THE DESIGN OF A FAMILY OF PROCESS COMPRESSOR STAGES


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ABSTRACT

The design of a new family of process compressor stages is described. The paper discusses the choice of master and derived stages to cover the required flow range and provides guidelines for the design of the stage components, including the impeller, diffuser and the return channel. Details are given of the mechanical and aerodynamic design process and the computational tools used for this. The test results show that the performance objectives have been achieved. Results from testing of some of the stages are compared with CFD simulations. These show that the inclusion of real geometry features, such as the shroud and hub leakage paths and the end-wall fillets, is necessary to obtain good agreement with the measured performance.

KEYWORDS: Process compressors, Design, CFD, Testing

NOMENCLATURE

\begin{align*}
\text{a} & : \text{inlet total speed of sound} & \text{u} & : \text{blade speed} \\
\text{A} & : \text{area} & \dot{\text{V}}_t & : \text{volume flow rate based on} \\
\text{b} & : \text{axial width} & \text{inlet total properties} \\
\text{D} & : \text{diameter} & \phi & : \frac{4\dot{V}_t}{\pi u_2 D_2^2} \text{ flow coefficient} \\
\text{h}_t & : \text{total enthalpy} & \rho_t & : \text{density based on total properties} \\
\text{l}_{\text{imp}} & : \text{impeller axial length} & \lambda & : \frac{\Delta h_t}{u_2^2} \text{ work coefficient} \\
\dot{\text{m}} & : \text{mass flow rate} & \psi_s & : \frac{\Delta h_{\text{is}}} {u_2^2} \text{ isentropic pressure rise coefficient} \\
\text{M}_{u2} & : \text{tip speed Mach number (u/a)} & & \\
\text{r} & : \text{radius} & & \\
\text{T}_t & : \text{total temperature} & & \\
\text{Subscripts} & & & \\
\text{ic, ih} & : \text{inlet casing, inlet hub} & 2 & : \text{impeller outlet} \\
\end{align*}

INTRODUCTION

Multistage industrial process compressors are designed to cover a wide range of volume flows, pressure ratios and gas properties for many different applications. In-line compressors are tailor-made from standard designs to meet the individual customer’s requirements, and to customize them to the specification, changes are introduced in the impeller diameters, number of stages, stage types,
speed and arrangements for intercooling. In order to have a short delivery time, the machines usually use families of pre-engineered stages for all the applications, where the discrete stages have already been modeled in CAD systems, tested and are ready for manufacturing. These stages must be developed in a cost-effective way and be adaptable enough to cover a range of applications. The performance of the multistage compressor is calculated by stage-stacking of the individual stage characteristics, which need to be accurately known to ensure this process works well as small errors can accumulate from stage-to-stage, see Dalbert et al. (1988).

The decision was made by Howden CKD Compressors, tailor-made centrifugal and reciprocating compressors manufacturer, to invest in the improvement of an existing compressor family by using a combination of the latest design rules and tools (CFD and mechanical analysis) to achieve improvements in efficiency, flow range and mechanical integrity. A family of new stages has been developed, consisting of several master stages and a series of derived stages to adapt these to the exact flow conditions required. The design of the stages has been completed and performance testing of selected stages from the new families using a dedicated single-stage test rig DARINA is currently ongoing.

The aim of the current paper is to describe the background to the decisions leading to the new family of designs and demonstrate some of the results. First, the methodology for selecting the master and derived stages is discussed. Then, guidelines for preliminary design of the stages are presented. In the last three sections, more information on the detailed design of the stages, testing and accuracy of the numerical calculations are provided.

