44TH TURBOMACHINERY & 31ST PUMP SYMPOSIA
HOUSTON, TEXAS | SEPTEMBER 14 – 17 2015
GEORGE R. BROWN CONVENTION CENTER

INTRODUCTION OF A NEW FAMILY OF STEAM TURBINE LOW PRESSURE STAGES – A COMPREHENSIVE REVIEW OF 10 YEARS OF EXPERIENCE IN DESIGN, MANUFACTURING, EXPERIMENTAL VALIDATION AND FIELD APPLICATION

Lorenzo Cosi
Lorenzo Cosi
General Electric Company, GE Oil&Gas
Via Matteucci 2, Florence, 50127 , Italy

Lorenzo Cosi is the manager of the Steam Turbine New Product Introduction group at GE O&G in Firenze. His organization is responsible for the design, validation and industrialization of the new steam turbine components. Lorenzo Cosi holds a master degree in mechanical Engineering from the University of Firenze. He is also a member of the ASME Turbo Expo Steam Turbine committee where he is promoting the collaboration between academia and manufacturers in the development of Industrial Steam Turbine technology.

ABSTRACT

Low Pressure stage design characteristics play a key role in steam turbine product line as they typically set limits to the maximum turbine flow and rotating speed and have a strong influence on the overall turbine efficiency.
LP stages design presents a combination of structural and aerodynamics challenges that oblige the manufacturers to develop and test the design well in advance with respect to the turbine design schedule.
For the same reason (development complexity) an LP stages design is usually scaled to cover a wide range of rotating speed and annulus area. The set of scaled LP stage is referred to as LP stage family.
10 years ago the author's company has started the design of a new family of LP stages for power generation and mechanical drive application. The initial phase of the development has focused on the design, manufacturing and experimental validation (wheel box test and full scale turbine test) of the master size for the reaction product line. Then master size design has been scaled to generate a wide family of LP stages. In parallel to this activity the application of this LP stage family on the impulse product line has been carried out leading to a hybrid rotor design. Eventually the dynamic behavior of the scaled blades design has been experimentally validated employing a combination of strain gages and tip timing frequency test.
This paper will present a comprehensive review of all the phases of development and the major experimental results on the master and on the scaled designs with the intent to share the key technical aspects involved in the development of this components.

INTRODUCTION

Hystorical trends shows that most of the Oil & Gas industries in which Mechanical Drive steam turbines are applied follow a continuous increase of size driven by the economy of scale effect. A practical example of this is the Ethylene and Fertilizer markets. Plant capacity for production of Ethylene is following a trend which began in the 60's. From initial values of 200-300 kton/y plants are now approaching outputs between 1.5 and 2 Mt/y. The global demand for fertilizer, specifically the synthesis of Ammonia and Urea, has seen a large increase in the last 25 years and the capacity of these plants has dramatically increased from the original 200-300 tons per day to the current daily production of 2000 tons per day. This plant size growth along with the continuous research of better trade-off between investment and operating cost is pushing the turbomachinery toward larger sizes, higher component efficiency and higher rotating speed (the so called power density approach). Speed increase, is beneficial in turbo compressor applications because the impeller diameter and the number of compression stages can be reduced keeping the same compression efficiency
On the industrial Power Generation industry side energy conversion efficiency, initial investment cost and flexibility are the main drivers of the technology development. Rotating speed increase (for a given annulus area) is an advantage also for power generation application. Higher peripheral speed allows increased enthalpy drop on the upstream stages while maintaining the optimal stage aerodynamic load coefficient. The result is a 10-20% of stage reduction depending on the steam path selection.
Industrial steam turbine customers (both in MD and PG applications) demand for a customized product capable to fit into their plants closing the steam balance cycle. For this reason in this market a full standardization of the product it is not possible and different approaches to optimize the performance and keep delivery time and cost under control has been followed in the ST Industry.
Steam turbines manufacturers follows a standard/custom mixed approach in which some components are optimized on the project basis (within a pre-determined design space), and other components are selected from standard libraries. For example HP/IP stages are optimized on the basis of the customer cycle data targeting the optimal flow and work coefficients and

keeping mechanical design parameters within a well experienced design space. This is usually done with automatic procedures which iterate between aero design and mechanical verifications. A similar approach is followed for the outer casing design. In this case the casting is customized composing standard patterns therefore for each project a different and optimized casing can be selected. On the contrary LP stages are part of the standard components families because as well described by Gyarmathy their development present many aerodynamic and mechanical design challenges that require a quite extensive design and testing activities which would not be compatible with the standard projects schedule.

Usually the LP section rotating speed capability limits the capability of the turbine because last stage blades have the highest A·N^2. Also in terms of efficiency LP stages have a significant impact on the entire turbine since they deliver from 20% to 40% of the power depending on the steam balance of the machine. Therefore an improve in LP stages capability and performance has a significant effect on the entire turbine but to fully exploit its benefit the match between the new section and the rest of the product structure has to be carefully taken into account since the beginning of the project.

To respond to the industry requirements of increased rotating speed and higher efficiency, 10 years ago the author company has started the development of a new generation of Low Pressure blades for MD and PG applications. The new design features a 60% increase in A·N^2 and a 6 points increase in section efficiency with respect to the traditional design. This new generation of blades has been scaled in order to produce a given number of sizes and thus to cover a wide range of exit annulus areas. The overall family is ranging from the 8'' (height of the last rotating blades) running at 11250 rpm to the 25'' last stage blade running at 3600 rpm and suitable for the 60 Hz power gen market.

