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Toolchain Development for Gas Turbine Optimization

Internship Report

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1/31/2019

The following report is the product of an internship work carried out at the *Flight Physics and Loads Department* of the *Netherlands Aerospace Centre NLR* in Amsterdam. From the 1^{st} of August until the 31^{st} of January I have been working full time as an intern under supervision of *Michel van Rooij*.

The internship project, part of the second year of the master's program in *Thermal and Fluid Engineering* at *University of Twente*, was supervised by *Prof. Kees Venner*.

The project was commissioned by *Sulzer RES Venlo*, which through *reverse engineering* provides repair services to the power generation industry. At Sulzer, they are specialized in delivering high quality components, often with design improvements over the original designs. They offer an alternative to the Original Equipment Manufacturer (OEM) for the supply of gas turbine and steam turbine. The project has been carried in constant collaboration with *Pim Goedhart* and *Sjef Mattheij* from Sulzer development department.

During my internship I have been supervised by *Edward Rademaker* for the development of the project in GSP, which is a gas turbine simulator made by NLR.

Besides, *SoftInway* provided me tutorial and live training to learn the theory of turbomachinery and how to use the axial turbine analysis and design software. *Nishit Mehta* gave me the live training and kept helping me during my project.

I would like to express my sincere gratitude to all the people mentioned here for the great and enriching experience and for the constant support provided.

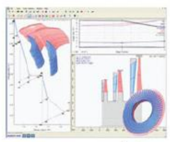
As part of the training, I have successfully achieved two certificates from SoftInway: Axial Turbine Design Online Training (Theory and Workshop) and AxSTREAM NET Online Training.

CERTIFICATE OF COMPLETION

This certification of completion is awarded to

Mr. Andrea Zanon

For successful completion of SoftInWay Turbomachinery University's
Axial Turbine Design Online Training (Theory and Workshop)



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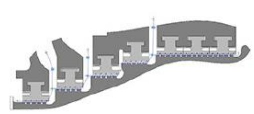

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Abstract

A modernized gas turbine can result in a more efficient solution for energy production, answering to an always increasing demand for more power at reduced fuel consumption. New turbine integrations are available or in development to reduce losses and to allow extraction of more work. In this report, the aerodynamic design of the turbine blades of one of the most used engines in service worldwide, the *Siemens V94.2* gas turbine, is investigated to assess the feasibility of a more efficient machine.

The purpose of this assignment is to analyse the third version of the engine. The turbine blades are not created from scratch, but the existing machine is reproduced through reverse engineering and the design analysed for possible improvements.

Several upgrades are in development to optimize the turbine, including improvements on aerodynamics of the blades, materials used and cooling system. This study focuses on aerodynamics of the blades exclusively. The profile characteristic angles like inlet and outlet metal angles, stagger angle, gauging angle, Leading Edge (LE) and Trailing Edge (TE) wedge angles and the unguided turning angle are investigated. Steady state streamline calculations are performed to observe the performance response. Special interest of this research focuses on cooled turbine total-to-total efficiency, power output, turbine reaction and total values of temperature and pressure at inlet and outlet of the turbine.

A toolchain is developed by modelling the turbine through several components of the *AxSTREAM software package,* and integrating the resulting performance map in *GSP* to have an overview of the entire gas turbine performance.

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Introduction

The *Siemens V94.2(3)* is a single shaft heavy duty gas turbine, with a capacity of 154 MW and designed for 50 Hz operation. It includes a 16-stage axial compressor and 4-stage axial turbine having a common rotor. It is equipped with two combustion chambers, vertically mounted on the sides of the gas turbine (fig.1a) [1].

Since the first model in the early 1980's, the gas turbine has been upgraded constantly, improving the efficiency and answering to an always growing demand of energy (fig.1b). Siemens introduced several improvements, among others re-designs of the compressor, combustor and turbine. More modern materials and coating technologies together with a more efficient cooling system were adopted as well. A significant role to make the gas turbine more competitive and cost-effective can be played by the aerodynamics of the turbine blades [2].

In 2005 *Siemens* developed a new aerodynamic design of turbine blades, called *Si3D*, introduced for stage 1 and 2, yielding an increase of 3.7 MW in power output and 0.5% in gas turbine efficiency [3]. In 2009 the *Si3D* upgrade was extended also to stage 3 and 4 [4]. *Si3D* retrofit has been successfully applied to a combined cycle plant at *Senoko Power Station* in Singapore, claiming a +1.51% improvement in gas turbine efficiency, with additional +5.2 MW of power output [5]. A further development at every stage, called *Si3D+*, is under research and it will be released in 2020 by *Siemens* [4].

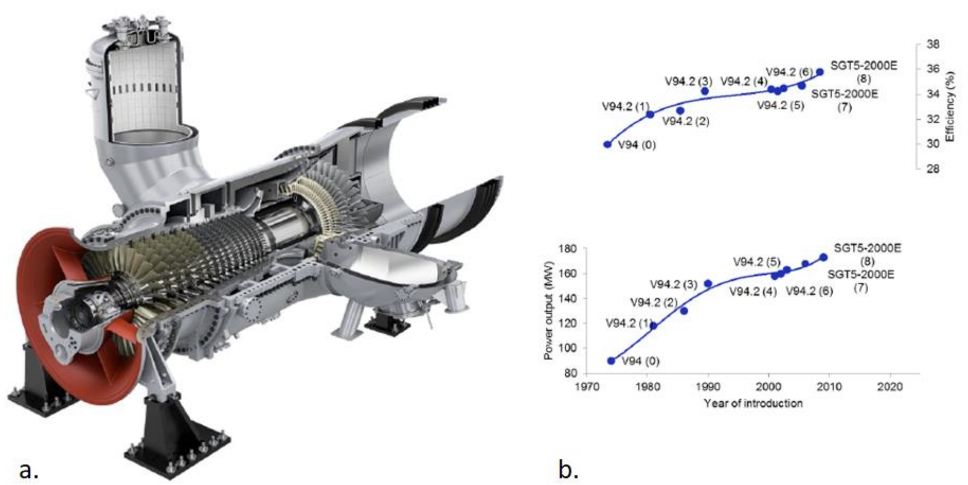


Figure 1 – a) V94.2 model; b) V94.2 performance evolution

In this report the third version of the engine's turbine (without *Si3D* upgrades) is taken as baseline. The feasibility study is performed on the entire model, considering every stage. The changes are applied starting from the first stage and including one by one all the others. The assignment is divided in two main tasks. First, the baseline models are created. After gathering all information, the models are built up in *AxSTREAM* and *GSP*. When performances are matching with experimental data the baseline is 'frozen'. Then, in the second part, only the aerodynamic parameters are edited to re-profile the turbine blades and vanes.

Baseline

Introduction

The baseline gas turbine model is achieved through creating the turbine model in AxSTREAM Axial Turbine, and the gas turbine model in GSP. AxSTREAM Axial Turbine is a platform for multidisciplinary turbomachinery design, analysis and optimization software tools. It provides a fully integrated and streamlined solution for the complete flow path design process of axial turbines. The package of tools used for the project includes: the stream line solver for meanline (1D) and axisymmetric (2D) analysis, profiling and 3D blade design, Design of Experiment (DoE) for optimization calculations (AxPlan), off-design calculations for generation of performance maps (AxMap) and reverse engineering module for extraction of profile geometry from blade's 3D CAD model (AxSlice). Besides, axial turbine cooling flows and secondary flows module (AxSTREAM NET) and design and process integration tool (AxSTREAM ION) have been considered but not adopted for the project [6].

In order to create the model of the turbine in AxSTREAM several information are mandatory. At *Sulzer*, through sophisticated 3D laser scanning, the blades can be reproduced in a CAD format. The first step to make the model in AxSTREAM is to import the blades geometry. AxSlice is the tool dedicated for this task, part of the package of AxSTREAM. Then, the complete operational sheet of the machine is composed through the information provided by Sulzer, and used to set the model. This includes mass flow, thermodynamic data of the turbine, clearances and cooling system.

Importing blade geometry

The domain is defined at hub and tip, and the number of sections to be imported in AxSTREAM selected (fig.2a). After several attempts with nine, seven and five sections, the last option is chosen. Fewer sections were creating smoother blades (fig.3). Besides, the streamline analysis will run faster with fewer sections.

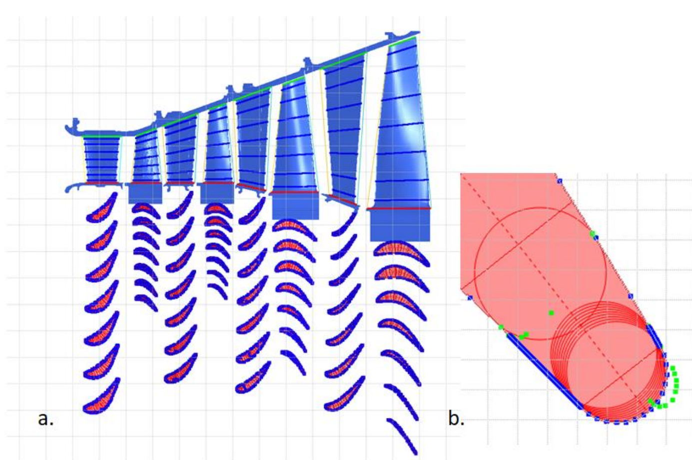


Figure 2 - a) Turbine sections slicing (7 sections) in AxSlice; b) R1 trailing edge approximation, in green the old points of the TE with cooling

During this task, the cooling system points are manually removed, as the aim of this optimization focusses on aerodynamic parameters only. As appreciable in fig.2b, the trailing edge is reshaped by removing the geometric step at the location of cooling flow injection.

