COMPARATIVE ANALYSIS OF R134a SUB-CRITICAL CYCLE VS. CO_2 TRANS-CRITICAL CYCLE. NUMERICAL STUDY AND EXPERIMENTAL COMPARISON.

J. Rigola, G. Raush, N. Ablanque, C.D. Pérez-Segarra, A. Oliva

Centre Tecnològic de Transferència de Calor (CTTC); Laboratori de Termotècnia i Energètica
Universitat Politècnica de Catalunya (UPC), C/ Colom 11, 08222 Terrassa (Barcelona), Spain

J.M. Serra, M. Escribà, J. Pons, J. Jover

CUBIGEL, S.A. - DIVISIÓN UNIDAD HERMÉTICA ACC Compressors
C/ Antoni Forrellat, s/n, 08912 Sant Quirze del Vallès (Barcelona), Spain

ABSTRACT

The present work is a numerical and experimental comparative study of the whole refrigerating cycles in general, and the hermetic reciprocating compressors in particular, between an R134a conventional refrigerating cycle and a carbon dioxide trans-critical refrigerating system. The comparative case presented has been specially designed for small cooling capacities and an evaporation temperature around 0^oC. A detailed numerical simulation model for hermetic reciprocating compressors performance has been used to numerically compare both a conventional R134a hermetic compressor and a first carbon dioxide compressor prototype. The conventional compressor has been experimentally tested in a calorimeter setup, while the CO_2 compressor has been validated in a specific experimental unit specially designed and built to analyze high pressure single stage vapour compression trans-critical refrigerating equipments. In order to validate this laboratory setup, a detailed numerical simulation of the thermal and fluid-dynamic behaviour of single stage vapour compression refrigerating unit has been developed and used. The numerical and experimental results obtained shows a good agreement, while the comparative globals values show very similar efficiencies under the same working conditions.

1. INTRODUCTION

The Montreal protocol [1] stipulated the phasing out of CFCs and HCFCs as refrigerants that deplete the ozone layer (ODP); while the Kyoto protocol [2] encouraged promotion of policies for sustainable development and reduction of global warming potential (GWP), including the regulation of HFCs, which have a high GWP. Thus, investigation and use of new and natural refrigerants is an important goal. During the last decade, the investigation indicates that carbon dioxide has an important interest as a natural fluid refrigerant [3][4][5][6], although there is no available commercial hermetic reciprocating compressors with carbon dioxide working under trans-critical conditions in single stage vapour compression refrigerating cycles.

The present work is a numerical and experimental comparative study of the compressors in particular and the whole cycle in general, under a R134a conventional cycle and carbon dioxide trans-critical refrigerating system. Two experimental units are presented in order to show an experimental comparative study. The first one is a conventional calorimeter setup to test hermetic reciprocating compressors working under common fluid refrigerants like R134a, R600a, R404A, etc. [7]. The second one is a specific laboratory setup that has been specially designed to be used with carbon dioxide as fluid refrigerant allowing the validation of the numerical results obtained [8].

A detailed numerical simulation of the thermal and fluid dynamic behaviour of hermetic reciprocating compressors commonly used in household refrigerators and freezers has been developed [9], numerically verified and experimentally validated [7]. The model has allowed to numerically compare the experimental results of the R134a compressor in the calorimeter test, and validate the results of the CO_2 compressor prototype under the single stage trans-critical vapour compression refrigerating unit developed.

A detailed numerical simulation of the thermal and fluid dynamic behaviour of a single stage vapour compression refrigerating unit has been developed [10], improved and specially adapted for carbon dioxide [11]. The modelization consists of a main program that sequentially calls different subroutines until the convergence is reached: i) four subroutines modelize the physical phenomena produced inside a double-pipe condenser and evaporator [12], [13]; ii) a subroutine simulates the phenomena for a capillary tube expansion device [14], [15]; iii) another subroutine carries out the reciprocating compressor behaviour solved as a simple model, although the global values are obtained from the advanced numerical simulation model cited above, and specially adapted to be used with carbon dioxide [16][17]; iv) finally, two more subroutines solve the different connecting tubes between the four main elements.

