

OIL-FREE TURBO-COMPRESSOR STAGE FOR LARGE-SCALE (100 KW) CO₂ HEAT PUMPS

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Abstract: Oil management is more difficult in transcritical CO₂ heat pumps than in subcritical heat pumps. Inadequate oil management can damage the compressor very quickly, so that suppliers' products use mainly standardised modules of complete CO₂ heat pumps and specially engineered large-scale installations are rare. The current project focuses on the development of an oil-free turbo-compressor for large-scale CO₂ heat pumps, which avoids the challenge of oil management, allows much simpler heat pump layouts and, for various reasons explained in the paper, leads to more efficient heat pumps. This paper presents design steps for a CO₂ micro-compressor, which operates at very high rotational speeds, resulting in high electrical efficiencies, a high power density and a compact machine. Optimization results and calculated performance data of a single stage are shown for a wide range of conditions required in this specific application. It is concluded that the stage presented allows the production of a highly efficient oil-free CO₂ compressor for application in heat pumps, resulting in a coefficient of performance (COP) of 4 at an adequately chosen reference point.

Key Words: heat pumps, CO₂, R744, radial compressor, oil-free, high-speed

1 INTRODUCTION

The thermodynamic and chemical properties of carbon dioxide (CO₂, R744) already raised interest in application as a coolant more than a century ago. The first developments of CO₂ as a refrigerant in closed cycles started in the second half of the 19th century, only half a century after the first closed cycle had been proposed for such applications (Pearson 2005). Carbon dioxide is non-toxic, the only non-flammable refrigerant suitable for vapor compression cycles below 0 °C, and for these reasons was the dominant coolant in marine applications until the mid 20th century (Kim 2004). A decline in the use of CO₂ followed, due to the availability of 'safe' synthetic coolants as (H)CFCs, the high discharge pressure required particularly for supercritical CO₂ cycles, and the difficult oil management. Interest in natural refrigerants was triggered again by legal restrictions on synthetic refrigerants in the last decades of the 20th century.

A variety of compressors for carbon dioxide have been developed and declared to have reached an 'advanced level' (Kim 2004). Both hermetic and open compressors have been realized, implementing amongst others reciprocating (piston) compressors, scroll compressors, rotary vane compressors and 'swing' or 'swash' compressors. A number of commercial compressors are available today, mainly from Japanese manufacturers, ranging from mobile air-conditioning applications to residential or commercial refrigeration systems (Kim 2004). To our knowledge, no attempts have been made to develop a radial compressor for application in CO₂ refrigerant cycles in a power range below 200 kW thermal.

In a series of academic projects, small and fast radial compressors were successfully developed for various applications (Zwyssig 2008, Schiffmann 2008, and others). It was not until a few years ago, that small radial oil-free compressors for commercial application in heat pumps or refrigerant cycles could be presented by the Danfoss Turbocor Company.

Their 2-stage hermetic radial compressor runs with R134a (250-600 kW nominal), a fluorinated synthetic refrigerant, while the use of magnetic bearings allows the compressor and refrigerant's cycle to operate without lubricant. This increases the efficiency, and simplifies the cycle. The presentation of the Turbocor compressor can be considered as a break-through.

In the present work, steps to the development of a highly-efficient, oil-free radial compression stage for application in mid-to-large-scale heat pumps (typically 100 kW thermal) are discussed. The compressor is designed to reach discharge pressure levels for transcritical cycles, allowing a reasonably high coefficient of performance (COP), but raising pressure levels to very high values compared to synthetic fluids as R134a. The compressor may be fitted into heat pumps or refrigeration cycles of the given thermal power, to produce hot water or cooling energy for freezers. The radial compressor has multiple stages with a suction-cooled electrical drive on the same spindle. With this, we aim at bringing the high-speed turbo compressor technology to CO₂ heat pumps.

In the following, design considerations and theoretical characteristics of the compressor, and details of a single compressor stage will be discussed

