Beyond the barrier of perfection

This article discusses the design optimization of an axial steam turbine (rated power of 160 MW), focusing on maximizing the total-to-total isentropic efficiency of the last three low pressure stages. The turbine, which is designed and produced by Franco Tosi Meccanica SpA, was optimized in collaboration with EnginSoft, by using a modeFRONTIER work flow to explore different designs and completely manage the fluid dynamics simulations. This simulation management involves geometry generation, model pre-processing, solving and post-processing, all achieved through the tools provided by ANSYS. Thanks to a strategic selection of input parameters, output values, targets and project constraints, a wide exploration of the possible parametric space is achieved by using efficient optimization algorithms.

Introduction

While everyone is familiar with the idea of boiling water for cooking or making a coffee, few consider that water has been boiled for nearly everything we do! Thanks to this, every day we can work at our computers, charge our smartphones or relax watching TV. Even if it sounds unusual, it's not too far from the truth. In fact, almost 80% of the electric power we consume comes from power plants that generate electricity from steam.

Hystory of Steam Turbine Technology

The first device to be classified as a steam turbine was the aeolipile, proposed by Greek mathematician Hero of Alexandria in the 1st century. Other steam-driven machines were described in the next centuries; in 1551 by Taqi al-Din in Egypt, in 1629 by Giovanni Branca in Italy and in 1648 by John Wilkins in England. No significant developments occurred until the end of the 19th century when various inventors laid the groundwork for the modern steam turbine. In 1884 Sir Charles Algernon Parsons, a British engineer, recognized the advantage of employing a large number of stages in series, allowing extraction of the thermal energy in the steam in small steps. The invention of Parsons' steam turbine made cheap and plentiful electricity possible and revolutionized marine transport and naval warfare.

After Parsons, a number of other variations of turbines have been developed that work effectively with steam. During the 1880s Gustaf de Laval of Sweden constructed small reaction turbines that turned at about 40000 RPM. From 1889 to 1897 de Laval built many turbines with capacities from about 15 to several hundred horsepower. Auguste Rateau of France first developed multistage impulse turbines during the 1890s. At about the same time, Charles G. Curtis of the United States developed the velocity-compounded impulse stage. One of the founders of the modern theory of steam and gas turbines was Aurel Stodola, a Slovak physicist, engineer and professor at the Swiss Polytechnical Institute in Zurich.

By 1900 the largest steam turbine-generator unit produced 1.2 MW, and 10 years later the capacity of such machines had increased to more than 30 MW. This far exceeded the output of even the largest steam engines, making steam turbines the principal prime movers in central power stations after the first decade of the 20th century. Steam turbines also gained preeminence in large-scale marine applications, first with vessels burning fossil fuels and then with those using nuclear power.

Despite the introduction of many alternative technologies in the intervening 120 years, nowadays it is estimated that more than 80% of the world's electricity is generated using steam turbine systems driving rotary generators. Steam to drive these turbines is raised by burning fossil fuels, mostly coal but also oil and gas (~65%), or by nuclear power (~15%). Less common thermal sources for steam generation are solar power and geothermal energy. Because of its ability to develop tremendous power within a relatively small space, the steam turbine has superseded all other prime movers, except hydraulic turbines, for generating large amounts of electricity and for providing propulsive power for large, high-speed ships. Today, units capable of generating up to 2 GW of power can be mounted on a single shaft.

How Engineers can get a better Steam Turbine Design?

Today, customers demand greater and greater performance. They continuously push to enhance the efficiency of steam turbines aiming to decrease CO_2 emissions from fossil power plants and to increase electrical power output from nuclear power plants. So suppliers are then asked to improve their designs, so steam turbine designers work on getting as much energy as possible out of the steam that is fed in by redesigning the turbine itself. Every day, they are practically asked to give an answer to the following questions: can the turbine be made lighter (so it spins faster) but still strong enough to withstand the heat? Can multiple stages be used to extract energy that would otherwise be wasted? Can heat losses be reduced (by insulating the machine)? What shape should the blades be and at what angle should they be made?
This article will focus on how Franco Tosi Meccanica and EnginSoft tried to give an answer to this last question.
Starting from Parson's concept, after more than a century of development, advances in blading design have contributed to improved steam turbine thermal efficiency. Considering that modern turbines' efficiency can reach values over and above 90%, it is clear that any further improvement is a very challenging task to accomplish.

