Aerodynamic development of a high-pressure ratio compressor for an advanced microturbine powerplant

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Abstract. Microturbine engines are increasing in popularity both for small poly-generation power plants and unmanned aerial vehicle; however, their overall efficiency is severely limited by their thermodynamic cycle and component efficiency. This compressor allows the real cycle of the engine to operate with lower specific fuel consumption, extending the range of the vehicle and lowering pollutant emissions. Optimization methods were implemented in the compressor design, in order to increase the aerodynamic efficiency. A 4.5:1 pressure ratio was selected in order to keep the exhaust jet subsonic. Linear models were used to optimize the speed and gross geometry while the TEIS and later fully viscous 3D models were used successively to optimize the end walls and blade shapes. For all numerical simulations, the Menter SST turbulence model was used. In order to minimize modelling errors, the rotor-stator interface was placed as far as possible downstream of the impeller.

1. Introduction
In the design of a turbomachinery an essential step is the dimensioning phase, having a major impact on its overall performance. Progress due to the applicability and accuracy of CFD programs, does not ensure improper preliminary design improvement; but together they can complement in an efficient way ensuring an optimal design process. The main trend in the development of the centrifugal compressor is the development of compact products, with a low weight, smaller frontal area and reduced cost. Nevertheless, these requirements are in contradiction with the needs for high efficiency and pressure ratios, but also with the limitations imposed by the mechanical properties of the materials used. Thus, an appropriate trade-off must be found to balance all requirements and reach a feasible specification. In the preliminary design phase an empirical approach can be applied, which correlates efficiency measurements, performed on similar turbomachines, with global parameters, such as specific speed or flow coefficient, size and tip clearance variation. Two known correlations of this type are those given by Casey and Marty [1] and those of Rodgers [2]. The first correlation offers an estimate for the total polytropic efficiency on the stage; and the second, the total isentropic efficiency of the rotor having high relative Mach numbers. Casey and Robinson [3] combined these two correlations into a more general form (where \( \phi_{1} \) - flow coefficient and \( M_{u2} \) - relative Mach number):

\[
\eta_{p} = f(\phi_{1}, M_{u2})
\] (1)
taking into account, also CFD methods for turbomachinery design.

Figure 1 presents the correlation between the flow coefficient, polytropic efficiency and relative Mach number, highlighting that the influence of the relative Mach number on efficiency is less than the effect given by the pressure ratio.

![Figure 1. Correlation between efficiency, flow rate coefficient and Mach number in rotor [4].](image)

In the 1950s, Cordier [5] conducted an empirical analysis for various turbomachines based on experimental data. The aim was to correlate specific diameter, specific velocity and efficiency to form dimensionless parameters. If for a specified diameter it is necessary to determine the operating point (mass flow and static pressure), using Cordier diagram can be determined the rotational speed necessary to reach that point, ensuring higher efficiency. Even though it is widely used in industry, does not provide any information about the blade type, angle, height, shape of the hub and the casing, etc. Thus, its determination is based on analytical results presented in literature and the designer experience.

Over the years, various methods for designing a centrifugal compressor stage were developed, some of them based on geometric parameters (number of blades, passage geometry, vaneless space, blade angles etc.) [6, 7], aerodynamic design and design parameters limitations [8-10], interaction between the impeller and diffuser [11, 12] and other methods that offer details about the overall performances, like CFD [13-18].

Rusch [19] developed a methodology for the preliminary assessment of a high flow capacity centrifugal compressor, showing that impeller geometry and material has a major impact on the maximum pressure ratio that can be achieved; also, proving that design flow coefficient decreases with total pressure ratio due to inlet casing relative Mach number. Kurauchi [20] presents a method for designing centrifugal compressors in three steps: the first one establish 1D preliminary design using empirical data, second step involves fluid flow study in the meridional plane and the last step representing CFD simulations. Li [21] developed an optimized design method for centrifugal
compressors based on one-dimensional calculations and improved the interaction between impeller and diffuser, obtaining an increase in efficiency by near 2%.

Another model was developed by Japikse [22] – TEIS model, having as purpose to discover more similarities between impeller flow and performance of conventional diffusers and nozzles. In TEIS model, the channels of an impeller are separated into two elements in series, conceptually modeled as diffusers and nozzles, called element "a" and "b". Based on accurate inputs, reliable predictions of a turbomachinery can be made using TEIS model, assuring in this way the 1D model and the physics of the flow.

