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The Design of Ultra-High-Speed Miniature Centrifugal Compressors

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THE DESIGN OF ULTRA-HIGH-SPEED MINIATURE CENTRIFUGAL COMPRESSORS

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ABSTRACT

This paper describes recent experience in the development and application of several ultra-high-speed miniature centrifugal compressors with an impeller diameter less than 30 mm and using high speed electric motors to provide rotational speeds between 200,000 and 600,000 rpm. In a growing number of applications at low flow rates such micro-compressors can be successfully used to replace much larger positive displacement devices or to replace larger centrifugal compressors operating at lower rotational speeds. A range of applications for heat pumps, fans and general air supply are considered in which the typical features of the stages developed are the low size, very low mass flows and high speeds. Dimensional analysis is used to show how the scaling laws for the compressor match those of the motor. These also show that a range of designs of different design styles are required; from low flow coefficient stages with two-dimensional geometry to high flow coefficient mixed flow stages. In addition to a description of the applications, and the stages developed for these, some information is provided on the design strategy, design tools and performance prediction methods used in the design process. Test data from a range of devices demonstrates that that an overall efficiency around 65% can be achieved. The measured performance is shown to agree well with the predicted performance of the stages to validate the design techniques used.

NOMENCLATURE

a	Speed of sound (m/s)
b	Channel width or height (m)
c	Chord length (m)
C	Esson's utilization factor
D_r	Motor rotor diameter (m)
D_2	Impeller tip diameter (m)
h	Specific enthalpy (J/Kg)
k_c, k_r	Coefficients in equations 3 and 4 (-)
l	Length of rotor shaft (m)
$\dot{m}$	Mass flow rate (kg/s)
M	Torque (N-m)
M_{u2}	Tip-speed Mach number (-)
n	Rotational speed (rpm)
P	Power (W)
r	Radius (m)
R	Gas constant (J/kgK)
Ra	Centre-line Average roughness (m)
T	Temperature (K)
u_2	Impeller tip blade speed (m/s)
V	Volume (m^3)
$\dot{V}_{t1}$	Volume flow rate (m^3/s)
Z	Real gas factor (-)
η	Polytropic efficiency (-)
ϕ	Flow coefficient (-)
γ	Isentropic exponent (-)
λ	Work coefficient (-)
ψ	Pressure or head rise coefficient (-)
π	Pressure ratio
ρ	Density (kg/m^3)
ω	Angular velocity (radians/sec)

Subscripts

1	condition 1, or inlet
2	condition 2, or outlet
c	compressor
m	motor
r	rotor shaft
t	total
s	isentropic

INTRODUCTION

The recent development of high speed permanent magnet (PM) electric motors has opened the way for several new innovative applications for miniature ultra-high-speed turbomachines of low flow capacity (see, Zwyssig, Kolar and Round (2009)). For convenience we define here miniature ultra-high-speed compressors as applications with an impeller tip diameter of less than 30 mm and a rotational speed greater than 200,000 rpm. These compressors are not as small as those being considered for hand-held power supply gas turbines (Epstein et al. (2004)) where an impeller of 4 mm diameter has been developed. But the compressors are smaller and rotate faster than those typically used in small turbochargers where diameters of around 30 to 50 mm are currently the smallest units in large-scale production and rotational speeds of up to 250,000 rpm may be found. In a growing number of applications such ultra-high-speed compressors can be successfully used to replace much larger positive displacement devices or to replace larger centrifugal compressors operating at a lower rotational speed.

This paper describes experience gained in the development of several miniature compressor types for a range of applications. The main concern in this paper is the aerodynamic design and performance, but some information is also provided with regard to the ultra-high-speed electrical machine design where this is appropriate, but other papers can be consulted for more detail on this aspect (Zwyssig at al. (2008)). At this stage in the development of such machines many possible applications are being investigated and these include refrigeration systems, heat pumps, ventilators, fuel cell air management systems, fuel gas compression, electrical supercharging, medical respiration assist systems and general air supply at high pressure or the production of a vacuum. Typical features of the stages developed are the low size and very low mass flows but the different applications require a range of designs covering a broad range of specific speeds from mixed flow stages to low flow coefficient stages.

The paper begins by considering fundamental issues related to the similarity rules for scaling compressors to small sizes and the associated scaling rules of the motor. This identifies some of the challenges in the design and these are then considered in more detail. Some information is provided on the design tools and performance prediction methods used in the design process. Key aerodynamic issues are the accurate prediction of the effects of tip clearance and of the Reynolds number and roughness effects on performance; together with an early estimate of the performance characteristics so that the appropriate design decisions with regard to the motor and ancillary equipment can be made. This is followed by a description of four applications and the prototype compressor stages developed for these. Test data from a range of devices is shown to agree well with the predicted performance of the stages and validates the techniques used.

