EXECUTIVE SUMMARY OF THE THESIS

Preliminary Design and Optimization of a Subsonic Turbine for Rotating Detonation Engine Applications

TESI MAGISTRALE IN MECHANICAL ENGINEERING – INGEGNERIA MECCANICA

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ACADEMIC YEAR: 2021-2022

1. Introduction

World energy consumption has increased in the last decades, and it is still increasing. Unfortunately, the main energy sources are coal, oil and gas, which are an important cause of pollution and climate change. So, there is the need to find alternative ways for the extraction of energy. Nowadays, Gas Cycles play a key role in energy field, but are mainly fueled by fossil fuels, having so a strong environmental impact. For this reason, in the past two decades, the interest on the novel technology called "pressure gain combustion" has increased, since it allows to increase the efficiency of the Joule-Brayton cycle and to use alternative fuels like hydrogen, reducing almost to zero the pollutants and CO_2 emissions. It works by using the detonation combustion mode, instead of classical deflagrative one. This allows to have a pressure increase during the combustion phase of the thermodynamic cycle, allowing to have a higher efficiency compared to cycles that use deflagrative combustion [1]. There are different technologies able to perform detonation combustion: standing wave detonation engines, pulsed detonation engines (PDE) and rotating detonation engines (RDE) [1]. The focus of this work is to design a turbine that can be efficiently coupled with an RDE. The basic geometry of RDE combustion chamber is an annular tube where one or more detonation waves travel circumferentially burning the fresh mixture injected from one end of the tube [2] (Figure 1).

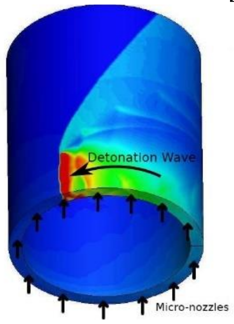

Figure 1: RDE combustion chamber basic geometry [2]

The flow coming out from the detonative combustor is supersonic, non-uniform and unsteady, so coupling it with a gas turbine is not trivial. Two main coupling solutions are possible: design a supersonic turbine or decelerate the flow bringing it to subsonic conditions. The disadvantage of the supersonic turbine is the presence of the shocks that are cause of losses, while the disadvantage of the subsonic turbine is the high inlet Mach number that reduces the flow defection achievable by stator and rotor to keep the stage subsonic. The inlet Mach number is high because a deceleration to conventional values of 0.2÷0.3 will cause too many losses, cancelling the advantage of detonative combustion. An efficient design of the supersonic turbine is possible, as demonstrated in [3]. So, the aim of this work is to study the feasibility of designing an efficient subsonic turbine stage, focusing mainly on the problem of high inlet Mach number rather than on non-uniformities.

