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SHROUD CONTOUR OPTIMIZATION FOR A TURBOCHARGER CENTRIFUGAL COMPRESSOR TRIM FAMILY

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ABSTRACT

A wide range of flow rates can only be covered by a family of compressors which are called trims. In the compressor map trims differ by a horizontal shift of the speed lines. Designs for lower flow rates are achieved by reducing the passage area. The goal is to maintain efficiency and pressure ratio of the base design as far as possible. In this study, an optimizing tool has been used to design the axisymmetric impeller shroud contour for each trim. Only the best efficiency point of the 100% speed line for each trim was optimized. Afterwards two complete speed lines were computed with the new geometry and analyzed. Whereas smaller trims typically show considerably lower efficiency levels, this optimization resulted in a relative efficiency loss of 0.94% only for an almost 20% smaller geometry combined with the same absolute tip clearance as the one of the base design. The pressure ratio of the best point of the base design could be maintained for all trims.

NOMENCLATURE

C_p [-] pressure recovery coefficient
$\dot{m}$ [kg s^{-1}] mass flow
N [revolution s^{-1}] rotational shaft speed
p [Pa] pressure
R [m] radial coordinate
T [K] temperature
$\dot{V}$ [m^3 s^{-1}] volume flow (p_{t1}=1 bar, T_{t1}=298 K)
y^+ [-] wall distance in normal direction
Z [m] axial coordinate
η [-] isentropic total to total efficiency
ρ [kg m^{-3}] density
Π [-] total to total pressure ratio
ζ [-] total pressure loss coefficient
Δ_{rel} [-] (opt. value – start value) / (start value)

Subscripts
1, 2 inlet and outlet of component or stage
1-7 trim sizes from low to high
arbitrary trim size
best best efficiency point on speed line
s, t static, total

Superscripts
* normalized with corresponding value of best point of base design

INTRODUCTION

Although probably all turbocharger and industrial compressor manufacturers use compressor trimming, there is surprisingly little published literature on this topic. Engeda (2007) lists the following methods to adjust an existing compressor design to a smaller flow range:

1 Varying the angle of the inlet guide vanes (if inlet guide vanes are existing)
2 Narrowing the impeller flow passage depth (impeller exit width trimming)
3 Varying the inlet angle of the diffuser vanes
4 Radial cropping of the impeller blades
5 Radial cropping of the diffuser vanes

A 6th method was introduced by Grigoriadis et al. (2012) where a conical nozzle-type element is inserted into the casing upstream of the impeller of an automotive turbocharger compressor without changing the impeller geometry. This principle implies no flow into the inducer at higher radii.

As a reason to apply trimming Engeda (2003) mentions that manufacturers (i.e. of compressors for process industry) have to respond instantly to new impeller designs or in an aftermarket situation to modification of an existing impeller. Turbocharger manufacturers in contrast have a different reason to use trimming: a turbocharging system is usually designed for a typical power class of gas or Diesel engines. The flow range for the entire power class cannot be covered by just one compressor, but by a family of compressors. The base design is applicable to the upper limit of swallowing capacity. Trims, the smaller members of the compressor family are derived from this base design. Every member has to cover a certain flow range preferably maintaining efficiency and pressure ratio of the base design. The so-called trim factor between two adjacent trims is defined that way that the entire family guarantees a gap free overlapping within a combined compressor map (Fig. 1). As the turbocharger shaft speed is the same for all trims, their designs are not “just scaled base designs” taking into account the rules of geometric similarity. In the compressor map trims differ just by a horizontal shift of the speed lines. The ratio of choke line volume flows is ideally proportional to the ratio of impeller (and diffuser) inlet cross sections. Comparisons of trim family members have been presented by Sapiro (1983) and Casey (1990), who applied method 4 from the list above by varying the shrouded impeller tip diameter. Rodgers (2001) for open impellers and Engeda (2003) for shrouded impellers have presented analog comparisons for method 2. Rodgers (2001) does not mention whether tip clearance was reduced while decreasing trim size. The present work also applies method 2 where the impeller shroud contour was separately optimized for each trim resulting in blade height reduction. In addition, the absolute tip clearance of the base design had to be maintained for all trims. This is due to mechanical reasons, because smaller gaps could cause blade rubbing. Blade rubbing is caused considerably by an impeller tilting moment, which is constant for the entire compressor family and does not allow to reduce the absolute tip clearance for smaller trims. *Absolute tip clearance* is the real distance between blade and casing with dimension “mm” while *relative tip clearance* is a percentage of blade height.

TRIM FAMILY

In the present case two adjacent trim family members of a turbocharger compressor differ in size and volume flow by a factor of 1.0388. Trim 7 is the base design and trim 2 is the smallest trim. The idea is that trim 2 can handle $1/1.0388^5 = 1/1.21 = 82.7\%$ of the volume flow of trim 7. The ratio of the flow cross sections at the impeller inlet and exit and in the entire vaned diffuser between biggest and smallest trim of this family is 1.21. Figure 2 shows an overlay of the 6 compressor trims.