SCALING CONSIDERATIONS

Similarity considerations for compressors

The well-known fundamental equations relating the head rise, efficiency and the volume flow in terms of non-dimensional parameters for a single stage compressor are given as:

$$
\begin{aligned}
&u_2 = \omega r_2 = \omega D_2 / 2 = (\pi n / 60) D_2 \\
&\dot{V}_{t1} = \phi u_2 D_2^2 = \phi \omega D_2^3 / 2 \\
&\eta = \Delta h_s / \Delta h \\
&\Delta h = \lambda u_2^2 = \lambda \omega^2 D_2^2 / 4 \\
&\Delta h_s = \psi u_2^2 = \eta \lambda u_2^2 = \eta \lambda \omega^2 D_2^2 / 4 \\
&M_{u2} = u_2 / a_{t1} = u_2 / \sqrt{\gamma Z R T_{t1}} \\
&\pi = p_{t2} / p_{t1} = \left[1 + (\gamma - 1) \lambda M_{u2}^2\right]^{\eta\gamma/(\gamma-1)}
\end{aligned}
\qquad (1)
$$

In general, the key elements of the compressor duty for the customer are the pressure rise, here also quantified as the isentropic enthalpy rise, Δh_s, the volume flow rate at inlet, V_{t1}, and the efficiency. To achieve the best efficiency the non-dimensional pressure coefficient, ψ, should be selected within a relatively small range between $0.45 < \psi < 0.55$, and then this determines the required mechanical tip speed of the compressor for a given pressure rise duty. The pressure rise thus fixes the value of the impeller tip-speed in meters per second, but not the rotational speed in radians per second, ω, or revolutions per minute, n. In non-dimensional terms the tip-speed Mach number is the appropriate aerodynamic parameter to categorize this tip speed.

If we need to design a compressor with a low volumetric flow capacity then a small machine with a low impeller diameter is needed and this then implies a higher rotational speed to retain the tip speed. For a constant tip speed, we see that the required flow coefficient for a given volume flow then increases inversely with the square of the tip diameter. Alternatively if the flow coefficient is retained then the volume flow rate reduces with the square of the diameter. These simple equations have important implications for small high speed machines. They naturally imply that higher speeds will lead to applications with lower volume flows or higher flow coefficients. There are then three interesting situations that may occur in the application of ultra-high-speed centrifugal compressors and these are described below.

The first situation occurs if the volume flow and the duty are already such that with conventional speeds and size, the flow coefficient for the given duty is below a low value of say 0.005. This implies that a conventional radial compressor is not really suitable for the application as at very low flow coefficients the friction and parasitic losses are high, leading to a very low efficiency. Typically this regime would be the realm of low-speed positive displacement machines such as reciprocating compressors, scroll compressors and screw compressors, but there are some specific turbomachinery types of very special design which may still be suitable for this region, for example, drag pumps, Balje (1981). The ability to make a smaller machine with a higher rotational speed for the same duty increases the flow coefficient, so in this case it may happen that a small high-speed radial compressor of medium to low flow coefficient can then be envisaged for this application, with substantial improvements in performance. Thus the move to higher speeds and smaller impeller diameters allows more conventional turbomachinery solutions of higher efficiency to be considered for applications where previously positive displacement machines were needed.

A second situation occurs when the current machine at a lower speed uses impellers operating below the optimum flow coefficient of around 0.09. In this case an increase in rotational speed moves the operating point towards the optimum flow coefficient for the application of radial compressors. The use of higher speeds will then probably be accompanied by an increase in efficiency, provided this is not outweighed by the penalty effects due to a decrease in size.

A third situation is where the conventional turbomachinery application already requires an impeller that has a high flow coefficient (larger than 0.09) giving a high non-dimensional swallowing capacity. In this case the adoption of a higher rotational speed with a smaller impeller requires the flow coefficient to move towards the region of application of high flow capacity centrifugal stages with mixed flow impellers, or even axial stages. Applications which currently make use of high flow radial impellers will move towards mixed flow impellers as the speed increases, with associated performance deficits, see Rusch and Casey (2012).

In this paper applications corresponding to all three situations are described and a wide range of stage types are needed to meet these requirements. The wide range of different stage types that are needed leads to an approach for their design that is a mixture of the techniques used for industrial process compressors (Dalbert et al. (1999)) or for turbocharger stages (Came and Robinson (1999)), making use of the non-dimensional parameters given above to categorize the designs and to highlight differences and similarities of different applications.

Power density and torque

From the equations given above (equation 1) we can note that for a constant tip-speed the power required by the compressor scales with the square of the impeller diameter.

$$P_c = \dot{m}\Delta h = \rho\phi\lambda\, u_2^3 D_2^2 \tag{2}$$

It is of interest to assess the power density of small machines, which is the power required for a given volume of machine. If we take the volume of the compressor to be proportional to the cube of the impeller diameter, $V_c=k_c D_2^{3}$, we obtain the power density as

$$\frac{P_c}{V_c} = \frac{P_c}{k_c D_2^3} = \frac{\rho\phi\lambda\, u_2^3 D_2^2}{k_c D_2^3} = \frac{\rho\phi\lambda\, u_2^3}{k_c}\frac{1}{D_2} \tag{3}$$

Clearly a geometrically similar machine which is down-sized, but operates at similar non-dimensional operating points (with same flow coefficient and work coefficient) and with the same blade speed will have a higher power density as the size decreases. Note however that the actual volume flow and mass flow are reduced as the size decreases so several machines may be required. As an example, if the rotational speed is doubled, reducing the size by one half, then four smaller machines would be required for the same duty. The machine volume and weight for the same duty is proportional to the cube of the diameter and reduces by a factor of eight, leading to the fact the four high speed machines still have a weight and volume advantage over the lower speed machines. This interesting and somewhat surprising result has also been pointed out by others working in this field (Epstein et al. (2006), Zwyssig et al. (2008))

