ONE-DIMENSIONAL PRELIMINARY COMPRESSOR DESIGN APPLIED TO A 500 kW GAS TURBINE AND THE EFFECT OF BLADE NUMBER

André Perpignan V. de Campos, andre.perpignan.campos@usp.br
Guenther C. Krieger Filho, guenther@usp.br
Universidade de São Paulo, Departamento de Engenharia Mecânica, Laboratório de Engenharia Térmica e Ambiental, 05508-900, Av. Professor Mello Moraes, 2231 – São Paulo – SP – Brasil

Abstract. *This work presents the thermodynamic cycle calculation for a 500 kW gas turbine and its first design choices to introduce the preliminary compressor design. The radial type compressor selection through charts involving the similitude principle is described. A one-dimensional impeller design procedure based on basic fluid mechanics and thermodynamics, as well as on empirical correlations, was implemented to provide the impeller main dimensions. As available correlations for the impeller number of blades are not conclusive, CFD was used to evaluate the performance and efficiency of the impeller with the designed blade in different numbers. Using the CFX solver with the RNG k-ε turbulence model, the design point was reproduced for the various blade numbers. Simulations were able to capture the increase and subsequent decrease in efficiency while the blade number was increased, characterizing an optimum point among friction and slip losses. In the range of blade number from 10 to 30, efficiency varies almost 10%, proving that a wise blade number selection is crucial for the whole gas turbine performance.*

Keywords*: impeller design, centrifugal compressor, empirical correlations*

1. INTRODUCTION

Gas turbines are very reliable used as electrical power generators. Besides, they are an attractive option in places where energy distribution costs are high, since the onsite energy generation is made possible. Considering the onsite generation niche, the Laboratory of Environmental and Thermal Engineering of the University of São Paulo intends to develop a small 500 kW gas turbine in order to extend the knowledge on gas turbines design, computational simulation and constructive features.

The actual work presents the first steps of the thermodynamic cycle selection and the compressor preliminary design. Although these design phases do not have high calculation complexity, choices made in these phases have a fundamental role on the design system performance (Saravanamuttoo, 2001).

Compressor type selection based on the similitude principle is also described. The radial type compressor was selected. Even with the increasing use of CFD (Computational Fluid Dynamics) in turbomachinery design, one-dimensional calculations based specially on empirical correlations are still appealing in first dimensioning because of their much faster results. A one-dimensional design procedure based mainly on the work by Whitfield and Baines (1990) was used to define the impeller main dimensions.

The number of blades employed in the impeller is an important parameter that affects the whole performance and efficiency. Impeller slip and blockage characteristics are dependent of the blade number as well (Dixon and Hall, 2010). Although there are some correlations intended to deliver the optimal blade number. They are based on different variables and their results must be further evaluated in order to guarantee the best configuration. To evaluate the trend of change in performance with the change in the number of blades, CFD was used in the impeller preliminarily designed.

2. INITIAL CYCLE CALCULATION

Before applying any compressor design method it was necessary to establish some requirements and parameters to serve as input to the design. Bearing in mind the 500 kW power output, a comparative research with commercial gas turbine systems was done. This research used data given by manufacturers added to a database collected by Giampaolo (2006). Based on the results, the compressor pressure ratio requirement was fixed as 4. Figure 1 shows the data points of pressure ratio in function of the output power. Despite the fact that the distribution may suggest the use of higher pressure ratios, the intention was to impose a comfortable requirement, even with no previous design experience.

To proceed with the simple thermodynamic cycle calculation, more assumptions had to be made. Turbine inlet temperature T_{03} was estimated to be 1150 K, according to Han et al. (2000) for turbines with no cooling system. The power turbine configuration, as shown in Fig. 2, was chosen for this system. The advantage is to avoid rotation restrictions in the compressor by not connecting it to the load. The thermodynamic procedure recommended by Saravanamuttoo et al. (2001) was used, taking into account estimated compressor and turbine isentropic efficiencies (η_c and η_t), combustion and mechanical efficiencies (η_{comb} and η_m), as well as combustion chamber and exhaust pressure losses ($\%L_c$ and L_e) into the adapted Brayton cycle.