The aim of this paper is to develop a compact high-pressure ratio compressor for a medium size, 80-daN micro gas turbine engine, improving the aerodynamic efficiency through optimization of the end walls and blade shapes.

2. Design development and case setup
Optimization methods were used at all levels of the design process, maximizing the impact on the thermodynamic cycle and the aerodynamic efficiency of the component. A 4.5:1 pressure ratio was selected in order to keep the exhaust jet subsonic.

2.1 Design development
Linear models were used to optimize the speed and gross geometry while the TEIS and later fully viscous 3D models were used successively to optimize the end walls and blade shapes. Correlations were obtained at every step of the design process, revealing the relevance of each free parameter in the overall penalty function. The findings from the 1D and 2D optimization studies helped with the parameterization effort for the more labor-intensive 3D studies. That is, only the critical free parameters were allowed to vary.

The 1D design of the impeller was generated using Vista CCD, followed by Direct Optimization of the interest parameters. The set of parameters selected for optimization are introduced through the Screening procedure; procedure that allows the user to generate a new set of samples and sort the entries based on the predetermined objectives and constraints. The method is based on sampling and sorting; supporting multiple objectives and constraints, but also the input parameters.

Two main methods can be used to determine the correlation between two variables:
- Pearson: parametric correlation;
- Spearman: non-parametric correlation.

Figure 2 presents the correlations between design of the impeller, parameterization and optimization.

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**Figure 2.** The process of design, parameterization and optimization the impeller.
The Pearson correlation method is used as a primary verification of the relationship between two variables. A Pearson correlation is a number between -1 and 1 which indicates the extent to which two variables are linearly linked. A Spearman rank correlation is a number between -1 and +1 which indicates to what extent two variables are monotonically related. A Spearman correlation is simply a Pearson correlation calculated on ranks instead of values or categories of data.

Figure 3 presents the Pearson correlation between 14 parameters considered important for impeller optimization. The main diagonal has the value 1, representing the correlation of the parameter with itself; thus, the important correlations are those with values as close to 1 as possible. According to the data presented in figure 3, the most relevant correlations are: a) P1 - P14 (rotational speed - isentropic efficiency), b) P8 - P12 (β angle - static pressure). In addition to the positive linearly correlated variables, there are also linearly negative correlated variables, such as: P3 - P13 (diffusion degree - absolute flow angle) and P1 - P13 (rotational speed - absolute flow angle), P1 - P10 (rotational speed - power), P1 - P11 (rotational speed - pressure ratio). There are also negative correlations, but a negative correlation suggests that the two variables are moving in opposite directions.

Figure 3. Linear Correlation Matrix – Pearson.

Where the main parameters are: P1 – rotational speed; P2 – meridional velocity gradient; P3 – diffusion degree; P4 – hub diameter at inlet; P6 – main number of blades; P7 – secondary number of blades (splitter); P8 - β₂ angle at discharge; P9 – θ angle, P10 – power; P11 – pressure ratio; P12 – static pressure; P13 – absolute flow angle; P14 – isentropic efficiency.

Figure 4 presents Spearman, quadratic determination matrix, where five important correlations are development: a) P1 - P10 (rotational speed - power), b) P1 - P11 (rotational speed – pressure ratio), c) P1 - P14 (rotational speed - isentropic efficiency), d) P3 - P13 (diffusion degree - absolute flow angle), e) P1 - P13 (rotational speed - absolute flow angle). Following the established correlations, it has been found that in the optimization of a radial impeller an important part is given by the connection between the rotational speed and the isentropic efficiency. In addition, the connection between the β₂ angle and the static pressure should not be underestimated.

Figure 4. Linear Correlation Matrix – Spearman.
In defining the parameterized model of the impeller, parametric curves of the hub and shroud are coupled with beta distribution, leading to blade loading by variation of β angle from leading to trailing edge. After establishing the parameterized model, supposing - defining β and θ angles - at hub and shroud - as the input parameters in the optimization process, the blades geometry is checked, in order to evaluate: Mach number and velocity distribution in the hub area, medium section and shroud, positions of the secondary blades (splitter); incidence angle, operation range etc.