If we now consider a more challenging application in which the physical mass flow of the compressor remains the same as the machine is downsized, by moving to a design with a higher flow coefficient, that is we do not retain geometrical similarity, then we obtain

$$\frac{P_c}{V_c} = \frac{P_c}{k_c D_2^3} = \frac{\dot{m}\lambda\, u_2^2}{k_c D_2^3} = \frac{\dot{m}\lambda\, u_2^2}{k_c}\frac{1}{D_2^3} \tag{4}$$

In this case there is a much more rapid increase in the power density with reduction in size. A doubling of the rotational speed, reducing the diameter by one half, requires a single machine but the power density goes up by a factor of eight with a massive weight and volume advantage.

In high speed motor design the torque required has a strong influence on the motor size. It is often of interest to design not for minimum power consumption but to design for minimum torque as torque is roughly proportional to the volume of the electrical machine, Rahman (2004). If we consider the case with a fixed duty (that is the mass flow and head rise are fixed and with a fixed work coefficient the tip speed is also fixed) then we obtain:

$$M = \frac{P_c}{\omega} = \frac{\dot{m}\lambda u_2^2}{2u_2 / D_2} = \frac{\dot{m}\lambda u_2}{2} D_2 \tag{5}$$

This indicates that the torque required for a given application scales with the impeller diameter, giving lower torque for smaller machines, so from the electrical machine there is a drive to smaller sizes. Note that these very general conclusions may be slightly affected by any changes in flow coefficient and efficiency causing a different head coefficient as the size decreases.

Scaling of high-speed electrical drives

The power P_m available from an electrical machine can be written as

$$P_m = CnlD_r^2 = \frac{ClD_r^2\, 60 u_2}{\pi D_2} = u_2 C \frac{60}{\pi}\frac{l}{D_r}\left(\frac{D_r}{D_2}\right)^3 D_2^2 \tag{6}$$

where D_r is the rotor diameter, l the active length of the rotor shaft, and n the rotational speed. C is Esson's utilization factor dependent on the machine type and other various variables such as the cooling system and size of the machine and may be considered to be a constant for a particular machine type. If the impeller tip speed, the aspect ratio of the rotor, l/D_r, and the ratio of rotor diameter to impeller diameter is kept constant, this equation indicates the motor power also scales with the square of the impeller diameter, as shown in equation 2 for the compressor. So we see that there is a good match between the motor and compressor scaling parameters for machines scaled while retaining aerodynamic similarity – the power required and the power produced are both proportional to the square of the impeller diameter.

If we consider that the volume of a motor is given by $V_m=k_r l D_r^2$ we see that for a given power, the volume of the machine decreases as the speed increases and this leads to very small machines for ultra-high speeds. If we scale the compressor while retaining geometric similarity, the motor also scales with size in a similar way to the compressor, see equation 3, such that the volume of the electrical machine decreases with increasing speed in a similar ratio to the compressor, as can be seen from the following expression for the power density

$$\frac{P_m}{V_m}=\frac{CnlD_r^2}{k_m l D_r^2}=\frac{Cn}{k_m}=\frac{C}{k_m}\frac{60u_2}{\pi D_2} \tag{7}$$

However, if we increase the rotational speed and retain a constant mass flow duty, then the power density relationship of the motor scales inversely with size whereas the compressor scales inversely with the cube of the size, see equation 4. The volume of the rotor of the electrical machine needs to increase compared to the diameter of the compressor, so that either the rotor diameter or the rotor length has to increase relative to the impeller diameter. From this we see that scaling the machine in this way but retaining the original mass flow leads to a combination of high rotational speeds and much higher power density. This is the main challenge in the design of ultra-high-speed electrical machines. The rotor robustness (length and size relative to the impeller diameter) also needs to be increased if a machine designed for an atmospheric inlet pressure is used at higher power levels with elevated pressure, or as a multistage machine with several impellers on a single shaft.

Scaling an electrical machine with a constant power rating and efficiency, and therefore constant losses, to higher speeds leads to increased losses per surface area, since the size of the machine decreases. This leads to lower utilization factors and the need for more sophisticated thermal designs for ultra-high-speed machines. This implies that for a given speed, there is a power limit depending on the machine design, materials used, rotor dynamics and thermal constraints, Zwyssig et al. (2008).

CHALLENGES OF ULTRA-HIGH-SPEED DESIGNS

High-speed electrical motor design

The high speed electrical machine considered here is a permanent magnet synchronous machine, and some details can be found in Zwyssig et al. (2008) and further references given there. The design comprises several challenges such as the mechanical rotor design, particularly the stresses in the permanent magnet (PM) and the retaining titanium sleeve. Additionally, high rotational speeds usually increase the losses, mainly due to eddy current effects in winding, stator iron and the entire rotor (magnet, iron, sleeves), but also in higher fluid friction losses. Increasing the speed with constant efficiency also results in higher losses per surface area, and therefore an improved thermal design is required. For this reason, an optimization method has been developed, which takes air-friction losses, iron losses, copper losses, and eddy-current losses into account (Luomi et al. (2009)). The electrical machine is designed for the rated specifications (speed, power, fluid) defined by the compressor specifications. For this purpose it is important to have a suitable preliminary design system for the compressor allowing power requirements and operating range to be well estimated. The stator magnetic field rotates with a high frequency (up to 10 kHz for 600 000 rpm); it is therefore necessary to minimize the losses in the stator core by using amorphous iron and the eddy-current losses in the skewed air-gap winding by using litz wire.

