Aerodynamic reconfiguration and multicriterial optimization of centrifugal compressors – a case study

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Abstract: *This paper continues the recent research of the author, with application to 3D computational fluid dynamics multicriterial optimization of turbomachinery parts. Computational Fluid Dynamics has been an ubicuous tool for compressor design for decades, helping the designers to test the aerodynamic parameters of their machines with great accuracy. Due to advances of multigrid methods and the improved robustness of structured solvers, CFD can nowadays be part of an optimization loop with artificial neural networks or evolutive algorithms. This paper presents a case study of an air centrifugal compressor rotor optimized using Numeca's Design 3D CFD suite. The turbulence model used for the database generation and the optimization stage is Spalart Allmaras. Results indicate a fairly quick convergence time per individual as well as a good convergence of the artificial neural network optimizer.*

Key Words: *RANS, CFD, Centrifugal compressor, optimization*

1. INTRODUCTION

This paper presents a case of centrifugal compressor off design point optimization. For practical applications it is often necessary to adapt the existing turbomachinery for different working conditions. In this case, the mass flow of the original compressor design point will be lowered to accommodate the new specifications. Typically this would be accomplished by scaling methods Refs [1-3].

However, the current design theme requires that the diameters and height of the rotor be kept constant (i.e. the new design rotor must fit without adaptations in the same machine as the old rotor). Also, because the compressor is powered by a synchronous electric motor, the rotational speed is also constant.

From classic turbomachinery calculations, the hub and tip incidence angles vary with the mass flow when maintaining the angular speed constant Ref [4]. Empirically there are optimal incidence angles which depend on the local Mach number Refs [5, 6] and therefore for optimal performance, a compressor rotor must correlate these parameters.

The method used relies on the optimization module of Numeca Design 3D which uses simulated annealing in order to estimate an optimal geometrical configuration, after analyzing a sufficiently large database. Because the method is iterative, it allows the artificial neural network to adapt itself by adding each optimization iteration to the database - which leads to convergence i.e. the CFD results confirm to a satisfactory degree the estimation of the artificial neural network Ref [7].

To summarize the design theme, the following points must be satisfied:

Minimal design modification,
Lower the mass flow from 4.4 kg/s to 4.0,
Maintain the pressure ratio of 1.8:1,
Maintain the wheel speed 14500 rpm,
Maintain the overall size of the rotor,
Maintain/Increase aerodynamic efficiency,
Maintain the 9 main + 9 splitter configuration due to aero-acoustic influences.

Since this is a mock-up design exercise, the so-called “existing design” had to be created. An in-house linear design method with experimental correlations was used. The geometry was then CFD tested in Numeca Fine/Turbo to confirm the aerodynamics.

Figure 1 depicts the solid mesh on the walls of the blades and the overall mesh quality parameters for the case above. The turbulence model used is Spalart-Allmaras with rotation and curvature corrections as presented in Ref. [8].

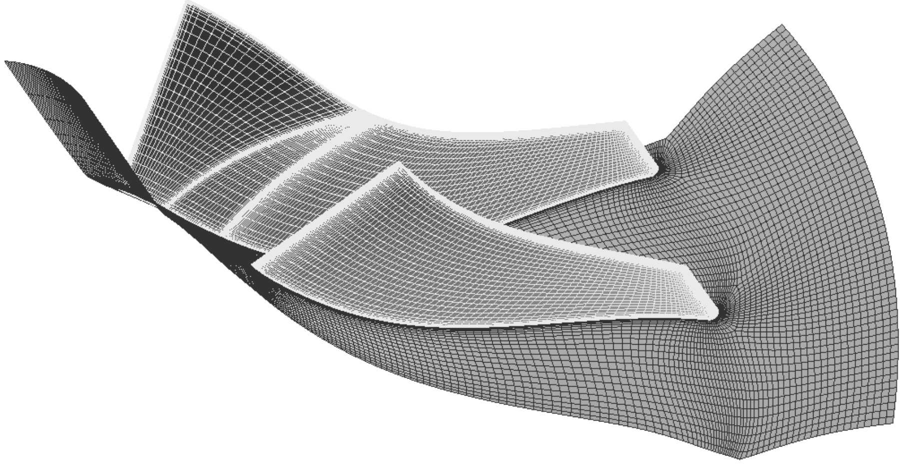


Fig.1 - Solid mesh near the walls of the rotor for a channel

The original rotor was obtained using linear design calculations and semi-empirical corrective correlations to account for the boundary layer blockage and flow retarding. The table below provides a comparison between the CFD results obtained with Numeca Fine Turbo and the linear design method.