2 COMPRESSOR DESIGN

An oil-free design is an essential requirement in the definition of the compressor's properties. Supercritical CO₂ is an excellent solvent for all kinds of organic substances, including oils, fats and polymers. Consequently, any non-solid lubricant used in the compressor is carried through the coolant cycle. To ensure oil transport, the rising pipes need to be built including siphons. All surfaces of the heat exchangers are then continuously covered with oil, significantly reducing the heat transfer. Trans-critical CO₂ cycles include a two-phase-container from which the trapped lubricant has to be separated from the liquid coolant and extracted. Oil-free design therefore offers large advantages in efficiency, as well as simplification of the coolant's cycle. Compressor types where only low lateral forces are exerted on the bearings offer the potential to be built oil-free, as for example in small, high-speed radial turbines. Further, oil-free design combined with hermetic encapsulation of the working fluid allows high efficiencies and long mean times between services (MTBS), as exchange and cleaning of the lubricant, as well as refilling of the coolant are not necessary. This compressor is expected to reach an MTBS of 60'000 hours, being virtually service-free. Hermetic compressors include the electric motor within the working fluid (or require a magnetic coupling). Here, the electric drive is built as an electronically commutated synchronous machine with a permanent 2-pole FeNdB-magnet, both stator and rotor cooled by the working fluid. The electric motor is able to run at 50% higher power than required for the compressor's reference state. This allows the speed to be varied over a large range, strongly increasing part load efficiencies often required in heat pumps with variable evaporation temperatures. Efficiencies of electrical drive and frequency converter are expected to be above 95%.

Table 1: Conditions for a compressor range

Condition	Value	Symbol
Electrical Power	10-50 kW	P_{el}
Mass flow	0.1-0.8 kg/s	\dot{m}
Inlet pressure and temperature	2-7 MPa 260-310 K	P_{in}, T_{in}
Pressure ratio	1.3-4.4	Π_{tot}
Induced-isentropic efficiency at reference point	0.7	$\eta_{ind,is}$

First estimations of the impeller dimensions showed that the thermodynamic requirements of power ranges below 200 kW thermal result in impeller sizes below 35 mm in diameter. For the fluid dynamic design in this size range, scaling effects have to be taken into consideration. Due to the adverse influence of the surface-to-volume—ratio when going to smaller dimensions, a potentially reduced flow coefficient of the wheel is expected. Similarly, the leakage flow from impeller exit to entry decreases the net flow through the impeller, as on such small scales the manufacturing of (labyrinth) seals is almost impossible. Due to a lack of design rules of such small radial stages, the effective flow coefficient will be determined experimentally.

The aerodynamic losses of the spindle rotating in the high pressure working fluid have to be taken into consideration when calculating the mechanical power available for compression.

3 STAGE PROPERTIES AND EXPERIMENTAL TEST RIG

A single compressor stage was designed following known turbomachinery principles (e.g. Traupel 1977, Balje 1981), for initial tests. This stage is not expected to cover the complete range in pressure ratios required for the final compressor as defined above, where it will be used in a multi-stage configuration, but is currently being tested experimentally to verify the design strategy of the single stage.

The stage is equipped with a shrouded three-dimensional impeller, diffuser and return channel. For multi-stage requirements (depending on the refrigerant's cycle evaporation conditions), the stages fulfilling dimensionless similarity are easily combined to achieve wider pressure ratios. Mach numbers were chosen to be subsonic at all times. Two software tools (VistaCCD, by PCAEngineers, UK, and COMPAL/AxCent, by ConceptsNREC, USA) were used to verify and refine the stage design resulting in coincident design rules and predicted properties. The details of blade shape, hub curvature and diffuser properties were optimized with a geometry code (VistaGEO) and a throughflow code VistaTF following some of the procedures and design rules of PCA Engineers (Casey 2008). Table 1 summarizes the key figures characterizing the stage, using the following symbols: \dot{V} : volumetric flow rate; Ω : aerodynamic cross section; u_{exit} : tip speed at impeller exit; h : specific enthalpy.

Table 2: Single stage properties in the reference state

$\phi = \frac{\dot{V}}{\Omega \cdot u_{exit}}$	$\lambda = \frac{\Delta h}{u_{exit}^2}$	$\psi = \frac{\Delta h_{isentr}}{u_{exit}^2}$	Exit Mach Nr.	$\eta_{isentr} = \frac{(h_{exit,is} - h_{in}) \cdot \dot{m}}{P_{mech}}$
0.22	0.72	0.56	0.74	0.75

The leakage mass flows were calculated separately, as a strong influence on the stage performance has to be expected. A simple substitution model was developed for the two rectangular seal channels; labyrinth seals cannot be included due to dimensional restrictions. The mass flow from impeller exit to impeller inlet (Figure 1) was found to be below 8% for all operating conditions, while the mass flow from stage exit to impeller exit was found to be small enough to be neglected.