DESIGN SPACE DEFINITION

The operating space of the last stage blade can be represented on a diagram with an x-axis as condenser pressure a y-axis as End Load (defined as mass flow divided the annulus area at the exit of the rotating blade), and a z-axis as rotating speed. For a given exit enthalpy the axial exit annulus velocity (VAN) is a function of end load and condensing pressure and can be represented on the diagram as a line. Depending on the application (Power Generation or Mechanical Drive) the last stage is demanded to operate in different parts of this diagram. In Power Generation where energy conversion efficiency needs are dominant, the turbine – condenser system is usually designed for low condensing pressure in a range of 0.07-0.15 bara, which result into an end load around 10-20 kg/s/m2. On the contrary Mechanical Drive ST's are usually operated at higher end load to reduce the size/cost of the turbine and of the condenser. Plotting these operating points on the condensing-pressure - end load diagram (see Figure 1) the difference between power generation and mechanical drive applications can be well illustrated. This difference in the required operating conditions has a significant impact on the design of the last stage blade. Basically a more robust and "damped", blade is required to operate in the high backpressure and end load region because it will be developing at higher power and eventually will be subjected to higher mechanical loading.

When considering the rotating speed, the difference between mechanical drive and power generation applications is even more remarkable. In power generation, the turbine has a single operating speed and during the start-up it is brought to synchronization speed at no load (very low mass flow). On the contrary in mechanical drive applications, the turbine must be able to operate in a speed range that is usually from 70% to 105% of the compressor rated speed. During start-up, the load increases as a cubic power of the rotating speed. This difference has a significant impact in the aeromechanic design of the last stage blade. Power generation blades have historically been designed to operate off-resonance. In mechanical drive applications, this is not possible because of the large speed range required, hence the design of the blade is aimed to provide an adequate level of damping so that even on resonance the vibratory stresses are low and sufficient margins on the Goodman diagram are guaranteed.

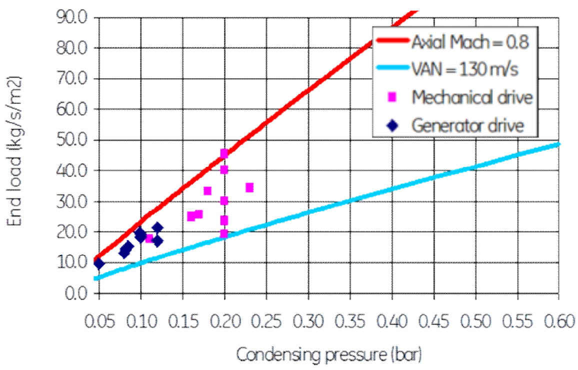

Figure 1: Map of typical applications

HS stages were designed to cover the full envelope of mechanical drive and power generation applications. They are capable of variable speed, high backpressure and high end-load operation. Flexibility and robustness implemented for MD applications can be an advantage also in PG applications. For example, high backpressure capability allows for continuous operation even in plants equipped with air-cooling condensers where operation at high backpressure in some period of the year can be required.

MASTER SIZE DESIGN

The smallest element of the family (HS8) was selected as master design taking into consideration the following aspects:

- Smaller components are more sensitive to

manufacturing tolerances therefore scaling up it is always a safer and easier process
- Smaller components would have allowed reducing the procurement time for hardware manufacturing and validation
- Given the high end-load design specification of this family full load testing requires such a high steam flow that choosing the minimum size would have contributed to contain testing cost.

Low Pressure stages design process is based on iteration between aerodynamics and mechanical design. A summary of the design process adopted at each iteration is reported in the following paragraphs.

Aerodynamic Design

The last four stages are designed aerodynamically to work together as a system using a combination of streamline curvature design methods, two-dimensional cascade analysis, and state-of-the-art three-dimensional computational fluid dynamics analysis techniques. This four stage group has been designed to meet up with an intermediate pressure section whose inner diameter is considerably lower than the exit diameter of the last stage blade. This design strategy allows for the retrofitting of existing LP rotors, while providing maximum work output from the LP section. Since power density is an important feature, the stage spacing were limited to maintain close wheel spacing while allowing for a high recovery exhaust diffuser configuration.
The last, or L-0 stage, is the first in the four stage group to be designed. The inherently wet steam conditions, high exit mach numbers and variable pressure ratio seen in the application of this stage makes the design most challenging. Of considerable importance is the requirement to ensure that the pressure ratio across the hub section of the last stage blade remain greater than unity for an extended operating range, which includes potentially high exhaust pressures as well as possible applications at a variety of rotational speeds for a given scale factor. The velocity vectors for the new design were determined using the following process. An optimization was performed using a through-flow analysis code, whereby profiles of nozzle and blade discharge angles were varied, as well as the mean level of blade average work, and geometric parameters of airfoil height and diameter. While a free vortex angle distribution was used as an initial guess, the optimization process allowed for several non-free vortex or ‘forced vortex’ distributions to be examined. The resulting steampath shape is provided in Figure 2. For the final blading, extensive studies were carried out using 3D CFD models to determine the optimal configuration and airfoil shape definition.
Also the exhaust hood diffuser was subjected to a 3D CFD optimization process in design and off-design conditions which is described in Verstraete, T. et al.