Table 1 - Design data versus numerical simulation data (Original rotor at design point)

Parameter	Linear design with correlations	Numerical simulation
Total-to-total isentropic efficiency [%]	92.2	92.2
Outlet static pressure [Pa]	147830	156532
Outlet total pressure [Pa]	190680	185727
Outlet velocity magnitude [m/s]	288.03	248.473
Power consumption [kW]	248.68	242.495

The new design requires that the rotor diameters and height remain constant and that the mass flow is reduced from 4.4 kg/s to 4 kg/s. Since the entire machine is an industrial compressor, powered by an electric motor through a geared multiplier, the rotation speed will also be kept constant in order to save cost.

Off design point calculations were also carried out for the baseline rotor at the new operating point of 4 kg/s, Table 2 synthesizes the results.

Table 2 - Optimized vs Off Design Pont Original performances

	Optimized	O.D.P.	
T*_in	288.15	288.15	[K]
T*_out	345.969	348.344	[K]
p*_in	101353	101353	[Pa]
p*_out	186185	186912	[Pa]
Press Rat	1.83699545	1.844168	
Eff_isent	0.94569974	0.914732	[%]
Power	232.43238	241.9799	[kW]
Mass flow	4	4	[kg/s]

The efficiency decreases with the shift in mass flow. This is due, in part, to:

- Less than optimal angle of attack on the leading edge,
- Less than optimal flow coefficient and specific speed.

2. THE CFD OPTIMIZATION METHOD

In order to optimize the geometrical model we must first describe it using parameters that can be modified so that the database has enough individuals which are varied in a relevant way. The following steps are recommended when undergoing such an optimization procedure: Setting up a flexible yet correct blocking topology in order to use the *.res file from the baseline CFD simulation for all the other simulations. This is a particularly important aspect since the structured mesh considers also the cells as part of the topology (not just the blocks). It is therefore impossible to use an initial solution from a case with a non-matching topology. This step is usually done in the CFD testing of the baseline geometry. Fitting the baseline geometry is critical since it is literally a translation from an unformatted geometry to a model which AutoBlade can fully describe and interpret. The geometric model must be described in as few parameters as possible.

Parameterization is coupled with the fitting process although it serves a somewhat different purpose: choosing which parameters will be varied and how much the variation limits will be. The Screening is a full rehearsal of the CFD process, including the geometry generation, meshing, boundary condition settings, initialization, computational run and post processing. It is especially useful when using meshing IGG and post-processing CFView scripts. For the optimization method to be effective as well as efficient, a database of various geometries is created.

In this step, a pseudo-random geometry set is created and then tested using either the Fine/Turbo CFD solver or an external solver. Due to the structured meshing and solver, and to the geometric multi grid methods, the ~ 2 million element mesh case is solved in ~ 3hours on 6 threads (3 cores x 2).

In the actual optimization process, simulated annealing is used to make sense of the database and correlate the free parameters with the optimization function. This is also an iterative process since a typical database is only a coarse representation of the full space of available variables.

Therefore, the initial “guessed” optimal geometry will almost never have all the tested parameters matching (i.e. the CFD simulation will only partially confirm the estimated values of the simulated annealing).

However, after 20-30 iterations, convergence is obtained as well as the optimal geometry – according to the desired specifications.

The last step is to analyze the resulting geometry and its flowfield in order to insure that there are no numerical effects that adversely influence the optimization process. Also another reason is to try and make sense of the optimal design and try to learn from it in order to optimize the linear design code.

Using Autoblade we can import the exact geometry used in the previous simulation and describe it in a way that is suitable to the design philosophy of the designer. In our case one design requirement is that the blade has ruled surfaces and so we opted that the blade will only be described by the hub and shroud airfoils. A number of 85 parameters were used to describe the 3D geometry (5/airfoil, 7/hub, and 9/shroud). In total, 26 out of the 85 geometrical parameters were left to vary freely, including hub and shroud curves as well as main and splitter blade endwall airfoil camber.