2. Mean line parametric analysis

Before describing the design procedure, let's set some necessary conditions: inlet total pressure set to 15 bar, as done in [3]; inlet total temperature set to 1773.15 K, as it can be found in [4]; stator and rotor outlet Mach number set to 1, to have the maximum deflection without going supersonic.
The first step of the turbine design is a mean line parametric analysis, to study the turbine behavior at the variation of stator inlet Mach number, hub radius, blades flaring, rotational speed and rotor outlet absolute flow angle. To perform the mean line calculations the software zTurbo was used. It has already implemented some losses correlations and among them we decided to use Traupel. Due to the non-conventional case studied in this work, we were out of the validity range of all the correlations. So, we applied some modifications to Traupel correlations for the extrapolation of losses values. First of all, we modified the correlation between deviation angle and deflection: Traupel correlation is valid for deflections greater than 45°, but in this case we reached much smaller values, like 20°. For this reason, we decided to put the deviation to 0° for deflections lower than 45°. Then we modified the correlation between blade angles and profile losses. Here the correlation was valid for blade outlet angles greater than 45° from axial direction, but we reach outlet angles of 20° in our analysis. So, we decided to put the profile loss of a straight blade equal to 1% and then linearly interpolate between the loss calculated from the smallest outlet angle valid for Traupel (i.e. 45°) and this 1%. About secondary losses, since they are a combination of profile losses and experimental coefficients, we decided to not modify their correlation since it is not the scope of this thesis, and it would have required too much time to validate their extrapolation strategy. For profile losses the extrapolation works well, but for secondary losses it is inadequate. Before running the parametric analysis, another problem had to be solved to use zTurbo for our scope: some of our input parameters, like stator and rotor outlet Mach number, blades flaring and rotor outlet absolute flow angle are zTurbo outputs, so they depend on losses. But zTurbo wants isentropic input values from which it calculates losses, and its input parameters are different from ours. So, it was created a code for mean line isentropic calculations that, from our inputs, calculates zTurbo inputs. The problem is that the isentropic zTurbo inputs are calculated form our parameters, but then zTurbo applies losses, so the values of the same parameters coming from zTurbo will be different from our wanted values. For example, if we want stator outlet Mach number equal to 1, we use this value to find the inputs to pass to zTurbo, then zTurbo performs mean line calculations applying losses, so it returns a stator outlet Mach number of 0.9, which is different from our desired value of 1. So, to solve the problem, an iterative algorithm was created to modify our parameters values up to when zTurbo returns our wanted values. After the creation of this algorithm, the parametric analysis could be run.
The main results are summarized in Figure 3 and Figure 4: we want to find the trend that maximizes efficiency, so it is possible to see that stator inlet Mach number and rotor outlet absolute angle must decrease to increase efficiency, while stator flaring, hub radius and rotational speed must increase. Obviously there are some limits, like the one already cited about Mach number that cannot be too small to avoid important losses in the transition duct. So, as a design choice, we have chosen a lower limit of 0.6. About rotational speed the limit is given by mechanical stresses, so the maximum rotational speed allowed will be found in the next steps, through FEM calculations. It will result 9500 rpm. The limit on flaring is given by boundary layer detachment from endwalls, and for stator we decided to use 1.2 as maximum value, for rotor 1.4

since the blade will be much longer than stator one. About rotor outlet absolute angle, since it influences rotor flaring, its lower limit is the one that gives a flaring of 1.4, so -27°. Finally, about hub radius we decided to use the upper limit of 0.345 m because it is the value used in [3] and because at its increasing corresponds a blade height decreasing, so if the blade height is too small the endwalls losses will be very high, cancelling the efficiency gain given by the higher radius.

3. CFD analysis

A mean line analysis is not able to account for geometrical features like camber line shape, blade thickness distribution and endwalls contouring, so the second step of the turbine design is the development of 2D and 3D models of the stage and the assessment of their behavior through CFD calculations, using the software Ansys CFX.

Stator analysis

We started by developing the stator 2D blade using the values coming from the mean line analysis. Then we analyzed the stator performances at various blades number, axial chord length, blade thickness distribution, camber line shape and endwalls contouring. We found that the best performances in terms of losses and flow field are for a stator with 35 blades, in which the axial chord is 70 mm, the camber line has the curvature near the leading edge (Figure 5 a) and endwalls diverge in correspondence of leading edge. The maximum thickness of the profile is 10% of the axial chord placed at 50% of axial chord. This profile was then subjected to a mechanical stress assessment to check if it can withstand the pressure loading at high temperatures. Considering Inconel 718 as blade material, which yield strength at 760° C is 758 MPa, and to apply different solutions to increase the blade resistance even at higher temperatures, like single grain crystal blade, thermal barrier coating and internal cooling, we can say that the blade is able to preserve its resistance to temperatures up to 900° C, with flow temperatures up to 1600° C (Figure 2). Considering also that the flow maximum temperature in our case is of 1500° C, we considered acceptable a yield strength of 758 MPa for mechanical safety coefficient calculation. The result is that safety coefficient is 1.97, so a good value.

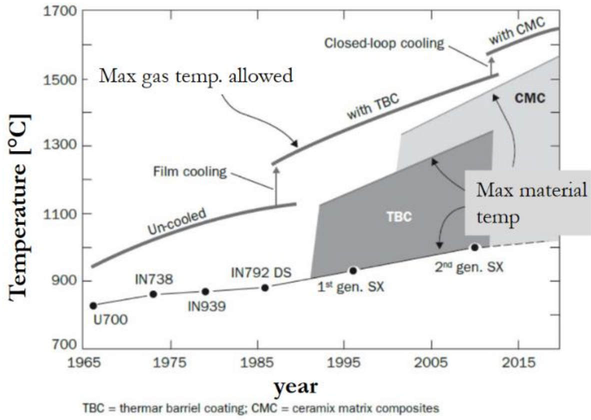

Figure 2: Max temperatures allowed for flow and blades material for different technologies