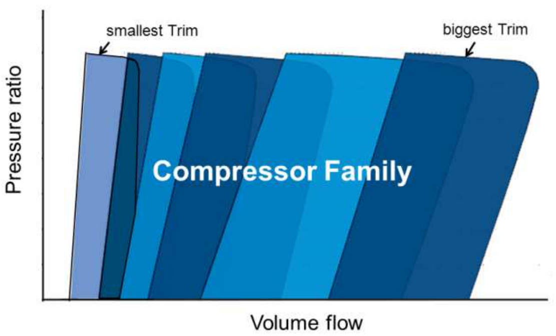

Figure 1: Compressor Map with 6 Trims

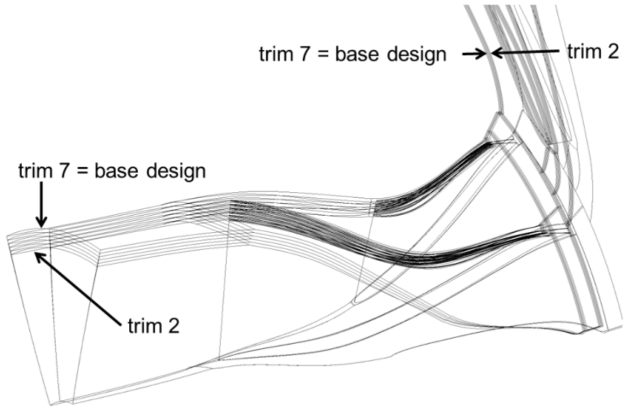

Figure 2: Overlay of 6 Trims

NUMERICAL DISCRETIZATION

The CFD model has been reduced to one pitch per component (impeller: 1’400’000 cells, diffuser: 320’000 cells) with periodic boundary conditions. The volute and the inducer casing bleed system have *not* been modeled. The mesh, consisting of structured blocks ($y^+ < 2$ along walls), is a result of mesh independence studies for a similar compressor stage. A CFD Solver from NUMECA (2010) has been applied. The Spalart-Allmaras turbulence model is used. The working fluid is air (ideal gas). Walls are assumed to be adiabatic. A central spatial discretization scheme of 2^{nd} and 4^{th} order of Jameson was used and an explicit 4-stage Runge Kutta discretization scheme in time. The mixing plane model is used for the rotor-stator interface. The relative error in mass flow balance is always less than 0.01%.

BOUNDARY CONDITIONS

All computations were run with p_t = 1 bar and T_t = 298 K at the inlet. The inflow direction is normal to the inlet boundary, here axial. For each speed line the procedure was as follows: starting at the choke line the averaged static pressure p_s at the outlet has been increased until it was impossible to arrive at a converged solution. In a second step, the mass flow at the best efficiency point computed with the previous technique was reduced until, again, no converged solution could be reached. For each trim all data base samples and optimization samples were calculated for one operating point only: the best efficiency point with a mass flow prescribed at the outlet.

OPTIMIZATION PROCESS

The trim optimization process, the geometry generation and parameterization followed by the grid generation were conducted with modules from NUMECA (2010).
Figures 3 and 4 describe the optimization process at the parameterized geometry and with a flow chart. For each trim the best efficiency point on the 100% speed line had to be improved. Before the optimization process was started, it had to be checked whether the base design (trim 7) could be improved as well. Based on this design first guesses for trims 6,5,4,3 and 2 had to be created, followed by the actual optimization. Each trim was optimized separately. The axisymmetric flow path for the first guess per trim geometry is achieved as follows (right part of figure 3):

- the hub contour (B1+B2) is kept constant for all trims
- the shroud contour has to be adjusted to get lower cross sections than for the base design:
  → C1 is shifted to lower R values
  → C3 and C4 are shifted to higher Z values
- C2 is changed manually with Bezier points to fit in between

Before starting the optimization process, for each trim the first guess or base design geometry has to be parameterized to be able to generate the basic mesh. According to Figure 3 all coordinates are kept constant except the majority of points which define curve C2 on top of the blades' tip contours. These points are allowed to change within a given reasonable interval. The blade geometry (main & splitter, each defined by 5 cuts) and the vane geometry input data never change. The vaned diffuser downstream of the interface remains "untouched" during the entire optimization process. This implies that its mesh is identical for all samples. The impeller blades have to be cut off due to the current shroud contour shape maintaining the absolute tip clearance of the base design. This is reached by the fact that only the shroud contour is exchanged in the input data for the impeller mesh generation process which adjusts the tip clearance automatically.
It is important to mention that no point on C2 should have a radial coordinate lower than on line C1. Otherwise the impeller cannot be inserted into the "one part" casing.
The number of the so-called free parameters to describe a new geometry is 11. There are 6 axial and 5 radial coordinates to define the shroud contour. The parameterization of the first guess (start solution) is called fitting. For all trims the genuine geometry and the fitted geometry match very well.
The computation of two speed lines (N*=100% and N*=70%) is repeated for the solution of the first guess for two reasons. First, one has to make sure that the parameterized geometry provides the same results as the previously non-parameterized geometry. Second, the solution of the first guess will be compared to the speed lines for the optimized geometry later on.

