Turbocompressors for Domestic Heat Pumps – A Critical Review

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Abstract

Small-scale, oil-free radial compressors on gas lubricated bearings represent a promising alternative to state-of-the-art compressor technology for driving domestic sized heat pumps. The inherent characteristics of turbocompressors manage to fit the heat pump load while the oil-freeness allows the implementation of advanced heat exchanger technology and the deployment of advanced multi-stage cycles, both validated means to significantly enhance performance and efficiency.

The paper presents experimental investigations and results of both electrically and thermally driven 20mm 2kW radial compressor rotating at rotor speeds of up to 210krpm. The compressor stage has been tested in R134a, reaching isentropic efficiencies above 70%. Dynamic, gas lubricated bearings are supporting the high-speed rotors that are lubricated with vapor phase working fluid, thus offering an oil-free and hermetic solution. Insights into the experimental investigation of a 6kW two-stage radial compressor for driving retrofit heat pumps with high temperature lifts are introduced as well.

Besides comparing experimental data, the paper also sheds light into critical design aspects related to reduced scale machines and compares them with regards to their impact on efficiency. Identified key issues are (1) aerodynamic challenges related to the severe downscaling of radial compressors such as tip clearance, surface roughness and non-adiabatic operation (2) stable gas bearing technology and (3) challenges related to high power density designs.

Keywords: residential heat pump; small-scale oil-free turbocompressors; gas bearings; radial compressors

1 Introduction

According to IEA residential heating and hot tap water production account for 20% of the total worldwide primary energy consumption [1]. Heating and cooling only need relatively low temperature levels, hence renewable energy sources offer an interesting alternative to fossil fuels. In view of categorizing promising technological combinations to provide residential heating Favrat et al. [2] clearly identify heat pumps as a key technology for reducing energy consumption for heating services. More recent work by Demierre et al. [3, 4] and by Mounier et al. [5] show further interesting alternatives to absorption heat pumps based on Organic Rankine Cycles directly driving a vapour compression heat pump cycle.

Although heat pump technology is widely spread in the residential sector a significant potential is still unexplored to improve its performance. The performance has increased significantly with the introduction of scroll compressor in the early nineties but has mainly been stagnating after that surge. Marginal improvements

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have been achieved mainly with enhanced controlling [6].

Based on this background the question rises whether a new step change can be introduced to further improve heat pump performance. An exergetic analysis by Zehnder [7] suggests that approximately 50% of the exergetic losses occur during the compression process, 30% are due to the expansion process whereas the remainder 20% occur during the heat transfer. A key enabler to enhance the heat pump performance is therefore to improve the compression efficiency. This can be achieved either through novel compressor technology and/or by splitting the compression process into several stages in combination with intercooling.

This is underlined through recent work by Arpagaus et al. [8] who suggest that multi-stage vapour compression cycles with open economizer offer the highest gain in Coefficient of Performance (COP) compared to state of the art cycles. Both the intercooled compression process and the two-stage expansion decrease the net cycles losses. A key challenge of multi-stage compression cycles with conventional components, i.e positive displacement compressors, is the oil-migration within the cycle. Oil needed for lubrication and sealing of the compressor is dragged along with the working fluid and therefore contaminates both cycle and heat exchangers. Experimental investigation by Zehnder [7] on a two-stage heat pump driven by two scroll compressors shows that no self stabilizing configuration could be found without an additional oil management system. In addition, the presence of oil in the heat exchangers is suggested to increase both pressure losses and the thermal resistance in particular in the evaporator [9, 10].

It follows that heat pump performance can be best improved by combining multi-stage cycles with oil-free and efficient compressor technology. Dynamic compression machines are known to achieve higher efficiency levels than volumetric machines and in addition, no oil is needed for sealing. Hence, combining a turbocompressor with oil-free bearings has been identified as a promising technology to further boost heat pump performance. As an example, Turbocor is well known for promoting efficient turbo compressor technology for industrial chillers. The oil-free R134a compressor is composed of two radial compressor stages supported on magnetic bearings.