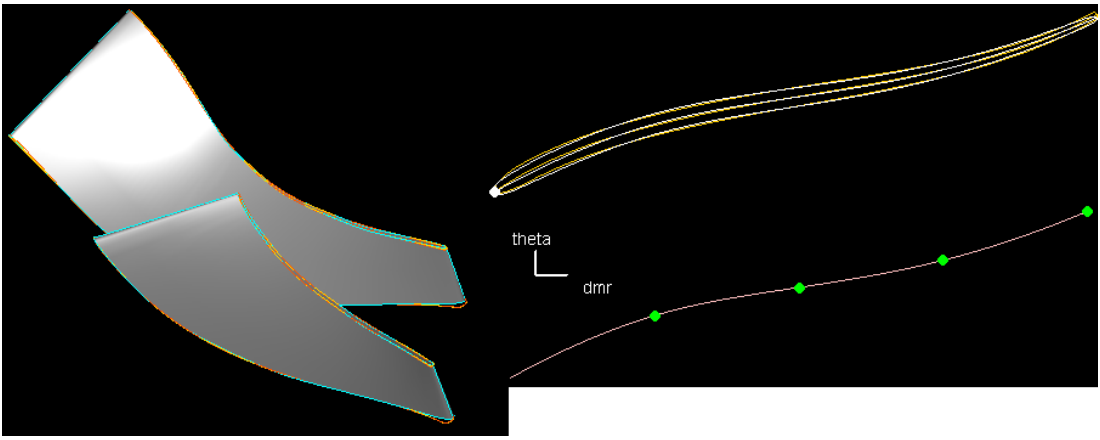


Fig. 2 - The fitted geometry overimposed on the baseline geometry

The database generation is a process in which a completely automated CFD simulation is carried out for a geometrical configuration - other than the original geometry. A rule of thumb for the number of individuals in the database required for the optimization stage is three times the number of free parameters.

In our case we opted for a database with 100 different geometrical variations - which satisfies and exceeds this empirical rule.

Since the optimization process uses simulated annealing for estimating the influence of each parameter and each parameter combination - in a holistic manner - the geometrical variations must be carefully generated. There are many methods to generate the database but one of the most robust is the Latin hypercube method Ref [9] - which is also the one used for this paper.

The optimization process begins with the database which is loaded and analyzed by the neural network (ANN).

A genetic algorithm is then used, based on the correlations estimated by the ANN, to predict an optimal geometry. The geometry is then built, meshed, simulated and post processed.

All relevant parameters are compared with the CFD results and then added to the existing database.

Reiterating the process insures that the convergence is likely to occur, since the new prediction will be near the previous ones.

The user dictates penalties on relevant parameters, also the weight of each parameter. Through the optimization process, the solver minimizes the penalties reaching a final geometry. Our case:

Massflow strictly equal to 4.0 kg/s,

Pressure ratio no less than 1.8:1.

Efficiency equal to 1 (note that this penalty will always be high so it will weigh in more than the others).

Figure 3 depicts two charts from the optimization process. The first is the whole objective function minimization vs the design loop index, which incorporates all the design restrictions imposed by the user. The second is the isentropic efficiency of the rotor vs the design loop index (i.e. optimization attempts).

Note that whereas in the first case we attempt to minimize the objective function, in the second case we wish to reach a maximum for the efficiency.

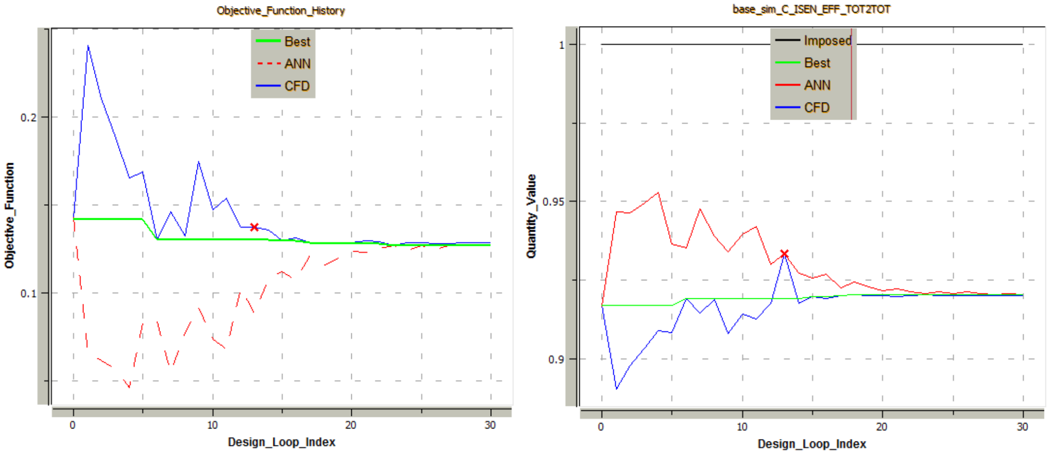


Fig. 3 – Objective function and isentropic efficiency vs. optimization attempt index