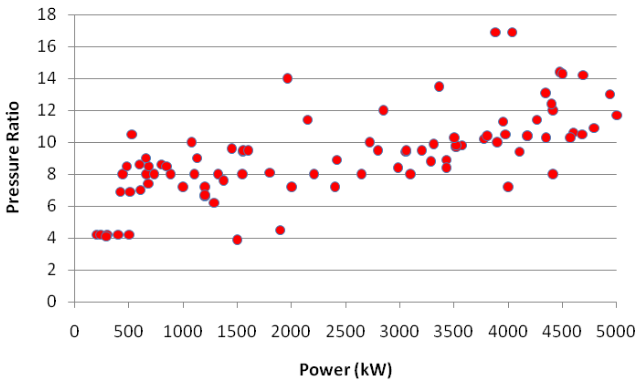

Figure 1. Pressure ratio of commercial gas turbines collected data in function of power output.

Ambient temperature T_{01} and pressure p_1 are, respectively, 300 K and 1 bar. Specific heat at constant pressure of incoming air c_P was assumed to be 1.005 kJ/(K.kg), and c_P^b is 1.148 kJ/(K.kg) for the burnt gases. Specific heat ratios were adopted as 1.4 and 1.333, for air (γ) and burnt gases (γ_b) respectively.

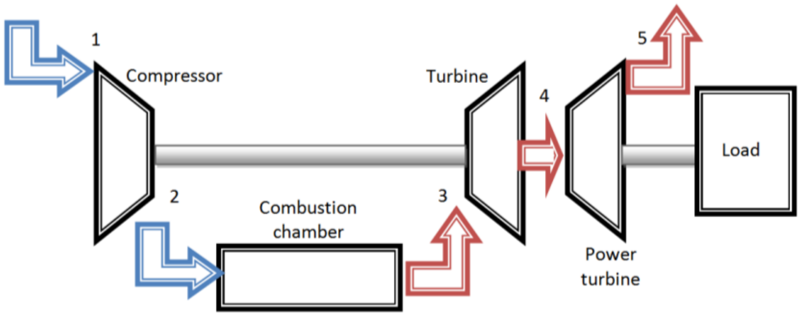

Figure 2. Selected cycle diagram with the power turbine configuration.

With this data, it is possible to estimate the required mass flow that, with a pressure ratio P_R of 4 is able to generate the required power output of 500 kW. Equations 1 to 9 show this calculation, where the subscripts are related to the positions in Fig. 2 and W_c is the compressor specific work and W_{pt} is the power turbine specific work. Both compressor and turbine isentropic efficiencies were assumed to be 85%, while shafts mechanical efficiency was 99%, following Saravanamuttoo et al. (2001) recommendations.

$$T_{02} = \frac{T_{01}}{\eta_c}\left[P_R^{(\gamma-1)/\gamma} - 1\right] + T_{01} \tag{1}$$

$$W_c = \frac{c_P(T_{02} - T_{01})}{\eta_{comb}} \tag{2}$$

$$T_{04} = -\frac{W_c}{c_P^b} \tag{3}$$

$$T_{03} - T_{04} = \eta_t T_{03}\left[1 - \left(\frac{1}{p_{03}/p_{04}}\right)^{(\gamma_b-1)/\gamma_b}\right] \tag{4}$$

$$p_{04} = P_R(p_{01} - \%L_c \cdot p_{01})\frac{p_{04}}{p_{03}} \tag{5}$$

$$p_{05} = p_{01} + L_e \tag{6}$$

$$T_{05} = T_{04} - \eta_t T_{04}\left[1 - \left(\frac{1}{p_{04}/p_{05}}\right)^{(\gamma_b-1)/\gamma_b}\right] \tag{7}$$

$$W_{pt} = c_P^b(T_{04} - T_{05})\eta_m \tag{8}$$

$$\dot{m} = \frac{500\ kW}{W_{pt}} \tag{9}$$

3. COMPRESSOR TYPE SELECTION

After the first global calculations it was possible to carry on to the compressor design. The decision of staring with the compressor relies on the fact that its design is considered more complex and vital than the other gas turbine components (Wilson and Korakianitis, 1998). One of the most important decisions on this design is the selection of the flow type, if it will be radial, axial or mixed. There is not a closed method for this definition and it relies on many issues, such as the manufacturer experience, area limitations and estimated costs (Cumpsty, 2004). The most widely accepted method was developed by Cordier in the 1950's and more explored by Balje (1981) that gathered data of the most efficient turbomachines and plotted their specific speed (N_s) against the specific diameter (D_s). As a result, the data points formed a shape close to be a line, called the Cordier line.

