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High Efficient CO2 trans-critical Reciprocating Compressor
Part I : Valve plate and cylinder head 1 D flow optimization

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ABSTRACT

In this article we will describe a new high efficient CO2 trans-critical compressor architecture where valves technology choices will be detailed.
A first compressor efficiency optimization step was done based on the following methodology:
- Experimental PV diagram analysis
- Valve plate design tool (lift, spring, ports optimization)
The article will be completed by experimental validation of this initial step.

INTRODUCTION

As explained by ASERCOM statement about Carbon Dioxide in RAC Systems, CO2 was one of the first refrigerants to replace early air cycle systems and was in use primarily for shipboard refrigeration in the beginning of the twentieth century. It was then superseded by chlorofluorocarbons. However, since CO2 is environmentally benign, non-toxic (in the classical sense), non-flammable, chemically inactive and offers a very high volumetric cooling capacity together with excellent heat transfer properties, it is increasingly considered for use today in RAC systems. Because of its very low global warming potential (GWP=1) and zero ODP, CO2 systems do not need the very stringent containment criteria necessary for HFCs and other refrigerants. Since CO2 is in the same safety class (L1) as HFCs the safety requirements may be less onerous than they would be for ammonia or hydrocarbons. Thermodynamic characteristics of CO2 are much different to refrigerants usually applied in RAC system. This is mainly related to the very low critical temperature of 31°C which, depending on the heat sink temperature on the discharge side, may require so-called trans-critical operation. The system design for trans-critical operation will differ from a conventional vapor compression cycle.
The energy efficiency tends to be lower as compared to a sub-critical conventional system. Nevertheless, in several applications or in specific circumstances CO2 systems can reach or exceed the energy efficiency of systems with established refrigerants. In any case, a high efficient CO2 compressor is required to be competitive versus HFC solutions in terms of TEWI.

A new semi-hermetic compressor has been developed with CO2 for refrigeration and heat-pump applications. The chosen cycle is a transcritical one, already developed and discussed elsewhere [1], with an upper high pressure of 140 bars, in one-stage and two-stage versions (figure 1).

1. COMPRESSOR DESIGN

The main characteristics of the compressor have been determined by calculating compression process, resulting bearings loads, leakage rate. The chosen design is a vertical motor/shaft position, with two opposite flat cylinders
and a direct suction for refrigeration application, while a motor gas-cooling option is possible for heat-pump application (figure 2). The target mass flow at design point is 850 kg/h and design speed of 2900 rpm.

Figure 1: vertical cut view of the compressor   Figure 2: Transcritical CO2 map

The main key-design points of such CO2 compressor are reliability issue, due to high pressure differential, and efficiency issue, due to high pressure and high gas density. We will develop here the second point.

1.1 Efficiency key point: the valve design
The care for optimizing efficiency of the compressor is piloted by the specific characteristics of the CO2 as refrigerant compared to traditional ones. Because of CO2 higher vapor phase density, parts size (cylinders, valves, tubing, etc) are 80 % reduced, compared to R404A for the same cooling capacity. Because of its thermo dynamical properties, superheating greatly affects the efficiency and the compressor pressure differential is 5 times greater. These 3 items, compactness, thermal aspect, pressure impact, deeply involve the valve plate design for the global efficiency of the compressor, regards to three main requirements:

- the valve plate should offer reduced pressure drops.
- the valve plate shall minimize thermal exchanges between suction a and discharge flows.
- the valve plate should allow a speed of 3000 rpm.

The valve design has been made in three main steps.

1.2 Valve design: refrigerant flow mean value
First step is to estimate the mean values of the refrigerant flow. The thermo-dynamical data (fluid properties, application points) and the main design data (assumed dead volume, cycle timing & rough valve timings) allow to built the theoretical cycle as shown in the Pressure-Volume diagram (Figure 4).
The main outputs of this chart are the suction and discharge flows, average and peak. For an application point of 35/95 bars SH10 K, the average value is 800 kg/h, 10 m³/h at suction and 4.4 m³/h at discharge.

### 1.4 Valve design: technology choice

Second step is to choose the valve technology, refer to these values. Annular valve was preferred to reed valve for 3 reasons: geometry, sizing, and speediness ability.