In the figure below, a relevant criterion for case verification is presented. The non-dimensional wall distance, y+ for both the baseline and the optimized geometry is presented. For the Spallart-Almaras viscosity model, the y+ value should be below one unit – as seen in Fig.4, this criterion is satisfied for both cases presented.

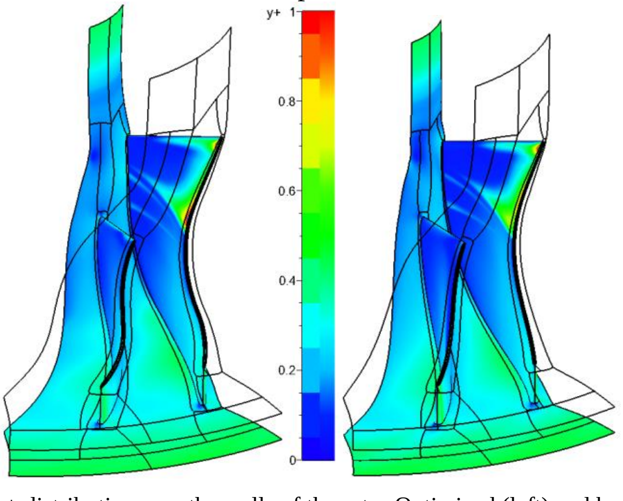


Fig.4 - The y+ distribution near the walls of the rotor Optimized (left) and baseline (right)

3. COMPARATIVE ANALYSIS

After reaching a final optimal design that satisfies our needs, we can start to analyze the various parameters which have changed in order to better understand the new design.

From Fig. 5 we observe that the backsweep angle of the blade is higher in the optimized geometry, which is particularly obvious for the shroud section. This would suggest a smaller amount of work transferred to the fluid. However, this is not the case, since the backsweep is actually negative just before the outlet section. Therefore, the optimization process leads to an individual which maximizes the work done on the fluid in the axial part of the rotor and minimizes the losses in the radial part of the rotor. Although there were no recirculation zones in the baseline rotor, it appears that high adverse pressure gradients are energy consuming and must be kept to a minimum. In fact, minimizing the pressure gradient absolute value on the tip of the blade leads to a more homogenous total pressure field as seen in Fig. 8.

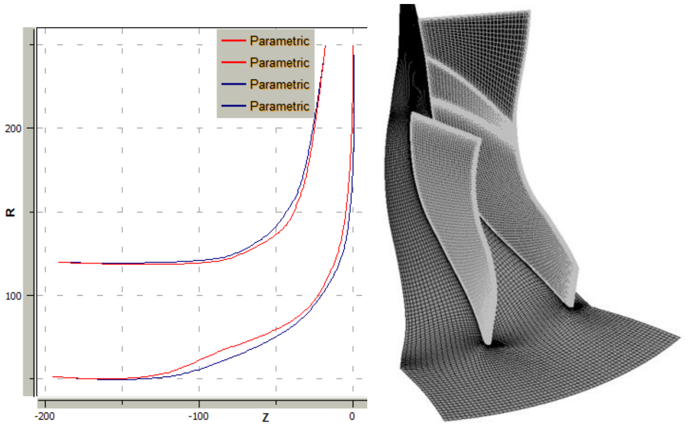
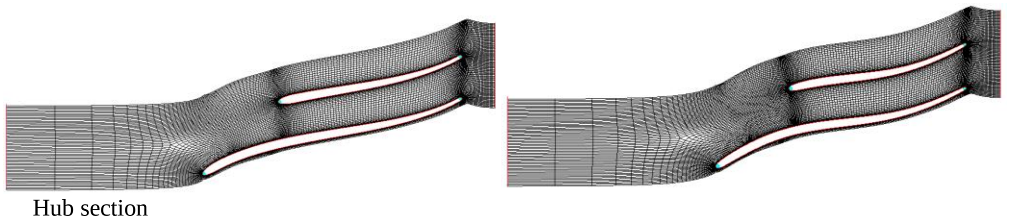
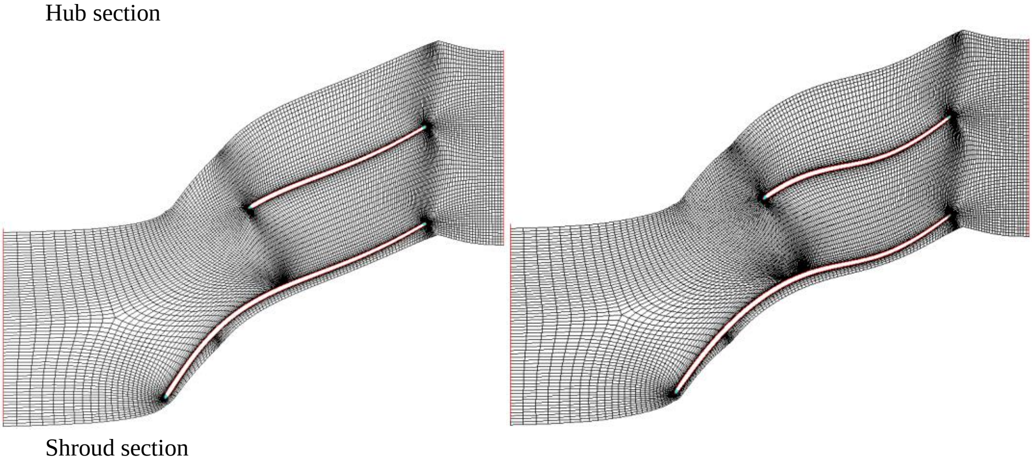


