Universidad Politécnica de Valencia
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Estudio teórico-experimental de compresores de pistón herméticos con diferentes refrigerantes

Programa de doctorado en tecnología energética

Tesis Doctoral

Presentada por:
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Dirigida por:
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Valencia, 19 de Diciembre de 2006

Universidad Politécnica de Valencia
Energy Engineering Institute
Applied Thermodynamics Department

Theoretical and experimental study about hermetic piston compressors working with different refrigerants

Doctoral Program in Energy Technology

Doctoral Thesis

Presented by:
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Valencia, 19 de Diciembre de 2006

A mi familia

Hay veces que pese a que sepas que ya no vas a recibir nada a cambio de alguien y que ni siquiera va a ser consciente que vas a estar ahí a su lado, pese a eso simplemente estas ahí:

I seem to recognize your face
Haunting, familiar, yet I can't seem to place it
Cannot find the candle of thought to light your name
Lifetimes are catching up with me
All these changes taking place, I wish I'd seen the place
But no one's ever taken me
Hearts and thoughts they fade, fade away...
I swear I recognize your breath
Memories like fingerprints are slowly raising
Me, you wouldn't recall, for I'm not my former
It's hard when, your stuck upon the shelf
I changed by not changing at all, small town predicts my fate
Perhaps that's what no one wants to see
I just want to scream...hello...
My god its been so long, never dreamed you'd return
But now here you are, and here I am
Hearts and thoughts they fade...away...
E.V.

Supongo que podría estar bastante cabreado con lo que me pasó,
pero cuesta seguir enfadado cuando hay tanta belleza en el mundo,
a veces siento como si la contemplase toda a la vez y me abruma,
mi corazón se hincha como un globo que está a punto de estallar,
pero recuerdo que debo relajarme y no aferrarme demasiado a ella
y entonces fluye a través de mí como la lluvia
y no siento otra cosa que gratitud
por cada instante de mi estúpida e insignificante vida.
No tienen ni idea de lo que les hablo seguro,
pero no se preocupen, algún día la tendrán.

A.B.

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Bueno supongo que para la mayoría de personas que puedan leer esta tesis, los nombres que aquí se citan únicamente serán conjuntos de letras puestas en un papel, de cualquier modo considero justo reiterar de nuevo mi agradecimiento a todos ellos. Pero dejemos estar ya los temas personales a un lado y pasemos a hablar de compresores.

Emilio Navarro

Resumen

La presente tesis se encuentra englobada dentro de los objetivos del proyecto del Ministerio de Ciencia y Tecnología DICORE. En este contexto este trabajo se centra en el estudio de compresores de pistón herméticos para sistemas de refrigeración con el fin de asistir a su adaptación y uso con nuevos refrigerantes.

Inicialmente se procedió a analizar una serie de cinco compresores de pistón de distintas capacidades lubricados con POE funcionando con propano y con R407C, observando a nivel experimental las diferencias observadas en su comportamiento. Puesto que tradicionalmente los compresores que han utilizado propano han venido siendo lubricados con aceite mineral de viscosidad alta, se realizó adicionalmente un estudio comparativo entre aceite MO clavus GV 68 y POE ISO 32 con el fin de determinar la posible aparición de algún problema relacionado con la lubricación y el aceite POE de menor viscosidad. En estas pruebas se procedió a caracterizar un compresor funcionando con ambos tipos de aceite, a realizar medida de OCR (Oil Circulation Rate) y por último se diseño un procedimiento de medida del aceite expulsado por el compresor en el arranque.

Una vez comprobada la viabilidad a nivel experimental del propano como refrigerante en este tipo de compresores se procedió a realizar un modelo semiempírico general de compresor que permitiese por un lado analizar desde un punto de vista teórico las diferencias de funcionamiento de ambos refrigerantes. Dicho modelo permitiría así asistir al diseño de compresores funcionando con otros refrigerantes. El modelo así obtenido proporciona obtener una estimación del rendimiento volumétrico y del compresor cuando se sustituye el refrigerante con un error inferior al 3% en la mayoría de las condiciones de trabajo evaluadas.

Por último, el desarrollo del modelo ha apuntado la posibilidad de la existencia de una condensación de parte del fluido frigorígeno durante el proceso de compresión. Dicho efecto que no es habitualmente contemplado en los modelos de la bibliografía permitiría explicar la discrepancia entre el volumen muerto estimado por el fabricante del compresor y el volumen muerto obtenido con el modelo.

Resum

La present tesi es troba englobada dins dels objectius del projecte del Ministeri de Ciència i Tecnologia DICORE. Dins d'aquest context este trevall es centra en l'estudi de compressors de pistó hermètics per a sistemes de refrigeració per a assistir a la seua adaptació i ús amb nous refrigerants.

Primerament es va procedir a analitzar una sèrie de cinc compressors de pistó de distintes capacitats lubricats amb POE funcionant amb propà i amb R407C, observant a nivell d'experimental les diferències observades en el seu comportament. Ja que tradicionalment els compressors que han utilitzat propà han vingut sent lubricats amb oli mineral de viscositat alta, es realitze addicionalment un estudi comparatiu d'estos dos olis a fi de poder establir l'existència d'algun problema relacionat amb la lubricació i l'oli POE de menor viscositat. En estes proves es va procedir a caracteritzar un compressor funcionant amb estos dos tipus d'oli, a realitzar mesura d'OCR (Oil Circulation Rate) i finalment se disseny un procediment de mesura de l'oli expulsat pel compressor en l'arrancada.

Una vegada comprovada la viabilitat a nivell experimental del propà com a refrigerant es va procedir a realitzar un model semi-empiric general de compressor que permetera d'una banda analitzar des d'un punt de vista teòric les diferències de funcionament d'estos dos refrigerants permetent així assistir al disseny de compressors funcionant amb altres refrigerants. El model així obtingut permet obtindre una estimació del rendiment volumètric i del compressor quan se substituïx el refrigeren-te amb un error inferior al 3% en la majoria de les condicions de treball avaluades.

Finalment, el desenrotllament del model ha apuntat la possibilitat de l'existència d'una condensació de part del fluid frigorigen durant el procés de compressió. Dit efecte que no és habitualment contemplat en els models de la bibliografia permetria explicar la discrepància entre el volum mort estimat pel fabricant del compressor i el volum mort obtingut amb el model.

Abstract

This thesis is included within the objectives of the project of the Ministerio de Ciencia y Tecnologia DICORE. Within the frame of this project, the thesis is focused in the study of hermetic piston compressors for refrigeration systems to attend their adaptation and use with new refrigerants.

Firstly a series of five reciprocating compressors of different capacities lubricated with POE working with propane and R407C were analyzed. Since traditionally the compressors working with propane have being lubricated with mineral oil of high viscosity, an additionally a comparative study between a MO of high viscosity and the used POE were developed to evaluate the possible convenience of using POE of lower viscosity with propane. These tests included a compressor characterization with both oils, OCR measurement (Oil Circulation Rate) and finally the design of a procedure to measure the oil expelled by the compressor in the start up.

Once verified the reliability in the experimental level of propane as refrigerant with this compressors. A general semi-empirical compressor model which allows on the one hand to analyze from a theoretical point of view the observed differences between both refrigerants and on the other hand to asses the design of compressors working with other refrigerants. The obtained model allows obtaining an estimation of the volumetric efficiency and the compressor when the refrigerant with an error lower than 3% in most of the tested working conditions.

Finally, the development of the model has pointed out the possibility of the exis-

tence of an effect of condensation of the refrigerant during the compression process. This effect that habitually is not contemplated in the models of the bibliography would allow to explain the discrepancy between the dead volume of the compressor considered by the manufacturer and the obtained dead volume with the model.

Preface

Before starting this thesis on the refrigeration field nearly four years ago, I had been working in a theoretical field. My previous work consisted of the development of a mathematical formalism for the polarization of light in a Minkowskian space-time. This topic might seem unrelated to the thesis that you are holding in your hands. At that time, I remember that after one year of hard work on the formalism for polarization, the only thing we had was a certain quantity of dirty pages with equations, and a bitter feeling of having wasted our time, due to the suspicion that perhaps we were looking for something that simply did not exist. Nevertheless, at the end, all the things fitted and we found a mathematical structure that was not so hidden in Nature.

Afterwards, I moved to the field of work of this thesis. During the first year and half I was working in the test rig for the characterization of components, which the IMST group had placed in the Instituto de Ingeniería Energética at the Universidad Politécnica de Valencia. There I worked in the characterization of refrigeration compressors of several types. This meant a significant change, as I began to work in an experimental field, totally unknown to me before. My first feelings were related with the confidence in the process of developing experiments, despite the large amount of hard technical work required and the daily problems to be solved. It looked as if to solve a problem was to give the first step to the following problem instead of a real solution to the problem. You had the advantage of feeling the certainty of being in front of a real problem with a real solution, with the confidence that if you speculated you could find the solution. If a machine is not working properly, for sure, there is a real reason that produces this effect, so you "only" have to find it.

But when the modelling process presented in this thesis started, the same "relativistic" insecure feeling turned up again. At first, the main objective was to develop a simple model for the compressor and volumetric efficiencies, which allowed us to infer the relative magnitude of the different losses in the compressor by knowing the aforementioned parameters in some working conditions. This model implied a large amount of simplifications of the real physical system. Thus, the possibility of being working with such a complex system that our initial objective became just impossible to achieve, came to our minds. In fact, several alternative models were developed before reaching this final version in which additional loss terms have been incorporated.

All these facts allowed me to realize how similar both of the research fields are. In spite of the a prior impressions which one could have, you can study a compressor or you can study the Universe, but what you basically work with is a system from which you have a limited amount of information, obtained by any means, which you want to turn into more knowledge about the system. This knowledge will allow to predict its future behavior or to explain its past behavior. The understanding of the system will then imply a better understanding of our medium, our Universe, or the possibility to use or create a device with different intentions. That is basically the game of science: we have limited information about a system, we make some hypothesis about it, based on assumptions, and after the development of these hypotheses, experimentally or theoretically, we arrive to certain conclusions which, after being compared with the system, can confirm or deny the hypothesis made. This game is played with the same rules for the compressors as for the Universe.

From this, maybe you can think that both knowledge areas are similar, but this is not exactly so because engineering goes one step further since the knowledge of the system is usually used to modify the system to improve its behavior. Some time ago, I read o a quotation of a physicist saying that they play to try to read the god's mind. In this context we could say that an engineer play to try to be small gods attempting to create the new realities.

Es sencillo darse cuenta de lo fácil que resulta enroscar un tornillo en una tuerca. Enroscarlo es otra historia.

Nomenclature

$\dot{Q}_{ev}$	Cooling capacity
$\dot{m}_{in}$	Mass flow rate
h_{ev}	Enthalpy exchange in the evaporator
Δh_i	Enthalpy in the state i of figure 2.4
$\dot{E}_k$	Compressor energy consumption
$\dot{E}_{mech}$	Energy dissipated in mechanical losses
η_k	Compressor efficiency
η_s	Volumetric efficiency
$\overline{\dot{m}}_{leak}$	Mass flow rate leaked
$\dot{V}_s$	Swept volume
$\dot{V}_d$	Dead space
ρ_i	Density in state i
η_{sth}	Theoretical volumetric efficiency
$\overline{\dot{m}}_{pc}$	Mass flow rate in phase change
η_{el}	Electric efficiency
$\mathbf{K}$	Model parameter vector
$\mathbf{G}$	Compressor parameters known
η_{sexp}	measured volumetric efficiency
η_{kexp}	measured compressor efficiency
$\sigma(x_i)$	standard deviation
ϵ	error
OCR	Oil circulation rate
COP	Coefficient of performance
$\dot{m}_{oil}$	Oil mass flow rate
δ	oil film thickness
Re	Reynolds number
d	Pipe diameter
τ	stress tensor
μ	dynamic viscosity
L_{tub}	Pipe length
v_{oil}	Oil velocity

ρ_{oil}	Oil density
ρ_{ref}	refrigerant density
f	Friction factor
M_{oil}	Oil mass
K_1	Losses parameter related to the percentage of heat released to the refrigerant by electric and mechanical losses.
K_2	Losses parameter related to heat transfer between inlet and outlet gases.
K_3	Losses parameter related to pressure losses in the cylinder inlet valve.
K_4	Losses parameter related to pressure losses in the cylinder outlet valve.
K_5	Losses parameter related with the leakages.
(V_d/V_s)	Dead space ratio obtained with the model without phase change inside the cylinder.
$(V_d/V_s)'$	Dead space ratio obtained with the model considering phase change inside the cylinder.
K_6	Losses parameter related with a possible phase change inside the cylinder.
K_7	Losses parameter related with mechanical losses derived from the power consumption.
K_8	Losses parameter related with mechanical losses derived from the motor velocity.

Contents

CHAPTER 1

Introduction

1.1 Motivation

Since the first refrigeration compression machines were developed during the 1830-1840 decade (Jacob Perkins patented the first refrigeration compressor in 1834), the research to find the most suitable fluid for these systems has been an important challenge from the point of view of safety and efficiency . This issue seemed to be solved in 1930 with the development of CFC based fluids by Thomas Midgley. However, in 1974 Sherwood Rowland and Mario Molina [1] predicted that these gases could cause a serious damage to the atmospheric ozone layer. In 1985 a *hole* in the Antarctic ozone layer was discovered. This fact was the departure point that motivated the phase out of chlorofluorocarbons (CFCs) and hydrofluorocarbons (HCFCs) at the first Montreal protocol [2] two years later. Since then, the group of suitable refrigerants were reduced to hydrofluorocarbons (HFCs) and natural refrigerants. After that, in 1997 the Kyoto protocol [3] assessed actions to achieve the stabilization in the atmosphere of the amount of greenhouse gases at levels that would prevent any interference with the climate system and has encouraged the promotion of policies to sustainable development. This regulation has affected the refrigeration systems in two ways:

- HFCs have a high global warming potential (GWP)(more than 1000 times the effect of CO_2), and thus they are included in the list of gases whose use must be reduced.

- Refrigeration systems represent quite an important amount (20% approximately) of the energy consumption of the developed world and their efficiency should be improved in order to reduce the indirect emissions of CO_2 to the atmosphere.

Due to these facts, a considerable research effort has been focused in finding an environmentally friendly refrigerant and also improving these systems efficiency during the last years.

This thesis has been developed inside the group "Investigacion y modelado de sistemas termicos" (IMST) from Universidad Politecnica de Valencia (UPV). This group started a research line in the refrigeration area in 1994, and since then a wide experience in this area backed by several doctoral thesis ([4], [5], [6], [7]) and contributions in international conferences and journals like [8], [9], [10], [11] or [12], was gained in the experimental and in the modelling aspects of the problem.

The activities of the group in the refrigeration branch has been centered in the following fields:

- The development of simulation tools for the different components of the refrigeration systems to aid their design.

- Experimental study of the new refrigerants either with comercial equipments or new developed prototypes.

- To study the different heat and mass transfer processes in these systems.

All these activities have been supported by several national and European projects. Among them it should be mentioned **JOULE** (*A new high-efficiency air to water reversible heat pump working with propane (R290) for commercial applications in southern europe*), **GEOCOOL** (*Geothermal heat pump for cooling and heating along european coastal areas*), **SHERHPA** (S*sustainable heat and energy research for heat* *pump applications*), **GEOCARE** (*To take profit of the residual heat of geothermal origin to improve the energy efficiency in heating and cooling of buildings through the use of ground coupled heat pumps*), **DICORE** (*Theoretical and experimental study of the main components of the refrigeration systems addressed to the synthesis of the design directives for their evolution to the new refrigerants and the development of models to asses the optimization of the systems*).

This thesis was started within the objectives of the project DICORE, focussing its attention in the compressor. Inside this context, this thesis increases the compressor data base using propane and R407C as refrigerants and develop a theoretical model to deepen our understanding of the compressor behavior with different refrigerants.

1.2 State of the art

1.2.1 Propane as an alternative natural refrigerant

As a consequence of the troubles appeared in connection with refrigerants, as exposed in section 1.1, , an intense search for appropriate substitutes has been developed.

Although, in a first moment HFCs were adopted as a good alternative (because they had a null ODP) but they already have a high GWP. Therefore they were included in the list of fluids whose production has to be reduced in the next years and can only be considered as transition fluids. As a sound alternative nowadays, people are looking for natural refrigerants as the most feasible long term solution, since they have been in the atmosphere for a long time and their effects on the environment are perfectly known.

Among the different alternatives CO_2, ammonia and HCs are the most usually considered. Regarding CO_2, it has the disadvantage of working at very high pressures (approximately 100 bar) which requires a complete redesign of the existing systems, a quite detailed review of CO_2 can be found in [13]. Ammonia is another natural fluid with suitable thermodynamical properties to be used as refrigerant. It has the disadvantages of a strong smell, is toxic, flammable and is not soluble in most commonly used oils in refrigeration systems. A complete review of recent applications of this fluid in small residential heat pumps can be found in [14].

Regarding HCs, and specifically propane to which this thesis is focused, it was presented as a working fluid in refrigeration systems in 1922 to substitute ammonia, But was substituted in most of the systems few years afterwards, when the CFCs were discovered, and since then their use was limited to big refrigeration plants mainly in the petrochemical industry. After the CFCs and HCFCs were forbidden, the interest in the use of propane returned to the refrigeration industry, because of its capability of allowing a direct retrofit in the old refrigeration systems working with HCFCs, its very good thermodynamical properties (a complete review about the properties of this refrigerant is given in [15], [16], [17]) and compatibility with most of the materials used.As a consequence of this, since the last decade, a significant research effort has been developed regarding the use of HCs in refrigeration and air conditioning systems.

From the point of view of the thermodinamical properties of hydrocarbons. Spindler [18] compares R290 properties with those of other commonly used refrigerants. Higashi [19] determine the critical parameters for propane and isobutane. Recently Saleh et al. [20] has published a work based on BACKONE equations to evaluate theoretically the different alternatives to HFCs, including R290.

From the point of view of the global system performance, James et al. [21] evaluated the behavior of a refrigeration equipment using R290, Lystad [22] studied a water to water heat pump working with R290, Halozan [23] compared the performance of a refrigeration system working with R290 and R22 and Purkayashta et al. [24] analyzed the influence of the condensation and evaporation temperature in a heat pump working with R290. Blanco [4] studied the optimization of a air to water heat pump working with R290, Hammilic [25] developed a comparative study between R410A, R290 and R134a and Domanski et al. [26] has been compared the performance of a refrigeration system working with HCs and HFCs.

Other authors have focused their studies in the behavior of HCs in heat exchangers. Setaro et al. [27] studied compact brazed plate heat exchangers for heat pump applications; Thonon et al. [28] reported data and design method for shell-side condensation. Nan et. al [29] studied in tube condensation of propane in plain, micro-fin and cross hatched tubes comparing their results to correlations from the open literature. Jung et al.[30] studied 6 flammable refrigerants for shell-side condensation. Chang et al. studied in-tube condensation of various hydrocarbons and proposed a new correlation. Beaugeois et al. [31] analyzed the condensation in plain and micro-finned tubes. Finally, Lee et al. [32] examined the heat transfer coefficient and the pressure drop in a specific heat exchanger.

In any case, regarding compressors there is still a considerable lack of information related to its behavior with propane and there are not too much companies that provide compressors for propane, *Danfoss* has a hermetic compressor line for this refrigerants (www.danfoss.com) but only for low capacity systems, *Bitzer* provides compressors for propane, but only semihermetic (www.bitzer.de), and the most important manufacturers do not supply any hermetic compressor for medium and large capacity systems for this refrigerant. This is one of the reasons which are preventing a wider propane use in refrigeration systems.

As a consequence of this, one of the motivations of this work was to increase the data base of hermetic compressors working with propane, and increase the experience of its safe use with compressors lubricated with POE oil, commonly used with HFCs.

A further problem in the development of propane as refrigerant is the way in which propane interacts with the commonly used oils which is quite different to CFCs HCFCs and HFCs because of its higher solubility. This high solubility affects the system in two ways, on the one hand it increases the amount of refrigerant solved in the oil increasing the total charge of the system, on the other hand, the high solubility affects the way in which the compressor is lubricated by the oil. A complete review of the most common lubricants used in refrigeration can be found in [33]. In this area one of the first works developed was done by Hans-peter [34], but also the work of Fernando et al. [35] should be mentioned which study the solubility of R290 with several lubricants or the complete experimental analyze developed in the university of Dresde [36] in which general properties of the hydrocarbons-oil mixtures were analyzed. From the theoretical point of view, Bertucco et al. [37] evaluated the mixtures of several refrigerant-oil mixtures including R290 based on a perturbed hard sphere chain equation. Quiñones-Cisneros et al. [38] developed what they call the f-theory to study the viscosity of several mixtures of oils with refrigerants. In any case, up to the knowledge of the author, there is a considerable lack of information in this field regarding the real behavior of the oils used nowadays in refrigeration systems in a system using propane as refrigerant.

In any case, probably the most important factor preventing the introduction of HCs in the market has been its flammability. Regarding this important drawback of HCs, since nineties the evolution of the standards to make a safe use of them in residential heat pumps has been in constant evolution. In connection with the state of the standards related with heat pumps, reference [39] constitutes a good review.

To reduce this problem mainly two methods to lower the charge of HC have been studied: To work with a mixture instead of a pure HC (see [40], [41], [42]), in these studies the system lose COP but increase the capacity or to redesign the refrigeration systems (mainly the heat exchangers) to reduce the charge (see [43], [44])

In this sense the role that the oil, or the solubility of the refrigerant in the oil can play in the reduction of the total charge of the system is noticeable, since compressor represent the second element (after the condenser) which contain more refrigerant charge.

1.2.2 Compressors

Nowadays, single stage vapor compression cycle is the most important refrigeration cycle used for small refrigeration systems and residential heat pumps. This cycle is divided in the following four parts:

- Pressure and temperature of the refrigerant in gaseous state is increased up to certain level (compressor part).
- Refrigerant is cooled down at constant pressure giving a part of its internal energy to the environment (condenser part).
- An abrupt reduction of the refrigerant pressure at constant enthalpy up to a pressure and temperature that makes possible the phase change (expansion valve).

- Evaporation of the fluid at constant pressure, absorbing certain amount of energy from the environment (evaporator).

In this cycle, the compressor is the one that supplies the required energy to make feasible the heat transfer from a cold place to a hot place. It is the most determinant element in the efficiency of a refrigeration system. Its function consists in delivering the refrigerant from the cold reservoir to the hot reservoir, transferring by compression a certain amount of energy to the refrigerant.

There are several compressor types commonly used in refrigeration [45], [46]: positive displacement (rotary, screw, scroll, reciprocating, etc...) and dynamic (turbo compressors). In the first type the gas is trapped in a space which is reduced during the rotation of the axis driving the compressor, as a consequence of the volume reduction the pressure will rise. In the turbo compressor the pressure increases due to the high velocity given by the impeller.

There are also important differences among the positive displacement compressors. As this work is centered in hermetic reciprocating compressors, further information about the other compressor types the reader should refer to the literature [47],[46].

The piston compressors were the first compressor invented. Its mechanism is relatively simple although, as it has a lot of mobile parts, it is a technical challenge to obtain a noiseless and vibrationless operation. Reciprocating compressors use a piston that is driven directly through a pin and connecting rod from the cranckshaft. Hermetic reciprocating compressors have the motor and the crankshaft placed in the same housing with direct contact with the refrigerant and oil.

Nowadays, reciprocating compressors are the most widely used technology in cooling systems. They are employed in household refrigerators and freezers, residential air conditioning, low capacity comercial refrigerating units and heat pumps.

A general scheme of the path followed by the refrigerant inside the piston can be seen in fig. 1.1.

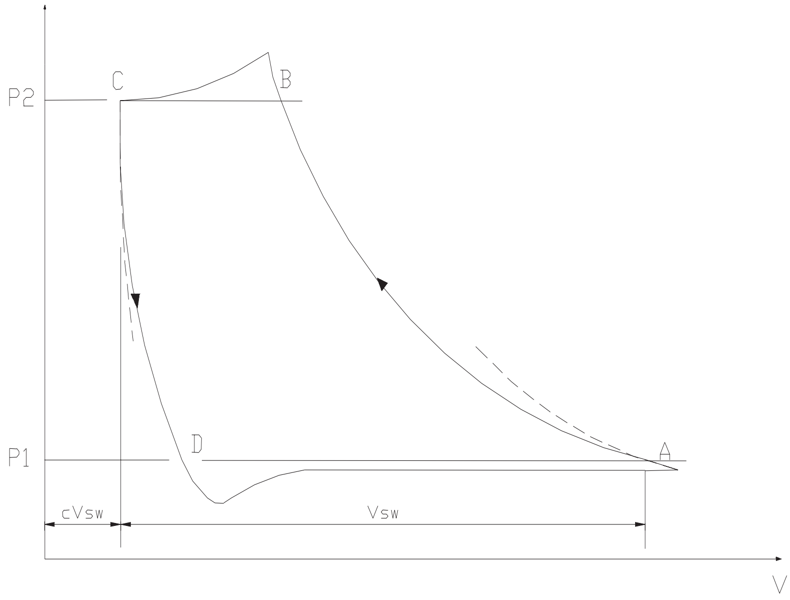

Figure 1.1: Compression cycle:(A-B) cylinder full of refrigerant begins the compression in an approximately adiabatic process up to the discharge pressure, (B-C) Discharge valve is opened and the refrigerant begins to flow outside the cylinder,(C-D) Discharge valve is closed and the adiabatic expansion of the refrigerant that remains inside the cylinder begins, (D-A) The suction valve is open and the refrigerant from the evaporator enters inside the cylinder.

1.2.3 Modelling

A *model* is a theoretical scheme of a system, expressed in most of the cases in a mathematical form, which is intended to allow an easier understanding and to predict its behavior.

There are several ways to face the problem according to the model purpose and the available information about the system.

Considering the hypothesis which have been taken into account and the used parameters, a possible way of classifying models is the following:

- **Empirical models**: These models are based on empirical correlations in which input data are related to output data. They do not fit exactly the previous definition of what a model is because they do not describe any physical phenomena.

- **Semi empirical models**: These models are based on experimental data, but they describe up to some extent the system using some of its characteristical variables combined with some empirical parameters. Even though they do not give information about all the physical processes involved, they allow more physical insight than the empirical models.

- **Theoretical models**: These models analyze all the processes involved trying to describe the whole system in terms of the physical laws implied. This is usually done dividing the system into individual parts (control volumes) that are coupled together.

This classification is arbitrary and each group can also be divided in several subgroups as it is done in [48]. This proposal has been done according to the criteria of generality and usefulness.

The empirical models have as their main advantage that they are fast and simple, but on the opposite side, they are mostly unable to predict the behavior of the system in non-tested conditions. They are useful for programming control systems or evaluating the system in conditions specified by some standard. In compressors, these kind of models are the ones used by the ARI and ISO normative in which a polynomial equation of 10 coefficients is required to describe the compressor. The coefficients of these polynomials are obtained from a least square fit of a set of experimental data.

The main advantage of the semi-empirical models is their simplicity. However, they allow to obtain an idea of the system performance and in some sense to assist in the analysis of empirical data. Their utility as a design tool is less clear because they do not allow to extrapolate data when some internal parameters of the system are changed. In the particular case of compressors, parameters like clearance volume ratio or compressor displacement are commonly used and the output of the model are usually the compressor performance characteristic variables like isentropic and volumetric efficiencies.

The theoretical models are also subdivided by some authors in subcategories according to the complexity and the amount of empirical information required. In general terms, the higher their complexity the more the running time and the required number of boundary conditions. They are mainly useful in the design of systems and to estimate the value of some magnitudes difficult to measured. Their main drawbacks are the larger time to develop this kind of models and that sometimes the a priori data involved are difficult to estimate precisely. In the particular case of compressors, as in nearly complex systems, as a consequence of the complexity and the amount of processes involved, to develop a complete theoretical model is nowadays nearly imposible. Thus, although they are more detailed than the semi-empirical type of models, they usually use some correlations and effective factors difficult to estimate.

It should be pointed out here that in the open literature the expression "compressor model" is used to describe many different types of models. Reviewing papers on this subject one may find this term used for models which describe only a part of a compressor, like contact forces between moving parts, the acoustic properties of the compressor shell or the dynamic response of the compressor suspension system. However, as this work is focused on the calculation of the compressor performance with respect to capacity and efficiency, the following lines are going to be dedicated to the description of the main trends and subjects.

If a compressor is seen as a single control volume, a useful description of its performance requires at least the analysis of the following characteristics:

- Capacity performance.
- Energy performance.
- Thermal performance.

Each of these characteristics can be described in terms of system oriented variables or in terms of compressor oriented variables.

The system oriented variables express the performance of the compressor in terms of variables such as cooling capacity, COP/EER and thermal performance or discharge gas temperature.

Sometimes these system oriented variables are misunderstood and their changes are associated to changes in compressor efficiencies, neglecting the possible influence of other parameters related with other system parts such as the heat exchangers.

A higher precision in characterizing the compressor performance is obtained by using directly the so called compressor oriented variables. Next, a classification taking into account the most commonly used variables is included:

- **Capacity performance**
  The capacity performance of a compressor relates the actual flow of refrigerant to an ideal flow having no losses. The most often used magnitud for this characterization is the volumetric efficiency which relates the volume flow of refrigerant in the compressor suction to the displacement rate of the compressor. The definition given here can only be applied to positive displacement compressors.
  All the effects having an influence on the reduction of the mass flow rate are included in this general definition of volumetric efficiency. Strictly, this efficiency can be subdivided in several terms reflecting the different factors responsible of the reduction of volumetric efficiency, such as thermometric efficiency (reduction of volumetric efficiency as a consequence of heating of the refrigerant), clearance volumetric reexpansion or leakages.

- **Energy performance**
  This performance can be described by several different efficiencies comparing the actual power consumption to the power consumption of a reference process. The difference between these efficiencies lies in the choice of the reference process. Among these reference processes the isentropic, polytropic and isothermal processes could be mentioned.
  In this work the isentropic process is considered as the reference state (compressor efficiency). The main reason for this decision was that, it was known from some previous work of our group in this type of compressors [7] that

the heat transfer inside the compressor chamber was nearly negligible. The polytropic reference process has the disadvantage of having a fixed exponent highly dependent on the refrigerant that should be determined. This exponent does not give considerable information about the energetic performance and it makes impossible a fair evaluation of the efficiency of a given compressor with different refrigerants.
The isothermal reference process is commonly used for compressors with low operating frequencies where it is more relevant to compare the actual compression process with an isothermal one.

- **Thermal performance**
This variable is not included in the developed model. This decision was taken because it was not possible to define a methodology to evaluate this magnitude using only catalogue data. Besides, it should be pointed out that regarding this performance there is not a generally accepted method to characterize it.

Compressor modelling

The first investigations carried out with refrigeration compressors were developed assuming an ideal compression process in which the fluid had a polytropic compression and the inlet and outlet valves had an infinite area. Under these assumptions, the obtained expressions for volumetric efficiency and for the compression work were:

$$\eta_v = 1 - c(r^{\frac{1}{k}} - 1) \tag{1.1}$$

$$\dot{W} = \frac{k}{k-1} m R T_{adm} (r^{\frac{k-1}{k}} - 1) \tag{1.2}$$

Throughout the forties more elaborated compressor models were developed [49], [50]. These models represented a significant step forward in compressor design due to the increase of the number of parameters to be considered by the designer. Nevertheless, those models required the solution of several non linear partial differential equations and since at that time computers were not developed enough, the equations could only be solved with graphical methods or mechanic calculators. Therefore as a consequence of their high computational cost and the uncertainty they had, they were not widely used. This was going to change with the emerging digital computers.

In 1958 Brunner [51] presented the first global compressor model. In the sixties, the compressor models began to be popular among the specialists. For instance, four contributions [52], [53], [54], [55] were presented in the XII International Refrigeration Conference organized by IIR (1967). In 1972, the Purdue University organized the first International Compressors Conference and since then this meeting has been considered as a reference in the refrigerant compressor research, being a good sample of the main trends in the field.

In 1972, Soedel [56] imparted a course in compressor modelling which was extended by Hamilton [57] in 1974. This course was focused on how to calculate the time dependent variations of the pressure and temperature in the cylinder and in the suction and discharge volumes using the first law of thermodynamics, coupled with a valve model. The model explained in that course was the basis of all the models later developed. The development of the next generations of models was done trying to improve the hypothesis of this model or to adapt it to analyze more accurately certain compressor processes. The areas where more work has been done in are:

- **Pulsation and acoustic waves:** The first research in this field was developed by Bannister [58], Czaplinski [59], Stein [60] and Jasper [61] although they did not include the interaction between pulsations and valve dynamics. The first integration of this phenomena was done by Brablik [62], [63].

  Benson and Ucer were, up to the authors knowledge, the first to integrate this effects in a global compressor model [64].

  Since then several alternatives have been considered increasing the details of the process, among them, it can be mentioned [65], [66], [67], [68].

- **Valve model:** This study has two important related aspects, the valve dynamics and its interaction with the thermodynamic matter of suction and discharge and the stress mechanical analysis to evaluate the durability of the valve. A good review on this field is presented in the courses [69], [70] which were focused in the compressor valves design and their effects in compressor performance.

- **Heat transfer model:** In the beginning, the analysis of this process was mainly focused on the cylinder walls [71], [72], [73], [74], [75], but then these studies were extended to all the other parts [76], [77] demonstrating their influence in the global compressor performance. A review about the main results in this direction can be found in [78].

These approaches to the compressor description are bounded mainly with the development of what is known as detailed models or what we called in the last section *theoretical models*. These models allow to obtain a deeper understanding of the compressor, however, they use to have the problem of requiring too much information (usually only available for the manufacturer) to be useful from a practical point of view. As a consequence of this, the evolution of models more similar to the ones described in eqs. 1.1 and 1.2 has not been abandoned. In literature, a certain number of such models with different deepness in their physical sense according to their purpose [79], [80],[81], [82], [83], [84] may be found. In a general way, they are quite accurate for the desired application but they usually show the following problems: i)they are too empirical to be able to extract relevant physical information ii) certain parameters of the model are difficult to be physically understood iii) when the number of parameters is increased, sensibility problems appear in the parameters values avoiding a direct physical interpretation without the help of an experienced user.

The author considers that it is not convenient to proceed to comment empirical models because it is considered that although they could be quite useful for the industry, when someone want to decide which compressor install among others, to control a refrigeration plant, to minimize energy consumption or to maximize production. As they usually are based on empirical correlations, they not give information about the compressor physical processes. Therefore they are not so useful from our scientist point of view.

1.3 Objectives

Within the frame exposed in the last section, and taking into account the objectives of the DICORE project, in the beginning of the project propane was thought as the best natural fluid to substitute the HCFCs and HFCs.

As said before, there is not much information regarding compressors working with this refrigerants and most of them use a mineral oil (MO) of high viscosity as lubricant. In any case, HFCs are not soluble in MO and refrigeration systems that use these refrigerants are commonly lubricated with POE oils. As a consequence of this, it was considered necessary to investigate the way in which propane works with POE to know the possible adverse effects of a direct drop in of a system working with HFCs. POE oils have the advantage of possesing lower propane solubility, which from the point of view of the total charge of the system is a very suitable property.

Going further in the project objectives, it was an important task to synthetize design directives for the new refrigerants and the development of models to asses the optimization of systems. Nowadays propane is not a common used refrigerant, and as the ARI polynomial coefficients, that according to the standard characterize the compressor behavior, are only valid for each refrigerant, it is difficult to extrapolate data from the refrigerants given in the catalogues to others not included. Consequently, to seek a compressor model able to estimate data for one refrigerant from data of other refrigerant was considered. The characteristics of this model should be near to what we call semi-empirical models as it only should require information commonly available in catalogues and not a deep knowledge of the compressor. In addition, this objective opens another interesting question: knowing the efficiency at some operating points for a given compressor with a given gas, is it possible to estimate the relative magnitude of the different losses?. Up to the knowledge of the author, there is no literature related to a model with these characteristics and for this reason a new one had to be developed.

Synthetically, the objectives of this thesis can be summarized as:

- To develop a comparative experimental study of a series of hermetic reciprocating compressors working with propane and R407C using a polyester oil (POE) and mineral oil (MO) as lubricants.
- To develop a new compressor model with three useful features:
  - It will allow quantifying the relative importance of the main sources of losses in compressor and volumetric efficiencies in a first approximation.

- It will allow predicting the performance of a compressor working with refrigerant different from those included in catalogue data.This second objective is especially relevant if a not common refrigerant like propane is used.
- It will be comparatively global and simple

This thesis is organized as follows: first of all, the definition of the problem and the nowadays state of the art is described in the introduction, then, the developed model and the way to solve it is described in chapter two, continuing with a description of the test rig used in chapter three. The experimental test and the corresponding results obtained are reported in chapter four and moreover, these results are used to check the model and to analyze the behavior of the studied compressors and a possible explanation of the different behavior observed between propane and R407C in chapter five. Finally, papers related to this work already published or under revision in international journals are annexed as appendices at the end of the document.

Some other publications produced during this thesis have not been included here as the author considers that they are only partial results already included in the papers given as appendices or they are not directly related with the issue of this thesis ([85], [86], [87], [88], [89], [90]) .

The main body should not be considered as a detailed technical description of the developed work, instead, the author has preferred to organize it as a review of the main results obtained including certain relevant information not published. Therefore, the reader is submitted to the appendices placed at the end of the document for a more extended analysis about the technical contents of this thesis.

CHAPTER 2

Compressor model

2.1 Compressor description

The compressors used in this work were hermetic reciprocating compressors of similar design, had approximately a range from 5 to 30 kW of cooling capacity, thus they were suitable for heat pumps of medium and high capacity. Their main characteristics are shown in table 2.1. A general scheme of their internal design is

Stroke (mm)	Pistons	Dead Space Ratio	Compressor name
30.23	1	0.037	SO
38.10	1	0.029	LO
30.23	2	0.037	ST
38.10	2	0.029	LT

Table 2.1: Tested Compressors.

shown in fig. 2.1

The refrigerant fills all the internal compressor sections. The oil in the compressor sump is in direct contact with the refrigerant and furthermore certain amount of refrigerant is solved in it. Despite the reduction of oil viscosity, the refrigerant must

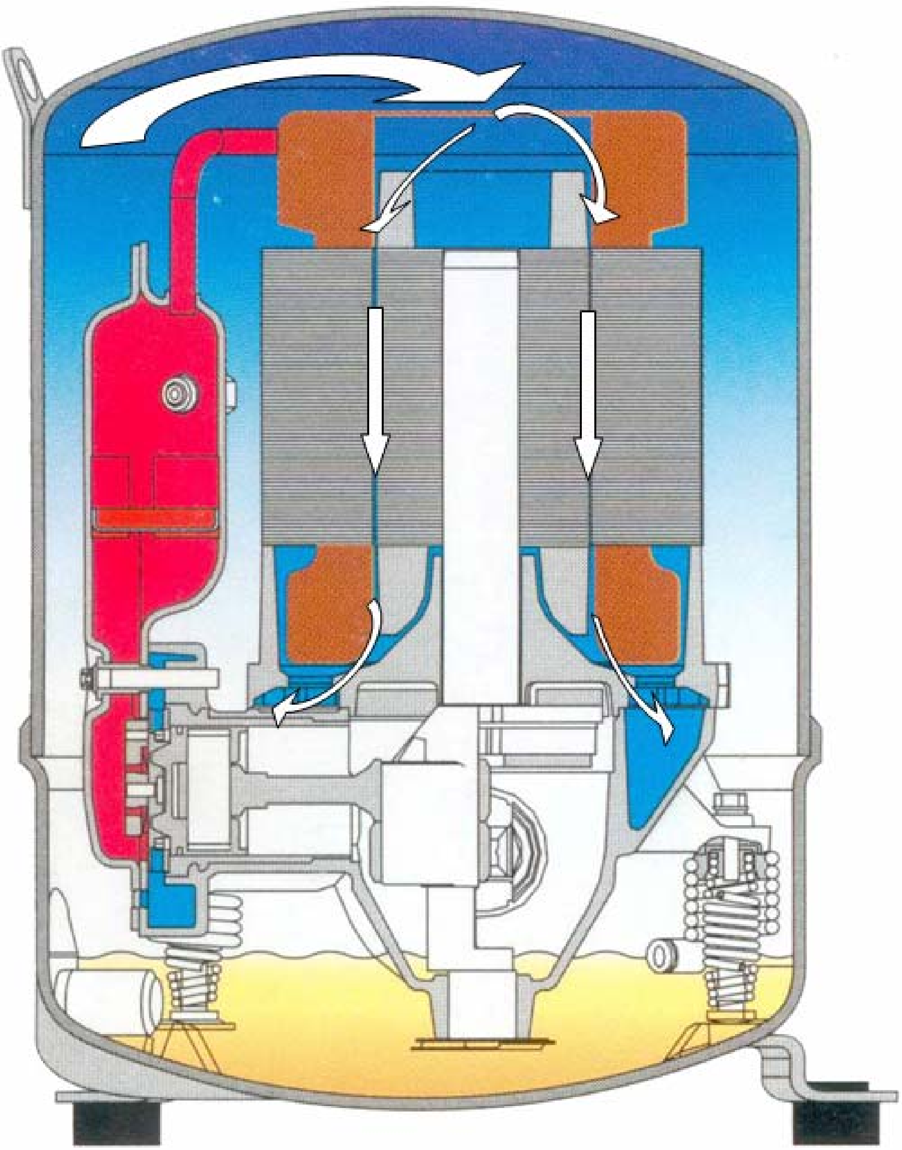

Figure 2.1: Compressor scheme

be soluble in the oil, since when certain amount of oil is expelled to the rest of the test rig, it must be dragged by the refrigerant to allow the return of the oil expelled

Figure 2.2: Compressor without cranckcase

by the compressor to the rest of the installation. Although, the oil is in contact with refrigerant coming from the evaporator, its temperature is mainly determined by the refrigerant temperature in the discharge as is going to be shown in chapter 4.