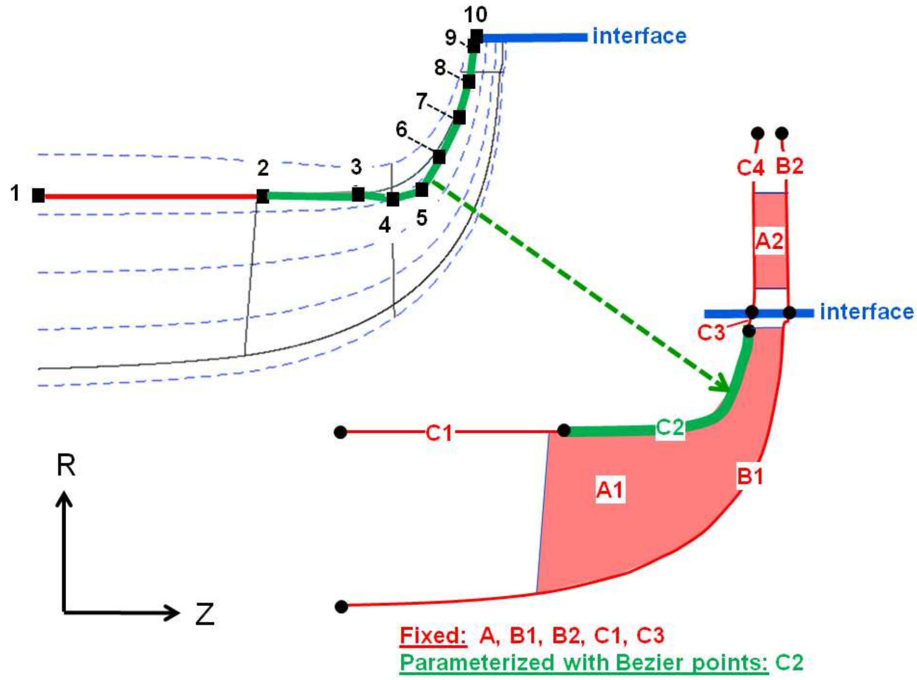

Fig. 3: Definition of Parameterized Geometry

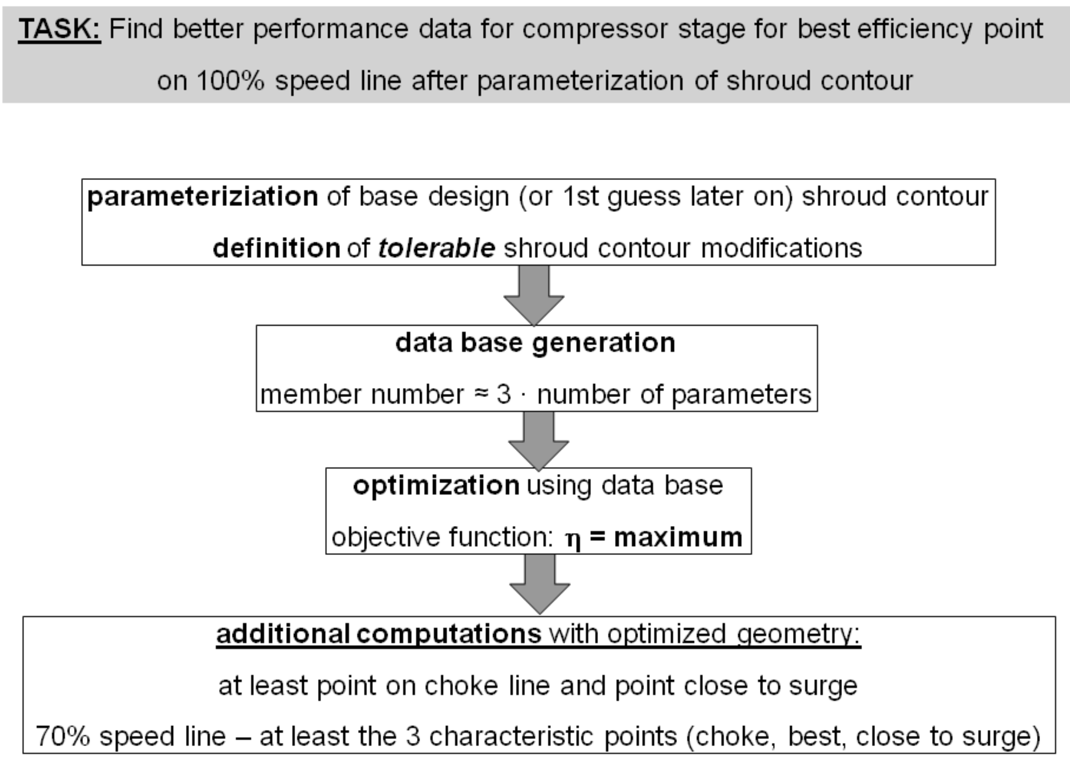

Figure 4: Flow Chart of the Optimization Process

The optimization process is based on a surrogated method. For the design of experiments (DOE), a data base was generated and calculated with CFD. The postprocessing was previously defined in order to obtain 1D parameters and typical turbomachinery CFD plots. Typically a number of samples around three times the number of free parameters is sufficient (Hildebrandt et al. (2009)).

Based on the DOE data base, the optimization is started with one objective function only: η, the total-to-total isentropic efficiency of the stage, should be maximized. Concerning all 6 optimizations after 50 samples maximum the process was stopped. It was observed that as some kind of rule η did not improve significantly after the first 25 samples.
To close the process, for each trim the computation of two speed lines had to be repeated with the optimized geometry to see the influence on other operating points of the N*=100% speed line, especially on the ones close to surge and at the choke line and on the N*=70% speed line.
The right part of figure 5 shows the results in form of optimized shroud contours for 6 trims in comparison to the start contours on the left part (base design for trim 7, first guesses for trims 6,5,4,3 and 2). Direct comparisons (before vs. after) of the contours of trim 7 and trim 4 are shown later on. The specialty of the first guesses of trims 3 and 2 are discussed in a later chapter ("Splitter blade choke as a result of a bad first guess").

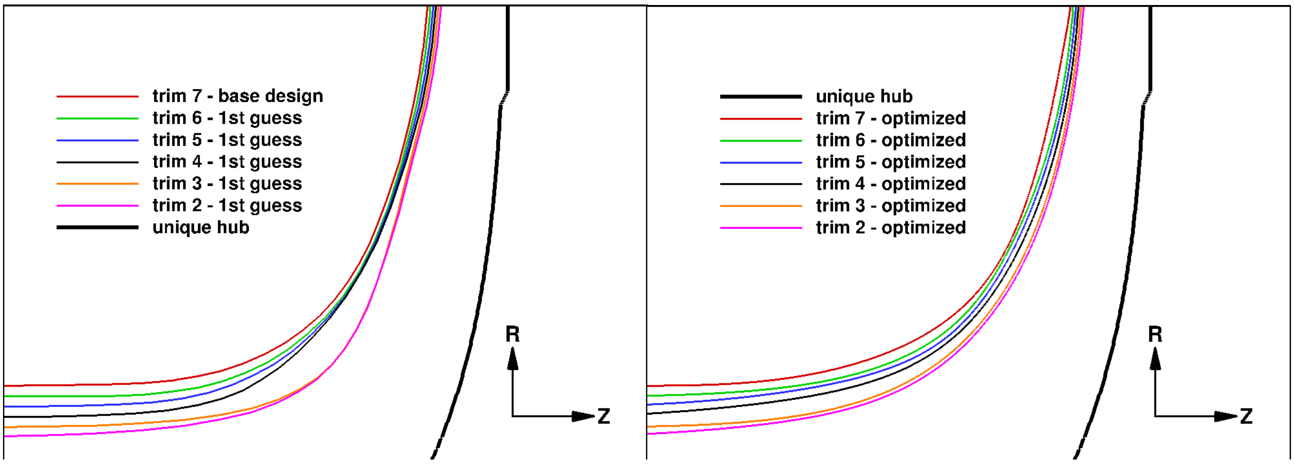

Figure 5: Shroud Contours: First Guesses (left) and optimized (right)