SELECTION OF THE MASTER AND DERIVED STAGES

The flow coefficient of the individual stages in a multistage compressor clearly depends on the specification of the thermodynamic process and cannot be known in advance. As an example, consider the performance of individual stages in a six stage compressor, designed with a fixed flow coefficient of the first stage, but where it is designed for use with different gases. With a low molecular weight gas with a high speed of sound, the tip-speed Mach number will be low, around 0.5. The density (and volume flow) change between stages is small, see equation (A-1) in the appendix, and all stages will have a relatively high flow coefficient. The last stage may have a flow coefficient as high as 90% of the first. If the compressor is designed to operate with air at a high tip speed Mach number of, say, 1.1, a larger change in flow coefficients occurs from one stage to the next, due to larger density change across the stages, and the last stage may have a flow coefficient of only 20% of that of the first stage. For a fixed first stage flow coefficient, the inlet flow coefficient of intermediate stages can thus vary though an infinite number of values, depending on the tip-speed Mach number.

Developing a stage for each possible flow coefficient is not really practicable. Therefore, a standardization process is needed in which certain “master stages” need to be designed and tested at defined fixed points within the flow range shown in Figure 1. The master stages need to be supplemented by many additional derived stages for intermediate flow coefficients, as developing a master stage for each possible flow coefficient would be impracticable and testing of all these would be excessively expensive. At these fixed points, the performance and the geometry of the stages are completely engineered. Several of the base stages and the derived stages are tested in advance so their performance is precisely known, and this leads to a high reliability of the performance prediction accuracy. The diversity and complexity of the process compressor applications is then accounted for by using the most suitable base impellers and derived stages.

One way to fix the flow coefficients of each family is to select a constant ratio between the flow coefficients of each derived stage such that a geometric sequence is produced. The selection of the geometric ratio determines the number of stages that are required to cover the whole flow range. The approach adopted here is to use the Renard R40 series which results in a uniform geometric sequence with a step size of about 6% (40th root of 10) between the flow coefficients. The Renard series (named after French army engineer Col. Charles Renard who proposed it in the 1870’s) was
adopted in 1952 as international standard ISO 3 and has been used for standardization of turbomachinery designs in the past. An example can be found in Dalbert et al. (1988) where R40 was used for the selection of impeller outlet width ratios. The application of the R40 series results in an overall number of 40 members to cover the variation in the flow coefficient over a decade of flow coefficient from 0.15 to 0.015 and 60 members if the lowest flow coefficient is 0.0075.

In addition to the size of the steps between the stages, the location of the master stages in the series should be decided. If the application is well-known and reasonably constant for all machines, then a logical possibility would be to use thermodynamic calculations of the most common type of machine to choose this distribution. Unfortunately, the operating conditions and thermodynamic processes of most gas compressors are so variable that this is not possible. An alternative would be, say, to select every \( n^{th} \) member of the Renard series and choose this as a master stage. This would lead to an equal ratio of the flow coefficient between two adjacent master stages. The approach adopted here is a modification of this, in which this flow coefficient ratio is made smaller for the high flow coefficient stages and larger for the low flow coefficient stages. A similar standardization can be found in the stages reported by Bygrave et al. (2010). This has a number of advantages for the high flow coefficient stages. Firstly, as these tend to be relatively long compared to the low flow coefficient stages, there is a rotor-dynamical advantage in switching earlier to the low flow coefficient stages as this leads to a shorter rotor. Secondly the high flow coefficient stages have lower performance than those near to the optimum flow coefficient, so it is sensible to switch as soon as possible. In addition, the two-dimensional impeller geometries at low flow coefficients are easily designed to have a larger number of similar stages as the widths of the flow channels are small.

Based on this, seven master stages were selected to cover the entire flow range. The flow coefficient range for each stage in the family is shown in Figure 1. The derived stages can be generated by trimming the master stages from the hub or shroud side with retention of the blade shape. Typically, the high flow coefficient stages are trimmed from the hub side giving a larger shaft for the derived stages with rotor dynamic advantages, and the lower flow coefficient, more two-dimensional stages are trimmed from the shroud side. Following the procedure suggested by Dalbert et al. (1988), the trim profiles for the intermediate or derived stages were determined from the calculated streamlines in the throughflow calculation of the original master stage. The mass fraction of the stream tubes represents the required step change in the mass flow. This ensures that each stage has a similar aerodynamic flow (in the throughflow calculation) with the same inlet and outlet flow angles, the same incidence and blade loading, which should ensure that the characteristics of the derived stages are similar in shape and performance level to those of the master stage.

