2004

Numerical Study and Experimental Validation of a Transcritical Carbon Dioxide Refrigerating Cycle

Joaquim Rigola
Universitat Politecnica de Catalunya

Sergio Morales
Universitat Politecnica de Catalunya

Gustavo Raush
Universitat Politecnica de Catalunya

Carlos C. Perez Segarra
Universitat Politecnica de Catalunya

Nicolas Ablanque
Universitat Politecnica de Catalunya

NUMERICAL STUDY AND EXPERIMENTAL VALIDATION OF A TRANSCRITICAL CARBON DIOXIDE REFRIGERATING CYCLE

Joaquim RIGOLA, Sergio MORALES, Gustavo RAUSH, Carlos. D. PÉREZ-SEGARRA, Nicolas ABLANQUE

Centre Tecnològic de Transferència de Calor CTTC,
Laboratori de Termotècnia i Energètica,
Dept. Màquines i Motors Tèrmics. Universitat Politècnica de Catalunya,
Colom 11, 08222 Terrassa (Barcelona), Spain
Fax: +34-93-739.89.20, Tel. +34-93-739.81.92
E-mail: labtie@labtie.mmt.upc.es, Web page: http://www.cttc.upc.edu

ABSTRACT

The present work is a numerical study of a transcritical carbon dioxide refrigerating cycle. The heat exchangers in particular, and the whole cycle in general, are simulated under a carbon dioxide refrigerating system. The experimental unit, specially designed to be used with carbon dioxide as refrigerant fluid, has allowed validate the numerical results obtained. A detailed numerical simulation of the thermal and fluid-dynamic behavior of a single stage vapour compression refrigerating unit has been developed. The modelization consists of a main program that sequentially calls different subroutines until convergence is reached. The different subroutines modelize the physical flow phenomena produced inside all the different elements that make up the compression cycle. The governing equations of the flow in the heat exchangers are solved by means of the Step by Step method. Some empirical information is required for the evaluation of the convective heat transfer, shear stress and void fraction. The compressor model is obtained using an advanced hermetic compressor numerical simulation model. An expansion valve device is modeled by means of the characteristic parameters. The software allows the analysis, in transient or steady state, of a wide range of situations, taking into account different working fluids, geometries and boundary conditions. The software gives the flow variables distribution, which are evaluated at each point of the discretized domain.

1. INTRODUCTION

The common refrigerants, CFCs and HCFCs, produce negative effects on the environment (UNEP, 1987) and (UNFCCC, 1997). The environmental problems caused by some refrigerants need a solution. Thus, the search, investigation and utilization of new substitutes is an important goal for researchers. During the last decade investigations indicated that carbon dioxide has an important interest as a natural refrigeration fluid (Lorentzen, 1994), (Jakobsen, 1998), (Kruse et al., 1999) and (Fleming, 2003). For these reasons, the obtaining of general and flexible design methods is very important in the applications and the optimization of refrigeration cycles 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.

A numerical simulation of the thermal and fluid-dynamic behavior of a single stage vapour compression refrigerating unit has been developed and improved (Rigola et al., 1998). The unit studied consists of a double-pipe gas cooler and evaporator, an expansion device, and a reciprocating compressor. Each element is simulated by means of a group of subroutines. The modelization consists of a main program that sequentially calls different subroutines until convergence is reached. These subroutines have been created to solve: i) the two-phase flow evaporation inside a tube (Escanes et al., 1995) and (García-Valladares et al.,In Press); ii) the two-phase flow condensation inside a tube; iii) the fluid flow inside an expansion device; iv) the single-phase fluid flow inside an annulus; v) the conduction heat transfer in a tube wall; and vi) the reciprocating compressor.

The compressor is modelized considering cyclic conditions on the basis of global mass and energy balances in the whole system. This compressor model also needs some empirical information: the volumetric efficiency, heat losses and input power transferred to the gas. They are obtained by means of the advanced simulation model (Pérez-Segarra et al., 2003), specially adapted to be used with carbon dioxide (Rigola et al., 2004). The advanced simulation model solves the one-dimensional and transient governing equation of the flow in the whole domain. The equations are discretized using an implicit control volume formulation and a SIMPLE-like algorithm.