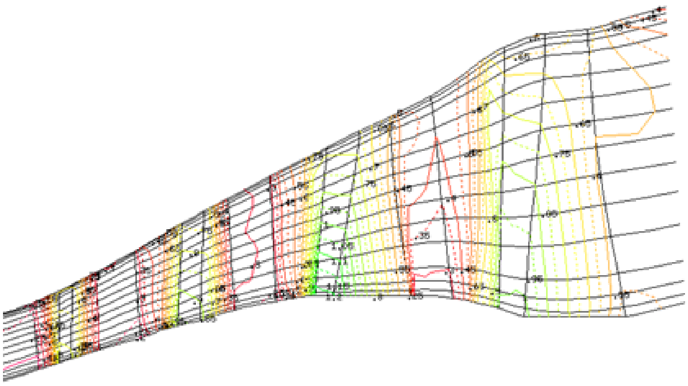

Figure 2: Streamlines distribution

The L0 stage is placed at a lower inner diameter to the L1 stage to encourage the radial flow of steam into the root of the L0 blade, thereby delaying the separation of flow down to lower exit velocities with improved operability. The L-0 nozzle also employs complex tangential lean to force the flow into the root section, while axial lean is used to control the nozzle to blade spacing in minimizing the potential for erosion. The L0 nozzle is contoured to minimize incidence losses across a wide range of application speeds.
The higher radial position of the L1 blade root allows for a lower work coefficient on this stage while still producing a considerable amount of output relative to the other front stages. Increasing L1 blade tip diameter and applying forced vortexing to the stage minimize diffusion between the L1 blade and L0 nozzle. This combination allows for a lower stage work coefficient and higher efficiency, higher reaction blading than would be obtained using a traditional design. The L-1 stage radial throat distributions are specially designed to provide optimal profiles of total pressure and enthalpy entering the last stage nozzle. Like the L0 nozzle, the L1 nozzle is contoured to provide optimal efficiency across a range of application speeds.
The steampath encompassing the L-2 and L-3 stages is designed to mate a lower diameter High or Intermediate Pressure section with the higher diameter of the L-1 stage, while minimizing losses in kinetic energy between blade rows. Secondary flows are reduced through appropriate blade count and flow coefficient selection. The bladed ring (BLING) technology used in the construction of the diaphragms allows obtaining a better vanes fidelity to the design intent compared to traditional assembled or welded technology. This geometry accuracy results in an improved control of the actual performance of the turbine. Multiple seal teeth are employed for all nozzle and blade sealing surfaces, resulting in minimal leakage flow and improved overall performance.
Three-dimensional CFD calculations performed on the newly designed blading confirm a considerable advantage in the performance over similar existing designs. The use of the

numerical test bed allows for investigation into the performance of the new design over a wide range of corrected speeds, Reynolds numbers, and moisture levels. The larger annulus area and higher speed capability further increases the application space of this design, resulting in higher unit output.

Rotating Blades Mechanical Design

The mechanical design process consists of a few interconnected steps: transforming the aerodynamic shape of the airfoil into the as-machined one; static design; dynamic design (or aeromechanics design) and erosion design. This section will describe those design steps. The design process for this blading builds upon the process reported in Mujezinovic et al. [1], with special design calculations and testing to ensure robustness through a greater range of applications.

The airfoil shape used in the aerodynamic calculations is the one that the airfoil will assume at the running speed. The low-pressure blades untwist during acceleration from rest to running speed. A necessary step during mechanical design is to follow an iterative process that seeks a zero-speed airfoil shape, which yields a shape nearly identical to the aerodynamic shape at the operating speed. The process is accomplished by using a computer-implemented finite element method. Since the blades have integral covers in this particular design, the maximum cover untwist is limited by a cover-to-cover gap at the assembly.

The basic premise in the static design of the blade and wheel is that both the maximum average stress in all the wheel and blade sections—and the maximum local stress throughout the blade and the wheel—is maintained under a certain predetermined level. The maximum average stress (assuming fixed material properties) determines the overspeed margin: when compared to the yield strength of the materials (blade and wheel) it determines overspeed at which section gross yielding would occur. When compared to the ultimate strength, it determines overspeed at which the ductile failure would occur. Maximum local stress determines the low cycle fatigue life (number of cycles to a crack initiation) and also plays a role in the level of stress corrosion cracking (SCC) risk.

The average stresses in blade and wheel can be easily calculated using the zero-speed blade geometry and computer-implemented finite element method. A relatively unrefined finite element mesh is sufficient for such a task. However, concentrated stress calculations— specifically in a dovetail region—require a very refined finite element mesh in all three dimensions, appropriate boundary condition between the rotor and the blade and the appropriate material model.

The simplest approach would be a coupling between mating finite element nodes on the rotor and blade as a boundary condition, and linear elastic material model. However, to model the actual problem with more fidelity a non-linear analysis was employed with contact between blade and wheel modelled and an elastic plastic material model used. In addition, to obtain a finite element mesh of sufficient refinement and reasonable calculation time, sub-modelling of the dovetail locations with the highest local stresses was employed. The local stress calculated in such a way was also used in a fracture analysis. Fracture analysis methods traditionally used in aircraft engine designs have been employed to show that the blade/wheel configuration meets an acceptable number of cycles before any potential crack would grow to an unacceptable size.