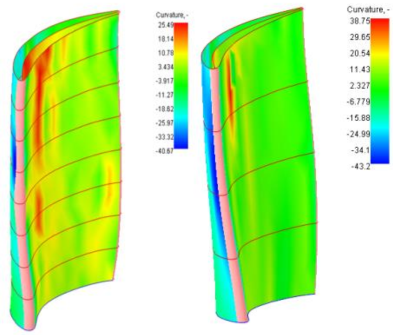


Figure 3 - R1 dimensionless curvature comparison with 9 and 5 sections, from AxSTREAM

The recognized properties can immediately be loaded into the AxSTREAM main project to calculate performance, kinematic and thermodynamic parameters and losses at design point and off-design conditions using the streamline solver [7].

Profiling the baseline

The Profiler and Blade Design software is used to create and edit 3D airfoils. A wide range of geometric tools and interactive charts are available [8]. In profiling, each section, after being saved as a ‘shadow’, is switched into profiling mode ‘custom side profiling’. Several splines automatically interpolate the profile (fig.4a), keeping the inlet and outlet metal angles and the throat fixed. The splines can then be edited with several points to match the shadow profile of the sections. Besides, parameters can be tuned to control the splines more precisely in the profiling grid shown in fig.4b.

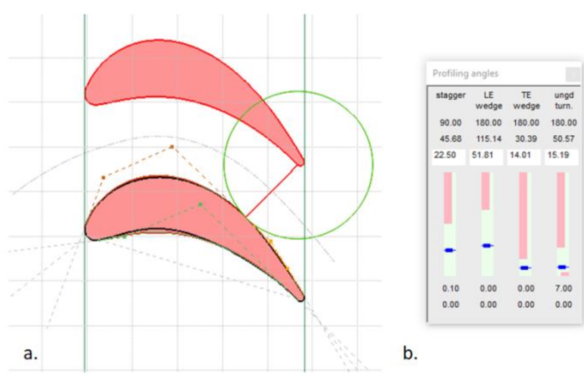


Figure 4 – a) R1 profiling mode with splines, in black is saved the shadow profile; b) The profiling grid.

Profiling at this point is needed for two reasons. First, the geometries are transferred multiple times, resulting in imperfect blade shapes, e.g. with wavy surfaces that require repair (fig.5). Secondly, optimization, which will see the re-profiling of the sections, is performed in profiling mode. In the following figures a comparison is shown of a processed and unprocessed blade in terms of surface curvature.

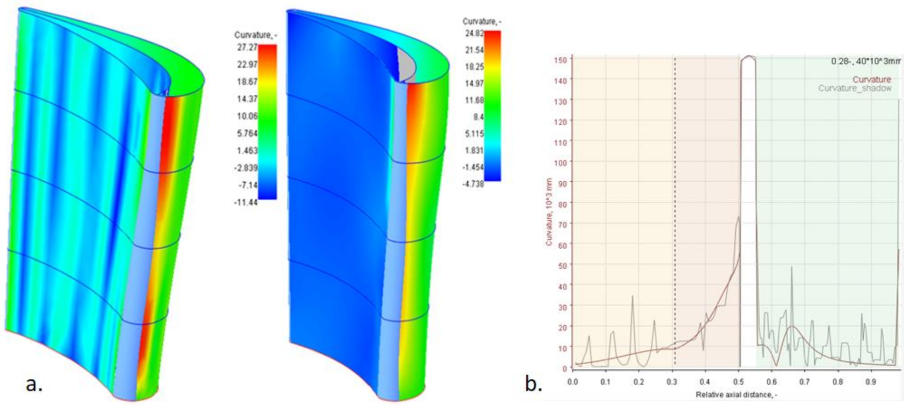


Figure 5 – a) S1 unprocessed and processed comparison; b) Curvature chart comparison

Working Fluid

At the turbine, the working fluid in use is the flue gas from pure methane combustion. The well-known stoichiometric chemical reaction is presented here [9].

$$CH_4 + 2(O_2 + 3.76N_2) \rightarrow CO_2 + 2H_2O + 7.52N_2$$

Equation 1 - Methane combustion reaction

In AxSTREAM, the air excess factor (usually known as lambda) is defined as follows:

$$af = \frac{l}{l_0} = \frac{AFR}{AFR_s} = \frac{\frac{mole\ of\ air}{mole\ of\ fuel}}{\left(\frac{mole\ of\ air}{mole\ of\ fuel}\right)_{stoich}} = \frac{49.407}{17.238} = 2.866$$

Equation 2 - Air excess factor

By setting pure methane as fuel also in GSP, the ratio AFR and AFR_s are provided, and *af* is estimated. Besides, to make the model run properly, the same fluid has to be chosen for cooling. AxSTREAM Tutorial advises an *af* of 1000 [10].

In the following table (table 1) the values are reported of air excess factor and stoichiometric air to fuel ratio for the working and cooling fluid in use in AxSTREAM. The flue gas from pure methane combustion was part of the fluid's library of AxSTREAM.

Table 1 - Fluid type characteristics

$\boldsymbol{Fluid\ Type}$	$\boldsymbol{Air\ excess\ factor, af}$	$\boldsymbol{Stoichiometric, l_0}$
Working fluid (AxSFLG_CH4)	2.866	17.238
Cooling fluid (AxSFLG_CH4)	1000	17.238

Cooling System

The turbine model cooling system is very important if the aim is to investigate the performance of the machine. For instance, the Mollier diagram of a turbine including cooling (fig.6) is significantly different from the one presented in every thermodynamics book. In fact, the cooling flow alters the entropy generation during expansion. The total-to-total turbine efficiency is also influenced, and in this project, the cooled total-to-total turbine efficiency will be used as the turbine efficiency. The definition of cooled stage turbine efficiency [11] is presented here.

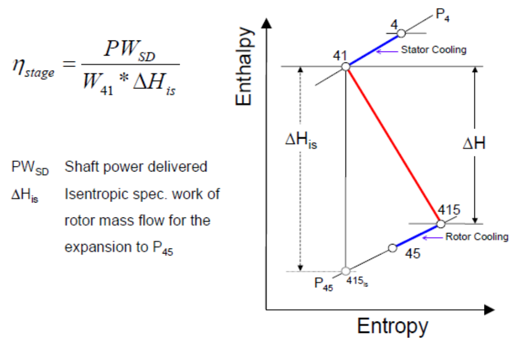


Figure 6 – Example of cooled turbine stage, efficiency and Mollier diagram

The cooling system data have been gathered from other models and typical values from AxSTREAM tutorials [12]. Here follows a sketch of the cooling system in use (fig.7), with a table assigning values to each mass flow (table 2). This version of the engine, provided by Sulzer, includes a cooling system which uses 11% of mass flow entering the compressor [13]. Two main flows, depicted in the figure, represent the bleeds from compressor adopted for the turbine cooling.

Table 2 - Cooling system

$\dot{m}_{in}$ $[kg/s]$	$\dot{m}_{cool}$ $[kg/s]$	$Cool.\%$	$B_1[kg/s]$	$B_2[kg/s]$
496	54.56	11	39.68	14.88

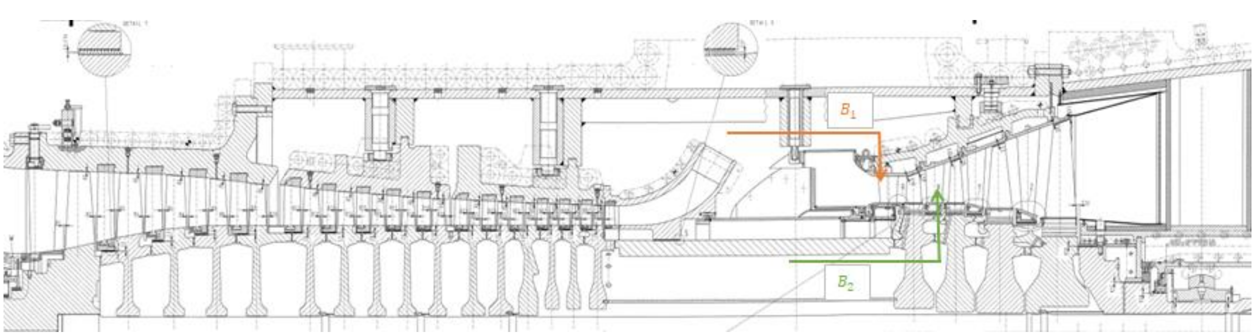


Figure 7 - Cooling flows adopted

To have a more realistic design, mandatory in AxSTREAM, an available and complete system from a similar version [14] has been used to estimate the mass fractions of each cooling stream, which are then used with the provided mass flows (B1 and B2) to recalculate the distribution of cooling flows (table 3). Furthermore, to run AxSTREAM, mandatory parameters to model the cooling system were adopted taking typical values from AxSTREAM tutorials [10].