The aim of this paper is to present a comparative numerical and experimental study between two refrigerating cycles under similar conditions although considering a sub-critical R134a cycle vs. trans-critical, CO_2 system. Illustrative comparative results of both compressors performance are presented. In this case, both numerical and experimental cases present a good agreement for evaporation temperatures around 0^oC. A comparative results of the whole carbon dioxide cycle is also shown in order to validate the global cycle numerical simulation model when carbon dioxide is used as fluid refrigerant. The experimental validation under the correlations used for gas cooler and evaporator when carbon dioxide is used has presented a very good agreement. The final results have also shown the promising improvements of carbon dioxide instead of R134a for small capacity refrigerating systems.

2. NUMERICAL SIMULATION MODEL OF HERMETIC RECIPROCATING COMPRESSORS

The numerical simulation model 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 extended to compressible flow. The complete set of discretized momentum, energy and pressure correction equations is solved by the direct method TDMA (Tri-Diagonal Matrix Algorithm). Parallel circuits and extra elements (double orifices, resonators, etc.) are also considered in the formulation. The motor torque equation system is linearly independent, thus is solved directly by means of the inverse matrix system LU resolution. Macro-volumes energy balances are also directly solved in the same way. The results obtained are the volumetric efficiency, isentropic efficiency and heat transfer losses efficiency vs. compression ratio relations, in order to obtain a general parametrization used in the global simulation cycle.

The theoretical basis of the numerical simulation model summarized above for hermetic reciprocating compressors is described in detail in [9], while the accuracy of the mathematical model developed and an extensive experimental validation is shown in [7].

3. NUMERICAL SIMULATION MODEL OF SINGLE STAGE VAPOR COMPRESSION REFRIGERATING UNITS

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 single phase and two-phase flow inside a characteristic duct control volume, together with the conduction heat transfer along solid tube control volume. The different elements of the equipment (evaporator, compressor, gas cooler or condenser, and expansion device) are solved by means of the mentioned subroutines called in a convenient way.

3.1 Compressor process

The possibility to include the numerical simulation model of the compressor in the whole cycle resolution would increase the CPU time needed to solve the cycle significantly. In this case, 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 information is obtained and parametrized from the advanced simulation model referred to above, and specially adapted to be used with carbon dioxide [16][8].

3.2 Heat exchangers

The numerical simulation of the double-pipe heat exchangers (condensers, evaporators or gas coolers) is explained in detail in [12][13]. The one-dimensional and transient governing equations of the fluid flow (continuity, momentum and energy) are numerically integrated using a fully implicit step by step numerical scheme. The empirical information needed in the governing equations are the convective heat transfer, shear stresses, and the void fraction. The general correlations implemented and adapted for condensers and evaporators when conventional fluid refrigerants are used are detailed in [11], while the correlations selected when carbon dioxide fluid is considered are described in [17]. The correlations used in the case of CO_2 have been taken from [18], [19], [20] and [21].

All the flow variables 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 are also used to solve the single phase flow in the annular duct, where a counter current water flows and a double-pipe is considered.

3.3 Expansion device

The numerical simulation model allows the use of capillary tube expansion device between the condenser and the evaporator [14],[15] . The numerical resolution fo the two-phase flow is solved like the two-phase in the heat exchangers, although taking into account that in this case, all the inflow conditions cannot be simultaneously input data, as the critical mass flow rate is fixed for a given capillary tube. 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.

In spite of this, the cases herewith presented have used a commercial valve as expansion device, selected to provide an accurate desired control flow rates. In this case, the numerical model is based on consider the flow through the valve as a sudden contraction along the tube. The contraction of the tube is function of the C_v which represents the coefficient of flow, obtained from the characteristic behaviour of the valve. The C_v is function of the number of turns open of the valve.

3.4 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 a result of the heat exchanger subroutine.

3.5 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. In the 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 the 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, being n 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 and in this case the outlet compressor pressure has been chosen.

Table 1 shows the information transfer scheme for the steady state algorithm. The different elements are: compressor (CMP), condenser or gas cooler (CND), expansion device (EXP) and evaporator (EVP) together with all connecting tubes. Thus, 8 points represent the inlet/outlet cross sections of each element. Table 2 shows the information transfer scheme for the transient state algorithm.