Theoretical methods and experimental tests are very useful in predicting and verifying the performance of every new design or redesign. However, the classical approach of "trial-and-error" through many experimental tests is very expensive in terms of time and money, while it is also unable to identify how to improve performances exactly.
With the availability of large computer power and efficient numerical algorithm, CFD becomes an essential tool for engineers, enabling a wide variety of complex flow situations to be simulated, reducing the amount of testing required, increasing understanding and accelerating development. As a result of these factors, CFD is now an established industrial design tool, helping to reduce design time scales and improve processes throughout the engineering world.
Today the designer has to cope with 2 key challenges to compete in the market: competitive market sales targets and strict energy efficiency regulations. In this complex sales scenario the designer is thus focused every day in "raising the bar", knowing that a few percent increase in performance often makes the difference. This is why a tool must be able to give an accurate, reliable, and automated prediction of the fluid flow behavior in steam turbines to allow Franco Tosi Meccanica to gain a new competitive edge in the market. In this context ANSYS proves to be a high-fidelity CAE tool to match a turbine designers' needs.

Fluid dynamics optimization of a LP Steam Turbine

The last three stages of a low pressure steam turbine is the focus of this study (Figure 1 and Figure 2). The objective is to optimize the three statoric rows by maximizing the total-to-total isentropic efficiency of the device at a given operating condition.
The turbine blades' shape is defined by the position of 5 airfoils in the spanwise direction. A Bezier distribution for both stacking angle (for flow incidence control) and bowing angle (for flow separation control) is used to recreate the blade shape. Another variable is used for the number of blades in the statoric row. See Figure 3 and Table 1 for details.

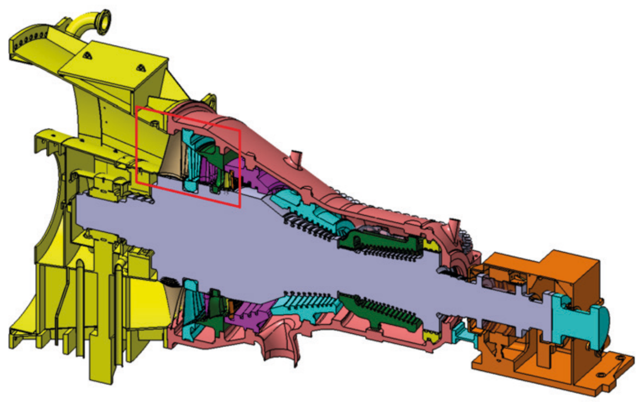

Figure 1 – Multi-stage steam turbine design by Franco Tosi Meccanica – The last three low-pressure stages are higlighted in the red box

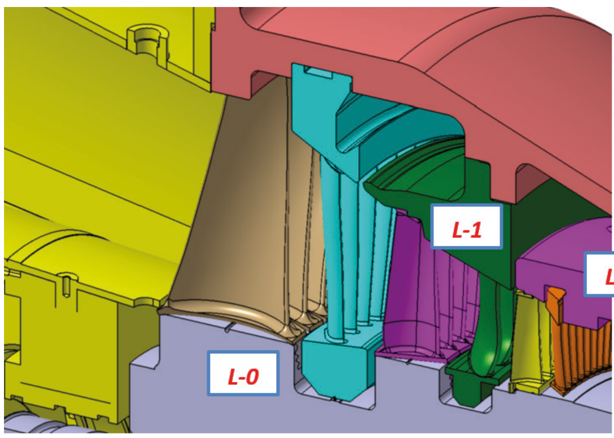

Figure 2 – Multi-stage steam turbine design by Franco Tosi Meccanica – Details of the last three low-pressure stages – Sealings and cavities