Next, the cycle followed by the refrigerant into the compressors is described. The

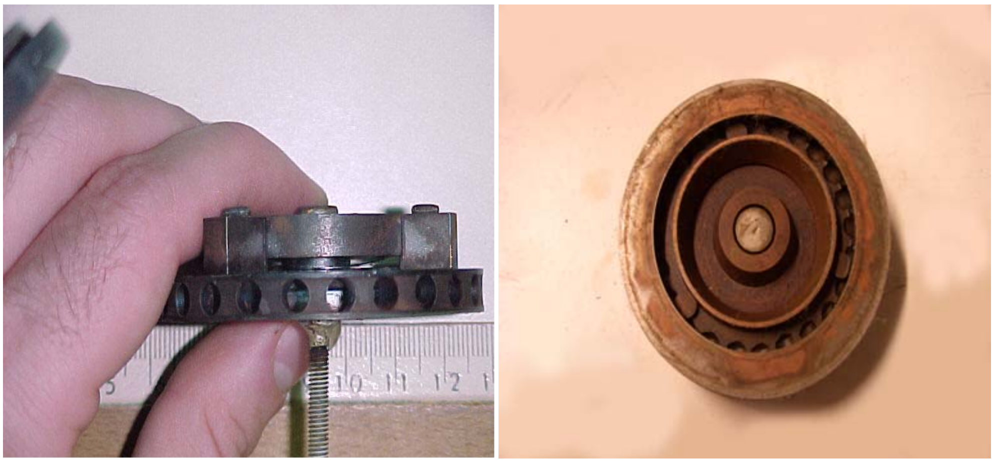

Figure 2.3: Compressor valve.

refrigerant coming from the evaporator goes inside the compressor and then follows the path shown in fig. 2.1. It is sucked out at the top of the compressor and then heated by the electrical motor before it arrives to the inlet port. This is the way in which the electric motor is cooled by the refrigerant. Thereafter it goes into the cylinder passing through the inlet valve (fig. 2.3) with the corresponding heating and pressure drop. Therefore the refrigerant inside the cylinder has a higher temperature and lower pressure than the refrigerant coming from the evaporator. Then the refrigerant is compressed and, when the pressure is near the condensation pressure, released through the discharge valve. Before the refrigerant leaves the compressor, it passes trough the muffler and afterwards the discharge pipe leads the refrigerant through the oil sump increasing its temperature and reducing its viscosity.

In this compressor, the muffler is placed quite close to the electric motor, as a consequence of this, in some operating conditions the motor may increase the temperature of the refrigerant in the compressor discharge part. Therefore, the refrigerant temperature could be higher at the compressor outlet than at the cylinder outlet. This effect could lead to certain confusion, because negative values of the heat losses coefficient may be obtained as its consequence ($\dot{Q} = \frac{\dot{m}_{in}(\Delta h_{1-8})-\dot{E}}{\dot{E}}$).

2.2 Compressor model

The most determining requirements of the compressor model developed in this thesis were:

- It shall reproduce compressor efficiency ($\eta_k = \frac{\dot{m}_{in}(h_{2s}-h_1)}{\dot{E}}$) and volumetric efficiency $\eta_s = \frac{\dot{m}\cdot v_{in}}{\dot{V}_s}$ with enough accuracy. These variables were selected because from the compressor design point of view they are more significant than others like COP and cooling capacity.

- It shall only require information available from performance characterization tests.

- The parameters that appear in the model should have a clear physical interpretation to be able to translate the model results into internal compressor information.

- It should be able to evaluate the response of a compressor when the refrigerant is changed.

According to these objectives, the model assumes that the compression is isentropic, thus all the irreversibilities take place before the refrigerant goes inside the cylinder. This approach is closer to reality as the compressor size increases. The size of the compressors used in this work is enough to make this approximation reliable. Therefore, the developed model assumes that the evolution of the refrigerant through the compressor can be divided in the following sequence of processes:

- 1–2: Vapor heating due to motor cooling and mechanical losses dissipation.

- 2–3: Vapor heating due to the heat transferred from the hot side of the compressor (discharge) to the inlet flow.

- 3–4: Isoenthalpic pressure lost at the suction valve.

- 4–5: Isentropic compression from real cylinder intake conditions (leaks and possible condensation also appear in this part of the process).
- 5–6: Isoenthalpic pressure lost at the discharge valve.
- 6–7: Vapor cooling due to the heat transferred to the suction side.

In addition, two additional effects that reduce the total mass flow rate are also considered:

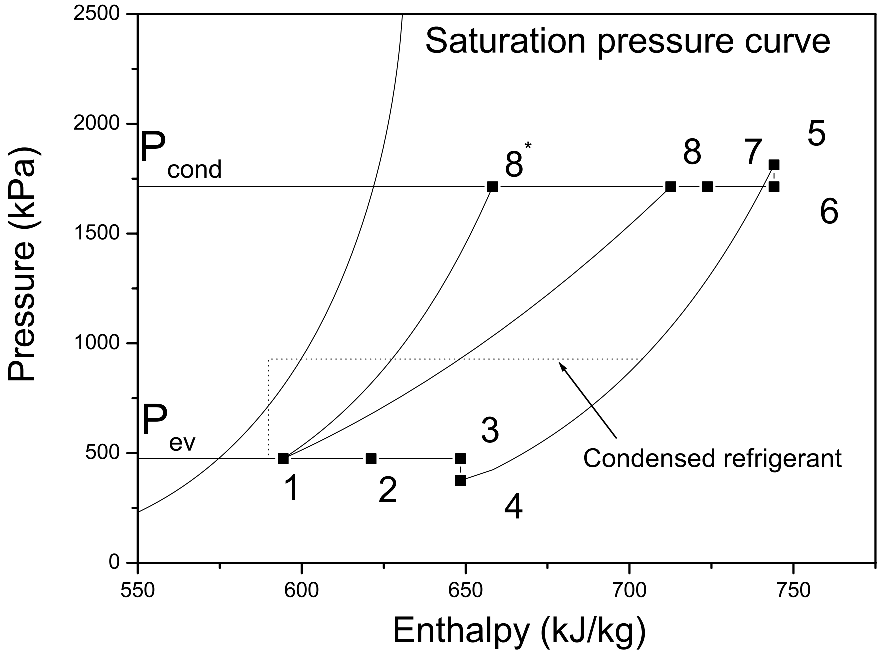

Figure 2.4: Modelled compressor cycle.

- The internal leakage of refrigerant through the piston rings. In order to simplify the treatment of this loss, it is considered as if all the loss of the circulating mass flow rate ($\overline{\dot{m}}_{leak}$) by means of leaks would take place at state 5. Therefore, the compressor is assumed to consume the work of compression for the circulating mass flow rate plus leaks ($\dot{m}_{in} + \overline{\dot{m}}_{leak}$).

- The possibility of condensation of a relatively significant fraction of refrigerant during compression on a cylinders cold spot in some conditions. Regarding this effect, it could appear as a consequence of the fact that the inlet valve is exposed to the relatively cold gas of the inlet. The valve plate may then be acting as a regenerative heat exchanger: if the valve surfaces had lower temperature than the condensing ones, then condensation may occur on the valve plate during compression and delivery phase. The condensate will evaporate during suction phase and, if we assume that this evaporation takes place on the valve plate, this will subsequently cool the valve plate surface, so that it is prepared for a new condensation process in the following compression cycle. Provided that the dynamics of these heat exchange processes are fast enough it will decrease the amount of gas delivered and have very similar net effect on the performance as an increased dead volume. It should be pointed out that this effect was added to the model to explain an anormal value in the predicted dead space ratio.

Other effects that influence refrigerant temperature before leaving the compressor (7-8 in fig. 1) such as heat release to the environment, oil heating and electric motor heating of the vapor among others are not considered in this study.

Under this considerations an estimation of the compressor and volumetric efficiencies is given:

$$\eta_s = \frac{\dot{m}_{in}}{\dot{V}_s \rho_1} = \frac{\rho_4}{\rho_1}\eta_{Sth} - \frac{\overline{\dot{m}}_{leak}}{\dot{V}_s \rho_1} - \frac{\overline{\dot{m}}_{pc}}{\dot{V}_s \rho_1} \tag{2.1}$$

$$\eta_k = \frac{\dot{m}_{in}\Delta h_{(1-8^*)}}{\dot{E}_k} = \frac{\Delta h_{(1-8^*)}\eta_{el}}{\Delta h_{(4-5)}(1+\frac{\overline{\dot{m}}_{leak}}{\dot{m}_{in}}) + \frac{\dot{E}_{mech}}{\dot{m}_{in}}} \tag{2.2}$$

where all the considered losses are used to calculate the diagram points 4 and 5 except the leaks and the condensation that are inserted directly inside the expressions of compressor and volumetric efficiency.

A detailed description of the considered losses can be found in appendix A. In the definition of the losses several parameters appear that depend on internal design characteristics such as the section of several pieces or effective areas, that have to be fixed by the user. Some of them (**K**(K_1, ..., K_8),η_{el}, $\frac{V_0}{V_s}$ according appendix notation) represents compressor design parameters difficult to determine, whereas **G** (G_1, ...G_n) stands for compressor design parameters that are easy to know like stroke (S), number of cylinders (nz), nominal speed (n), and the like. If for any reason some of these **G** parameters were not known, the model allows to regroup them inside the **K** parameters.

Once all the compressor design parameters **K** are known, the system of two equations can be solved for compressor and volumetric efficiencies, η_k and η_s, for a given working condition (P_1, P_{8^*}, ΔT_{sh}). Any solver for a system of non-linear equations can be employed. The results shown in this paper were obtained by using the Gauss-Seidel procedure [91]. With this algorithm, given an initial value of 0.5 for compressor and volumetric efficiencies , the convergence to the solution is typically reached in less than fifteen iterations.

As commented above, the parameters **K**, η_{el}, $\frac{V_0}{V_s}$ are difficult to estimate. A set of data for a number of working conditions obtained either from experiments or from manufacturer's catalogs are required to obtain the proper value of **K** by a fitting procedure. The developed fitting procedure to find the best estimation of **K** is explained in section (3).

From the best obtained values of **K**, it is possible then to know the value of compressor and volumetric efficiencies in conditions different from those tested. Besides, it is possible to obtain physically relevant information about the internal processes inside the compressor and to quantify the different losses.

A key assumption regarding **K** is that the different parameters are not significantly dependent on the test conditions or employed refrigerant. The obtained results indicate that this assumption is fairly reasonable.

2.3 Solution to the inverse problem

As explained in section 2, the last step to close the model is to estimate the compressor losses parameters **K** from experimental or catalog data. This is quite a critical issue in this kind of models because the dependency of the target functions on the parameters is non linear, and there is also some sort of indetermination, in the sense that several possible combinations of parameters could adjust the model properly with a final deviation in the predicted efficiency values which is smaller than the experimental uncertainties. In fact, several conventional non linear regression techniques (the standard routines in MsExcel, Origin, simplex algorithm [91]) were tested, yet they failed in the fitting process of the proposed model.

For this reason and to avoid possible problems arising from a step–by–step exploration of the parameter space (existence of a local minimum solution, too smooth dependence on certain parameters and the like), an heuristic algorithm based on a Monte Carlo type approach was designed. A review of the main trends in this field can be found in [92]. Although computationally not the most efficient these methods show great versatility and reliability.

A general scheme of the designed algorithm is shown in fig. D.2. In this scheme, the program starts by assigning pseudo-random values (according to the uniform distribution) to the parameters and the "best" combination of them to fit the compressor and volumetric efficiencies data is sorted out (this is called the *first process*). The routine used to generate the pseudo-random numbers was the one proposed by Park and Miller [93]. This routine has a long enough period for this specific application. To select the "best" combination of parameters, an error function or residue (ϵ) must be defined. In this case, the Euclidean norm weighted by the standard deviation of each experimental point i was selected:

$$\epsilon = \sum_i \frac{\sqrt{\eta_k^2(x_i) - \eta_{kexp}^2(x_i)}}{\sigma(x_i)} + \frac{\sqrt{\eta_s^2(x_i) - \eta_{sexp}^2(x_i)}}{\sigma(x_i)}$$

The set of parameters with a lower value of ϵ is selected as a solution to the first process.

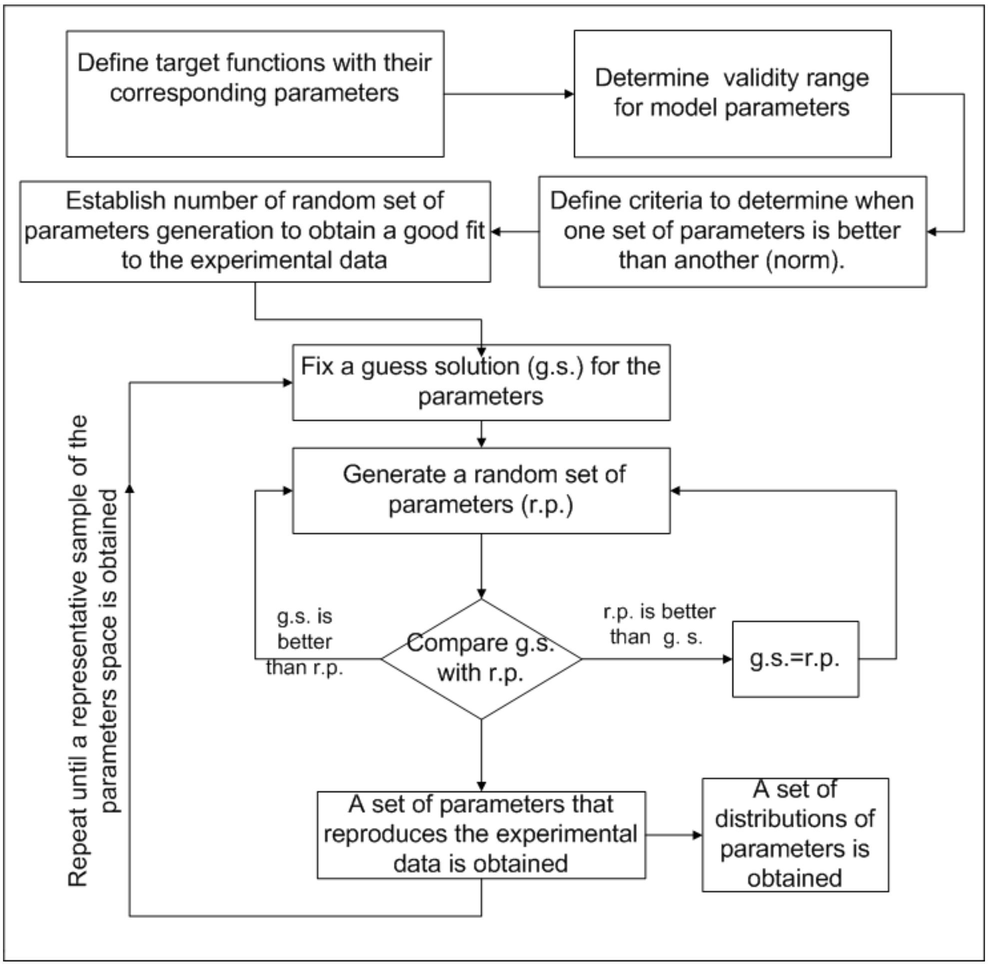

Figure 2.5:

Nevertheless, as a consequence of experimental errors, intrinsical errors associated to the model and the nature of the mathematical functions involved, if the process is repeated, a different set of parameters is obtained as a solution which may also give a good value for the error function ϵ.

Therefore, this process is repeated until a representative map of the solutions in the parameter space is obtained (this is called the *second process*). This means that the probability distribution of the solutions for each parameter is obtained.

The result of this process is a set of probability distributions for each of the model parameters. From these distributions, the most probable value for each parameter is selected as the best fit value.

Some comments should be made regarding this scheme:

- An interval in which the value of the parameters must be found, must be defined. As a result of the intrinsical stability of the method, this is not a critical point in the model, yet a good selection of this interval reduces the number of iterations needed to find a suitable solution.
- Preliminary studies have been developed to determine the proper number of iterations in the first and the second process. For the first process, this number is reached only if, increasing the number of iterations, the order of magnitude of ϵ does not change. For the second process, this number is reached if, by increasing the number of iterations, the obtained parameter distribution function does not change.
- To reduce the high computational cost linked to the direct use of REFPROP [94] in the evaluation of the thermodynamical properties of the refrigerants, an approximation based on linear interpolations of bidimensional meshes from REFPROP was employed (see [95] for details).

CHAPTER 3

Test rig

The test rig used to develop the experimental research is located in the *Instituto de Ingenieria Energetica* at *Universidad Politecnica de Valencia* and is designed to allow characterization of the different components of a refrigeration system. To allow this, the pressure at the compressor inlet and outlet, the superheat of the gas at compressor inlet and the subcooling at the inlet of the expansion valve are controlled in the test rig. A basic scheme of the test rig is presented in 3.1. In the particular case of compressors this test rig is designed according to fully comply with standard ISO 918 [96] for compressor characterization.

Later on, the different components of the test rig are described.

3.1 Main test rig parts

3.1.1 Compresor

As it is commented above, the test rig allows to fix the compressor inlet and outlet pressure and the vapor superheat in the compressor inlet, the compressor behavior can be studied in any operating conditions.

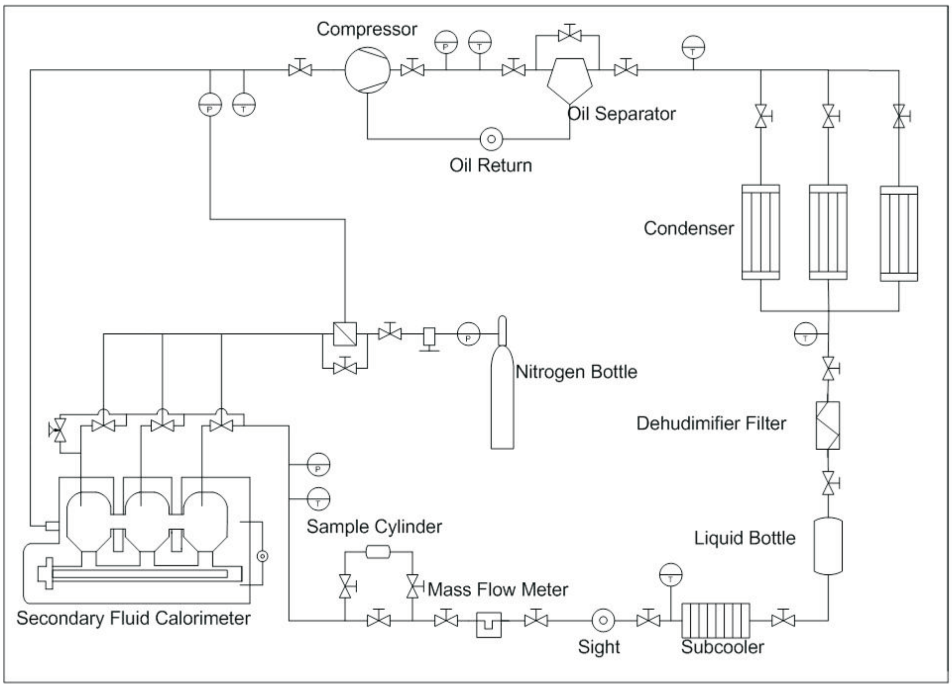

Figure 3.1: Test rig.

Compressors are mounted on several dampers (fig.3.2) termed "*silent block*" fixed on a piece of concrete to isolate the rest of the system from the vibrations that the compressor cause. The compressor is connected to the rest of the test rig by two rotolock junctions that allow to change the tested compressor in relatively short time.

3.1.2 Condensers

The condensation takes place in three plate heat exchangers (figure 3.2) in parallel. In this way, the heating capacity of the system can be fitted to the requirements of the different tests and compressors.

The plate heat exchangers work with a closed water circuit which also exchange heat with an open water circuit. In this way, the fluctuations derived from direct heat

Figure 3.2: Compressor (left), condensation unit (right)

exchange of the refrigerant circuit with water from the net are reduced. In addition, in order to further damp this temperature fluctuations and improve the stability of the condensation pressure, a water container was installed. This water container increases the amount of water in the closed circuit . The scheme of the water circuit is presented in figure 3.3.

3.1.3 Pneumatic expansion valves.

The expansion valves connect the high and the low pressure sides. These valves control the evaporation pressure independently from the caloric charge of the evaporator. Avoiding overcharges of the electric motor of the compressor in the start up. However, it has the backward of using the evaporator surface totally only when the working conditions are closer to the steady state.

3.1.4 Evaporator

The evaporator of this test rig is a secondary fluid calorimeter (figure 3.4). The calorimeter is composed of a container isolated from the environment which is divided in three parts. A coil is placed inside each part allowing heat exchange

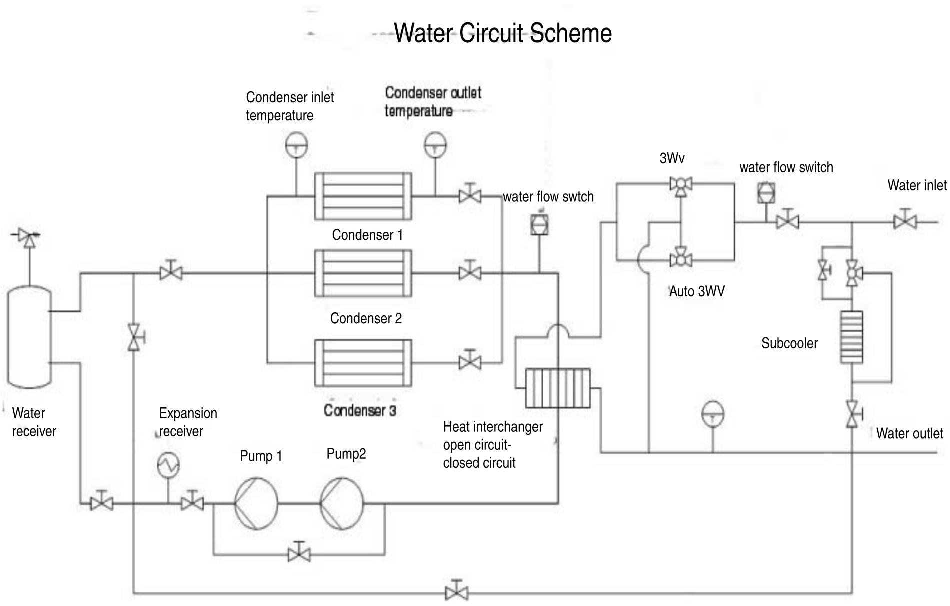

Figure 3.3: water circuits

between the primary and the secondary calorimeter fluid (R134a). The division of the calorimeter in three parts allows the control of the cooling capacity of the test rig according to the tested compressor size.

To allow a steady heat transfer process in the calorimeter, the secondary refrigerant is in saturation conditions and three electric resistances immersed in the liquid phase of the secondary fluid provide the required heat according the test conditions. The measurement and control of the electric consumption of these resistances assists superheat control on the compressor inlet and , after a proper calibration of the electric and thermal losses, allows to derive the mass flow rate trough the refrigerant circuit as is mentioned in the standard for compressor characterization [96].

The calorimeter has also the following security devices:

- Explosion proof heaters.
- Liquid sight.

Figure 3.4: Pneumatic expansion valves (left), calorimeter (right)

- Switch off liquid level.
- Shut down for higher temperatures.
- Fill obus.
- Protection device for overpressures.

3.2 Further test rig parts

3.2.1 Pipes

They join the different components of the test rig. In the beginning of this study, they were redesigned to allow a proper oil draw through the test rig. The following modifications were implemented according to [97]:

- Resizing of the pipe diameter to allow a minimum refrigerant velocity of 7.5 m/s in all the test conditions.
- Leaning the pipes in the flow direction to stimulate the oil flow.
- Removing the oil traps of the test rig and adding proper U junctions to increase the oil circulation in the vertical pipes and avoid the possible oil return to the

compressor through its inlet.

In addition, a receiver installed in the calorimeter outlet to prevent that some liquid refrigerant could reach the compressor inlet was removed to avoid that some oil could be trapped in it. Furthermore, for compressor safety, three pneumatic valves were installed at the inlet of each expansion valve that were switched on and off during compressor start up and shut down.

3.2.2 Oil separator

It is placed at the compressor outlet and its function is to avoid that the oil expelled by the compressor reaches the rest of the test rig components. This device retains the oil up to certain level and then the stored oil is reinjected trough the compressor inlet avoiding that the compressor could operate without lubrication.

The oil separator was bypassed to carry on oil circulation rate measurements in the test rig. This device was also used to measure the amount of oil expelled by the compressor in the start up. With this purpose, a gauge and a capillary tube were mounted in the rotolock connections that join the oil separator to the rest of the circuit.

Figure 3.5: Oil separator (left), unmounted oil separator (right)

3.2.3 Storage drum

It is placed after the condenser (fig.3.6) and its functions are to store vapor that could leave the condenser and to store charge in some working conditions.

3.2.4 Drying filter

It removes dampness that may exist in the test rig. This device was bypassed during the OCR tests to avoid that oil could be trapped in it.

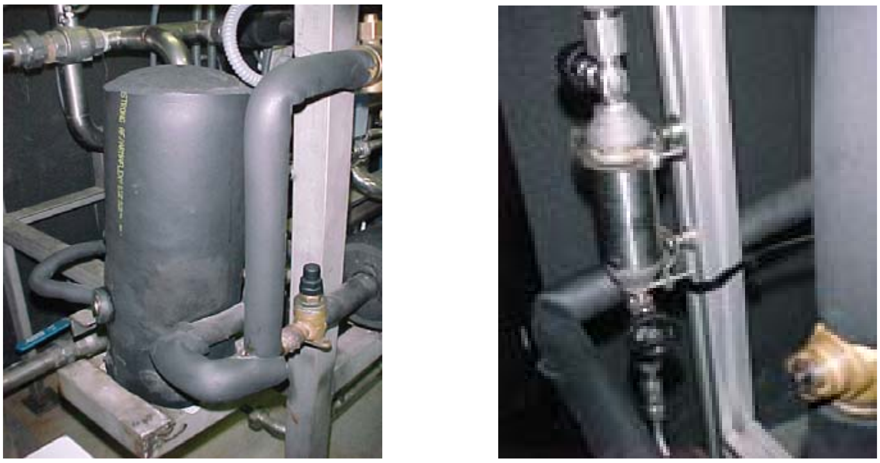

Figure 3.6: Storage drum (left), sample cylinder (right)

3.2.5 Subcooler

It is a plate heat exchanger placed after the storage drum. Its functions are on the one hand to assist to subcooling control at the inlet of the expansion valves and on the other to avoid that, as a consequence of the pressure drop through the pipes, vapor could arrive to the mass flow meter or the expansion valves.

3.2.6 Sample cylinder

This cylinder is placed between the subcooler and the expansion valves, in a liquid line parallel to the main refrigerant circuit line (fig. 3.6). The function of the cylinder is to allow to extract homogeneous refrigerant samples from the test rig. It was used to determine the oil circulation rate through the installation according to the standard refereed in [98].

3.3 Control

The test rig has four control loops to control the basic parameters of a refrigeration system: condensation and evaporation pressure, superheat and subcooling.

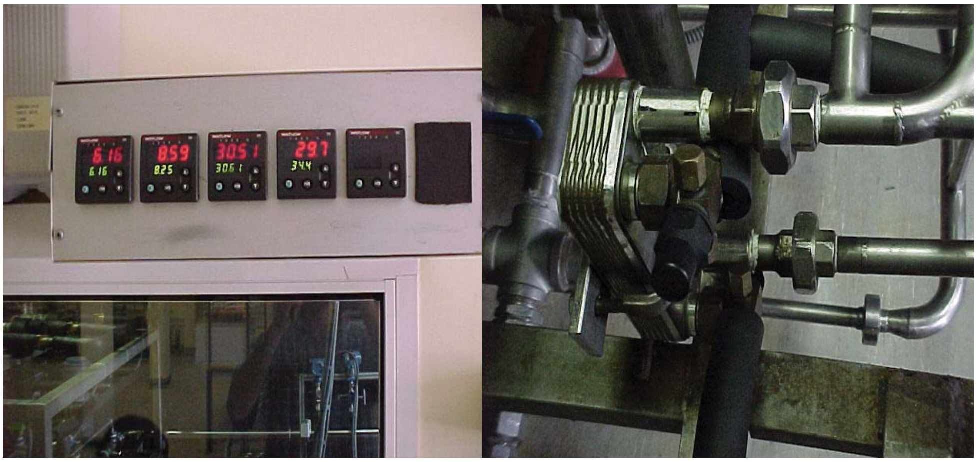

Figure 3.7: Watlow 96 controlers (left), subcooler placed in the condenser outlet (right)

The controllers used are PID brand Watlow serie 96. They are quite general devices and with a widely adaptable working range. The parameters of these controllers were fitted according to the iterative process of Ziegler-Nichols [99].

The control loops allow to fix the variables of the system in the range shown in the table 3.1.

Temperature	$\pm 0.1°C$
Evaporation pressure	$\pm 20 mbar$
Condensation pressure	$\pm 20 mbar$

Table 3.1: Maximum variable variation obtained by the control loop.

The different control loops will now be described in further detail.

3.3.1 Condensation pressure control loop

The mass flow rate of the open water circuit is used to control the condensation pressure.

This water circuit has two three ways valves (fig.3.8) that control the amount of water passing through the plate heat exchanger that exchanges heat with the closed water circuit. One of this valves is controlled manually through a rheostat, and allows a first control of the saturation pressure. The fine tuning is performed by the other three ways valve which is controlled by a PID. This PID receives the compressor outlet pressure and according to the prefixed parameters, controls the opening of the valve with a signal from 0-10 V. The valves used are the SKD62 and SQX61 models of Landis & Steafa.

3.3.2 Evaporation pressure control loop

The evaporation pressure is controlled by three presostatic valves installed at the calorimeter inlet, they are pneumatic valves operated by dry nitrogen (99% pure) to avoid dirt problems.

Nitrogen pressure on the valves is controlled by a pressure regulator of the brand SMS model ITV2050, with a nominal range between 3-8 bars although for especial applications this range could be slightly changed (fig. 3.8). This device is in direct communication with a PID and is the responsible of the evaporation pressure control by regulating the opening of the pneumatic valves (fig. 3.8).

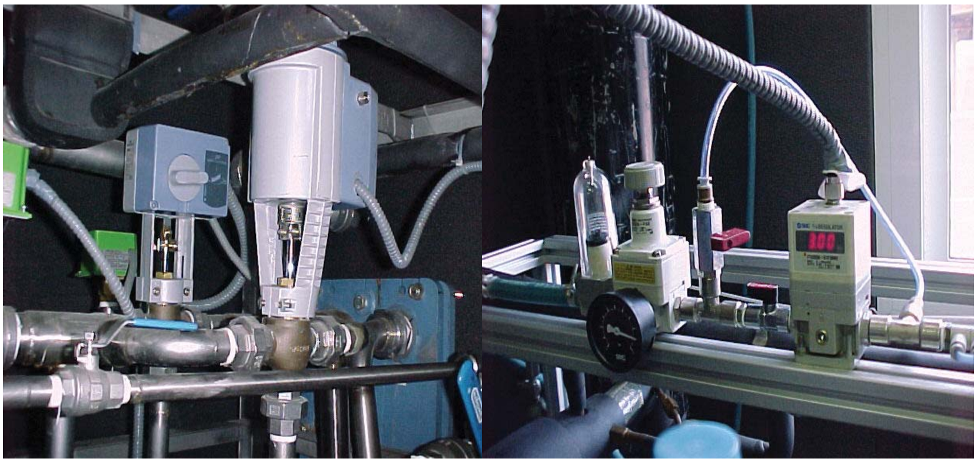

Figure 3.8: Three ways valve responsible of the control of the condensation pressure (left), pneumatic device responsible of the control of the evaporation pressure (right)

3.3.3 Superheat control loop

This parameter is controlled by the calorimeter where the secondary fluid exchange heat with the primary fluid.

In order to obtain the desired superheat, the calorimeter has three resistance groups connected in triangle with a total nominal power of 36,25 kW. The electric power supplied to them is controlled by a PID that has as input parameter the pressure of the secondary refrigerant.

In order to allow an easier control the power supplied to the resistances is divided in two fixed stages on/off of 8.75 kW in mode off/on and one variable stage of 18.75 kW. The last resistance is controlled by the PID by means of an analogic signal which is sent to an inverter of the brand Watlow model IN-a-mite solid State Power control.

3.3.4 Subcooling control loop

The way in which this parameter is controlled is similar to the one used to control the saturation pressure, although in this case, as a consequence of the fact that the working range is not so wide, only one three ways valve is required to perform the regulation.

To control subcooling, some water coming from the open water circuit is deviated by a manual valve to the subcooler already described section 3.2.5 (fig. 3.7). A three ways valve, placed after the manual one and before in the subcooler controls the water mass flow rate and is connected with a PID which performs the automatic control of subcooling.

It should be said that this parameter is the less stable one as a consequence of the fact that the system exchange heat directly with water from the net. In any case, for compressor tests this is not a very important issue.

3.4 Sensors and data acquisition

Pressure and temperature are measured in the points indicated in fig. 3.1. Temperature sensor are platinum RTD PT-100 with an intrinsic error of 0.1 K. The pressure sensors were Fisher-Rosemunt 3051 (Fig.3.9) with an error of 20 $mbar$.

Mass flow rate is measured with a Coriolis [1] Fisher-Rosemount CMF025M (fig. 3.9) which gives a current signal proportional to the mass flow rate. It is required that there are not bubbles in the flow to guarantee a right mass flow measurement (this was the main function of the subcooler in the developed tests). Another characteristic of these elements is that they are very sensible to vibrations, for this reason all the test rig was isolated from the compressor vibrations as described in section 1.

[1] An accurate measurement of mass flow rate is the most important objective in the test that were developed, for this reason, the measurement of this magnitude was also performed with the calorimeter. The divergence obtained between both methods was lower than 1% in almost all the test. This fact gives trustability to the obtained results

Figure 3.9: Fisher-Rosmount 3051 pressure sensors mounted in the test rig (lef°. t), mass flow meter type coriolis Fisher-Rosmount CMF025M (right)

The energy consumption of the compressor and the calorimeter are measured with a Schlumberger Quantum D-200 power meter.

All the measurements were centralized in a multiplexer HP34970A connected to a PC by means of a GP-IB port.

HP 34970A is a modular system for data acquisition with a significant record capacity due to its multiplexor by relays of 22 differential channels for each acquisition card with a maximum of three slots. A digital internal multimeter with an accuracy stability removing noises of 6 μV is another feature of this device. Each inlet recognizes units defined by the user, in a linear scaling (other scales are also allowed but defined by an external software) and alarm limits.

To develop the acquisition and analysis of data of this work, a program in VEE [100] was developed. This program allows to obtain the results of a characterization test (COP, cooling capacity, efficiencies) in real time.

The developed program also incorporates an error analysis module. The standard

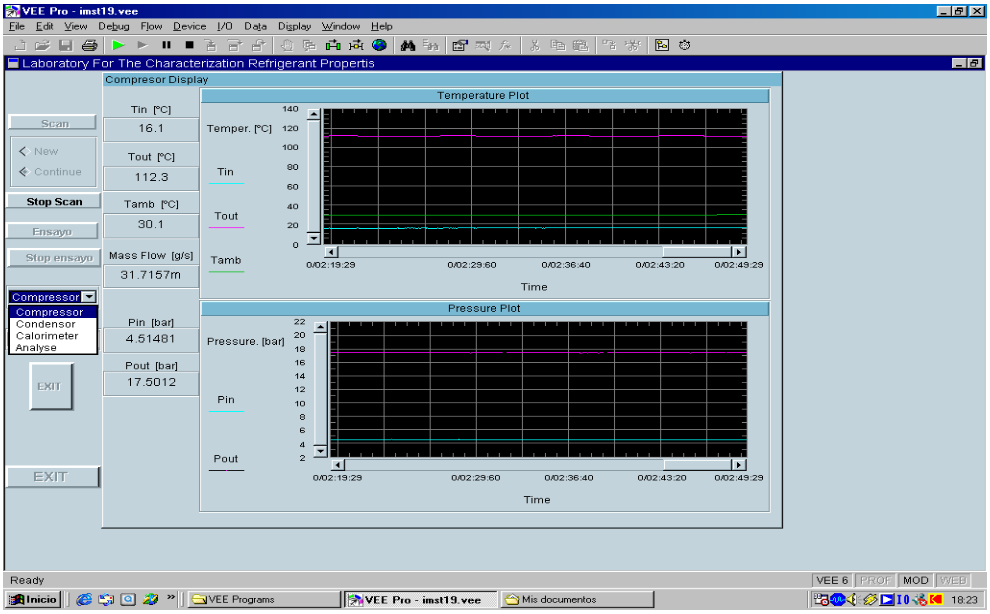

Figure 3.10: Veeprogram

deviation of experimental data was taken as the error of the variables obtained experimentally such as pressure or temperature. The rules of error propagation were applied for indirectly measured variables:

$$\epsilon(h) = \sqrt{\mid \frac{\partial h}{\partial x}\epsilon_x \mid + \mid \frac{\partial f h}{\partial y}\epsilon_y \mid + ...} \tag{3.1}$$

where h is a function of x, y...

There are some indirect variables like enthalpy or entropy whose dependency on the experimental variables are not given in a explicit mathematical expression. In this case, the derivatives were substituted by increments of small step resulting:

$$\epsilon(h) = \sqrt{\mid \frac{\Delta h}{\Delta p}\epsilon_p \mid + \mid \frac{\Delta h}{\Delta t}\epsilon_t \mid + ...} \tag{3.2}$$

where the function is calculated in $h(p)$ and $h(p + \Delta p)$ and $h(p + \Delta t)$. In our particular case, the step was defined arbitrarily as 1% of the absolute value of the experimental variable.

It should be noticed that the disposal of these errors in real time allows to evaluate the quality of the test in a short time. In the next page an example of a the results page of a test is shown.