Table 3 - Cooling flows distribution

S1 Hub $\left[\frac{kg}{s}\right]$	S1 TE $\left[\frac{kg}{s}\right]$	R1 Tip $\left[\frac{kg}{s}\right]$	R1 TE $\left[\frac{kg}{s}\right]$	S2 Hub $\left[\frac{kg}{s}\right]$	S2 Shroud $\left[\frac{kg}{s}\right]$	S2 TE $\left[\frac{kg}{s}\right]$	R2 Tip $\left[\frac{kg}{s}\right]$	S3 Hub $\left[\frac{kg}{s}\right]$	S3 Shroud $\left[\frac{kg}{s}\right]$	R3 Hub $\left[\frac{kg}{s}\right]$	S4 Shroud $\left[\frac{kg}{s}\right]$	R4 Hub $\left[\frac{kg}{s}\right]$
10.18	15.08	12.74	1.69	3.65	2.34	2.23	0.82	3.06	1.36	0.27	0.71	0.42

Streamline analysis

The Streamline solver module (fig.8) allows performing meanline (1D) or axisymmetric (2D) calculations of turbomachines to determine streamwise and spanwise distributions of kinematics, thermodynamics and loss parameters as well as leakages and secondary air flow for a given set of boundary conditions [6]. If cooling is included, it can perform aero-thermodynamic calculations of blades and endwall cooling taking into account the mixing losses and the change of working fluid temperature and properties as well as the composition due to mixing at each streamwise and spanwise location with the flow path [6].

This tool can perform steady state calculation at the mean section or considering all the sections simultaneously. It can accurately check the performance of designed machines at the design point operation [15].

The problem formulations used for this assignment are:

- Find mass flow rate for given inlet total pressure and outlet total/static pressure
- Find inlet total pressure for given mass flow rate and outlet total/static pressure

These formulations are preserving the geometry of the blades when running [15]. The boundary conditions set for the problem were provided by Sulzer [13], based on empirical values (table 4). Both problem formulations were used depending on the specific task.

Table 4 - Baseline mandatory data

$\dot{m}_{in}\left[\frac{kg}{s}\right]$	$T_{0_{in}}[K]$	$p_{0_{in}}[bar]$	$p_{0_{out}}[bar]$	$p_{out}[bar]$
451.3	1426	11.282	1.013	0.899

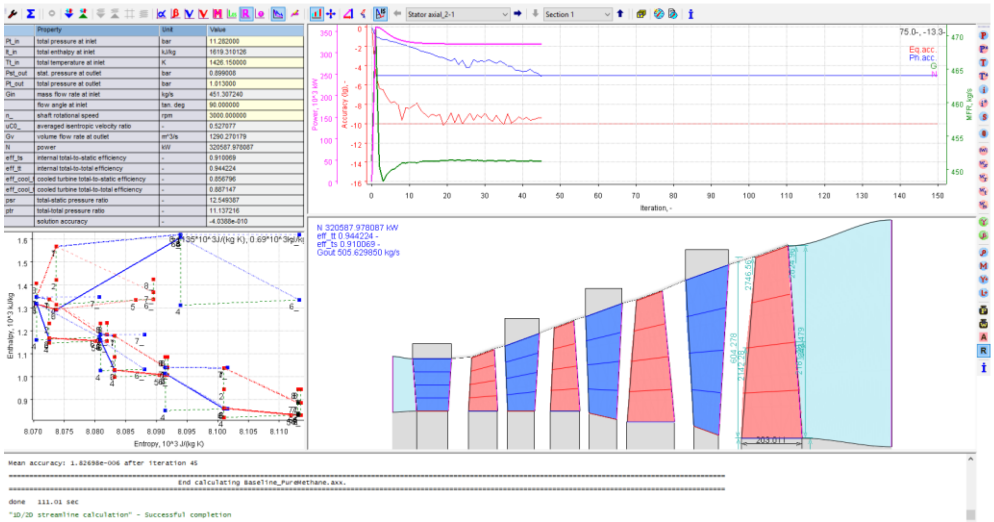


Figure 8 - Streamline solver overview

The primary profile losses model adopted in AxStream is “Craig&Cox corrected”, as recommended when using flue gas as working fluid [10]. Loss limit and scale are tuned to match results with experimental data.

Off-design performances calculations

AxMAP uses automatically the meanline and streamline solvers to generate performance maps. It runs off-design calculations for a minimum number of two variables, studying the influence of operational parameters on turbine performance [15].

The off-design calculations are estimated considering a constant rotational speed of 3000 rpm (50 Hz). Total inlet temperature and pressure are assigned with an operational range and the objectives specified are: mass flow rate at inlet of the turbine, total-to-total pressure ratio and cooled turbine total-to-total efficiency (fig.9).

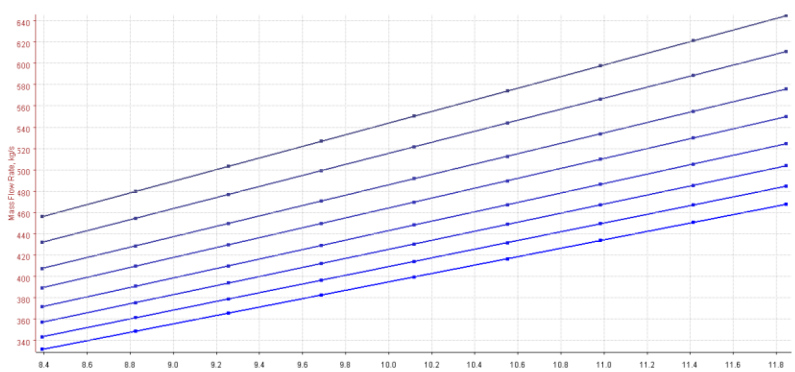


Figure 9 - Pressure ratio and mass flow rates for several inlet temperatures (1500 to 800 K from bottom) at 3000 rpm (not corrected)

The data are gathered and corrected to generate the turbine performance map to be used in GSP. The performance characteristics are usually drawn in terms of the mass flow parameter [16]

$$\frac{\dot{m}\sqrt{T_{0_{in}}}}{p_{0_{in}}}$$

Equation 3 - Corrected mass flow

and efficiency versus overall pressure ratio

$$\frac{p_{0_{in}}}{p_{0_{out}}}$$

Equation 4 - Pressure ratio

at different rotational speeds

$$\frac{N}{\sqrt{T_{0_{in}}}}$$

Equation 5 - Corrected rotational speed

GSP – Gas turbine simulation

GSP is a gas turbine simulation program developed by NLR, with which both steady state and transient simulations of complete gas turbines can be performed [17]. It is a powerful tool for performance prediction and both design point and off-design analysis and performance optimization. Gas turbine simulation is based on non-dimensional modelling of the process in the different gas turbine components with aero-thermodynamic relations and steady state characteristics. The model is created by arranging different components in the configuration desired to be simulated. The process in gas turbine components are determined by relations among up to five parameters defined by components maps and thermodynamic equations. These parameters are air or gas properties and other parameters such as rotor speed and efficiencies determining the component operating point.

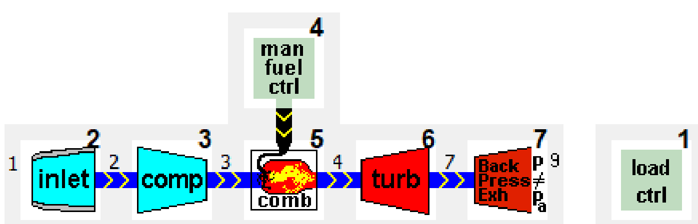


Figure 10 - GSP components viewing

A predefined design point is calculated first from a set of user specified design point data. The deviation from the design point is calculated by solving a set of non-linear differential equations. The equations are determined by the mass balance, the heat balance, the equation for conservation of momentum and the energy balance for all components.

The model of the gas turbine, including the characteristic maps of each component, was already present in NLR library due to previous work on the engine. The project is updated setting the new data for the desired design point, providing useful information for the building of the turbine in AxSTREAM. The purpose of using GSP is to test the turbine maps generated in AxSTREAM.

A back pressure exhaust component enables to impose the outlet static pressure at 0.899 bar (value estimated in AxSTREAM). The diffusor component, not essential for the purpose of the project, needed to convert kinetic energy exiting the turbine into static pressure is then not employed. And besides, by defining the outlet static pressure, the turbine outlet section can be estimated. In AxSTREAM a first part of the exhaust is defined. Some attempts with longer outlet duct have been tried, but they did not show relevant variations at the rotor outlet station.

The problem in GSP is set adopting the load control component (table 5). The amount of fuel is controlled by the power output, defined at 154 MW. This allows to quantify the specific fuel consumption and to make further comparisons.