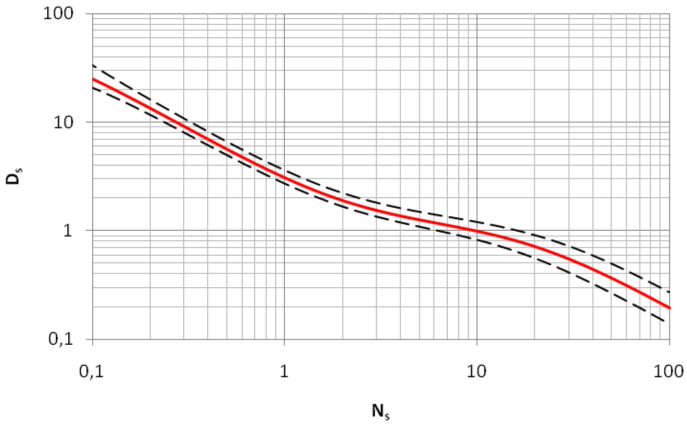

Figure 3. The Cordier line developed by Casey et al. (2010).

Balje (1981) noted that machines of the same type gathered in specific regions of the line and was even able to create efficiency contours. Approximately, machines with N_s smaller than 1 tend to have radial compressors, between 1 and 2 to have mixed and higher than 2, axial. This approach was used to define the machine type, with the highest efficiency available in the contours. The radial flow compressor was chosen.

More recently, Casey et al. (2010) proposed a mathematical formulation of the Cordier line, using more recent database. The equation's parameters have bands of values, resulting on a band in the $N_s D_s$ graph, not an exact line, as seen on Fig. 3. This equation was used to have a first estimate of the machine diameter, since it tends to be more reliable than older developments. Specific speed is defined by Eq. 10 and specific diameter by Eq. 11, where N is the rotation, Q is the volumetric flow rate, g is gravitational acceleration and H is the head.

$$N_s = \frac{N\sqrt{Q}}{(gH)^{3/4}} \tag{10}$$

$$D_s = \frac{D(gH)^{1/4}}{\sqrt{Q}} \tag{11}$$

4. IMPELLER ONE-DIMENSIONAL PRELIMINARY DESIGN

The preliminary design is intended to define or at least have a good estimate of the impeller main dimensions. The procedure used in this computation was proposed and described by Whitfield and Baines (1990). Basic empirical correlations, thermodynamic principles, velocity triangles and non-dimensional parameters are used. Additionally, correlations provided by Came and Robinson (1999) and Dufour (2006) were included.

The design system controls theoretical absolute and relative Mach number throughout the impeller in order to avoid sonic velocities. The slip factor is also a controlled variable and its initial estimate is reviewed with empirical correlations.

Table 1 summarizes the inputs and outputs used. The slip factor is assumed at first and is recalculated with a correlation, so that the whole calculus is redone until slip factor converges. Various blade angles, hub to shroud radius ratio and inlet shroud to discharge tip radius ratio were used and evaluated with abacuses presented by Whitfield and Baines (1990).

Table 1. One-dimensional impeller design parameters

Inputs	Value	Units	Outputs	Value	Units
Initial slip factor	0.85	-	Inlet hub radius	39.2	mm
Impeller total to total efficiency	85%	-	Inlet shroud radius	98.0	mm
Stage total to total efficiency	80%	-	Outlet radius	196.1	mm
Inducer shroud relative flow angle	-55	degrees	Outlet blade height	10.8	mm
Impeller discharge blade angle	-40	degrees	Axial length	117.4	mm
Absolute flow angle at discharge	65	degrees	Rotation	16600	rpm
Inlet stagnation temperature	300	K			
Inlet stagnation pressure	1	bar			
Mass flow	4.13	kg/s			
Pressure ratio	4	-			

5. BLADE NUMBER EFFECT

The number of blades used in the compressor impeller is a very important feature. The impeller efficiency is clearly affected. The use of fewer blades tends to increase losses due to turbulence, as well as increasing slip, while a larger number of blades is likely to increase friction losses (Dufour, 2006). However, the influence of each loss type depends on many factors, such as the operational speed, flow coefficient and blade angles. Some attempts of generalizing the optimal blade number were developed and are shown in section 5.1.