In view of targeting domestic applications oil-free single stage 2kW radial compressors have been designed and tested by Schiffmann and Favrat [11, 12] achieving pressure ratios in excess of 3.3 and measured isentropic efficiencies above 80%, while rotating at speeds up to 210krpm supported on gas bearings lubricated with vapour phase R134a. More recently Carré et al. [13] have presented promising results using a two-stage radial compressor for driving a twin-stage heat pump cycle with an open economizer for residential heating.

The objective of this article is to offer a critical review and present challenges related to directly driven small-scale turbomachinery supported on dynamic gas-lubricated bearings for residential heating and cooling applications.

2 Turbocompressor scaling and design issues

2.1 Scaling Analysis

In order to assess the evolution of dimensional aspects such as impeller tip, bearing and motor diameters as a function of compressor power, a scaling analysis has been performed for a direct-driven single stage radial compressor. The compressor is assumed to deliver the pressure ratio required for a condensation at 35°C and an evaporation at 5°C, which corresponds to a typical application of a ground source heat pump for floor heating. R134a has been selected as working fluid. In order to capture the effect on bearing and motor dimensions the following assumptions have been made, based on prior work by Schiffmann and Favrat [11, 14]:

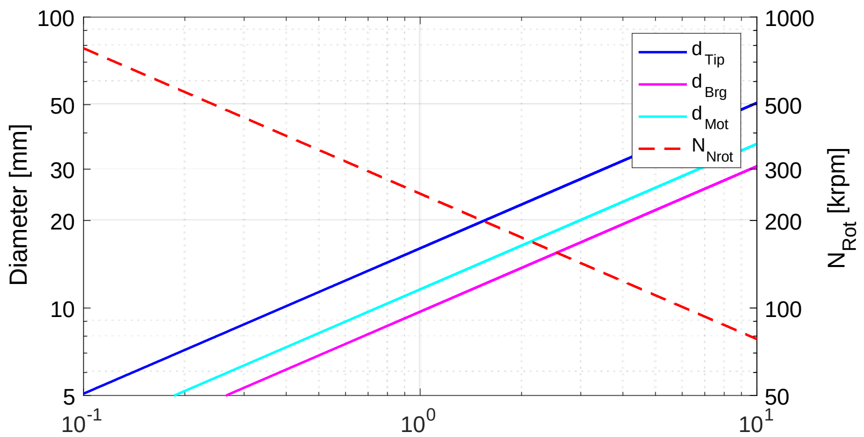
- Constant DN-bearing value, length to radius and clearance to radius ratio for the bearings
- Constant length to radius ratio for the rotor of the electric motor

Concerning the impeller it can be shown that the achievement of a given pressure ratio is directly linked to the impeller tip speed [15]. It follows that the same duty can be delivered through a large low speed impeller or by a small high-speed machine. An analysis of the aerodynamic losses, however, shows that if the impeller is too large, disk friction on the back of the impeller depreciates performance whereas if the impeller is too small, high relative velocity increases losses. From this follows that for a given duty (specific work and volume flow) an optimum rotor speed and diameter can be found. In preliminary design the Balje's charts [16] are often used to identify the optimum impeller speed and diameter for a given set of specifications. The electric motor dimensions can be estimated though the output equation, which is based on the concept of airgap shear stress [17]. A constant electromechanical efficiency is assumed in order to assess the thermal heat fluxes. A consequence of this assumption is that the specific heat flux as a result of motor losses through the stator outer diameter surface remains constant. The bearing losses are predicted via viscous shear as suggested by Cunningham et al. [18]. Table 1 summarizes the scaling dependencies and orders of the various features of a turbocompressor for refrigeration application as a function of compressor power, while keeping the required pressure ratio constant. The analysis clearly suggest that reducing turbocompressor power for a given pressure ratio, results in reduced impeller tip, bearing and motor diameters, and needs increasingly high rotor speeds. A 1kW R134a compressor would requires tip diameters in order of 15mm and rotor speeds in the order of 240krpm.