![Figure 1: Flow coefficient range for each stage in the family](image-url)
DESIGN GUIDELINES

Prior to detailed optimization using CFD and FEA, a set of guidelines is required to generate the basic stage geometries for various flow coefficients. Some of the guidelines are presented here and were generated using data available to the authors from past experience as well as information available in the open literature. The final design is generally within 5% of the guidelines.

**Impeller Axial Length and Eye Radius**

The impeller axial length has to increase as the flow coefficient is increased. Impellers of short axial length for high flow coefficient impellers have too much curvature in the flow channel and have a large change in meridional velocity across the leading edge of the impeller. This leads to unnecessary transonic shock losses at the impeller inlet on the casing.

Lindner (1983) has published data that shows the stage axial length has a relatively weak effect on performance, but his data do not include stages with a very high flow coefficient. Casey and Roth (1984) showed a through-flow calculation for a radial stage of short axial length and a flow coefficient $\phi = 0.140$ demonstrating the strong effect of the curvature on the meridional velocity distribution. Sorokes et al. (2009) demonstrated that the axial length is an important aerodynamic parameter, but point out that increasing the axial length also leads to mechanical and rotor dynamic problems. The paper of Sorokes et al. (2009) also refers to the guideline of Aungier (2000) for the axial length, measured from the eye on the shroud wall to the impeller outlet (Figure 7), as follows

$$l_{\text{imp}} / r_2 = 0.08 + 3.16 \phi$$

(1)

This again demonstrates the need to increase the axial length of high flow coefficient stages. An analysis of the geometry of many typical process stages has been carried out to produce the results shown in Figure 2. In the interests of generating stages that are economical in terms of axial length the guideline recommended for the axial length of the 3D impellers is given as

$$l_{\text{imp}} / r_2 = 0.1 + 2 \phi$$

(2)

This guideline may be a little on the short side for high flow coefficients and high tip speed Mach number impellers, but matches the data of the known stages better than the correlation of Aungier. Sorokes et al. (2009) commented that the guideline of Aungier may lead to stages that are too long and the only stage that matches this correlation in Figure 2 is a high Mach number stage.

![Figure 2: Axial length of the impeller ($l_{\text{imp}}/r_2$) (left) and impeller casing inlet eye radius ($r_{ie}/r_2$) (right) as a function of the flow coefficient $\phi$.](image-url)
It should be noted that a typical value of this ratio for turbocharger and gas turbine compressor impellers is between 0.6 and 0.7, so the process stages are much shorter than the equivalent impellers without a shroud used at higher Mach number. The final acceptable axial length is determined during the detailed design process for each individual master stage.

The impeller inlet eye radius of a range of process compressors (see Figure 2) has been analyzed to provide initial guidelines for this parameter in the 3D stages, as follows

\[ r_{ic} / r_2 \approx 0.5 + 1.5\phi \] (3)

The hub radius ratio is taken as radius \( r_{ih} / r_2 = 0.35 \), to avoid a shaft that is too small in diameter.

**Impeller and Diffuser Widths**

The impeller outlet width is mainly determined by the selection of the deceleration ratio in the impeller. The diffuser width is determined by the need to achieve a certain flow angle of around 55° to 60° in the vaneless diffuser for good stability of the flow. Generally an axial ‘pinch’ is needed to achieve this. Again a study of available impellers has been made to derive an approximate initial guideline for the impeller outlet width, as follows

\[ b_2 / r_2 = 0.05 + 0.8\phi \] (4)

Note that the impeller becomes generally narrower as the flow coefficient decreases but the ratio of the outlet width ratio to the flow coefficient increases as the flow coefficient gets smaller. The impellers need to be relatively narrow at high flows to avoid high mechanical loading and relatively wider at low flow coefficients to increase the width of the flow channels to reduce losses. A similar effect can be seen in the correlation given by Lüdtke (2004) for the impeller exit flow coefficient which increases as the flow coefficient increases.

\[ \phi_2 \approx 0.2 + 1.2\phi \] (5)

The guideline for the diffuser is to use a pinch of between 10% and 20% so that the diffuser is 80% to 90% of the impeller outlet width. The exact amount of pinch needed is determined in the detailed design process. This acceleration in the meridional velocity at the impeller outlet is known to be advantageous in removing flow distortion at the impeller outlet. If possible, the pinch should be placed on the shroud side, where detailed flow calculations always show that the lowest meridional velocities occur in the impeller outlet flow.