Table 5 - GSP project input values

$\dot{m}_{in}\left[\frac{kg}{s}\right]$	$\dot{m}_f\left[\frac{kg}{s}\right]$	$N\ [rpm]$	$\Pi[\sim]$	$\eta_{comp}[\%]$	$\Delta p_{comb}[\%]$
496	9.09	3000	11.33	88.5	1.74

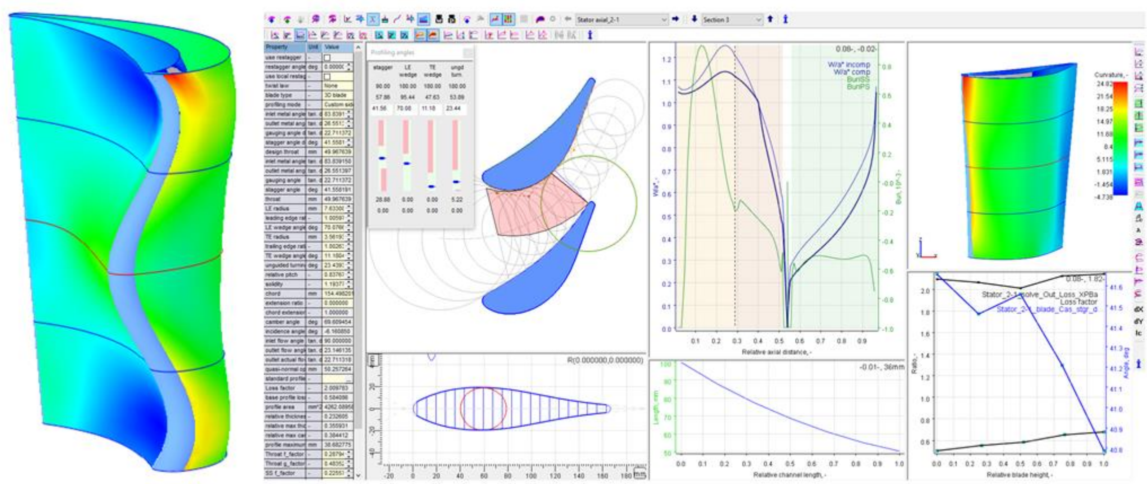
$LHV\left[\frac{MJ}{kg}\right]$	$T_f[K]$	$\eta_{turb}[\%]$	$\eta_{turb.mech.}[\%]$	$p_{exh}[bar]$	$P_{ctrl}[MW]$
50.03	293.15	88.7	98.1%	0.899	154

Optimization

Introduction

The optimization of the engine aims to extract work more efficiently through a more even distribution of work extraction along the entire turbine. This approach diverges from the classical point of view of work extraction focussed at the first stage [18].

The intention was to perform the optimization of the turbine using the DoE tool AxPLAN. This DoE tool analyses an important number of parameters, builds the response surfaces and finds function extrema with a minimal number of calculations [15]. Unfortunately, this tool proved ineffective for this task. The parameters designed for the optimization, like the characteristic angles of the airfoil, were available at the mean section only. It was not possible to include other sections and perform optimization of the complete blade. Therefore, when applying the advised optimal values at mean section, the blade shape becomes greatly distorted (fig.11a), as the other sections stay fixed. By applying the modifications at the meanline to the other sections, the outcome in performances tended to be unsatisfactory



a. b.

Figure 11 – a) Distorted blade after applying the optimal solution from AxPlan of stagger angle at mean section only; b) Profiling overview

AxPLAN can be a powerful tool if used for designing the blade from scratch. For existing machine analysis, blade cascade parameters might be optimized in AxSTREAM ION, where through an automatized process some programming might be possible.

Due to time constraints, it was decided to perform the optimization in a 'manual' mode. Parameters have been changed individually and at every section of the blade, and the impact on performances using streamline analysis observed. The Profiling tool (fig.11b), presented earlier, is used to apply the desired changes and simultaneously have an overview of the sections contour, cascade parameters, characteristic flow charts, losses and 3D geometry. New configurations were then tested using the streamline solver.

Theoretical background

Analysing the baseline, the reaction degree was in some stages negative, especially near the hub. The problem is approached looking at possible increment in reaction, and how this influences performance. First, some theoretical background on axial turbines is provided.

In general, the stage reaction is defined as the change in static enthalpy across the rotor as a fraction of the change in static enthalpy across the stage (eq.6). Assuming the absolute velocity entering and exiting the stage is constant, the stage static enthalpy drop can be approximated by the stage total enthalpy drop, which represents the work extraction of the stage [19].

$$R = \frac{\Delta h_{Rot}}{\Delta h_{Stage}} \cong \frac{\Delta h_{Rot}}{\Delta h_{0,Stage}} = \frac{\Delta h_{Rot}}{W}$$

Equation 6 - Degree of reaction

This expression suggests that a reduction in work extraction leads to a change in reaction. Therefore, the idea becomes to reduce the work extraction where the reaction is poor and to observe the interactions and how efficiency is influenced. The work in a pure axial flow turbine stage with arbitrary reaction can be expressed as follows [19]:

$$W = U\Delta w_{\theta}$$

Equation 7 - Work in a turbine stage with arbitrary reaction

Where Δw_{θ} is the difference in relative tangential velocity between the rotor inlet and outlet. By decreasing the energy transfer at the rotor, the work decreases and also the reaction is influenced. A more specific expression can be written considering all terms of velocities, as follows [20].

$$R = \frac{(U_1^2 - U_2^2) + (W_2^2 - W_1^2)}{(C_1^2 - C_2^2) + (U_1^2 - U_2^2) + (W_2^2 - W_1^2)} = \frac{(U_1^2 - U_2^2) + (W_2^2 - W_1^2)}{W}$$

Equation 8 - Reaction expressed with characteristic velocities

As appreciable in eq.8, the blade velocity and the absolute velocity differences are from inlet to outlet, while for the relative velocity it is the opposite. Therefore, by applying changes to the relative velocity a different impact is obtained with the other terms. Anyway, from this analysis the reaction strongly depends on the relative velocity. When they are equal, the relative velocity is not accelerated and the reaction is zero. A positive reaction configuration will see a rotor outlet relative velocity higher than the rotor inlet one. Besides, for a rotor blade row, reaction is defined as [20]:

$$R_{Rotor} = \frac{W_2^2 - W_1^2}{W_2^2} = 1 - \frac{W_1^2}{W_2^2}$$

Equation 9 - Rotor reaction

It should be noted that the analysis above is based on certain assumptions , while the software is simulating real turbine conditions. In purely axial flow e.g., the rotational velocity is constant (constant radius) while the radius and hence the rotational speed might be different in reality. For instance, the sections in AxSlice are cut with some inclination; they are not precisely parallel to the axial direction.

Methodology

The turbine work extraction is redistributed streamwise. A loss is provoked at the first stage and the new configuration studied at every stage to maximize the energy transfer. For the first two stages the change applied is uniform with radius, with increased reaction leading to improved stage efficiency. While for the last two stages, a different radial distribution giving optimized stage efficiency is reached through several attempts.

The degree of reaction and work are influenced by the characteristic stage velocities constituting the typical velocity triangles. To increase the reaction, the outlet flow angles of both stator and rotor have been changed (fig.12). As visible in the next figure, the outlet angle of the stator is increased, while the outlet angle of the rotor is reduced.

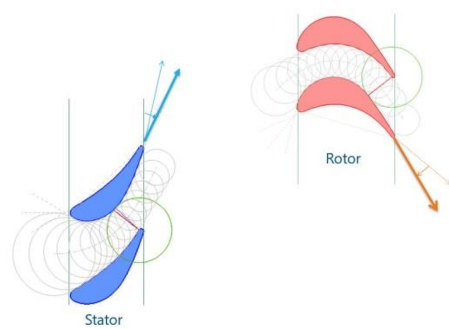


Figure 12 - Action on the outlet angles, general stage

For the first two stages a change in blade angle of approximately 2 degrees is applied almost uniform spanwise, giving a net positive impact on reaction and stage efficiency. For the other two stages, the increase in reaction led to a lower efficiency. A radial distribution of outlet flow angles is therefore applied for stage 3 and 4, based on performances.

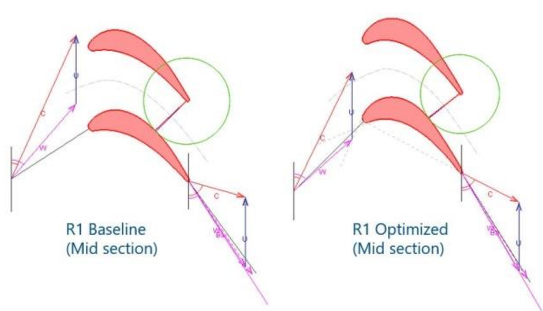


Figure 13 - Comparison of velocity triangles of R1

The outlet flow angle is changed by playing with both the outlet metal angle and the gauging angle, of stator and rotor. Also, the inlet metal angle is adjusted, to better align the flow from one blade to another (fig.13). As advised in tutorial [8], the incidence angle is kept in a range of -2 to -6 degrees (in AxSTREAM the incidence angle is defined as the metal angle minus the relative angle). Besides, re-profiling of the airfoil is performed, adjusting LE and TE wedge angles, stagger angle and unguided turning angle. The characteristic flow charts are simultaneously checked to avoid flow separation and the losses factors minimized.