Table 1: Scaling dependencies of geometrical variables of a single stage direct driven turbocompressor

	Parameters	Scaling dependency on compressor power
Rotor speed	N_{Rot}	$E_K^{-1/2}$
Impeller tip diameter	D_{Tip}	$E_K^{1/2}$
Bearing diameter	D_{Brg}	$E_K^{1/2}$
Motor diameter	D_{Mot}	$E_K^{1/2}$
Specific thermal motor losses	q_{Mot}	1
Specific bearing losses	q_{Brg}	$E_K^{-1/2}$
Specific bearing load	$m_{Rot} / L_{Brg} D_{Brg}$	$E_K^{1/2}$

A graphical representation of the scaling analysis is given in Figure 1. As expressed above, no mechanical stress issues are to be expected through the high rotor speeds, since tip speeds of impeller, motor and bearings remain constant (governed by pressure ratio & working fluid). In addition, the specific bearing load decreases with power, which makes the implementation of advantageous bearing technology with low load capacities such gas lubricated bearings much easier at reduced scale. While downscaling does not seem to rise any dimensional and mechanical stress issues, it clearly generates challenges at a thermal management level, since the specific heat flux to be dissipated by the bearings increases with decreasing power.



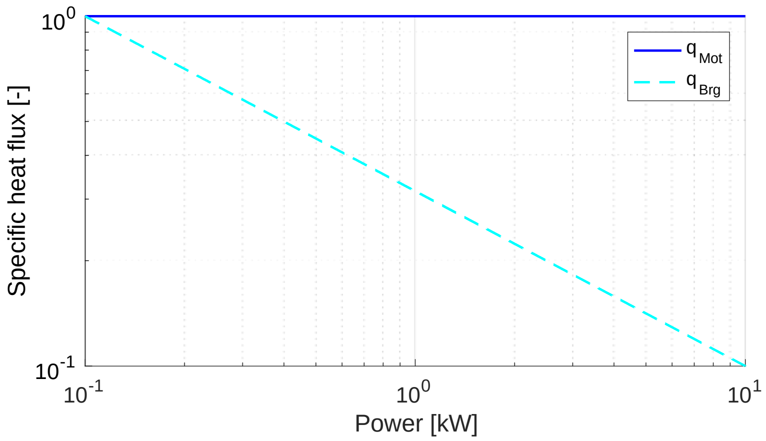


Figure 1: Scaling analysis on dimensional and thermal aspects of radial compressors for driving a heat pump with evaporation and condensation temperatures of 5 and 35°C for floor heating

2.2 Aerodynamic Scaling of Turbocompressor

Prior work by various authors demonstrates the technical feasibility of small-scale turbocompressor [19-22]. Sirakov [23] and Johnston [21], however, have clearly identified aerodynamic challenges related to turbomachinery downscaling, which may lead to lower efficiencies compared to large scale machines:

Reynolds-number. Both relative surface roughness and Re-number drop with size and therefore yield increased friction losses in small-scale turbomachinery. Casey's investigation suggest a 10 point efficiency drop when the Re-number of a compressor is reduced from 10^5 to $2{\cdot}10^4$ [24]. The advantage of processing refrigeration fluid is the high density and low viscosity (compared to air), which results in relatively high Re-numbers even for reduced-scale machines.

Tip clearance. Prior work clearly demonstrates that tip clearance has a significant impact on the loss generation [25-27]. In general, a rule of thumb suggests a 3-4 point efficiency drop for a 10% increase in tip clearance. Modern, large-scale impellers are generally operated at relative tip clearances between 1 to 5%. A 20mm tip diameter impeller with a tip blade height of 1mm would therefore need a tip clearance in the order of 10-40μm. Considering manufacturing and assembly tolerances this is very difficult to achieve. As a consequence small-scale compressors inevitably need to operate at larger tip clearances and therefore yield lower efficiencies than larger machines. As will be shown later, the bearing selection can have a significant impact on the allowable tip clearance.

Minimum feature size. Often impeller blades are relatively thin, even for large machines. Geometric downscaling can therefore not be performed deliberately, since blading would eventually become too thin to be manufacturable and mechanically sound. Hence, a downscaled impeller will suffer from increased aerodynamic blockage. To the author's knowledge, however, no experimental investigation on the effect of minimum feature size has been published so far.