The overspeed capability and life parameters calculated by using the described method are also compared to the same parameters in the previous successful designs. In a case when a sophisticated analysis was run for the first time on this particular design (as it was a case for a fracture analysis) the previous designs were also analyzed and the results were compared. In all instances, this design compares favorably to the previous ones.

In variable speed application, the crossings between blade natural frequencies and a number of multiples of engine speed (so-called per rev lines), will occur in the operating range. Per-rev excitations come from different sources, most of which are associated with the non- uniformity in the steam path, and have different stimulus intensity. Experience collected in field and test turbines allows classifying the stimulus intensity of the different per rev lines. Based on this categorization and on the type of damping device used it is required to avoid crossings between some determined per rev lines and the first modes of the blade. Therefore an accurate calculation of the blade row natural frequencies and a correct choice of the damping devices are the key points of the aeromechanic design. At this time the blade natural frequency calculations are linear calculations (i.e., the finite element boundary conditions and material models have to be linear) due to the limitation in the methods available. The analysis has to take into account the centrifugal effects in addition to the inertia and stiffness terms. Since the motions of the blades in this particular construction are strongly coupled, both through the cover, a simple calculation of the individual blade properties would lead to false results. Assuming that all the blades have the same vibratory characteristics (and considering tight tolerances on the blade manufacturing that is a reasonable assumption), a finite element model of a single blade and the corresponding wheel sector is constructed. This model is then considered as a basic element in a cyclically repetitive structure that is analysed. Thus the frequency (modal) analysis of the entire row is performed by modelling only one sector of it and applying cyclic symmetry boundary conditions. Using this method, both the single modes (zero nodal diameter and N/2 nodal diameter modes, where N is a number of blades in the assembly) and double modes (modes of higher nodal diameter that appear in pairs) are identified.

In addition to the synchronous per-rev excitation, long low-pressure blades (in particular last stage blades in some of the off-design operating regimes) can experience flow induced vibration and aeroelastic instability. Design solutions dealing with aeroelastic instability have been primarily empirical in nature and specific to a particular application.

The design of the L0 stage blade consists of an axial entry dovetail skewed in the axial-tangential plane. The blade is shrouded at the tip with an integral cover, providing a rigid form of continuous coupling for robust operation. No coupling was included within the steam path to minimize performance losses. A lock-wire is used to restrain the blade from axial motion and springs have been used under the blade dovetails to ensure hook engagement at low speed revolutions. The cover was optimised to provide the lowest possible stresses without compromising the aerodynamic design. The blades have an initial gap at assembly and rely on blade untwist at speed to establish the connection. Figure 3 outlines the mechanical design of the L0 stage.

The L1 (next to the last), stage blade is composed of a 3D airfoil, a two hooks axial entry dovetail, and an integral V-shape cover (see Figure 4). On the top part of the cover two seal teeth are machined realizing a stepped labyrinth seal between the blade and the stationary end wall that minimize the leakage losses. Cover geometry has been optimized in order to minimize tip stresses reducing the overhung mass while keeping an adequate coverage of the throat area. Blades are assembled with an initial gap between the cover contact faces. This gap is closed at speed in consequence of the cover rotation caused by the untwist of the blade. Once the covers come in contact, the blades behave like a single continuously coupled structure which shows a superior stiffness and damping characteristics leading to very low vibratory stresses when compared to an uncoupled design.

L2 and L3 blades (third and fourth to the last), blades are composed of a 3D vane, a tangential entry T shape root and a V shape integral cover (see Figure 5). Tip sealing is realized as a stepped labyrinth with strip seals on the stationary end wall which mates with cylindrical surfaces on the top of the cover. Blades are assembled with initial pretwist and when the rotor is spun at speed contact forces between the covers increase and the entire row of blades behave like a continuously coupled structure.

Figure 3: Solid Model of L0 Stage Blade

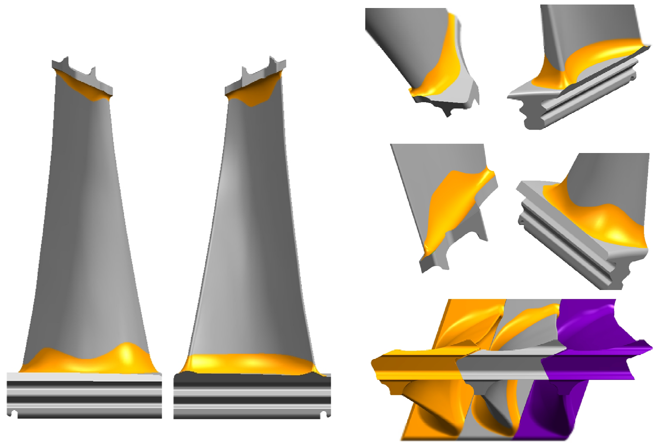

Figure 4: Solid Model of L1 Stage Blade

Figure 5: Solid Model of L3 Stage Blade

Stationary part design

A double shell construction currently used with existing LP sections was adopted for the new stages. The standard reaction technology solution in which separate blades and spacers are installed directly in the inner casing (for this reason called blade carrier) has been changed introducing integral diaphragms that are installed in the inner casing.