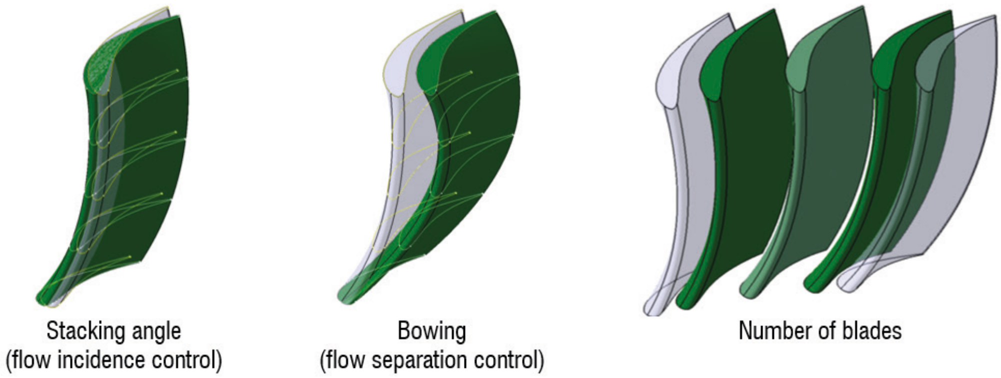

Figure 3 – Blade geometry – Description of parameters

A parametric model is generated with this characteristics in ANSYS DesignModeler. Starting from here, through ANSYS BladeEditor features, it is possible to extract the computational domain for a single passage. In fact, the first assumption made here is to consider just a single passage instead of a full wheel (Figure 4). The rotating speed compared to the mean stream velocity is such that a mixing plane approach is well suited. Consequently we obtain a huge advantage in terms of reduction in the computational costs, both for the size of the CFD model and for the steady state approach with mixing plane.

Another assumption made is the simplification of the actual geometry of the steam turbine: sealings, cavities, and rotor are neglected in the computational domain. This is required to make a completely automatic optimization workflow possible, in particular for the computational grid generation of ANSYS TurboGrid (Figure 5). A grid independency study is performed in order to apply the best compromise between speed and accuracy. The result is a very fast and robust procedure able to achieve a high quality mesh and a well-defined boundary layer treatment for every configuration which is identified by a unique parameter set, or design point.

		Minimum	Maximum	Step
STACKING ANGLE	Rotation angle of the profile sections around their centroid ($\Delta\theta$)[1]	-5.0°	+5.0°	0.2°
	Maximum difference between 2 consecutive sections		±5°	
BOWING ANGLE	Rotation angle of the profile sections around turbine axis ($\Delta\beta$)[1]	-3.0°	+2.0°	0.1°
	Maximum difference between 2 consecutive sections		1.5°	
NUMBER OF BLADES	Stage L-2	72	76	2
	Stage L-1	64	72	2
	Stage L-1	146	154	2

Table 1 – Blade geometry – Description of parameters

The result of the optimization is the geometry for three statoric blades. This generation depends on the simplified ("ideal") layout of the flowpath considered here. Then, after the optimization campaign, a final comparison between the "ideal" flow path and the actual one is performed in order to measure the losses due to sealings and cavities.

Another assumption made is to consider each stage independently from each other in the optimization. In this way the three stages, namely L-0 (the last one before the diffuser), L-1, and L-2, are treated separately in three different optimization stages. Boundary conditions for each CFD model are obtained combining 1D data supplied by Franco Tosi Meccanica and a preliminary analysis of the single passage, three-stage turbine performed by EnginSoft.

An additional analysis of the single passage, three-stage turbine is performed after each optimization stage. The baseline geometry of the statoric blade is replaced by the geometry that represent the result of the current optimization stage. In this way we want to verify if the new layout of the turbine performs better than the baseline configuration. Once this is assured, the optimization carries on to the next stage.

ANSYS CFX is used to set up and solve the CFD analyses. The fluid flow is considered in steady, compressible, and turbulent conditions. The advection term is resolved with "High Resolution" scheme (bounded 2nd order accuracy). The RANS 2 equations Shear Stress Transport (SST) is chosen for turbulence model. The IAPWS library is employed to characterize the steam as a real fluid. The liquid-vapor phase transition is considered in equilibrium conditions. The Stage (or mixing plane) approach is considered for multiple frame of reference (MFR). The numerical setup has been