The rotor consists of a diametrically magnetized Sm_2Co_{17} permanent magnet which is not segmented cylindrically and is encased in a retaining titanium sleeve ensuring sufficiently low mechanical stresses on the magnet. The eccentricity is minimized by shrink-fitting the sleeve onto the PM and grinding the rotor. A cross section of the compressor D, discussed later in the paper, is shown in figure 1 to highlight the main components. The two high-speed ball bearings are assembled at each end of the rotor giving an outboard bearing configuration for this design in order to be able to service the bearings without the need for disassembling the impeller. Other configurations make use of an overhung impeller with only the motor between the bearings.

In addition to the design of the individual components, rotor dynamics of the common rotor of the electrical machine and the turbomachine is required, together with high precision balancing. The critical speeds of the rotor for compressor D are depicted in figure 2, and the rated speed is between the second and third critical speed. The critical speed calculations have been made during the electrical machine optimization process with an analytical approach in order to define geometric constraints for the machine. The final rotor dynamic design has been verified with 3-D finite-element (FE) simulations.

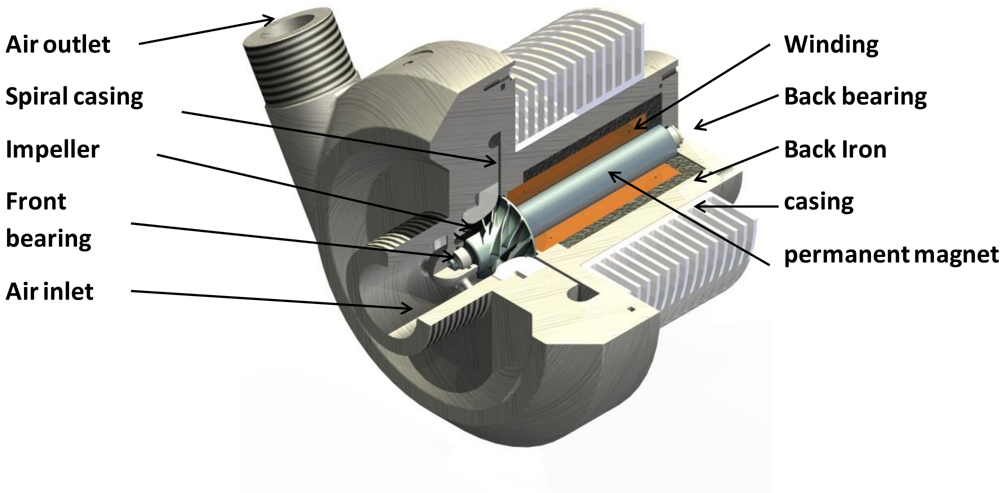
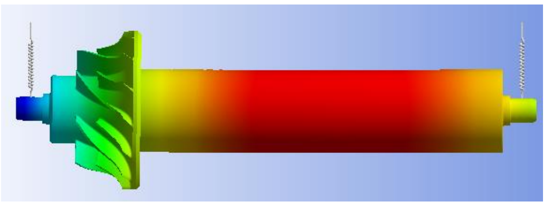
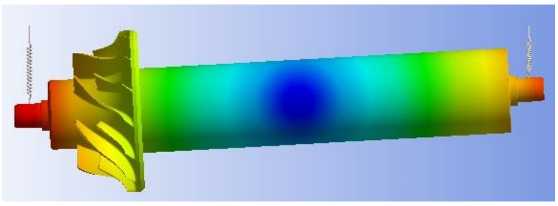


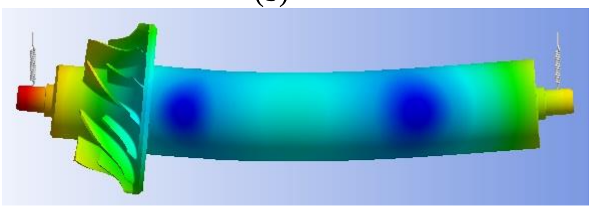
Figure 1: Cross section view of the compressor D.



(a)



(b)



(c)

Figure 2: The critical speeds of the rotor of compressor D.

(a) First critical speed at 66,000 rpm
(b) Second critical speed at 156,000 rpm
(c) Third critical speed at 480,000 rpm

Blue indicates no displacement and red indicates maximum displacement.