The flow chart below shows the three curves considered to be the most useful for profiling design: the Buri criterion in green, the momentum thickness in purple and the velocity profile in blue [8]. The left-hand side in orange is the suction side (SS), while the green side on the right is the pressure side (PS). The LE is the white region at center, the dash line on the left represents the throat. The main criteria are that curves should be as smooth as possible, and the Buri criterion should be below 0.05 to avoid flow separation. A sudden peak in the Buri line represents reattachment.

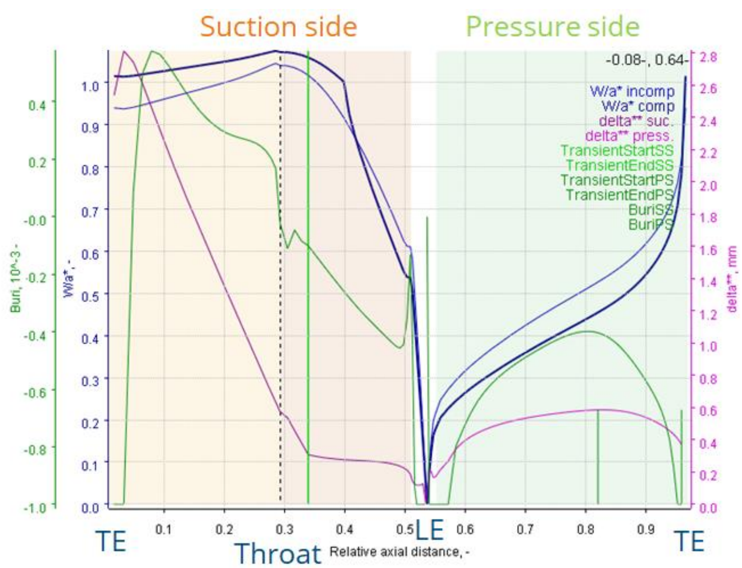


Figure 14 - Profiling flow chart stator example

Gauging angle

The gauging angle is the theoretical approximation of the outlet flow angle, which is the main parameter to be edited. The definitions for the stator and rotor gauging angles are:

$$Stator_{gaug.ange} = \arcsin\left(\frac{Throat_{nozzle}}{pitch}\right)$$

$$Rotor_{gaug.ange} = \arcsin\left(\frac{Throat_{blade}}{pitch}\right)$$

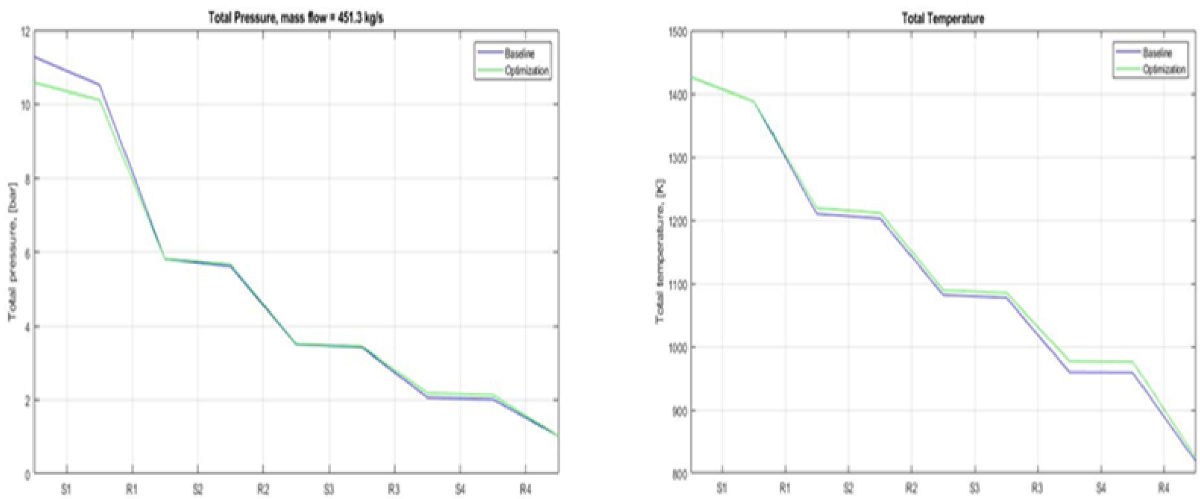
Equation 10 - Gauging angles

As the number of blades is constant, editing these angles influences the throat between nozzles and blades. If the throat is opened, the mass flow rate rises and/or the inlet total pressure drops. As the mass flow rate is a major constrain in design and has to be kept fixed, the problem formulation of the streamline solver is changed into 'Find inlet total pressure for given mass flow rate'. A small drop in inlet total pressure is then expected.

Results

Performance

The blades have been reshaped and the new model has been tested to compare perfomances. As predicted, the boundary conditions at the streamline solver are changed. Due to enlargment of the throat, following the gauging angles change, the inlet total pressure is reduced. Besides, new interstage pressure drops are established, as each stage has been edited.



a. b.

Figure 15 – a) Total pressure from inlet to outlet; b) Total temperature behavior

The inlet total temperature is kept equal to that of the baseline. As appreciable in the figure, the total temperature through the turbine is slighly higher than the baseline. As the outlet temperature is similar/higher than the baseline, it is still positive by considering the combined cycle option, since more heat is available for steam generation

The steady state calculations on the two turbine models, at design point, produce all data that are stored in the main project. The more characteristic parameters are listed in the following table (table 6).

Table 6 - Performances of the two models

	$\dot{m}_{in}\left[\frac{kg}{s}\right]$	$T_{0_{in}}[K]$	$p_{0_{in}}[bar]$	$T_{0_{out}}[K]$	$p_{0_{out}}[bar]$	$p_{out}[bar]$	$pr_{tt}[\sim]$	$P[\mathrm{MW}]$	$\eta_{tt_{cool}}[\%]$
Baseline	451.3	1426	11.28	816.18	1.013	0.90	11.14	320.58	88.71
Optimized	451.3	1426	10.58	827.15	1.013	0.89	10.44	317.01	90.03

The power output made by the turbine is reduced, but with an higher efficiency. Besides, the power net of the gas turbine has to take into account a lower power required at the compressor and to pressurize the fuel.

The Mollier diagram with average section values is compared (fig.16). The expansion in the turbine has a different behaviour than the conventional one, as cooling system is included. At stage 4, the expansion through stator (in blue) and rotor (in red) is without cooling representing a 'normal' behaviour. The optimized version evidences a larger drop in enthalpy, especially at the first and second rotor.

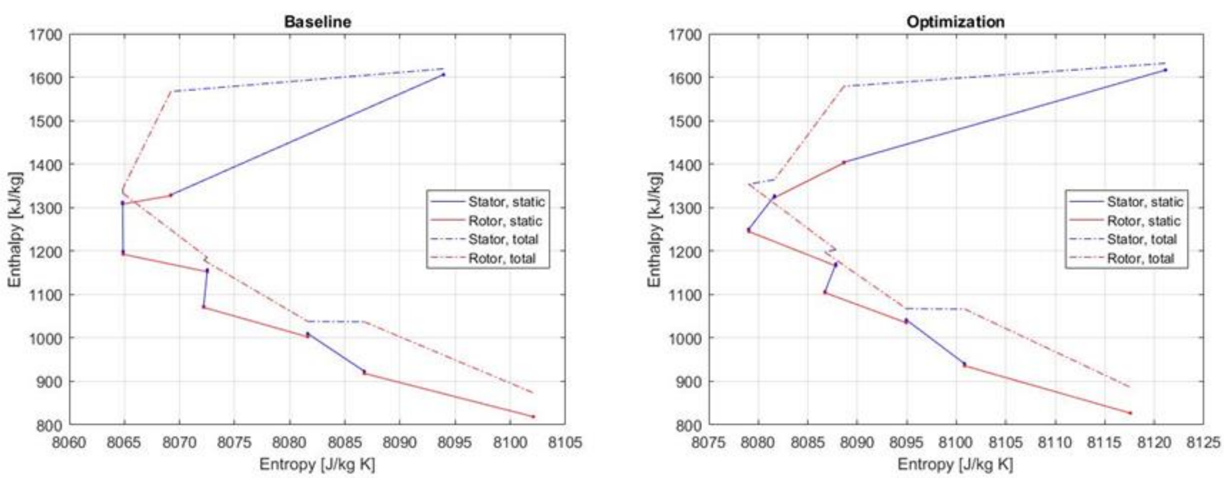


Figure 16 - Mollier diagram, turbine expansion comparison between baseline and optimized version

The enlargement in enthalpy drop at the rotor produces an increase in degree of reaction (fig.17). In the next figures, the spanwise distribution of reaction is shown for each stage. As expected, stage 1 and 2 present a higher reaction along the span. A more complex response is seen in stages 3 and 4, where the changes in outlet flow have been distributed differently along the span.