INFLUENCE ON THE BEST POINT OF 100% SPEED LINE PER TRIM

The efficiency gain per trim on the best point of the N^*=100% speed line is shown in figure 6 and table 1. η^* is normed by η of the base design's best point on the N^*=100% speed line. The relative gain of η^* and Π is defined as "(optimized value – start value) / start value" in percent.
All 6 optimizations were successful. The relatively low gain in η^* for the base design can be explained by the fact that the base design was developed by experienced engineers and was already improved during several iterations. As the derived smaller trims of the base design had been created primarily to produce a start solution of the optimization process, it is not surprising that the gain in efficiency is comparatively higher here.
The first guesses of trims 3 and 2 can be regarded as either "bad on purpose" or as "not checked well enough" especially in comparison to the previous trims (geometry and speed lines) before starting the optimization. But even these incidentally bad guesses did not prevent the optimization module to arrive at an optimum that fits onto a linear regression line for a "rule of thumb" which indicates the efficiency for smaller trims in the present case. The specialty of trims 3 and 2 in figure 6 are explained as follows:

1) For trims 4 to 7 there were no better efficiencies for lower or higher mass flows than for the one which was kept constant through the optimization process.
2) As the first guesses for trims 3 and 2 were choked (see later on) and the mass flow was kept constant during optimization, it turned out that the optimized shroud contour provided better efficiencies for slightly lower mass flows than for the one which was kept constant. Therefore these two points were inserted as well into figure 6 and used to compute the linear regression line.

This line indicates for the present case that between two trims the smaller one looses 0.195 % normalized efficiency η* compared to the higher one or 0.973 % between trim 7 and trim 2 (0.94% is the difference between the computed optimized points). The differences between absolute efficiencies η (both values < 100%) are even smaller, of course.

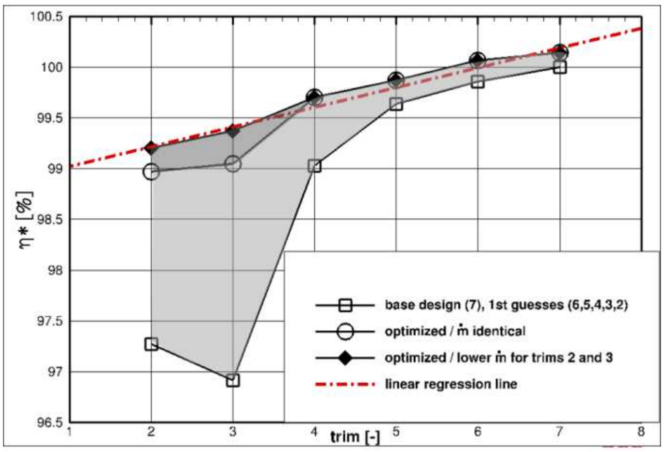

Fig 6: Efficiency Gain per Trim Optimization

η* [%]

trim	2	3	4	5	6	7
start	97.27	96.91	99.03	99.64	99.86	**100.00**
optimized	98.97	99.04	99.70	99.87	100.07	100.14
lower m	99.20	99.37				

$\Delta\eta_{rel}*$ [%]

trim	2	3	4	5	6	7
optimized	1.74	2.20	0.68	0.23	0.21	0.14
lower m	1.99	2.53				

Π [%]

trim	2	3	4	5	6	7
start	**0.975**	**0.976**	**0.990**	**0.996**	**0.996**	**1.000**
optimized	**1.022**	**0.988**	**1.016**	**1.013**	**1.008**	**1.008**
lower m	1.014	1.008				

ΔΠ* [%]

trim	2	3	4	5	6	7
optimized	4.80	1.16	2.59	1.71	1.20	0.78
lower m	4.00	3.28				

Table 1: Efficiency and Pressure Ratio per Trim Optimization

Table 1 shows efficiency and pressure ratio at the beginning and of the optimized design and their relative changes for all trim optimizations. The base design pressure ratio Π^* of 1.000 or even the optimized one (1.008) could be maintained for all trims in their best efficiency points.

It was of interest what could be gained if the *relative* tip clearance of the base design is maintained for a smaller trim. Smaller tip clearance minimizes losses caused by the flow from the pressure side to suction side of a blade. Therefore the optimized shroud contour of trim 4 was combined with the scaled tip clearance. It was possible to increase η* from 99.70% (table 1) to 99.79% for the trim 4 compressor stage. Π^* changes from 1.016 to 1.020.

INFLUENCE ON REST OF 100%-SPEED LINE AND ON ENTIRE 70%-SPEED LINE

As the optimization was applied to the best point of the N*=100% speed line only, the influence on the rest of this speed line and on part load (N*=70%) had to be investigated. This is i.e. not mentioned in Rodgers (2003) and Engeda (2007). Figure 7 shows these influences for trim 4 and trim 7 (base design). The right side (100%) shows again that the base design did not experience very much change compared to a first guess of a small trim (difference between filled and empty squares per trim). For turbocharger compressor maps usually a volume flow relating to p_{t1}=1bar and T_{t1}= 298K is applied. Therefore it is directly proportional to mass flow via ρ_{t1}=constant: $\dot{V} = \dot{m} / \rho_{t1}$.

The optimized shroud contours allowed to compute more converged results for points in the direction of small volume flow. This *could* be interpreted with a gain in surge margin (to be proved by an experiment) which is positive as well as in addition to the gain in efficiency for the best point. The left part of figure 7 shows the influence of the optimized shroud contour on part load. It would have been a lucky coincidence to achieve an improved part load speed line by optimizing with respect to the best point of the design speed line. Except for the choking part the η* curve looks as if shifted slightly downward in case of the base design. For trim 4 it is even worse: It was impossible to compute converged operating points for the optimized shroud contour as far to the left as for the base design. Counterintuetively, in part load it can happen that smaller trims show better efficiencies than higher trims do (filled and empty squares in left part of figure 7). This effect was even measured for a different trim family at the OEM. The source of this behaviour has not been investigated yet. In the work of Rodgers (2001) the same behavior is even visible for a design speed line of a Π=1.5 compressor.