Non-adiabatic flow. When downscaling, the ratio between the wetted surface and the internal flow volume increases. A consequence is that the heat exchange with blades and boundary walls increases with reducing scale, thus leading to a non-adiabatic flow. For compressors, heat addition during the compression process is known to depreciate performance.

2.3 Compressor Design

One of the challenges for a heat pump compressor is that the system itself does often not offer a stable nominal operation point. Both heating power and temperature levels are dependent on the external air temperature. In cold weather more heating power at a higher temperature level is required to keep the housing warm. In mild weather the heating power decreases, as well as the delivery temperature. Hence, the compressor needs to be able to operate in an efficient manner on a wide range of mass-flows (driven by power) and pressure levels (driven by temperatures). Schiffmann and Favrat [12] have addressed this challenge by formulating the aerodynamic compressor design as a multi-objective optimization problem. An experimentally validated 1D-compressor code was linked to an evolutionary optimization algorithm in order to find the ideal trade-off between rotor speed and seasonal efficiency. The deviation in operating conditions was captured by expressing the compressor efficiency as a weighted seasonal compressor performance, taking account for the occurrence of the various operating points as a function of a selected climatic profile. Optimizations performed on individual operation points and their comparison with the overall optimum design reveals that the wide spread of operating conditions has a significant impact on the aerodynamic compressor design. The same approach has been enhanced by Schiffmann [28] by including the bearing and the rotordynamic design of the complete compressor system. The results highlight the advantage of an integrated design and optimization approach compared to the widely spread component view philosophy.

More recently, Javed et al. [29] have analysed the effect of operational deviations of the heat pump on the compressor map and reviewed best practice compressor design applied to a 1kW radial compressor with a 15mm tip diameter and a vaneless diffuser. The results of the investigation clearly showed that operational deviation of heat pumps may be challenging for radial compressors, since its operation is limited by surge and by choke towards low and high mass flows respectively. Inlet guide vanes may be an interesting feature to further widen the operation range of the compressor, at the cost of efficiency, though. Blading optimization using 3D CFD suggests that a careful consideration of the blade loading distribution can have a beneficial impact on performance, in particular when the small-scale impeller is operated at large tip clearances. CFD based results show that an aft loaded impeller design limits the flow disturbance generated by tip clearance flow.

3 High-Speed Bearings

3.1 Bearing technology comparison

As highlighted by the scaling analysis above, turbocompressor technology for domestic scale refrigeration applications is exposed to the requirement for rotor speeds above 100krpm. In addition, a compressor for a domestic heat pump is supposed to achieve a lifetime above 120’000h without maintenance. These competing objectives call for a careful bearing selection procedure. Some particular challenges related to high rotor speeds can be summarized as follows:

- *Losses* generally increase over-proportionally with speed, resulting in temperature gradients around and to high temperatures in the bearing itself. Excessive temperatures may lead to a depreciation of both material and lubricant, whereas thermal gradients may lead to excessive mechanical distortion and eventually to failure.
- *Vibrations* due to unbalance increase with the square of the rotational speed and need to be supported by the bearings. Ideally, high-speed bearings offer high damping properties.
- *Centrifugal forces* also increase with the square of the rotational speed and generate mechanical stress in the rotor, thus limiting the maximum rotor and bearing diameter.
- *Lubrication.* The high rotational speeds generate strong windage velocities that may endanger the continuous supply of lubricant. A situation may occur where some of the relevant surfaces are not supplied with lubricant, which would lead to pre-mature failure.