Aerodynamics of small compressors

The major aerodynamic technical challenges in the development of miniature compressors compared to larger machines are the effects of the small scale. A key aspect is manufacturing with micron precision to avoid the limits of relative accuracy that would occur with the straightforward downscaling of macro system manufacturing techniques, including the roughness which is generally specified as Ra = 0.8μm (N6) and is achievable by point-milling. As explained above the duty in terms of tip speed does not change with size, so that for a given inlet condition and gas leading to a particular speed of sound, the aerodynamic Mach number levels also remain constant as the design is scaled. If the mass flow is retained as the machine is made smaller then the Mach number levels naturally increase. In general no real performance deficit is related to this, unless the flow capacity causes local Mach numbers above 1.3; see Rusch and Casey (2012).

The key performance deficits which can be attributed directly to the small size are:

- Low Reynolds numbers with narrow flow channels and high relative roughness, all leading to relatively high viscous losses and larger boundary layer blockage in the channels. This loss in performance can be reduced by polishing the blades, but otherwise simply has to be estimated and accepted as unavoidable.
- Tip clearances that are high relative to the diameter due to manufacturing tolerances and bearing limitations, also leading to higher losses and more flow blockage, and possibly to a reduction of operation range. Here this effect can only be weakened by minimizing the clearances, but otherwise has to be accepted.
- Mechanical robustness during manufacture and in operation requires blades that are relatively thick (>0.3 mm), compared to values scaled from larger impellers, with leading and trailing edges that are relatively thick compared to the chord. The thicker blades require more effort on profiling the edges, whereby typically elliptic edges are used, but the thickness to chord ratio of the blades remains small and is not considered to be a major effect on the performance.
- Lower numbers of blades due to manufacturing limitations in the narrow flow channels leading to high aerodynamic loading. Splittter vanes are used to increase the blade number at outlet, if possible. Many small turbochargers are equipped with 5, 6 or 7 main blades and experience from these identifies that the blade number does not have a large effect on the efficiency on the performance of small turbochargers
- Relatively high auxiliary system losses due to the low power output level, which again cannot be avoided, except by highly polishing the impeller back-plates..

If we consider a very large turbocharger or process impeller with a diameter of 500 mm, and a typical blade span at outlet of 25 mm then it is quite possible to achieve a typical clearance between the impeller and the casing of 0.5 mm in the large machine, leading to a clearance ratio of 0.1% of the diameter or 2% of the span. If we now scale this impeller geometrically down to a diameter of 25 mm we have a blade span of 1.25 mm and the clearance scaled to this size would be 0.025 mm, that is 25 μm. Retaining the original clearance level at the small size leads to a clearance to span ratio of 40% such that the blade at outlet would now extend only over 60% of the channel. There are three strategies for dealing with this intolerably high clearance level. Firstly every attempt is made to achieve extremely low clearance, with special care in the machine tolerances and control of the rotor position axially and the rotor dynamics, such that typically in the small machines a clearance of 50 μm is achieved. Secondly in the design of small impellers the design strategy is changed compared to that generally used in larger machines in order to increase the blade span at impeller outlet, as described for compressors B and C below. Finally the additional losses and blockage of the larger clearance flows need to be taken into account in the design calculations. The losses are assumed to be related to the clearance following the work of Senoo and Ishida (1987) and an additional blockage caused by the clearance is included and also assumed to be related to the clearance to span ratio.

AERODYNAMIC DESIGN TECHNIQUES

The design techniques used for the stages are based on commercially available software tools from PCA Engineers Limited and ANSYS Inc. These have been published in detail in other sources and so only a rudimentary description of these is needed here.

The stage performance at the design point and the scoping calculations for the overall geometry were generally made with a preliminary design tool described by Came and Robinson (1999). This is a one-dimensional model of the impeller aerodynamics, including correlations for performance and real gas equations called Vista CCD, which is now also freely available for ideal gases as an iPhone App. Working from the target mass flow, pressure ratio and shaft speed, the method gives a rapid estimate of the key geometrical features of the impeller and a prediction of the efficiency at the design point. A key element of the performance estimation is the Casey-Robinson correlation, now published and described in some detail by Rusch and Casey (2012). This is coupled with additional correlations for the effect of the type of diffuser system (vaned or vaneless), for the effect of Reynolds number and roughness on radial compressors (Casey (1985)) and for tip clearance (Senoo and Ishida (1987)). With this tool it is possible to manipulate the design by adjusting a range of parameters, such as back-sweep and impeller diffusion, to explore design options within a set of constraints or to determine what shaft speed and diameter would lead to the optimum design for the motor and the compressor to maximize efficiency. The sensitivity of the performance to the tip clearance and the roughness can also be examined. Following this, an estimate of the full performance map over a range of speeds and flows can be made with a map prediction tool, Vista CCM, as outlined by Casey and Robinson (2011), which allows the operating points of the application to be compared with the expected speed characteristics and identifies possible conflicts between the required operating points at low speed and high speed. With this tool these conflicts can be evaluated early in the design, without the need for fully defined stage geometry.

The initial estimates were followed by more detailed design calculations using a throughflow method, known as Vista TF, which provides a model of the whole stage also including real gas effects where necessary, see Casey and Robinson (2010). A parameterized geometry definition system, Vista GEO, similar to that described by Casey (1983) was used to define the stage component geometry. In some cases, but not always, these steps were followed by a final design optimization by means of full 3D CFD simulations using the ANSYS BladeModeler and ANSYS CFX software. No fundamental changes were made to these tools to deal with the particular small size of these machines although, following early experience and validation against measurements, it has become general practice to include additional corrections to the blockage predictions to allow for the very small size of these machines and relatively large clearance.

