Experimental Studies Of Hermetic Reciprocating Compressors With Special Emphasis On pV Diagram

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ABSTRACT

A detailed experimental investigation of the thermal and fluid dynamic behaviour of hermetic reciprocating compressors has been carried out. The absolute transient pressure evolutions and the temperature maps along different strategical points inside an hermetic reciprocating compressor have been experimentally obtained. The present experimental study allows to determine pV diagram without any previous hypothesis to determine absolute pressure level. This work is focussed not only to generate an important and useful experimental data to know compression work, energy losses through valves, gas pulsation, etc., but also to validate experimentally in detail the numerical simulation results of hermetic reciprocating compressors performance. Numerical results have been obtained by means of numerical simulation models developed to be used in the whole compressor domain.

1 INTRODUCTION

Compressor performance design is defined by different parameters: mass flow rate or volumetric efficiency, compression work or isentropic efficiency, mechanical, electrical and heat losses, COP, etc. To have a better understanding of the complex physical processes involved in this kind of elements it is necessary not only to know these global values, but also its detachment and specific evolution under a wide range of different working conditions.

One of these meaningful parameters that defines compressor behaviour is compression work and its non-dimensional ratio with isentropic or ideal work, i.e. isentropic efficiency. Together with this, both works can be split into four different compression steps: compression, discharge, expansion and suction, and its different contributions to isentropic efficiency.

Several authors present different experimental investigations on pV diagrams [1-3], where the main problem was to not have absolute pressure reference in compression chamber. The main objective of this work is to present an experimental study of pV diagram, with special emphasis on absolute compression chamber pressure evaluation. Thus, the area enclosed by pV diagram is the compression work per cycle, and its detachment shows compression, discharge, expansion and suction works detachment. Therefore, comparison between both real and ideal pV diagram allows to know isentropic efficiency, isentropic detachment and energy losses through valve processes. Then, all this information is a useful tool for prototypes design and compressor performance optimization.

A complete and advanced numerical simulation model of the thermal and fluid dynamic performance of hermetic reciprocating compressors has been developed and is used in the optimization of commercial compressor design [4-6]. During the last few years several global experimental validations and detailed thermal experimental comparisons have been carried out.

The second objective of this work is to present a detailed experimental validation of the numerical simulation model by means of the experimental results obtained in transient pressure evolutions, temperature maps and pV diagrams.

2 EXPERIMENTAL SET-UP

A commercial hermetic reciprocating compressor has been encased in a bigger special shell, in order to keep compressor crankcase and encoder together (see Fig. 1).

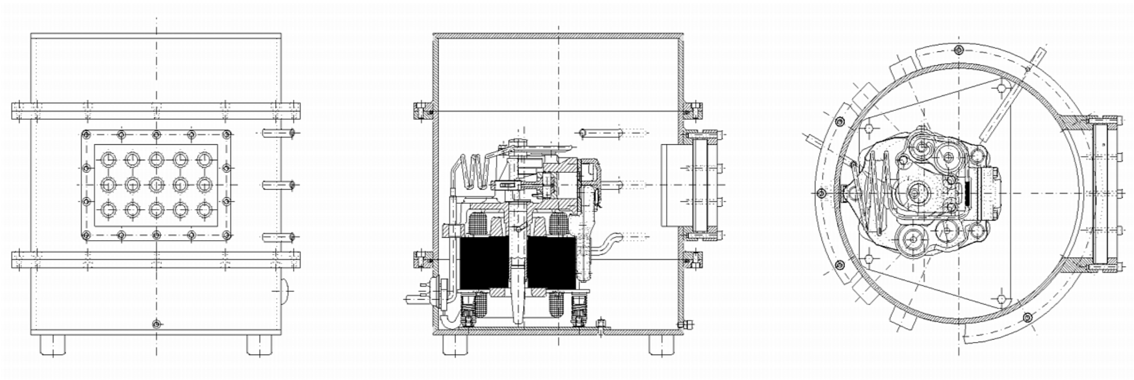

Fig. 1: Hermetic reciprocating compressor prototype scheme: front, side and top views respectively.