Table 1: Information transfer scheme for the steady state algorithm.

element	CMP	TCC	CND	TCE	EXP	TEE	EVP	TEC
input data	$p_2, h_1, \dot{m}$	$p_2, h_2, \dot{m}$	$p_3, h_3, \dot{m}$	$p_4, h_4, \dot{m}$	p_5, h_5, p_6	$p_7, h_6, \dot{m}$	$p_8, h_7, \dot{m}$	$p_1, h_8, \dot{m}$
output data	p_1, h_2	p_3, h_3	p_4, h_4	p_5, h_5	$h_6, \dot{m}$	p_6, h_7	p_7, h_8	p_8, h_1

Table 2: Information transfer scheme for the transient state algorithm.

element	CMP	TCC	CND	TCE	EXP	TEE	EVP	TEC
input data	$p_2, h_1, \dot{m}_1$	$p_3, h_2, \dot{m}_2$	$p_4, h_3, \dot{m}_3$	$p_5, h_4, \dot{m}_4$	$p_6, h_5, \dot{m}_5$	$p_7, h_6, \dot{m}_6$	$p_8, h_7, \dot{m}_7$	$p_1, h_8, \dot{m}_8$
output data	$p_1, h_2, \dot{m}_2$	$p_2, h_3, \dot{m}_3$	$p_3, h_4, \dot{m}_4$	$p_4, h_5, \dot{m}_5$	$p_5, h_6, \dot{m}_6$	$p_6, h_7, \dot{m}_7$	$p_7, h_8, \dot{m}_8$	$p_8, h_1, \dot{m}_1$

4. EXPERIMENTAL SETUP DESCRIPTION

Experimental data have been obtained in two different setups. The first, hereafter referred to as the calorimeter test, uses the CUBIGEL company standard experimental units that are designed to obtain the main global parameters. The second, hereafter referred to as the CO_2 laboratory setup is a specially designed setup by the CTTC Research Center to be used under trans-critical carbon dioxide fluid conditions.

4.1 Calorimeter setup

The calorimeter setup, constructed according to ISO 917, consists mainly of a single-stage vapor compression unit. The evaporator is suspended in the upper part of a pressure-tight heat-insulated vessel. A heater in the base of this vessel is charged with a volatile secondary fluid. An expansion valve controls refrigerant flow into the compressor unit.

The secondary fluid that flows through the condenser is water. Different thermocouples with a precision of $\pm 0.2^o$C and a standard deviation of $\pm 0.3^o$C measure the fluid flow temperature at each inlet and outlet refrigerating system cross section and evaporator secondary fluid. Pressure regulation at the inlet and outlet compression sections is maintained below $\pm$5mbar and$\pm$1bar, respectively. Pressure transducers precision guarantees $\pm$0.2% F.S. Calorimeter test ISO 917 regulations guarantees mass flow rate and refrigerating capacity within $\pm$3%, and compressor power accuracy within $\pm$1% of values measured.

4.2 Laboratory setup

An experimental unit specially designed to test carbon dioxide trans-critical cycles has been built. Figures 1 and 2 shows a schematic diagram and a general view of the refrigeration system. The experimental unit are made up of the following elements: a carbon dioxide hermetic reciprocating compressor prototype, one dual heat transfer coil gas cooler and evaporator together with a metering valve. The auxiliary fluid used in the gas cooler and the evaporator annuli is water.

Table 3 shows the cycle components and instrumentation elements parameters. Two thermostatic heating and cooling units control the inlet auxiliary water temperature in the condenser and evaporator auxiliary circuits, respectively. The volumetric flow in these secondary circuits is controlled by two modulating solenoid valves and measured by means of two magnetic flowmeters, with an accuracy of $\pm$0.01 l/min. from 0 to 2.5 l/min., and $\pm$ 0.5% F.S. from 2.5 l/min. to 25 l/min. The compressor CL15 is the first CO_2 compressor prototype.

Table 3: General characteristics of cycle elements and instruments of measurement.