- **Geometry**: The gas flow is perfectly symmetrical, avoiding spotted flows created by reed valves. The suction and discharge are concentric and naturally well-sized according to their volumetric flow ratio.
- **Sizing**: The flow path is maximized. There is no lost surface by the strips, and the section surface / lift ratio is higher. One can compare two classical suction configurations for a 50 mm diameter piston, one with 3 suction reed valves 10 mm diameter and another with an annular suction valve of max diameter 50 mm (Figure 3). In that case the section surface gain is 50% in the whole lift range.
- **Speed**: Its limitation is higher than for reed valve, 3000 rpm instead of 1500, and even up to 7000 rpm with a quite similar design. The inconvenience of annular valve is that heat exchange is more difficult to eliminate in
concentric flows, requesting an optimized cylinder-head. The global geometry of the valve plate is thus determined with our supplier. It consisted in a valve plate including two annular concentric valves, their springs and stoppers, large suction paths with a common circular recess, discharge holes with a discharge recess.

### 1.5 Valve design: geometry design

Third step was to set the geometry factors with a strong coupled code developed by our valve plate supplier. The lifts, valves, springs, port geometries should be carefully designed to maximize the inlet and discharge flow, with regard to mass flow, application range and efficiency (figure 5). The port and gap area, coupled with the maximum lift, determine the flow velocity and thus the pressure force on the valve. The valve movement, limited by its stoppers and springs, implies back-force on the valve. The combination is done by a cycled loop that determines the real valve dynamics (figure 6). At output, valve oscillations, time pressure allows to predict the pressure-volume diagram and then the isentropic efficiency of the compression process.

The balance was to be done between maximizing the flow path and reducing the dead volume. Large flow path limits the pressure drops and thus the pressure losses. A low dead volume decreases the cycle losses and increases the compressor capability at high compression ratio.

A first design was done with 4 large inlet paths and 8 discharges. A prototype has been tested; the global isentropic efficiency was measured and need some improvement. The compression prediction is shown on figure 6, on the Pressure-Volume diagram of the cylinder compression cycle.

![Figure 5: Chart of strong coupled valve–design code](image)
1.6 Valve design: discussion

We can notify on that diagram the different powers by discrimination of the surfaces of the cycle (figure 6, b). The indicated power is proportional to the area defined by the actual PV curves. The useful compression power is proportional to the surface between the cylinder compression curve, the cylinder pressure release curve, and the high-side and low-side system pressures (as dash lined in the figure n°6a). The powers ratio (indicated/useful) is used as estimation of the compressor inlet and discharge process efficiency (called latter in the article “valve plate” efficiency).

A redesign was to be done, and the Pressure-Volume diagram revealed us where the possible gains are. On figure 6b, the surface between the low-side system pressure and the flat low cylinder pressure represents the suction losses, is composed by valve and plenum losses. The surface between the high-side system pressure and the cylinder high pressure represents the discharge losses, composed by valve and plenum losses.

We note on figure 6a that the pressure losses at discharge are much too high. This over-pressure was due to a badly combination of the flow pressure and spring back-load on the discharge valve (figure n°7). The spring stiffness is too high and creates an over-pressure in cylinder to open the valve and begin the discharge process.

We made a redesign of the valves springs stiffnesses, divided by 2 at discharge and 4 at suction, and of the port geometry (discharge cylindrical holes replaced by 4 slots, as well as major enhancements in path flows with 3D
CFD simulation, as presented in a second article “Part II: Valve plate and cylinder head 3D thermal & flow optimization”). These changes provided an optimization of the predicted compression cycle with regard to the condensing pressure (figure 8). The gain was -50% of discharge losses, however the dead volume was increased from 4.4 cc to 5.6 cc.

1.7 Prototype phase & experimental P-V diagram measurements

A compressor prototype was built. To optimize its efficiency level, experimental measurement of the P-V diagram were done on it. Objectives were the identification of losses during compression cycle phases, and the evaluation of secondary losses of the compressor.