SOME APPLICATIONS

The applications described below have been selected from a range of different stages that have been designed in the recent past to highlight specific features of the technology and the compressor design issues. In some cases only limited information can be supplied as the applications and designs are commercially sensitive. The tests are carried out with high quality instrumentation giving an estimated error in the mass flow of ± 2%, and in pressure rise ± 2%. The efficiency cannot be measured as accurately as this and is estimated from the motor power, with a correction for the motor losses, giving an estimated accuracy in efficiency measurement of ± 5%.

Compressor A (air vacuum blower)

Compressor A is high flow coefficient stage operating as an air vacuum blower to produce a reduced air pressure at inlet. With a design pressure ratio of 1.24, and an inlet volume flow of 0.04 m^3/s, a conventional aerodynamic design (with a radial stage at optimum flow coefficient) for this application might use a radial impeller with a diameter of around 50 mm and a speed of 80,000 rpm. In this case, however, a constraint on the overall machine diameter for this specific application forces the design to be smaller with a higher flow coefficient and guides the selection of a mixed flow impeller design with an axial outlet. A design study was carried out with the throughflow code to optimize the impeller diameter and speed, see figure 3, where the increase in meridional velocity and the change to mixed flow as the machine diameter is reduced can be seen.

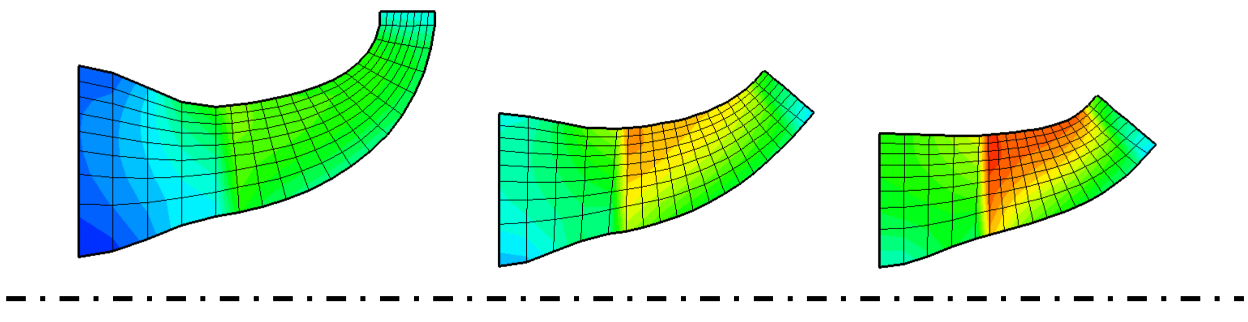


Figure 3: Impeller geometries determined for rotational speeds of 100,000, 150,000 and 200,000 rpm, showing meridional velocity contours from Vista TF throughflow calculations.

The final impeller diameter selected is less than 30 mm and the rotational speed is 200,000 rpm. The mixed flow impeller followed by a mixed flow diffuser and an axial de-swirl vane is shown in figure 4. This example is typical of the case where the requirement for a small high speed stage retaining the mass flow causes the design to be made at an extremely high flow coefficient, but nevertheless an overall efficiency with motor losses of 65% was achieved at the design point.

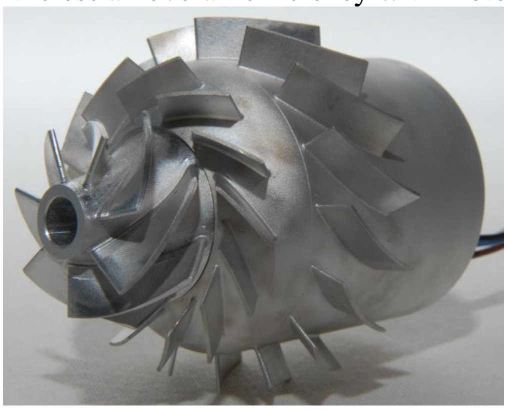


Figure 4: Prototype of compressor A, showing mixed flow impeller, vaned diffuser, axial de-swirl vanes and enclosed electric motor with a length of 85 mm, and overall diameter of 60 mm

The design rules and design strategy of compressor A have already been outlined by Casey, Zwyssig and Robinson (2009), so no further details are given. A comparison between the predicted performance map and the measured map is shown in figure 5. Note that the predicted map is generated from equations including coefficients that were derived for a range of radial stages and even without adjustment lead to a remarkably similar performance map compared to that measured. The prediction method cannot predict the flow at flows below the stability point, whereas test data can be obtained down through a region of rotating stall to zero flow, as is typical for fan applications.