Component characterization laboratory
IMST – Institute of Energy Engineering
Polytechnical University of Valencia
Edificio I–4, Camino de Vera, 14
46022 Valencia – Spain

Compressor Characterization Report

Proyect		**Compressor**	**Meas. standard**
Maneurop MTZ compressors with R290		MTZ64	ISO 917
Date/Hour	**Refrigerant**	**Oil**	**Person in charge of test**
4-25-03 / 1:48:37	R290	POE 160 PZ	Emilio Navarro

Reference Conditions

Evap. Temp.[C] : 0	Cond.Temp.[C] : 50	Superheat.[K]: 11

Measurement Results

Inlet and oulet Compressor Conditions			
Inlet Temp.[C]:	11.15 ± 0.01 [C]	Inlet Press.[bar]:	4.741 ± 0.006
Outlet Temp.[C]:	81.45 ± 0.03 [C]	Outlet Press.[bar]:	17.14 ± 0.01
Calorimeter Conditions			
Inlet Calorm. Temp.[C]:	26.1 ± 0.07 [C]	Inlet Calorm. Press.[bar]:	16.96 ± 0.01
Outlet Calorm. Temp.[C]:	10.43 ± 0.02 [C]	Outlet Calorm. Press.[bar]:	4.552 ± 0.01
134a Temp.:	11.131 ± 0.001 [C]	134a Press.:	4.3055 ± 0.0002
Condenser Conditions			
Inlet Refr. Temp.[C]:	78.97 ± 0.02	Oulet Refr. Temp.[C]:	48.27 ± 0.02
Inlet Water Temp.[C]:	38.71 ± 0.02	Outlet Water Temp.[C]:	40.1 ± 0.02
Other Results			
Casing Mean Temp. [C]:	37.1	Oil Sump Temp. [C]:	53.1 ± 0.04
OCR [%] :	—		
File cointaining Comp. noise:	sound7.wav		

Characterization Results

Compressor and Calorimeter Electric Data			
Phase 1 Intensity [A]:	8.00 ± 0.03	Phase 1 Voltage [V]:	391 ± 2
Phase 2 Intensity [A]:	7.2 ± 0.4	Phase 2 Voltage [V]:	391 ± 2
Phase 3 Intensity [A]:	7.95 ± 0.03	Phase 3 Voltage [V]:	392 ± 2
Comp. Active Power [kW]:	3.733 ± 0.007	Comp. Reactive Power [kW]:	3.55 ± 0.03
Calorimeter Active Power [kW]:	11.57 ± 0.05	Calorimeter Electric losses [kW]:	0.0103 ± 0.0005
Calorimeter and Coriolis massflowmeter results			
Calor. Mass Flow [kg/s] :	0.0358 ± 0.0008	Coriolis Mass flow [kg/s]:	0.03596 ± 0.00008
Difference [%]:	0.53		
Performance			
Cooling Capacity [kW]:	9.21 ± 0.01	COP :	2.468 ± 0.003
ARI Cool. Capacity [kW]:	10.112 ± 0.01	ARI COP :	2.71 ± 0.003
Further Performance Results			
Spec. Vol. Capacity [kW m^{-3}]	2506.19	Compression Work [kW]:	3.643 ± 0.004
Vol. Efficiency:	0.7062 ± 0.0003	Compressor Efficiency:	0.6148 ± 0.0002
Isoentropic COP:	4.0136	Heat losses:	2.425 ± 0.005 %
Pressure ratio:	3.616 ± 0.006	Isoentropic work [kW]:	624.422

Comments

Signature:

Figure 3.11: Example of test report

CHAPTER 4

Experimental results

4.1 Introduction

Five reciprocating compressors for small and medium capacity heat pumps were tested in the test rig described in chapter 3. These compressors were working with propane and R407C as refrigerants and a POE ISO 32 and a MO CG68 oil as lubricants.

The developed experiments had the following objectives:

1. To extend the data base of compressors operating with propane.
2. To analyze the possible differences of a compressor operating with propane and R407C from an empirical point of view.
3. To analyze the suitability of using common POE oil (HFC common lubricants) with propane.
4. To obtain experimental characterization data to fit and verify the model explained in chapter 2.

Propane has higher solubility in oils than other more common refrigerants such as HCFCs or HFCs, as a consequence of this, the third objective analyzes the possible effects that could appear when a hermetic compressor designed to work with HFCs is working with propane and the feasibility of changing the refrigerant of a test rig working with R407C for propane.

The fourth objective will allow to develop a performance analysis of the studied compressors in terms of the model losses to understand in a better way the observed differences in a compressor working with different refrigerants, in our particular case propane and R407C.

4.2 Developed measurements

The main characteristics of the selected compressors are shown in table 2.1. The test matrix designed to test these compressors is shown in fig. 4.1 covering a wide rage of working conditions for a constant superheat of 11.1 K. According the ISO procedure subcooling is not an experimentally significant parameter for compressor characterization, it only has to be high enough to avoid the presence of bubbles in the mass flow meter or the expansion valves. A theoretical value of 8.3 K. was taken for this parameter in the calculation of COP and cooling capacity. So that capacity was directly comparable to ARI test conditions.

When the test rig is operating in steady state, compressor electric consumption, mass flow rate (with calorimeter method and coriolis method) and inlet and outlet compressor temperature and pressure are registered and the following compressor characteristics calculated:

- COP (Refrigeration performance coefficient):

$$COP = \frac{\dot{m} \cdot h_{ev}}{\dot{E}_k}$$

- Cooling Capacity:

$$Q_{ev} = \dot{m}_{in} \cdot h_{ev}$$

- Compressor efficiency:

$$\eta_{comp} = \frac{\dot{m}_{in}(h_{8^*} - h_1)}{\dot{E}_k}$$

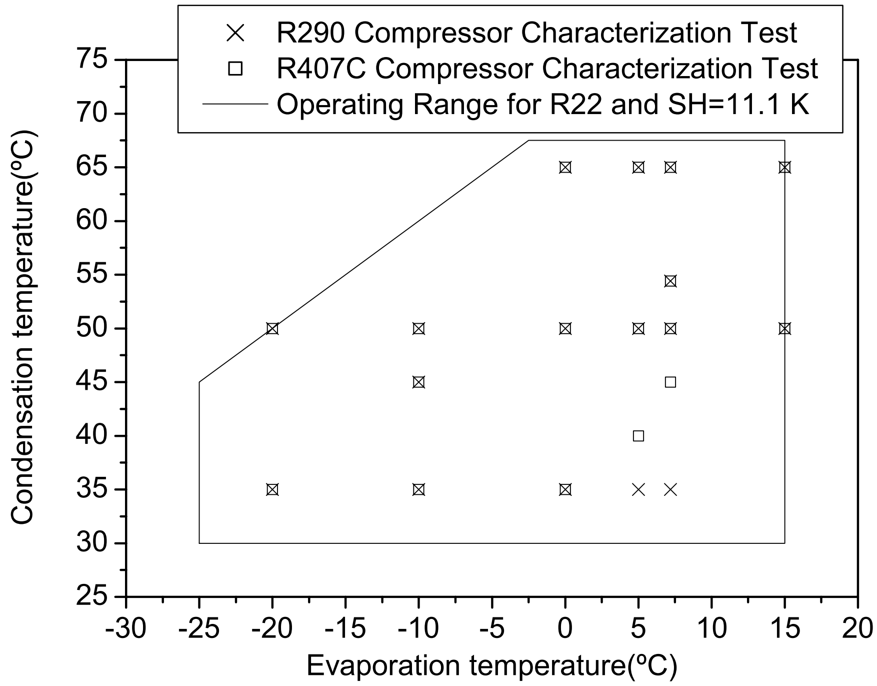

Figure 4.1: Test matrix used.

- Volumetric efficiency:

$$\eta_{vol} = \frac{\dot{m} \cdot \upsilon_{in}}{\dot{V}_s}$$

Mainly, these parameters characterize the global compressor performance. NIST REFPROP [94] was used to calculate refrigerant thermodynamical properties.

According to this procedure, all the compressors were tested working with propane. Additionally, the compressor ST were tested working with R407C to evaluate, from an experimental point of view, both refrigerants and to test the model reliability to describe compressor performance when the refrigerant is changed. The compressors use the comercial oil used by the manufacturer with HFCs (POE ISO32) in all the tests.

Regarding the oil experiments, measurement of oil circulation rate through the test rig (OCR) was considered as a good first approach of the possible suitability of the used oil since:

- It gives information about the amount of oil expelled by the compressor when it is working in steady state. If this amount of oil is too different between both refrigerants (more than 5-10 %) possible compressor lubrication problems and some heat transfer problems in the heat exchangers may be expected [101], [102].

- It gives information regarding the proper draw of the oil through the test rig to the compressor.

These oil measurements were developed according the standard ANSI/ASHRAE [98].

The used POE oil has a lower viscosity and surface tension than mineral oil commonly used with propane. This property joined to the higher propane solubility related to R407C may produce an intense foaming effect in the start up process that could due to compressor degradation. As a consequence of this, additionally, a technic to measure the oil expelled by the compressor in the start up was designed and a comparison between mineral and POE oils was developed. A thermography camera recorded these experiments to monitor the compressor casing temperature evolution during the start up.

4.3 Comparative study of several compressors of different capacities working with propane and R407C

A detailed review of this experimental work was published in international journal of refrigeration (see appendix A). The most important conclusions of this experimental work were:

- Propane improves compressor efficiency approximately 3%- 4% at low and mean pressure ratios.
- The electric motor efficiency is similar for both refrigerants. Therefore, it is not a determinant influence factor to explain the observed differences.
- Propane showed better behaviour at low and medium pressure ratios compared with R407C. That tendency is inverted at high pressure ratios (see A.8-A.11).
- OCR is independent of the test conditions inside the range of the test matrix used and for a constant superheat. However, this magnitude could be increased considerably (100%) for low superheats.
- OCR is similar for both refrigerants and never higher than 2%.
- The use propane in a HFCs system lubricated with POE oil is reliable according the developed experiments.

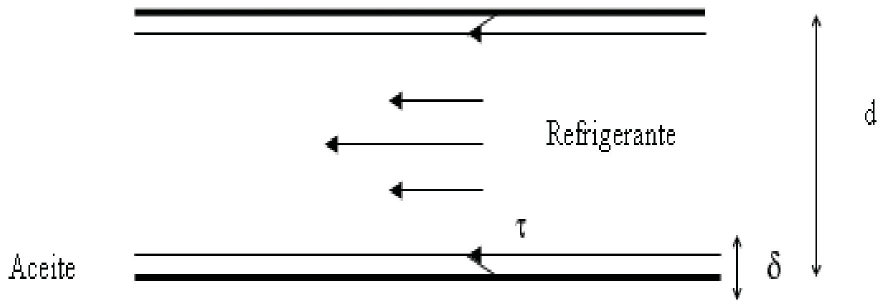

Figure 4.2: Oil flow in a pipe

To gain some insight in the OCR results obtained, it might be interesting to make an estimation of the oil film thickness that these results imply if it is supposed that all the amount of oil is flowing through the pipe walls as a separated flow. This estimation gives information about the maximum oil film which could difficult the heat transfer in the evaporator. Looking fig.4.2, taking in consideration mass flow rate and OCR and making an estimation of the lubricant density and viscosity, the oil film thickness can be calculated as it is shown here :

$$\dot{m}_{oil} = \frac{v_{oil}}{2} \cdot \delta \cdot \pi \cdot d \cdot \rho \tag{4.1}$$

where the oil velocity is given by:

$$v_{oil} = \frac{\tau \cdot \delta}{\mu} \tag{4.2}$$

and the stress tensor:

$$\tau = \frac{dp}{dz} \cdot \frac{d}{4} \tag{4.3}$$

where the derivative of the pressure is given by

$$\frac{dp}{dz} = f \cdot \rho_{ref} \cdot v_{ref}^2 \cdot \frac{1}{d} \tag{4.4}$$

$$f = 0.092 \cdot Re^{-0.2}$$

The oil mass flow rate is also a known magnitude from the total mass flow rate and the OCR

$$\dot{m}_{oil} = \dot{m}_{ref} \cdot OCR \cdot \frac{\rho_{oil}}{\rho_{ref}} \tag{4.5}$$

Being the thickness of the oil layer

$$\delta = \sqrt{\frac{\dot{m}_{oil} \cdot 2 \cdot \mu}{\tau \cdot \pi \cdot d \cdot \rho}} \tag{4.6}$$

If the pipe is placed vertically the development is the same but adding the term related with the gravity acceleration.

The obtained results for the oil thickness is approximately 0.5 mm. (this estimation was made at the conditions (0ºC, 50ºC) and a mean value of OCR of 1.5%). This value seems to be too small to have a significant influence in the global installation performance (see [101], [102]).

4.4 Comparative study of a compressor working with propane and lubricated with POE and MO

This part of the work tried to compare the compressor behavior working with propane and lubricated with mineral and POE oil. A detailed description of this part of the work can be found in appendices B and C. This study has been developed in compressor SO (in the appendix literature) and consisted of:

- Measurement of compressor and volumetric efficiency.
- Measurement of OCR.
- Measurement of start up oil emission at different conditions.

The aim of the first to points was to find possible differences when the compressor is working in steady state. The third test was related with the study of possible lubrication differences in the start up.

There is not too much experience using POE as a lubricant in refrigeration systems working with propane. The low surface tension of POE oil compared with mineral oil plus the high propane solubility in oils may due to an intense foaming effect in the start up that could potentially produce compressor lubrication problems. This phenomena might down to a faster compressor deterioration in commercial heat pump systems in which the device is often switched on and off.

Since propane has been used traditionally with mineral oil a comparative study between both oils has been developed to observe if POE oil has a significantly different behavior that could have any disadvantage.

The conclusions of this work were:

- POE and MO oils presented a similar behavior regarding compressor and volumetric efficiency.
- OCR values are 100% higher for POE oil, nevertheless the amount measured is not significant in any case.

- The amount of oil expelled by the compressor in the start up were similar for both types of oil. The lower surface tension of POE oil has been compensated by the higher solubility of propane in mineral oil.

Therefore, the used POE oil is a suitable lubricant to be used with propane. This fact has two direct consequences: the direct drop in of a system working with HFCs without requiring an oil cleaning of the system is feasible and the refrigerant charge of a system working with propane is reduced due to the lower solubility of propane in POE oil than in MO.

It should be pointed out that the start up experiments were quite complicated as a consequence of the intrinsic difficulty of the processes involved . Nevertheless, the developed experimental procedure seems to be accurate enough to the purposes of this work.

The amount of oil in the pipes between the compressor and the oil separator used in the start up as receiver can be evaluated with the estimation given in last section about the thickness of the oil layer ($M_{oil} = \pi \cdot L_{tub} \cdot [d^2 - (d - \delta)^2] \cdot \frac{\rho_{oil}}{4}$). Thus, The amount of oil in the pipes between the oil separator and the compressor were approximately 20 g. This value was in perfect agreement with some preliminary results obtained in the test rig.

In addition to this measurements, a thermographic study of the compressor in the start up was developed (see appendix C). During this tests a temperature gradient was observed in the compressor discharge pipe, this could be a consequence of a temperature difference between the refrigerant and the lubricant. This may open the possibility of using this technique to the study of multicomponent biphasic flow in this line of the test rig. Unfortunately, time limitations of that project did not allow to develop a deeper study in this direction.

CHAPTER 5

Model results

5.1 Introduction

The objective of this part of the thesis is to describe the procedure and results obtained using the model developed in the last sections in combination with the compressor characterization test results described along chapter 4. This will allow to verify the model reliability and robustness as well as to obtain insight into the compressor operating characteristics.

To show the model results this chapter is organized as follows. First, the physical significance of the values obtained for the parameters given by the model is discussed. Including, an analysis of the coherence between the results obtained for the different compressors. Subsequently, a global analysis of the relative influence of all the losses is performed and finally the capability of the model to predict compressor performance with a refrigerant different to the one used for parameter adjustment is evaluated. At the end, an analysis of the observed differences between propane and R407C is done according to the model.

5.2 Model validation

The model described in chapter two includes some internal parameters grouped in the vector **K** that, in principle, are unknown. The Monte Carlo fitting procedure which is in the core of our model, finds the best approximation to these parameters using only experimental data of compressor and volumetric efficiencies that are commonly available in catalogues.

The first test of model consistency was to check that the obtained parameters values are physically meaningful. In a first step, the condensation effect was not taken into account and the value obtained for the dead space ratio was too large compared with the information that was available has about this compressor. In order to avoid this divergence, the condensation term was added to the model improving the predictions of the dead space ratio. A detailed analysis of the obtained values for all the parameters K_i compared with their expected value is explained in section 5 of appendix D. In this section it can be seen that all the parameters except K_4 (pressure drop at the cylinder outlet) show a consistent value.

The fitting procedure was repeated for all compressors tested and the resulting constants are shown in table 5.1. Compressor and volumetric efficiencies were adjusted with an error lower than 3% in almost all operating conditions in all the compressors. The parameter values obtained for all the compressors were similar. However, slight tendencies were appreciated in the parameters depending on the compressor size. In any case, they were so small that it is not easy to deduce if they are a consequence of a real tendency within the model uncertainties.

Once the physical meaningfulness of the model was shown, an analysis of the obtained results has been done and the following conclusions regarding each parameter has been obtained:

- K_1 : This parameter represents gas heating from motor cooling and mechanical losses dissipation. It is coupled to parameter K_2 (gas heating due to the heat transferred from the hot side of the compressor - discharge- to the inlet flow) in such a way that it is not possible to obtain their values separately. Thus, a good estimation for this K_1 is needed, and, assuming that K_1 should not depend on the analyzed compressor, its value was fixed at 0.9 for all compressors, which

	SO	LO	ST	LT
K_1	0.9	0.9	0.9	0.9
$K_2^{'}$	3.02	2.88	2.80	2.78
K_3	$2.08 \cdot 10^7$	$1.09 \cdot 10^7$	$1.94 \cdot 10^7$	$0.98 \cdot 10^7$
K_4	$3.7 \cdot 10^8$	$2.39 \cdot 10^8$	$3.85 \cdot 10^8$	$2.54 \cdot 10^8$
$K_5^{'} \cdot D^2$	$0.96 \cdot 10^{-6}$	$0.89 \cdot 10^{-6}$	$0.95 \cdot 10^{-6}$	$0.86 \cdot 10^{-6}$
V_d/V_s	$7.12 \cdot 10^{-2}$	$5.71 \cdot 10^{-2}$	$6.77 \cdot 10^{-2}$	$5.32 \cdot 10^{-2}$
$(V_d/V_s)^{'}$	$4.3 \cdot 10^{-2}$	$3.5 \cdot 10^{-2}$	$3.9 \cdot 10^{-2}$	$3.4 \cdot 10^{-2}$
K_6	1.70	1.65	1.73	1.63
K_7	$6.08 \cdot 10^{-2}$	$13.05 \cdot 10^{-2}$	$5.11 \cdot 10^{-2}$	$5.19 \cdot 10^{-2}$
$K_8[kW]$	0.113	0.148	0.2052	0.483
Electrical motor eff.	0.88	0.89	0.86	0.90

Table 5.1: Parameter values for the compressors under study.

means that 90% of the electrical and mechanical losses are wasted increasing the temperature of the inlet gas. This assignment provides very good results that justify the assumption.

- K_2 : As explained earlier, this parameter represents gas heating due to the heat transferred from the hot side of the compressor (discharge) to the inlet flow. The difference between the value of K_2 for all the compressors is small enough to consider that it is included within the parameter error. In any case, the value of K_2 shows a slight tendency to decrease as the compressor size increases. The differences of this parameter among the compressors leads to differences in the increase of the suction gas temperature at the cylinder inlet of approximately 1 K which may be considered as a slight effect.

- K_3 : This parameter represents the pressure lost through the inlet valve. Similar values were obtained for the different compressors. In any case, the compressors with a larger stroke (LO, LT) show a slightly smaller value for this constant. The differences in the value of K_3 between the long and short stroke were also found in the value of K_4, meaning that the pressure losses through the valves are higher for shorter strokes. This tendency in the values of K_3 and K_4, taking into account the fact that pressure losses were assumed to depend on the square of the gas velocity, allows one to draw the conclusion that they actually depend on the velocity with an exponent slightly higher than the supposed 2.

- K_4 : This parameter represents the pressure lost through the outlet valve. As discussed before, this parameter also seems to be slightly lower for compressors with a larger stroke.

- $K_5^{'} \cdot D^2$: This parameter represents the leakage mainly through the gap between the piston and the cylinder during the compression process. The value obtained for this parameter for all the compressors can be considered as constant. This parameter influences compressor performance in two ways:

  1. *Reduction of the mass flow rate pumped by the compressor*: The relative leakages significantly increase with pressure ratio. On the one hand, the leakages are higher when the pressure ratio increases; on the other, as the pressure ratio increases, the total mass flow rate decreases. Consequently, these two effects combine to produce the relative increase of leakages. In addition, the relative leakage is slightly more significant when the number of cylinders is increased and that for the same number of cylinders the amount of refrigerant leaked is greater for larger stokes.

  2. *Increase of the inlet temperature at the cylinder inlet*: This temperature increase added to the refrigerant discharge temperature could have a significant influence in the high oil sump temperature measured for high pressure ratios because the higher pressure ratio the higher temperature increase as a consequence of leakages.

- $\frac{V_d}{V_s}$: This parameter (dead space ratio) shows a quite a significant deviation from the real geometric value for the four compressors. Compressors SO and ST have a geometric dead space ratio of 0.037, whilst the one obtained from the fitting process of the model is approximately 0.07. By contrast, compressors LO and LT have a geometric dead volume ratio of 0.029, and the one obtained from the model is approximately 0.055. Thus, it seems that the fitting process tends to overpredict the geometric value by a factor of approximately 2. As discussed in Appendix D, a possible phase change inside the cylinder was considered in the model to explain this deviation in the dead space ratio and much better results were obtained.

- K_6 : This parameter represents the quantity of refrigerant which undergoes a phase change in the cylinder (dropwise condensation at the inlet valve).

In appendix D it was established that all the parameters except the dead space ratio remain mainly unaltered when the phase change term is considered in the model. Taking this into account, the results shown in table 5.1 correspond to the parameters of the model without condensation, except for the values of K_6 and $(\frac{V_d}{V_s})'$, which correspond to the model considering phase change effects.

The value obtained for the dead space ratio in the model if phase change is taken into account is quite close to the expected geometrical one. Regarding K_6 (phase change parameter), the values for the four compressors are also quite similar. Furthermore, the absolute value obtained for this parameter was found to be quite meaningful, where the implicated heat transfer coefficient for this process is approximately 25000 $W/(m^2K)$ which is inside the possible values of dropwise condensation and is of the same order of the heat transfer coefficient given by [103] for a compressor working in wet conditions. These facts support the hypothesis that this effect indeed exists, and weilds a considerable influence particularly when dealing with propane.

Basically the hypothesis is that condensation may occur on a "cold spot" at certain points in the cylinder. For instance, the inlet valve is exposed to relatively cold gas at the inlet. The valve plate may then be acting as a regenerative heat exchanger: if the reed valve temperature is lower than the dew point temperature, then condensation could occur on the valve plate during the compression and delivery phase. Any condensate formed will evaporate during the following suction phase and, if we assume that this evaporation takes place on the valve plate, this will subsequently cool the valve plate material so that it is ready for a new condensation process in the following compression cycle. Provided that the dynamics of these heat exchange processes are fast enough, the amount of gas delivered by the compressor will decrease and cause a similar net effect on the performance as an increased dead space.

This effect could be quite significant when the compressor is working at high pressure ratios. In these conditions the percentage of mass flow rate lost as a consequence of this effect could be up to 12% for propane.

- K_7 and K_8: These parameters represent the two terms of mechanical losses. The former is proportional to energy consumption (K_7) and the latter, to the velocity of the compressor (K_8). Regarding the results shown in table 5.1, two comments should be made:

  1. The values obtained for K_7 are similar for the four compressors except for LO. The fact that the value obtained for compressor LO is completely different from the others seems to point to some kind of problem or malfunction in this compressor. This agrees with the experimental results obtained with these compressors in which LO showed slightly lower values for compressor efficiency compared to the other three compressors.

  2. K_8 shows the highest scattering among all the parameters considered. The values obtained for the short stroke compressors (SO and ST) seem to be consistent since they only differ in the number of cylinders (SO has one cylinder and ST has two cylinders). However, the results obtained for compressors LO and LT do not maintain that relationship; in fact, a factor of 3 is observed. This difference could be explained as a mechanical problem in compressor LO, as mentioned before.

- η_{el}: This parameter represents motor electric efficiency and was considered constant for all the experimental points of each compressor, since it was known that the compressors are working in the flat region of the motor efficiency curve in all conditions within less than 1%-2%. The values obtained from the model for the electric efficiency (table 5.1) are in very good agreement with the available data for these compressors.

Finally compressor outlet temperatures were estimated considering that this temperature should not differ too much from the cylinder outlet temperature. The results obtained represent a good approach except for high pressure ratios when a divergence of approximately 5 degrees from the measured data was found. These results seems to indicate that at those conditions not considered effects such as losses to the environment become more important. Taking into account that no experimental compressor outlet temperatures are fed into the model, the results obtained for discharge temperature give more consistence and confidence to the developed model. Additionally, a slightly lower temperature (approximately 1 K) of compressor outlet temperature compared with calculated cylinder outlet temperature at low pressure ratios was detected for all the compressors. This fact could imply a heating of the discharge refrigerant in the muffler by the electric motor that is very close to it.

A deeper analysis of all these results can be found in appendices D and E.

5.2.1 Relative influence of each loss considered in the compressor and volumetric efficiencies

Figs. 5.1 and 5.2 illustrate the relative influence of the losses on compressor and volumetric efficiencies in all available test conditions for compressor ST as a function of pressure ratio. Fig. E.9 shows that electric and mechanical losses are most influential in the reduction of compressor efficiency (totalling approximately 75%), the pressure losses being quite important at low pressure ratios. The opposite behavior is observed for the leakages, reaching 15% relative influence in the reduction of compressor efficiency at the highest pressure ratios. The heat transfer between suction and discharge is responsible for approximately 7% of the reduction in compressor efficiency.

Regarding volumetric efficiency, from fig. E.10 it can be concluded that electric and mechanical losses are also considerable. Their influence remains approximately constant under all working conditions (totalling approximately 55%). Pressure losses at the suction valve are as great as mechanical and electric losses are for low pressure

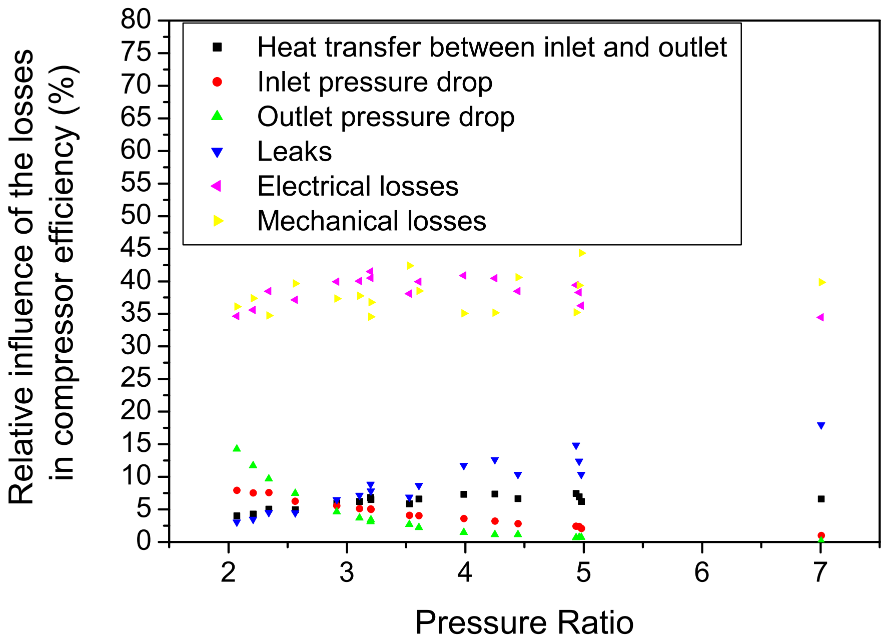

Figure 5.1: Relative influence of the different losses considered in compressor efficiency as a function of pressure ratio.

ratios. The influence of heat transfer between compressor inlet and outlet is also quite significant for volumetric efficiency, representing more than 15% of the total losses under most conditions. The phase change effect is considerably more influential at high pressure ratios, where the temperature difference between the inlet and the outlet of the cylinder is quite high and the mass flow rate is lower, being at these conditions the second most influential factor in the loss of volumetric efficiency.

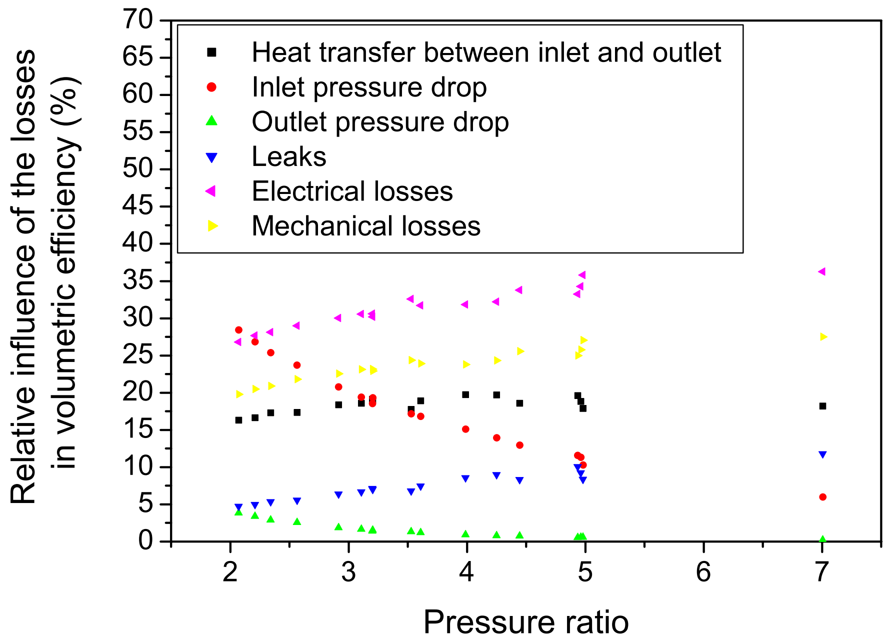

Figure 5.2: Relative influence of the different losses considered in volumetric efficiency as a function of pressure ratio for the ST compressor.

5.3 Differences between refrigerants

To evaluate the response of the model when the refrigerant is changed, the compressor ST was also characterized experimentally working with R407C and the obtained results were compared with the model estimation using the compressor parameters obtained with propane.

In figure 5.3, compressor efficiencies are reproduced for both refrigerants with an error lower than 3% in almost all conditions. Compressor efficiency for R407C is properly reproduced using the compressor parameters obtained for propane, however calculated volumetric efficiency (fig 5.4)shows a systematic deviation of 3% from the

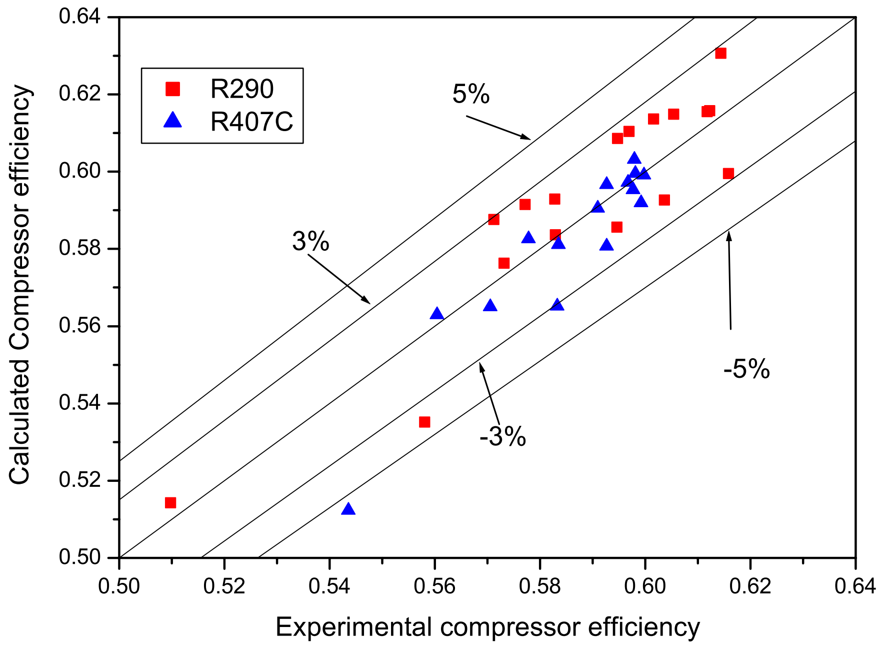

Figure 5.3: Calculated compressor efficiencies versus measured compressor efficiency for both refrigerants are represented.

experimental value. As a consequence of this deviation in volumetric efficiency for R407C, the assumption that all the parameters are constants when the refrigerant is changed was reconsidered.
After a detailed study over all parameters, a reduction of 50% in the parameter that represents the condensation losses provides a noticeable improvement of the obtained results for volumetric efficiency as it can be seen in the refitted values of volumetric efficiency for R407C on figure 5.4. An explanation to the different value of this parameter between both refrigerants could lay in the fact that the heat transfer coefficient, that parametrices the condensation evaporation process, probably shows a dependence in the refrigerant thermodynamical properties that is not considered in the model.

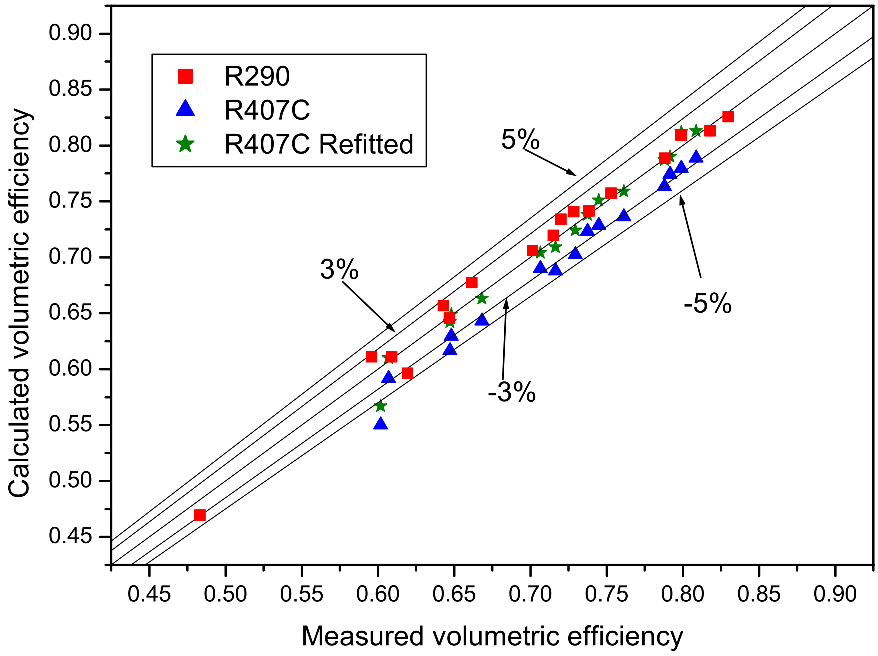

Figure 5.4: Calculated volumetric efficiencies versus measured compressor efficiencies are represented for both refrigerants.

5.3.1 Analysis of the different behavior between both refrigerants according the model

The electrical and mechanical losses are the most important factors in the reduction of compressor and volumetric efficiencies. Electric efficiency was considered constant for both refrigerants as it is commented in appendix A. Regarding mechanical losses, figure 5.5, on the left, presents the relative influence of mechanical losses on the compressor energy consumption as a function of pressure ratio. On the average, the percentage of energy lost is approximately the same for both refrigerants. Therefore, a notorious difference in the compressor performance is not expected when the compressor is working with any of these refrigerants.

Figure 5.6, illustrates the pressure drop for the inlet and outlet valve for both refrigerants. On the average, R407C has higher pressure drop through the valves than propane (approximately 4% higher through the inlet valve and 11% through the outlet valve).

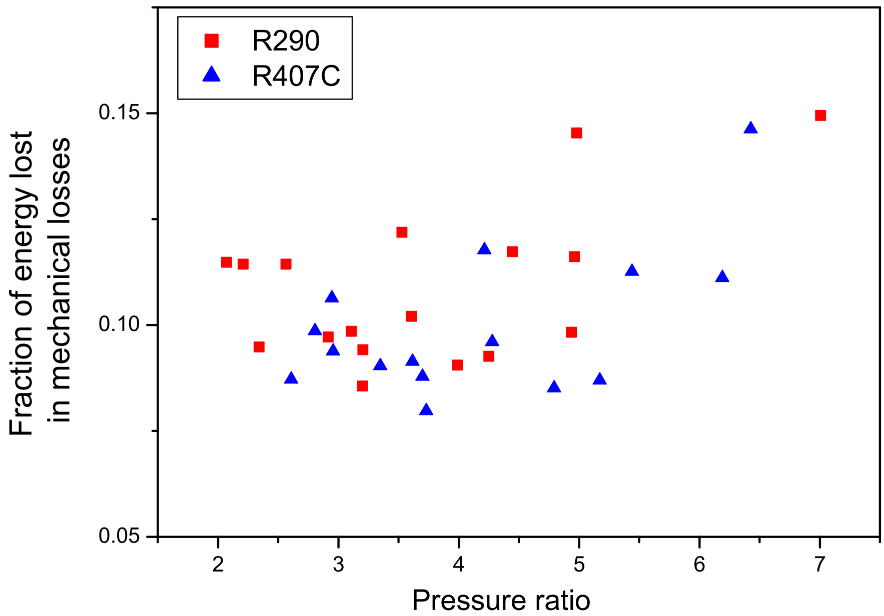

Figure 5.5: Fraction of mechanical losses as a function of pressure ratio for R407C and R290.

Figure 5.7 plots the relative mass flow rate leaked versus the pressure ratio for both refrigerants. The relative influence in the reduction of mass flow rate as a consequence of the leaks is higher for propane; this is probably related to the fact that propane has a lower density than R407C, thus a lower viscosity, allowing this refrigerant to flow throughout the piston ring more easily. The relative difference between the leakage percentages can be up to 50% at high pressure ratios.

Figure 5.8 shows the relative mass flow rate lost by the effect of condensation

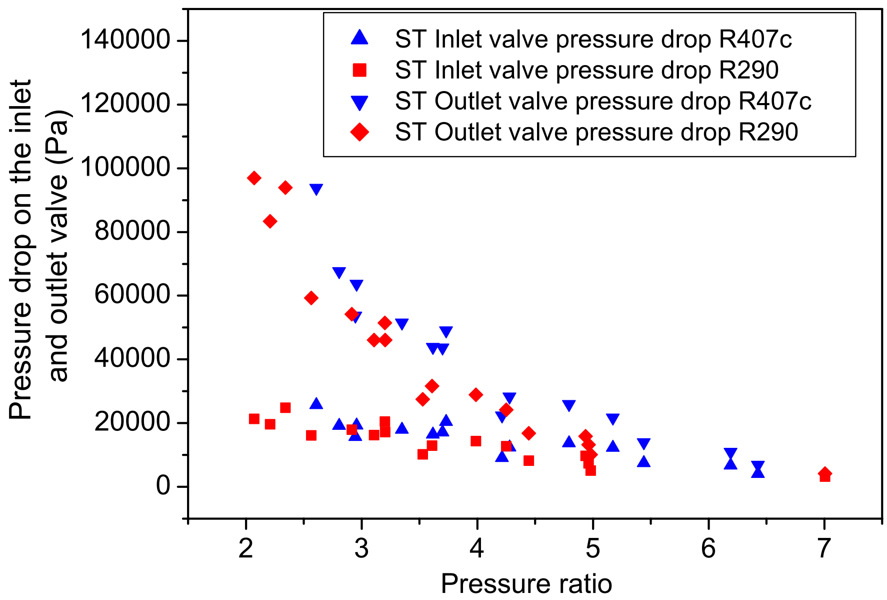

Figure 5.6: Absolute pressure drop is represented as a function of pressure ratio for the inlet and outlet valve for both refrigerants.

versus the pressure ratio for both refrigerants. This effect is not so important for R407C as for propane; in fact it does not represent a relative loss in mass flow rate higher than 4% up to a pressure ratio of 4.5.

Figure 5.9 illustrates the calculated temperature of the refrigerant on the cylinder outlet versus the measured temperature of the refrigerant at the compressor outlet for both refrigerants. As it is explained in the last section, the model is not able to calculate the compressor outlet temperature because it does not evaluate the evolution of the refrigerant between the points 7-8 of figure D.1. However, as it can be seen in figure 5.9, the calculated temperature of the refrigerant at the cylinder outlet is a very good approximation of the temperature of the refrigerant at the compressor outlet when the temperature of the refrigerant at the compressor outlet is not too high (90ºC-100ºC for propane), beyond this, heat losses to the environment and other

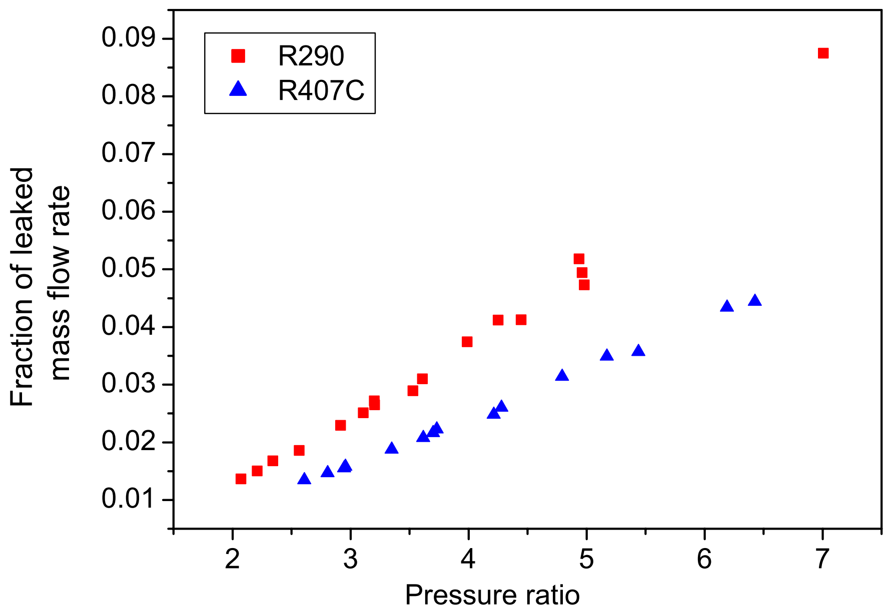

Figure 5.7: Fraction of leaked mass flow rate as a function of pressure ratio for R407C and R290.

not considered effects which could have an influence between states 7-8 of figure D.1. begin to be important, causing the difference between both temperatures to be higher. The difference between both temperatures is not so important for R407C. Therefore, it can be concluded that losses to environment are not so important for R407C. This figure also gives reliability to the model, because no refrigerant outlet temperature data were used in any part of the definition of the model parameters.