The main principle of this experiment is a synchronous measurement of cylinder pressure, piston volume, and valve plate acceleration, the latter to detect the valves closings. These 3 values allow identifying the compression work, by determining indicated work and volumetric flow. But it is not sufficient to understand the whole refrigerant compression process, in particular refrigerant flow phenomena before and during the cylinder suction phase. So we have planned additional measurements for inlet, discharge and housing pressures, inlet and discharge temperatures.

The pressure sensors used are piezo-resistive, with a highly dynamics (f0 9 MHz) relative pressure range, up to 140 bars, and a third-scale possible extension. The position sensor chosen is magnetic and electronic exterior to avoid oil
distortion and pressure infiltration. It is screwed in the low bearing, and detects teeth positions; those are directly milled on the bottom crankshaft. This arrangement avoids possible distortion and vibration interferences.

All the channels are piloted by a signal acquisition & processing hardware, LMS Testlab, with 12 channels up to 102 kHz in a time acquisition. Then a Matlab post-processing calculates the pressure volume from piston-connecting rod kinematics, thermodynamics values, pressure-volume diagram and resulting efficiencies. The first results were not in phase with objectives. Major defects have then been detected with the first P-V diagram analysis and are described in [4]. Important enhancements have then been decided, including pressure balancing. The compressor prototype measured results are exposed on figure 9.

![Compression cycle](image)

**Figure 9: Scheme of measurements for a full P-V diagram**

Now we can see that the valve plate efficiency (and consequently the indicated efficiency) was drastically improved (figure 9 versus figure 6). The target is now fulfilled: the global compressor efficiency reached the best available market efficiency without reduced volumetric efficiency.

### 1.8 Additional values from experimental P-V diagram

The temperature and acceleration level of the valve plate are very useful for optimization. With their correct exploitation, two compressor key points were revealed: the suction phase duration, and the mechanical efficiency.

The valve timings of the suction and discharge phase are detected with the acceleration level of the valve plate. This is particularly important for the suction phase. Suction valve opening is detected by the plenum pressure changes, and suction vale closing is detected by combination of cylinder pressure and valve plate acceleration (figure 10).

![CO2 Suction 35 b](image)

**Figure 10: Suction process in a cylinder.**
This exploration could improve the volumetric efficiency with the reduction of three main defaults, first the suction valve closing delay, then the suction gas heating in the plenum chamber, and finally the dead volume of the cylinder. We can notice on figure 10 that the suction valve closing has a great delay of 2.2 ms. This delay leads to a possible 6% backflow in the suction chamber. With a reduced or null delay, the volumetric efficiency should increase, without isentropic impact if the suction phase is not perturbed. In addition, the P-V diagram tool can predict the volumetric gain: +2% each 1ms reduced delay.

The heating of the refrigerant flow in the suction chamber is increased by the thermal exchanges in the cylinder head. This is reducing the mass-flow by increasing its specific volume. We can also predict how much volumetric efficiency gain could be done with better thermal isolation of suction paths: +1% by 1K reduction of the suction flow in the plenum.

The dead volume is also responsible for a low amount of refrigerant inlet in the cylinder, because the pressurized and heated refrigerant trapped in the cylinder is re-expanded and take place of fresh refrigerant sucked in the cylinder. The predicted related gain is +3% of volumetric efficiency each cc reduced.

The whole suppression of these three defaults would lead to a perfect 100% volumetric efficiency.

By comparing the indicated power calculated by the P-V tool to the input power, we have deduced the mechanical efficiency of the compressor and quantify further improvements around motor and friction.

**CONCLUSION**

In this article we have described a new design of high efficient CO2 trans-critical compressor. Its main characteristics are a vertical shaft, with two opposed cylinders with direct suction, and a 15 kW input power. As a key design point for efficiency, valves technology choice and design have been described, as well as their optimization steps.

Experimental PV diagram set-up and analysis have been carried on a first prototype test, and results presented here. Their exploitation leaded to further improvements, presented in a second paper [4], and implemented in a final prototype.

We presented the results of this final compressor prototype tests conducted in a dedicated CO2 test bench. The compressor reached a volumetric efficiency of 74 % and the best available global efficiency of the market.

**NOMENCLATURE**

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**REFERENCES**