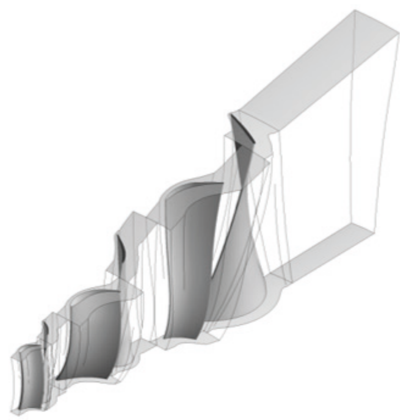

Figure 4 – Three stage, single passage – CFD model

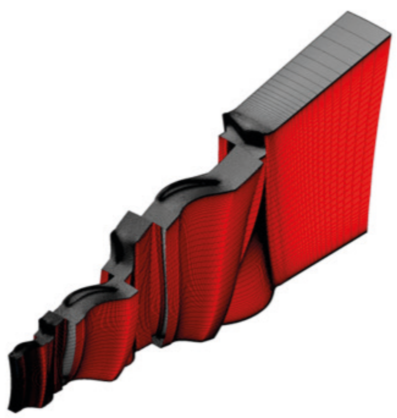

Figure 5 – Three stage, single passage – Computational grid

optimized in order to achieve a good level of convergence in less than 100 iterations. In this way, each design point selected in the optimization campaign took just about 25 minutes to be estimated (from the selection of the parameter set to the output of the post-processing procedure).

The results collected by the post-processing procedure are useful in understanding if the new design performs better with respect to the baseline configuration. The objective is to maximize the total-to-total isentropic efficiency:

$$\eta_{TOT-TOT} = \frac{h_{0\,TOT} - h_{1\,TOT}}{(h_0 - h_{1\,ISO}) + \frac{1}{2}({v_0}^2 - {v_1}^2)}$$

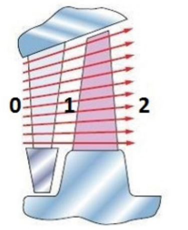

Figure 6 – Reference for post-processing

Several operating constraints have to be satisfied in order to guarantee the feasibility of a design point. In particular, such constraints are (see Figure 6 for reference). The choice of input parameters, output values, targets and project constraints greatly define an optimization process. Building from this point, several techniques can be selected defining what kind of optimization method to apply to an engineering problem is appropriate, especially in terms of time and cost.

		Minimum	Maximum
EXPANSION RATIO	$\frac{p_0}{p_2}$	-5%	+5%
REACTION RATIO	$\frac{h_1 - h_2}{h_0 - h_2}$	-10%	+10%
VAPOUR FRACTION	L-0	90%	
$\frac{\dot{m}_{vapor}}{\dot{m}_{total}}$	L-1	94%	
	L-2	100%	

Table 2 – Operating constraints (with respect to the baseline configuration)

Traditional engineering based on "trial-and-error" and was widely used in the past when automatic optimization tools were not available. Starting from a baseline configuration, the design is perturbed in order to get a new design point (hopefully with improved performance). The aspects of such perturbation is selected and applied by the designer, basically according to his experience. This process is repeated iteratively until the desired performance target is reached. It is clear that this approach cannot be completely automated, because decisions are made by the designer, and it is a time consuming and costly method. A schematic representation of this approach is shown in Figure 7.

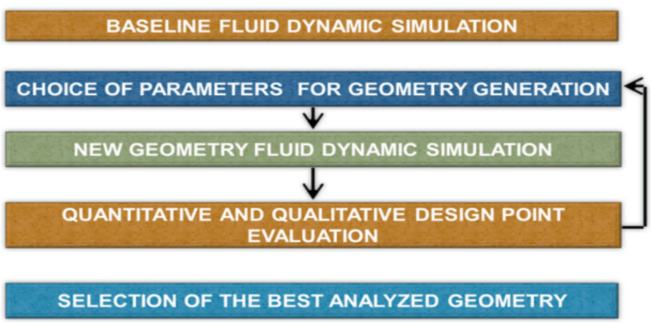

Figure 7 – Optimization methods – "Trial-and-error" approach

The evolution of this concept lead to the birth of modern optimization approaches, where algorithms took the place of the designer in the selection of the new design points to be evaluated. By means of efficient optimization algorithms, a wide and intelligent exploration of the parametric space can be performed in a completely automated way in much less time compared to traditional "trial-and-error" approach. Two different approaches are available:

- direct optimization: after a first exploration of the parameters' space through a DoE (Design of Experiment), each of the design points is explicitly evaluated, aiming to reach a desired target. See Figure 8;

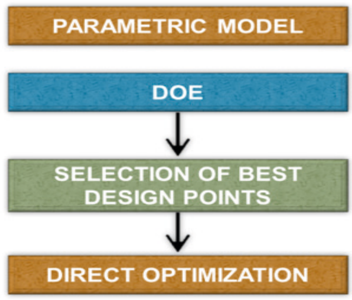

Figure 8 – Optimization methods – Direct optimization approach

- virtual optimization: after a first exploration of the parameters' space through a DoE, a response surface is generated in order to have a continuous representation of this space. In turn, this response surface is explored with a large number of virtual design points (not explicitly simulated) in order to find good candidates. Then, these good candidates are explicitly evaluated, enriching the database on which the response surface can be redefined. This process is repeated iteratively until the possible inaccuracy of the response surface goes below a certain threshold. See Figure 9.