Diaphragms are composed of two half bladed-rings bolted together (see Figure 6), which are inserted in the blade carrier. Diaphragms are supported at the horizontal joint with bolted bars. Tangential positioning is controlled through use of a pin fixed to the lower half of the blade carrier. L1 and L0 diaphragms features back spring packing ring which is inserted in a tangential groove machined into the inner ring as shown in Figure 8. The packing ring is mounted on the diaphragms, thereby requiring a cylindrical surface on the mating rotor surface. In the L2 and L3 diaphragms, a stepped semi-labyrinth is realized with strip seals on the rotor that require inner cylindrical surfaces of the diaphragms.

The mechanical design was verified with 3D Finite Element models. All the parts up to the connections to the blade carrier were modelled (see Figure 7). Steam load was obtained from the pressure distribution and applied on the FE model as a pressure distribution. It was verified that local and average stresses are well below the allowable values.

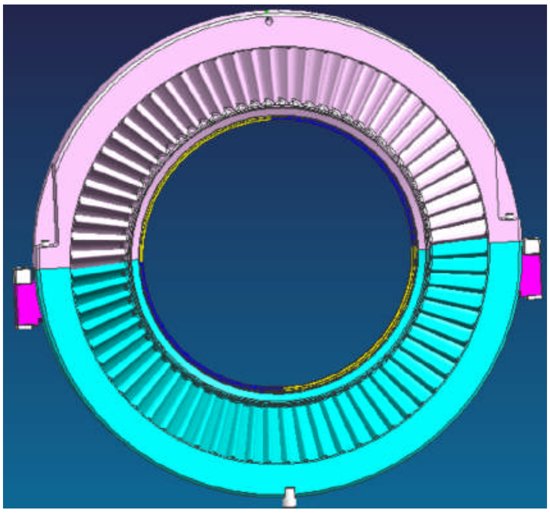

Figure 6: L1 Stage Bladed Ring

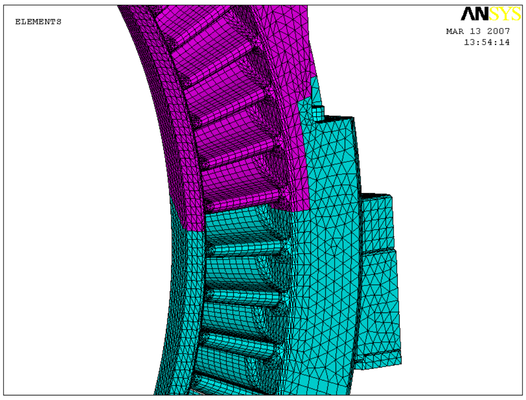

Figure 7: Finite Elements model of L3 Stage Bladed Ring

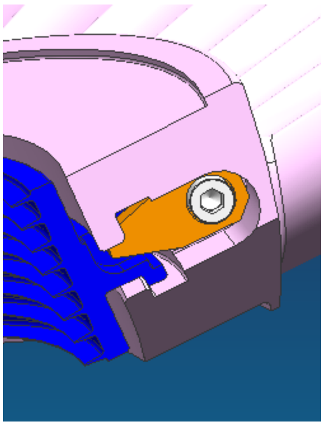

Figure 8: Detail of the Packing Ring of L0 Bladed Ring

Layout design

The four stages layout is shown in Figure 9. This part of the design is focused on defining axial and radial clearances between rotating and stationary parts and on the verification of inner casing and shaft stress distribution. On the basis of the local stress casing and shaft life is evaluated. For the shaft also the tolerance to existing defect and overspeed capability is assessed. For the stationary components (inner and outer casing) the capability to contain rotating blades liberation is also verified.

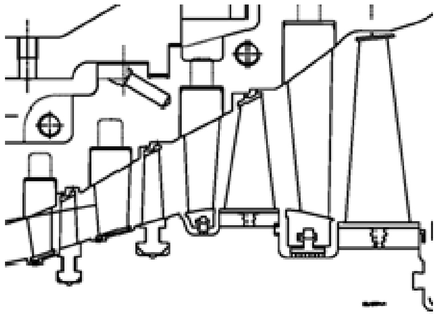

Figure 9: HS LP Stages Layout

MASTER SIZE MANUFACTURING

Rotating blades were constructed using 5-axis machining technology. Test hardware was constructed starting from forged bars whereas for production units bars or shaped forgings (with stock material) are used for blades of different sizes to optimize cost and lead time (larger blades are machined from shaped forgings).

Main challenge of blades manufacturing is the realization of the cover contact face design tolerances (in terms of position with respect to the dovetail). This aspect is particularly critical for L0 blade where the highly twisted shape of the blade makes the tip machining very critical (see Figure 10).