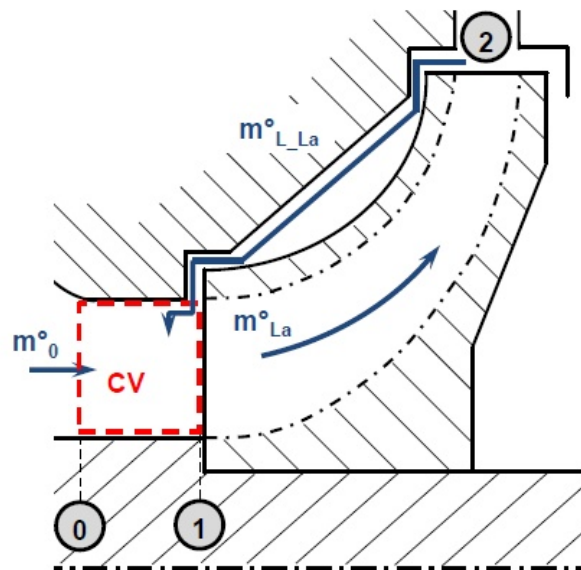


Figure 1: Cross-section of the stage, with indication for the calculated mass flow from impeller exit (2) to impeller inlet (1) ($m^\circ_{L_{La}}$).

The leakage flow was included to determine the stage performance (see Fig. 2), resulting in (theoretical) values for isentropic aerodynamic stage efficiency of about 0.75 (total-total) in the reference point. The chart in Fig. 2 shows a good performance for high Mach numbers as well as a large tolerance for variation in the flow coefficient. Both of these are required in heat pump applications due to the variability in inlet conditions.

As the leakage mass flows were calculated including the angular momentum imposed on the fluid, the resulting values could be used to calculate the axial thrust force generated by the compression stage. Axial forces as high as 300 N have to be expected.

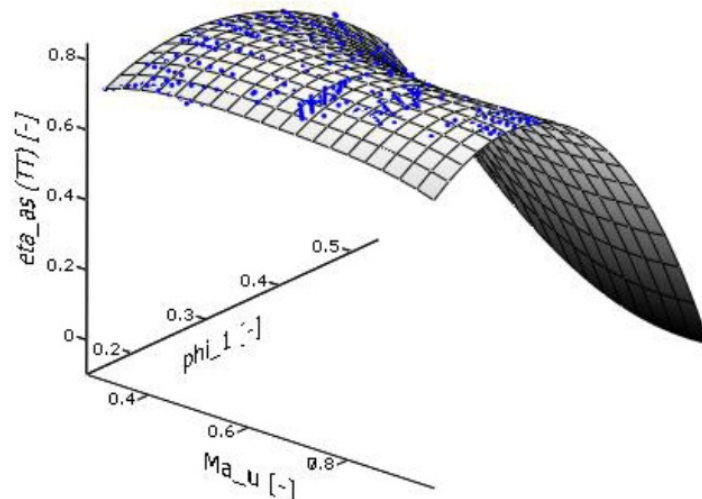


Figure 2: Calculated total-total isentropic aerodynamic stage efficiency (η_{as}) as a function of circumferential Mach number (Ma_u) and flow coefficient (ϕ_1), including losses from leakage, not including windage losses.

An impeller model/prototype resulting from these design considerations was machined out of TiAlV (Fig. 3), an alloy with low thermal conductivity (6.7 W/mK), and similarly low thermal expansion coefficient ($8.6 \cdot 10^{-6}/K$), while showing high specific modulus ($\sim 26 \cdot 10^{-6} m^2/s^2$) and

strength (~250 Nm/g). To match thermal properties, the diffuser and the return channel are produced from the same material.



Figure 3: Impeller sample, as machined.

In order to test the stage, a non-hermetic compressor was built, including a frictionless-floating suspension of the stator which allows the mechanical momentum available for compression to be measured. Measurement points for temperature and pressure are installed at the inlet, at the impeller outlet, between diffuser and return channel, and after the return channel. With this setup, the complete stage as well as its components can be characterized. Only static conditions can be measured within the stage due to space restrictions, while total pressure is obtained at the stage exit.

Upcoming tests will show if the design pressure ratios and efficiencies can be achieved.

Further testing will be performed to confirm the flow coefficient and speed tolerance, and the surge/stall boundaries. As no friction is involved in the concept, the compressor is expected to run several thousands of hours without wear.

4 CONCLUSIONS AND OUTLOOK

The design of a single radial compression stage for CO₂ compressors is shown and the test procedure and validation of the design is described. The underlying oil-free compressor concept was designed for use in heat pumps. Oil-free coolant cycles allow much simpler heat pump layouts and lead to more efficient heat pumps. Model calculations of commercial tools as well as preliminary experiments suggest that reasonably high compression efficiencies can be achieved, results of calculations show a potential of 0.75 (isentropic). The stage was designed for high tolerance towards variable evaporation conditions, which can be seen in the wide range of high efficiencies as a function of Mach number and flow coefficient. If the projected efficiencies calculated for this stage can be reached, heat pumps with a coefficient of performance (COP) of >4 at an adequately chosen reference point can be built.

5 ACKNOWLEDGEMENTS

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