Different K type thermocouples have been put inside compressor to measure fluid temperature distribution along specific points: compressor shell, inlet muffler, discharge tube, wall temperatures, etc. Three special pressure transducers have been introduced to measure: suction muffler, compression chamber and cylinder head absolute fluid pressure. An incremental encoder has been connected to crankshaft connecting rod mechanism to evaluate instantaneous piston position. Fig. 2 shows compressor unit. A specific Dynamic Analyzer Signal DAC NI PXI 4472 unit has been employed to measure fast variables like pressures and encoder position. A more general purpose Data Acquisition and Control System DACS unit to show variables like single stage vapour compression refrigerating experimental unit temperatures, mass flow rate, etc. is used. This DACS unit also allows to control pressure ratio work unit and secondary fluids control. Finally, a specific Data Acquisition Switch Unit DASU hp 34970A is used to measure temperatures inside compressor. Fig. 3 shows compressor unit with DAS and DASU systems together.

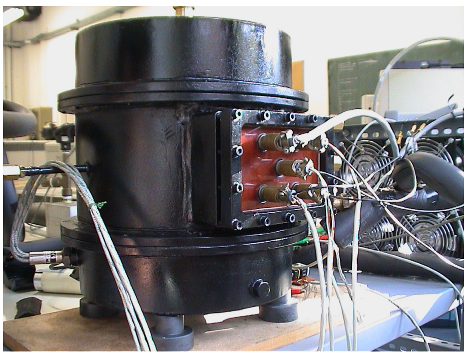

Fig. 2: General instrumented compressor view.

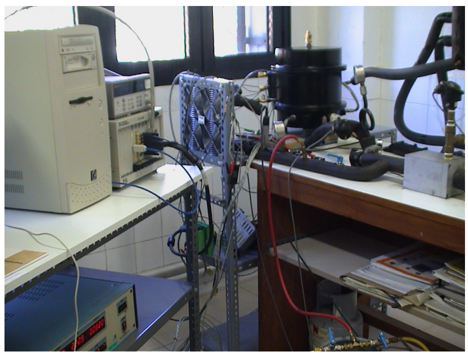

Fig. 3: Compressor view with acquisition units.

2.1 Hermetic reciprocating compressor instrumentation

The instrumentation has been carried out under a commercial hermetic reciprocating compressor. A few changes have been necessary to instrument the compressor. A special piece has been built to connect encoder with connecting rod. Specific holes through cylinder head to connect pressure transducers and several holes through the shell for transducers connections together with temperature groups feedthroughs have also been employed.

A relative quadrature encoder of 720 pulses/revolution with a maximum frequency output of 100kHz has been used to measure instantaneous piston position at each half crank angle degree. From the electrical power consumption, the relative encoder adds losses around 6% of measured power consumption, for the studied cases. Figs. 4 and 5 show general view encoder position and pressure transducers distribution.

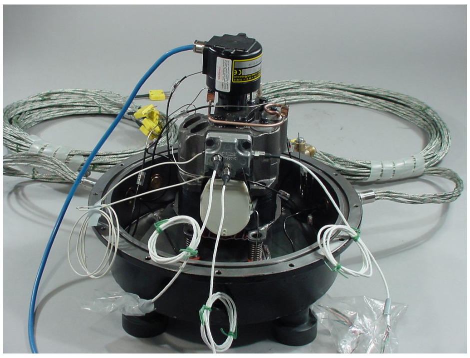

Fig. 4: Instrumented compressor general view.

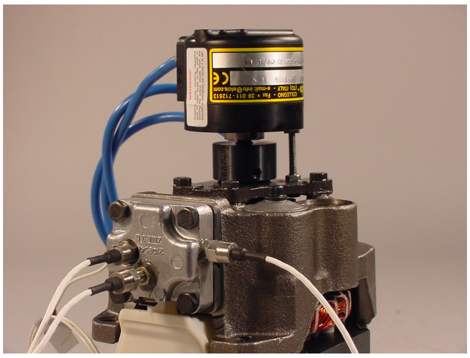

Fig. 5: Detailed crankcase instrumentation.

Three piezo-resistive miniature absolute pressure transducers have been introduced to measure the following points: suction pressure at the end of suction muffler, compression chamber pressures under valve plate and discharge pressure inside cylinder head. Figs 6 and 7 shows transducers position in detail.

The transducers used are high sensitive, with high natural frequency and thermal compensated between 80 and 160^{o}C. The range is from 0 to 10 bar, although it can works with an overpressure of 2 times F.S. without losing linearity. Then, it has an accuracy of 0.1% F.S. combining non-linearity and hysteresis.