The heat exchangers are solved on the basis of a control volume formulation of the governing equations (continuity, momentum and energy), considering one-dimensional flow. The formulation requires the use of empirical information for the evaluation of shear stress, convective heat transfer and void fraction. Most of the empirical correlations implemented for carbon dioxide are referenced in (Ortiz, 2002). The longitudinal conduction in heat exchangers and connecting tubes has also been considered. The expansion valve device has been modelized by means of the characteristic valve parameters. The thermodynamic and transport properties database has been implemented to work with carbon dioxide as a refrigerant (NIST, 1996).

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 the secondary circuits. The mass flow rate and the inlet temperature are independently fixed in the condenser and evaporator secondary circuits. The refrigerant mass flow rate depends on the expansion device opening position. Once this has been carried out, the rest of the results are a consequence (in the experimental unit and numerical simulation).

The following aspects to be presented in this work are: i) a comparison for each element and for the whole system between experimental and numerical results for CO_2; ii) a study of the thermal behavior of the transcritical carbon dioxide refrigerating unit. Significant results, i.e. the pressure-enthalpy diagram and temperature-entropy diagram, will be presented.

2. 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 duct control volume, the single phase flow inside a characteristic annular duct control volume and the conduction heat transfer through a characteristic tube control volume. The compressor process is formulated based on mass and energy balances between inlet and outlet conditions in the hermetic compressor. The different elements of the equipment (evaporator, compressor, gas cooler and expansion device) are solved by means of the mentioned subroutines called in a convenient way.

2.1 Heat exchangers

The numerical simulation is explained in detail in (Escanes et al., 1995) and (García-Valladares et al.,In Press). The one-dimensional and transient governing equations of the fluid flow (continuity, momentum and energy) are integrated numerically using a fully implicit scheme. The Step by Step numerical method is used as a solver. The empirical information, i.e. the convective heat transfer, shear stresses, and the void fraction, is needed in governing equations. The correlations implemented for carbon dioxide in the gas cooler and the evaporator are taken from (Bennett and Chen, 1980), (Hwang et al., 1997), (Pitla et al., 1998), (Pitla et al., 2002), (Ortiz, 2002) and (Yoon et al., 2003). All the flow variables (temperature, pressure, quality, void fraction, mass flux, heat flux, enthalpy etc.) are evaluated at each point of the grid at which the domain is discretized. Depending on the case, inflow and/or outflow conditions, and/or wall temperatures are taken as boundary conditions. The governing equations of the flow have been integrated for pressure, mass flux and enthalpy. This subroutine has solved the two phase flow inside the inner duct, where carbon dioxide flows, and the single phase flow in the annular duct, where a counter current water flows.

2.2 Conduction heat transfer through the tubes

The one-dimensional heat conduction equation in the solid element has been discretized considering one-dimensional phenomena (in the longitudinal directions), on the basis of a central-difference numerical scheme. The set of discretized equations has been solved using a line by line algorithm. The fluid temperature and local heat transfer coefficient distribution (inside the tube) are taken as result of the heat exchanger subroutine.

2.3 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 (Pérez-Segarra et al., 2003) and experimental validated (Rigola et al., 2003), specially adapted to be used with carbon dioxide (Rigola et al., 2004), which solves the thermal and fluid-dynamic behavior 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 behavior is based on heat global balances at each solid component. The subroutine for the compressor process uses the results obtained from this modelization. These results are the efficiencies vs compression ratio relations.

2.4 Expansion device

The expansion device used in the refrigerating cycle is a commercial valve designed to provide accurate and stable control of flow rates in analytical and research applications. The numerical model of the valve is based on coefficient of flow simulations criteria. This model find a coefficient of flow Cv as function of the number of turns open of the valve. The Cv value can be used to determine flows through a valve. The reference of the valve is shown in Table 1.

2.5 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 convergence is reached. 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 gas cooler and evaporator, the compressor speed, the ambient temperature and pressure, and the opening position in the valve. 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.

In transient state, the pressure input data in the subroutines of the gas cooler 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. Thus, 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).