Fig. 5 – a. The optimal rotor geometry compared with the initial rotor



Hub section



Shroud section

Fig. 5 – b. The optimal rotor geometry compared with the initial rotor

Figure 6 presents a meridian section of the flow channel, the optimization version and the baseline rotor. It can clearly be seen that the optimized design diminishes the “hotspot” with the high Mach number, reducing the losses.

This is the direct result of the increased backsweep angle on the shroud airfoil profile. Having a smaller Mach number in that region also leads to a beneficial side effect, minimizing the potential recirculation counter-rotating bubble which may tend to form downstream in the case of vaneless diffusers such as the one studied here.

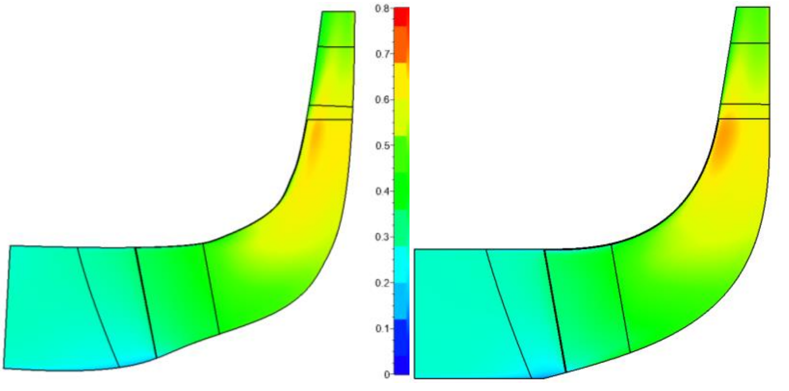


Fig. 6 - Absolute Mach number across the channel Optimized (left) and baseline (right) meridian plane