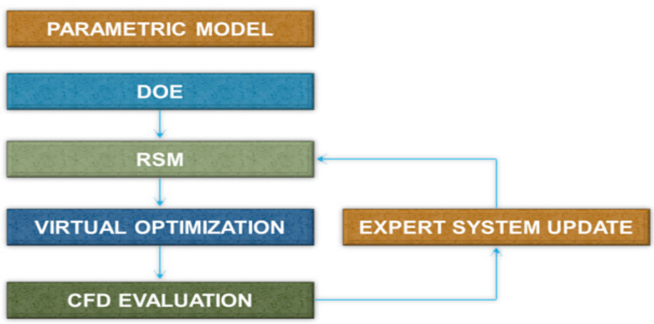

Figure 9 – Optimization methods – Virtual optimization approach

For direct or virtual optimization approaches we need an optimization software that is able to control the workflow that automatically manages both the exploration of different design points and the management of the fluid dynamics simulations, such as geometry generation, model pre-processing, solution and post-processing by means of the CAE tools provided by ANSYS. The software employed for this purpose is ESTECO's modeFRONTIER.

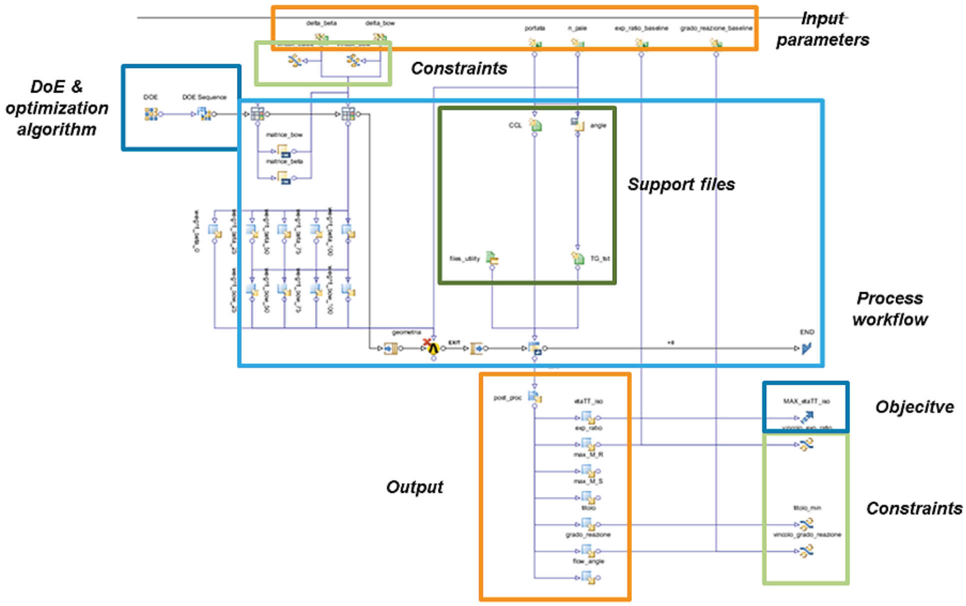

Figure 10 – ESTECO modeFRONTIER – Optimization worklow

Stage	Optimization approach	Optimization algorithm	Performed by	Number of explicit design points evaluated	Human time consumed
L-0	"Trial-and-error"	N.A.	Franco Tosi Meccanica	~ 100	~ 1 month
L-1	Virtual	MOGA-II + RSM	EnginSoft	~ 1000	~ 1 month
L-2	Direct	MOGA-II	EnginSoft	~ 5000	~ 1 month

Table 3 – Optimization strategy

Stage	Single stage improvement	Improvement of the three-stage turbine
L-0	+1.0%	+0.4%
L-1	+1.0%	+0.6%
L-2	+0.9%	+0.7%

Table 4 – Results of the optimization campaign

modeFRONTIER's optimization process could be split into 3 steps:

- the first step includes the creation of a logic workflow in order to graphically formulate the engineering design problem at hand, i.e. how the simulations have to be performed and in which order. The "how" implies the choice of values/measures to be used and generated (inputs and outputs), the definition of the optimization objectives and the configuration of the most adequate algorithms for design space exploration and optimization. A representation of the optimization workflow defined for this study is shown in Figure 10;
- the second step consists in the evaluation of designs, as defined by the workflow. The evaluation, or "run", can be monitored in real-time by means of charts and graphs, and direct access to log and process files;
- The final step is the assessment and visualization of results. The available tools allow understanding of a problem's important parameters on the basis of the design space exploration, reducing the number of significant parameters considered making the optimization more efficient, re-arranging data in a comprehensible manner and extracting a clear meaning in order to make informed decisions. Specific analysis tools help convey relevant insights on the interaction effects and visualize optimization trends. The RSM (Response Surface Models) tool allows for the training, comparison and validation of meta-models, speeding up the entire optimization process.

In this study, all the three different approaches have been used, as summarized in Table 3.

In the classical "trial-and-error" approach, all the design points (almost 100) have been explicitly evaluated by Franco Tosi Meccanica designers on their local workstations. This optimization stage took about one month to be accomplished.

For the optimization stages in which L-1 and L-2 were studied EnginSoft adopted an automatic optimization procedure by means of modeFRONTIER™. MOGA-II (Multi-Objective Genetic Algorithm) is the optimization

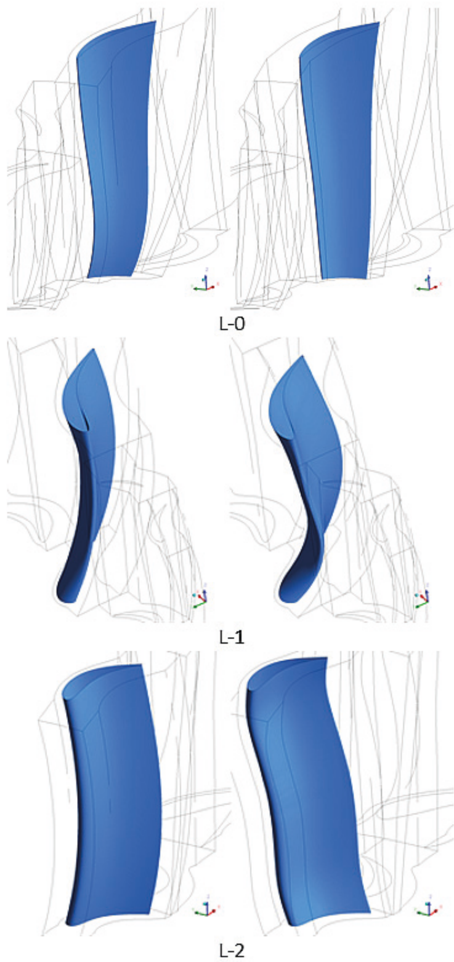

Figure 11 – Comparison between baseline (left) and optimized (right) design of the statoric blades – Blade geometry