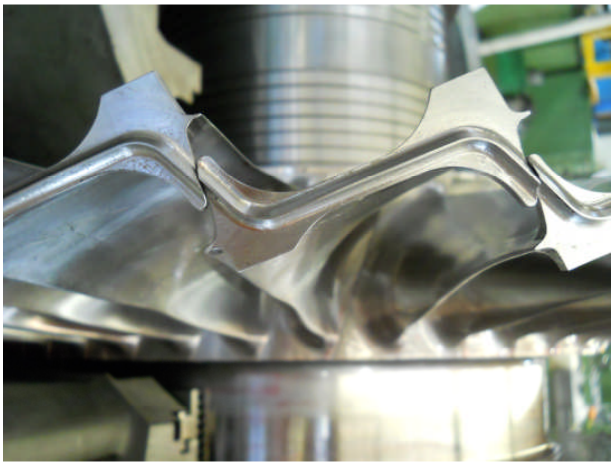

Figure 10: L0 Stage rotating blade tip

Master size diaphragms are constructed with combination of milling from a solid disc and Electro-Discharge Machining. This technology is today possible in light of the significant improvements in 5-axis milling machine capability. In the last ten years the company has accumulated a vast and successful experience in fully milled closed impellers for centrifugal compressors. This full milling technology has now been transferred for use in steam turbine stationary part production. Traditionally steam turbine stators were constructed as composed structure of blades and spacers. In reaction machines they were directly mounted on the blade carrier whereas in impulse technology they were usually welded on diaphragms. The fidelity of the realized nozzle throat distribution to the aero-design intent was dependent on the capability to control single parts tolerances and adjust pieces during the assembly. With full milling, precision and repeatability of the throat distribution along the nozzle row is directly connected to the milling machine accuracy, therefore a very good uniformity is achieved (throats measurement operation is illustrated in Figure 11).

Figure 11: Bling throats measurement operation

Master size shaft manufacturing was qualified performing machining test on a full size dummy shaft (see Figures 12 and 13). Focus of this test was in the verification of the quality of the axial entry dovetails considering the small dimension of the fir tree necks (5 mm) and the tight tolerances.

Figure 12: Shaft axial entry dovetail machining qualification

Figure 13: Machined shaft (WBT rotor)

MASTER SIZE VALIDATION

As a first step, a rotating vibration test of a full scale rotor in a vacuum cell (also known as Wheel Box Test) was performed. This type of test is used to determine the frequencies of the blades over the operating speed range and hence, to build the Campbell diagram. Cover locking speeds of the L0 and L1 blades can also be determined from this test.
The test setup (see Figure 14), consisted of a rotor with the four blade rows assembled on it. The blades were instrumented with strain gauges and the entire assembly placed in a vacuum cell (Figure 15) usually utilized for production rotor over-speed testing.

This facility is equipped with an electric motor and a fluid drive system so that the turbine rotor can be spun to any desired speed. Oil jets (see Figure 16) are used to generate an artificial excitation. Strain gages signal is transmitted to a data acquisition system through a telemetry system (Figure 17)
Experimental Campbell diagrams were generated for all the blades confirming the analytically predicted frequencies and cover locking speed. Blade row nodal diameters were also identified as described in Mitaritonna et al.