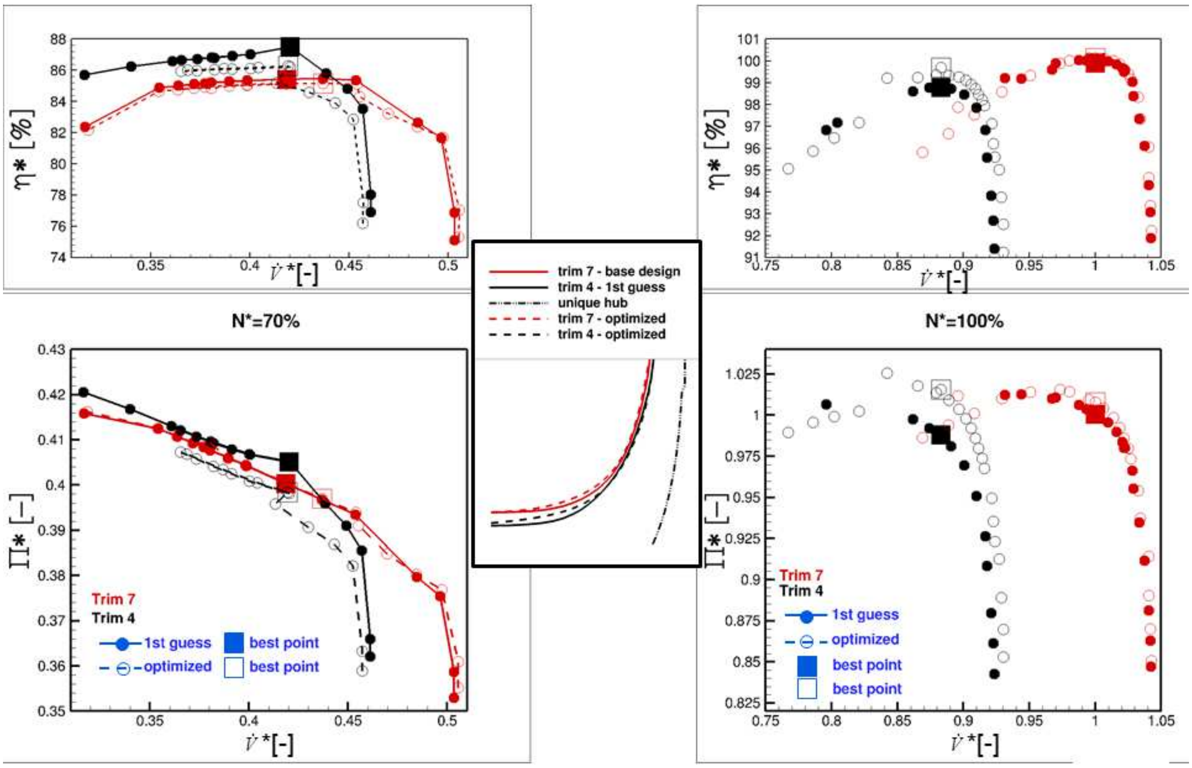

Figure 7: Compressor Map – Off-Design Speed (left)– Design Speed (right)

REASON FOR PERFORMANCE IMPROVEMENT

Table 2 shows the characteristic 1D parameters for the first guess and the optimized design of trim 4. Δ_{rel} is "(optimized value - start value)/ start value". The comparison in changes of the total-to-total efficiencies η^* for the entire stage „+0.67%" (impeller+diffuser) and the impeller only „+0.06%" points already to the fact that the optimized impeller shroud contour has primarily changed the downstream component: the vaned diffuser.

Comparisons of flow variables like static or total pressure, Mach number and entropy on several S1-, S2 and S3 planes did not show any significant change between the two impeller designs. Typical characteristic 1D parameters for diffusers like pressure recovery coefficient C_p and total pressure loss coefficient ζ account for the improvement of the diffuser performance. C_p^* increases by 11.1% and ζ^* decreases by 10.0%. Figures 8 and 9 show the change of these values in S1-planes at 50% and 95% diffuser height. The source of the favorable change of C_p^* and ζ^* is the reduction of lower velocities in the trailing edge region at the suction side of the diffuser vane and further downstream (wakes). The flow of the optimized design is better attached to the vanes.

Summarized: The impeller shroud contour optimization did not improve the impeller performance but *did* improve the diffuser performance due to more favorable inlet flow conditions did improve. The ζ^*-distribution confirms this effect and is also an explanation for the rise of η^*, because the higher total pressure values at the diffuser exit are not taken into account in the C_p^* formula. Stage balancing planes are inlet and outlet of the computational domain. Impeller balancing planes are inlet and impeller/diffuser interface of the computational domain. Diffuser balancing planes are impeller/diffuser interface and outlet of the computational domain.

isentropic efficiency

$$\eta = \frac{\left(\frac{p_{t2}}{p_{t1}}\right)^{\frac{\kappa-1}{\kappa}} - 1}{\frac{T_{t2}}{T_{t1}} - 1}$$

pressure recovery coefficient

$$C_p = \frac{p_{s2} - p_{s1}}{p_{t1} - p_{s1}}$$

total pressure loss coefficient

$$\zeta = \frac{p_{t1} - p_{t2}}{p_{t1} - p_{s1}}$$

1 – inlet of component or stage

2 – outlet of component or stage

			1st guess	*optimized design*	Δ_{rel}
normalized isentropic efficiency	$\eta^* = \frac{\eta_{\text{trim \# best}}}{\eta_{\text{base design best}}}$	*stage*	*100.00 %*	*100.67 %*	*+0.67 %*
		impeller	*100.00 %*	*100.06 %*	*+0.06 %*
normalized pressure recovery coefficient	$C_p^* = \frac{C_{p\,\text{trim \# best}}}{C_{p\,\text{base design best}}}$	*diffuser*	*89.77 %*	*99.74 %*	*+11.10 %*
total pressure loss coefficient	$\zeta^* = \frac{\zeta_{\text{trim \# best}}}{\zeta_{\text{base design best}}}$	*diffuser*	*120 %*	*108 %*	*-10.00 %*