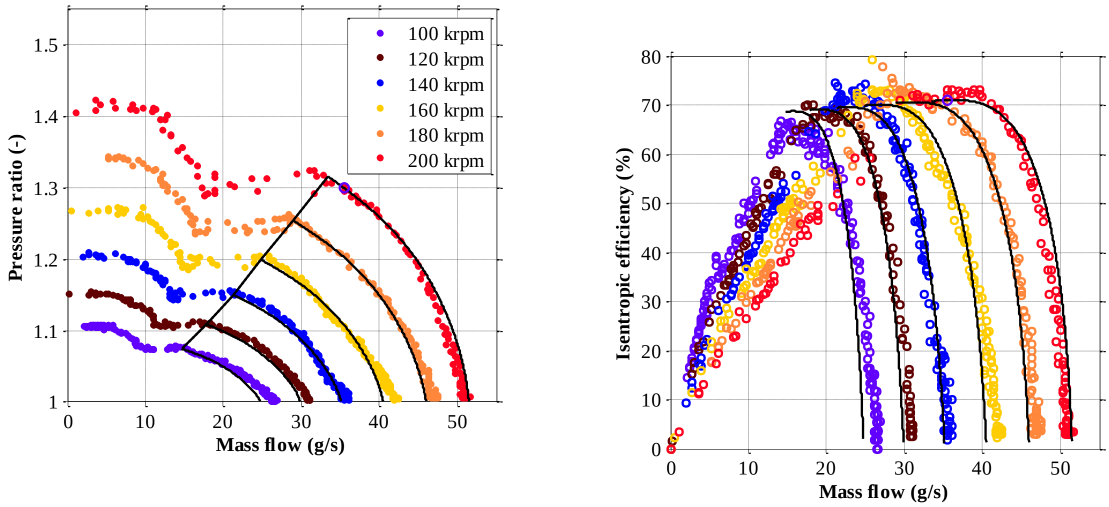


Figure 5: Measured performance of compressor A, left: pressure ratio, right: isentropic efficiency

Compressor B (water vapor vacuum blower)

Compressor B is a design for a special application which requires water vapor to be exhausted from a plenum and replaces alternative solutions with a positive displacement compressor for this purpose. The specified pressure ratio (which varies between 2 and 3.3 during the evacuation of the plenum) requires two stages of compression in series with a relatively low inlet flow (about 0.0025 kg/s). Even though the design tip-speed of the impellers is extremely high, in the neighborhood of 550 m/s and requiring the use of titanium as impeller material, the tip-speed Mach number of the stages is around 1.2 as the speed of sound in water vapor is high. The Mach number in the impeller is however subsonic. For reasons of cost the impeller was also specified as unshrouded. Initial optimization of the two-stage compressor suggested a design for the first stage at a low flow coefficient of 0.015 and a very low flow coefficient of only 0.0086 for the second stage.

Figure 6: Impeller of compressor B, diameter: 21 mm, rotor length: 36 mm

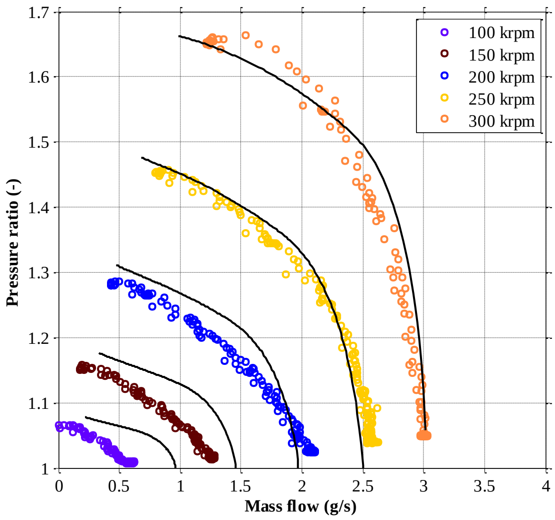


Figure 7: Predicted and measured performance map of compressor B, first stage. Measured in air at ambient inlet conditions.

At these low flow coefficients two-dimensional impellers with axial blade elements may be used without loss of performance, as shown in figure 6. This has a number of advantages over 3D impellers:

- A 2D impeller with a constant channel width across the impeller considerably simplifies the manufacture of the impeller.
- The adaptation of the impeller to different flows is easier as both the diffuser and the impeller need to only to be trimmed to a different width. In this case the second stage is essentially adapted directly from the first stage, but just trimmed to a narrower channel.
- If the stage does not meet its flow requirements it can be quickly remanufactured or trimmed with a different width to adjust the flow capacity.
- The control of the clearance is easier as only axial clearance of a purely radial disc needs to be considered, rather than axial and radial clearance of 3D designs with curved meridional channels.

The design strategy adopted in this low flow coefficient stage is similar to that described by Casey et al. (1990) and Dalbert et al. (1999), and involves the use of wider flow channels relative to the flow capacity at impeller outlet than would be used at higher flow coefficients. This leads to lower losses but to large flow angles at the impeller outlet and requires the use of a vaned diffuser to avoid vaneless diffuser rotating stall. In this case, a vaned diffuser with pinch was used, the impeller outlet width is 0.84 mm and the diffuser width is only 0.63 mm for the first stage. The overall efficiency achieved in this case was not measured but estimated as 48% for the first stage and 38% for the second, which are considered to be high values for an unshrouded low flow coefficient stage at this size. The predicted and measured pressure rise performance map of stage 1 is shown in figure 7.