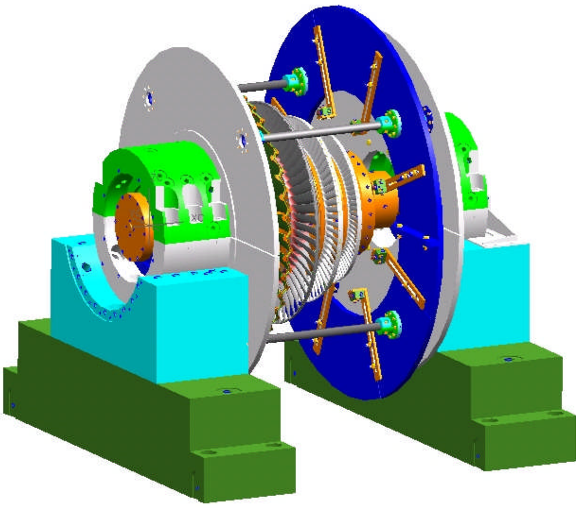

Figure 14: Wheel Box Test Assembly

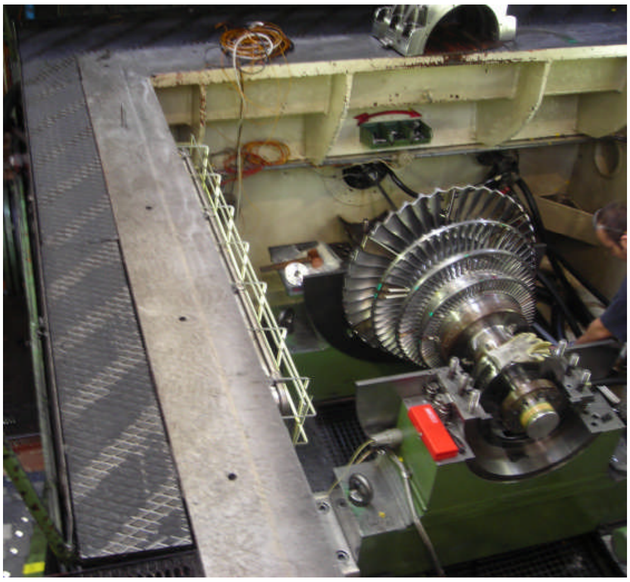

Figure 15: Wheel Box rotor in the vacuum cell

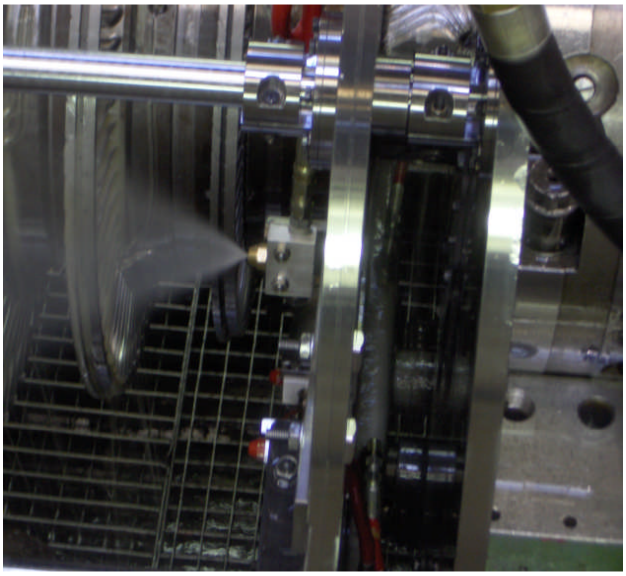

Figure 16: Oil jet excitation

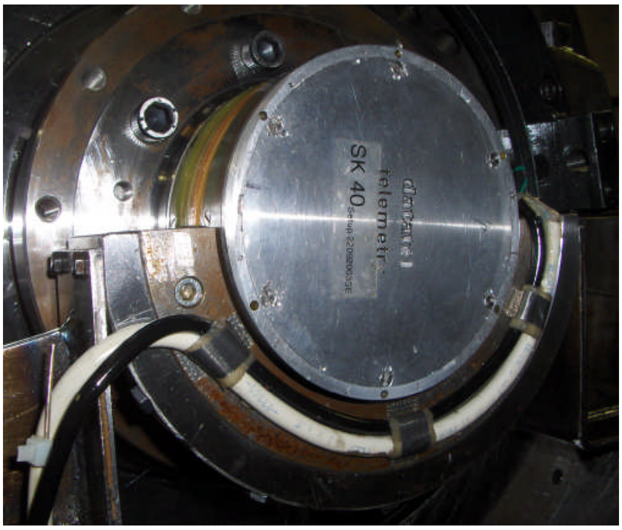

Figure 17: Telemetry System

As a second validation step, verification of the predicted aeromechanical behavior and aerodynamic efficiency was conducted by mean of a full scale full load instrumented test performed in an experimental test turbine (Figure 18). The test turbine facility is used to carry out aeromechanical and aero performance tests of low pressure steam turbine stages. It consists of a steam source (with steam conditions appropriate for low pressure turbine testing), the stages of a low pressure steam turbine and two water-brakes that absorb the energy that the LP turbine produces. Stages are instrumented with strain gages and thermocouples on the blades and with static temperature, total temperature, static pressure and total temperature probes in the flow path and in the diffuser.

Figure 18: Experimental Test Turbine

The aeromechanics test space is a three-dimensional space with the three critical variables being mass flow, exhaust pressure and turbine speed. Test points are established by fixing the mass flow first. With the mass flow constant, the exhaust pressure assumes distinct values between the minimum and maximum exhaust pressure (in this case between 0.05 and 0.7 bara). At each of those combinations of mass flow and exhaust pressure, the speed of the turbine is varied between 50% of Maximum Continuous Speed 100% of Maximum Continuous Speed while acquiring the strain gages data.
Main result of the aeromechanics validation consists in the mapping of blades vibratory response over the entire operating range. Vibratory response is post-processed on a statistical basis to take into account blade to blade variability and build to build variability. Eventually the experimental Goodman diagrams of the blades are built for any operating condition. A broader description of the test set-up and results is presented in "Aeromechanical Validation of a New Steam Turbine LP Section: Test Major Outcomes" by Piraccini et al.
Turbine performance is as a function of several operating parameters, namely: mass flow (Reynolds number), rotating speed, condensing pressure and wetness content. For this reason the performance test is conducted in a set of operating points which are selected in order to map the dependence of the efficiency upon the above parameters.
For each point the operating conditions are set and maintained until all the temperatures stabilize and eventually all the pressure probes are purged to remove water lags and the data are recorded.
Data are then post-processed to measure all the performance characteristics of the turbine like: section efficiency, individual stages efficiency, diffuser recovery factor and moisture loss contribution.

APPLICATION ON IMPULSE PRODUCT LINE

This LP section was initially developed for reaction technology turbines which are characterized by drum construction type (see Figure 20). In a later stage of the project it was decided to apply the same LP section also to impulse type steam turbine which is characterized by disc and diaphragm architecture (Figure 19). This application led to a mixed configuration disc and diaphragm in the front section and drum in LP section that was called "Hybrid Rotor". Rotor design was deeply analyzed focusing on 4 main aspects:

- Thermal transient behavior considering the different thermal inertia of the discs and the drum rotor sections
- Rotordynamics behavior considering the different stiffness of discs and drum rotor sections
- Axial thrust impact on the turbine architecture considering discs section is design to nullify axial thrust whereas the drum section as any reaction turbine generates axial thrust
- Aerodynamic design of the transition between the last IP impulse stage and the first LP stage.