Table 2: Characteristic Numbers to evaluate the Optimization Success for Trim 4

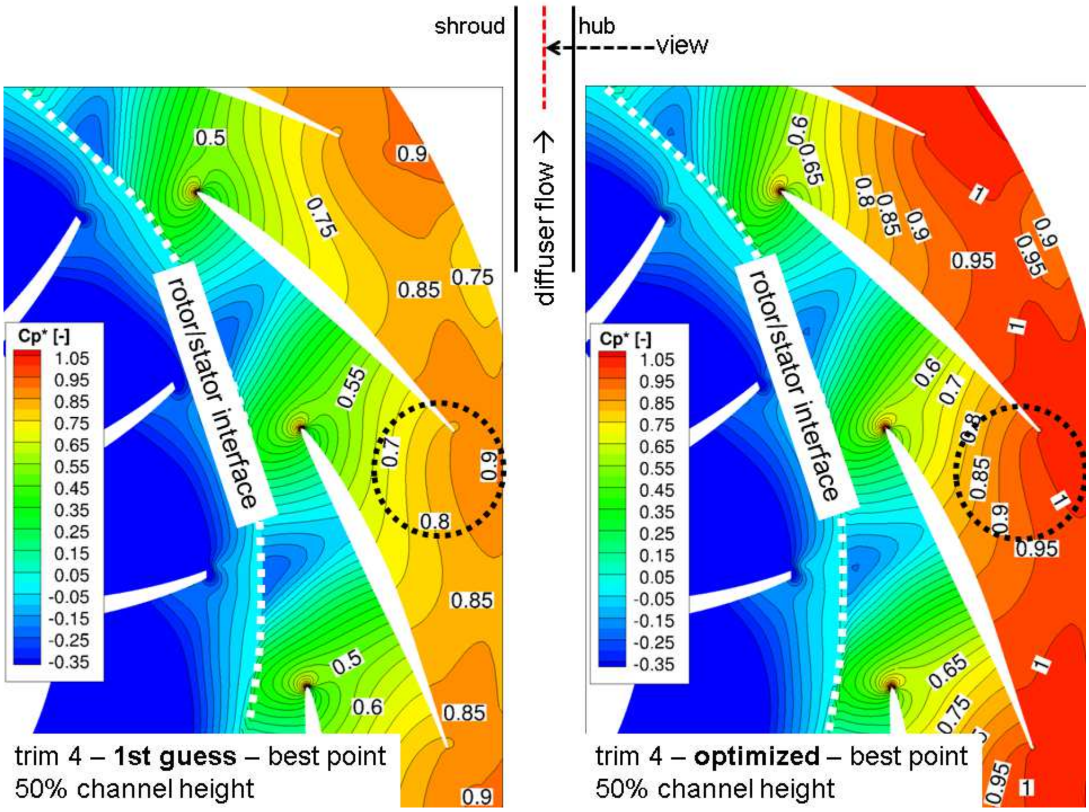

Figure 8: Diffuser: Pressure Recovery Coefficient at 50% Channel Height

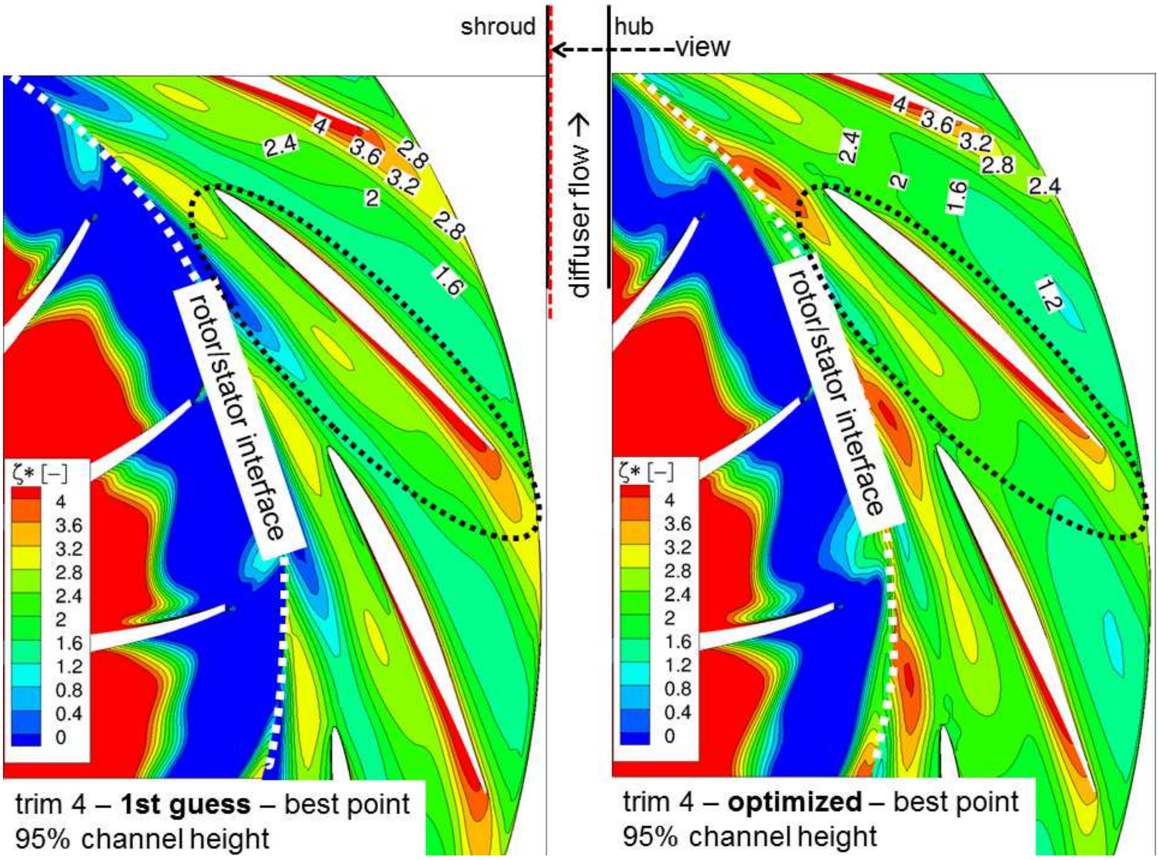

Figure 9: Diffuser: Total Pressure Loss Coefficient at 95% Channel Height

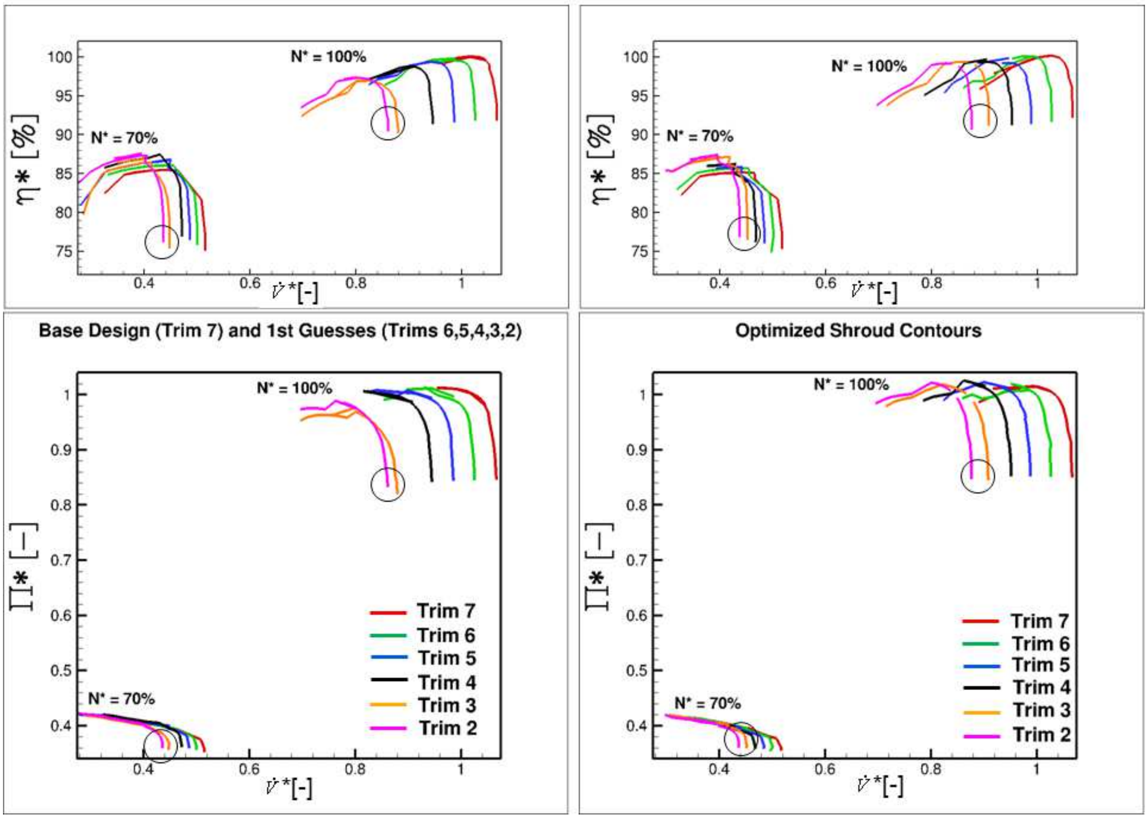

Figure 10: Set of 2 Speed Lines – Base Design+1st Guesses (left) – Optimized (right)

SPLITTER BLADE CHOKE AS RESULT OF A BAD FIRST GUESS

The “bad” first guesses concerning the shroud contours for trim 2 and trim 3 as shown in Figure 5 can be detected by just one glance when they are combined with the other four shroud contours or when all speed lines are combined in one compressor map as shown in figure 10. The contour shapes are completely different than the ones of the other trims and the gap between the speed lines of trim 4 and trim 3 is almost double the size than between trim 5 and trim 4. The shroud contour of trim 3 leads to choking at smaller volume flows than intended – as one could have expected. Figure 11 shows that in case of trim 3 choking takes place already in the inlet region of the splitter blades which of course does not exist in case of trim 7 = base design. It is obvious from figures 10 and 11 that the optimized shroud contours avoid choking in the splitter channel and lead to equidistant speed lines for full load and part load.