Depending on the application different bearing technologies can be envisaged for high-speed rotors. As an example, the designer of a drilling spindle for printed circuit boards will focus more on load capacity and stiffness whereas for a turbocompressor minimal losses, life time and reliability are more important features. Bearing technologies used in high-speed systems can be subdivided into the following categories:

- *Rolling Element Bearing (REB)* are widely spread and robust technology. The main advantages are that it can support loads down to zero speed, its compactness and the fact that they are easily available in normalized sizes. REB presents no cross-coupled stiffness and is therefore inherently stable. Lifetime expectation reduces significantly with rotor speed since the inertial forces generated by the rolling elements may exceed the rotor loading. Unfortunately, they require oil or grease based lubrication, which can be a problem if contamination with hydrocarbons is an issue.
- *Active Magnetic Bearing (AMB).* The rotor levitates in a magnetic field without any contamination of the fluid. The bearing system is large compared to other technologies, since it requires touchdown bearings in case of power failure and an auxiliary feedback control with proximity probes and controller [30]. Stacked laminations are required on the rotor in order to decrease eddy current and hysteresis losses. These shrink-fitted laminations on the rotor do not contribute to the flexural rotor stiffness and therefore decrease the natural bending frequency. Hence, high-speed rotors on AMB are often ran overcritical. The relative space requirement increases with decreasing rotor size and therefore makes

these systems less interesting for very small applications. In addition, the overall unit becomes more complex and expensive compared to other passive technologies.

- *Fluid Film Bearing (FFB).* The lubricating fluid can be of incompressible or compressible nature. The advantages of gaseous over liquid lubricants lies most of all in the cleanliness and in the elimination of contamination caused by liquid lubricants. Furthermore seals can be avoided as the processed gas is generally usable as the bearing's lubricant. A non-negligible advantage of gas over liquid is the wide temperature range without relevant changes in properties. The lower viscosity and density of gas compared to liquids, however, considerably decrease the bearing load capacity and damping. The lower viscosity, however, decreases the mechanical losses enabling much higher surface speeds. Externally pressurized bearings require an external source of pressurized gas or liquid, needing an auxiliary pump or compressor, which represents a net energy consumption and a significant reduction of the total system efficiency. Dynamic fluid film bearings generate load capacity through the rotation of the shaft and therefore require no auxiliary systems. As these bearing types generate no load capacity and stiffness at zero speed the material selection is an important part of the deign process to avoid any damage related to stop & go cycles.

The comparison of technology clearly suggests that neither REB nor AMB correspond to an ideal choice for reduced scale, high-speed turbomachinery for energy conversion systems. REBs due to their limited life expectation and the potential contamination of the processed fluid with grease or oil, AMBs due to their complexity and requirement of auxiliary elements. It follows that small-scale turbomachinery would ideally be based on fluid film bearings lubricated with a compressible fluid, ideally with the gas processed through the machine. Further, the gas lubricated fluid film bearing should be of dynamic type in order to avoid additional energy consumption.

3.2 Gas lubricated bearings

The first theoretical investigations concerning gas-lubricated bearings have been performed by Reynolds [31] in the late 19th century. Sommerfeld [32] published first investigations on hydrodynamic lubrication and Harrison [33] developed the lubrication theory using compressible fluids as lubricant. The main drawback of gas-lubricated bearings is their cross-coupled behavior, which drives the rotor into a whirling orbit. Depending on the whirling velocity and the rotor mass the orbit size can increase with time, hence leading to an unstable behavior. The low stability margins of plain journal bearings motivated the research to offset the low stability threshold. An interesting overview of gas bearing technologies is given by Fuller [34]. The main technologies with interesting load capacity, stability and loss characteristics are foil bearings (FB) and the herringbone grooved journal bearings (HGJB), which can also be implemented as thrust bearings. The two technologies follow different philosophies to increase the stability threshold.

FBs are generally composed of thin metal foils (Figure 2, right), which are shaped such as to offer a compliant structure [35]. The motion of the spinning rotor is then used to act on the compliant structure and to deform it elastically. The deformation and the induced relative motion between foils generates Coulomb friction, which is suggested to produce damping, and thus stabilize the bearing [36]. The consequence of this working principle is that the rotor assembly needs to be able to work on large orbits, which seen from a perspective of the turbocompressor results in increased relative tip clearance and increased losses. The compliant mechanical structure, though, makes the bearing tolerant to misalignment errors and enables it to cope with significant thermal gradients. Recent investigations in foil bearings present means to improve the FB stability threshold by increasing the damping capabilities through the use of metal meshes [37] or by tailoring the fluid film through selective shimming [38].