At this point the blade and endwalls shape were optimized through the software FORMA, developed by Politecnico di Milano, to increase its performances. The optimized blade shape with its Mach field is reported in Figure 5 b) and the losses goes from 3.2% before optimization, to 1.85% after optimization. This profile was also mechanically assessed, finding a safety coefficient of 2.37. So, the optimized profile was developed in 3D to look at the 3D flow. It results in some small separations of boundary layer from endwalls in correspondence of blade leading edge and a small separation on the rear part of the suction side. The losses are of 7.07%, so a satisfactory value considering that no 3D optimization was performed. Finally, the 2D optimized profile performances were assessed in off-design conditions, considering to change the inlet total pressure to 20 bar and to use the incidence values of -10° and +10°. The losses increased a little bit, but the results are promising.

Rotor Analysis

For 2D rotor generation, we started calculating the number of blades. The flow at stator outlet is at high relative velocity (relative Mach number of 0.8 for a rotational speed of 8000 rpm), so we wanted to avoid having a normal shock at rotor inlet due to blade leading edge thickness, which is of 3 mm. So, considering the area restriction, the maximum number of blades is calculated to avoid the presence of shocks at leading edge. Then, from Zweifel solidity, we calculated rotor axial chord, which results 0.161 m. We tried to shorten it, but it is not possible without worsening a lot the turbine performances. Later we analysed different camber line shapes, finding two promising shapes: one front loaded (Figure 6 a) and one central loaded (Figure 6 b). The blade thickness distribution is the same of stator. We performed a mechanical assessment on both the blades, before continuing with shape optimization. For centrifugal loading we considered the rotational speed of 8000 rpm, used for all the previous simulations. For both the profiles the mechanical assessment is good, indeed we can increase rotational speed still having a safety coefficient greater than 1.5. But before increasing the rotational speed, we decided to optimize profile and endwalls shapes of the front loaded blade, obtaining results that were not satisfactory. So, at this point, we decided to study the effects of rotational speed on mechanical stress of central loaded blade, to find the maximum velocity for which the safety coefficient is 1.5: it results 9500 rpm. Since the relative velocity changes with rotational speed, the blade angles were adjusted to match the relative angles given by that rotational velocity, while the number of blades is changed always avoiding shocks at rotor inlet (for this rotational velocity they are 26 blades). At this point, we optimised the improved central loaded profile that rotates at 9500 rpm. The optimized stage efficiency results 96.88%, while the extracted work is 26.878 MW (Figure 7). So, this profile was firstly mechanical assessed, finding a safety coefficient of 1.53, then it was developed in 3D, finding an efficiency of 90.33%, with a work of 24.682 MW. The performances are still good, the reductions are due to the formation of two big vortexes on rotor suction side and to a limited boundary layer separation on hub and shroud. Finally, the stage was simulated in the same stator off design conditions, finding also here optimistic results.

4. Conclusions

In this thesis work, a subsonic turbine stage was designed to deal with non-conventional inlet conditions. This work is needed to explore a way to couple a gas turbine with a rotating detonation combustor. The design started from a mean line parametric analysis, finding the parameters trends that increase efficiency. Then, from mean line results, we started the CFD analysis of stator and rotor cascades. Different two-dimensional profile shapes were analyzed, finding the ones that give best performances in terms of efficiency and flow field. Then, these best profiles were optimized to improve their performances. In the optimization process, also the endwalls contouring was included. The optimized profiles were mechanically assessed and their behavior in off-design condition was analyzed, finding optimistic results. Finally, they were developed in 3D and, even if the performances have a small worsening, the results are satisfactory.

Many developments to couple turbine with RDEs are possible, like improving 3D flow or make a deeper study on non-uniformities effects, but the results obtained up to now are promising, showing that different solutions can be applied to couple turbomachinery with RDEs.

5. Bibliography

References

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6. Acknowleggements

I want to express my gratitude to Prof. Paolo Gaetani, Dott. Noraiz Mushtaq and Prof. Giacomo Persico, that made this work possible.

I also want to express my gratitude to my family and all the important people that supported me during my years at Politecnico di Milano.