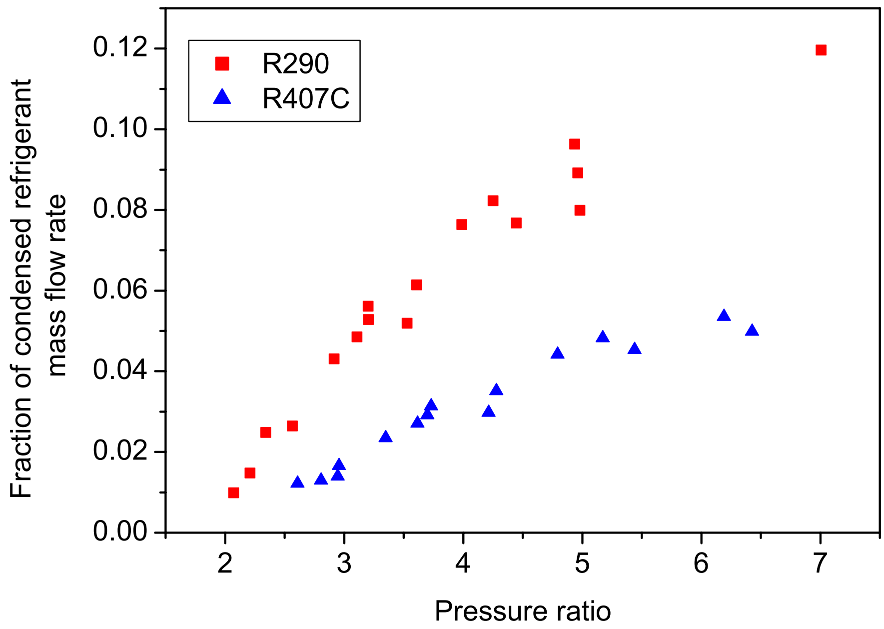

Figure 5.8: Fraction of condensed mass flow rate as a function of pressure ratio for R407C and R290.

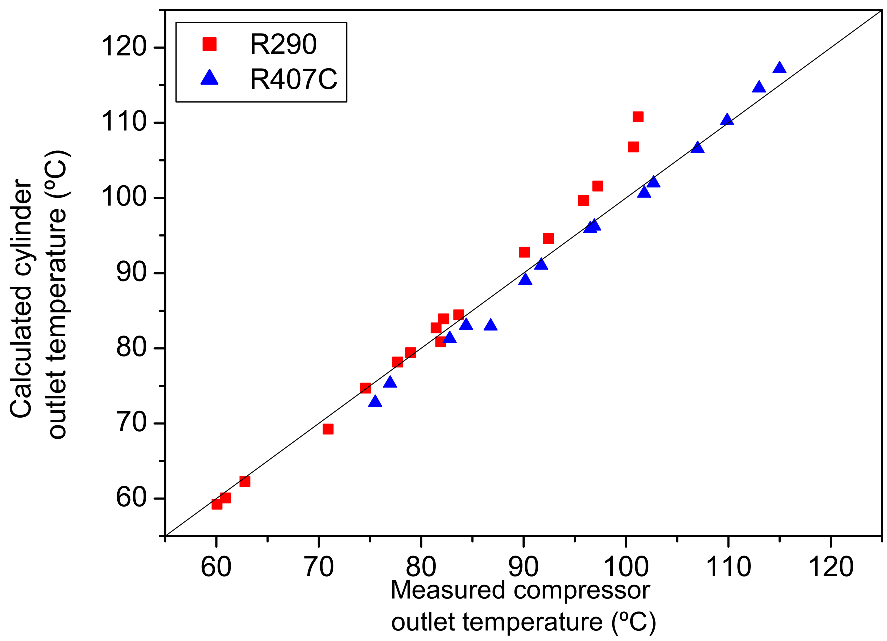

Figure 5.9: Calculated temperature of the refrigerant on the cylinder outlet versus measured compressor outlet temperature for both refrigerants.

CHAPTER 6

Conclusions

6.1 Concluding remarks

The objectives of this thesis have been accomplished. First, it was the aim to get a vast amount of empirical experience and performance data on the behavior of compressors working with propane as refrigerant.

Throughout an extensive 3-year programme that included a substantial re-development of our compressor characterization test rig, the set up of original experiments (like OCR and start up measurements) and the realization of large number of tests on 5 commercial R407C reciprocating compressors, a complete experimental database was developed which offers a comprehensive picture about the functioning of compressors originally designed for R407C working with propane. Some R407C tests were also done to allow comparative observations.

The second main objective, partially linked to the first one, was to gain physical insight into the reasons why the behavior R407C vs. propane is different. This would allow not only to improve the design of compressors, specifically for propane, but also our general understanding of the compression process, with a wide application potential to any compressor with any refrigerant.

After a thorough consideration of possible alternative model approaches, discussed in chapter 1, it was concluded that the development of a new semi-empirical model with novel characteristics compared to the previous existing was the right answer to this challenge.

The conclusion extracted form both different but intimately interconnected parts of this thesis are the following:

Experimental work

The main difference of propane related with HFCs is the higher solubility in oils. No significant differences in OCR has been detected between R407C and propane when the two are using a POE oil ISO 32 as lubricant. A comparison between POE oil and MO VG 68 (more common with propane) in terms of compressor performance, OCR and oil start up emission has been developed too. This study has pointed out the similar compressor performance with both oils, being the use of propane completely reliable. However, propane has lower solubility in POE oil, and this fact could represents a reduction up to 100 g. of refrigerant charge in a heat pump with a compressor charged with one liter of oil, that is quite relevant amount in a system working with this flammable gas.

Theoretical work

A model for reciprocating compressors was developed. This model can reproduce the compressor and the volumetric efficiency with an error lower than 3% under a wide range of operating conditions. Although this model was developed for hermetic piston compressors, as a consequence of its general conception, it may be applied to analyze and describe any kind of positive displacement compressor.

The compressor model is based on the hypothesis that the most important sources of losses occur before the refrigerant arrives to the compression chamber, being the compression isentropic. This model has 10 empirical parameters, which, if they are unknown, must be fitted using data of compressor and volumetric efficiency either from experiments or catalogue.

A statistical fitting methodology based on Monte Carlo techniques was designed with this purpose. This methodology was tested on one compressor and the results with 16 experimental points were quite good. To apply the developed fitting methodology, only data commonly available in catalogs are required.

All the model parameters have a direct physical interpretation and characterize the design and performance of the compressor. In general, the developed model could be quite useful:

1. *As a prediction and design tool*: Estimating compressor performance at operating points, different from the experimental points used on the fit. Furthermore, the model could be used to estimate the compressor performance with another refrigerant, other compressor speeds or with slight modifications in the cylinder geometry.
2. *As a diagnosis tool*: Characterizing the compressor performance with 10 parameters and analyzing its adequacy from their absolute value or by comparison with reference parameters. For example, unusual values of motor electric efficiency, mechanical losses or valve losses, can point to internal compressor problem.

When the model robustness was established, 4 hermetic piston compressors with slight changes in their internal piston geometry were analyzed in terms of the model working with propane and the results were compared with the ones obtained when the compressor was working with R407C.

Certain anomalies in the performance of one of the compressors working with propane were detected, and a mechanical problem was indicated by the model as a possible reason for this outlying behavior.

The dead space ratio obtained was larger than the expected one. This fact seems to point to an additional loss not considered previously in the model which reduces the total mass flow rate that the compressor is able to pump. The possibility of a condensation effect inside the piston was postulated as a suitable explanation for this result. When this effect is considered in the model, the values obtained for the dead space ratio becomes quite close to those expected.

According to the model, the mechanical and the electric losses are the most relevant ones in all the tested conditions, they represent approximately 75% of the total compressor efficiency losses and 55% of the total volumetric efficiency losses. At low pressure ratios (1.5-2.5) the pressure losses are noteworthy (more than 15% on compressor efficiency and more than 25% on volumetric efficiency). At high pressure (5-7) ratios, the leakages become a significant factor (more than 10% for compressor and volumetric efficiencies). The relative influence of heat transfer losses between the inlet and outlet remains constant at all pressure ratios and its overall influence on volumetric efficiency is significant (15%).

The response of the model when it is used to predict compressor performance with a different refrigerant was tested and the relative compressor performance differences between propane and R407C outlined. Concerning volumetric efficiency, the model underestimates the experimental results by approximately 3%. If the phase change parameter is released and adjusted to the experimental volumetric efficiency data, it becomes approximately 50% of the value obtained for propane, improving significantly volumetric efficiency predictions. Some not considered dependence in the refrigerant thermodynamical properties of the heat transfer coefficient related with this process are supposed to be as a possible explanation to this result.

Regarding the differences between both refrigerants, it appears that propane tends to improve its performance compared with R407C at low pressure ratios, while worsening it at high pressure ratios. This seems to be related with the lower pressure losses with propane at low pressure ratios, as well as higher leakages for propane at high pressure ratios. Both effects may be attributed to the lower propane density. Finally it should be commented that the efficiency of propane has yet another beneficial practical effect, since at the same condensation and evaporation temperatures propane works at lower pressure ratios as a consequence of its saturation pressure-temperature curve.

6.2 Future research

The possible future research derived from this thesis can be summarized in the following main lines:

- To generalize the same model philosophy to describe any kind of positive displacement compressors as in principle there are no strong bounds to the compressor design.
- To go further in the understanding of the postulated propane condensation effect inside the compressor as an important source of potential losses.
- To analyze other lubricants like PAG or PVE with lower solubility to be able to reduce the charge of a system working with propane.
- To apply the new modelling philosophy to predict and analyze the behavior of other potential refrigerants as a benchman-king tool and a guide for improving their design.

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APPENDIX A

Test results of performance and oil circulation rate of commercial reciprocating compressors of different capacities working with propane (R290) as refrigerant

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A.1 abstract

In this experimental investigation five R407C positive displacement hermetic reciprocating compressors, covering different capacities, displacement, stroke–to–bore ratios and number of cylinders, have been characterized using propane as refrigerants by means of a specifically designed characterization test rig. Test results have been systematically compared with their R407C reference performance data to obtain a complete picture on changes on the volumetric efficiency and compressor efficiency amongst others. The compressors used POE oil as lubricant and additional oil circulation rate (OCR) tests at steady state conditions were done to evaluate possible effects and differences to the traditionally used mineral oils.

Nomenclature

Tev	Evaporation temperature
Tcond	Condensation temperature
R_c	Pressure Ratio
$\dot{Q}_{ev}$	Cooling capacity
$\dot{m}$	Mass flow rate
h_i	Enthalpy in *i* point of the refrigeration cycle
$\eta_{v,1}$	Compressor volumetric efficiency referred to compressor inlet conditions
$\Psi_{v,1}$	Volumetric capacity
$\dot{V}_{sw}$	Compressor swept volume
v_1	Refrigerant specific volume
ϵ	Coefficient of performance
$\dot{P}_{el}$	Electric power consumption
ψ_{is}	Compressor efficiency
ϵ_{is}	Ideal cycle performance
OCR	Oil circulation rate

A.2 Introduction

Within the global search after new environmentally safe alternatives to CFC and HCFC– based refrigerants, HFC's are seen by many as an almost unavoidable choice. However, HFC's have mainly two important disadvantages: first, despite their null ODP they show a high GWP, this fact has led some UE countries to put limits to their use; secondly, they are new chemical products that have never been present in the atmosphere and, for they have long mean life, nowadays unknown long term effects may arise. For these reasons research has been conducted to find other, environmentally unquestionable, alternatives to HFC's based on gases *actually* present in the atmosphere. In this context, hydrocarbons (HC's), and in particular, propane (R290), are long time known as excellent refrigerants and their reintroduction in refrigeration systems is being supported by different research groups, companies and institutions.

From the theoretical side, propane is a pure HC which offers excellent thermodynamic properties, good compatibility with most materials already present in refrigeration plants and null ODP and GWP, see [104] for a more systematic account of the advantages of HC's in refrigeration systems, see also [18] for a comparative account of the advantages and disadvantages with respect to the most used refrigerants. There are also many experiences with R22 drop–in systems as well as specifically designed equipment, see for example [104], [21], [105], [106], which support the suitability of R290 in different types of refrigeration plants.

Regarding the main drawback of hydrocarbons, their flammability, new standards and codes have been recently released (see [107], [108]) or are in preparation to allow its safe use (see e.g. the documents around the IEC 60335 standard for use of flammable gases in electrically driven refrigeration equipment, [109]). Also special attention should be paid to the one released by the European Committee for Standardization, [107], fixing certain restrictions to the use of HC's as refrigerant (mainly to be used in indirect systems) and setting maximum allowable charge figures for given applications.

According to statistics of heat pumps tests at the WPZ Töss Switzerland, [110], after R407C, propane is the second most used refrigerant in heat pumps, being carried by about 12% of the total number of heat pumps tested in 2002 (31% of the air–water systems). It should be though noted that the share of propane was even higher in 1999 (14%), when R22 was the mostly used refrigerant. This means that the substitution of R22 has not given advantage to R290, but to R407C which, despite its zeotropic character and its poorer performance compared to R22 and R290, is being widely used as *de–facto* replacement for R22 in heat pumps and air conditioning systems, because of its adequate pressures and temperatures.

One of the most serious obstacles to a more widespread use of R290 in refrigeration systems lies in the the main compressor manufacturers being unwilling to give support to these applications. Beyond the safety issue, one of the reason for this may lie in the lack of information on how compressors behave with propane in a wide range of conditions.

The use of HFC's has also brought changes in the types of compressor lubricants used because of the different solubility patterns that these refrigerants show when mixed with mineral oils. For this reason new oils like POE have been developed to allow the refrigeration system to work in a more efficient way with HFC's. There is also a considerable lack of information related to the interaction of these new oils with other refrigerants like R290.

The aim of the present work was to develop a comparative study between R290 and R407C for five different reciprocating compressors working with POE oil. The compressors and test conditions were selected to cover a wide range in capacities as well as working conditions. With this philosophy three compressors with different number of pistons (one, two and four) and a stroke of 38.1 mm, and two compressors, with one and two pistons, and with a stroke of 30.23 mm were tested. In the analysis we have tried to separate those changes that are based on the refrigerant thermodynamical properties and those related to the way the compressor works with the different refrigerants. In addition, oil circulation rate measurements were made to evaluate possible anomalies or differences in behavior of these refrigerants with POE oil.

A.3 Test setup

Compressors selected to be tested are shown in table E.1. The standard POE oil included by the manufacturer in the compressors was used in all tests. In all propane tests almost pure (99.95%) propane was used to ensure traceability of our results.

The compressor rating procedure was performed according to the relevant standards in the field such as the ISO-917, [96], and American ANSI ASHRAE 23-1993, [111], as shown in the scheme in fig. A.1. According to the aforementioned standards, the circulating refrigerant mass flow is the determining parameter to be measured and primary and confirming measurements are to be made. The primary test procedure chosen is the secondary refrigerant calorimeter method. A Coriolis-type mass-flow meter was used as the confirming test method. In all cases, confirming tests were carried out simultaneously with the primary mass-flow rate determination.

Several PID control loops (compressor inlet and outlet pressure, superheat and subcooling controls) were incorporated to allow a precise adjustment of the refrigerant conditions at compressor inlet (evaporating temperature and superheat) and outlet (condensing temperature) with a precision of 1 kPa. The rig is fully automated, and designed to make possible to reach any allowable test conditions without manual adjustments.

The mass flow rate directly measured by means of a Coriolis-type (Fisher-Rosemount Micro-Motion CMF025M) was compared with the secondary refrigerant calorimeter based result. The instrument accuracies of pressure transmitter (Fisher-Rosemount

3051) and temperature transmitter (RTD-PT 100) are 0.02% and 0.05ºC, respectively.

Oil circulation rate (OCR) measurements were done following standard ANSI/ASHRAE 41.4-1996, [98]; Care was taken that the oil level at the sight glass was stable during the test.

Safety was a major concern during the design of the test facility. Specific procedures and standards regarding the handling and use of flammable gases were taken into account. Specific measures included the use of intrinsically safe electric material, special propane sensors, the use of emergency switches and alarms and appropriate air renewal procedures to ensure non-critical concentrations in case of leakage.

A.4 Test and evaluation procedure

Comparing the performance of a given compressor with different refrigerants, there are changes caused just by differences in the thermophysical refrigerant properties whereas others could in principle be optimized with a better compressor design. According to the rating procedures described in the above section, compressor capacity is given as:

$$\dot{Q}_{ev} = \dot{m}(h_1 - h_{3^*}) \tag{A.1}$$

where $\dot{m}$ is the mass flow rate and $(h_1 - h_{3^*})$ the enthalpy difference between two specific states of the refrigerant. According to the ISO procedure, see [96], these states are the inlet of the compressor, 1, and the saturated liquid state at compressor outlet pressure, 3*. The last is not a real state of the refrigerant in the compressor test loop. In this sense the rated capacity and COP do not depend on the actual cycle, but only on the conditions upstream and downstream of the compressor. In this work,instead of the ISO standard, this point was fixed at 8.3K. degrees of subcooling, to compare the obtained results with the available catalog data for R407C .

A different way to write the above expression is to consider it from the point of view of the compressor:

$$\dot{Q}_{ev} = \eta_{v,1}\Psi_{v,1}\dot{V}_{sw} \tag{A.2}$$

being $\dot{V}_{sw}$ the compressor swept volume, $\Psi_{v,1} = (h_1 - h_{3^*})/v_1$ the volumetric capacity (or refrigerating effect) of the given fluid for the given cycle and $\eta_{v,1}$ the compressor volumetric efficiency referred to compressor inlet conditions. v_1 is the refrigerant specific volume at compressor inlet for given evaporation temperature and superheat.

A coefficient of performance (COP), ϵ, is defined as:

$$\epsilon = \frac{\dot{Q}_{ev}}{\dot{P}_{el}} = \psi_{is}\epsilon_{is} \tag{A.3}$$

with $\dot{P}_{el}$ the electric power consumption of the compressor. The coefficient of performance of the ideal cycle with isentropic compression – $\epsilon_{is} = (h_1 - h_{3^*})/(h_{2s} - h_1)$ – is again a refrigerant property, whereas the compressor efficiency, $\psi_{is} = \dot{m}(h_{2s} - h_1)/\dot{P}_{el}$, combines compressor related and refrigerant features.

From these considerations it follows that propane possesses a higher capacity per mass unit, but, due to its much lower density, a lower capacity per volume flow unit (specific volumetric capacity). On the other hand discharge temperatures are also expected to be substantially lower on a compressor working with propane.

The ideal results indicate that, at equal volume flow of the compressor, when switching to propane, a capacity decrease of about 9% for high evaporation temperatures, and a capacity increase around 3% for low evaporation temperatures should be expected. For the same range of conditions, the coefficient of performance of the machine would be expected 6% higher with propane.

All this facts can be seen in fig. A.2 and fig. A.3 that exemplifies the ideal cycle performance data of R290 and R407C, for 8.3 K. of subcooling and 11.1 K of superheat.

The matrix used for the compressors characterization was chosen taking into account the conditions in which these compressors would work (relatively high condensation temperatures and evaporation temperatures) in order to provide service-representative results. It covered a set of condensation temperatures ranging from 35 ºC to 65ºC and -20 ºC to 15ºC evaporation temperature. A standard superheat of 11.1 K was chosen.The detailed used test matrix can be seen in fig. A.4. The unfilled circles indicate the points in which OCR measurements were performed. For SO compressor additional measurements of OCR at 35ºC and 50ºC were done to analyze some possible dependences on working conditions.

After a careful characterization of the calorimeter losses, test results showed a high consistency between the result from the primary and confirmation test method as specified in standards. The mean discrepancy between both methods were less than 1% in most cases.

A.5 Results and Discussion

Measured values of cooling capacity and COP, corresponding to a condensation temperature of 50ºC, are shown in fig. A.5, fig. A.6 and fig. A.7. Table A.2 offers a more detailed comparative analysis at two standard conditions which can be regarded as significant, labelled as MT (-10ºC of evaporation temperature, 45ºC of condensation temperature and 11.1 K of superheat) and ARI (7.2ºC of evaporation temperature, 54.4ºC of condensation temperature and 11.1 K of superheat). In the referred tables as well as the curves, propane based experimental results are compared to R407C performance data obtained from the manufacturer's catalogue.

Some additional test were developed to check that the tested compressor actually behave as expected from the catalog when they are working with R407C. The results show a reasonable consistency between catalogue and experimental data. The dew point was used to evaluate the properties in R407C.

Although in general terms the actual performance curves respect the tendencies that may be expected from the theoretical refrigerant curves, fig. A.2 and fig. A.3, there are quantitative differences arising from differences in the volumetric and compressor efficiency figures.

Measured cooling capacities, fig. A.5, show that the evaporation temperature at which R407C capacity equals R290 capacity is higher than expected from the volumetric specific capacity figures of fig. A.2. This can be understood from volumetric efficiency data that for the same temperature working conditions is higher for propane.

In fig. A.8 and fig. A.9 propane gives higher volumetric efficiency than R407C for low pressure ratios, this tendency is inverted as the pressure ratio goes up. As the number of cylinder grows, for the long stroke version, see fig. A.8, propane maintains a better volumetric efficiency at comparatively higher pressure ratios. In general terms volumetric efficiency depends more on compressor stroke than on compressor size (number of cylinders). This is expected from the ideal compressor behaviour as a consequence of compressors with larger stroke present a larger dead space ratio although some other factors like leakages must be taken into account.

It is important to clarify that the plotted lines in fig. A.8, fig. A.9, and the next ones fig. A.10, fig. A.11 only intend to represent the general trend of data for both refrigerants. The represented functions show other test condition dependencies like the inlet temperature although the pressure ratio is the most important one.

Relating to COP, a mean relative improvement of about 9 % was achieved using propane instead of R407C. This is an important and significant result, which also relies partly on theoretical refrigerant behavior and is quite sensitive to operation conditions: the improvement is larger at high evaporation temperatures, but falls down to less than 6% at low evaporation temperatures which may be expected from the theoretical isentropic COP data.

In fact compressor efficiency, fig. A.10 and fig. A.11, shows a complex behavior, being higher for propane at low and medium pressure ratios and lower at high pressure ratios.

Considering the compressor design, efficiency increases with the number of cylinders for the same cylinder volume (fig.A.10 and fig.A.11) and decreases when increasing the stroke while keeping the cylinder diameter constant.

A further aspect which was carefully considered was possible changes in the electric motor efficiency. Looking for instance at the operating conditions included in table A.2, the electric efficiencies at those points, showing a rather flat dependency on load, do not differ in more than 1% at most conditions when comparing R290 with R407C conditions. This leads us to conclude that the improvement in overall efficiency when using propane instead of R407C does not rely primarily on a change in the efficiency of the electric motor but on an improvement of the mechanical and thermal characteristics of the compression.

In this sense, the observed differences in compressor efficiency for both refrigerants could be related to the lower temperatures in the discharge for propane than for R407C (between 10^0C and 15^0C) limiting the amount of irreversible heat transfer between the relatively hot and the relatively cold parts of the circuits inside the compressor. Furthermore, as known from many experimental studies, see for instance [15], propane shows considerable reduced pressure losses at equivalent flow velocities. This leads to reduced losses especially at sections with relatively high velocities (mainly at the inlet and exhaust valves).

Another interesting factor considered in our investigation was Oil Circulation Rate. The obtained results are shown in fig. A.12 and table A.6.

From the SO compressor wider OCR test serie, it was observed that the proportion of oil leaving the compressor is quite independent on operating conditions –at least within the tested range (between -10ºC and 15ºC evaporation temperature for a constant superheat of 11.1 K), fig.A.12. For this reason only three OCR measurement were taken for each of the tested compressors at different operating points (see table A.6) and their associated mean results are shown in table A.6. The resulting quantities of oil do not seem large enough to cause a significant damage or effect on the normal operation of the system working with this R290/POE oil combination. Furthermore they are quite comparable to OCR figures for the R407C/ POE oil combination, see for instance [89].

On the other hand there is some variability in OCR values for different compressors.

A.6 Conclusions

A large and systematic test campaign including 5 different compressors and a representative span of operating conditions was conducted to derive general conclusions with respect to the suitability of propane (R290) as refrigerant in combination with modern POE oil. Performance data were systematically compared with those of R407C derived from catalogue data. These measurements and the following analysis lead us to the following general conclusions:

- R407C shows a worse thermodynamical behavior than propane, theoretically as well as in the real compressor. A mean COP improvement of 9 % resulted when using propane instead of R407C.
- Regarding the cooling capacity, results expected from the refrigerant properties where obtained although slight changes are recorded due to differences in the volumetric efficiencies.
- On the basis of the observed behavior, two main factors can be thought to have an influence: the reduced irreversible heat flow due to the lower discharge temperatures of propane and the reduced pump losses in the inlet and discharge valves.
- Related with the POE oil used to perform these tests, there is no significant difference between the oil behavior with propane and with R407C, at least in

standard working conditions. The influence of oil in the refrigeration system should be negligible if the installation is designed correctly to avoid oil traps [112], letting the oil to be drawn by the refrigerant.

During these tests the viability and the efficiency of a refrigeration system working with propane and POE oil has been proven. To look further, additional tests should be performed to check if long term durability problems could arise as a consequence of differences in solubility and viscosity of the oil–refrigerant mixture [35].

Furthermore, a deeper understanding of the rather complex confluence of effects on the basis of the differences observed deserves the employment of the computer codes that are being developed and adjusted in parallel with the described tests. Test results and models, altogether, will show the way to the final aim of this research: a better understanding of what is happening inside a refrigeration compressor (with or without propane).

Acknowledge

This work has been partially supported by the Spanish Ministerio de Ciencia y Tecnología through the project, ref. DPI 2001-2661-C02-01. The authors are grateful for the given support.

Figure Index

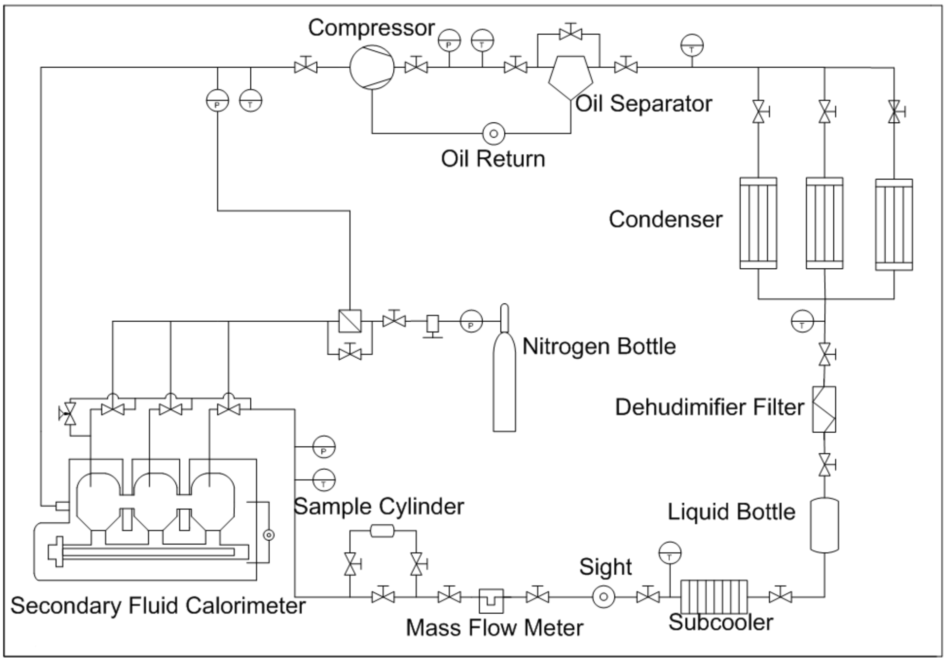
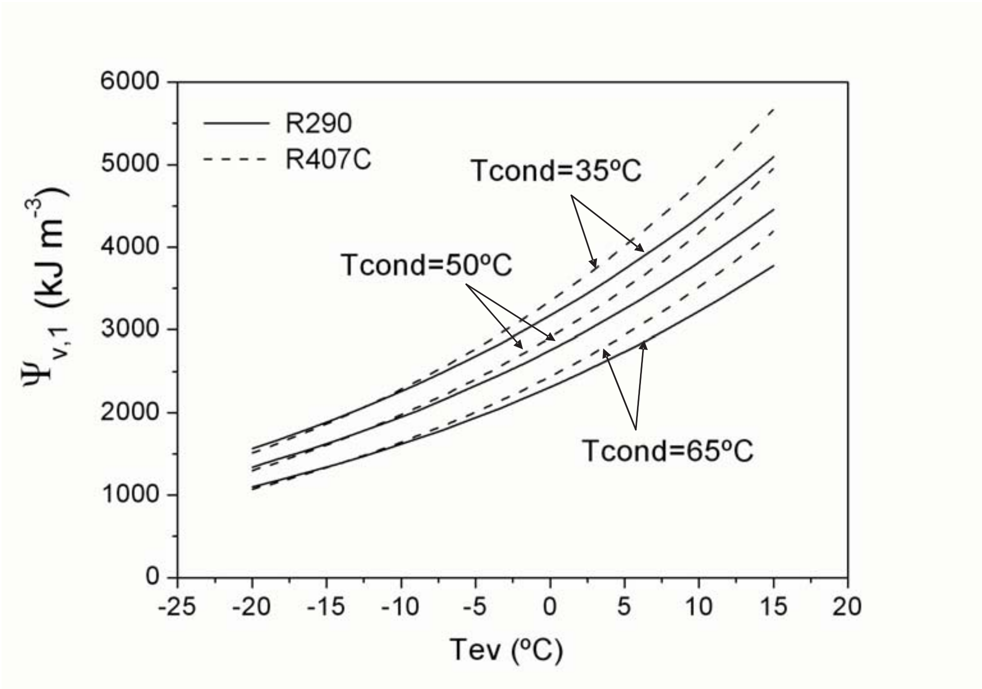

Compressor
Oil Separator
Oil Return
Condenser
Nitrogen Bottle
Dehudimifier Filter
Sample Cylinder
Liquid Bottle
Sight
Secondary Fluid Calorimeter
Mass Flow Meter
Subcooler

Figure A.1:



R290
R407C
Tcond=35ºC
Tcond=50ºC
Tcond=65ºC
$\Psi_{v,1}$ (kJ m^{-3})
Tev (ºC)
6000
5000
4000
3000
2000
1000
0
-25
-20
-15
-10
-5
0
5
10
15
20

Figure A.2:

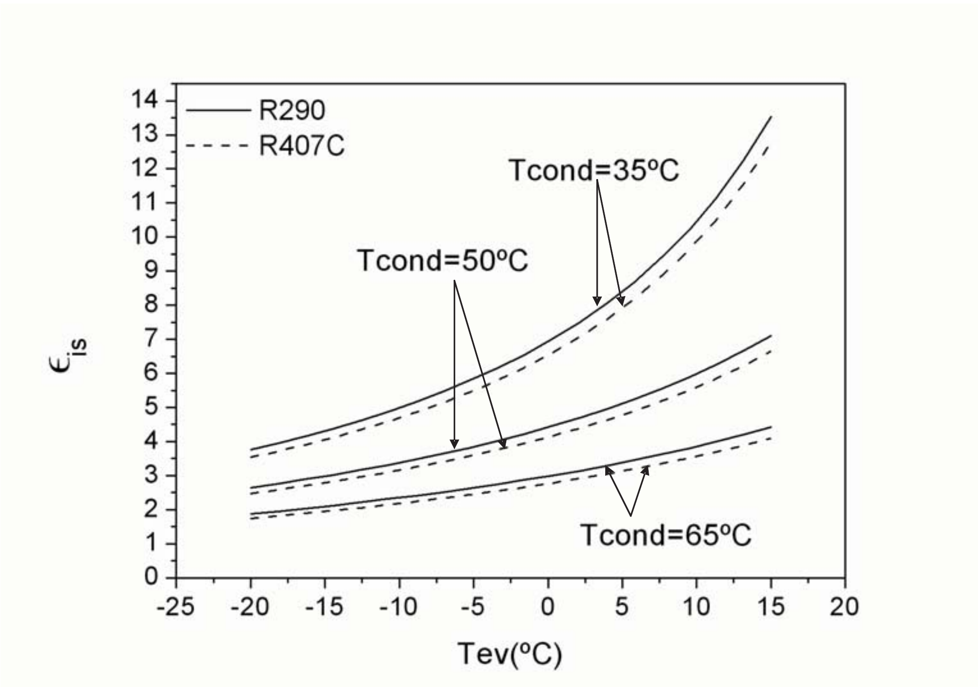

R290
R407C
Tcond=35ºC
Tcond=50ºC
Tcond=65ºC
ϵ_{is}
Tev(ºC)
14
13
12
11
10
9
8
7
6
5
4
3
2
1
0
-25
-20
-15
-10
-5
0
5
10
15
20

Figure A.3:

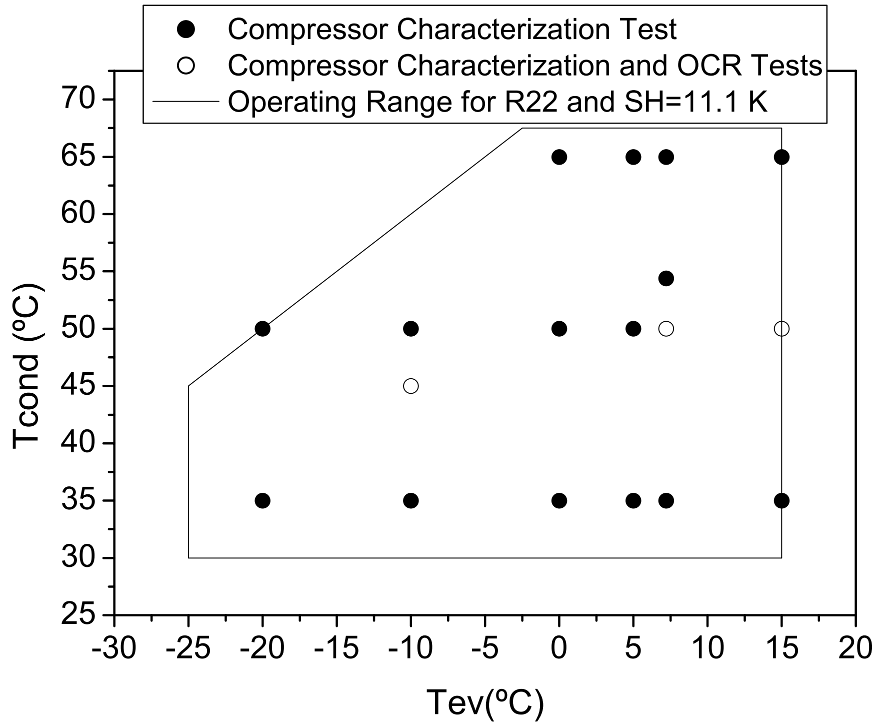

Compressor Characterization Test
Compressor Characterization and OCR Tests
Operating Range for R22 and SH=11.1 K
Tcond (ºC)
Tev(ºC)
70
65
60
55
50
45
40
35
30
25
-30
-25
-20
-15
-10
-5
0
5
10
15
20

Figure A.4:

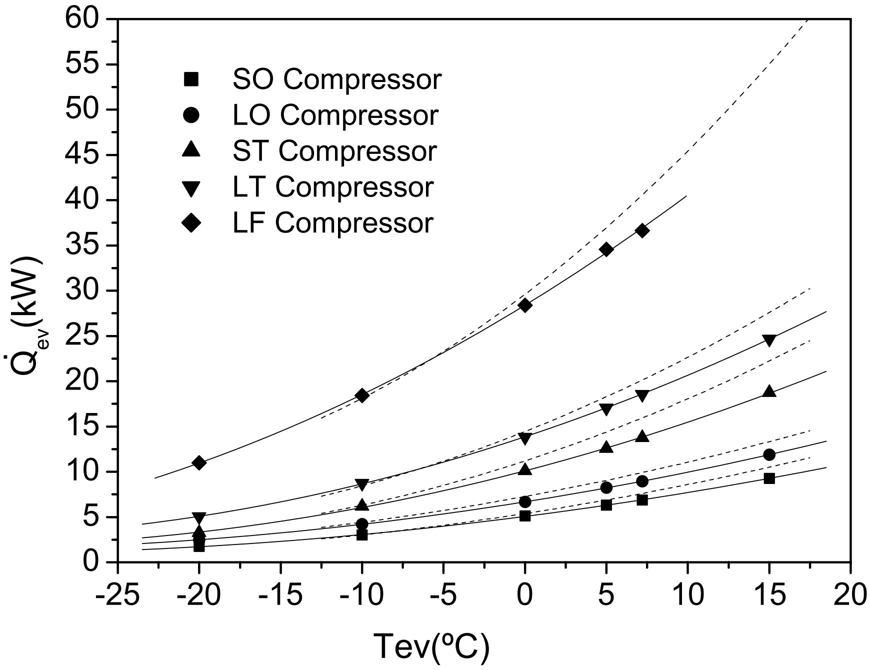

SO Compressor
LO Compressor
ST Compressor
LT Compressor
LF Compressor
$\dot{Q}_{ev}$(kW)
Tev(ºC)
0
5
10
15
20
25
30
35
40
45
50
55
60
-25
-20
-15
-10
-5
0
5
10
15
20

Figure A.5:

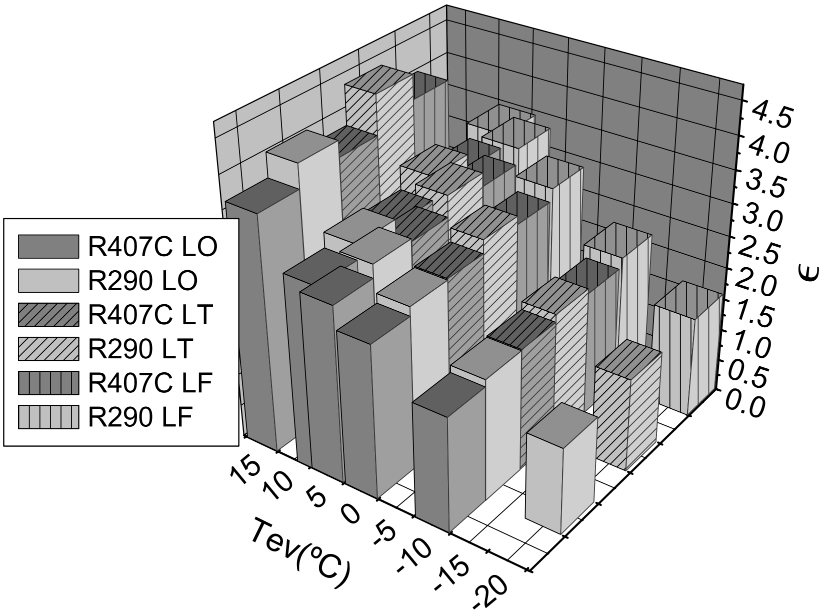

R407C LO
R290 LO
R407C LT
R290 LT
R407C LF
R290 LF
4.5
4.0
3.5
3.0
2.5
2.0
1.5
1.0
0.5
0.0
ϵ
15
10
5
0
-5
-10
-15
-20
Tev(ºC)

Figure A.6:

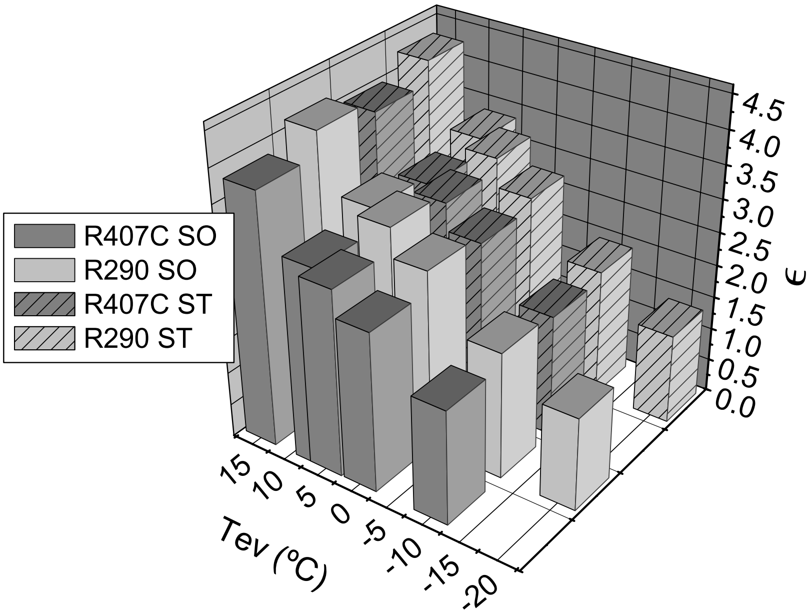

R407C SO
R290 SO
R407C ST
R290 ST
4.5
4.0
3.5
3.0
2.5
2.0
1.5
1.0
0.5
0.0
ϵ
15
10
5
0
-5
-10
-15
-20
Tev (ºC)

Figure A.7:

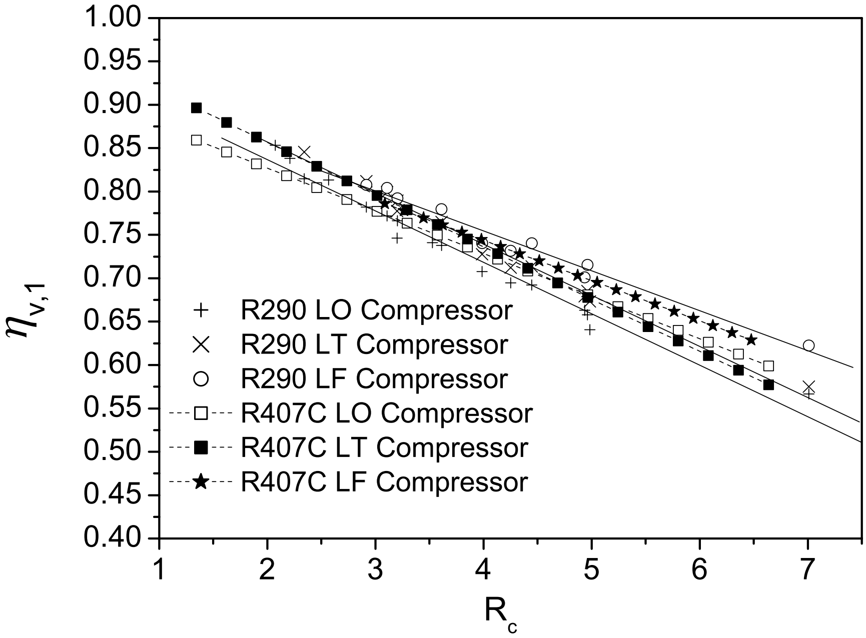
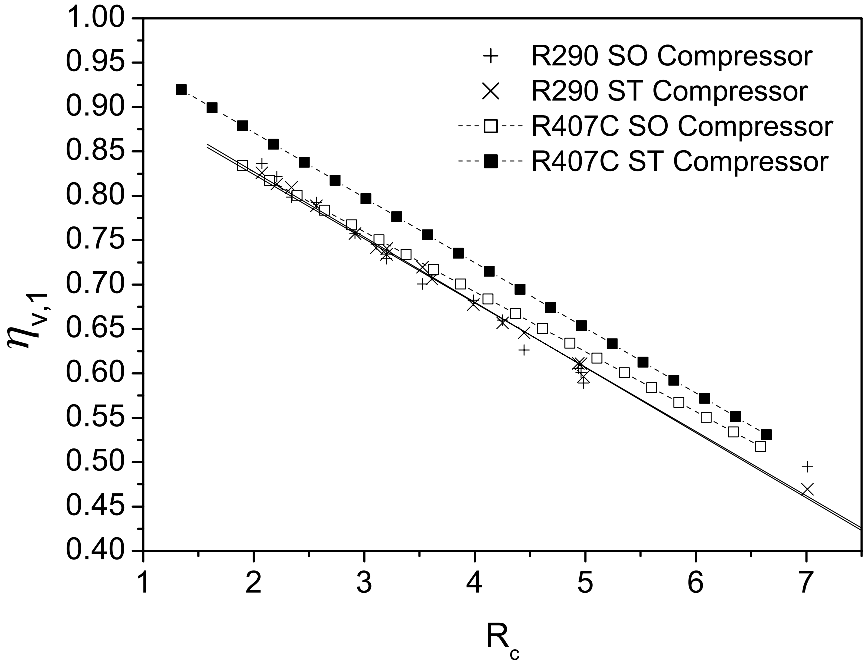

1.00
0.95
0.90
0.85
0.80
0.75
0.70
0.65
0.60
0.55
0.50
0.45
0.40
$\eta_{v,1}$
R290 LO Compressor
R290 LT Compressor
R290 LF Compressor
R407C LO Compressor
R407C LT Compressor
R407C LF Compressor
1
2
3
4
5
6
7
R_c

Figure A.8:



1.00
0.95
0.90
0.85
0.80
0.75
0.70
0.65
0.60
0.55
0.50
0.45
0.40
$\eta_{v,1}$
1
2
3
4
5
6
7
R_c
R290 SO Compressor
R290 ST Compressor
R407C SO Compressor
R407C ST Compressor

Figure A.9:

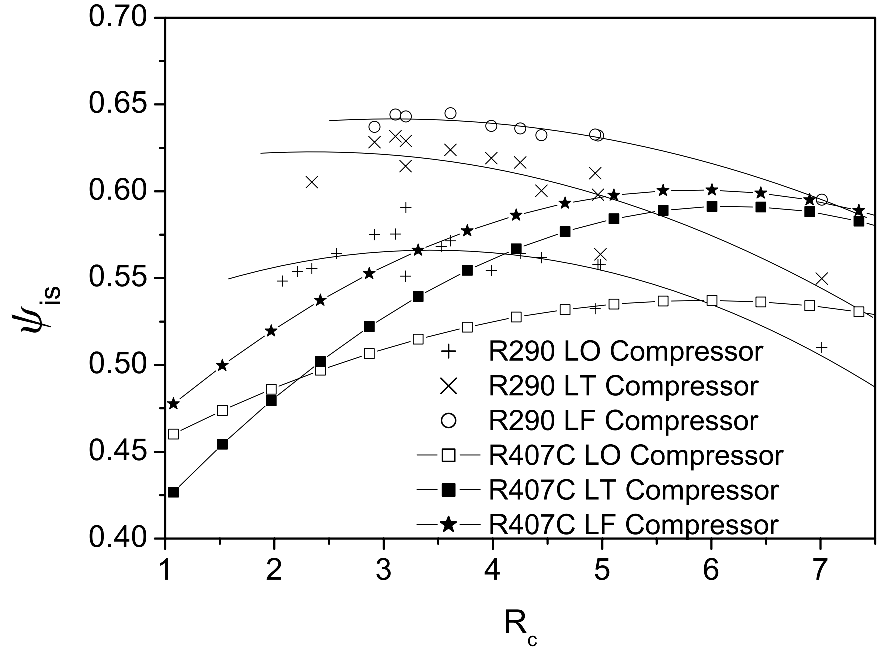

0.70
0.65
0.60
0.55
0.50
0.45
0.40
ψ_{is}
1
2
3
4
5
6
7
R_c
R290 LO Compressor
R290 LT Compressor
R290 LF Compressor
R407C LO Compressor
R407C LT Compressor
R407C LF Compressor

Figure A.10:

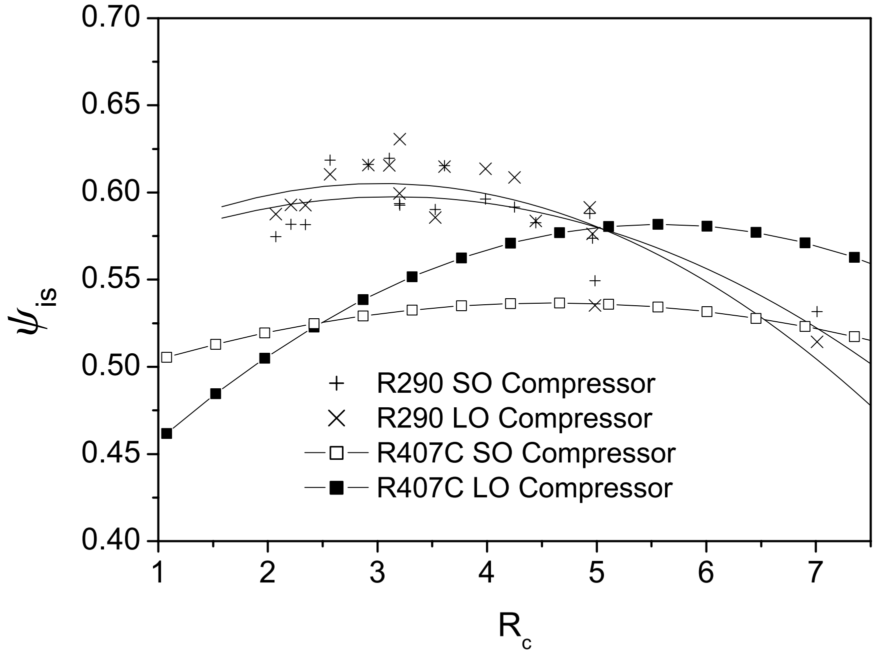

0.70
0.65
0.60
0.55
0.50
0.45
0.40
1
2
3
4
5
6
7
ψ_{is}
R_c
R290 SO Compressor
R290 LO Compressor
R407C SO Compressor
R407C LO Compressor

Figure A.11:

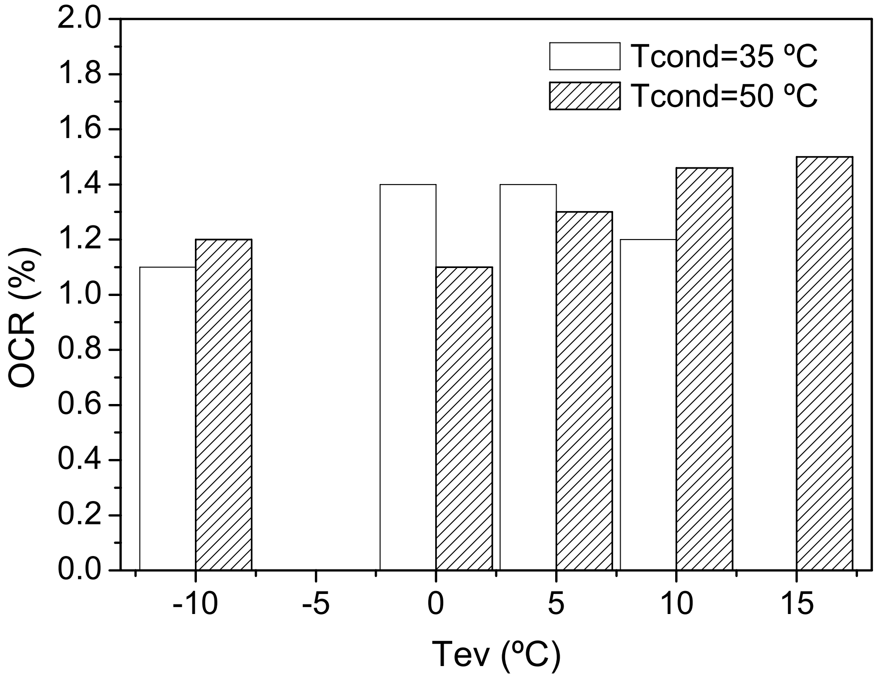

2.0
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
0.0
OCR (%)
Tcond=35 ºC
Tcond=50 ºC
-10
-5
0
5
10
15
Tev (ºC)

Figure A.12:

Stroke (mm)	Numb. of pistons	Dead Space ratio	Compr. name
30.23	1	0.037	SO
38.10	1	0.029	LO
30.23	2	0.037	ST
38.10	2	0.029	LT
38.10	4	0.029	LF

Table A.1: Tested Compressor

		MT Conditions				ARI Conditions			
Compressor		Cooling (kW)	Vol. eff.	COP	Comp. Eff.	Cooling (kW)	Vol. eff.	COP	Comp. Eff.
SO	R407C	3.41	0.60	1.99	0.53	6.94	0.70	2.60	0.55
	R290	3.34	0.63	2.22	0.58	6.37	0.73	2.84	0.59
	%	-2.1	4.7	11.0	9.0	-8.6	3.5	8.8	6.5
LO	R407C	4.86	0.68	2.03	0.54	9.13	0.74	2.53	0.54
	R290	4.68	0.69	2.15	0.56	8.36	0.77	2.83	0.59
	%	-3.8	1.5	5.7	6.6	-8.8	4.5	11.2	9.0
ST	R407C	7.06	0.62	2.10	0.56	14.59	0.74	2.78	0.59
	R290	6.90	0.64	2.23	0.58	12.82	0.74	3.03	0.63
	%	-2.2	2.8	6.0	3.9	-12.9	-0.13	8.6	3.4
LT	R407C	9.38	0.66	2.16	0.57	18.52	0.75	2.71	0.58
	R290	9.60	0.71	2.30	0.60	17.25	0.79	3.02	0.63
	%	2.3	7.9	6.3	4.6	-7.14	5.6	10.8	8.9
LF	R407C	19.93	0.70	2.21	0.59	37.39	0.75	2.82	0.60
	R290	20.10	0.74	2.40	0.63	34.90	0.79	3.08	0.64
	%	0.8	6.0	8.2	7.6	6.9	4.8	8.8	6.6

Table A.2: Experimental compressor comparative data for MT and ARI points. Positive percentage values of the relative difference indicate a better propane behavior.The errors of the measurements were lower than 1% in all the cases.

Compressor		OCR Meas	Mean Value
SO	(-10,45)	1.1	1.4
	(7.2,50)	1.5	
	(15,50)	1.5	
LO	(-10,45)	1.5	1.4
	(7.2,50)	1.7	
	(15,50)	1.0	
ST	(-10,45)	0.4	0.35
	(7.2,50)	0.4	
	(15,50)	0.3	
LT	(-10,45)	0.6	0.55
	(7.2,50)	0.5	
	(15,50)	0.5	
LF	(-10,45)	0.6	0.7
	(7.2,50)	0.8	
	(15,50)		

Table A.3: OCR measured values for the different compressors.

APPENDIX B

Comparative experimental investigation of oil behaviour in a hermetic piston compressor using propane (R290) as refrigerant.

E. Navarro, J.F. Urchueguía, J. Gonzalvez, J.M. Corberán
Proceeding of VI IIR Gustav lorentzen conference in Natural Fluids 2004

B.1 abstract

In this paper we report the result of a number of tests using an hermetic refrigerant piston compressor working with propane as refrigerant and two kinds of oil (POE and MO) as lubricant. Measurements were focused on three aspects: compressor performance differences between oils, oil circulation rate differences, and oil loss at start-up in different conditions in which the compressor could start-up in the normal operation of a heat pump. To develop the start-up measurements a new experimental method was designed to characterize the process as a consequence of the lack of information in this kind of test. All these tests let to establish some conclusions on the different behavior of both kinds of oils in a propane refrigeration system.

B.2 Introduction

In the global search after new environmentally safe alternatives to CFC and HCFC based refrigerants, HC's, and particullarly, propane, has been presented as a good alternative. From the theoretical point of view, propane is a pure HC which offers excellent thermodynamic properties, good compatibility with most materials already present in refrigeration plants and null ODP and GWP ([16] for a more systematic account of the advantages of HCs in refrigeration systems). Regarding the main drawback of hydrocarbons, their flammability, new standards and codes have been recently released (see [107], [108]) to allow its safe use. In this sense special attention should be paid to the one released by the European Committee for Standardization, [107], fixing certain restrictions to the use of HCs as refrigerant (mainly to be used in indirect systems) and setting maximum allowable charge figures for given applications.

An essential issue concerning the use of hydrocarbons in refrigeration is the interaction of the refrigerant with the lubricating oil, which has a strong impact on the performance and durability of the compressor and system. Some of the physical and chemical properties of the hydrocarbon-oil mixtures seem to point to the fact that the high solubility of hydrocarbons into mineral oil and consequent reduction of viscosity of the mixture have negative influence on the durability of compressors, pointing to the need of using oils with a higher viscosity. Another choice is to use an oil with a lower solubility (with propane) than mineral oil, but soluble enough to let a right oil drawn when the lubricant is inside the pipes and coils. This has the additional advantage of reducing the amount of refrigerant solved in the oil so less refrigerant charge is necessary to get the same refrigerant effect. The relatively new lubricants like POE and PAGs seem a good way to be checked, because of their potential in reducing the system charge.

In this work the different compressor behavior of conventional mineral oil ISO 68 and POE ISO 32 nowadays widely used in HFC's compressors is analyzed from the point of view of compressor efficiency, volumetric efficiency and oil circulation rate differences.

Other problems related with lubrication can occur in the compressor start-up phase, in which a sudden change in the compressor inside pressure is produced, with a possible intense foaming and a high amount of oil could be expelled. This can make the compressor to work a certain interval of time without enough lubrication and if this process is repeated many times, what occurs when the compressor is operating on an installation, can lead to its breaking. For this reason, a detailed study of the start-up phase to evaluate the amount of oil expelled by the compressor during this phase for both oils was developed. To do this, a new experimental methodology has to be implemented based on two complementary measurement techniques.

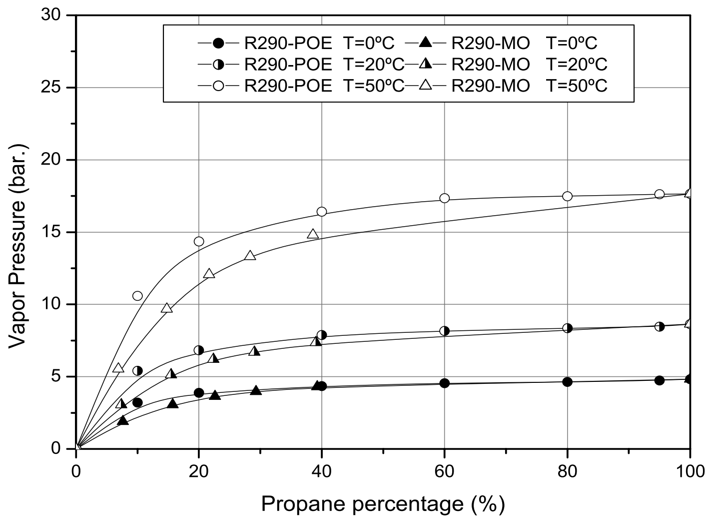


Figure B.1: Propane solubility for POE and mineral oil

B.3 Experimental set up

The selected compressor to be tested is a one cylinder reciprocating compressor. The selected lubricants to develop this comparative study were the traditional mineral oil used with propane (Clavus G68) and a POE oil (ISO 32) commonly used with HFC's. In graph B.1 can be seen the solubility of MO (from [36]) and POE oil with propane. It should be noted that the data for the mixture POE-propane are not for the used oil but for another one used for the same applications [35], this is a consequence of the fact that no data about this POE oil were available for the authors, although the properties of the used oil should not be very different from the one represented on graph B.1. In all propane tests almost pure (99.95 %) propane was used to ensure traceability of our results.

B.3.1 Characterization test

The compressor rating procedure was performed according to the relevant standards in the field such as the ISO-917, [96], and American ANSI ASHRAE 23-1993, [111].

Oil circulation rate (OCR) measurements were done following standard ANSI/ASHRAE 41.4-1996, [98]; Care was taken that the oil level at the sight glass was stable during the test.

B.3.2 Start-up test

To perform the start-up phase oil emission measurements, as it is mentioned above, a methodology has to be implemented as a consequence of the lack of standards about these kind of measurements. The same standard compressor characterization test rig was used, although the oil separator has the oil return blocked, to trap the oil expelled by the compressor. The oil separator is arranged as in fig. B.2. To perform the test the standard for oil circulation rate measurements was used as a guide, although in this case absolute quantities of oil are measured, a detailed guide of the followed steps are shown in the appendix. This measurement process was cumbersome, and had some intrinsically associated imprecisions that made that the obtained value only be useful to get an estimation of the oil outflow on the start-up. For this reason another attempts to estimate this magnitude were designed. With this motivation in mind, an additional oil level sight,designed to avoid the foaming process, was installed and filmed during the start-up process to see the oil down and correlate it with the amount of oil expelled by the compressor. This is not an exact method to quantify the exact mass that goes out of the compressor but it could be a good indication for the same kind of oil to see if some problems related with lubrication could appear in the start up process.

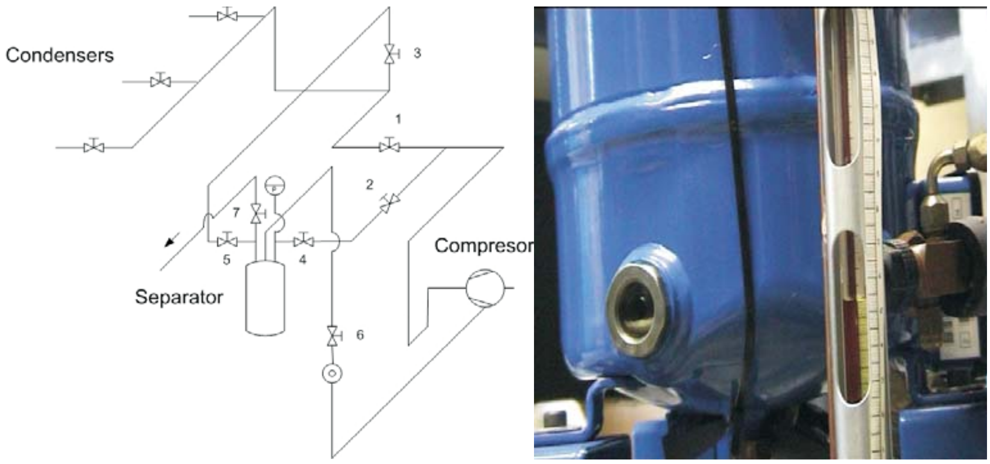


Figure B.2: On the left, start up experimental set- up. On the right, an image of the oil sight installed in the compressor

Both processes were used in parallel as complementary measurement methods to get a global image of the start-up oil outflow.

B.4 Developed Test

B.4.1 Compressor characterization

The matrix used for the characterization of the compressors working with POE and MO was chosen taking into account the conditions in which they would work in order to provide service-representative results. It covered a set of condensation temperatures ranging from 35 ºC to 50ºC condensation temperature and -10 ºC to 7.2ºC evaporation temperature. A constant superheat of 11.1 K was chosen.

After a careful characterization of the calorimeter losses, test results showed a high consistency between the result from the primary and confirmation test method as specified in standards. The mean discrepancy of evaluated capacity was lower than 1% in most case.

Oil circulation rate for a given compressor is independent of the tested conditions for a constant superheat according to[89]. For this reason three working conditions were selected to perform the measurements and its mean value was selected as the oil circulation rate.

B.4.2 Start up measurements

Measurements of the oil outflow at compressor start-up were performed in three quite different work conditions:

- The compressor works steadily at 50ºC condensation temperature, 0ºC of evaporation temperature, and a superheating of 10 K. The compressor is turned off during 10 minutes and then is restarted.
- Similar conditions to a), but with a superheating of 0-2 K at the compressor inlet. The compressor is turned off during 10 minutes and afterwards is also restarted.
- The compressor is warmed up with an electrical resistance during 8 hours and then powered on.

The objective of the two initial tests is to check the compressor start up in the usual standard conditions of a heat pump system. At the first test the superheating is high as it is usual in the real refrigeration system and the second one tries to reproduce the start up in conditions of low superheat. In this case the compressor can work in flooded conditions and this fact could due to a higher oil ejection in the start-up.

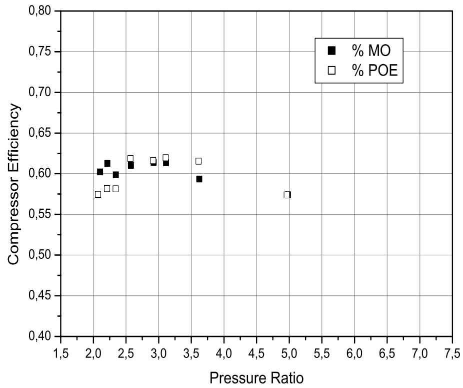
The third test tries to simulate the conditions in which a refrigeration system remains stopped during a long period of time. In these conditions, there is a substantial increase in the amount of refrigerant inside the compressor (as a consequence of the lower partial pressure of refrigerant in this part of the test rig), to avoid this the compressors are heated before the start-up.

It should be pointed out here that an important aspect to consider is the duration of start up phase relatively to lubricant emission. To do this some preliminar test in the test conditions a) and b) were developed with POE (as a consequence of its higher oil circulation rate with propane), in these tests the compressor was working during 10 minutes and the oil level down on the adapted sight in intervals of 20 seconds were registered to know in which time interval were the more significant change on the oil level. From this test, oil start-up time was fixed in 2:30 minutes, this time interval was assumed to be independent of the performed test for both oils to obtain comparable results.

B.5 Results and discussion

Volumetric and compressor efficiency for compressor working with both lubricants are shown in fig. B.3. An explicit expression for these compressor variables can be found on [89]. From fig.B.3 it can be seen that volumetric efficiency is not affected by the used oil. But regarding with compressor efficiency, fig B.3 shows that the curves for both oils, in spite of having a similar behavior, seem to be shifted, and the fall of compressor efficiency at low pressure ratios appears before for POE oil than for mineral oil. As the volumetric efficiency curve do not shows any significant divergence for both oils, the only important factor to explain the observed differences on the compressor efficiency has to be sought on the different mechanical losses as a consequence of the different viscosities of mixture oil-refrigerant between both lubricants.

The oil circulation rate results are shown in table B.5. From this data it can be seen that mineral oil presents an OCR value lower than POE, these results points out the fact that the mixture MO-propane has a higher viscosity than the mixture POE-propane, although in both cases this magnitude seems to be small enough for not being an important factor in the global performance of a refrigeration system.
In fig B.4 the oil sight level down evolution in the start-up process between intervals of 20 s. is plotted. From these curves it can be seen that for the same oil the results are similar, but there is a quantitative difference in the absolute level down between both graphs. This is an expected result taking into account the higher solubility of propane in MO. Another relevant aspect is the fact that the down oil process begins before for POE than for MO. This behavior is a straightforward consequence of the fact that POE oil has a lower surface tension than MO. This lower surface tension, additionally, due to a more violent foaming process for POE in spite of having less



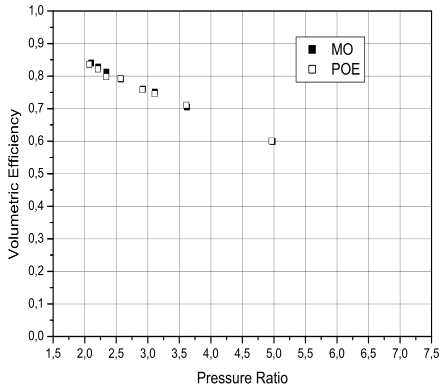


Figure B.3: Volumetric and compressor efficiency for the tested compressor working with mineral and POE oil.

(T_{ev},T_{cond})	POE(%)	MO(%)
(-10,45)	1.1	0.6
(7.2,50)	1.5	0.8
(15,50)	1.5	0.7

Table B.1: Oil circulation rate for mineral and POE oil working with propane.

solubility with propane.

In POE test, it should be noted that the curve for the superheat of 0-2 K. shows a different trend than the rest of the curves. This behavior could be explained on the basis of a higher oil circulation rate for low superheat, described in [113], that is accumulated on the evaporator producing a higher controlled temperature than the real refrigerant temperature leading the compressor to work in flooded conditions. As a consequence of this, an amount of liquid refrigerant goes into the compressor, this amount of liquid is suddenly evaporated on the start-up process making the oil level going down faster. This phenomenon is not observed for MO because of its lower oil circulation rate values.

The amounts of oil weighted are presented on table B.5 and from them it can

Test	Oil	Oil weight (g.)	Liquid sight glass level down (mm.)
Sh=10 K	POE	150	3
	MO	110	4
sh=0-2 K	POE	80	7
	MO	140	5
PTC	POE	100	3
	MO	70	5

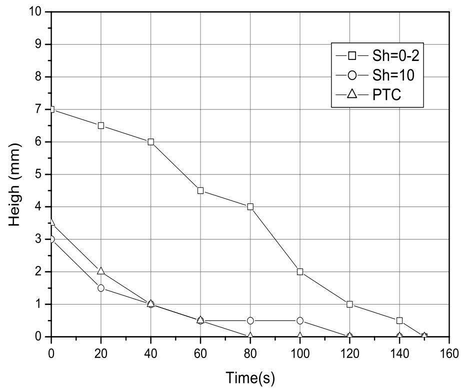
Table B.2: Start up oil outflow measurements for POE and mineral oil.

be seen that, for the accuracy of the tests, there are not significants difference in the amount of expelled oil by the compressor for both lubricants, so any problem derived from an excessive oil ejection in the start up with a compressor working with POE and propane as refrigerant should be discarded.

B.6 Conclusions

From compressor's point of view it has been seen that the behavior for both oils is nearly the same so this POE oil, commonly used with HFC's, seems to be suitable to be used with propane in spite of the small observed differences and has the advantage of a lower solubility that lets to reduce the refrigerant charge in a heat pump system, which is a very important factor for the propane. Nevertheless lower viscosity of POE compared with propane produce a higher oil circulation rate that joined to its lower solubility can generate some oil drawn problems, as it has been observed working with low superheat.

In this point it is convenient to observe the fact that in this work no durability compressor test has been developed and from the fig B.4 POE oil shows a higher mechanical losses than MO which can have a significant effect on the durability of a reciprocating compressor working with propane and this kind of oil. Based on these arguments, a POE oil with a slight increment on the viscosity respect the HFC's POE oil used in these tests should be suitable.



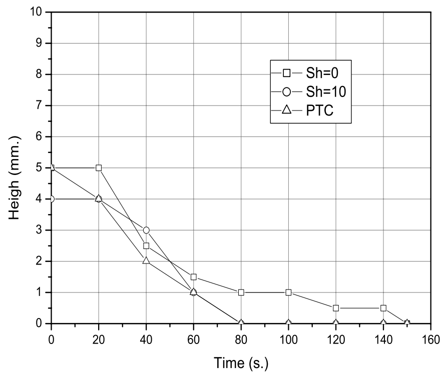


Figure B.4: Relatively compressor oil level down for POE oil (on the left) and mineral oil (on the right) in the start up process.

Regarding with the start-up process two complementary methodologies to develop these kind of measurements have been developed, and even they are not very accurate, as a consequence of the intrinsic difficulty of the process, combined show enough precision to let a qualitative evaluation about the existence of significant behavior differences between both oils. In the developed test, the initial more violent foaming for POE that should lead to a higher oil ejection is compensated by its lower solubility with propane that due to a similar expelled quantities of oil in both cases.

B.7 Appendix

Start-up weight test method description:

- Remove the separator from the installation and clean all the pieces with acetone .
- Weight the oil separator and insert it on the test rig.
- Made the vacuum inside the separator and open the valve 2 to let the refrigerant go inside the separator closing then it. In this process an small amount of oil

from the pipes goes inside the separator. We made the assumption that this amount of oil is the same that the one that do not be introduced inside of the oil separator when the compressor is turned off.

- Put the compressor in the test conditions, and open the oil separator to the test rig.
- Switch on the compressor and keep it working during the predefined start up time and then isolate the oil separator from the test rig.
- Open the valve 7 and let the refrigerant go out from the separator through the capilar without drawing the oil.
- When the manometer pressure is low enough (2 bar), remove the capilar from the rotolock valve and let it to go outside faster, supposing that, for this case the separator filter is enough to avoid that the oil goes out.
- Remove the separator from the installation, heat it since 80º C degrees and weight it since the separator keep its weight constant. This is the amount of oil expelled by the compressor.

APPENDIX C

Study and characterization of a propane heat pump installation, by the use of the infrared thermography techniques.

R. Royo, J. Gonzalvez, E. Navarro.
Proceeding of XXVI Thermosense Conference.

C.1 abstract

The use of the infrared thermography techniques is progressively showing as an important tool in the research of very different thermal applications, in this case air conditioning systems. The Instituto de Ingeniería Energética de la Universidad Politécnica de Valencia has developed a Joule Project, financed by the European Community for the Study and Characterization of propane heat pumps, adapted to the use in the South European countries. The substitution of the fluorocarboned refrigerants by natural fluids as propane is the main objective of this project. With the study, important efficiency improvements have been obtained, in relation with the use of the classical refrigerants, for both summer and winter conditions. The use of the thermography in this project has been quite useful in the achievement of several important tasks as the study of the compressor start up process, as it is described in the paper.

C.2 Introduction

In the global search after new environmentally safe alternatives to CFCs and HCFCs based refrigerants, the HFCs are the most extended alternative nowadays. However, HFCs have mainly two important disadvantages: First, despite their null ODP they present a high GWP, and this fact is causing that some UE countries put some limits to their use, and second, they are new chemical products that have never been present in the atmosphere and, as they have long mean life, the effects that these chemical products can produce in the long term are unknown. For this reasons some research are been developed to find other, environmental safety, alternatives to HFCs. In this line, HCs, and in particular, propane, has been presented as a good alternative. From the theoretical side, propane is a pure HC which offers excellent thermodynamic properties, good compatibility with most materials already present in refrigeration plants and null ODP and GWP ([104], [16] for a more systematic account of the advantages of HCs in refrigeration systems).

Regarding the main drawback of hydrocarbons, their flammability, new standards and codes have been recently released [107], [108] to allow its safe use. In this sense special attention should be paid to the one released by the European Committee for Standardization [107] fixing certain restrictions to the use of HCs as refrigerant (mainly to be used in indirect systems) and setting maximum allowable charge figures for given applications.

The use of HFCs as refrigerant has also brought the change in the kind of lubricants used as a consequence of the low solubility that these refrigerants have in the traditionally used mineral oils. For this reason new oils like POE has been developed to let the refrigeration system work in a more efficient way with these refrigerants. Despite some effort [89], there is still a considerable lack of information related with the interaction of these new oils with refrigerants.

The aim of this work is to analyze the oil emission process in the start up phase of a compressor that is working with propane as refrigerant and synthetic oil as lubricant with the help of infrared thermography techniques. These results are compared with obtained results when these tests are performed with a traditional mineral oil. This study is motivated by the fact that HCs present a higher solubility in synthetic oils than HFCs at low temperatures that joined to its lower density can lead to some lubrication problems during this process.

C.3 Description of the Infrared equipment

A FLIR ThermaCAM SC 2000 Infrared camera is used during the research project which will be described. This type of camera is quite adequate for research of transient processes, because it is allows direct digital computer acquisition, as fast as the hardware capabilities of the computer. The sequences and different type of images obtained are post processed with the use of the ThermaCAM Researcher, complete and very useful software, which allows the complete treatment and analysis of the infrared images: change of thermal focus, color palettes, radiation characteristics, etc. The responsible of the infrared tests during this project research is the author of this paper, Rafael Royo, ITC Level II certified thermographer.

C.4 Compressor characterization test rig

The installation used for the compressor characterization tests is outlined in figure C.1. It is adequate to test refrigerating compressors within the range of 5-40 kW cooling power.

The compressors are characterized according the corresponding International Standards [96], [111]. The "secondary fluid calorimeter" was selected as the main testing method.

The secondary fluid calorimeter is made up of an assembly of parallel coils of direct evaporation, acting as main evaporator. This device is held on top of an isolated container. It contains a very volatile secondary fluid and an electrical heating device. The mass of the cooling fluid is controlled by constant pressure expansion valves, located at the inlet of the calorimeter. In this installation, the calorimeter is composed of three bodies and three independent coils, so that it is possible to divide the cooling power of the installation. In that way a wide range of compressor powers can be tested.

The measurement of the refrigerant mass flow rate is carried out with 1% accuracy. The refrigerant mass flow meter is mounted on the pipe between the sub-cooler outlet and the expansion valve. In this installation a Coriolis-type measurement device with 0.15% accuracy is used.

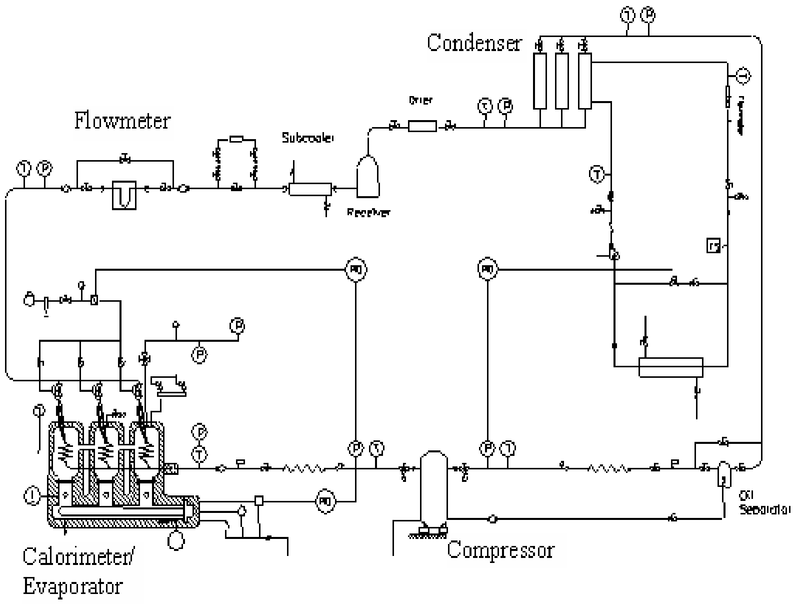


Figure C.1: Compressor characterization test rig.

Several control loops (compressor inlet and outlet pressure, superheating and sub cooling controls) have been incorporated to allow an accurate adjustment of the refrigerant conditions at compressor inlet (evaporation temperature and superheating) and outlet (condensing temperature). The rig is fully automated, making it possible to reach any allowable test conditions without manual adjustments.

C.5 Infrared overview of the main elements of the research installation

Figure C.2 shows a thermography of a compressor mounted in the research installation working at steady conditions.

Figure C.2: Visual and infrared image of a compressor characterized in the test rig.

The upper part of the case displays a uniform temperature about 38 ºC. The bottom part of the compressor is the hottest, because the lubricant oil that circulates through the different moving parts of the compressor, and is heated from the friction finally is back returned to the bottom part of the case that acts as carter or oil reservoir. It is important to point out that by the use of infrared techniques it is possible to observe the oil level inside the compressor. This is a very important feature for the tests that afterwards will be described.
As it is usual, hot refrigerant vapor exits from the compressor (see figure C.3). This hot vapor is afterwards cooled at the next element of the installation, the condenser heat exchanger unit, where the global amount of energy is finally transferred to the room or the atmosphere, according the work conditions of the cycle (respectively winter or summer conditions). It is possible to observe the hot temperature of the refrigerant vapor even although the outside part of the pipe is isolated to prevent thermal losses.

In the condensation system (see figure C.4) three plates exchangers are used to

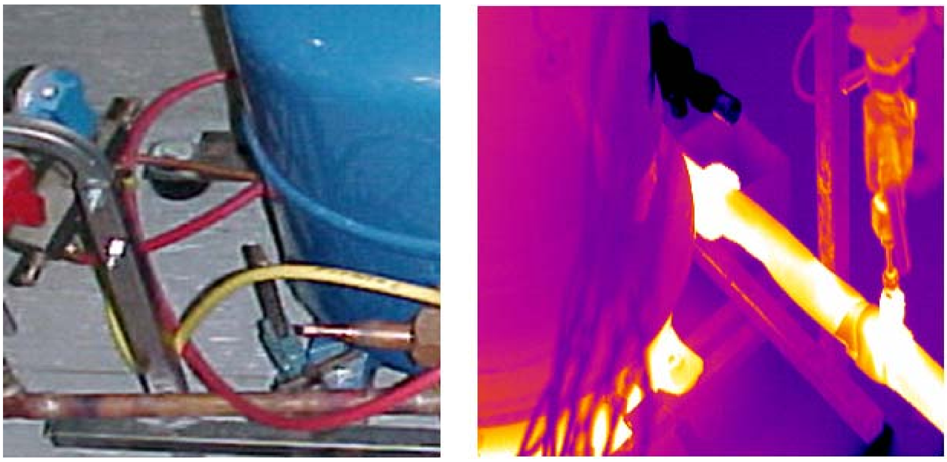


Figure C.3: Visual and infrared images of the outlet pipes of the compressor.

optimize the control of compressor outlet conditions. At the image, the usual isolation material usually used to avoid thermal losses has been removed for a better observation of the external surface temperatures.