3. EXPERIMENTAL UNIT DESCRIPTION

An experimental unit has been presented and explained in detail in (Rigola et al., 2004). The geometric parameters of the equipment and the measurement instruments have also been explained in the reference mentioned above. This unit has been built to study single stage vapour 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 1.

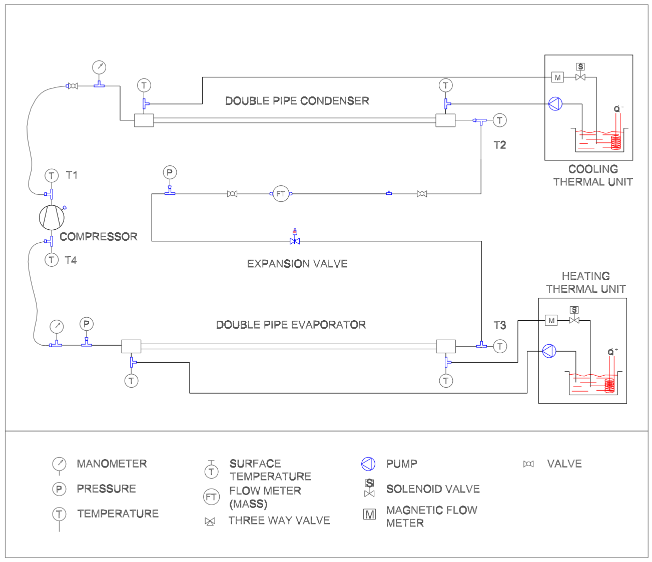

Figure 1: Schematic representation of the experimental unit.

Figure 2 shows a top view of the refrigeration system where the compressor, gas cooler, evaporator and expansion device are shown. Figure 3 depicts a controlling and acquisition data system.

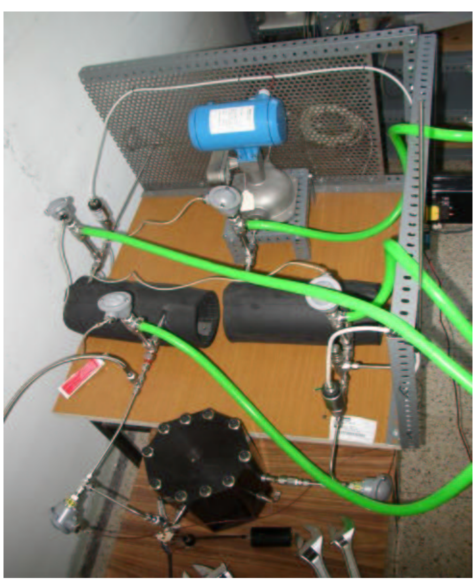

Figure 2: View of the refrigeration system.

Figure 3: View of the controlling and acquisition data system.

4. NUMERICAL RESULTS

In this section a comparison between numerical results and experimental data (obtained from the test facility described above in Figure 1 and using carbon dioxide as the refrigeration fluid) is presented. The comparison is made for each element of the system (compressor, double tube gas cooler, expansion device and double tube evaporator) and for the whole system.

Three different experimental cases for the whole cycle are presented, and their corresponding boundary conditions are shown in Table 1.

Table 1: Parameters of the experimental tests.

Fluid	Carbon dioxide
Compressor	parameters: V_{cl}: 1.5e-6 m^3; f_n: 50 Hz; V_m: 4.5e-8 m^3
Gas cooler	geometry: diameters: 3.86e-3, 6.35e-3, 1.021e-2 m; length: 4.5 m; smooth tube; counterflow
Expansion device	4A-NSL Parker Instrumentation, metering valve
Evaporator	geometry: diameters: 3.86e-3, 6.35e-3, 1.021e-2 m; length: 4.5 m; smooth tube; counterflow
Case a	
Gas cooler	annulus: water at 25.92 C; mass flow rate: 0.034500 Kg/s
Evaporator	annulus: water at 24.73 C; mass flow rate: 0.026666 Kg/s
Cycle	condensation pressure: 10.263 MPa; steady state
Case b	
Gas cooler	annulus: water at 25.91 C; mass flow rate: 0.034167 Kg/s
Evaporator	annulus: water at 24.73 C; mass flow rate: 0.027500 Kg/s
Cycle	condensation pressure: 10.264 MPa; steady state
Case c	
Gas cooler	annulus: water at 25.87 C; mass flow rate: 0.03533 Kg/s
Evaporator	annulus: water at 24.70 C; mass flow rate: 0.02833 Kg/s
Cycle	condensation pressure: 10.131 MPa; steady state