COMPONENTS				MEASUREMENTS			
Tubing		*Tube fittings*		*Mass flow meter*		*Security valve*	
Stainless Steel	1/4 OD	Stainless Steel	1/4 OD	limits	0.3-350 kg/h	chocked x_T	0.67
				accuracy	± 0.015%**	C_v	0.41
Dual HTC gas cooler		*Dual HTC evaporator*		repeatability	± 0.015%*	Spring	155-206 bar
Sample tube	1/4 OD	Sample tube	1/4 OD	stability	± 0.015%*		
Annulus	1/2 OD	Annulus	1/2 OD				
Length	4.5 m	Length	4.5 m	*Temperature sensors Pt100*		*K-type thermocouples*	
Insulation	20 mm	Insulation	20 mm	deviation	< 0.1 ^{o} C	deviation	< 0.4^{o}C
				accuracy	± 0.03 ^{o}C	accuracy	± 0.2^{o}C
Compressor		*Metering valve*					
CL15		PARKER		*Pressure transducers*		*Metering valve*	
Cylinder capacity	1.5 cm^3		4Z(A)-NSL-V-SS-V	limits	0-150 bars	limits	0-100 bars
		C_v	0.039	accuracy	<0.1% span	accuracy	<0.1% F.S.

* nominal flow ** measured

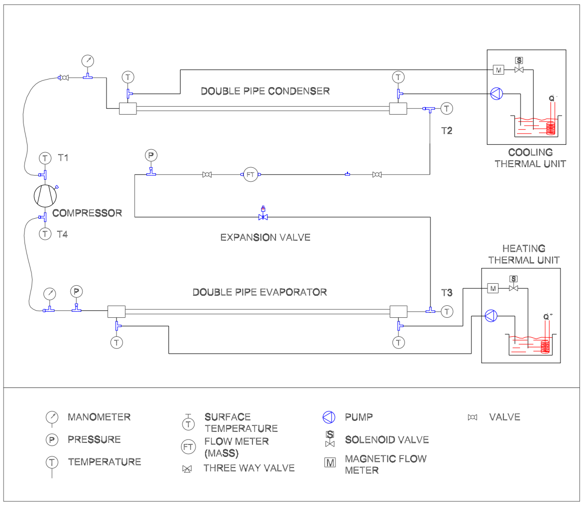

Figure 1: Carbon dioxide unit scheme.

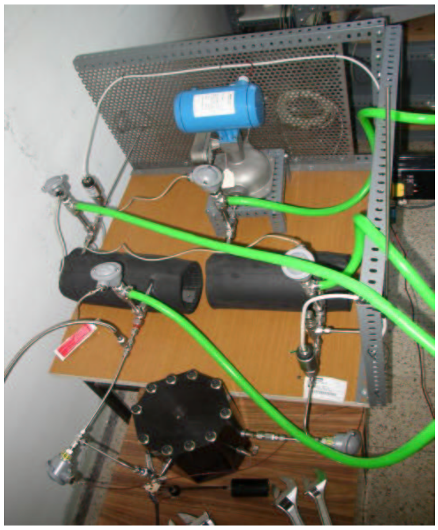

Figure 2: Carbon dioxide unit view.

5. RESULTS

Different comparative results between numerical simulation models (compressors, heat exchangers and whole refrigerating cycle) are presented in order to show an acceptable good agreement between experimental data and the numerical results obtained.

Table 4 shows comparative results for a domestic R134a hermetic reciprocating compressor with 8.0cm^3 of cylinder capacity. The values in brackets are input data in all Tables presented. The results show fairly good agreement. Differences in mass flow rate and power consumption are around 5%, while discrepancies in COP are less than 3% in the studied cases. Although there is no experimental information available for 0^{o}C of evaporation temperature, the comparative results guarantee the reliability of the numerical case presented. Under these conditions the small cooling capacity is around 600W and the COP is 2.123.

Table 4: Global comparative results with R134a cycle vs. calorimeter setup

	T_{evap} (C)	P_{in} (bar)	P_{out} (bar)	T_{out} (C)	$\dot{m}$ (kg/h)	η_v (%)	$\dot{W}_e$ (W)	$\dot{Q}_{ev}$ (W)	ω (rpm)	η_s (%)	COP
numerical	(-10.0)	(2.01)	(14.89)	92.5	8.61	66.9	232.5	405.7	2946	69.5	1.745
experimental	(-10.0)	(2.01)	(14.89)	83.7	8.13	73.4	222.3	375.5	2938	72.6	1.687
numerical	(0.0)	(2.82)	(14.89)	92.0	12.99	70.2	284.5	604.0	2922	67.6	2.123
numerical	(7.2)	(3.77)	(14.89)	87.3	18.43	72.7	329.3	847.6	2904	64.4	2.574
experimental	(7.2)	(3.77)	(14.89)	83.8	17.42	81.5	310.7	789.6	2938	68.2	2.538

In order to validate the heat transfer correlation for carbon dioxide implemented in the two-phase flow heat exchangers, Tables 5 and 6 show the comparative results of the heat exchangers shown in Table 2. The results show a good agreement and the discrepancies are around 1.5% in the outlet gas cooler temperature, while differences in the outlet evaporator are around 6%.