From a thermal point of view, it is very interesting the infrared image enclosed at figure C.4: the plates heat exchanger used at the condensation unit at the research heat pump installation. This type of devices is composed of very thin plates of aluminium, with high conductivity. So, it is possible an optimal observation of the flow distribution of both flows, hot and cold fluids through the whole surface of the heat exchanger. That important feature is very useful to check if the size selection and the design of the heat exchanger are adequate. Even also to analyze if some problem exists inside this device and in that way to prevent dangerous problems as fouling or excessive pressure drop along the channels.

The next enclosed infrared images show two thermal bridges, typical places were large, even critical heat losses could be produced. With infrared thermography it is very easy to observe every place where any isolation failure exists, or simply if these thermal losses have not properly been considered at the design of the installation. That is the case at this element, the metallic flange, in direct contact with the pipe, which holds on an aluminium rod of the installation.

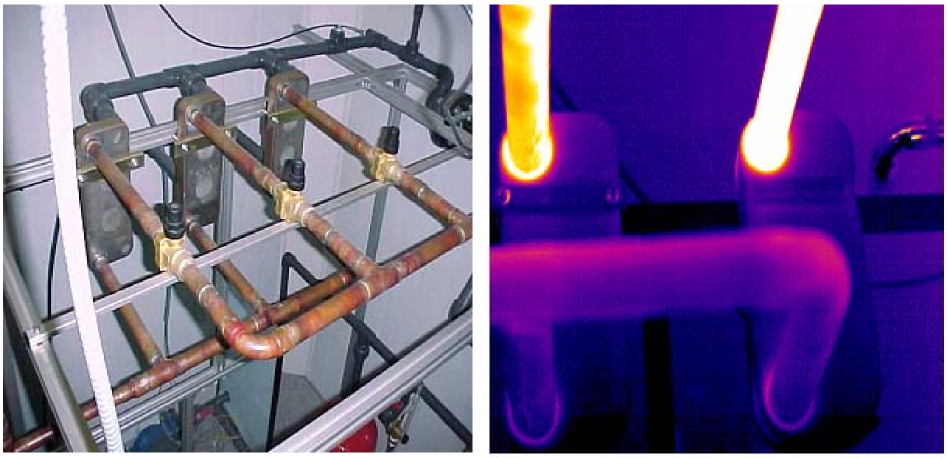


Figure C.4: Overview of the condensation heat exchanger arrangement.

C.6 Start up oil emissions measurements

In the actual heat pump systems, it is very important an adequate characterization of the compressor start up [114]. This is a transient process, typically only few minutes, in which the operation conditions are not the pre design standards, and from that reason several important problems could exist. The first objective of this study is the evaluation of the oil expelled by the compressor during the described start up process. This type of studies is relatively complex, because in reciprocating compressors the lubricant is in direct contact with the refrigerant of the heat pump, and usually the oil is soluble in it, so after the start up process, the fluid inside the compressor is a mixing of oil and refrigerant, with very different properties of the initial ones of both fluids depending of the real concentration.

The lubrication problems produced during the start up process are caused by two abnormal phenomena related with the particular conditions during this period: on one hand sudden changes in the internal compressor pressures. If a large amount of refrigerant is initially solved inside the oil, this sudden change of pressure determines a massive evaporation of refrigerant that generates large quantities of foam, on another one, in the start up process a large quantity of oil could be expelled from the compressor. In those ways the compressor could work without lubrication during

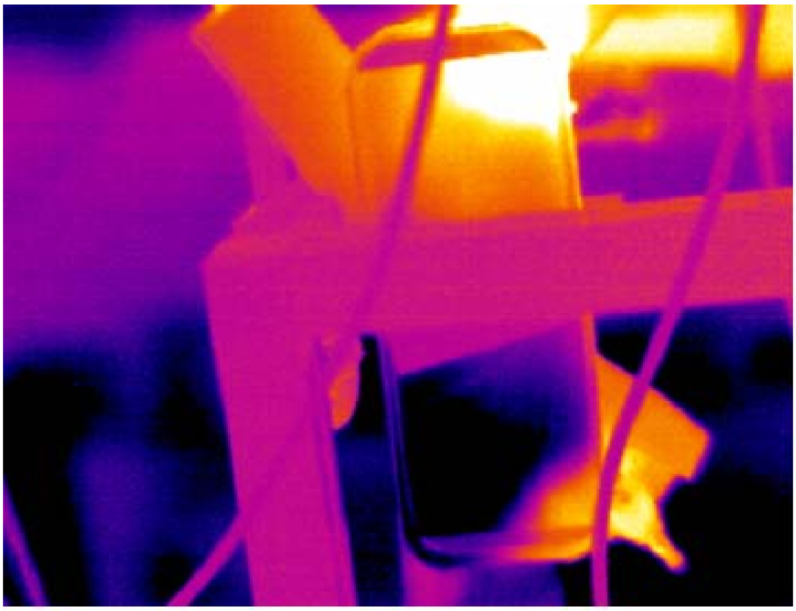


Figure C.5: Infrared image of the plates heat exchanger, showing the internal flow distribution.

few but critical minutes. Those conditions are very dangerous because the increasing of friction and wear at the moving parts of the compressor.

At the described research installation, a complete experimental work has been carried out to study the oil behavior during the start up of the compressor, even with different oils type (mineral and synthetic).
Trying to consider quite different work conditions during the compressor start up, the tests are developed at the following ones:

1. The heat pump installation works steadily with the usual condensation and evaporation temperatures, and a superheating of 10 K of the refrigerant fluid at the compressor inlet. The compressor is turn off during 10 minutes and then is restarted.

2. Similar conditions but with a superheating of 0-2 K at the compressor inlet. The compressor is turn off during 10 minutes and afterwards is also restarted.

3. The compressor starts in refrigerant flooded conditions.

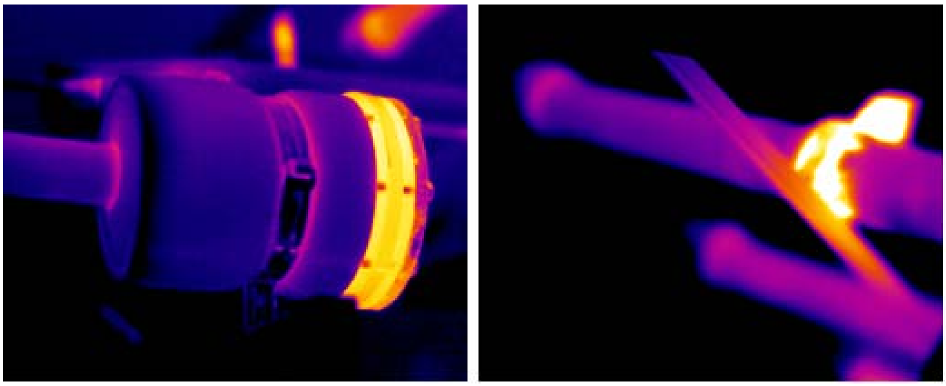


Figure C.6: Examples of thermal losses: hot surface without isolation and thermal bridge between a hot pipe and a bracket. An intense thermal bridge can be observed in this position, that afterwards was avoided by the use of the necessary isolation materials. This is only an example, because real dangerous problems could be avoided with the use of thermography at any other type of thermal installations: boilers, furnaces, engines or any other equipments at very high temperatures.

4. The compressor is warmed up with an electrical resistance during 8 hours and then powered on.

The objective of the two initial tests is to check the compressor start up in the usual standard conditions. During the normal operation of the system, the compressor regularly stops and restarts. These cycles are the normal operation of the compressor, and they have the objective to maintain the temperature of the room inside the range that is prefixed in the thermostat unit. In these usual conditions it is possible that some portion of the liquid refrigerant could arrive to the compressor. This fact is very dangerous for this component, because some moving parts could break. At the first test the superheating is very high as it is usual in the real heat pump installations.

The third test tries to simulate the conditions at a heat pump installation that remains stopped during a long period. In these conditions, there is a substantial increase in the amount of refrigerant inside the compressor (as a consequence of the lower partial pressure of refrigerant in this part of the test rig), this fact produces that a large quantity refrigerant condenses in the compressor.

The latest fourth test tries to observe the same phenomenon of start up but when the compressor is internally heated with an electrical resistance. The objective of this heating is to avoid the accumulation of liquid refrigerant inside the compressor.

C.6.1 Experimental results

During these described tests it was also measured the mass of oil expelled by the compressor during the start up period. The method consists of gather the oil in an oil separator situated in the compressor's outlet and weight it according the standard [98]. Obviously a high accuracy of the used balance is necessary.

An infrared camera was used to observe the evolution of the transient start up process, and also for the temperature measurement. The thermal pattern at the bottom part of the compressor where the oil is placed is recorded, this measurements was confirmed by a RTD situated at the cranckcase's bottom part. The temperature scale is the same for all the shown thermograms, except the third test as a consequence of the low temperature of it.

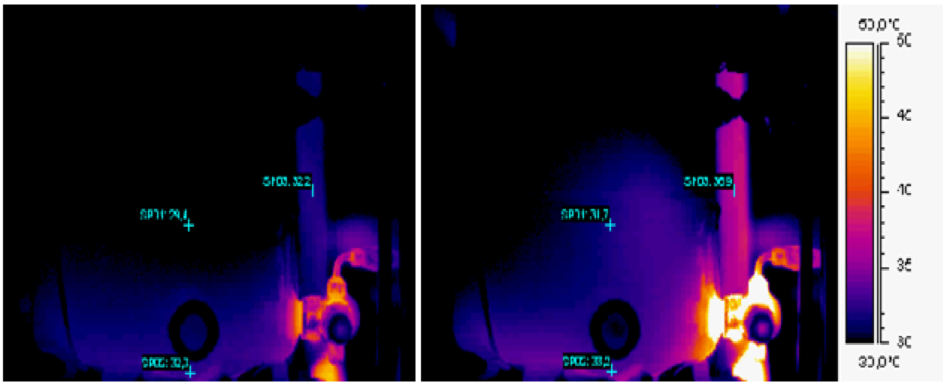
The pressure at the inlet of the compressor at every test was also measured. During the research, these data have been correlated with the properties of the mixture oil-refrigerant inside the compressor that us determined by the pressure-temperature conditions inside the compressor [115], [35], [36].

The results of these measurements are shown in the enclosed figures:

1. Figure 1: 10 K of superheating

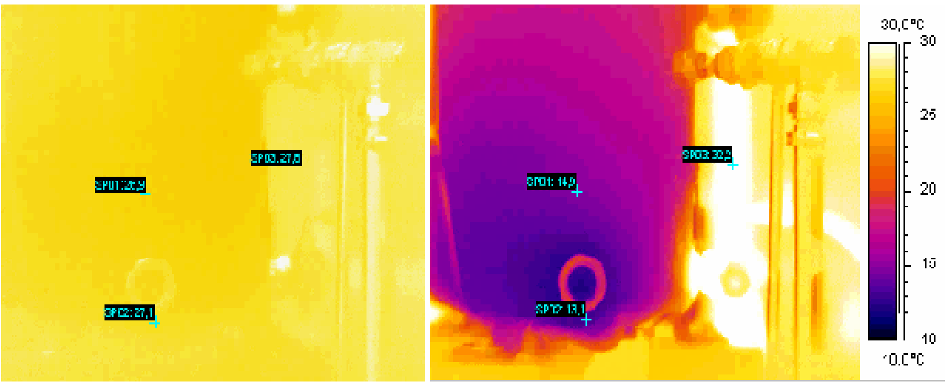
For these start up conditions, it could be observed that the compressor temperature is quite high although the refrigerant amount solved inside the oil is not important.

2. Figure 2: superheating of 0-2 K



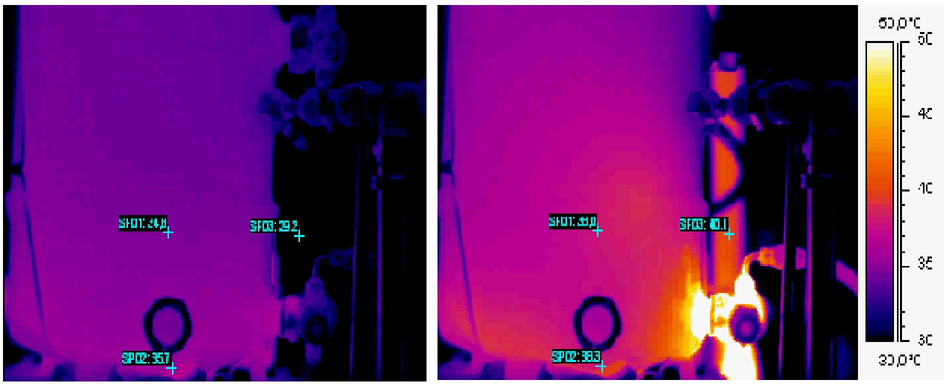
At this test with low superheating it was possible to observe little refrigerant drops coming into the compressor case. These little drops evaporate and the associated cooling effect reduces the compressor temperature. Finally the refrigerant amount solved into the oil increases. From this reason, the lubricant viscosity decreases and the foaming effect during the compressor start up is larger. All these phenomena produce dangerous conditions for the compressor performing.

3. Figure 3: compressor start up with refrigerant flooding.



At this test the lowest level of temperatures inside the compressor is achieved. So, it was necessary to change the temperature camera scale for an adequate observation of the thermal details of the transient start up process inside the compressor. The temperature is significantly reduced from the sudden evaporation of the refrigerant. These ones are the worst conditions, considering the problems related with the compressor lubrication that finally implies higher friction and wearing of the moving parts.

4. Figure 4: warmed compressor set up.



In these conditions, although the compressor is previously warmed with an electrical resistance, the compressor temperatures are even lower than those ones at the high superheating test. In that way, there are not expected lubrication problems in the compressor.

From the global experimental results it is also possible to conclude that mainly for the first case a, at the compressor outlet there is a temperature gradient along the pipe that goes from the compressor to the condenser. Although it could not be adequately verified, this fact could be related with the oil expelled from the compressor, because this fluid has a quite different thermal capacity compared with the refrigerant, so it could explain the temperature differences observed. This can suggest another interesting application of the thermography to determine the oil-refrigerant flow pattern.

APPENDIX D

A phenomenological model for analyzing reciprocating compressors.

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Accepted in International Journal of Refrigeration

D.1 abstract

A new model for hermetic reciprocating compressors is presented. This model is able to predict compressor efficiency and volumetric efficiency in terms of a certain number of parameters (10) representing the main sources of losses inside the compressor. The model provide users with helpful information about the way in which the compressor is designed and working.
A statistical fitting procedure based on the Monte Carlo method was developed for its adjustment. The model can predict compressor performance at most points with a maximum deviation of 3%.
A possible gas condensation on cold spots inside the cylinder during the last part of the compression stroke was also evaluated.

Nomenclature

η_k	Compressor efficiency
W_{ref}	Ideal isentropic work transferred to refrigerant by compresor
$\dot{E}_k$	Electric power consumption
η_s	Volumetric efficiency
$\dot{m}_{in}$	Mass flow rate
$\dot{V}_s$	Compressor swept volume flow
ρ_{in}	Density in compressor inlet
$\overline{\dot{m}}_{leaks}$	Mass flow rate leaked
$\overline{\dot{m}}_{pc}$	Mass flow rate in phase change
η_{sth}	Theoretical volumetric efficiency
ρ_i	Density in state i
V_d	Dead space
V_s	Swept volume
η_{el}	Electric efficiency
$\Delta h_{(4-5)}$	Enthalpy difference between states $4-5$
$\Delta h_{(1-8^*)}$	Enthalpy difference between states $1-8^*$
$\dot{E}_{mech}$	Energy lost in mechanical losses
Z_{elgas}	Fraction of electric motor losses going to the suction gas
Z_{mgas}	Fraction of mechanical losses going to the suction gas
c_{pi}	Specific heat at constant pressure in state i
$(UA)_{ht}$	Overall heat transfer coefficient per area for the heat transfer between the hot and the cold gas
h_{ht}	Heat transfer coefficient per area for the heat transfer between the hot and the cold gas
T_i	Temperature in state i

D	Cylinder diameter
S	Cylinder stroke
n	Compressor nominal speed
d_h	Effective hydraulic diameter
k	Specific heat at constant pressure
μ	Dynamic viscosity
k	Thermal diffusivity
P_i	Pressure in state i
ξ	Drag factor for compressor inlet and outlet valve
ζ	Equivalent flow resistance for leakages
w	Refrigerant velocity
R	Universal gas constant
A_L	Leakage effective area
h_{fg}	Heat of vaporization
A_v	Area for phase change inside the cylinder
h_{pc}	Heat transfer coefficient per area for phase change inside the cylinder
nz	Number of cylinders
K_i	Representative parameters for different model losses
ΔT_{sh}	Compressor inlet gas superheat
Rp	Pressure ratio

D.2 Introduction

Reciprocating hermetic compressors have been known since the 19^{th} century and, due to their simplicity and flexibility when working in a wide range of conditions, they are still used nowadays in refrigeration and air conditioning systems.

A complete empirical characterization procedure for this type of compressor is described in [116]. However, a certain theoretical effort based on analysis and modelling may be useful at this point to estimate how a given compressor is likely to work under different operating conditions or with different refrigerants, but also in more general terms, to assess the proper operation of the compressor.

Two main categories of models are discussed in the literature:

- Models whose aim is to explain in a detailed and accurate manner, the behavior of specific parts or processes (mechanics of the valves, vibration, heat transfer) inside the compressor. In [117] one can find numerous examples of models of this type. Here the main objective is to assist the compressor optimization.
- Models whose objective is to describe the compressor globally. In this category, three main basic approaches to the problem can be established:
  1. Correlations from experimental data for some of the compressor significant variables such as COP, cooling capacity, [116], [118]. This is the methodology most commonly used by compressor manufacturers, but it does not give any valuable physically information about the processes inside the compressor. The correlations obtained can only be used for the range of conditions in which they were obtained.
  2. Other authors ([56], [119], [120], [121]) have attempted to model the most important physical compressor processes using numerical methods to solve the differential equations implied in the conservation laws of these processes. Although these kinds of models may deliver considerable information about the way in which the compressor is working, they usually require numerous data available only to the manufacturer. These models aims to optimize compressor design.
  3. The so–called semi–empirical models, for instance [83], [80], [84], seek to reproduce the main compressor performance variables like COP and cooling capacity using empirically adjusted, simple models retaining at least some portion of the physical background. Given their simplicity, these models do not need as much information as the detailed models described in approach (2. As a consequence, the information obtained is not as accurate, and there are usually problems in the physical interpretation of certain results.

This research describes a global analysis of a series of hermetic reciprocating compressors covering different strokes, piston numbers, sizes and refrigerants.

The results of this research were divided in two papers. In this first paper, a new phenomenological model which aims to identify the most important phenomena occurring inside the compressor and the corresponding empirical coefficients required for their adjustment were described. In the second paper [122], the resulting adjustment of the model to experimental results from a vast experimental test campaign is discussed, and the empirical coefficients obtained are analyzed. Finally, a full discussion about the physical sense and interpretation of the obtained values, as well as the capabilities of the model to help in the analysis of the behavior and internal characteristics of the studied compressors is presented.

The proposed compressor model aims to reproduce the compressor efficiency ($\eta_k = \frac{W_{ref}}{\dot{E}_k}$) and the volumetric efficiency ($\eta_s = \frac{\dot{m}_{in}}{\dot{V}_s \rho_{in}}$) as a function of a set of parameters which may be obtained by correlations of standard characterization performance data. The philosophy is that these parameters have a physical background, so that once correlated, the model can be used to predict compressor performance under operating conditions which are not tested, for example, at extreme temperatures or at lower speeds. Furhter, the correlated model may estimate the compressor performance with other refrigerants for which there are no available data.

In the literature, one can find other models which follow this same approach (see [83], [84]). The present study attempts to move forward towards that objective, targeting the selection of the parameters in such a way that they retain the maximum physical significance. The values obtained in the correlations are expected to show a clear agreement with the reasonable order of magnitude of the compressor characteristics they represent.

Additionally, the fitting techniques employed have shown to have a considerable influence on the suitability of the obtained coefficients. The authors found that classical least square correlation methods are not useful in this kind of non linear systems. Monte Carlo based fitting methods, on the contrary, provide much greater freedom and stability in the definition of the model parameters, better final results and a better way of avoiding excess sensitivity problems.

This first paper is structured in three main parts. First, the model is presented;

then the Monte Carlo fitting procedure is described and finally the model is validated for a compressor providing an initial assessment of the order of magnitude of the obtained parameters.

D.3 Compressor model for a reciprocating compressor

Examining the evolution of the refrigerant in a p-h diagram, the model will assume the evolution shown in fig. D.1. The refrigerant enters the compressor at point 1 ("inlet conditions"). The reference for the compressor efficiency is given by an isentropic condition from the inlet to the outlet of the compressor: 1-8*. The real conditions at the outlet of the compressor are indicated by state 8 in fig. D.1. The developed model assumes that the evolution of the refrigerant through the compressor can be divided in the following sequence of effects:

- 1–2: Vapor heating due to motor cooling and mechanical loss dissipation.
- 2–3: Vapor heating due to the heat transferred from the hot side of the compressor (discharge) to the inlet flow.
- 3–4: Isoenthalpic pressure lost at the suction valve.
- 4–5: Isentropic compression from real cylinder intake conditions (leaks and possible condensation also appear in this part of the process).
- 5–6: Isoenthalpic pressure lost at the discharge valve.
- 6–7: Vapor cooling due to the heat transferred to the suction side.

Regarding the evolution from 4 to 5, inside the cylinder, measurements from [7] show that the real compression from conditions at the bottom of the piston (beginning of the compression stroke) to the top dead centre are very close to isentropic. The main source of irreversibilities is the heat transfer to and from the wall during the compression stroke. However, the process is very fast and wall temperatures are quite close to fluid temperatures; thus, heat transfer effects per mass flow rate unit are slight. For this reason, a generalized polytropic compression was avoided since the polytropic exponent must be fitted for each refrigerant.

The internal leakage of refrigerant through the piston rings has a considerable effect on compressor and volumetric efficiencies. In order to simplify the treatment of this loss, an evaluation of the leak is made at state 5. It is considered as if the loss of the circulating mass flow rate ($\overline{\dot{m}}_{leak}$) takes place at state 5. Therefore, the compressor is assumed to consume the work of compression for the circulating mass flow rate plus leaks ($\dot{m}_{in} + \overline{\dot{m}}_{leak}$).

The possibility of condensation of a relatively significant fraction of refrigerant during compression was also evaluated throughout this study. The condensed mass does not flow towards the discharge, but it is later evaporated during the suction stroke. The effect of this possible condensation is similar to an increase of the dead space ratio. In the model, this possible effect is treated as a loss of the circulating mass flow rate ($\overline{\dot{m}}_{pc}$).

Other effects that influence refrigerant temperature before leaving the compressor (7-8 in fig. 1) such as heat release to the environment or oil heating and electric motor heating of the vapor are not considered in this study. With these assumptions, a model describing compressor and volumetric efficiencies of the compressor is proposed.

The circulating mass flow rate can be calculated from the ideal flow rate by means of the expression:

$$\dot{m}_{in} = \eta_{Sth} \dot{V}_s \rho_4 - \overline{\dot{m}}_{leak} - \overline{\dot{m}}_{pc} \tag{D.1}$$

In expression (D.1), $\dot{V}_s$ is the swept volume flow given by $\dot{V}_s = n \cdot V_s$, being V_s the swept volume, n the nominal speed of the compressor and $\eta_{Sth} = 1 - \frac{V_d}{V_s}(\frac{\rho_{8*}}{\rho_1} - 1)$ the ideal volumetric efficiency [123]. It should be noted here that the compressor nominal speed was considered constant throughout the study. For the line frequency of 50 s^{-1}, the manufacturer estimates a value of 2900 rpm for these compressors.

Expression (D.1) implies that the total mass flow rate is given by the total cylinder refrigerant capacity at conditions corresponding to point 4 in fig. D.1, subtracting the cylinder volume losses from the cylinder clearance, the vapor that is leaked during the compression process and the possible formation of small refrigerant droplets in some part of the cylinder surface that are not subsequently pumped out of the cylinder during the compression process.

Electric compressor power input $\dot{E}_k$ can be assumed as the energy that the refrigerant consumes to change from state 4 to state 5 in fig. D.1 plus the energy that the compressor consumes in mechanical ($\dot{E}_{mech}$) and electrical losses. So $\dot{E}_k$ can be expressed as:

$$\dot{E}_k = \frac{1}{\eta_{el}}[\Delta h_{(4-5)}(\dot{m}_{in} + \overline{\dot{m}}_{leak}) + \dot{E}_{mech}] \tag{D.2}$$

From both equations, (D.1) and (D.2), the corresponding expressions for the volumetric and compressor efficiencies are obtained:

$$\eta_s = \frac{\dot{m}_{in}}{\dot{V}_s \rho_1} = \frac{\rho_4}{\rho_1}\eta_{Sth} - \frac{\overline{\dot{m}}_{leak}}{\dot{V}_s \rho_1} - \frac{\overline{\dot{m}}_{pc}}{\dot{V}_s \rho_1} \tag{D.3}$$

$$\eta_k = \frac{\dot{m}_{in}\Delta h_{(1-8^*)}}{\dot{E}_k} = \frac{\Delta h_{(1-8^*)}\eta_{el}}{\Delta h_{(4-5)}(1 + \frac{\overline{\dot{m}}_{leak}}{\dot{m}_{in}}) + \frac{\dot{E}_{mech}}{\dot{m}_{in}}} \tag{D.4}$$

The different processes considered in expressions (D.3) and (D.4) will now be described in further detail.

D.3.1 Vapor heating due to motor cooling and mechanical loss dissipation

The heating of the inlet refrigerant by motor cooling and mechanical loss dissipation are quantified by the following expression:

$$\dot{Q}_{1-2} = (1 - \eta_{el})\dot{E}_k Z_{elvapor} + \dot{E}_{mech} Z_{mvapor}$$

where $Z_{elvapor}$ and Z_{mvapor} are the fractions of the losses transferred to the suction vapor as heat. It is assumed that the fraction of absorbed heat is the same for both losses, that is $Z_{elvapor} = Z_{mvapor}$, so these factors can be renamed as K_1. The remaining heat is released to the environment either through the outlet gases or the compressor surface.

The increase in the suction vapor temperature 1-2 is thus given by:

$$\Delta T_{1-2} = \frac{\dot{Q}_{1-2}}{\dot{m}_{in} c_{p1}} = K_1 \left(\frac{(1-\eta_{el})}{\eta_k} \frac{\Delta h_{(1-8^*)}}{c_{p1}} + \frac{\dot{E}_{mech}}{\dot{V}_s \eta_s \rho_1 c_{p1}} \right) \tag{D.5}$$

D.3.2 Vapor heating due to heat transferred from the hot side of the compressor (discharge) to the inlet flow

Before leaving the compressor, the hot vapor flowing outside the cylinder heats the refrigerant at the low pressure side. As a first approximation, the heat transferred between both sides can be given by $\dot{Q}_{2-3} = (UA)_{ht}(T_{8^*} - T_1)$, where the temperature difference $(T_{8^*} - T_1)$, calculated from the isentropic – ideal process, is considered as an effective temperature difference, characteristic of the process.

The global heat transfer coefficient U_{ht} is related with a heat transfer coefficient h_{ht} using these approximations:

- The heat transfer between both sides takes place mainly in the suction and discharge pipes near the cylinder.
- U_{ht} is considered proportional to the heat transfer coefficient h_{ht}, $U_{ht} = C' \cdot h_{ht}$.
- The heat transfer coefficient h_{ht} is considered as the one given for the turbulent internal flow in a pipe.

$$Nu = C \cdot Re^{0.8} Pr^{0.4} \rightarrow h_{ht} = \frac{k}{d_h} \cdot C \cdot \left(\frac{\dot{V}_s \cdot \eta_s \cdot \rho_1}{d_h \cdot \mu} \right)^{0.8} \left(\frac{\mu \cdot c_p}{k} \right)^{0.4} \tag{D.6}$$

Finally, considering these approximations, the temperature increase is expressed as follows:

$$\Delta T_{2-3} = K_2 \frac{(T_{8^*} - T_1)}{(\dot{V}_s \eta_s \rho_1)^{0.2}} \frac{k_2^{0.6}}{d_h^{1.8} c_{p2}^{0.6} \mu_2^{0.4}} = K_2' \frac{(T_{8^*} - T_1)}{(\dot{V}_s \eta_s \rho_1)^{0.2}} \frac{k_2^{0.6}}{c_{p2}^{0.6} \mu_2^{0.4}} \tag{D.7}$$

where d_h is a hydraulic diameter characteristic of the narrow flow passages around the cylinder. If no reasonable estimation of these passages is available, the hydraulic diameter can be included within a new constant $K_2^{'} = \frac{K_2}{d_h^{1.8}}$.

D.3.3 Isoenthalpic pressure losses at the inlet valve

This pressure drop is estimated as $\Delta P_{3-4} = \xi_3 \rho_3 \frac{w_3^2}{2}$. Expressing the velocity as $w_3 = \frac{\dot{V}_s \eta_s}{nz A_{c3}}$ where nz is the number of cylinders and A_{c3} the effective inlet valve area per cylinder, the pressure drop is given by:

$$\Delta P_{3-4} = K_3 \rho_3 (\frac{\dot{V}_s \eta_s}{nz})^2 \tag{D.8}$$

where K_3 is given by $K_3 = \frac{\xi_3}{2A_{c3}^2}$.

From a practical point of view, it may be useful to express the inlet valve area A_{c3} in terms of the cylinder diameter $A_{c3} = \widetilde{K_3} \cdot D^2$. Thus, if the value of K_3 is known for one compressor, an estimation can be made for other compressors of similar characteristics but different cylinder diameters.

D.3.4 Isoenthalpic pressure losses at the outlet valve

Using the same arguments as in the previous section, the pressure loss for the outlet valve is given as:

$$\Delta P_{5-6} = K_4 \rho_5 (\frac{\rho_0}{\rho_5} \frac{\dot{V}_s \eta_s}{nz})^2 \tag{D.9}$$

where K_4 is given by $K_4 = \frac{\xi_5}{2A_{c5}^2}$, and A_{c5} is the effective outlet valve area per cylinder.

D.3.5 Leakages

To describe leakages, it is assumed that they are a function of the overall pressure difference in the compressor and are not dependent on the vapor flow. The mass flow rate leaked can be deduced using the ideal vapor non compressible flow through an orifice as:

$$\overline{\dot{m}}_{leak} = nz \cdot A_L K_5 \sqrt{\frac{\Delta P_m}{\zeta \rho_m}} \rho_m \tag{D.10}$$

where ζ represents the equivalent flow resistance, and A_L is the effective leakage area per cylinder. A_L can be considered proportional to the cylinder area ($A_L \approx \widetilde{K_5} D^2$), as done for the compressor inlet and outlet valve area. The pressure difference $\Delta P_m = K_5^2(P_{8^*} - P_1)$ is assumed to be proportional to the overall pressure difference ($P_{8^*} - P_1$) and the mean density for leaked vapor $\rho_m \approx \sqrt{\rho_{8^*}\rho_1}$. With these assumptions $\overline{\dot{m}}_{leak}$ is given by:

$$\overline{\dot{m}}_{leak} = K_5^{'} \cdot nz \cdot D^2 \sqrt{\Delta P_m \rho_m} \tag{D.11}$$

where $K_5^{'} = K_5 \cdot \widetilde{K_5} \cdot \zeta^{-\frac{1}{2}}$.

$\overline{\dot{m}}_{leak}$ influences the reduction of the total mass flow rate (see eq. (D.1)) and also affects the refrigerant temperature in the cylinder inlet, which is estimated as the corresponding mixing temperature:

$$\Delta T_{3-3'} = \frac{\overline{\dot{m}}_{leak}(T_{8^*} - T_1)}{\dot{m}_{in} + \overline{\dot{m}}_{leak}} \tag{D.12}$$

The use of more accurate expressions, like the compressible flow equation for unchoked flow -maybe more precise- would not significantly improve the results of the model,as commented in [124], and they would certainly lead to much longer computation times. The authors consider that the above approximation is a reasonable compromise between simplicity and physically relevant information regarding the internal process in the present model.

D.3.6 Refrigerant phase changes inside the cylinder

Some refrigerant liquid droplets can be formed on the colder areas of the cylinder. These droplets can be in continuous phase change, leading to a reduction in the total amount of vapor flowing through the compressor and thus affecting volumetric efficiency. The inlet valve is exposed to relatively cold vapor on the suction side; therefore, its temperature could fall below the dew point of the refrigerant during some fraction of time of the compression stroke. Thus, condensation may occur and some droplets could appear on the valve plate. These droplets would then evaporate during suction and, assuming that this evaporation takes place on the valve plate, a cooling of the valve plate would take place to allow again condensation in the subsequent cycle.

It is worth noting that the condensation part of this phase change process is quite a critical point because it has very little time to be produced (the pressure inside the cylinder is only higher than the refrigerant dew point for some fractions of a second).

Considering that the most important factor to facilite this process is the temperature difference between the cylinder inlet and outlet, the temperature difference $(T_{8^*} - T_1)$ may be considered as an effective temperature difference to characterize this process. The amount of condensing refrigerant for one compression cycle of the cylinder may be expressed as:

$$\dot{m}_{pc} = \frac{\tau \cdot nz \cdot h_{pc} \cdot A_v(T_{8^*} - T_1)}{h_{fg}(T_1)} \tag{D.13}$$

where τ is the time during the compression in which the condensation can be produced (time in which the pressure in the cylinder is higher than the dew point for the inlet temperature) and h_{pc} is an effective heat transfer coefficient, considered as approximately constant.

Expression (D.14) to be inserted into the model must be expressed in terms of the mass flow per time unit and not per cycle:

$$\overline{\dot{m}}_{pc} = \frac{\overline{\tau} \cdot nz \cdot h_{pc} \cdot A_v(T_{8^*} - T_1)}{h_{fg}(T_1)} = K_6 \frac{nz(T_{8^*} - T_1)}{h_{fg}(T_1)} \tag{D.14}$$

where $\overline{\tau}$ represents the percentage of the total cycle in which the pressure is high enough to produce the refrigerant condensation.

Regarding this phase change effect, previous researchs can be found in literature, for instance, in [103] for low speed piston compressors working in wet conditions. Nevertheless, there is no experimental information available regarding this phenomena in the present research and here, it is more intended as a proposal to explain some model results and to inspire further research.

D.3.7 Mechanical loss influence on the compressor efficiency

According to [125], the mechanical losses may be considered as a sum of two terms, one proportional to the energy consumption rate and the other dependent on the compressor speed:

$$\dot{E}_{mech} = K_7 \dot{E}_k + K_8 n^2 = K_7 \frac{\dot{V}_s \eta_s \rho_1 \Delta h_{(1-8^*)}}{\eta_k} + K_8 \cdot n^2 \tag{D.15}$$

D.3.8 Final model structure

To formulate the global model, the equations governing the different losses described in sections (2.1)-(2.6) are either introduced into eq. (D.3) and (D.4) by direct substitution of the obtained equations (eq. (D.11), (D.14) and (D.15)) or are used to calculate the refrigerant state 4 and 5 (eq. (D.5), (D.7), (D.8), (D.9) and (D.12)). This leads to a system of two implicit equations for the compressor and volumetric efficiencies:

$$f_1(\eta_k, \eta_s, \mathbf{G}, \mathbf{K}, \eta_{el}, \frac{V_0}{V_s}, P_1, P_{8^*}, \Delta T_{sh}) = 0 \tag{D.16}$$

$$f_2(\eta_k, \eta_s, \mathbf{G}, \mathbf{K}, \eta_{el}, \frac{V_0}{V_s}, P_1, P_{8^*}, \Delta T_{sh}) = 0 \tag{D.17}$$

where $\mathbf{K}(\mathbf{K} = (K_1, ..., K_8))$,$\eta_{el}$, $\frac{V_0}{V_s}$ represents compressor design parameters difficult to determine, whereas $\mathbf{G}$ ($\mathbf{G} = (G_1, ...G_n)$) stands for compressor design parameters that are easy to know like stroke (S), number of cylinders (nz), nominal speed (n), and the like. If for any reason some of these $\mathbf{G}$ parameters were not known, they can easily be regrouped inside the $\mathbf{K}$ parameters.

Once all the compressor design parameters **K** are known, the system of two equations can be solved for compressor and volumetric efficiencies, η_k and η_s, for a given working condition (P_1, P_{8^*}, ΔT_{sh}). Any solver for a system of non-linear equations can be employed. The results shown in this paper were obtained by the Gauss-Seidel procedure [91]. With this algorithm, given an initial value of 0.5 for compressor and volumetric efficiencies, the convergence to the solution is typically reached in less than fifteen iterations.

As commented above, the parameters **K**, η_{el}, $\frac{V_0}{V_s}$ are difficult to estimate. A set of data for a number of working conditions obtained either from experiments or from manufacturer catalogs is required to obtain the proper value of **K** by a fitting procedure. The developed fitting procedure to find the best estimation of **K** is explained in section (3).

From the best obtained values of **K**, it is then possible to determine the value of compressor and volumetric efficiencies in conditions different from those tested. Besides, it is possible to obtain physically relevant information about the internal processes inside the compressor and to quantify the different losses.

A key assumption regarding **K** is that the different parameters are not significantly dependent on the test conditions or employed refrigerants. The obtained results indicate that this assumption is fairly reasonable.

D.4 Statistical fitting procedure

As explained in section 2, the last step to close the model is to estimate the compressor losses parameters **K** from experimental or catalog data. This is quite a critical issue in this kind of models because the dependency of the target functions on the parameters is non linear, and there is also some sort of indetermination, in the sense that several possible combinations of parameters could adjust the model properly with a final deviation in the predicted efficiency values which is smaller than the experimental uncertainties. In fact, several conventional non linear regression techniques (the standard routines in MsExcel, Origin, simplex algorithm [91]) were tested, yet they failed in the fitting process of the proposed model.

For this reason and to avoid possible problems arising from a step–by–step exploration of the parameter space (existence of a local minimum solution, too smooth dependence on certain parameters and the like), an heuristic algorithm based on a Monte Carlo type approach was designed. A review of the main trends in this field can be found in [92]. Although computationally not the most efficient these methods show great versatility and reliability.

A general scheme of the designed algorithm is shown in fig. D.2. In this scheme, the program starts by assigning pseudo-random values (according to the uniform distribution) to the parameters and the "best" combination of them to fit the compressor and volumetric efficiencies data is sorted out (this is called the *first process*). The routine used to generate the pseudo-random numbers was the one proposed by Park and Miller [93]. This routine has a long enough period for this specific application. To select the "best" combination of parameters, an error function or residue (ϵ) must be defined. In this case, the Euclidean norm weighted by the standard deviation of each experimental point i was selected:

$$\epsilon = \sum_i \frac{\sqrt{\eta_k^2(x_i) - \eta_{kexp}^2(x_i)}}{\sigma(x_i)} + \frac{\sqrt{\eta_s^2(x_i) - \eta_{sexp}^2(x_i)}}{\sigma(x_i)}$$

The set of parameters with a lower value of ϵ is selected as a solution to the first process.

Nevertheless, as a consequence of experimental errors, intrinsical errors associated to the model and the nature of the mathematical functions involved, if the process is repeated, a different set of parameters is obtained as a solution which may also give a good value for the error function ϵ.

Therefore, this process is repeated until a representative map of the solutions in the parameter space is obtained (this is called the *second process*). This means that the probability distribution of the solutions for each parameter is obtained.

The result of this process is a set of probability distributions for each of the model parameters. From these distributions, the most probable value for each parameter is selected as the best fit value.

Some comments should be made regarding this scheme:

- An interval in which the value of the parameters must be found, must be defined. As a result of the intrinsical stability of the method, this is not a critical point in the model, yet a good selection of this interval reduces the number of iterations needed to find a suitable solution.
- Preliminary studies have been developed to determine the proper number of iterations in the first and the second process. For the first process, this number is reached only if, increasing the number of iterations, the order of magnitude of ϵ does not change. For the second process, this number is reached if, by increasing the number of iterations, the obtained parameter distribution function does not change.
- To reduce the high computational cost linked to the direct use of REFPROP [94] in the evaluation of the thermodynamical properties of the refrigerants, an approximation based on linear interpolations of bidimesional meshes from REFPROP was employed (see [95] for details).