To fully define the impeller geometry, the meridional contour, camber line, thickness distribution and number of blades are necessary (Verstraete, 2010). The one-dimensional procedure defined only the main dimensions of the meridional contour. The definition of the curved lines on the meridional contour, the camber line and thickness distribution, necessary to evaluate the blade number effect, was done following guidelines provided by Cumpsty (2004), Dixon and Hall (2010) and Whitfield and Baines (1990). The CFD simulations were also intended to assist the evaluation of these choices, which will be refined throughout the design process.

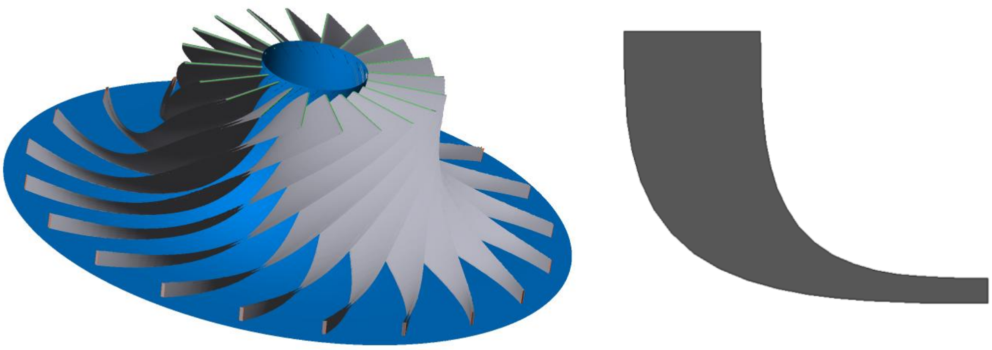

Figure 4. Designed impeller for the 20 blade case (left) and the blade meridional profile (right).

5.1 Empirical correlations

Literature reports some empirical correlations intended to determine the optimum number of impeller blades Z. Came and Robinson (1999) based their relation, presented by Eq. 12, on the space-chord ratio ψ, the impeller outlet radius r_2 and the impeller axial length L. The authors reported that ψ is usually 0.17 for a pressure ratio of 6 and varies from 0.25 to 0.35 as the pressure ratio goes from 3 to 2. Based on these values, ψ was estimated to be around 0.22 for the design pressure ratio 4.

Rodgers (2000) presented a relation based on the specific speed and the impeller discharge blade angle β_{B2}, shown in Eq. 13. Xu and Amano (2012) related the blade number to the pressure ratio of the centrifugal compressors using a graph with various designs values. There is significant dispersion in the data, as shown on Fig. 4, and the linear relation between the variables is questionable.

$$Z = \frac{\pi r_2}{\psi L} \tag{12}$$

$$Z = 25 \cdot \frac{cos(\beta_2)}{N_s} \tag{13}$$

$$Z = 2\pi \cdot tan(\alpha_2) \tag{14}$$

$$Z = \frac{\pi(110 - \alpha_2)}{30} \cdot tan(\alpha_2) \tag{15}$$

Anyhow, the three correlations gave very similar results for the designed impeller and its operational conditions. Even with this similarity, Came and Robinson (1999) and Rodgers (2000) stressed that it is necessary to evaluate the best blade number for each case, once there are design particularities that may escape from the .

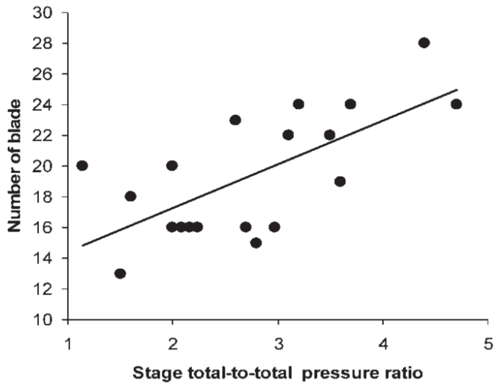

Figure 5. Relation between the blade number and pressure ratio for some impeller designs. Source: Xu and Amano (2012).

Additionally, both Glassman (1976) and Jamieson (1955) have developed relations for minimum blade number in radial turbines rotors that use only the absolute flow angle at discharge α_2 as input parameter, presented by Eq. 14 and Eq. 15. These relations might be used as a reference for radial compressors, bearing in mind that there are usually less blades in turbines than in impellers for the same pressure difference. Table 2 presents the results of the correlations for the derived geometry and operational conditions.