**Diffuser Radius Ratio**

A radius ratio of 1.6 (diffuser outlet diameter/impeller diameter) is selected for the vaneless diffuser. This has been selected on the basis of data given by Lüdtke (2004) which suggests that a drop in the radius ratio below 1.6 leads to a drop in stage efficiency. Lüdtke shows some of his own test data and also refers to the paper of Lindner (1983). This data is in agreement with the experience of the authors in this area. Longer diffusers have been used in practice but the additional pressure recovery in the flow does not generally outweigh the additional dissipation losses.

The recent work of Aalburg (2008) which suggests that a radius ratio of 1.3 is possible with no loss in performance is not considered to be appropriate. The return channel components have a relatively high loss coefficient (with a typical value of 0.4) and it is important to reduce the kinetic energy at inlet to the crossover bend to avoid large losses in these components. This is also supported by the work of Bygrave et al. (2010) where the optimum stage for a given casing diameter was found to be one with a long radius ratio for the vaneless diffuser. A shorter diffuser could be used (say with a radius ratio of 1.5) for vaned diffusers.
Crossover and Return Channel

In a multistage environment, the return channel is responsible for delivering the flow to the downstream impeller with, ideally, no swirl and low disturbance at the inlet. Any disturbance or separation in the inlet flow from the upstream return channel may cause flow blockage at the inlet of the downstream impeller and cause it to operate differently to the performance measured in the test stand DARINA with no upstream stage. To avoid possible separations, the return channels are normally designed to provide sufficient acceleration to the flow in the bend leading to the downstream impeller inlet. Unfortunately, the inlet of the downstream stage is not known in advance as it depends on the gas properties and the application. Hence, in a multistage compressor, the matching of the return channel with the downstream impeller needs special consideration. In the appendix, a guideline for the selection of the return channel outlet width from the inlet area of the downstream stage is presented for use during the preliminary design of a compressor stage.

As shown in equation (A-2) in the appendix, the inlet flow coefficient of a stage depends on the tip-speed Mach number of the upstream stage and is highest for low tip-speed Mach numbers. If a series of stages are designed to be used for a range of gases, the design of the return channel should be considered at low values of $M_{in2}$ to avoid excessive deceleration into the inlet of the next stage. Design for a high Mach number case would lead to a smaller return channel outlet which would then not be acceptable for the low-speed cases. This generally, results in a return channel outlet width which is much wider than the diffuser channel. Therefore, there is necessarily a large increase in channel width across the crossover bend and the return channel to the inlet of the next stage. The increase in channel can be arranged in various ways. Three basic choices for the change in width schedule are:

- increase the width across the crossover bend,
- increase the width in the return channel blading
- an evenly distributed increase in width in both components.

Previous experience suggests that the second or third option is best for high flow coefficient (3D stages) with wide flow channels, while the narrow 2D stages may be designed satisfactorily with the first option leading to a parallel walled return channel.