D.5 Results from the analysis of a two–piston hermetic compressor

To validate the model and the fitting methodology, a two–cylinder reciprocating compressor working with propane was analyzed using the set of experimental data from [126] (compressor ST, in the aforementioned nomenclature). A comparison between the obtained results with and without the phase change term is also presented to understand the possible relevance of this term.

After an initial study of possible correlations among the different parameters **K** for several compressors, it was found that K_1 and K_2 showed some kind of coupling, which can only be solved with a very high number of trial points (in the first process of the Monte Carlo method). In order to keep the fitting time moderate and after finding that the value of K_1 was almost the same for several compressors, the criteria of assigning a constant value of 0.9 provided very good results.

Electric efficiency is a function of the operating conditions, yet this dependence is usually small in the nominal operation range of the compressor. For this study, information from the manufacturer about experimental electric motor efficiency was available. This information shows a maximum deviation of $\pm 1\%$ over the mean value of the electric efficiency in almost all the tested points, so that electric efficiency can be considered independent of the operation conditions throughout the whole study. Although this parameter was known, to check the model capacities, this parameter was left free during the fitting process because it is usually not a parameter available in catalogs.

Table D.1 shows the value obtained for parameters **K** in both cases (including or not the phase change term), together with the interval in which the parameters were searched. The search interval was chosen to include all the possible values that the compressor parameters might have in an attempt to cover the widest possible range bounded only by the physical constraints.

Regarding the Monte Carlo methodology, 1,000,000 trial runs were required to obtain a stable value for the error function in the first process and 1,000 iterations were needed in the second process to obtain a representative distribution of probability of the space of parameters.

The values obtained for the parameters, despite the large number of assumptions involved in the model, provide a very good prediction of the compressor performance throughout the entire test sample. Further, they have values consistent with the compressor geometry and the physical process involved, as described in the following:

- K_2: As seen in table 1, the obtained value for $K_2^{'}$ is 2.80, if a value of 0.023 for the constant C in expression D.6 is taken (dittus-boelter correlation [127]), and considering that $U \sim \frac{h}{2}$, the constant $K_2^{'}$ represents the relation between an effective heat transfer area and an effective hydraulic diameter and is given by the term $K_2^{'} = \frac{0.023 \cdot 4^{0.8}}{2 \cdot (\pi)^{0.8}} \frac{A_{ht}}{d_h^{1.8}}$. Therefore, the relation between both magnitudes is $\frac{A_{ht}}{d_h^{1.8}} \sim 215$. As d_h should be quite small, this value supports the hypothesis that the heat transfer area of this process should be small.

- K_3: The measured value for the inlet valve orifice section A_{c4} was 160 mm^2, assuming that it is approximately the value of the valve flow section. The value obtained for K_3 gives a value for the drag factor of the inlet valve of $\xi = 0.99$, which is quite reasonable considering the approximations involved in the process.

- K_4: The measured value for valve orifice section A_{c5} was 100 mm^2, assuming that it is approximately the value of the valve flow section. The value obtained for K_4 also gives a value of $\xi = 3.77$ for the drag factor of the outlet valve. This value seems too high, perhaps indicating that irreversibilities around the discharge valve are much higher.

- K_5: According to the estimations of [7], $A_L \sim 1 \cdot 10^{-6} m^2$. This value is consistent with the one obtained in the model for the constant $\frac{K_5 A_L}{\sqrt{\xi}}$.

- K_6: In [126] a value of 0.037 was given for the dead space ratio of the compressor, the value obtained for the dead space ratio in the model with non phase change inside the cylinder was 0.068. This value was considerably larger than that expected (0.037) thus, the possible condensation of some fraction of refrigerant over a possible cold spot in the cylinder head was considered to explain this loss of mass flow. This assumption resulted in to an estimation of 0.039 for the dead space ratio. This value was quite near the expected.
  In the model with a phase-change inside the cylinder, the value obtained for the constant representative of this process was $K_6 = \overline{\tau} \cdot h_{pc} \cdot A_v = 1.73$. Considering, as a rough estimation, that:

  1. The moment in which the condensation can occur is $\overline{\tau} \sim 0.12\%$ of the

total time of one revolution. This time is supposedly equal to the time of the process of delivering vapor through the outlet valve (period of time when the pressure inside the piston is higher).

2. The area A_v in which this process can be produced is the inner surface of the inlet valve ($A_v \sim 7 \cdot 10^{-4}\ m^2$).

The obtained heat transfer coefficient is $h_{pc} \sim 20000(W \cdot m^{-2} \cdot K^{-1})$; it is quite reasonable if, for example, a dropwise condensation phenomena is produced, since [128] and [129] reported heat transfer coefficients for dropwise condensation up to $300000(W \cdot m^{-2} \cdot K^{-1})$.

- K_7, K_8: The obtained values for the mechanical losses, approximately 10%, are in agreement with the expected value for this type of compressor.

- η_{el}: This is the most influential factor in compressor efficiency. The obtained value for η_{el} is 0.859, which is very close to the actual mean electric motor efficiency data $\eta_{el} = 0.862$.

Compressor and volumetric efficiencies obtained using both model versions and their relative errors are shown in figs. D.3, D.4, D.5, D.6. The differences between model results, using the best fit parameter values, and the experimental data are always lower than 5%. The correlation coefficients obtained between experimental and calculated compressor and volumetric efficiencies are shown in table D.2.

The results show that both versions of the model (with and without phase change) reproduce the experimental results accurately enough. The values obtained for the different parameters in both models are in the same range. In brief, the effect of the phase change parameter seems to be equivalent to an increase in the dead space.

In fig. D.7, the vapor temperatures at point 7 of the pressure–enthalpy diagram (fig. D.1) are plotted against measured compressor outlet temperatures. The temperature given by the model in state 7 of this figure, in principle, can only be considered as a rough estimation of the real temperature at the exit of the compressor (point 8 in fig. D.1). In any event, this estimation is actually quite good for low and medium pressure ratios where the temperature of the refrigerant is not high, and it is worse at high pressure ratios, where the temperature of the refrigerant is higher and some processes not considered in the model, like the heat transfer of the hot refrigerant to the crankcase, by radiation, could become more relevant.

D.6 Conclusions

A model for reciprocating compressors was developed. This model can reproduce the compressor and the volumetric efficiency with an error lower than 3% under a wide range of operating conditions. Although this model was developed for hermetic piston compressors, as a consequence of its general conception, it may be applied to analyze and describe any kind of positive displacement compressor.

Considering the main sources of losses, the model is based on an ideal evolution of the refrigerant throughout the compressor . This model has 10 empirical parameters, each with a direct physical interpretation. If these parameters are unknown, they must be fitted with some empirical data.

To this end, a statistical fitting methodology based on Monte Carlo techniques was designed. This methodology was tested on one compressor and the results with 16 experimental points are quite good. To apply the developed fitting methodology, only data commonly available in catalogs are required.

One drawback of the developed model is that the volumetric and compressor efficiencies are presented as a system of implicit equations. Fortunately, this is not a major drawback nowadays, given the existence of many numerical solvers adequate for this kind of system. Although the developed fitting methodology is very stable, the computational time involved in the fitting process is quite long (6 h. in a Pentium IV processor, 3.4 Ghz., 1 Gb RAM).

All model parameters have a direct physical interpretation and characterize the design and performance of the compressor. In general, the developed model could be quite useful in:

1. Estimating compressor performance at operating points, different from the experimental points used on the fit. Furthermore, the model could be used to estimate the compressor performance with another refrigerant, other compressor speeds or with slight modifications in the cylinder geometry, for instance.

2. Characterizing the compressor performance with 10 parameters and analyzing their adequacy from their absolute value or from comparison with a set of reference parameters. For example, unusual values of motor electric efficiency, mechanical losses or valve losses, can point to internal compressor malfunctions.

The value for the dead space ratio found for the studied compressor is too large (6.9%). Considering the possibility of a condensation phenomenon on a cold spot inside the cylinder, like the inner surface of the inlet valve, during the end of the compression stroke leads to a value of this parameter (3.9%) more similar to the real value (3.7%). A study of the order of magnitude of this process revealed that a dropwise condensation could explain this process. Together with similar results obtained from other compressors studied, this result seems to support the possible existence of such a phenomenon.

In the paper [122], the model proposed here will be used to analyze a set of hermetic reciprocating compressors working with propane and the capabilities of the model to predict compressor behavior using other refrigerants will be evaluated.

Figure Index

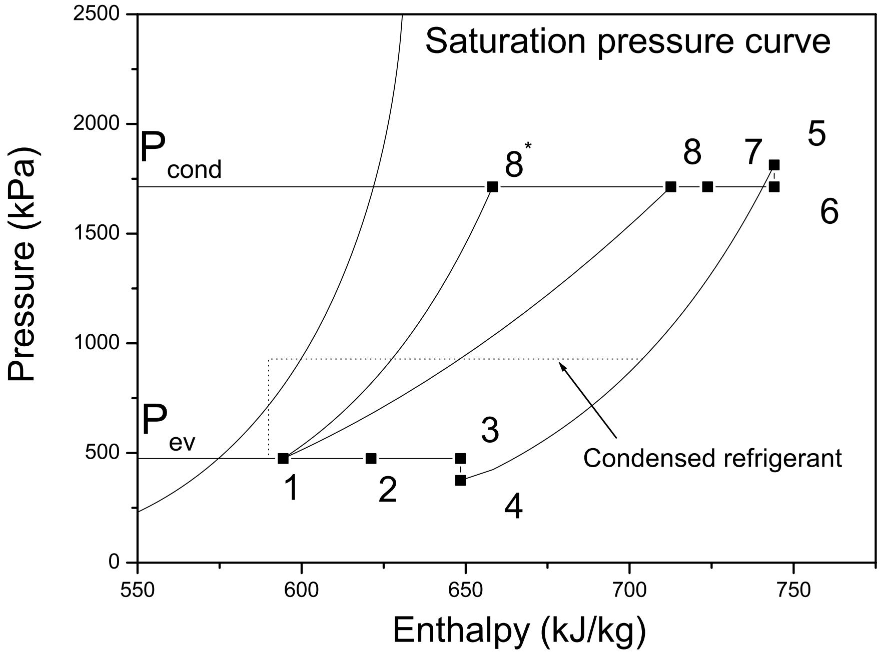

Saturation pressure curve
Pressure (kPa)
Enthalpy (kJ/kg)
P cond
P ev
Condensed refrigerant
8*
8
7
5
6
3
1
2
4
2500
2000
1500
1000
500
0
550
600
650
700
750

Figure D.1:

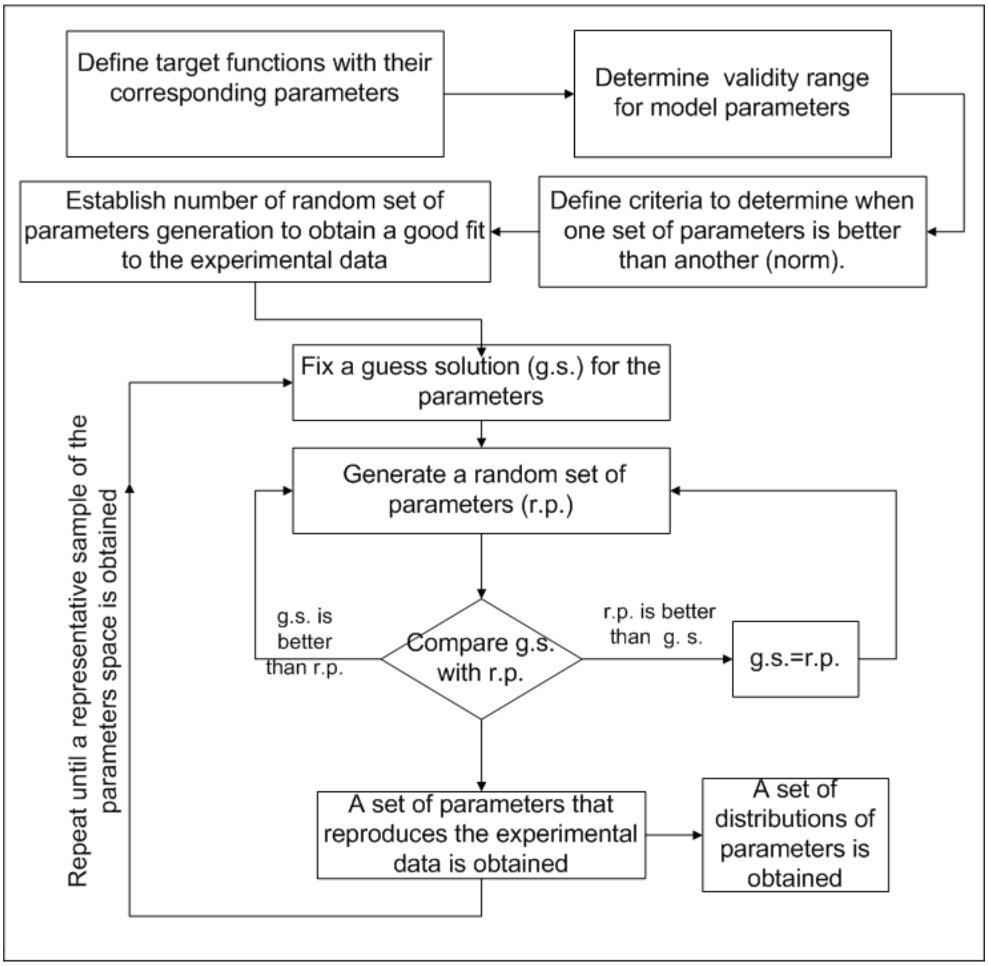

Define target functions with their corresponding parameters
Determine validity range for model parameters
Establish number of random set of parameters generation to obtain a good fit to the experimental data
Define criteria to determine when one set of parameters is better than another (norm).
Fix a guess solution (g.s.) for the parameters
Generate a random set of parameters (r.p.)
g.s. is better than r.p.
Compare g.s. with r.p.
r.p. is better than g. s.
g.s.=r.p.
A set of parameters that reproduces the experimental data is obtained
A set of distributions of parameters is obtained
Repeat until a representative sample of the parameters space is obtained

Figure D.2:

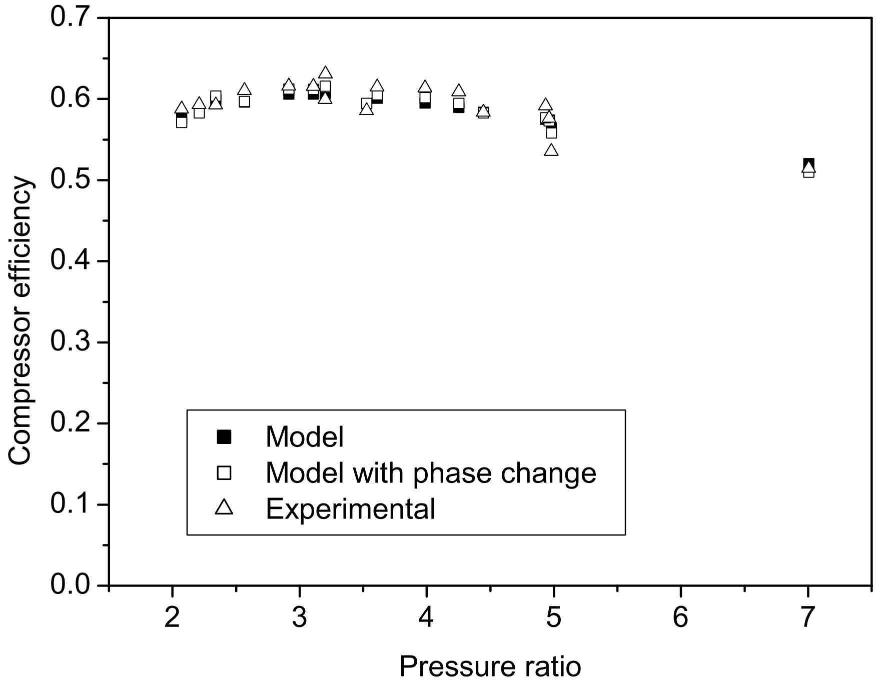

0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
Compressor efficiency
2
3
4
5
6
7
Pressure ratio
Model
Model with phase change
Experimental

Figure D.3:

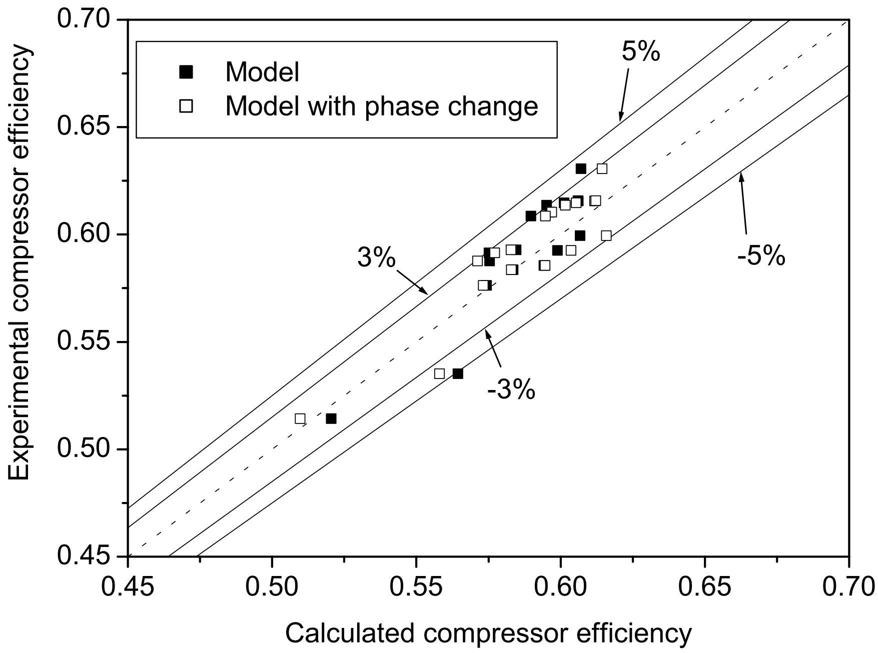
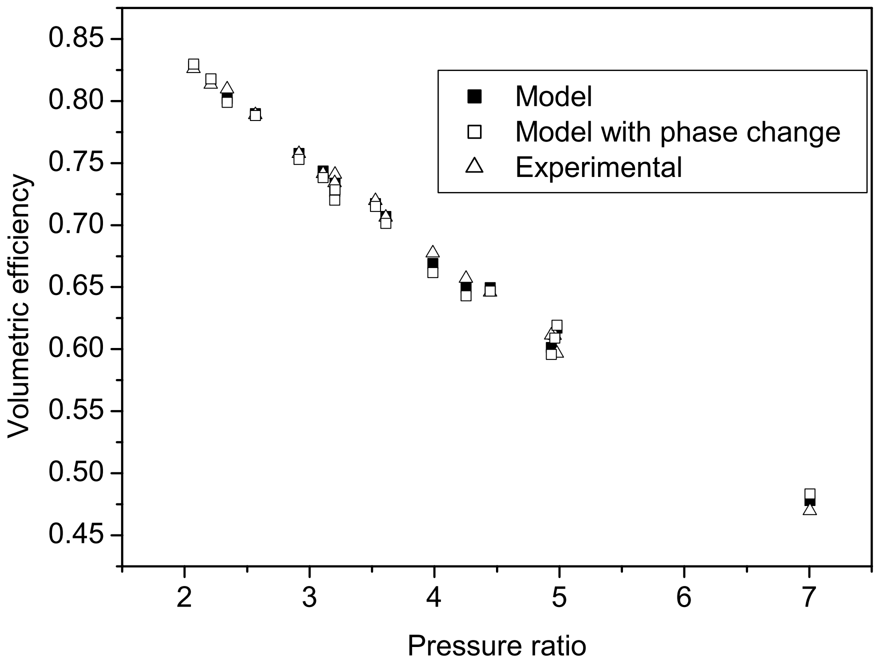

0.70
0.65
0.60
0.55
0.50
0.45
0.45
0.50
0.55
0.60
0.65
0.70
Experimental compressor efficiency
Calculated compressor efficiency
Model
Model with phase change
5%
3%
-5%
-3%

Figure D.4:



0.85
0.80
0.75
0.70
0.65
0.60
0.55
0.50
0.45
Volumetric efficiency
2
3
4
5
6
7
Pressure ratio
Model
Model with phase change
Experimental

Figure D.5:

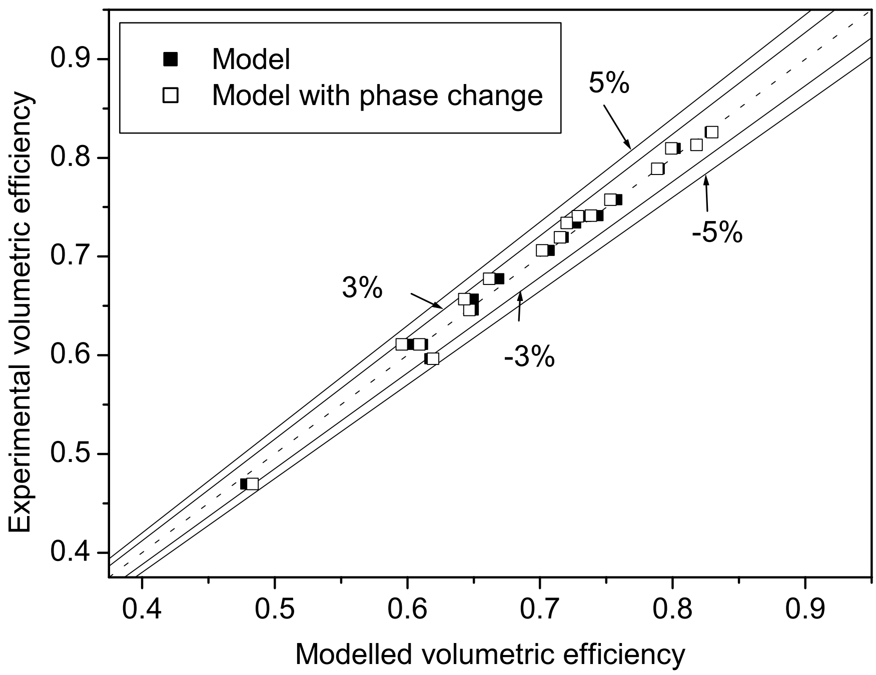

Model
Model with phase change
5%
3%
-3%
-5%
Experimental volumetric efficiency
Modelled volumetric efficiency
0.4
0.5
0.6
0.7
0.8
0.9

Figure D.6:

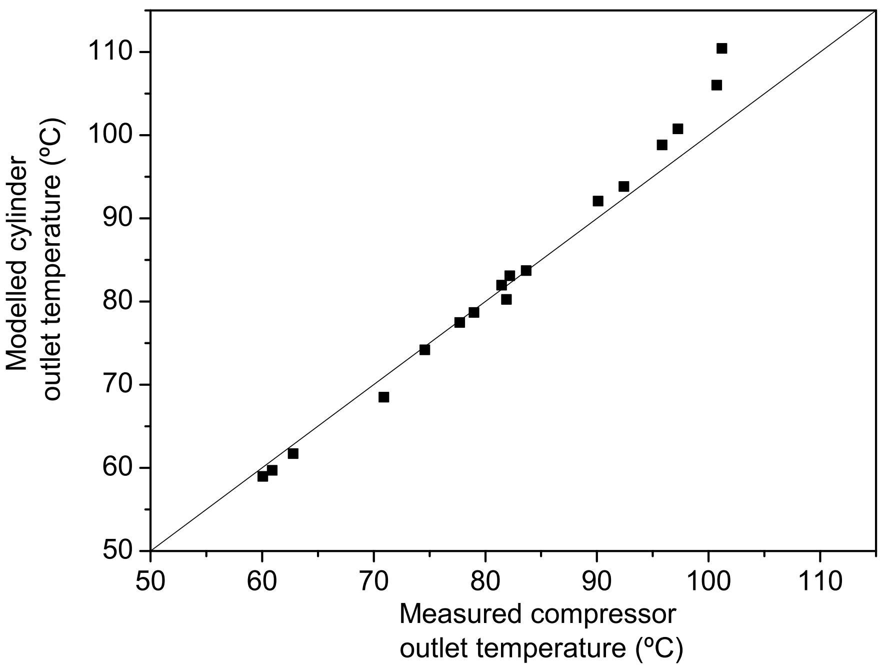

110
100
90
80
70
60
50
Modelled cylinder
outlet temperature (ºC)
50
60
70
80
90
100
110
Measured compressor
outlet temperature (ºC)

Figure D.7:

	Parameter definition	Model without phase change	Model with phase change inside the cylinder	Search interval
K_1	K_1	0.90	0.90	[—]
$K_2^{'}$ (m^{-5})	$C^{'} \cdot C \cdot \frac{A_{ht}}{d_h^{1.8}}$	2.80	2.69	[0-18]
K_3 (m^{-2})	$\frac{\xi_3}{2A_{c3}^2}$	$1.94 \cdot 10^7$	$1.89 \cdot 10^7$	$[0 - 8.74 \cdot 10^7]$
K_4 (m^{-2})	$\frac{\xi_4}{2A_{c4}^2}$	$3.85 \cdot 10^8$	$3.4 \cdot 10^8$	$[0 - 2.5 \cdot 10^9]$
$K_5^{'} \cdot D^2$ (m^2)	$\frac{A_L \cdot K_5}{\sqrt{\zeta}}$	$0.95 \cdot 10^{-6}$	$0.85 \cdot 10^{-6}$	$[4 \cdot 10^5]$
V_d/V_s	V_d/V_s	$6.77 \cdot 10^{-2}$	$3.90 \cdot 10^{-2}$	[0-0.4]
K_6	$\overline{\tau} \cdot h_{pc} \cdot A_v$	0.0	1.73	[0-18]
K_7	K_7	$5.11 \cdot 10^{-2}$	$4.64 \cdot 10^{-2}$	[0-1]
K_8 (Kw)	K_8	0.2052	0.2051	[0-3.73]
η_{el}	η_{el}	0.859	0.862	[0.5-1]

Table D.1: Definition and obtained value of each model parameter for the compressor studied.

	Model without phase change	Model with phase change inside the cylinder
η_k	0.932	0.943
η_s	0.996	0.994

Table D.2: Correlation coefficient obtained for the compressor and volumetric efficiencies for both versions of the model.

APPENDIX E

Performance analysis of a series of hermetic reciprocating compressors working with R290 (propane) and R407C.

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Accepted in International Journal of Refrigeration

E.1 Abstract

In this paper, a series of compressors with different capacities and geometries working with propane as refrigerant are analyzed in terms of the compressor model developed in [130]. The relative influence of the diverse compressor losses is estimated as a function of the operating conditions.

In addition, a comparison study between propane and R407c was carried out for one compressor and the observed differences were analyzed in terms of the compressor model. This study was also useful to verify the model's goodness with the aim of predicting the compressor performance with a not tested refrigerant.

E.2 Introduction

A model for compressor and volumetric efficiencies of reciprocating compressors is developed in [130]. This model characterizes compressor performance in terms of certain compressor design parameters, each having a physical interpretation. To determine the value of these parameters, specific data of compressor and volumetric efficiencies are required to fit the model to this data. Theoretically, when these parameters are known, the model can be applied:

1. to obtain information as to how the compressor works.
2. to use the performance data for one refrigerant to estimate of the compressor performance for another refrigerant.
3. to obtain data regarding the causes of the different performance between refrigerants.

This paper aims, on the one hand, to verify the capabilities of the aforementioned model, and on the other, to use the model to analyze a compressor series working with propane, examining the relative expected differences when this refrigerant is replaced by R407C. Furthermore, this work could complement other developed works such as [32] or [43] in the direction to optimize the design of a heat pump working with a natural refrigerant like propane.

E.3 Scope of the study

Four compressors of different capacities working with propane were analyzed in this study. There were three main objectives:

- To use the model described in [130] to verify its reliability and robustness when the refrigerant is changed (from R290 to R407C).
- To obtain information from the adjusted model as to how compressor design and operating conditions influence the different losses and, specifically, to gain insight into the relative importance of such losses under different operating conditions.

- To gain insight into how the different thermophysical characteristics of R290 and R407C affect compressor performance.

The main characteristics of the selected compressors are detailed in table E.1. In these compressors the stroke and the cylinder number are changed to study of the dependence of the compressor performance on both variables. The cylinder diameter of all the compressors is constant.

The four compressors were characterized working with propane in the test rig described in [126]. The compressor rating procedure followed the relevant standards in the field such as the ISO-917, [96] and the American ANSI ASHRAE 23-1993, [111].

To evaluate the response of the model when the refrigerant is changed, the compressor ST was also characterized working with R407C. As R407C is a mixture, special care was taken in the process of charging the test rig to avoid any modification in the mixture properties. In addition, certain modifications in the test rig described in [126], such as the removal of the liquid bottle at the condenser outlet, had to be made to avoid changes in mixture properties.

The matrix for these tests is shown in fig. E.1. All the tests were carried out with a superheat of 11.1 K. It was not possible to reach the specified conditions: (5ºC, 35ºC) and (7.5ºC, 35ºC) with fluid R407C in the test rig used in this study. Instead, conditions (5ºC, 40ºC) and (7.2 ºC, 45ºC) were characterized for this refrigerant. The compressor characterization results for propane are presented in [126]. REFPROP [94] was employed in the evaluation of the thermodynamical properties of the refrigerants and dew point of R407C was used to evaluate the properties of the refrigerant mixture.

E.4 Results for propane

The model was adjusted to the experimental results for all the compressors following the fitting process described in [130]. The relative deviations of the calculated compressor and volumetric efficiencies from the experimental results are presented in figs. E.2 and E.3. In almost all conditions, the error is lower than 3%.

Table E.2 shows the obtained correlation factor between calculated and measured efficiencies.

The adjusted model parameters for each compressor are shown in table E.3 (see [130] for the definition and meaning of these parameters). In this table, the K_i obtained for the different compressors are quite similar, an indication of the consistence and robustness of the developed model. The values for the parameters K_i provide an estimation of compressor performance over the entire test matrix for the four compressors with an error lower than 5% (see figs. E.2 and E.3).

E.4.1 Discussion of results

In this subsection, the meaning and consistency of the obtained K_i values is discussed.

- K_1 : This parameter represents gas heating from motor cooling and mechanical losses dissipation. It is coupled to parameter K_2 (gas heating due to the heat transferred from the hot side of the compressor - discharge- to the inlet flow) in such a way that it is not possible to obtain their values separately. Thus, a good estimation for this K_1 is needed, and, assuming that K_1 should not depend on the analyzed compressor, its value was fixed at 0.9 for all compressors, which means that 90% of the electrical and mechanical losses are wasted increasing the temperature of the inlet gas. This assignment provides very good results that justify the assumptions.

- K_2 : As explained earlier, this parameter represents gas heating due to the heat transferred from the hot side of the compressor (discharge) to the inlet flow. The difference between the value of K_2 for all the compressors is small enough to consider that it is included within the parameter error. In any case, the value of K_2 shows a slight tendency to decrease as the compressor size increases. The differences of this parameter among the compressors leads to differences in the increase of the suction gas temperature at the cylinder inlet of approximately 1 K which may be considered as a slight effect.

- K_3 : This parameter represents the pressure lost through the inlet valve. Similar values were obtained for the different compressors. In any event, the compressors with a larger stroke (LO, LT) show a slightly smaller value for this constant. The differences in the value of K_3 between the long and short stroke were also found in the value of K_4, meaning that the pressure losses through the valves are higher for shorter strokes. This trend in the values of K_3 and K_4, taking into account the fact that pressure losses were assumed to depend on the square of the gas velocity, allows one to draw the conclusion that they actually depend on the velocity with an exponent slightly higher than the supposed 2. Fig. E.4 represents the pressure drop versus the pressure ratio indicating that the pressure drop in the inlet valve is almost not dependent on the test conditions.

- K_4 : This parameter represents the pressure lost through the outlet valve. As discussed before, this parameter also seems to be slightly lower for compressors with a larger stroke. Fig. E.4 shows that the absolute differences in the pressure drop are small and that the pressure drop for the outlet valve is higher for small pressure ratios, decreasing considerably for large pressure ratios in which the pressure drop in the inlet and outlet valve is almost the same.

- $K_5^{'} \cdot D^2$: This parameter represents the leakage mainly through the gap between the piston and the cylinder during the compression process. The value obtained for this parameter for all the compressors can be considered as constant. This parameter influences compressor performance in two ways:

  1. *Reduction of the mass flow rate pumped by the compressor*: Fig. E.5 illustrates, according to the model, the relative mass flow rate leaked versus the pressure ratio. The relative leakages significantly increase with pressure ratio. On the one hand, the leakages are higher when the pressure ratio increases; on the other, as the pressure ratio increases, the total mass flow rate decreases. Consequently, these two effects combine to produce the relative increase of leakages. In addition, fig. E.5 shows that the relative leakage is slightly more significant when the number of cylinders is increased and that for the same number of cylinders the amount of refrigerant leaked is greater for larger stokes.

2. *Increase of the inlet temperature at the cylinder inlet*: In fig. E.6 the increase in the refrigerant temperature on the suction side of the compressor as a consequence of the leaked gas reinjection is depicted. As observed, the higher the pressure ratio, the higher the superheat.

- $\frac{V_d}{V_s}$: This parameter (dead space ratio) shows quite a significant deviation from the real geometric value for the four compressors. Compressors SO and ST have a geometric dead space ratio of 0.037, whilst the one obtained from the fitting process of the model is approximately 0.07. By contrast, compressors LO and LT have a geometric dead volume ratio of 0.029, and the one obtained from the model is approximately 0.055. Thus, it seems that the fitting process tends to overpredict the geometric value by a factor of approximately 2. As discussed in [130], a possible phase change inside the cylinder was considered in the model to explain this deviation in the dead space ratio, and much better results were obtained.

- K_6 : This parameter represents the quantity of refrigerant which undergoes a phase change in the cylinder (dropwise condensation at the inlet valve).

  In [130] it was established that all the parameters except the dead space ratio remain mainly unaltered when the phase change term is considered in the model. Taking this into account, the results shown in table E.3 correspond to the parameters of the model without condensation, except for the values of K_6 and $(\frac{V_d}{V_s})'$, which correspond to the model considering phase change effects.

  The value obtained for the dead space ratio in the model with this phase change is quite close to the expected geometrical one. Regarding K_6 (phase change parameter), the values for the four compressors are also quite similar. Further, The absolute value obtained was found to be quite meaningful (see discussion in [130]). These facts support the hypothesis that this effect indeed exists, and weilds a considerable influence particularly when dealing with propane.

Basically the hypothesis is that condensation may occur on a "cold spot" at certain points in the cylinder. For instance, the inlet valve is exposed to relatively cold gas at the inlet. The valve plate may then be acting as a regenerative heat exchanger: if the reed valve temperature is lower than the dew point temperature, then condensation could occur on the valve plate during the compression and delivery phase. Any condensate formed will evaporate during the following suction phase and, if we assume that this evaporation takes place on the valve plate, this will subsequently cool the valve plate material so that it is ready for a new condensation process in the following compression cycle. Provided that the dynamics of these heat exchange processes are fast enough, the amount of gas delivered by the compressor will decrease and cause a similar net effect on the performance as an increased dead space.

Fig. E.7 shows the calculated percentage of the mass flow rate in phase change versus pressure ratio. The increase in the percentage with the pressure ratio is due to the increase in the temperature difference. Fig. E.7 also shows that at high pressure ratios, the percentage of mass flow rate lost as a consequence of this effect could be quite high for propane (12%). A clear dependence on the stroke appears in fig. E.7, being the condensation fraction larger for the shorter stroke. This could be a consequence of the fact that the supposed cold area in the cylinder where the condensation occurs is the same for both kinds of cylinders since all of them have the same valve plate, but the one with a shorter stroke has less refrigerant inside.

- K_7 and K_8: These parameters represent the two terms of mechanical losses. The former is proportional to energy consumption (K_7) and the latter, to the velocity of the compressor (K_8). Regarding the results shown in table E.3, two comments should be made:

  1. The values obtained for K_7 are similar for the four compressors except for LO. The fact that the value obtained for compressor LO is completely different from the others seems to point to some kind of problem or malfunction in this compressor. This agrees with the experimental results

obtained with these compressors (see [126]). Thus, LO showed slightly lower values for compressor efficiency compared to the other three compressors.

2. K_8 shows the highest scattering among all the parameters considered. The values obtained for the short stroke compressors (SO and ST) seem to be consistent since they only differ in the number of cylinders (SO has one cylinder and ST has two cylinders). However, the results obtained for compressors LO and LT do not maintain that relationship; in fact, a factor of 3 is observed. This difference could be explained as a mechanical problem in compressor LO, as mentioned before.

- η_{el}: This parameter represents motor electric efficiency and was considered constant for all the experimental points of each compressor, since it was known that the compressors are working in the flat region of the motor efficiency curve in all conditions within less than 1%-2%. The values obtained from the model for the electric efficiency (table E.3) are in very good agreement with the available data for this value.

Regarding the compressor outlet temperature, the model is only able to calculate the refrigerant temperature at the cylinder outlet. Fig. E.8 illustrates the calculated cylinder outlet temperature for all the compressors versus the measured compressor outlet temperature. This figure shows that the cylinder outlet temperature calculated by the model can provide a useful estimation of the compressor outlet temperature if the compressor outlet temperature is not very high. At high temperatures other effects not considered in the model, like heat transfer to the environment or discharge vapor heating by the motor, could increase the difference between the cylinder outlet temperature and the compressor outlet temperature. Even though no experimental data regarding temperatures are provided to the model, it is able to predict the outlet temperature (quite a sensitive parameter) with reasonable accuracy. This fact gives more consistency and confidence to the developed model

E.4.2 Relative influence of each loss considered in the compressor and volumetric efficiencies

Figs. E.9 and E.10 illustrate the relative influence of the losses on compressor and volumetric efficiencies in all the available test conditions for compressor ST as a function of pressure ratio. Fig. E.9 shows that electric and mechanical losses are most influential in the reduction of compressor efficiency (totalling approximately 75%), the pressure losses being quite important at low pressure ratios. The opposite behavior is observed for the leakages, reaching 15% relative influence in the reduction of compressor efficiency at the highest pressure ratios. The heat transfer between suction and discharge is responsible for approximately 7% of the reduction in compressor efficiency.

Regarding volumetric efficiency, from fig. E.10 it can be concluded that electric and mechanical losses are also considerable, their influence remains approximately constant under all working conditions (totalling approximately 55%). Pressure losses at the suction valve are as great as mechanical and electric losses are for low pressure ratios. The influence of heat transfer between compressor inlet and outlet is also quite significant for volumetric efficiency, representing more than 15% of the total losses under most conditions. The phase change effect is considerably more influential at high pressure ratios where the temperature difference between the inlet and the outlet of the cylinder is quite high and the mass flow rate is lower, being at these conditions the second most influential factor in the loss of volumetric efficiency.

Figs. E.11 and E.12 graphically illustrate the relative influence of losses at specific conditions: 50 ºC condensation temperature and 0 ºC evaporation temperature.

E.5 Influence of the tested refrigerant

The results obtained for compressor ST performance with propane and R407C are given as a function of pressure ratio in Figs. E.13 and E.14. Propane leads to slightly higher compressor efficiencies and lower volumetric efficiencies.

In this section, the model's ability to estimate compressor performance with different refrigerants is evaluated and the differences between the operation characteristics with the two refrigerants are analyzed in terms of the model.

To estimate compressor performance with one refrigerant when the parameters of the compressor have been obtained for another, theoretically, only the refrigerant properties need to be changed, maintaining the value of all the parameters as constant since they depend mainly on compressor design. Of course, this will only be valid if the refrigerants and the operating conditions are similar. For instance, the compressor torque should not vary too much; otherwise the operating point of the motor would be far from that of the design and the electric efficiency would be quite different.

Based on the values of the model parameters used to describe the performance of compressor ST working with propane, the compressor model was used to estimate the performance of the compressor working with R407C. Figs. E.15 and E.16 show the values obtained, and the agreement is in general quite good (deviations lower than 5% in almost all the conditions). Specifically, the compressor efficiency is well predicted. By contrast, the volumetric efficiency is underpredicted for all the conditions (mean deviation of approximately 3% for all the points).