Table 2. Optimal (or minimal) blade number correlations

Empirical correlation	Blade Number *Z*
Came and Robinson (1999)	23/24
Glassman (1976)	11
Jamieson (1955) – minimum	14
Rodgers (2000)	22
Xu and Amano (2012)	22/23

5.2 CFD simulations

As the impeller main geometry and blade angles were defined with the one-dimensional design procedure, blade geometry was digitalized following those parameters as Fig. 4 illustrates. Only the impeller was simulated, not having the diffuser. Meshes used for the all blade number configurations were generated using the *TurboGrid* software. Blade geometry was kept while the blade number was varied.

All mesh elements used are hexahedral, and meshes had at least one hundred thousand elements. The *RNG k-ε* was the chosen turbulence model, which was shown to provide good results for centrifugal compressors (Moura, 2008). Boundary conditions imposed total pressure in domain entrance, mass flow rate in the exit, rotation specified in Tab. 1 and shroud tip clearance. These boundary conditions are a recommendation of the software manufacturer, as well as from Moura (2008) and Souza (2011), regarding the fact that these boundary conditions let the pressure ratio as a free parameter.

The *Ansys CFX* code was used in the simulations. Iterations were made until residuals RMS were below 10^{-5}. The main values controlled were the impeller isentropic efficiency and its pressure ratio. At first, five blade numbers were simulated: 10, 15, 20, 25 and 30. Results analysis led to other blade numbers. As the highest efficiencies were for 20 and 25 blades, other three blade numbers were investigated between them, until a peak was found for 22 blades.

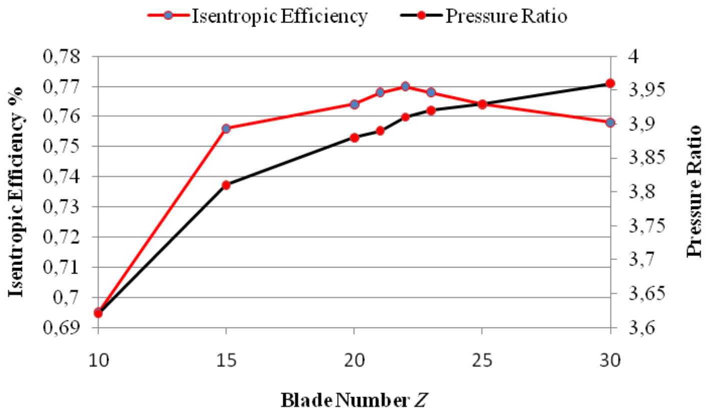

Figure 6. CFD simulations results for the various numbers of blades, showing isentropic efficiency and pressure ratio.

These results of simulations in terms of isentropic efficiency and pressure ratio are shown by Fig. 6. Simulations analysis showed that the efficiency drop from 22 to 30 blades is related to the discharge narrowing, which seems to cause more friction losses. Figure 7 presents the comparison between the velocity fields in the meridional plane for the cases with 10 and 30 blades. For the case with 30 blades, there is a clear deceleration right before the outlet, providing evidence that the passage blocking in generating losses.

On the other hand, the impellers with less than 20 blades were not able to cause enough head rises, having also lower efficiencies. Figure 8 shows a comparison of head along the impeller. Even with less work required, impellers with fewer blades generate less pressure rise proportionally.

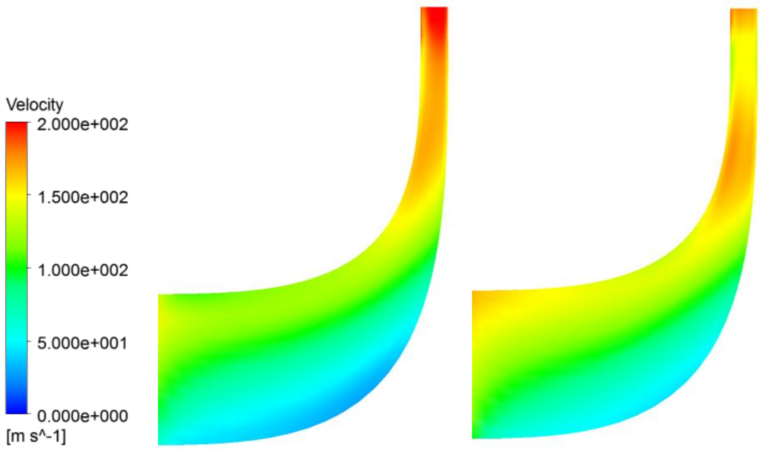

Figure 7. Velocity fields in the meridional plane for the case with 10 blades (left) and with 30 blades (right).