HGJB are composed of two counteracting helicoidal pumping grooves (Figure 2, left), which increase the pressure within the fluid film and thus increase the stability threshold [14, 39]. The rigid mounting and the small bearing clearances ($C/D \approx 0.001$) require tight manufacturing tolerances, perfect alignment and particular attention with regards to thermal management to avoid large thermal gradients across the bearing clearance. However, since the bearings are generally rigidly mounted, the compressor can be run at tight tip clearances and therefore limit the efficiency penalty at reduced scale.

Figure 2: Examples of a 10mm herringbone grooved journal bearing for a 280krpm 1kW compressor (left) and of a 40mm foil bearings (right) designed, manufactured and tested at EPFL.

One of the main considerations in designing small-scale, high-speed gas bearing supported rotors is to control the forced response and to avoid rotordynamic instability. The forced response is mainly driven through unbalance and can easily be controlled by appropriate balancing procedures. The onset of instability can be monitored, however, the orbit amplitudes often increase very rapidly with rotor speed, barely leaving enough time to react to avoid catastrophic bearing failure [40]. Further, since gas-lubricated bearing offer only limited damping, crossing critical speeds involving flexural bending is very difficult. Only one example is known in literature so far for a foil bearing supported rotor [41]. An additional complication comes from the fact that both stiffness and damping matrices of fluid film bearings not only depend on rotor speed but also on the excitation frequency. As a consequence an appropriate rotordynamic model of a gas bearing supported rotor is required to predict critical speeds, modes, unbalance response and stability margins and to design an appropriate small scale turbocompressor system [42].

4 Experimental Results

4.1 Single stage proof-of-concept compressor

An electrically driven single stage proof-of-concept compressor for R134a has been designed, manufactured and successfully tested by Schiffmann and Favrat [11], which, to the author's knowledge, represents the first experimental evidence of the technical feasibility of a small-scale and oil-free turbocompressor for driving domestic heat pumps. Figure 3 represents the comparison of the oil-free proof-of-concept turbocompressor system with an equivalent scroll compressor. The turbocompressor power density is increased by one order of magnitude compared to the positive displacement machine and offers a truly oil-free and hermetic compressor system. The radial compressor has a tip diameter of 20mm and is supported on herringbone grooved journal bearings lubricated with vapour R134a.

The proof-of-concept compressor has been tested in a hermetic gas-phase loop, in order to have a better control of the compressor inlet conditions. Figure 4 represents the measured compressor map at an inlet pressure of 1.44 bara and compares the experimental data with the predictions of an in-house 1D-compressor model. Pressure ratios in excess of 3.3 have been achieved at a rotor speed of 210krpm. In addition, good agreement is shown between the 1D-model prediction and experimental data, suggesting that simple models based on empirical loss correlations are a good choice for predicting radial compressor performance. This is noteworthy, since most of the loss correlations have been derived based on large-scale impellers. Good prediction agreement is obtained, in spite of reduced size, since the Re-number are high despite the small geometrical features as a consequence of the high density and low viscosity of the working fluid.

Figure 3: Comparison of the single-stage proof-of-concept radial compressor on gas-lubricated bearings and an equivalent scroll compressor [12]

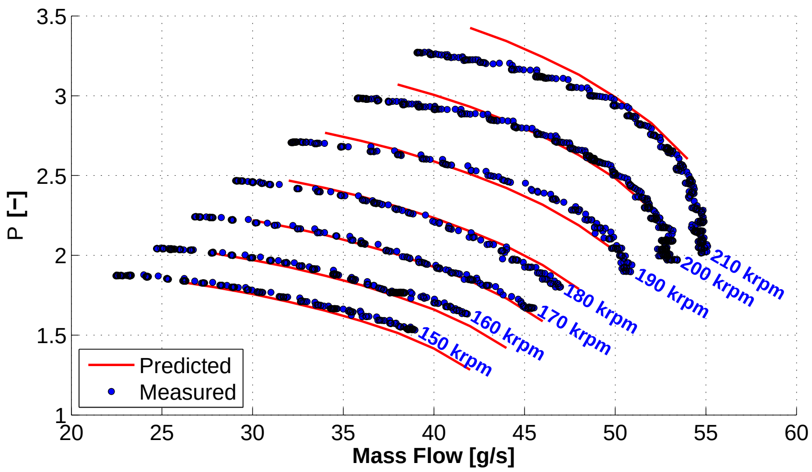


Figure 4: Compressor map measured at an inlet pressure of 1.44 bara (corresponding to an air temperature of -12°C) and comparison with a 1D compressor model prediction, showing good agreement between the model and experimental data [12]