The main purpose is to increase efficiency and to find a suitable pressure ratio, targeting the value of 7.1. For the case studied, 700 samples were generated; table 1 presents the candidates who best fulfil the imposed objectives - resulting in three candidates. Of the three candidates, the most suitable choice for the impeller application is selected, being: a pressure ratio of 7.1002 and 0.8551 efficiency. The obtained data will be implemented as input variables for sizing the new geometry.

### Table 1. Optimal candidates corresponding to the imposed objectives.

|                | C1     | C2     | C3     |
|----------------|--------|--------|--------|
| Total isentropic efficiency | 0.8552 | 0.8551 | 0.8551 |
| Total pressure ratio         | 7.0981 | 7.1002 | 7.1003 |

#### 2.2 Case setup

Numerical simulations were performed using Numeca Fine/Turbo package. The structured grid resulted following the numerical discretization is presented in figure 5, were can be observed the mesh structure for the leading and trailing edge and the inflation near blades. In addition, the structured grid was determined in such a way as to insure $y^+$ values below one unit across the aerodynamic surfaces while maintaining a growth ratio of 1.1:1, figure 6.

![Figure 5. Computational grid for trailing and leading edge.](image)

![Figure 6. Distribution of $y^+$ number in compressor.](image)
In order to minimize modelling errors, the rotor-stator interface was placed as far as possible downstream of the impeller. It also depicts the rotor-stator interface placement as well as the limits of the rotating domain considered. This limiting is possible without the use of an interface because the equations associated to the rotation can be limited to a selected cone inside the rotor mesh itself. For all numerical simulations, the Menter SST turbulence model was used. The model provides good prediction of the flow and accurate results, capturing phenomena from the entire domain and near walls [23].

As main boundary conditions, the following where imposed: *Inlet*: total pressure (101325 Pa), total temperature (288 K); *Outlet*: mass flow (1.4 kg/s); *No slip wall*: blades, hub/shroud walls.

### 3. Results and discussions

In defining the compressor characteristic, different operating conditions were analyzed; the most important ones (rotational speed, mass flow) are presented in table 2. The table also reveals other defining parameters in choosing the operation point of the compressor.

Based on the results it was found that the optimum working point is at 45000 rpm and a mass flow of 1.4 kg/s.

| Rotational speed | Mass flow | Total Pressure [Pa] | Relative velocity [m/s] | Atan (Wt/Wm) [°] | Total Temperature [K] |
|------------------|-----------|----------------------|------------------------|-----------------|----------------------|
| 15000 rpm        | 0.2 kg/s  | 124575               | 54.5689                | -28.3182        | 311.378              |
|                  | 0.25 kg/s | 124308               | 53.5739                | -24.4884        | 310.51               |
|                  | 0.3 kg/s  | 122156               | 71.3948                | -28.4481        | 309.51               |
| 30000 rpm        | 0.5 kg/s  | 214122               | 109.966                | -15.2979        | 379.94               |
|                  | 0.6 kg/s  | 212362               | 113.071                | -17.6703        | 376.041              |
|                  | 0.7 kg/s  | 219220               | 112.336                | -17.3511        | 373.928              |
|                  | 0.75 kg/s | 209395               | 157.613                | -24.2184        | 372.34               |
| 35000 rpm        | 0.8 kg/s  | 276186               | 129.703                | -19.2968        | 407.048              |
|                  | 0.9 kg/s  | 281470               | 117.508                | -20.0395        | 404.797              |
|                  | 1 kg/s    | 273071               | 131.234                | -21.7132        | 403.135              |
|                  | 0.9 kg/s  | 346286               | 148.543                | -18.9386        | 447.832              |
|                  | 1 kg/s    | 358798               | 255.224                | -14.0070        | 444.138              |
|                  | 1.1 kg/s  | 364000               | 144.125                | -18.7195        | 442.138              |
|                  | 1.2 kg/s  | 359120               | 143.906                | -22.3848        | 438.544              |
|                  | 1.25 kg/s | 350563               | 152.093                | -20.1991        | 438.039              |
| 40000 rpm        | 1.15 kg/s | 458151               | 168.458                | -18.6905        | 488.72               |
|                  | 1.2 kg/s  | 471120               | 166.212                | -18.1492        | 487.614              |
|                  | 1.3 kg/s  | 477084               | 158.373                | -17.4558        | 484.473              |
|                  | 1.4 kg/s  | 476201               | 163.68                 | -17.8228        | 481.452              |
| 45000 rpm        | 1.4 kg/s  | 476201               | 163.68                 | -17.8228        | 481.452              |
|                  | 1.5 kg/s  | 458909               | 177.222                | -18.2638        | 478.084              |
|                  | 1.53 kg/s | 445425               | 187.197                | -19.1053        | 476.568              |