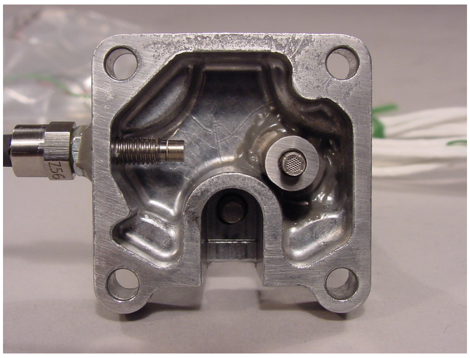

Fig. 6: Pressure transducers inside cylinder head.

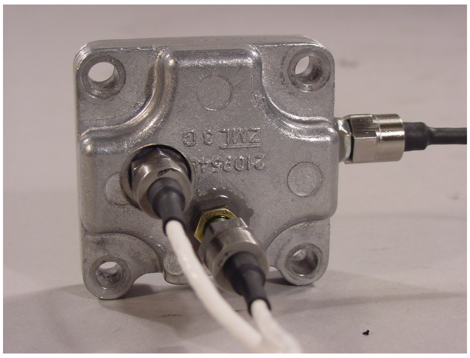

Fig. 7: Pressure transducers outside cylinder head.

Calibration and linearity has been certified against precision pressure controller which has an accuracy of 0.003% F.S. and stability of 0.01% F.S. This second calibration has verified that pressure measurements error is lower $\pm$ 10 mbar from 0 to 20 bars.

Ten internal K type thermocouples have been built in two thermocouple groups with five thermocouples in each group. Thermocouple groups pass through the shell with vacuum pressure feedthroughs. Thermocouples inside the compressor shell are encapsulated with steel, and electrically insulated with MgO between two wires and the steel thin stick. A third pin connects the steel capsulate inside the shell with a steel grid outside the shell and it is connected to the ground.

Thermocouples have been calibrated by means of a thermostatic refrigerating unit working at different temperatures and using mineral oil. A precision Platinum Resistance Thermometer is used as the reference value. Both thermometers have been put together. PTR is read with an individual data acquisition model. The PTR system accuracy is $\pm$ 0.025^{o}C. A Data Acquisition Switch Unit DASU used guarantees an accuracy of $\pm$ 0.2^{o}C. Thermocouples K type have an accuracy after calibration of $\pm$ 0.08^{o}C. Thus, total accuracy temperature measured is less than $\pm$ 0.3^{o}C.

2.2 Experimental unit description

An experimental unit is used to study single stage vapour compression refrigerating systems. This unit has been designed to validate mathematical models of the thermal and fluid dynamic behaviour of single stage vapour compression refrigerating units in general, and specially for condensers, evaporators and hermetic reciprocating compressors.

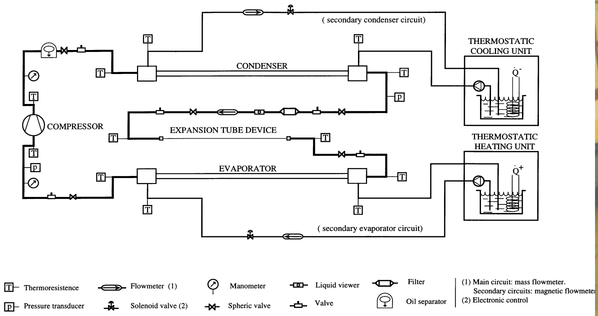

Fig. 8: Experimental unit scheme.

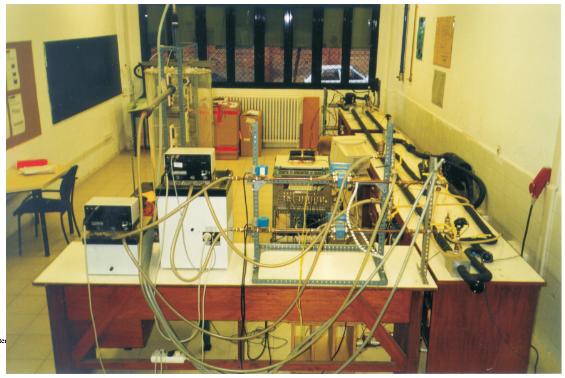

Fig. 9: Experimental unit general view.