Figures

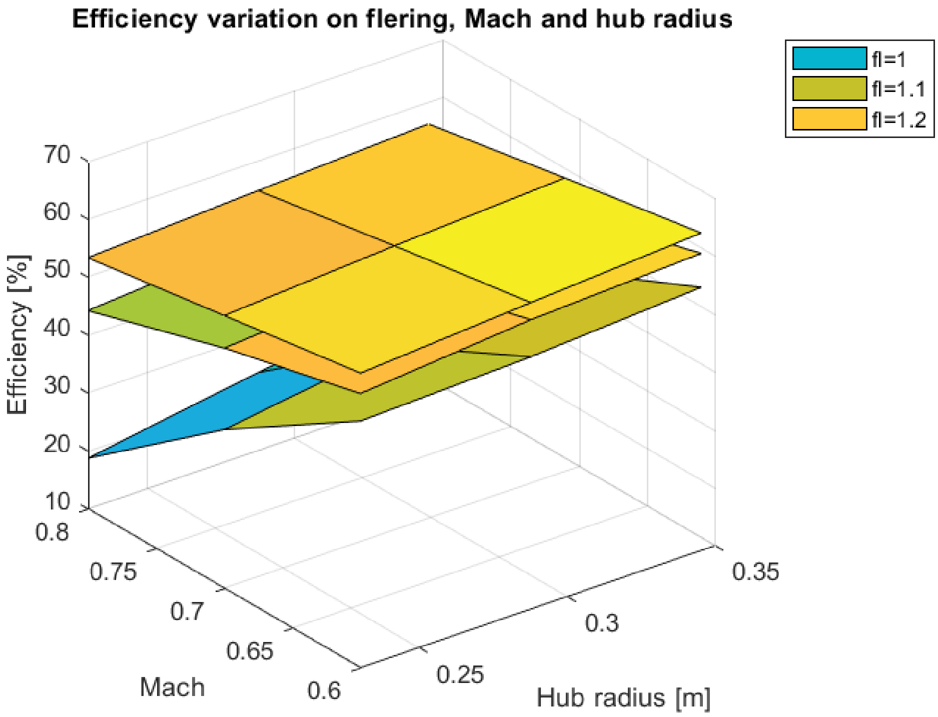


Figure 3: Stator inlet Mach number, flaring and hub radius effects on efficiency

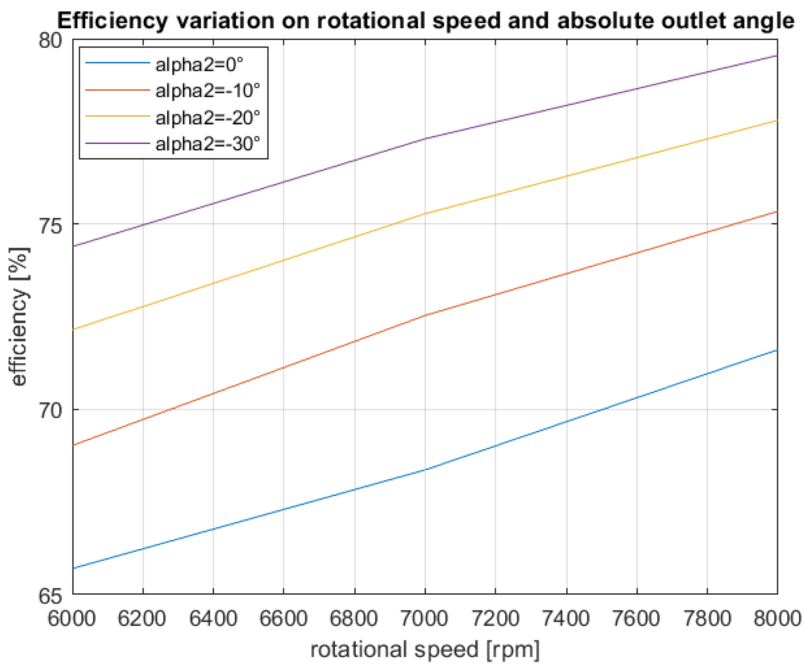
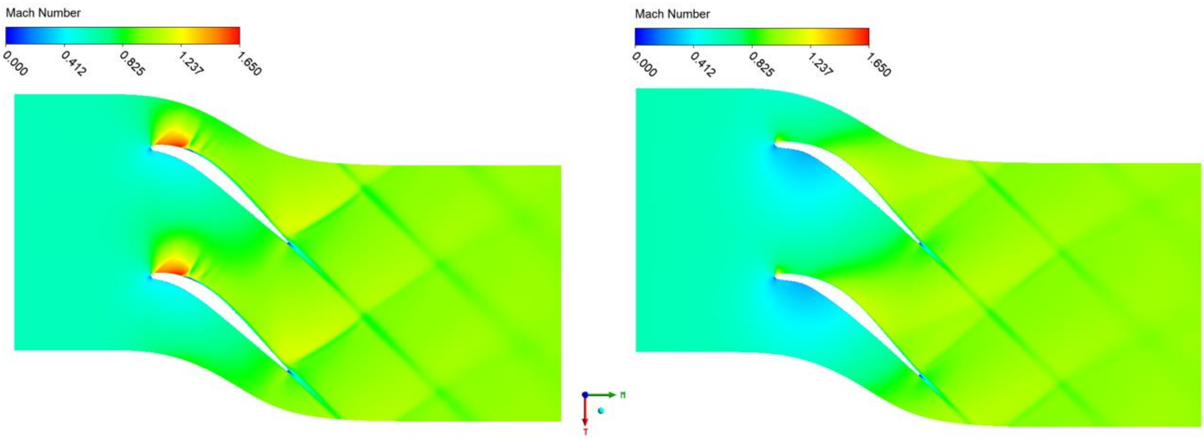
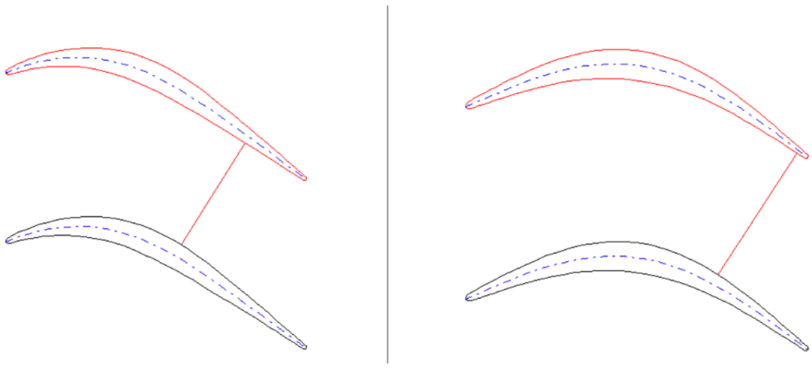


Figure 4: Rotational speed and rotor outlet absolute angle effects on efficiency



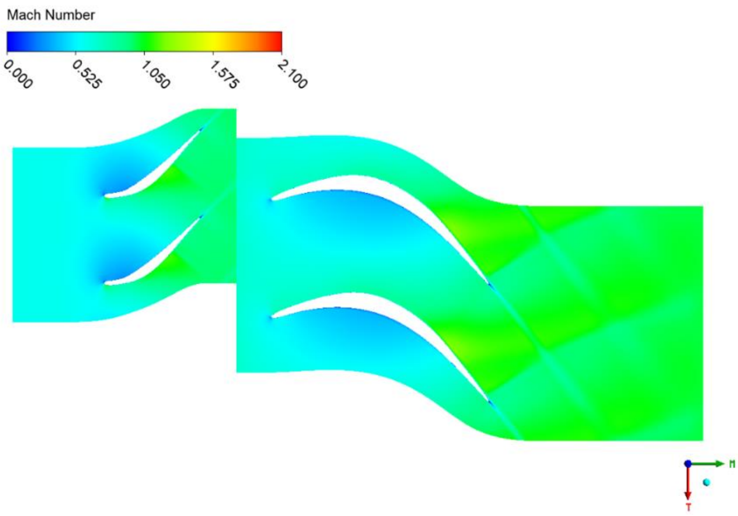
a) b)

Figure 5: Comparison between Mach fields of stator base profile a) and optimized profile b)



a) b)

Figure 6: Camber line comparison: a) front loaded, b) central loaded



$M_0 = 0.608$
$M_1 = 0.966$
$M_{1w} = 0.714$
$M_{2w} = 1$
$M_2 = 0.68$
$\eta_{is} = 96.88\%$
$Power = 26.878\ MW$

Figure 7: Optimized stage Mach field with principal performances