The empirical information of the compressor for all the numerical results has been evaluated by means of the advanced compressor model (Rigola et al., 2004). The results obtained for the studied cases are presented in Figure 4 (the electrical/mechanical efficiency is nearly constant in all the studied cases and has a value of approximately 75%-76%).

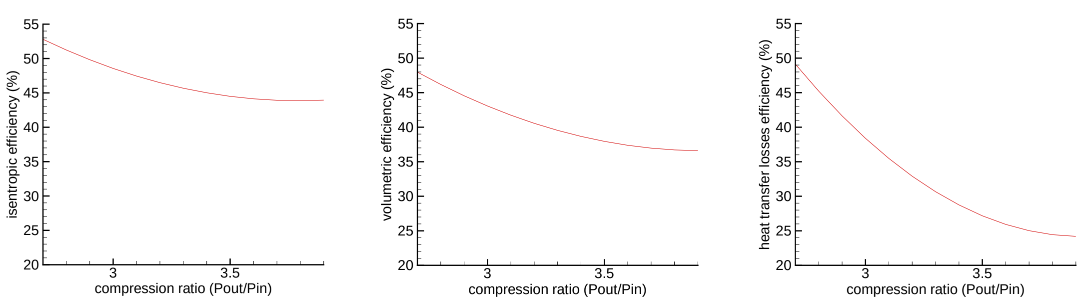

Figure 4: Isentropic, volumetric and heat-transfer-losses efficiencies vs. compression ratio.

In Table 2 some comparative results for the simulation of the compressor alone are shown. A high level of accuracy is reached.

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

results	T_{in} (C)	T_{out} (C)	P_{in} (MPa)	P_{out} (MPa)	$\dot{m}$ (Kg/s)
experimental	24.28	119.44	2.6610	10.2630	0.001528
numerical	(24.28)	119.47	2.6615	(10.2630)	(1.001528)

Comparative results between numerical simulation and experimental data for the gascooler are shown in Table 3. Values T_{ig} and T_{og} are the inlet and outlet temperatures in the auxiliary gascooler circuit, and its mass flow rate is $\dot{m}_{aux}$. As shown, good agreement is obtained (e.g. outlet secondary fluid temperature prediction is within 1.1%). Similar evaluations were made for the evaporator and the expansion device with the same grade of accuracy.

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

results	T_{in} (C)	T_{out} (C)	P_{in} (MPa)	P_{out} (MPa)	$\dot{m}$ (Kg/s)	$\dot{m}_{aux}$ (Kg/s)	T_{ig} (C)	T_{og} (C)
experimental	119.44	25.31	10.2630	10.2630	0.001528	0.0345	25.92	28.55
numerical	(119.44)	25.92	(10.2630)	10.2625	(0.001528)	(0.0345)	(25.92)	28.86

For the complete refrigerating system, and for the three cases described above (see Table 2), a comparison between experimental data and numerical results is presented in Table 4 (in only those cases the inlet flow conditions of the secondary circuits, the gas cooler pressure and the mass flow rate are fixed as boundary conditions). A reasonable accordance between experimental and numerical results has also been obtained. For all variables compared, except for the inlet temperature in the evaporator, the numerical prediction is within 3% of the experimental data. However the largest temperature difference at the evaporator entrance is relatively small (0.61^{o}C for T_3 in Case c).