DETAILED DESIGN

The design of the impeller is a multidisciplinary process aiming at achieving maximum aerodynamic performance with acceptable mechanical and manufacturing properties. In this case, all impellers were of the shrouded (closed) type with three dimensional, flank-millable vanes used for the high and medium flow range ($\phi > 0.046$) and 2D impellers for the lower flow range. The 3D impellers were designed as single-piece milled impellers which allow a higher surface quality and higher tip speeds to be achieved. The 2D impellers were designed as brazed impellers, giving low thermal stress and no deformations during manufacturing, see Figure 3, whereby the location of the brazing plane was decided based on stress analysis. Generally, operating tip-speeds up to 380 m/s (X5CrNi) were tested, and maximum tip-speeds of around 500 m/s are possible with the use of titanium alloys.
Vaneless diffusers have been used at high and medium flow coefficients to give a wide range. At low flow coefficients, larger impeller tip widths have been used to avoid high loss generation in excessively narrow diffuser passages. This increases the flow angle at the diffuser inlet and therefore, vaned diffusers have been used to avoid possible flow instabilities and high losses associated with high flow angles in vaneless diffusers. This very successful design strategy for low flow coefficient impellers is discussed in detail in Casey et al. (1990), Dalbert et al. (1999) and Lettieri et al. (2014).

The stage components including the impeller, diffuser, cross-over bend and return channel were set up in PCA’s geometry definition code, VistaGEO, and designed using PCA’s throughflow code, VistaTF, see Casey and Robinson (2010) and Casey et al. (2008). The geometry definition is based on a series of parameters, so that each member of a single family has many parameters in common, except for the channel width. The different families use a similar parametrization but with different parameter values to achieve a change in the axial length and inlet radius ratio. VistaGEO generates an ANSYS BladeGen file to provide an interface to the ANSYS software. The final optimised geometry was obtained with the help of CFD analysis using ANSYS CFX. The finite element code ANSYS Mechanical was used to ensure mechanical integrity of the designs. The meridional flow paths of the stages used in a six-stage compressor are shown in Figure 4.
PERFORMANCE TESTING

The compressor stages were tested in a newly built, dedicated test facility at Howden CKD Compressors. The open loop facility enables testing of single stages with an impeller outlet diameter of 440 mm at a rotational speed of up to 18,000 rpm. The master stages were tested along with their smallest trim or derived stages to determine the performance for the entire flow range. The compressors were tested at a range of tip speed Mach number from 0.3 to 1.1. Detailed flow measurements were taken at five planes at impeller inlet, impeller outlet, diffuser outlet, return-channel inlet and return-channel outlet. Overall, more than 120 pressure and 50 temperature probes were used to study the performance of each stage. Detailed measurements also included Kiel probes, 3-hole probes and high frequency pressure transducers. The test section is shown in Figure 5.

Figure 5: Compressor test facility

VALIDATION

The main purpose of the measurements is to provide accurate test data for stage-stacking calculations of multistage compressors. The measurements have also been used to validate the results of the numerical calculations. Figure 6, shows the predicted and measured performance of two stages, representing low and high flow coefficient designs across the flow range, at a tip speed Mach number of 0.9. To demonstrate the importance of the real geometry effects, two sets of results are presented. The full model included the hub and shroud leakage paths as well as the fillets on the blade end-walls. The gas path only model does not include any of these features. Both are single passage steady-state calculations, performed using SST turbulence model in ANSYS CFX 17.1. Impeller and leakage flow paths were modelled using tetrahedral elements with an overall mesh size of about 3.5 million nodes. The flow in the diffuser and return channel were modelled using structure meshes with about 500,000 nodes. With leakage paths and fillets included (Full model) an excellent agreement can be observed between the predicted and the measured parameters. The effect of leakage flow paths is generally more important in low flow coefficient stages. The maximum flow capacity, the work coefficient and the stage peak efficiency match the test data precisely in both stages. The surge flow is predicted well for the low flow coefficient stage but over-predicted for the high flow coefficient stage. In general, surge is an instability of the entire compression system and cannot be always predicted with a high confidence by the CFD calculations.

Calculations with gas path only resulted in significantly higher efficiency levels and less steep work characteristics. The effect on the peak efficiency increases as the flow coefficient is reduced.
The calculations also over-predict the choke flow in the high flow coefficient stage which is attributed mainly to the larger fillet radius used for mechanical reasons in this stage. Including the real geometry features, such as the shroud and hub leakage paths and the end-wall fillets, is computationally expensive and therefore they are often not included in the early design iterations. As seen above, this can result in an error of few percentage points in predicted efficiency and flow capacity and must be done with caution. It is important that the impacts on the overall performance of the real geometry effects are accounted for by carrying out more detailed CFD calculations or by applying empirical correction factors.