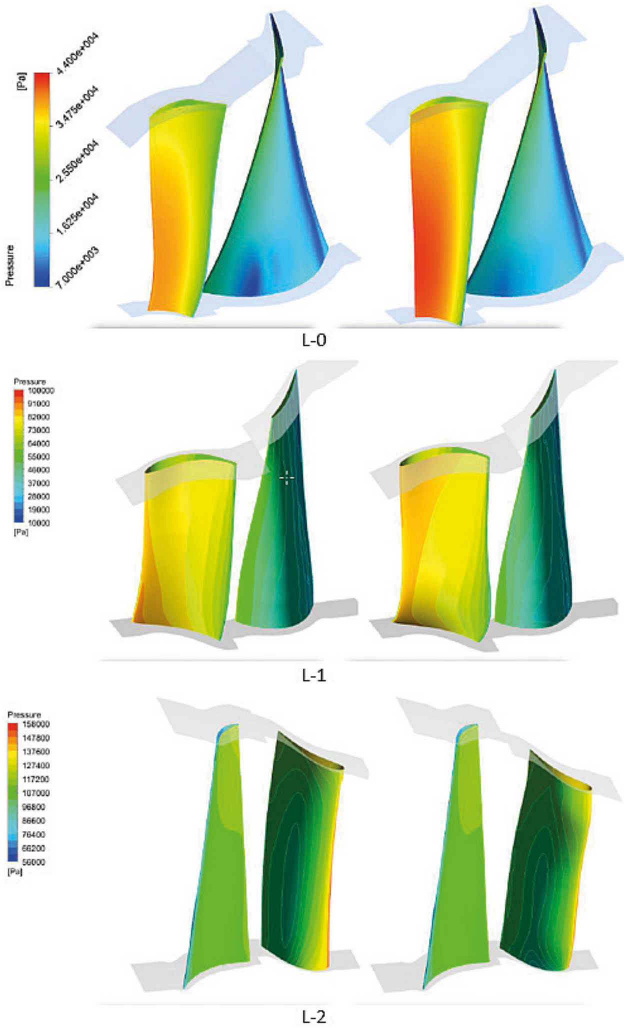

Figure 12 – Comparison between baseline (left) and optimized (right) design of the statoric blades – Pressure field on blades

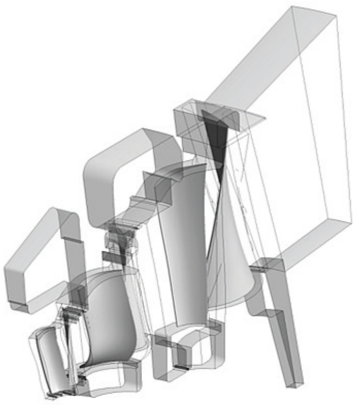

Figure 13 – Three stage, single passage – CFD model with actual sealings and cavities

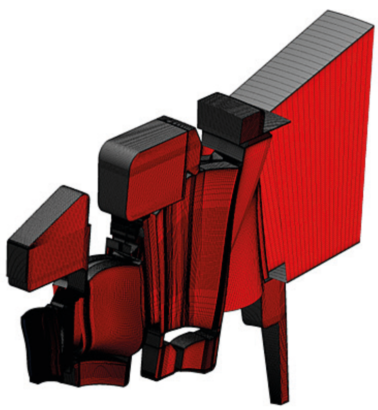

Figure 14 – Three stage, single passage – Computational grid with actual sealings and cavities

algorithm selected for this campaign. The optimization stage for L-1 required the evaluation of about 1000 design points, while for L-2 the CFD calculations performed were about 5000. All these computations were performed on the EnginSoft cluster. A concurrent design points strategy has been adopted to reduce the total wall clock time, so both the optimization stages took about one month to be completed.

The results of the optimization campaign is summarized in Table 4. The second column represents the improvement of the total-to-total isentropic efficiency evaluated on the respective turbine stage. In other words, these are the results of the three optimization stages. The last column represent the results of the additional analyses of the single passage, three-stage turbine that are performed after each optimization stage. As mentioned before, in these analyses the baseline geometry of the statoric blade is replaced by the geometry that represent the result of the respective optimization stage.

Even if the optimization stages are independent from each other, it is clear that a good overall trend is achieved. It is good to highlight that the efficiency of the baseline design is quite high (above 90%). From this point of view, such results are very remarkable.

The last phase of the study was the evaluation of the single passage, last three-stage steam turbine with actual flowpath, i.e. including sealings and cavities (see Figure 13 and Figure 14). A comparison between baseline and optimized geometries of the statoric rows has been performed. In real conditions, the improvement of the total-to-total isentropic efficiency is about 0.5%.

Conclusion

The object of the study was to optimize the last three low pressure stages of a steam turbine. A global optimization has been performed on such system. Acting on the geometry of the statoric blades, the purpose was to find the optimal performance in terms of isentropic total-to-total efficiency.
The optimization strategy adopted here was defined on 3 different stages with different approaches:

- "Trial-and-error" for L-0
- Virtual optimization (by means of response surface methods) for L-1
- Direct optimization for L-2

Once the optimal design for each stage has been selected, it has been fit into the three stages steam turbine in order to confirm the improvement of the system.
After the 3 optimization stages, a final verification of the steam turbine was performed, taking into account the real flow path with sealings and cavities. In this complex scenario, the isentropic total-to-total efficiency gain is about 0.5%.