As noticed in Fig. 5, all the configurations efficiencies were below the design target of 85%. Analyzing the computational data, these low efficiencies might be due high ratio between blades height at inlet and blades height at outlet. As pointed by Whitfield and Baines (1990), high ratios like in the designed blade are useful to decrease the Mach numbers at the impeller outlet. In fact, simulations reported considerably low Mach numbers for all configurations, with an average around 0.6. This value might be too low if one intends to reach a high efficiency (Cumpsty, 2004).

Figure 6 helps explaining the effect of a high blade height in the inlet. The velocity increase in the hub region is very low, denoted by the blue region in the contours. These low velocities tend to reduce the energy transfer to the fluid. The curvature of the meridional plane may be modified to be more inclined in this region as an attempt to force higher velocities. Figure 7 also shows the tendency of low head rises in the region of high blade heights, as the curves slope rises when approaching the outlet.

As expected, pressure ratio increases as the number of blades increases, as shown by Fig. 5. Blade loading is reduced by the increase in blade number, but not enough to balance the work applied to the fluid by a larger number of blades. This happens because mass flow and rotation were imposed in the simulations. Being so, it is considered that the system is able to provide the work necessary to keep this rotation. The efficiency of the different blade numbers could be compared for the same pressure ratios and, therefore, for different rotations.

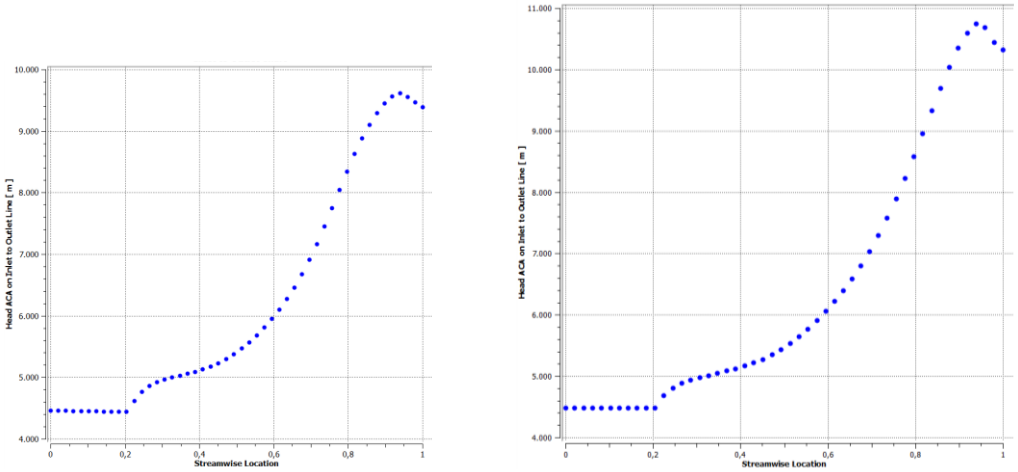

Figure 8. Graphs for the head averaged by the path area along the impeller, from inlet to outlet. Case with 10 blades (left) and with 22 blades (right).

6. CONCLUSIONS

For all tested blade numbers efficiency is below the indicated by Balje's (1981) charts and the assumed for the design procedure. It is clear that the further refinement of blade geometry is needed, with more attention to the inlet blade height, that might be too large, and to the outlet blade height, that might cause excessive friction losses for being too narrow. Varying thickness distribution must also be explored.

However, the study of blade number showed how greatly the performance and efficiency are affected. In the range of blade number values indicated by the empirical correlations, efficiency changed almost 8% and pressure ratio had significant variation, decisive to meet or not the compressor imposed requirements. The correlations presented by Came and Robinson (1999), Rodgers (2000) and Xu and Amano (2012) predicted well the optimum blade number of 22 blades. It is recommended to perform simulations in other rotations to monitor the responses. It would be interesting to check the different blade numbers generating the same design point pressure ratio.

7. ACKNOWLEDGEMENTS

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