CONCLUSIONS
The design of a new family of process compressor stages using modern design methods is described. The guidelines for preliminary design of the stages are presented and aspects of the detailed design are discussed. The test results show that the performance objectives have been achieved and the design tools have been effective. Results from testing of some of the stages are compared with CFD simulations and these show that the inclusion of real geometry features, such as the shroud and hub leakage paths and the end-wall fillets, is necessary to obtain good agreement with the measured performance.

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REFERENCES


**APPENDIX: RETURN CHANNEL MATCHING TO DOWNSTREAM STAGE**

The equations are derived with reference to the notation in Figure 7, showing the flow out of the return channel to the downstream impeller.

![Figure 7: Notation of the appendix, showing the acceleration of the flow from the return channel outlet to the impeller inlet](image)

The variation of the flow coefficient from one stage to the next is first considered. Assuming that the compression takes place as a polytropic process with polytropic exponent n, then the density ratio across a stage is given by

\[
\frac{\rho_2}{\rho_1} = \left(\frac{T_2}{T_1}\right)^{\frac{1}{n-1}} = \left(1 + (\gamma - 1)\frac{2}{\gamma} M_{u2}^2\right)^{\frac{1}{n-1}}
\]

(A-1)
Assuming constant diameter for both stages, the inlet flow coefficient of next stage, \((\phi_{n+1})\), can be calculated from that of the previous stage \((\phi_n)\) with the density ratio as

\[
\phi_{n+1} = 4m_l(\rho_{i_2}\pi D^2 u_2) = (\rho_{i_1} / \rho_{i_2})\phi_n = \frac{\phi_n}{(1 + (\gamma - 1)\lambda M^2_{u_2})^{\gamma/2}}
\]

This equation identifies that the non-dimensional volume flow coefficient of the next stage depends on the density ratio and hence on the tip speed Mach number of the upstream stage. For instance, for the case of a first stage with a flow coefficient of \(\phi = 0.15\), a work coefficient of 0.65 and a polytropic efficiency of 0.84 and \(\text{Mu}_2\) of 0.9, the downstream stage will have a flow coefficient of 0.104.

The analysis of available data shows that if the flow coefficient of the downstream \((\phi_{n+1})\) stage is known, the hub and tip radius ratios at the impeller inlet (see figure 7) can be estimated as

\[
r_{ic} / r_2 \approx 0.5 + 1.5\phi_{n+1}, \quad r_{ih} / r_2 \approx 0.35
\]

From this, the inlet area of downstream stage can be calculated as

\[
A_i / r_2^2 = \pi[(r_{ic} / r_2)^2 - (r_{ih} / r_2)^2]
\]

Defining the effective outlet radius of the return channel as a constant ratio \((R)\) of the next stage inlet tip radius, the effective area at the return channel outlet is given by

\[
r_{rc\_out} = R r_{ic}, \quad A_o = 2\pi r_{rc\_out} b_{rc\_out}
\]

From this the outlet width of the upstream return channel can be calculated as follows

\[
b_{rc\_out} / r_2 = \left(\frac{A_o}{A_i}\right)^{(1/2)}\left(\frac{r_{ic}}{r_{rc\_out}}\right)^2 = \left(\frac{A_o}{A_i}\right) \left(\frac{r_{ic}}{R r_{ic}}\right)^2 \left(\frac{(r_{ih} / r_2)^2 - (r_{ih} / r_2)^2}{2}\right)
\]

where, \(R\) and \(A_o/A_i\) are two empirical design parameters. A value of \(R = 1.25\) to 1.35 is typical to ensure a smooth bend at the inlet. As an approximate guideline, it is suggested that the mean velocity in the inlet bend should increase by 25 to 35% through the inlet bend into the downstream impeller to avoid possible flow separations, leading to a value of the area ratio \(A_o/A_i\) of between 1.25 and 1.35.