It is worth noting that, as summarized in Table 3, each optimization stage requires the same amount of human engineering time (~1 month). This time includes all the CPU time and man hours involved. In the last optimization stage, the engineer's time is almost close to zero while the simulations are run automatically. Given that each design takes about 25 minutes to run on the

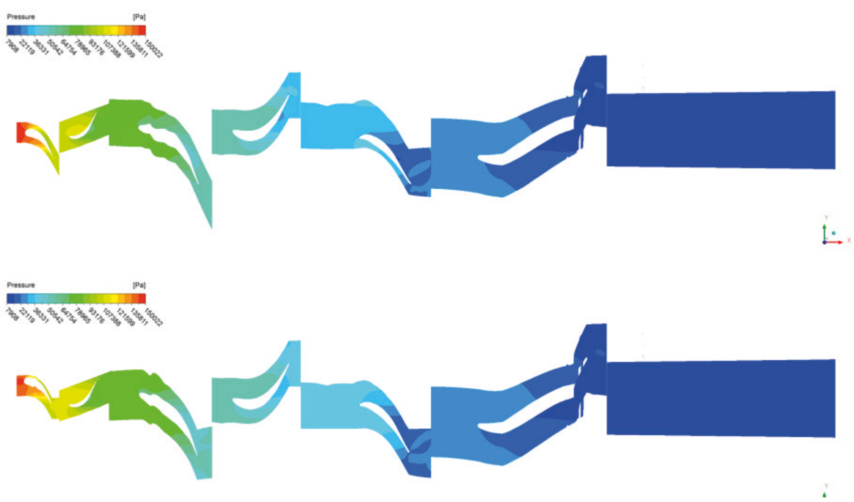

Figure 15 – Three stage, single passage – CFD model with actual sealings and cavities – Static pressure field comparison @ 50% span – Baseline (above) and optimized (below)

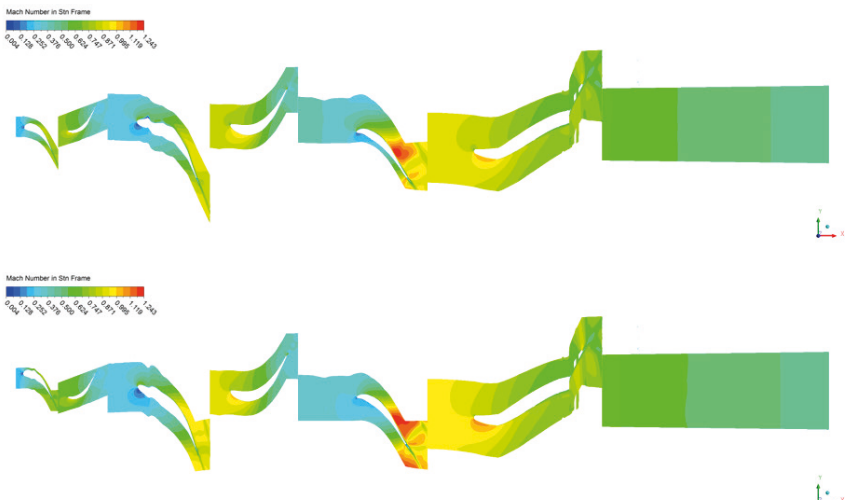

Figure 16 – Three stage, single passage – CFD model with actual sealings and cavities – Mach number field comparison @ 50% span – Baseline (above) and optimized (below)

EnginSoft cluster, and 3 concurrent design points were running simultaneously during the optimization stage, the total CPU time is about 1 month.

Now, in a typical medium enterprise, where HPC computing clusters of over a hundred CPUs are now quite common, it is clear that such times could be dramatically reduced. For example, for a medium installation of 256 cores, the same job could be performed in just 1 week! This is a remarkable result, especially if we think about the time that can be saved in the development of a large machine like the one considered in this study.

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The Fortissimo Project

Fortissimo is a collaborative project that enables European SMEs to be more competitive globally through the use of simulation services running on a High Performance Computing cloud infrastructure. The project is coordinated by the University of Edinburgh and involves 123 partners including Manufacturing Companies, Application Developers, Domain Experts, IT Solution Providers and HPC Cloud Service Providers from 14 countries. These partners are engaged in 53 experiments (case studies) where business relevant simulations of industrial processes are implemented and evaluated. The project is funded by the European Commission within the 7th Framework Programme and is part of the I4MS Initiative.

Fortissimo Booklet and Case Studies

The Fortissimo booklet contains a collection of 15 case studies from the first wave of initial experiments. They demonstrate the wide verity of HPC-cloud solutions and the impact these solutions had on the business of manufacturing SMEs as Fortissimo experiment partners. You can download your free copy of the Fortissimo booklet at:
http://www.fortissimo-project.eu/
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