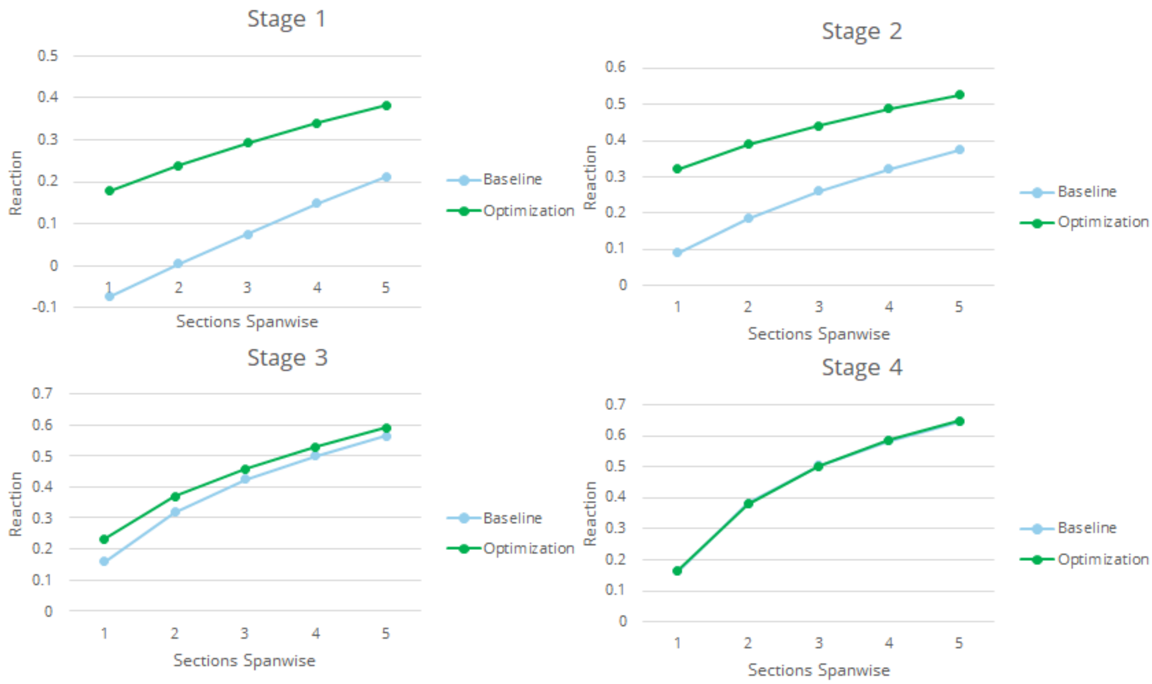


Figure 17 - Degree of reaction span-wise for each stage

The average values of reaction and cooled stage total-to-total efficiency against the stages are shown (fig.18). A different response for the first two and last two stages is observed. For stage 1 and 2, increasing the reaction leads also to an increase in stage efficiency, while the opposite was happening with stages 3 and 4.

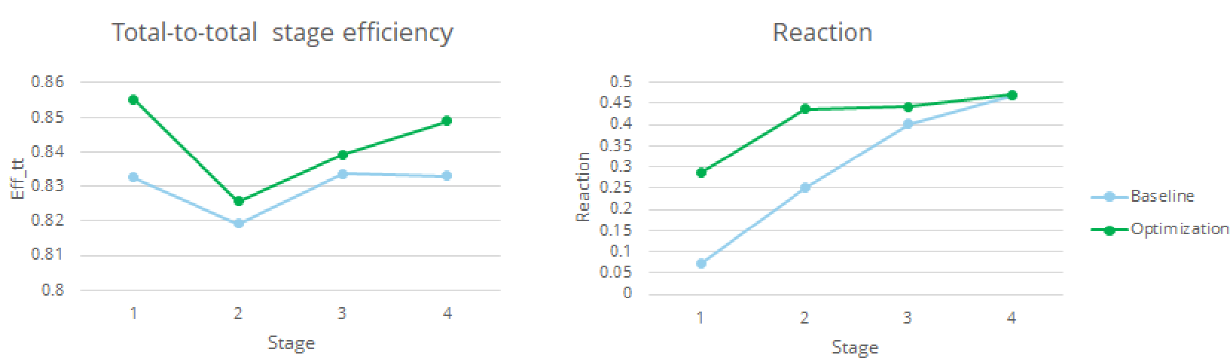


Figure 18 - Stage efficiency and reaction

The focus for them became to increase specific work and efficiency, to overcome the reduction in work extraction in the first stage.

In the following picture (fig.19), the magnitude in degrees of the change in gauging angle applied at every section is presented. The difference between the angles of the baseline and optimized geometries is given by the orange line. The first two stages have a similar distribution, with an almost uniform increase at the stator and decrease at rotor. For stage 3 a different distribution, based on results in stage efficiency and profile thickness constraint, is applied. Inversely, at stage 4 opposite changes are applied to allow a higher work extraction.

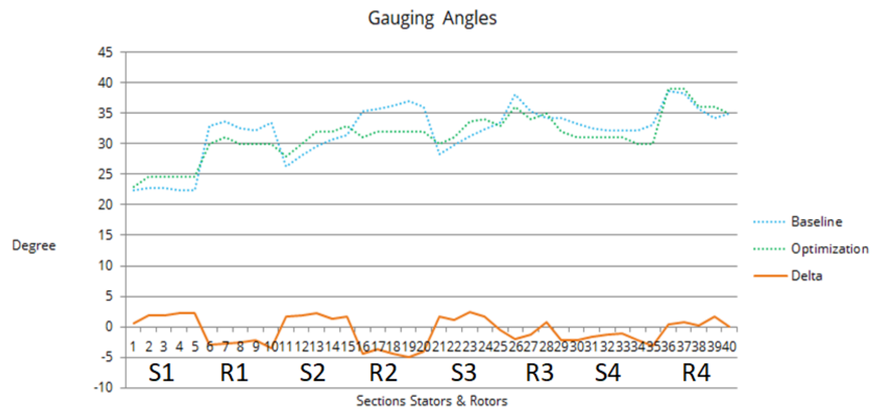


Figure 19 - Gauging angle distribution at sections

The specific work can be described by the following expression [20]:

$$W = (C_1^2 - C_2^2) + (U_1^2 - U_2^2) + (W_2^2 - W_1^2)$$

Equation 11 - Work expressed with characteristic velocities

All terms of difference in velocity, using sections average values, have been plotted to assess which value is dominant when looking at specific work extraction and degree of reaction (fig.20).

As mentioned earlier, for the relative velocity term, the difference is outlet minus inlet, while for absolute and blade speeds it is the opposite. This analysis evidences that specific work has been re-distributed; stage 2 and stage 4 are extract more work, and stage 1 and 4 less. This was expected for stage 1, as the idea was to reduce the work extraction here. Stage 3 on the other hand, was thought to be higher. Several attempts at stage 2 and 3 have been tried, but it seems that by increasing work extraction in one, it was reduced in the other . A sensitivity analysis is therefore strongly recommended to better understand the interaction between these two stages. As for stage 4, the new set of blades produces a large increase in specific work. By looking at the speed charts, the first two stages show linearity between absolute and relative velocity. Stage 3 seems to need more in outlet relative speed. Both relative and absolute terms are positively influencing the specific work at stage 4, which is significantly increased. Lastly, blade speed difference, usually constant in books, is much smaller and not playing a significant role.

Those data have been elaborated using Microsoft Excel and MATLAB, separately, slowing down the overall process. AxSTREAM ION is therefore strongly recommended when developing a toolchain for optimization, as through programming in C# allows connecting software to speed up the process [21].

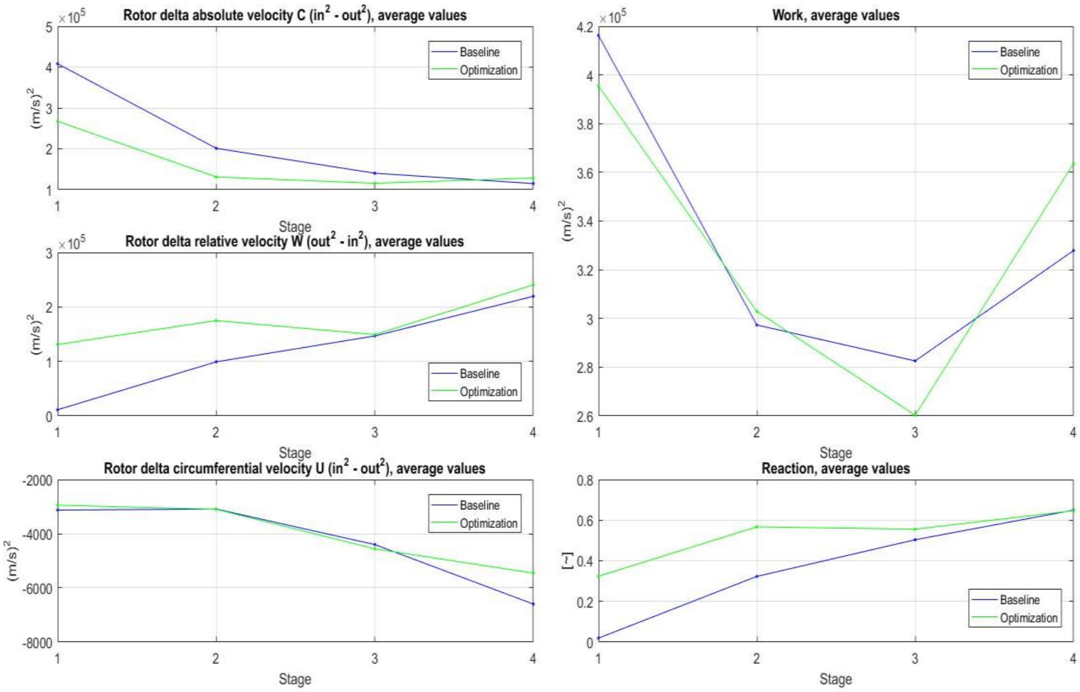


Figure 20 - Work distribution, velocity terms and degree of reaction at average values

Operational Condition

Turbine performance has been calculated for a set of operational conditions, varying the mass flow rate from 65% to 115% (fig.21). The efficiency reaches a maximum of around 95%. A constant positive shift in cooled turbine total-to-total efficiency is produced. The surplus in efficiency registered in AxSTREAM is around 1.3%.