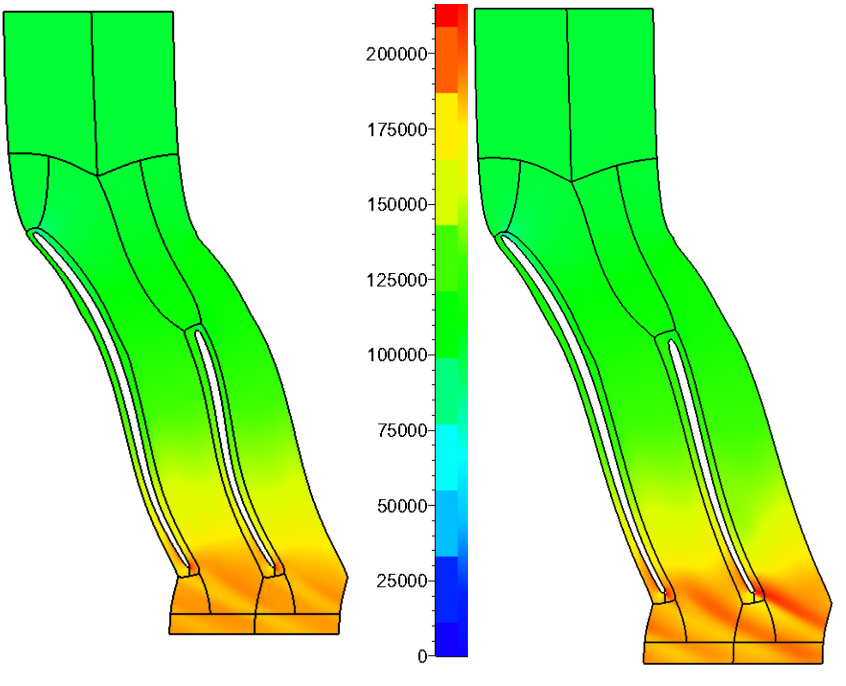


Fig. 7 - Absolute total pressure blade to blade Optimized (left) and baseline (right) midspan section

The real difference between the optimized and baseline rotor is in the total outlet temperature, which in the case of the optimized design is significantly lower, as seen in both Fig. 8 and in Table 2. This positively impacts the total to total isentropic efficiency of the machine. Both the total pressure and total temperature values have been mass flow integrated in order to provide the necessary information during the database generation and the optimization process.

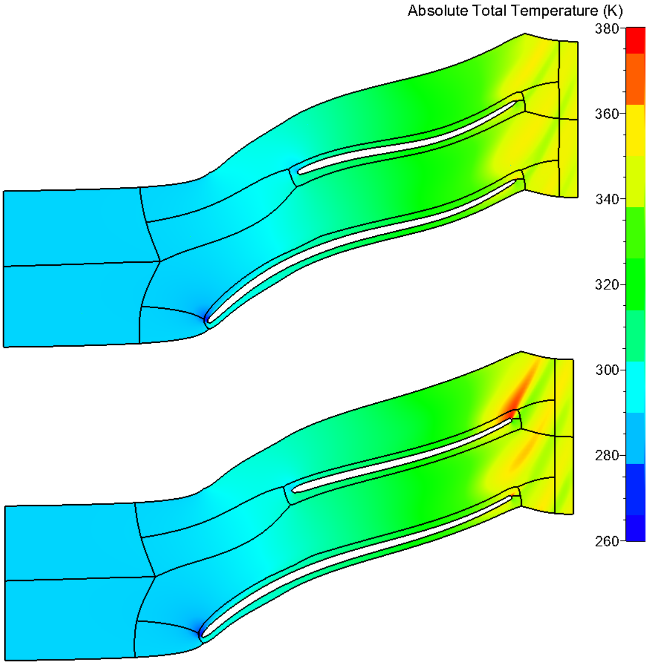


Fig. 8 - Absolute total temperature flow field of the optimized (top) and baseline (bottom) rotor in the midspan section (blade to blade view)

4. CONCLUSIONS

The paper presents a case study of a centrifugal compressor off-design point optimization. Computational fluid dynamics simulations were carried out to validate the linear design calculations of the design point.

Those simulations were in good agreement with the linear calculations for the rotor, providing a coherent baseline for the rotor. The isentropic efficiency of the rotor was estimated at 92.2% at the design point.

Off design point calculations were carried out at a mass flow of 4 kg/s instead of the initial 4.4 kg/s - obtaining, as expected, a lower isentropic efficiency of 91.5%.

After fitting the original geometry, a parametric model was elaborated using AutoBlade. The blades were described as ruled surfaces with only the hub and shroud airfoils which had the camber lines described by 5 point spline curves. The splitter was allowed to vary independently of the main blade in order to provide more degrees of freedom to the model. Furthermore, the hub and shroud curves were allowed to vary - without changing the outlet and inlet sections. In total there were 26 different independent parameters which were allowed to vary.

A database containing 100 individuals was generated, having variations on the original geometry, having individuals with isentropic efficiencies between 89.1% and 92.1%.

Based on this set, an optimization project was launched, having the following equally weighted penalties: mass flow = 4 kg/s, pressure ratio above 1.8:1, isentropic efficiency =1. After 30 design iterations, convergence was obtained on an individual having a pressure ratio of 1.837:1 and an isentropic efficiency of 94.57% was obtained for the 4 kg/s imposed conditions.

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