Compressor C (high pressure air supply)

Compressor C is a design to supply small quantities of air from atmospheric conditions to a plenum at an elevated pressure of 2 bar. The specific application has severe limitations on the available space so that a stage of extremely small size is needed, see figure 8. In this case particular care has

been taken to demonstrate the motor-compressor technology at an extremely high rotational speed of 600,000 rpm, with an impeller diameter of only 13 mm, a swallowing capacity of 0.005 m^3/s, and an impeller tip speed of 400 m/s. The stage is designed close to the optimum specific speed (with a flow coefficient of 0.075) so that the impeller, as shown in figure 9, appears to be extremely similar to a typical turbocharger stage except for its small size.
More detailed observation of the vanes identifies that the blades are relatively thick compared to a turbocharger (thickness at blade tip of 0.3 mm, which relative to the diameter is 5 times that in a typical turbocharger), but nevertheless the thickness chord ratio is low. In order to reduce the trailing edge losses the trailing edge is rounded in an elliptic form. This stage also makes use of the design strategy in which the outlet of the impeller is wider than that of an equivalent turbocharger stage, in an attempt to reduce the tip clearance losses. In this case however the pinch at inlet to the diffuser lowers the flow angle sufficiently that a vaneless diffuser can be used. The final overall efficiency (including motor losses) achieved at this extremely small size was 63%.

Figure 8: Compressor C prototype. Length 50 mm, motor casing diameter 20 mm, compressor casing diameter 48 mm.

Figure 9: Impeller of compressor C with a 13mm diameter.

Compressor D (heat pump application)

Compressor D has been designed for a small heat pump application for domestic heating where it might be used as a replacement for conventional systems with positive displacement or scroll compressors. The refrigerant gas considered is butane (denoted as R600) and the design point during heating is taken as a pressure ratio of 2.5, a mass flow of 0.025 kg/s at a speed of 250,000 rpm. Constraints on the mechanical design suggested an optimum hub diameter of 9 mm and the preliminary design process leads to an impeller diameter of 21 mm. The stage comprises an un-shrouded impeller with splitter vanes and a vaneless diffuser, as shown in figure 1 and 10. The measured and predicted performance map in air is shown in figure 11, whereby peak efficiency in air was 65%.

The similarity of the impellers for the two very different applications of compressor C and D can be identified from figures 9 and 10. This similarity is related to the fact that in terms of the non-dimensional aerodynamic coefficients both stages are close to the optimum flow coefficient (D: 0.074, C: 0.075) and have a very similar tip-speed Mach number (D: 1.3, C:1.2). The impeller for compressor D has a slightly lower back-sweep than that of compressor C as the duty requires a slightly higher work coefficient, as can be seen in figures 9 and 10.

Based on stages C and D several similar designs have been applied to fuel cell applications with similar size, speed and performance levels. Due to the space restrictions of this paper no further details can be provided here but some information can be found in Zhao et. al (2012).

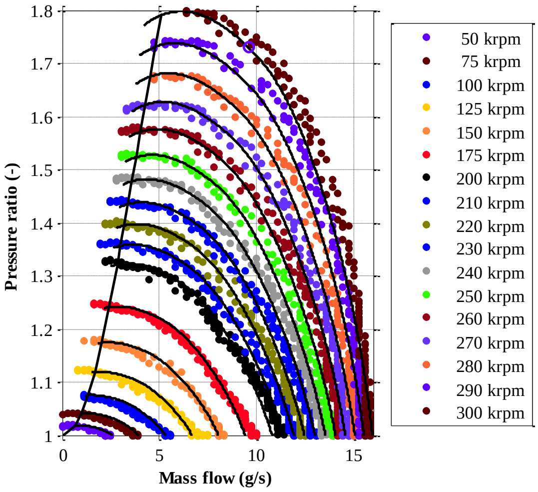


Figure 10: Assembled rotor D including impeller and high-speed ball bearings. Length 69 mm, impeller diameter 21 mm.

Figure 11: Performance map of stage D as measured in air.

CONCLUSIONS

The requirement for gas compression with low mass flow rates can now often be achieved with very small turbocompressors (less than 30 mm impeller diameter) driven by high speed permanent magnet motors, at speeds above 200,000 rpm. In a growing number of applications at low flow rates such micro-compressors achieve overall efficiencies (electrical machine and compressor) of 65% and can be used successfully to replace much larger positive displacement devices or to replace larger centrifugal compressors operating at a lower rotational speed.

The scaling laws for the downsizing of radial compressors and for high speed motors have been documented and their consequences examined:

- It is shown that with a geometric scaling of a larger compressor to a smaller size the power produced by the motor is well matched to that required by the compressor.
- It is shown that if the downsized machine requires the same mass flow as the larger machine the motor becomes more challenging and needs to be more robust as it has to produce both a high speed and a higher power density. In addition, downsizing of this nature increases the compressor flow coefficient and depending on the requirements this may move it closer to the optimum flow coefficient or push it into the range of mixed flow machines.

The examples given show that a range of different design styles are needed for different applications, ranging from low flow coefficient stages to mixed flow compressors. Key elements of the performance estimation of the small stages are correlations for the effect of Reynolds number and roughness and for the performance deficit due to tip clearance flows. It is demonstrated that the preliminary design methods used, which have been developed for much larger machines, are reasonably satisfactory for the design of such small machines and for estimating their performance.

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