On the right hand-side of the following figure, the power output is similar, even though the pressure ratio has been decreased, as the figure on the lower left corner is showing. For overloaded condition the efficiency would still be optimal. Increasing the mass flow would linearly enlarge the power output. Upgraded versions of the engine already use bigger mass flow with larger capacity [18]. The amount of inflow can be increased regulating the variable inlet guide vane (VIGV).

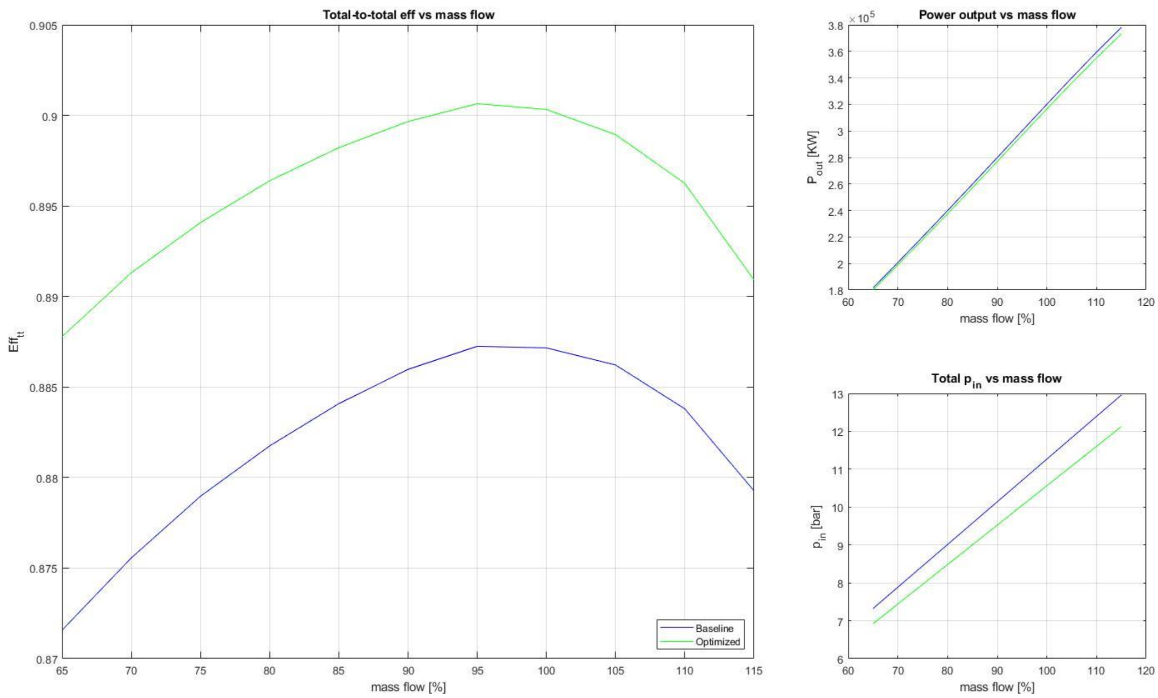


Figure 21 - Operational condition performance

Gas turbine overview

The turbine maps produced by AxSTREAM are tested in GSP. It has to be reported that the maps turned out to be different from the typical profiles of turbine performance. Since AxSTREAM is simulating real condition operation, the cooling system is included which probably influences the outcome. Therefore, the maps generated are not entirely reliable to be used in GSP. To have an overview of the gas turbine, these are used to perform design point calculation and make a comparison of the two versions. Two GSP configurations are presented here. First, the models are run with the single map turbine (table 8), without including the influence of the new map with the other components maps, like compressor and combustor. All maps and components are scaled to the design point using map scaling factors, which means that the hardware is different from the original. Design point calculation shows then a virtual gain of 1.1% in specific fuel consumption. This is achieved with a lower pressure ratio, and lower turbine inlet and outlet total temperatures. Besides, the exhaust area decreases as outlet total values changed and an ambient static back pressure is imposed at outlet.

Table 7 - Design point comparison in GSP using AxSTREAM maps

	$\dot{m}_{fuel}\left[\frac{kg}{s}\right]$	$\eta_c[\%]$	$\eta_t[\%]$	$T_{0_{in}}[K]$	$p_{0_{in}}[bar]$	$T_{0_{out}}[K]$	$A_{exh}[m^2]$	$P_{shaft}[\text{MW}]$	$SFC_{shaft}gain\ [\%]$
Baseline	9.09	88.5	88.7	1429.49	11.28	823.17	5.50	154	/
Optimized	8.99	88.5	90.0	1412.58	10.58	815.20	5.47	154	1.1

Then, the multi map turbine component is adopted. This allows considering how the new maps impacts with others original map of the other components (table 8).

Table 8 - Multi-map comparison in GSP using AxSTREAM maps

	$\dot{m}_{fuel}\left[\frac{kg}{s}\right]$	$\eta_c[\%]$	$\eta_t[\%]$	$T_{0_{in}}[K]$	$p_{0_{in}}[bar]$	$T_{0_{out}}[K]$	$P_{shaft}[\text{MW}]$	$SFC_{shaft}gain\ [\%]$
Baseline	9.01	88.4	88.7	1419.39	11.28	817.61	154	/
Optimized	9.29	88.0	88.8	1438.13	10.58	840.28	154	-3.1

The comparison in multi-map is made on fuel consumption to generate the same amount of power, including the impact of the new (AxSTREAM) maps on the other components, and the operating point of the turbine itself is estimated based on the interactions with the compressor. This includes turbine efficiency, which is approximately the same as in the baseline. Therefore, to conclude the turbine map made by AxSTREAM is not increasing the efficiency of the turbine in GSP. Hence, using a lower pressure ratio and due to the losses at the others component (caused by the new map), a higher fuel consumption is reported.

Last, it should be noted that the starting GSP project with the original maps was designed with a much lower capacity of 80 MW.

Machine axial load

The axial load applied to the disk is an important parameter when designing the turbine blades. The energy transfer has been maximized by changing the outlet flow angles. This creates an increased axial load to be transferred to the disk.

In the following table (table 9), the values of disk axial load (daf) at each rotor row and the machine axial load are listed for the two models.

Table 9 - Axial load comparison

	$R1\ daf(kN)$	$R2\ daf(kN)$	$R3\ daf(kN)$	$R4\ daf(kN)$	$Turbine\ A.L.(kN)$
Baseline	-77.8	53.9	43.7	28.3	821.9
Optimization	176.5	180	63.8	30.1	1444.5

The negative value at first rotor might be due to the negative reaction near hub. A similar response has been observed in AxSTREAM tutorial projects also, leading to assume a linear dependency between reaction and daf. Besides, it has to be noticed that the clearance values adopted are taken from assumptions, typical values from AxSTREAM training. In particular, the tip gap cold values are used at the rotor rows [14]. A more precise model is desirable to have the proper impact of axial load and losses in general.

In the following figure presents how machine axial load is influenced by the change in the outlet angles of stator and rotor at the first stage (same behaviour is observed for the others).

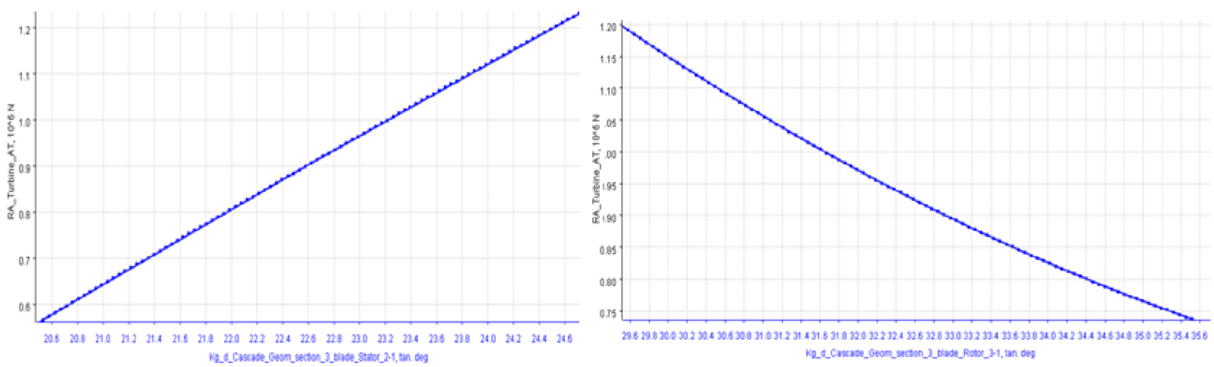


Figure 22 - a) Turbine axial load vs stator gaug. angle; b) Turbine axial load vs rotor gaug. angle

In the optimization, the outlet angle of the stator is increased, and the rotor outlet is decreased. As appreciable, the actions applied increase the axial load, which can reach really huge values. A limit can be imposed to respect operational constraints. The purpose of this assignment is to study the re-design of the aerodynamics of the blades. It is part of an iterative process, because the findings must be validated through CFD, structural and vibrational analysis.

