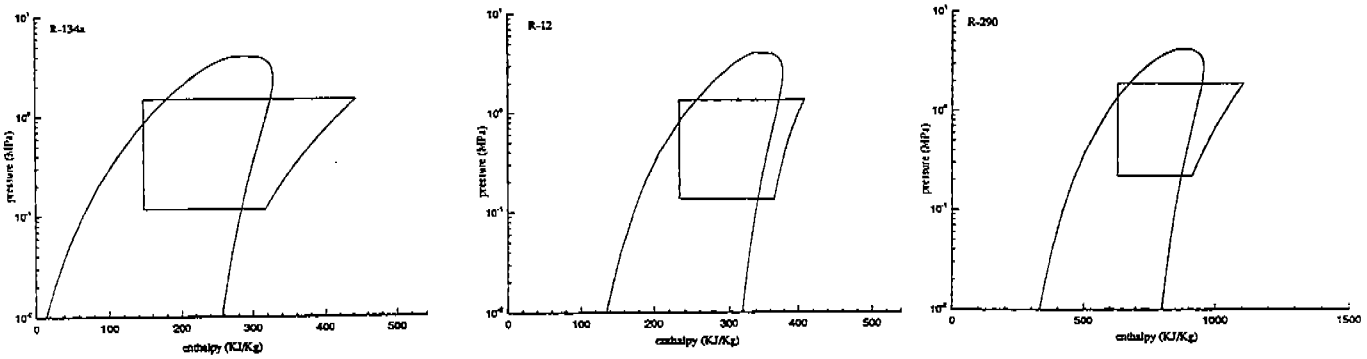

Figure 3. Pressure vs. enthalpy for R-12, R-134a and R-290.

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. For the compressor, the empirical information needed (volumetric efficiency, heat losses and compressor power consumption) has been obtained from an advanced simulation model. The global simulation solves sequentially the different elements of the system (compressor, condenser, expansion device, evaporator and connecting tubes) 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 has also been presented.

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REFERENCES

[1] J.Chi and D.Didion, A Simulation model of the transient performance of a heat pump, Int. J. of Refrigeration, vol.5, no.3, pp.176-184, 1982.

[2] W.E.Murphy and V.W.Goldschmidt, Cyclic characteristics of a typical residential air conditioner-modeling of start-up transients, ASHRAE Transactions, vol.91, no.2, pp.427-444, 1985.

[3] N.Rajendran and M.B.Pate, A computer model of the start-up transients in a vapor compression refrigeration system, Progress in the Design and Construction of Refrigeration Systems, ed. by D.R.Tree, pp.129-140, 1986.

[4] J.W.McArthur, Transient heat pump behaviour: a theoretical investigation, Int. J. of Refrigeration, vol.7, no.2, pp.123-132, 1984.

[5] X.Yuan and D.L.O'Neal, Development of a transient simulation model of a freezer part I: Model development, 1994 International Refrigeration Conference at Purdue, pp.213-224.

[6] F.Escanes, A.Oliva, C.D.Pérez-Segarra and J.Flores, Numerical simulation of a single stage vapor compression refrigerating unit, 1994 International Compressor Engineering Conference at Purdue, 1, pp.139-144, 1994.

[7] F.Escanes, C.D.Pérez-Segarra and A.Oliva, Thermal and fluid-dynamic behaviour of double-pipe condensers and evaporators. A numerical study, Int. J. of Num. Meth. for Heat and Fluid Flow, vol.5,no.9, pp.781-795, 1995.

[8] F.Escanes, C.D.Pérez-Segarra and A.Oliva, Numerical simulation of capillary-tube expansion devices, Int. J. of Refrigeration, vol.18, no.2, pp.113-122, 1995.

[9] C.D.Pérez-Segarra, F.Escanes and A.Oliva, Numerical study of the thermal and fluid-dynamic behaviour of reciprocating compressors, 1994 International Compressor Engineering Conference at Purdue, pp.145-150.

[10] F.Escanes, C.D.Pérez-Segarra, J.Rigola, Numerical simulation of reciprocating compressors. Recent improvements and experimental validation, to be presented at the 1996 International Compressors Engineering Conference at Purdue.