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

results	P_{evap} (Mpa)	P_{gc} (MPa)	T_1 (C)	T_2 (C)	T_3 (C)	T_4 (C)	Xg_3	$\dot{m}$ (Kg/s)
Case a								
experimental	2.6610	10.2630	119.44	25.31	−10.04	24.28	0.308	0.001528
numerical	2.6681	(10.2630)	119.71	25.92	−9.69	24.54	0.314	(0.001528)
Case b								
experimental	3.0020	10.2640	116.41	25.38	−5.72	24.21	0.283	0.001862
numerical	3.0120	(10.2640)	116.78	25.91	−5.36	24.53	0.288	(0.001862)
Case c								
experimental	3.5560	10.1310	113.26	25.41	0.49	24.18	0.242	0.002723
numerical	3.5836	(10.1310)	113.42	25.87	1.10	24.42	0.245	(0.002723)

Finally, global results are depicted in Figures 5 and 6. The pressure vs. enthalpy and temperature vs. entropy diagram of the system working with carbon dioxide are shown.

Global results have shown the behavior of the transcritical carbon dioxide refrigerating cycle and, in general, how the system works at supercritical condition.

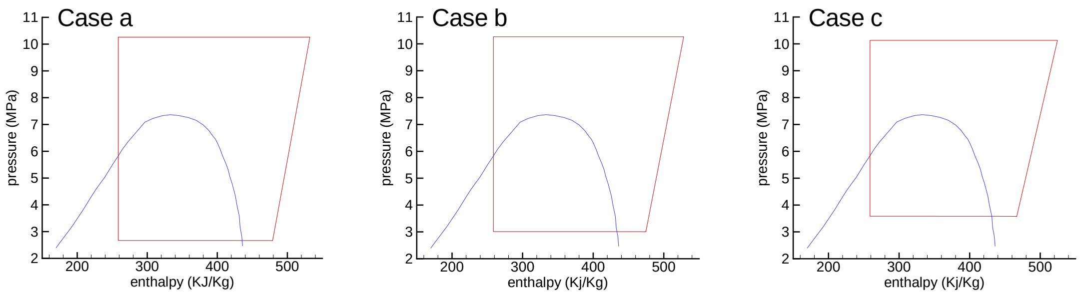

Figure 5: Pressure vs. enthalpy diagram

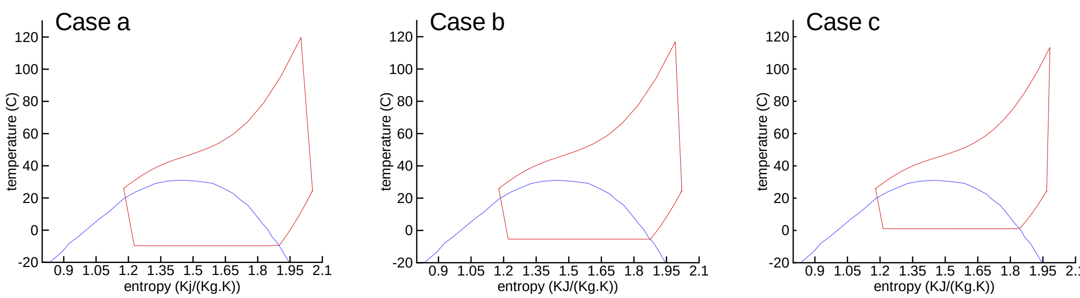

Figure 6: Temperature vs. entropy diagram

5. CONCLUSIONS

A detailed numerical simulation of the thermal and fluid-dynamic behavior of a vapour 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 sequentially solves the different elements of the system (compressor, gas cooler, expansion device, and evaporator) until convergence is reached. Good agreement between numerical results and experimental data has been obtained. Finally, some illustrative results comparing three different cases working with carbon dioxide as refrigerant are shown.

REFERENCES

Bennett, D. and Chen, J., 1980, Forced Convective Boiling in Vertical Tubes for Saturated Pure Components and Binary Mixture, *AIChE Journal*, 26:p. 454–462.

Escanes, F., Pérez-Segarra, C., and Oliva, A., 1995, Thermal and Fluid-Dynamic Behaviour of Double-Pipe Condenser and Evaporators, A Numerical Study, *International Journal of Numerical Methods for Heat and Fluid Flow*, 5, no. 9:p. 781–795.

Fleming, J., 2003, Carbon dioxide as the working fluid in heating and/or cooling systems, *Bulletin of the International Institute of Refrigeration*, 83, no. 4:p. 7–15.