4.2 Thermally driven heat pump compressor

An interesting experimental investigation has been performed by Demierre et al. [3, 4] where the same 20mm proof-of-concept compressor has been driven by an ORC turbine rather than by an electric motor. The particular cycle is composed of a topping Organic Rankine Cycle, which drives a bottoming vapour compression heat pump cycle. Both the heat pump and the ORC condenser are operated at the same pressure level. The heart of this tri-thermal cycle is the so-called compressor-turbine-unit (CTU), which is composed of a radial inflow ORC turbine that directly drives the radial heat pump compressor. The common rotor is supported on R134a lubricated herringbone grooved journal bearings. A comparison of the isentropic compressor efficiency shown in Figure 5 reveals that the values measured with the thermally driven compressor yields significantly lower levels compared to the ones obtained when electrically driven. Since both compressors are geometrically identical it is hypothesized that a heat flux from the hot ORC turbine towards the cold heat pump compressor is the main cause of the measured efficiency drop. These results are supporting the challenge to achieve adiabatic compression and expansion processes at reduced scales.

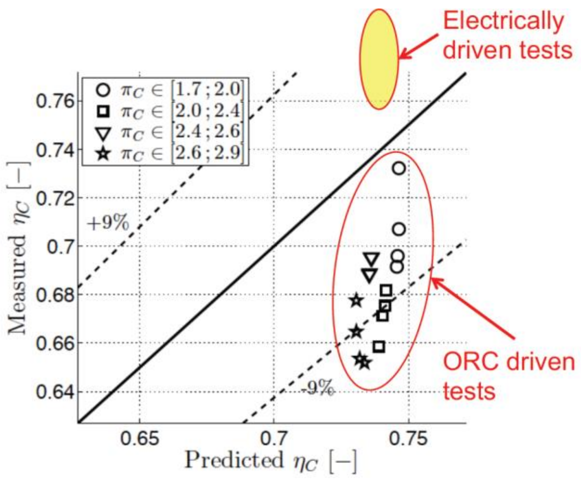


Figure 5: Comparison between the predicted and measured isentropic compressor efficiency for the electrically and the thermally driven compressor system [3]

4.3 Twin stage turbocompressor

Carré et al. [13] have recently presented their experimental investigation of a two-stage heat pump with an open economizer driven by a twin-stage turbocompressor supported on gas lubricated bearings. The tested heat pump was designed as an air-water-system for retrofit applications. At an air temperature of -7°C and a water temperature of 35°C very competitive COP have been measured, which is a remarkable achievement for a first prototype. Figure 6 schematically represents the layout of the tested compressor with two overhung back-to-back radial compressor stages and a bearing-motor cartridge that is aerated with vapour phase R134a. The motor is cooled with liquid refrigerant, which is not penalizing the cycle since the heat pump is designed for heating mode.

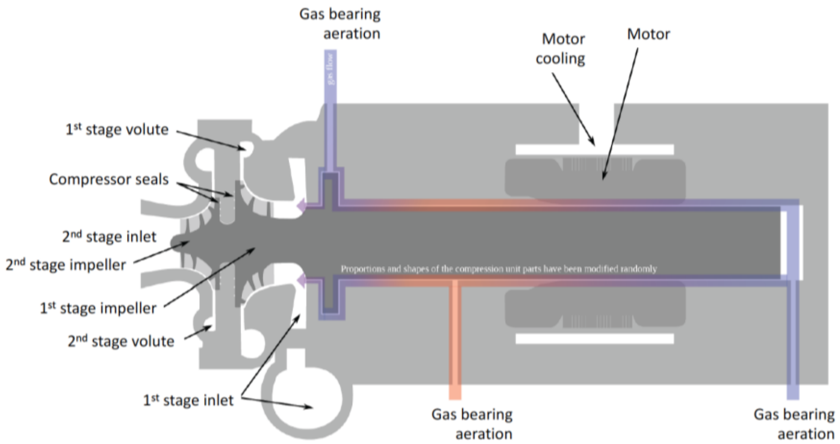


Figure 6: Schematic view of the two stage radial compressor on gas-lubricated bearings driven by a 6kW electric motor