Elements that make up the experimental unit are basically: hermetic reciprocating compressor, double-pipe condenser and evaporator, expansion devices and different tube connections. Auxiliary fluid for condenser and evaporator is water. The fluid flow temperatures inside tubes and annuli are measured with calibrated platinum resistance thermometer sensors Pt-100. These sensors are located at the inlet and outlet sections of each element of the main circuit and secondary circuits. Condenser and evaporator pressures are measured by transducers, the accuracy is within $\pm 0.1\%$. Mass flow rate inside main circuit is measured with a Coriolis type mass flow-meter, the accuracy is within $\pm 0.2\%$. Volumetric flow in secondary circuits is measured with magnetic flow-meters, the accuracy is from $\pm 0.1\%$ to $\pm 0.5\%$. Temperature in auxiliary circuits is controlled by thermostatic units and two modulating solenoid valves control mass flow rate in these circuits. The Data Acquisition and Control Unit hp E1300A, and a personal computer, process all compressor and experimental unit information.

3 RESULTS

Based on the experimental set-up described above a first group of illustrative results have been obtained. The experimental data facilitates instantaneous pressure evolution along suction muffler, compression chamber and cylinder head vs. half crank angle degree. The temperature map allows to evaluate the direct gas mass fraction from inlet compressor to suction muffler. Finally, instantaneous piston position together with compression chamber pressure allows to know experimental pV diagram.

Both pressure distributions and pV diagrams have been compared numerically by means the complete numerical simulation model also referenced above. The comparison between experimental and numerical pV diagrams with ideal pV diagram facilitate differences between numerical and experimental isentropic efficiency, suction and discharge energy losses.

3.1 Global comparative results

The results obtained correspond to the hermetic reciprocating compressor instrumented with a 7.5 cm ^{3} cylinder capacity, working with R134a and a nominal frequency of 50 Hz. The following single stage vapour compression cycle under periodical conditions has been tested: both inlet compressor temperature and ambient temperature of 25 ^{o}C, condensation temperature of 55 ^{o}C and evaporation temperature of -23.3 ^{o}C.

The numerical results have been obtained by means of numerical simulation model running in a PC cluster with 48 AMD-K7 processors working at 900 MHz with a RAM memory of 512MB. Software code language is FORTRAN 95 and all variables type are double precision.

Accuracy required for considered converged results have been 10 $^{-6}$, while temporal mesh has been 720 time steps per cycle. All input data information is: inlet compressor temperature, inlet and outlet shell pressure, compressor geometry, valve dynamic modal analysis and electrical motor curve. No other data is necessary, the rest of the information is output data. However, direct gas mass fraction from inlet compressor to suction muffler has been experimentally approximated to 0.2% considering experimental temperature map obtained [7].

Table 1 shows global comparative results between experimental data and numerical simulation. Differences are typically lower than 10% [8] in mass flow-rate, volumetric efficiency, COP, etc.

Table 1: Comparative results between numerical simulation and experimental data.

	$\dot{m}$ (kg/h)	η_v $(\%)$	T_{out} (^oC)	ω_{real} (Hz)	$\dot{W}_e$ (W)	$\dot{Q}_{evap}$ (W)	COP
experimental	3.52	57.0	89.7	49.22	150.8	185.7	1.232
numerical	3.81	60.4	84.9	49.24	147.8	202.1	1.367

3.2 Pressure maps comparison

Both numerical results and experimental data allow to compare absolute instantaneous pressure in different strategic points under periodical conditions. Fig. 10 shows comparative pressure maps in the three following points: outlet suction muffler, compression chamber and cylinder head.

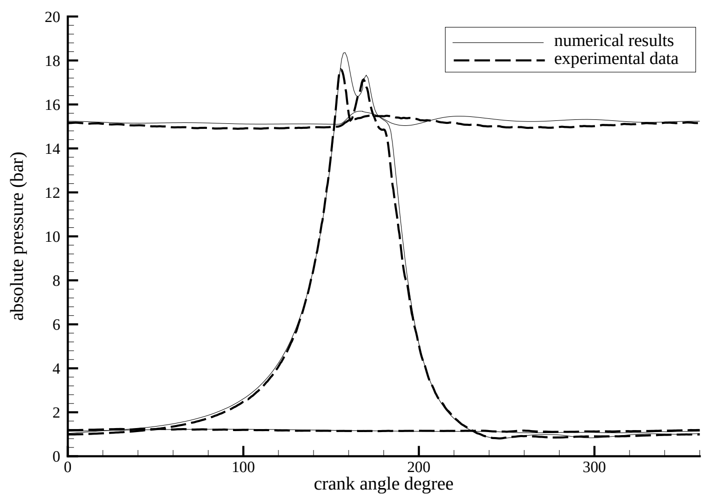

Fig. 10: Comparative pressure results at the: suction, compression and discharge chambers.