García-Valladares, O., Pérez-Segarra, C., and Rigola, J., In Press, Numerical Simulation od Doble-pipe Condesers and Evaporators, *International Journal of Refrigeration*.

Hwang, Y., Kim, B., and Radermacher, 1997. Boiling Heat Transfer Correlation for Carbon Dioxide. Technical report, IIF-IIR - Commission B1, College Park, USA.

Jakobsen, A., 1998, Improving efficiency of trans-critical CO_2 refrigeration systems for reefers, *IIF-IIR comission D2/3*, Cambridge, UK, p. 130–138.

Kruse, H., Heidelck, R., and Süss, J., 1999, The application of CO_2 as a Refrigerant, *Bulletin of the International Institute of Refrigeration*, 79, no. 1:p. 2–21.

Lorentzen, G., 1994, Revival of carbon dioxide as a refrigerant, *International Journal of Refrigeration*, 17, no. 5:p. 292–301.

National Institute of Standards and Technology, 1996. REFPROP v.5.0, NIST Thermodynamic Properties of Refrigerants and Refrigerant Mixtures Database. Gaithersburg, MD, USA.

United Nations Framework Convention on Climate Change UNFCCC, 1997. Kyoto Protocol.

Ortiz, T., 2002. *Deveopment of a New Model for Investigation of the Performance of Carbon Dioxide as a Refrigerant for Residential air conditioners*. PhD thesis, Purdue University, In USA.

Pitla, S., Groll, E., and Ramadhyani, S., 2002, New Correlation to Predict the Heat Transfer Coefficient During In-Tube Cooling of Turbulent Supercritical CO_2, *International Journal of Refrigeration*, 25:p. 887–895.

Pitla, S., Robinson, D., Groll, E., and Ramadhyani, S., 1998, Heat Transfer from Supercritical Carbon Dioxide in the Flow: A Critical Review, *International Journal of Heating, Ventilating, Air-Conditioning and Refrigerating Research*, 4, no. 3:p. 281–301.

Pérez-Segarra, C., Rigola, J., and Oliva, A., 2003, Modeling and numerical simulation of the thermal and fluid dynamic behavior of hermetic reciprocating compressors. Part 1: Theoretical basis, *International Journal of Heating, Ventilating, Air-Conditioning and Refrigerating Research*, 9, no. 2:p. 215–236.

Rigola, J., Pérez-Segarra, C., García-Valladares, O., Serra, J., Escribà, M., and Pons, J., 1998, Numerical study and experimental validation of a complete vapour compression refrigerating cycle, *International Compressor Engineering Conference*, Purdue University, IN, USA, p. 201–206.

Rigola, J., Pérez-Segarra, C., and Oliva, A., 2003, Modeling and numerical simulation of the thermal and fluid dynamic behavior of hermetic reciprocating compressors. Part 2: Experimental Investigation, *International Journal of Heating, Ventilating, Air-Conditioning and Refrigerating Research*, 9, no. 2:p. 237–249.

Rigola, J., Pérez-Segarra, C., and Oliva, A., 2004, Thermal and Fluid Behaviour of Transcritical Carbon Dioxide Hermetic Reciprocating Compressors: Numerical analysis and Parametric Study, *International Compressor Engineering Conference*, Purdue University, In USA.

United Nations Environmental Programme UNEP, 1987. Montreal Protocol on Substances that Deplete the Ozone Layer.

Yoon, S., Kim, J., Y., H., Kim, M., Min, K., and Kim, Y., 2003, Heat Transfer and Pressure Drop Characteristics During the In-Tube Cooling Process of Carbon Dioxide in the Supercritical Region, *International Journal of Refrigeration*, 26:p. 857–864.

ACKNOWLEDGEMENTS

The authors gratefully acknowledge the financial support provided by CUBIGEL, S.A. - División Unidad Hermética Spain (ref. no. C-04749) and by the Comisión Interministerial de Ciencia y Tecnología (ref. n^o. TIC2003-07970). We would also like to acknowledge the technical support of the R&D group of Unidad Hermética.