After determining the influence of each parameter, it was found that a value for the phase change parameter K_6 of 0.8 instead of 1.7 significantly improves the predictions of the model (solid dots in fig. E.16). This could mean that the phase change effect inside the cylinder is substantially lower for R407C. The final correlation factors for compressor and volumetric efficiencies are 0.945 and 0.993, respectively. These values are similar to the correlation factors obtained for propane.
The different value of the phase change parameter K_6 may be explained by the fact that the definition of this parameter includes in its expression a heat transfer coefficient h_{ph} which, given the lack of information about this process, was considered constant in the definition of the model (see [130] for a detailed explanation). It is possible that this heat transfer coefficient, characterizing the dropwise condensation of this process is dependent on certain thermophysical properties of the refrigerant such as viscosity, surface tension or differences in the refrigerant-oil mixture properties that could differ significantly among the tested refrigerants. This unknown dependence may be responsible for the behavior observed.

Regarding the performance differences between the two refrigerants and according to the model, the following comments should be made:

- The cylinder inlet temperature is approximately 5 K higher for R407C than for R290. This is a consequence of the higher cylinder outlet temperature for R407C.

- On average, R407C shows a higher pressure drop through the valves than propane (approximately 4% higher through the inlet valve and 11% through the outlet valve).

- The relative influence in the reduction of the mass flow rate as a consequence of the leaks is higher for propane, probably because propane has a lower density than R407C, thus a lower viscosity, allowing this refrigerant to flow throughout the piston ring more easily. The relative difference between the leakage percentage could reach 50% at high pressure ratios.

- Fig. E.17 presents data regarding the calculated cylinder outlet temperature versus the measured compressor discharge temperature. The deviations between the cylinder outlet temperature and compressor outlet temperature are smaller for R407C than for propane (compare with fig. E.8). Therefore, the losses to the environment and other losses not considered in the model seem to be more pronounced for propane than for R407C.

From the conclusions of this analysis and from the experimental results presented in [126], it appears that propane compared with R407C tends to improve its performance at low pressure ratios, while worsening it at high pressure ratios. This seems to be related with the lower pressure losses with propane at low pressure ratios, as well as higher leakages for propane at high pressure ratios. Both effects may be attributed to the lower propane density.

Finally, it should be pointed out that the efficiency of propane has yet another beneficial practical effect. At the same condensation and evaporation temperatures, propane works at lower pressure ratios given its saturation pressure-temperature curve.

E.6 Conclusions

In this paper, the compressor model developed in [130] was tested with other compressors and refrigerants. The results obtained are in very good agreement with those obtained in the experiments, proving the capability of the model to estimate compressor performance. In addition, the values obtained for the model parameters are very reasonable and meaningful, providing a good characterization of the design and performance of the analyzed compressors.

In this study, certain anomalies in the performance of one of the compressors working with propane were detected, and a mechanical problem was indicated by the model as a possible reason for this outlying behavior.

The dead space ratio obtained was larger than expected. This seems to point to an additional loss not considered previously in the model which reduces the total mass flow rate that the compressor is able to pump. The possibility of a condensation effect inside the piston has been postulated as a reasonable explanation for this result. When this effect is considered in the model, the values obtained for the dead space ratio become quite close to those expected. Furthermore, this effect could be a relevant factor in the dependence of volumetric efficiency with superheat [131].

According to the model, the mechanical and the electric losses are the most significant ones for all the tested conditions; they represent approximately 75% of the total compressor efficiency losses and 55% of the total volumetric efficiency losses. At low pressure ratios (1.5-2.5), the pressure losses are noteworthy (more than 15% on compressor efficiency and more than 25% on volumetric efficiency). At high pressure (5-7) ratios, the leakages become a significant factor (more than 10% for compressor and volumetric efficiencies). The relative influence of heat transfer losses between the inlet and outlet remains constant at all pressure ratios and its overall influence on volumetric efficiency is significant (15%),therefore the relative importance assigned to this effect in [78] is justified.

The response of the model when used to predict compressor performance with a different refrigerant and the relative compressor performance differences between propane and R407C is outlined. Model parameters obtained from the direct fitting of propane data can accurately reproduce compressor efficiency for R407C. Concerning volumetric efficiency, the model underestimates the experimental results by approximately 3%. If the phase change parameter is released and adjusted to the experimental volumetric efficiency data, it becomes approximately 50% of the value obtained for propane, significantly improving volumetric efficiency predictions. Some not considered dependence in the refrigerant thermophysical properties of the heat transfer coefficient related with this process are thought to be a possible explanation for this result.

The compressor model described in [130] has demonstrated its capacity to describe with reasonable accuracy hermetic reciprocating compressor performance. As a consequence of its structure , in future, the capabilities of the model to describe other positive displacement compressor types such as Scroll compressors will be tested.

Figure Index

- Figure 13. Experimental compressor efficiency vs. pressure ratio for the ST compressor working with propane and R407C.
- Figure 14. Experimental volumetric efficiency vs. pressure ratio for the ST compressor working with propane and R407C.
- Figure 15. Measured compressor efficiency vs. calculated compressor efficiency for the ST compressor working with R407C.
- Figure 16. Measured volumetric efficiency vs. calculated volumetric efficiency for the ST compressor working with R407C.
- Figure 17. Calculated cylinder outlet temperature vs. measured compressor outlet temperature for the ST compressor working with R407C.

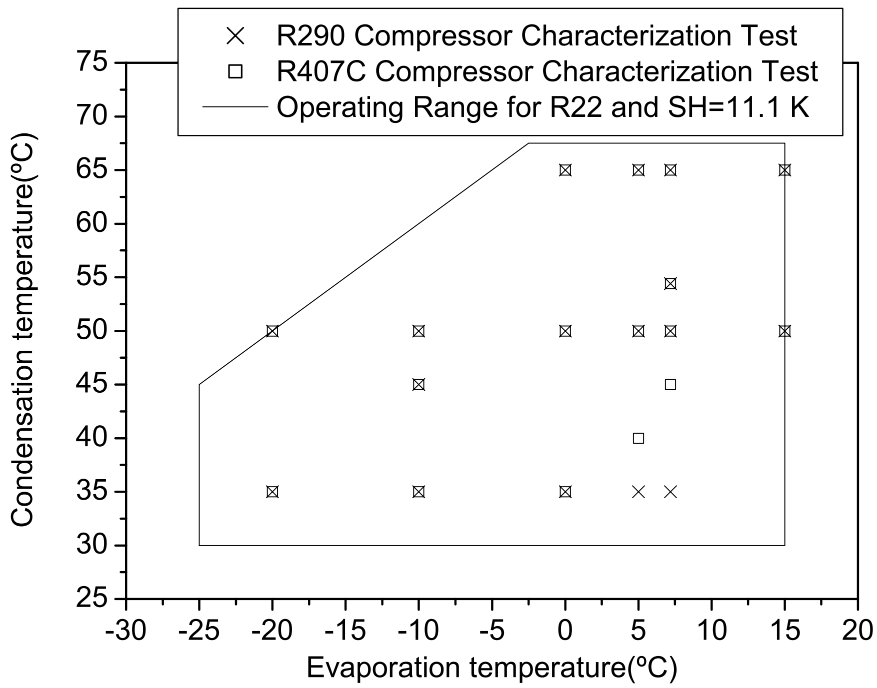
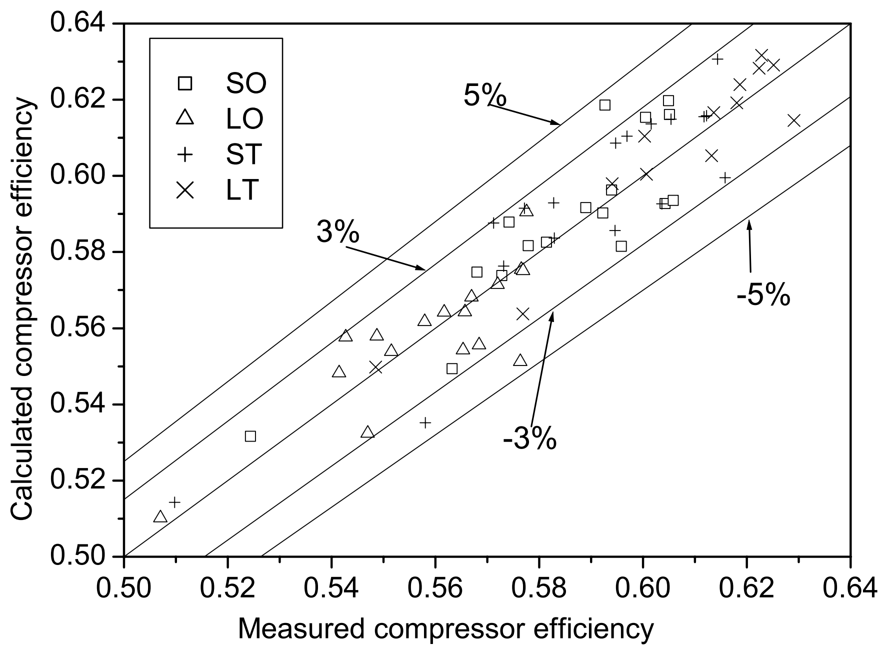
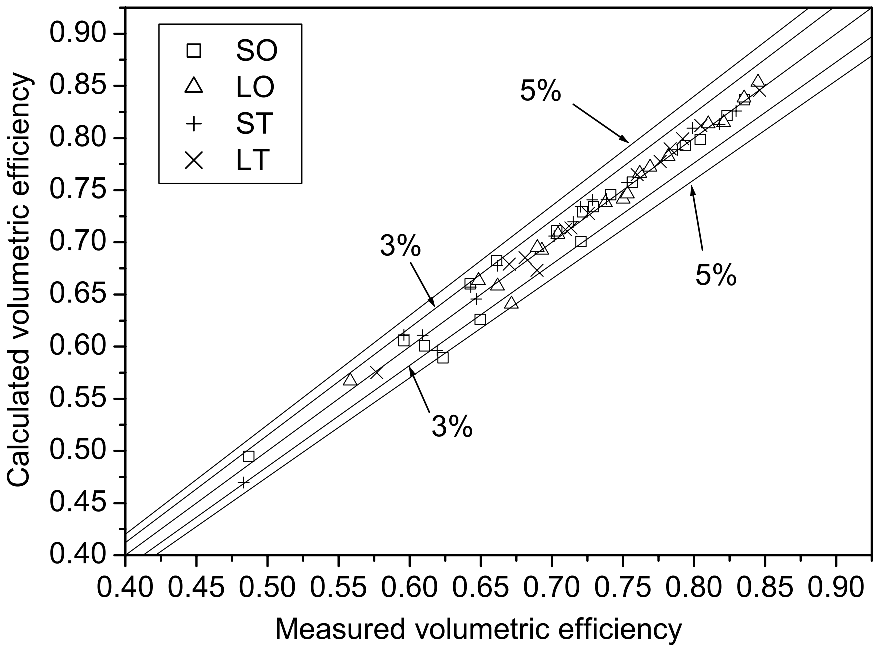

R290 Compressor Characterization Test
R407C Compressor Characterization Test
Operating Range for R22 and SH=11.1 K
75
70
65
60
55
50
45
40
35
30
25
Condensation temperature(ºC)
-30
-25
-20
-15
-10
-5
0
5
10
15
20
Evaporation temperature(ºC)

Figure E.1:



0.64
0.62
0.60
0.58
0.56
0.54
0.52
0.50
Calculated compressor efficiency
0.50
0.52
0.54
0.56
0.58
0.60
0.62
0.64
Measured compressor efficiency
SO
LO
ST
LT
5%
3%
-3%
-5%

Figure E.2:



SO
LO
ST
LT
5%
3%
5%
3%
0.90
0.85
0.80
0.75
0.70
0.65
0.60
0.55
0.50
0.45
0.40
Calculated volumetric efficiency
0.40 0.45 0.50 0.55 0.60 0.65 0.70 0.75 0.80 0.85 0.90
Measured volumetric efficiency

Figure E.3:

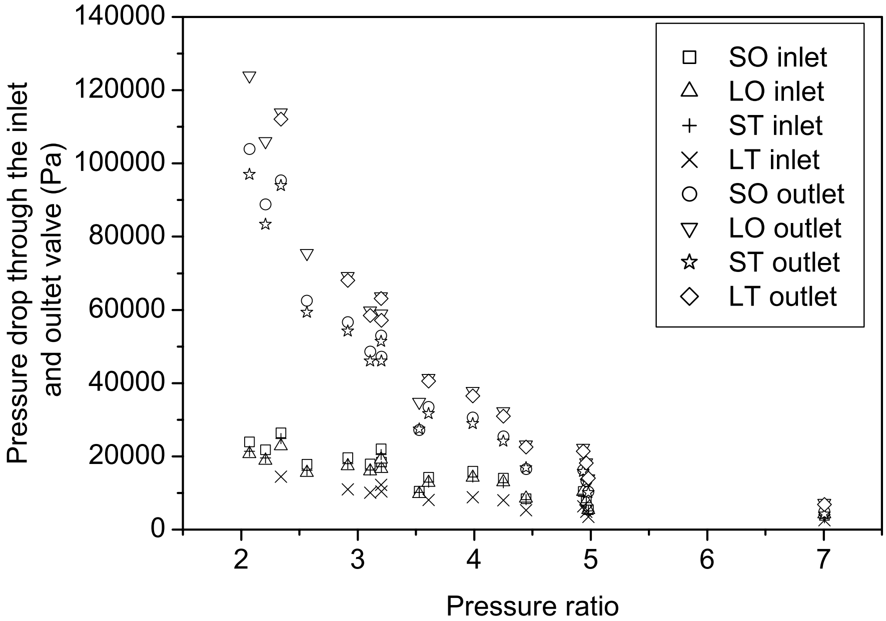

140000
120000
100000
80000
60000
40000
20000
0
2
3
4
5
6
7
Pressure drop through the inlet and oultet valve (Pa)
Pressure ratio
SO inlet
LO inlet
ST inlet
LT inlet
SO outlet
LO outlet
ST outlet
LT outlet

Figure E.4:

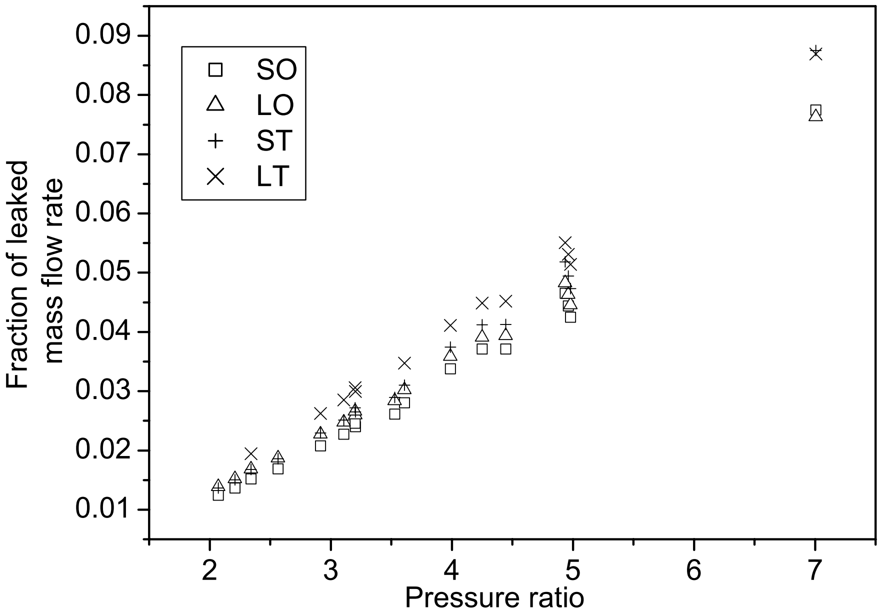

0.09
0.08
0.07
0.06
0.05
0.04
0.03
0.02
0.01
SO
LO
ST
LT
Fraction of leaked mass flow rate
2
3
4
5
6
7
Pressure ratio

Figure E.5:

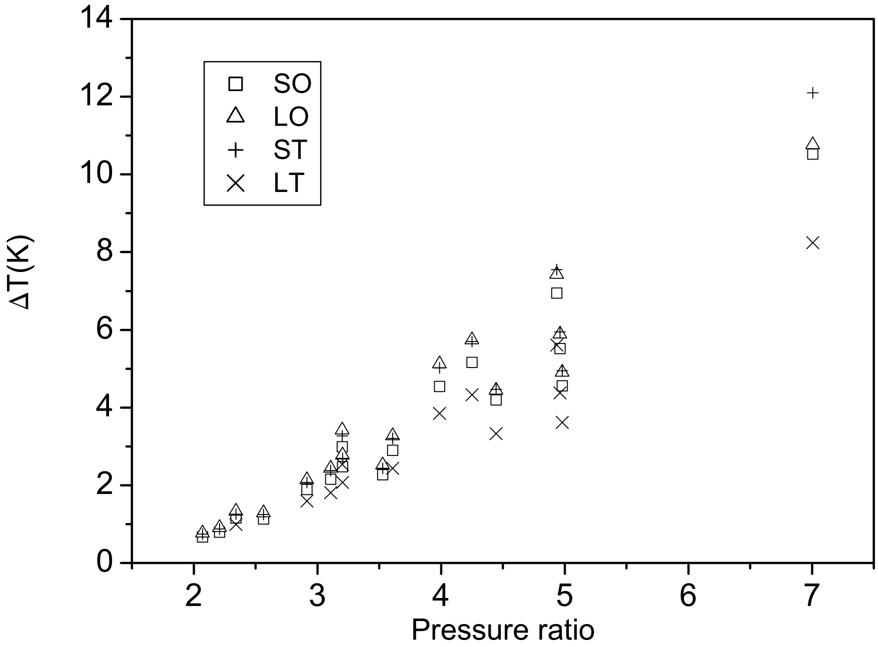


Figure E.6:

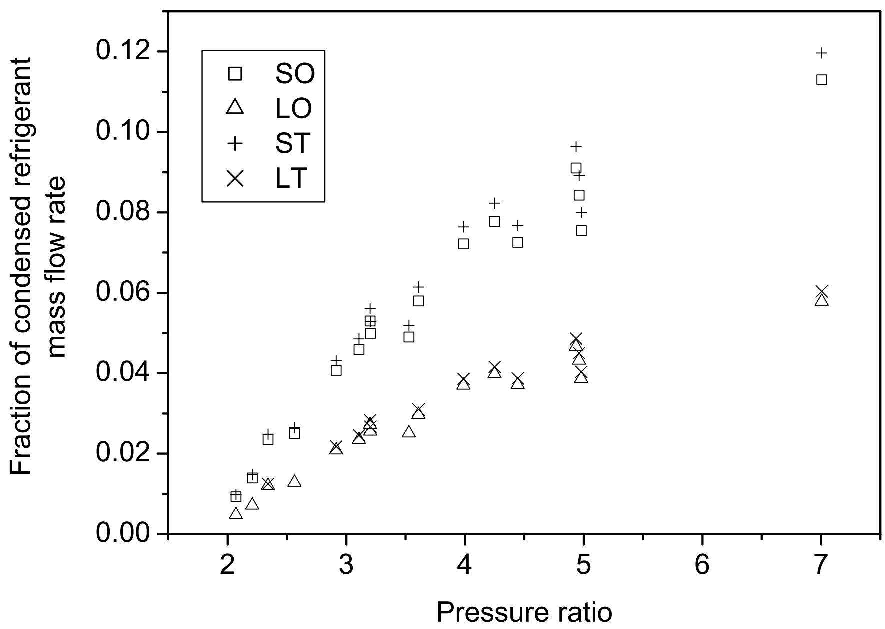

0.12
0.10
0.08
0.06
0.04
0.02
0.00
Fraction of condensed refrigerant
mass flow rate
2
3
4
5
6
7
Pressure ratio
SO
LO
ST
LT

Figure E.7:

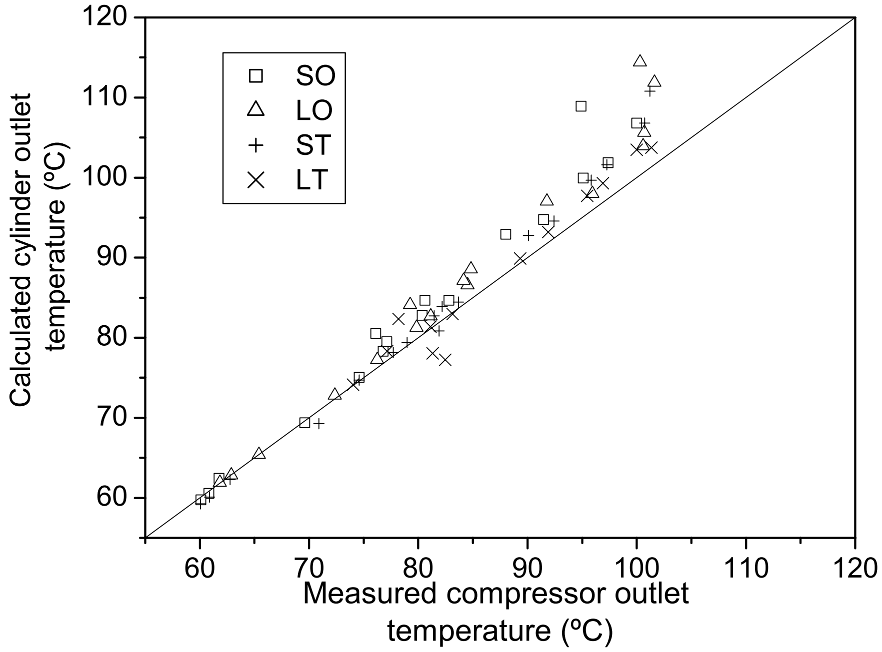
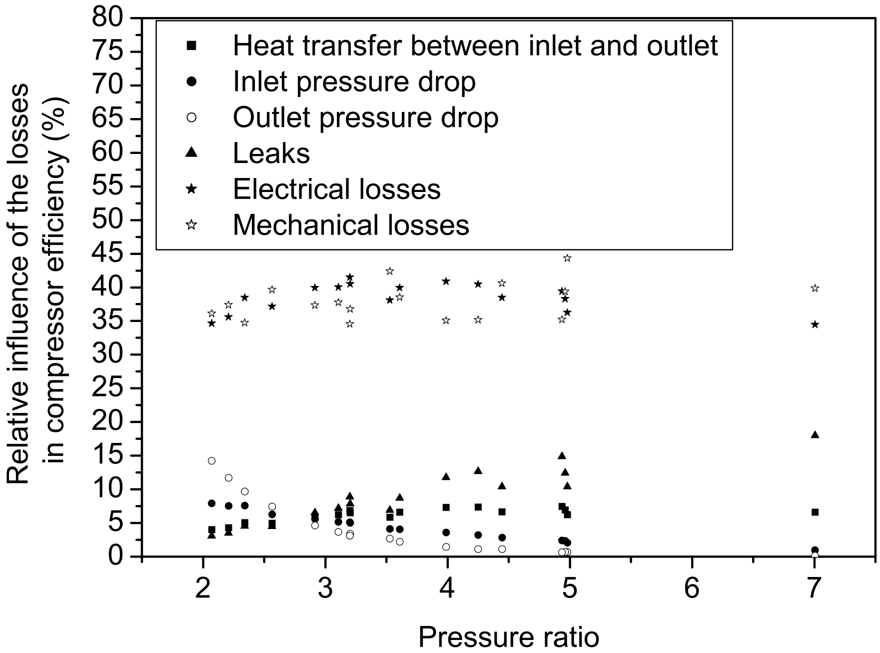

120
110
100
90
80
70
60
60
70
80
90
100
110
120
SO
LO
ST
LT
Calculated cylinder outlet temperature (ºC)
Measured compressor outlet temperature (ºC)

Figure E.8:



80
75
70
65
60
55
50
45
40
35
30
25
20
15
10
5
0
Relative influence of the losses in compressor efficiency (%)
Heat transfer between inlet and outlet
Inlet pressure drop
Outlet pressure drop
Leaks
Electrical losses
Mechanical losses
2
3
4
5
6
7
Pressure ratio

Figure E.9:

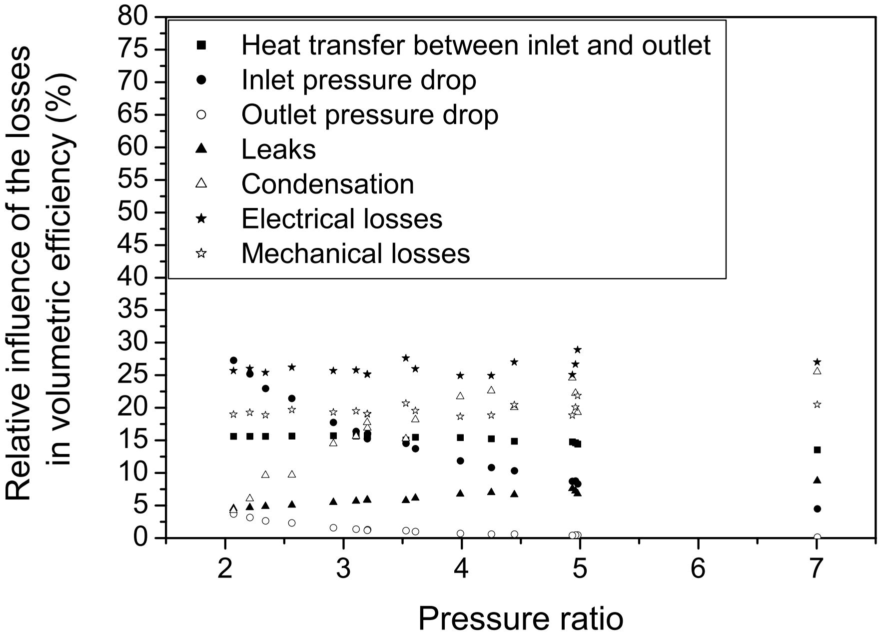

Relative influence of the losses in volumetric efficiency (%)
0
5
10
15
20
25
30
35
40
45
50
55
60
65
70
75
80
2
3
4
5
6
7
Pressure ratio
Heat transfer between inlet and outlet
Inlet pressure drop
Outlet pressure drop
Leaks
Condensation
Electrical losses
Mechanical losses

Figure E.10:

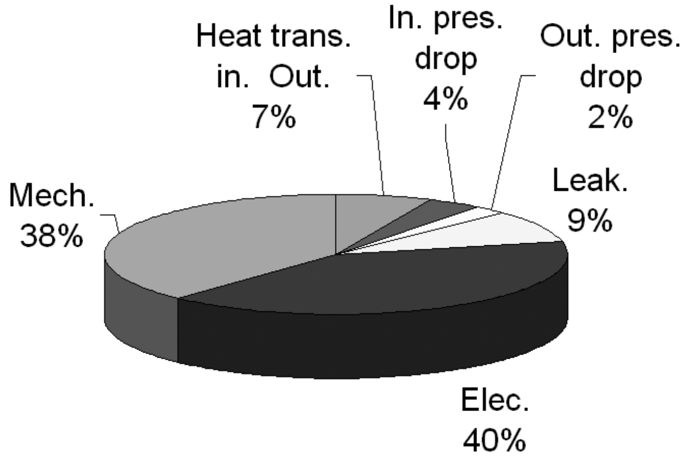

Heat trans.
in. Out.
7%
In. pres.
drop
4%
Out. pres.
drop
2%
Leak.
9%
Mech.
38%
Elec.
40%

Figure E.11:

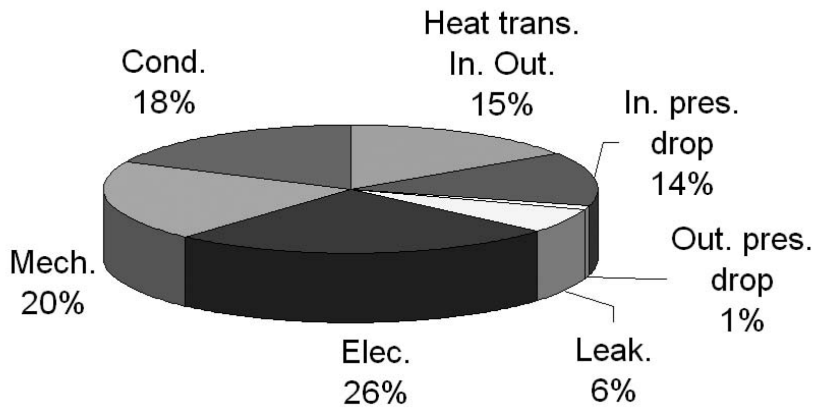

Heat trans.
In. Out.
15%
Cond.
18%
In. pres.
drop
14%
Out. pres.
drop
1%
Mech.
20%
Elec.
26%
Leak.
6%

Figure E.12:

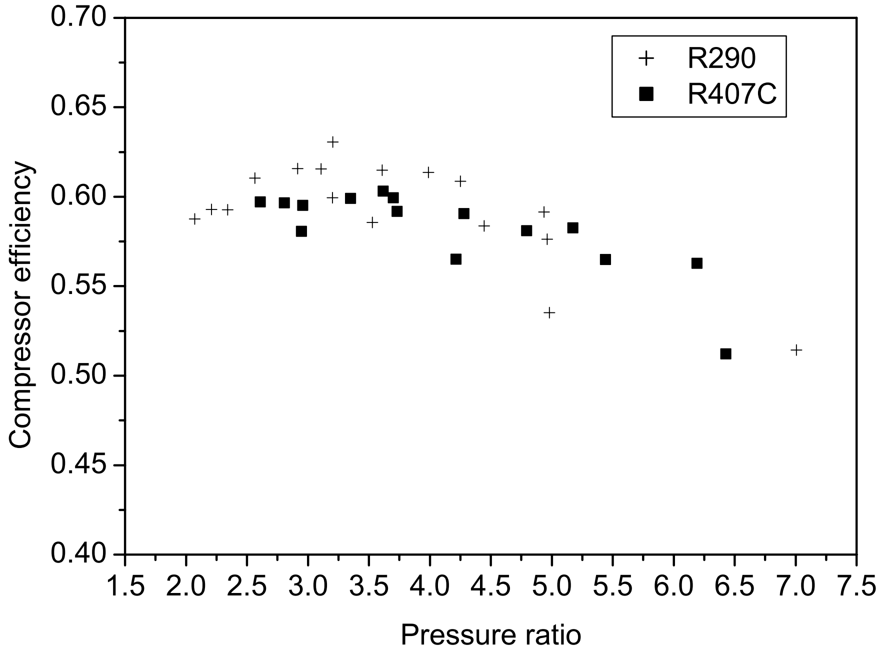
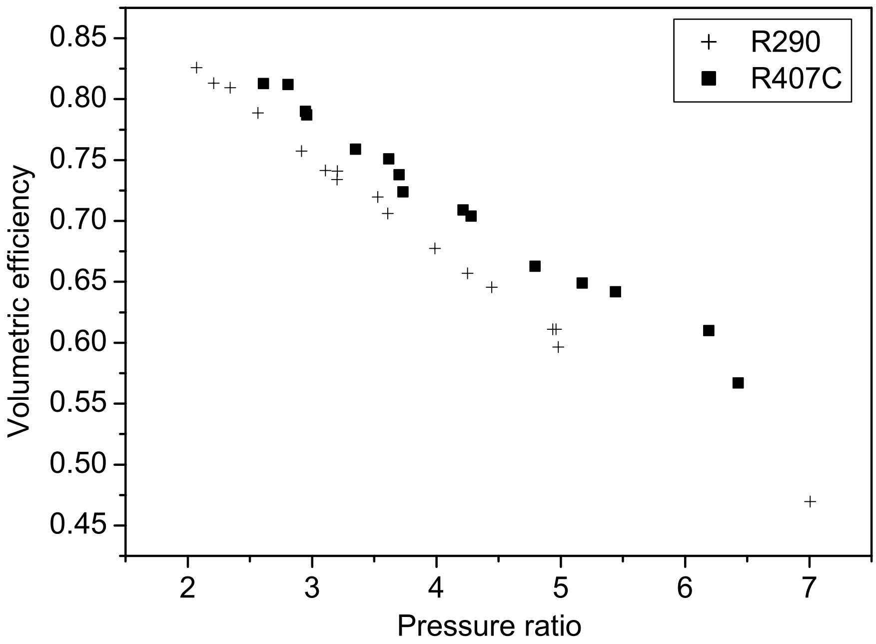

0.70
0.65
0.60
0.55
0.50
0.45
0.40
Compressor efficiency
1.5 2.0 2.5 3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5
Pressure ratio
R290
R407C

Figure E.13:



R290
R407C
Volumetric efficiency
0.85
0.80
0.75
0.70
0.65
0.60
0.55
0.50
0.45
2
3
4
5
6
7
Pressure ratio

Figure E.14:

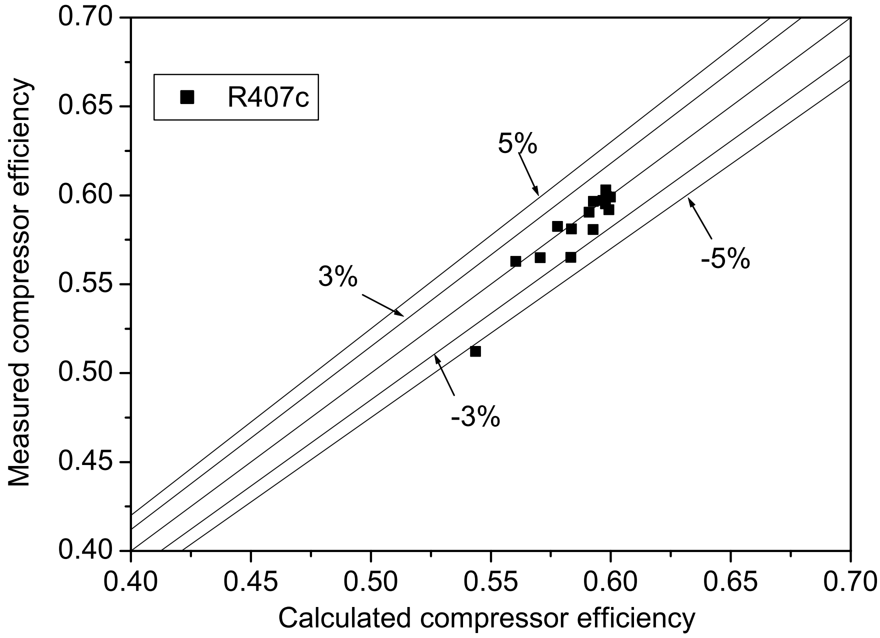

0.70
0.65
0.60
0.55
0.50
0.45
0.40
0.40
0.45
0.50
0.55
0.60
0.65
0.70
R407c
5%
3%
-5%
-3%
Measured compressor efficiency
Calculated compressor efficiency

Figure E.15:

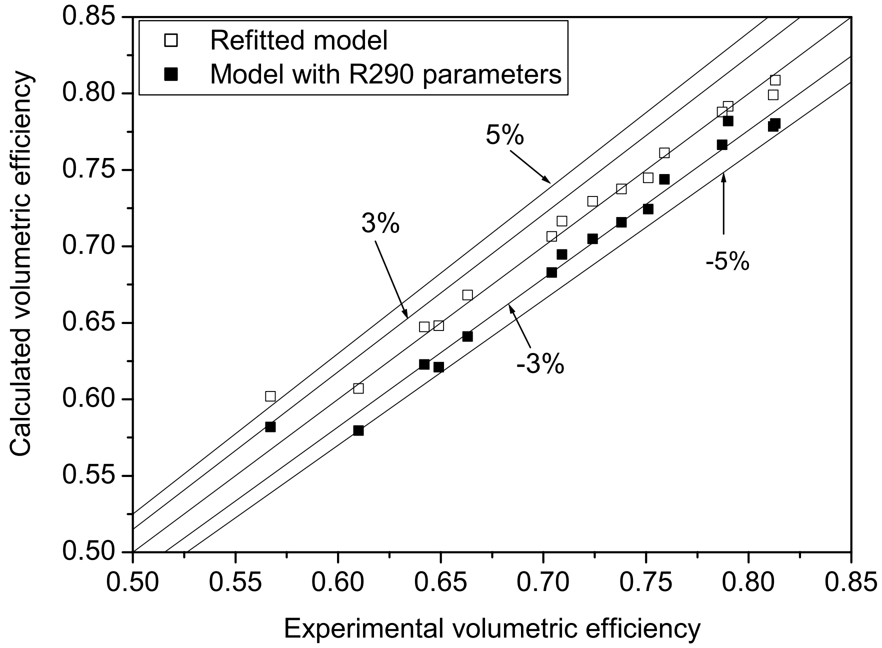

Refitted model
Model with R290 parameters
5%
3%
-3%
-5%
Calculated volumetric efficiency
Experimental volumetric efficiency
0.50
0.55
0.60
0.65
0.70
0.75
0.80
0.85

Figure E.16:

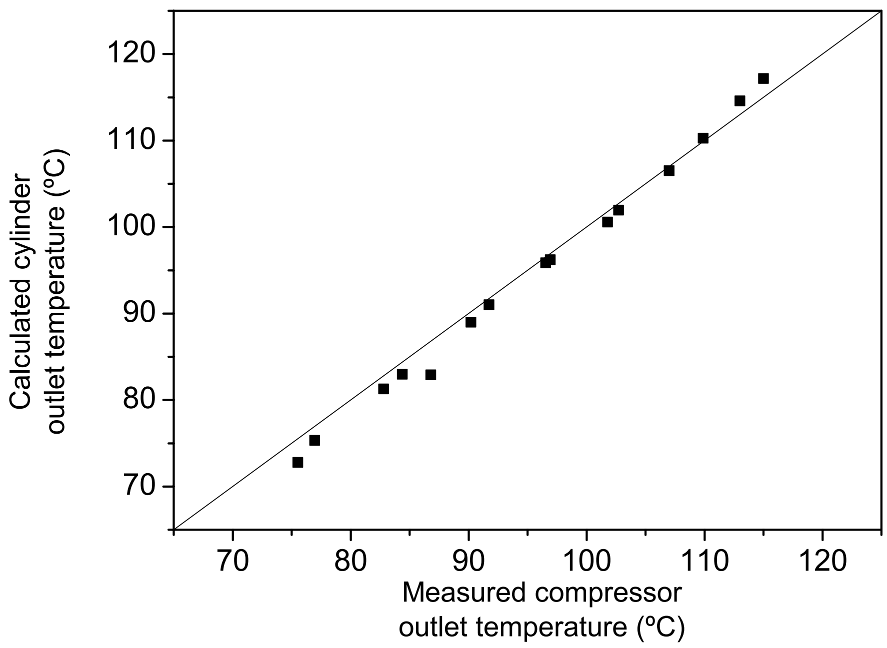

120
110
100
90
80
70
Calculated cylinder
outlet temperature (ºC)
70
80
90
100
110
120
Measured compressor
outlet temperature (ºC)

Figure E.17:

Stroke (mm)	Pistons	Dead SpaceRatio	Compressor name
30.23	1	0.037	SO
38.10	1	0.029	LO
30.23	2	0.037	ST
38.10	2	0.029	LT

Table E.1: Tested Compressors.

	SO	LO	ST	LT
η_k	0.954	0.943	0.932	0.968
η_s	0.988	0.991	0.996	0.993

Table E.2: Correlation factor obtained for the compressor and volumetric efficiency for all the studied compressors.

	SO	LO	ST	LT
K_1	0.9	0.9	0.9	0.9
$K_2^{'}$	3.02	2.88	2.80	2.78
K_3	$2.08 \cdot 10^7$	$1.09 \cdot 10^7$	$1.94 \cdot 10^7$	$9.76 \cdot 10^6$
K_4	$3.7 \cdot 10^8$	$2.39 \cdot 10^8$	$3.85 \cdot 10^8$	$2.54 \cdot 10^8$
$K_5^{'} \cdot D^2$	$0.96 \cdot 10^{-6}$	$0.89 \cdot 10^{-6}$	$0.95 \cdot 10^{-6}$	$0.86 \cdot 10^{-6}$
V_d/V_s	$7.12 \cdot 10^{-2}$	$5.71 \cdot 10^{-2}$	$6.77 \cdot 10^{-2}$	$5.32 \cdot 10^{-2}$
$(V_d/V_s)^{'}$	$4.3 \cdot 10^{-2}$	$3.5 \cdot 10^{-2}$	$3.9 \cdot 10^{-2}$	$3.4 \cdot 10^{-2}$
K_6	1.70	1.65	1.73	1.63
K_7	$6.08 \cdot 10^{-2}$	$13.05 \cdot 10^{-2}$	$5.11 \cdot 10^{-2}$	$5.19 \cdot 10^{-2}$
$K_8(kW)$	0.113	0.148	0.2052	0.483
Electrical motor eff.	0.88	0.89	0.86	0.90

Table E.3: Parameter values for the compressors under study.