Once the heat exchangers have been validated, a global comparison of the whole carbon dioxide trans-critical cycle has been developed. Table 7 shows the global comparative results of the four temperature cycle points: outlet compressor temperature (T_1), outlet gas cooler temperature (T_2), inlet evaporator temperature (T_3) and inlet compressor temperature (T_4).

Table 5: Global comparative results within the CO_2 gas cooler

	T_{in} (C)	p_{gc} (bar)	$\dot{m}$ (kg/h)	T_{out} (C)	$\dot{m}_{aux}$ (l/min)	T_{win} (C)	T_{wout} (C)
numerical	(94.05)	(75.83)	(13.02)	25.45	(2.01)	(24.92)	31.39
experimental	(94.05)	(75.83)	(13.02)	25.04	(2.01)	(24.92)	31.00

Table 6: Global comparative results within the CO_2 evaporator

	T_{in} (C)	p_{ev} (bar)	$\dot{m}$ (kg/h)	T_{out} (C)	$\dot{m}_{aux}$ (l/min)	T_{win} (C)	T_{wout} (C)
numerical	(0.0)	(35.01)	(13.02)	14.41	(2.01)	(14.91)	9.93
experimental	(0.0)	(35.01)	(13.02)	14.33	(2.01)	(14.91)	10.55

The comparative results of Table 7 show differences less than 7%, without considering (T_3), which is very close to 0^oC; and differences around 3.5% in the global mass flow rate.

Table 7: Global comparative results CO_2 refrigerating cycle

	p_{gc} (bar)	p_{ev} (bar)	T_1 (C)	T_2 (C)	T_3 (C)	T_4 (C)	x_{g3}	$\dot{m}$ (kg/h)
numerical	(75.83)	35.26	96.07	25.38	0.49	14.36	0.285	13.48
experimental	(75.83)	35.01	94.05	25.04	0.18	14.33	0.281	13.02

Table 8 shows the global comparative parameters of the case presented in Table 7. Comparative results on power consumption, cooling capacity and COP are lower than 7% between numerical results and experimental data.

Table 8: Global comparative results with CO_2 cycle vs. laboratory setup

	p_{gc} (bar)	p_{ev} (bar)	T_{ev} (C)	$\dot{m}$ (kg/h)	$\dot{W}_e$ (W)	$\dot{Q}_{ev}$ (W)	COP
numerical	(75.83)	35.26	0.49	13.48	319.54	699.85	2.190
experimental	(75.83)	35.01	0.18	13.02	297.07	687.01	2.313

Finally, it is interesting to remark the comparative results between Tables 4 and 8, where both the conventional R134a cycle and the trans-critical carbon dioxide system work under similar conditions for an equiparable cooling capacity. Tables 4 and 8 shows a similar numerical COP, nearly greater for CO_2 cycle. These results present promising perspectives for carbon dioxide once the compressor prototype is able to be improved and the refrigerating cycle can be optimized.

6. CONCLUSIONS

A comparative study between a conventional R134a cycle and carbon dioxide trans-critical system has been presented. A numerical and experimental comparison of the hermetic reciprocating compressors of both systems has been developed. A numerical and experimental study of the carbon dioxide heat exchangers has also shown. Finally, both numerical and experimental results has present a similar results of both cycles under an equal evaporation temperature and similar cooling capacity. The results shown a promising perspective, due to the CO_2 is a first prototype under development and the trans-critical cycle is able to be optimized.

7. ACKNOWLEDGEMENTS

The authors gratefully acknowledge the financial support provided by the Comisión Interministerial de Ciencia y Tenología (ref. no. TIC2003-07970).