Conclusion

An analysis of the design of the turbine blades and vanes of the Siemens V94.2(3) has been carried out to investigate the feasibility of aerodynamic optimization. This has been achieved through designing a toolchain using the AxSTREAM axial turbine software package and GSP.

A baseline model has been defined after matching of performance with experimental data. On that, the investigation regarding aerodynamic parameters of the blades exclusively has led to an optimal version of the turbine and gas turbine.

Following theoretical consideration regarding the work redistribution through the machine, the outlet flow angles, both of stator and rotor rows, have been edited. This was intended to lower the work extraction at the first stage and to increase the energy transfer at the following rotor rows. The problem formulation changed, leading to a lower pressure ratio configuration and an improved turbine cooled total-to-total efficiency of 1.3%. Steady state calculations, performed relatively fast at the 5 sections of each blade, have been performed to produce design and off-design solution to produce a performance map of the turbine. This was imported into GSP to assess overall gas turbine performance. The gas turbine model in GSP produced a virtual gain in specific fuel consumption of 1.1%. On the other hand, the right approach, considering the interactions between the new AxSTREAM map and the original software, shows a gas turbine efficiency decrease of -3.1%. However, in this case, the pressure ratio is lower and the turbine efficiency (estimated by GSP itself) is the same as the baseline.

Recommendation

The analysis performed at this third version of the engine evidences that an increased intake of air is recommended in order to restore a higher pressure ratio and produce an enlarged power output with higher turbine efficiency.

The clearances and the cooling system adopted for this project are based on typical values in AxSTREAM, and other design versions. In order to make the predictions in AxSTREAM as reliable as possible, every specification should be as close to reality as possible. The rotor tip gaps for example are assigned with half cold values, while the streamline solver does not considering the elongation of the blades. This can severely influence the machine axial load. Furthermore, attempts with different cooling systems have been tried, sometimes giving significant changes in performances. The first rotor tip cooling is higher than the TE cooling, while in other engine version the opposite is designed, just to make an example. AxSTREAM NET, another tool of SoftInway that focuses on the cooling system, could be integrated in the design process, giving a more precise model.

It is evident just by looking at the Mollier diagram (fig.16) that the cooling system influences the entropy generation and therefore needs to be included in the aerodynamic optimization of the blades. More clarifications are needed on use of the performance maps calculated with AxStream in GSP. The AxSTREAM maps produced results different from the original. It is not clear if maps are usually made including cooling. If not, the turbine should be analysed without cooling, integrated in GSP, and then the cooling should be added in GSP. However, this will be achieved with a significant loss in accuracy of the model. The interactions between the new maps and the other components are not completely understood yet. Furthermore, based on the working fluid, the primary losses model adopted in AxSTREAM is Craig&Cox corrected, and it has been tuned to match with experimental data. More clarifications on how this can impact results in GSP are advised.

Regarding the optimization, AxPlan was not able to perform DoE at every section of the blades simultaneously. Besides to process the data, MATLAB has been used, slowing down the overall process. Through a faster automatized chain, stage 2 and 3 sensitivity analysis might be performed to investigate a better work redistribution. The toolchain should be designed through process diagram automation, which enables the creation of a network of software, including a DoE tool, and that is capable of manipulating variables and objectives and elaborate on them. AxSTREAM ION can be employed to accomplish this target. The feasibility to employ DoE at every section simultaneously needs more clarification from SoftInway.

Machine axial load results show that by increasing the reaction, especially near the hub, the force transmitted to the disk can increase enormously. More specific constraints should be adopted in order to make the optimization feasible. The aerodynamic re-design of the turbine blades is only the first step, and part of an iterative process, and must be validated through CFD, structural and vibrational analysis. These tools exist in AxSTREAM and might be integrated.

List of symbols

A_{exh}	Turbine exhaust surface
af	Air excess factor
AFR	Air to fuel ratio
AFR_s	Air to fuel ratio stoichiometric
B_1	Cooling flow 1
B_2	Cooling flow 2
C_1	Absolute velocity rotor inlet
C_2	Absolute velocity rotor outlet
Δh_{Rot}	Rotor enthalpy drop
$\Delta h_{0,Stage}$	Stage total enthalpy drop
Δh_{Stage}	Stage enthalpy drop
Δp_{comb}	Combustor pressure drop
Δw_θ	Relative tangential velocity difference
η_{comp}	Compressor efficiency
$\eta_{tt_{cool}}$	Cooled turbine total-to-total efficiency
η_{turb}	Turbine efficiency
l	Air to fuel ratio
l_0	Stoichiometric air to fuel ratio
LHV	Low heating value
$\dot{m}_{cool}$	Cooling mass flow rate
$\dot{m}_f$	Fuel mass flow rate
$\dot{m}_{in}$	Inlet air mass flow rate
N	Rotational speed
Π	Pressure ratio
P	Power net
P_{ctrl}	Control Power (GSP)
P_{Shaft}	Net power gas turbine
pr_{tt}	Total-to-total pressure ratio
R	Degree of reaction
R_{Rotor}	Rotor degree of reaction
R1	Rotor 1
S1	Stator 1
$SFC_{shaft}gain$	Specific fuel consumption gain percentage
U_1	Inlet blade rotational speed
U_2	Outlet blade rotational speed
T_f	Fuel temperature
$T_{0_{in}}$	Inlet total temperature
$T_{0_{out}}$	Outlet total temperature
W_1	Inlet rotor relative velocity
W_2	Outlet rotor relative velocity

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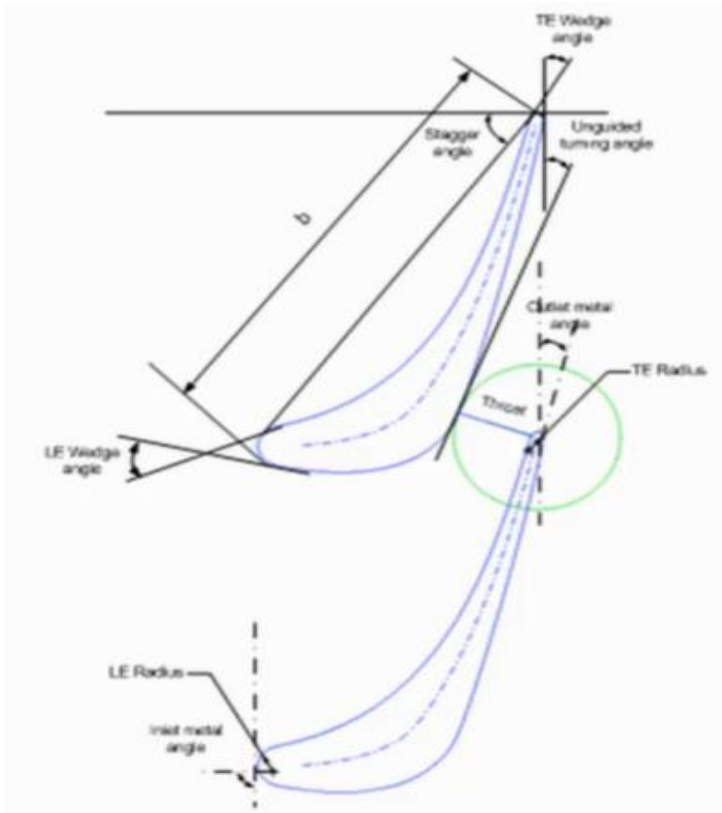
Appendix: Characteristic Angles Definition



- Inlet/Outlet metal angle - Is the angle between the extension of the chamber line and the tangential velocity, both at inlet and outlet.
- Stagger angle - Angle between the chord (b) and the axial direction.
- Incidence angle – difference between the inlet metal angle and the inlet flow angle (angle in absolute frame).
- Throat – It is the minimum distance between TE and suction side of a neighbouring profile
- Unguided turning angle – Is the angle between the tangent to suction side at throat point and the tangent to the TE on the suction side
- LE wedge angle – It is the angle between the common tangent to the suction side at the LE and the common tangent to the pressure side at the LE
- TE wedge angle – It is the angle between the common tangent to the suction side and the common tangent to the pressure side at the TE
- Gauging angle – is the theoretical approximation of the outlet flow angle for the turbine cascade.