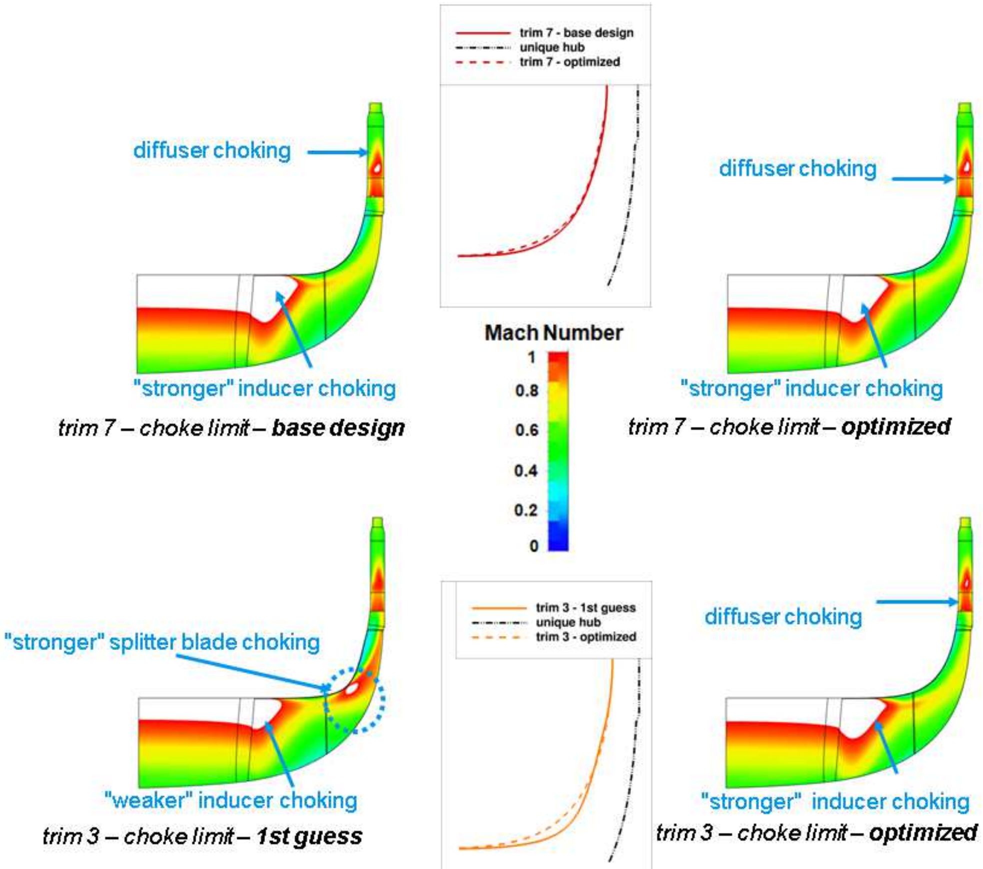

Fig 11: Corresponding Operating Points on Choking Line for Trims 7 and 3 – for start and optimized Shroud Contours, circumferentially averaged Mach number in Meridional Plane

APPEARANCE OF THE OPTIMIZED TRIM FAMILY

Operating Point		7 (Base Design)	Trim 6	Trim 5	Trim 4	Trim 3	Trim 2
peak η	$\dot{V}_{Trim+1}/\dot{V}_{Trim}$	No Trim 8	1.037	1.044	1.046	1.055	1.037
	Rel. to Trim Factor	./.	-0.17%	+0.5%	+0.7%	+1.6%	-0.17%
choke line	$\dot{V}_{Trim+1}/\dot{V}_{Trim}$	No Trim 8	1.0391	1.0389	1.038	1.048	1.037
	Rel. to Trim Factor	./.	+0.06%	+0.03%	-0.05%	+0.90%	-0.16%

Table 3: Ratio of Volume Flows of Adjacent optimized Trims at Design Speed Line

Volume ratios of the best efficiency points and points on the choke line of adjacent trims are listed in table 3 for comparison with the intended trim factor (1.0388). The relative deviations of these volume flow ratios to the constant intended trim factor are very low. They are lower for operating points on the choke line than for those of peak efficiencies. Besides the accuracy of the simulation it is supposed that the reason why these ratios differ from the exact trim factor is that the optimization was done for peak efficiency points only and that the ratio of cross sections at inlet and outlet of the impeller and in the diffuser correspond to the exact trim factor only.
The applied procedure should be improved for future optimization processes: Instead of optimizing the best efficiency point of the first guess with fixed mass flow it might be better to optimize the efficiency of a trim at the mass flow corresponding to the scaled powers of the trim factor corresponding to the base design, i.e. the fixed mass flow for trim 2 is $1/1.0388^5$ of the mass flow of the best efficiency point of the base design (trim 7).
Even if the base design has successfully passed blade vibration tests the blade vibration situation for the smaller trims has to be assessed again. Rodgers (2001) reminds that due to blade trimming there are also mechanical design concerns such as inducer blade thickness and vibrational frequency, the ratio of tip clearance height and axial thrust changes.

CONCLUSIONS AND OUTLOOK

A process to optimize the shroud contours of a 6 member trim family has been presented. The optimization was limited to the curved part of the impeller shroud contour which automatically implied that main and splitter blade tips were adjusted. The absolute tip clearance of the base design had to be maintained for all smaller trims.
All optimizations succeded in arriving at better efficiencies. A linear regression could be found that tells approximately how much loss in efficiency can be assumed for a trim family member compared to the base design. The difference between normalized efficiencies η^* for the optimized trims 7 and 2 (size ratio 1.21) is 0.94%. The difference between absolute efficiencies η is even smaller. The base design pressure ratio could be maintained for all trims in their best efficiency points. In some cases even the (numerical) surge margin could be improved.
The source of the stage efficiency improvement are better inflow conditions into the diffuser. The impeller efficiency was not influenced significantly by the change of the shroud contour – if the shroud contour was not a too bad guess. The diffuser performance however was improved, verified by higher pressure recovery and lower total pressure loss coefficients. The flow around the diffuser vanes is more attached than it is in the first guess geometry.
An optimization of the best efficiency point on the 100% speed line does not simultanously improve the entire speed line as well and can even worsen the efficiencies for an entire part load speed line. It depends on the field of compressor application whether this can be accepted or not.
A multi-point-optimization for 3 operating points on one speed line, as done by Hildebrandt et al. (2009), or even for 6 points distributed on the two speed lines of the presented case could be performed to obtain a shroud contour that improves definitely more than just one operating point.
The quality of a “first guess shroud contour” depends on the experience of the designer and on the time and effort spent to arrive at it.
It is recommended, not to rely on just one operating point improvement, but to compute entire speed lines with a shroud geometry optimized for just one operating point.
The compressor model itself was reduced to one pitch of the impeller and one pitch of the diffuser. An inlet duct systems, inducer casing bleed system and volute have not been modelled. Therefore it is of big interest to know the improvement which can be gained in comparative runs on the test facility. Such a test is planned for one of the trims to evaluate the optimization results.

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