NOMENCLATURE

COP	coefficient of performance	T_{win}	inlet water temperature (C)
h	specific enthalpy (J/Kg)	T_{wout}	outlet water temperature (C)
$\dot{m}$	mass flow rate (kg/h)	$\dot{W}_e$	power consumption (W)
$\dot{m}_{aux}$	auxiliary water mass flow (kg/min	ω	compressor frequency (rpm)
p	fluid pressure (bar)	x_g	vapour quality
$\dot{Q}_{ev}$	cooling capacity (W)	η_s	isentropic efficiency (%)
T	fluid temperature (C)	η_v	volumetric efficiency (%)

REFERENCES

[1] (UNEP). United Nations Environmental Programme. Montreal Protocol on substances that deplete the ozone layer, 1987.

[2] (UNFCCC). United Nations framework convention on climate change. Kyoto Protocol, 1997.

[3] G. Lorentzen. Revival of carbon dioxide as a refrigerant. *International Journal of Refrigeration*, 17(5):292–301, 1994.

[4] A. Jakobsen. Improving efficiency of trans-critical CO_2 refrigeration systems for reefers. In *IIF-IIR Commission D2/3*, pages 130–138, Cambridge, UK, 1998.

[5] H. Kruse, R. Heidelck, and J. Süss. The application of CO_2 as a refrigerant. *Bulletin of the International Institute of Refrigeration*, 79(1):2–21, 1999.

[6] J. Fleming. Carbon dioxide as the working fluid in heating and/or cooling systems. *Bulletin of the International Institute of Refrigeration*, 83(4):7–15, 2003.

[7] J. Rigola, C.D. Pérez-Segarra, and A. Oliva. 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(2):237 – 250, 2003.

[8] J. Rigola, S. Morales, G. Raush, and C.D. Pérez-Segarra. Analysis of two-phase flow in double-pipe condensers and evaporators with special emphasys on transition zones: numerical model and experimental comparison. In *International Compressor Engineering Conference, R067*, Purdue University, IN, USA, 2004.

[9] C.D. Pérez-Segarra, J. Rigola, and A. Oliva. 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(2):215 – 236, 2003.

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

[11] J. Rigola, S. Morales, G. Raush, C.D. Pérez-Segarra, and N. Ablanque. Numerical study and experimental validation of a transcritical carbon dioxide refrigerating cycle. In *International Compressor Engineering Conference, R070*, Purdue University, IN, USA, 2004.

[12] F. Escanes, C.D. Pérez-Segarra, and A. Oliva. 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(9), 1995.

[13] O. García-Valladares, , C.D. Pérez-Segarra, and J. Rigola. Numerical simulation of doble-pipe condensers and evaporators. *International Journal of Refrigeration*, In Press.

[14] F. Escanes, C.D. Pérez-Segarra, and A. Oliva. Numerical simulation of capillary-tube expansion devices. *International Journal of Refrigeration*, 18(2), 1995.

[15] O. García-valladares, C.D. Pérez-Segarra, and A. Oliva. Numerical simulation of capillary tube expansion devices behaviour with pure and mixed refrigerants considering metaestable region. Part I: Mathematical formulation and numerical model. *Applied Thermal Engineering*, 22, 2002.

[16] C.D. Pérez-Segarra, J. Rigola, and A. Oliva. Thermal and fluid behaviour of transcritical carbon dioxide hermetic reciprocating compressors: Numerical analysis. In *International Compressor Engineering Conference, R062*, Purdue University, In USA, 2004.

[17] J. Rigola, C.D. Pérez-Segarra, and A. Oliva. Thermal and fluid behaviour of transcritical carbon dioxide hermetic reciprocating compressors: Experimental investigation. In *International Compressor Engineering Conference, R063*, Purdue University, IN, USA, 2004.

[18] D.L. Bennett and J.C. Chen. Forced convective boiling in vertical tubes for saturated pure components and binary mixture. *AIChE Journal*, 26:454–462, 1980.

[19] Y. Hwang, B.H. Kim, and S. Ramadhyani. New correlation for the heat transfer coefficient during intube cooling of turbulent supercritical carbon dioxide. *International Journal of Refrigeration*, 25:887–895, 2002.

[20] S.S. Pitla, D.M. Robinson, E.A. Groll, and S. Ramadhyani. Heat transfer from supercritical carbon dioxide in the flow: A critical review. *International Journal of Heating, Ventilating, Air-Conditioning and Refrigerating Research*, 4(3):281–301, 1998.

[21] T.M. Ortiz. *Development 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, 2002.