Several challenges have been identified on this twin-stage heat pump system, which are summarized as follows:

Thrust forces and surging. Stopping a running heat pump cycle driven by a turbocompressor is challenging, since the pressure ratio across the compressor is governed by the source and sink temperatures. Hence, reducing the compressor speed while operation will inevitably result in strong compressor surge and axial forces, which might damage the thrust bearing. A compressor bypass system with check valves is therefore needed in order to protect the compressor during start-stop operation. This represents an additional cost, of course.

Thermal management. Measurements and data post-processing reveal very high temperatures in the compressor seals and in the region of the thrust bearing, which corroborates with thermal management getting increasingly challenging at reduced scale as pointed out by the scaling analysis introduced above.

Cycle pollution. A characteristic issue of gas-lubricated bearings is the tight fluid film clearance, which makes them prone to pollution and potentially failure. Experiments have revealed that a typical refrigeration loop is far from being sufficiently clean for gas-lubricated bearings. Contamination occurs from the manufacturing process of heat exchangers grease and valves. In particular oil pollution through contaminated working fluids represents a serious challenge. As a consequence great care needs to be taken in order to avoid either contamination of the cycle loop or at least of the bearing cartridge.

Compressor map matching. Difficulties have been reported to achieve stable heat pump operation, in particular with regards to the economizer pressure level. A thermodynamic analysis reveals that the pressure level in the economizer is significantly influenced by the two compressor mass-flows. It is therefore hypothesized that slight perturbations of the two compressor maps is the cause for this issue. This is corroborated by the fact that compressor speed lines are often very flat close to the surge line, i.e. minimal pressure ratio oscillations have a significant effect on the delivered mass-flows. The consequence of this phenomenon is that an additional control variable is needed to better control the matching between the two compressor stages in the heat pump loop. This can either be achieved by individually controlled single stage compressors or possibly by variable compressor geometry such as inlet guide vanes.

5 Conclusions

The article gives an overview of the current state of the art on direct driven small-scale turbocompressor for domestic heat pumps. A scaling analysis clearly shows that reducing the compressor power results in smaller impellers requiring increasingly high rotor speeds. Very high lifetime expectations and high rotor speeds are a challenging set of specifications for the bearings. Dynamic gas-lubricated bearings have been identified as the most promising technology for small-scale turbomachinery as a result of their low specific losses, their simple geometry and of their potential to offer a clean and oil-free solution. A summary of published experimental data underlines the technical feasibility of reduced scale turbocompressor for domestic heat pump applications and identifies major technological challenges related to their operation in heat pump cycles.

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Nomenaclature

C	Bearing clearance	(m)	**Acronyms**	
D_{Tip}	Compressor tip diameter	(m)	*COP*	Coefficient of performance
D_{Brg}	Bearing diameter	(m)	*CTU*	Compressor turbine unit
D_{Mot}	Compressor tip diameter	(m)	*ORC*	Organic Rankine cycle
E_K	Compressor power	(kW)	*HGJB*	Herringbone grooved journal bearing
L_{Brg}	Bearing length	(m)	*FB*	Foil bearing
N_{Rot}	Rotor speed	(rpm)	*REB*	Rolling element bearings
m_{Rot}	Rotor mass	(kg)	*AMB*	Active magnetic bearings
q_{Rot}	Specific bearing heat flux	(W/m^2)	*FFB*	Fluid film bearings
q_{Mot}	Specific motor heat flux	(W/m^2)		
Π	Compressor pressure ratio	(-)		
η_C	Compressor isentropic efficiency	(-)		

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