Comparative pressure results obtained between numerical solution and experimental data show that differences are: between 1 - 2% on discharge pressure, with a maximum value of approximately 2.2% in crank angle positions 190 degrees and 230 degrees; between 2 - 4% on suction pressure only with a maximum value of 7% around crank angle position of 260 degrees; about 8% on compression chamber pressure, except crank angle positions between 0 and 150 degrees with differences around 10%, and between 180 and 200 degrees with differences of approximately 12%.

In general, pressure comparative results show same differences as global comparative results presented in previous section. Although, these are preliminary results for the case studied, it is interesting to remark some relevant aspects. All comparative pressure results during compression and expansion processes present a remarkable agreement. In general, there is a slight shift between numerical results and experimental data, even though both results depend on instantaneous piston position, which is taken into account. Differences appear basically when discharge valve is opened or during suction process. These differences are essentially due to effective flow area and modal analysis of the reed valve used in the numerical model.

3.3 pV diagram results

Fig. 11 shows both numerical results and experimental data on pressure-volume diagram for the case studied. With these values it is possible to compare numerical and experimental compression work, or indicated work, together with isentropic efficiency and its detachment.

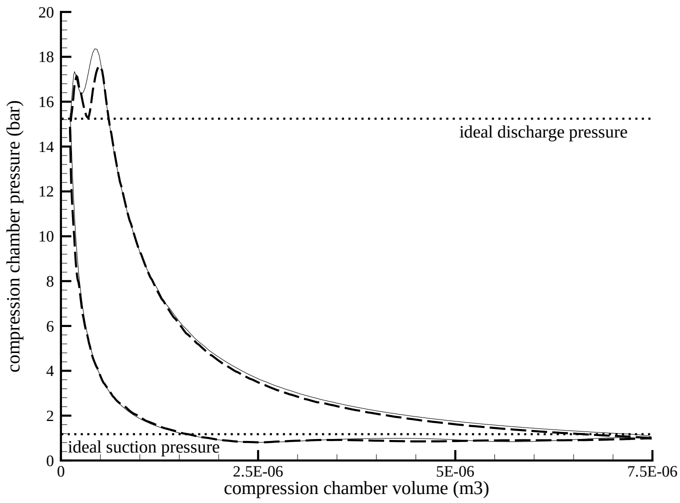

Fig. 11: Comparative pressure volume diagram.

Table 2 shows compression work per mass unit, and its detachment for each of the four work processes: compression, discharge, expansion and suction. Even though the experimental work per cycle is lower than the numerical one, it should be kept in mind that experimental mass flow rate is lower than the numerical one.

Table 2: Comparative results of indicated work per mass unit and its detachment.

	w_{cp} (kJ/kg)	w_c (kJ/kg)	w_d (kJ/kg)	w_e (kJ/kg)	w_s (kJ/kg)
experimental	100.0	104.71	40.87	-23.24	-22.30
numerical	98.26	105.83	40.07	-23.26	-24.38

Based on isentropic efficiency detachment explained in detail in ref. [9], Table 3 shows isentropic efficiency for numerical results and experimental data, together with its different contributions: compression, discharge, ç expansion and suction efficiencies. Due to the lack of experimental information of opening and closing valve time, these efficiencies have been estimated considering the areas below and above the suction and discharge ideal pressures.

Table 3: Comparative results of isentropic efficiency and its detachment.

	η_{iso} (%)	η_{wc} (%)	η_{wd} (%)	η_{we} (%)	η_{ws} (%)
experimental	70.9	76.1	90.5	111.6	97.4
numerical	72.2	74.9	91.0	111.8	99.4

The last two Tables show that differences between numerical results and experimental data on different work per mass unit detachment are approximately 2%, except suction work where differences are about 8%.

Isentropic efficiency and all its detachment have differences lower than 2%, which really shows reasonable good agreement.

Finally, it is interesting to highlught that discharge work efficiency and suction work efficiency are approximately 90% and 98.5% for the case studied, while expansion work efficiency has a value higher than 100%.

4 CONCLUSIONS

A detail experimental study of hermetically sealed reciprocating compressor has been performed. Absolute instantaneous pressures have been obtained for the compression chamber together with suction and discharge chambers. The methodology allows to directly measure pressure level for pV diagrams. A comparison of the experimental data and the numerical results obtained using an advanced detailed code have been presented. Even though the agreement is remarkable, more effort should be made in improving some parts of the model. Specifically some aspects related to effective flow area estimation.

5 ACKNOWLEDGEMENTS

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