Following the numerical simulations, compressor characteristic was defined, being represented by a set of functions, like:

\[
\pi^* = \pi^*(n, M_s) \tag{2}
\]

\[
\eta^*_{lcon} = \eta^*_{lcon}(n, M_s) \tag{3}
\]
Where: \( p_0^* \) and \( T_0^* \) - total pressure and temperature for atmospheric conditions, \( n \) - rotational speed and 
\( M_a \) - mass flow. The relation (2) representing the compression characteristic, and the relation (3) the efficiency characteristic.

Figure 7 presents the compression characteristic, emphasizing that the pressure ratio is influenced by rotational speed, increasing proportionally. For rotational speed of 45000 rpm, six mass flows between 1.15 - 1.53 kg/s were analysed, resulting a mass flow of 1.4 kg/s for best overall performances. At the same time, for this rotational speed there is also a minimum flow rate, under whose value flow in the compressor becomes unstable.

The relations used to determine isentropic and polytropic efficiency are as follows:

\[
\eta_{iscn} = \frac{\frac{p_2^*}{p_1^*}^{\gamma - 1}}{\frac{T_2^*}{T_1^*} - 1} \quad (4)
\]

\[
\eta_p = \ln \left( \frac{T_2}{T_1} \right) \ln \left( \frac{p_2}{p_1} \right)^{\gamma - 1} \quad (5)
\]

Through these relationships, were defined the two characteristics, for isentropic and polytropic efficiency, see figure 8. At 45000 rpm and a mass flow of 1.4 kg/s, the highest isentropic efficiency was 0.8282 for the entire stage, and a polytropic efficiency of 0.861. The differences between the two efficiencies, figure 8(c), occur since the polytropic efficiency does not take into account the thermodynamic effects, considering only the aerodynamic performance of the compressor.

Between velocity and pressure ratio is a direct connection, thus increasing of one quantity leads to increase of the other; leading implicitly to differences between the two efficiencies and to the secondary flows, large Mach number, flow detachment etc.
Figure 8. Efficiency curves for different rotational speeds: (a) Isentropic efficiency; (b) Polytropic efficiency, (c) Isentropic efficiency vs. Polytropic efficiency.

Figure 9 shows the assembly of the gas-dynamic passage, from the compressor inlet to the turbine outlet. It also depicts the rotor-stator interface placement as well as the limits of the rotating domain considered, figure 10.

Figure 9. The centrifugal compressor integration with the other major components of the engine.

Figure 10. Local Mach number on the hub— the limitation of the rotating equation within the impeller mesh.

Figure 11 presents total pressure variation in the compressor. Most of the total pressure losses take place in the vaneless region between the two components, but also in the diffuser. At impeller discharge, the local flow speed is high, from the effects of curvature of the streamlines, surpassing Mach 1 in the region, figure 12.
Moreover, it can be observed that the tip area of the blades is predisposed to occurrences of attached whirlpools that generate entropy, leading to flow instabilities in the rotor, figure 13. Another effect of is the efficiency decrease.

Figure 14 presents Mach number variation at three different heights of the blade: (a) 10%, (b) 50% and (c) 90%. Due to high velocity and blade curvature, towards the blade trailing edge, Mach number value exceeds 1 (figure 14(a) and (c)). As the fluid flow approaches the impeller discharge, Mach number value drops to near 0.37.
4. Conclusions

It is the goal of this paper to present a compact high pressure ratio compressor for a medium size, 80 daN micro gas turbine engine. Optimization methods were used at all levels of the design process, maximizing the impact on the thermodynamic cycle and the aerodynamic efficiency of the component. Different rotational speeds and mass flows were analysed in order to determine the operating range and operating point of the compressor, capable of assuring higher overall performances. All numerical simulations were performed using Numeca software, and SST turbulence model. Following the results of the numerical analysis, a rotational speed of 45000 rpm was chosen, and a mass flow of 1.4 kg/s. The pressure ratio was selected in order to keep the exhaust jet subsonic. This compressor allows the real cycle of the engine to operate with lower specific fuel consumption, extending the rage of the vehicle and lowering pollutant emissions.

5. References

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