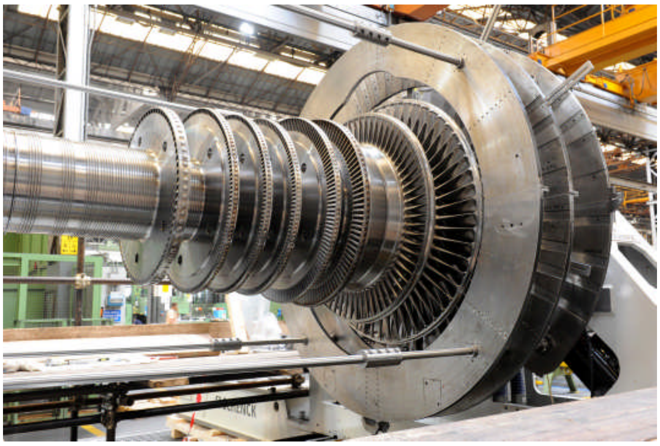

Figure 19: Impulse Turbine equipped with HS LP stages being assembled for tip timing blades frequency test

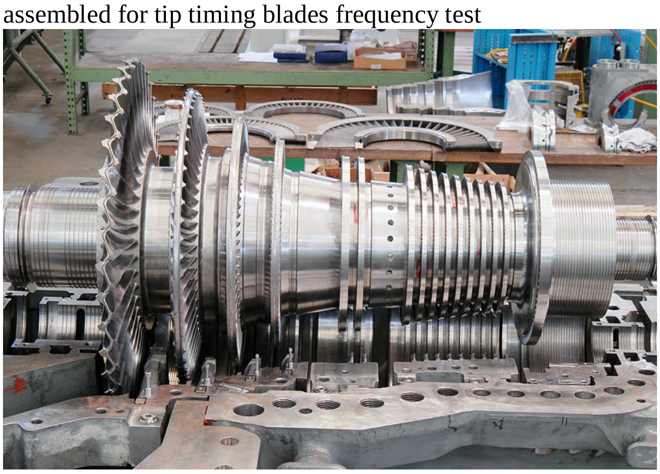

Figure 20: Reaction turbine with HS LP Stages

Thermal transient behavior was assessed analyzing the entire rotor by mean of a Finite Element axial symmetric model. Complete warm-up, steady state and shut-down mission was analyzed.

Rotordynamic was analyzed performing a back to back gravity sag and lateral analysis on a reference unit. The effect of the drum section is actually to increase the stiffness of the rotor, this is clearly visible from the gravity sag analysis a s shown in Figure 21. This effect is visible also in term of critical speed increase and amplification factor decrease as shown from the back to back analysis performed on the same unit.

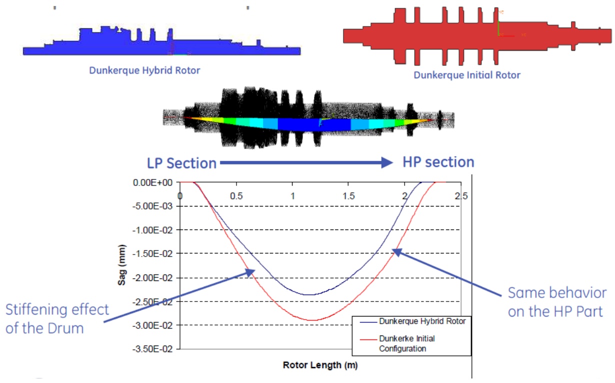

Figure 21: Hybrid Rotor gravity sag analysis

	First Translation Mode		First Bending Mode	
	Dunkerque + HS	Dunkerque Initial Job	Dunkerque + HS	Dunkerque Initial Job
Critical Speed Frequency (rpm)	4400	3990	17400	14910
Amplification Factor	2.84	3.2	< 2.5	1.55

Figure 22: Hybrid Rotor rotordynamic analysis

Axial thrust increase has been evaluated assessing thrust balance over the entire surface of the rotor including the blades contribution. On the same reference turbine used for the rotordynamic assessment the thrust increase is of the order of 2 tons well within the existing thrust bearing capacity (8 tons continuous operation capability).

Transition between the last impulse HP/IP stage and the front LP stage has been assessed using 3D CFD (see analyzed volumes model in Figure 23). Main challenge is due to the different flow coefficient of the traditional impulse HP/IP stage and modern LP 3D reaction stage which obliges to design a diffusing transition zone. Several configurations have been tested defining an optimal diffusion length and angle.

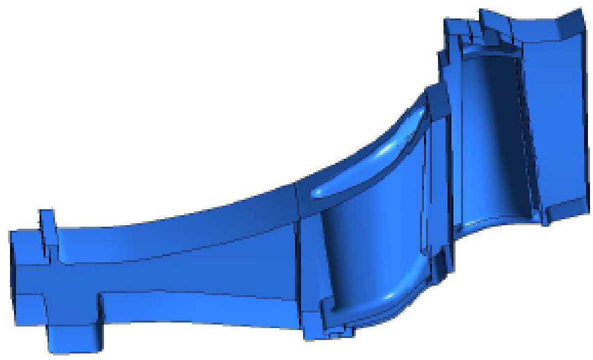

Figure 23: HP/IP Section to LP section transition geometry

GENERATION OF THE LP SECTIONS FAMILY

Rotating speed and absorbed power of the turbo compression trains in which steam turbines are utilized depends on the application and on the size/optimization of the plant. They are typically in an application range where design speed is between 4000 and 15000 rpm, and power is from 1 to 90 MW. Plant sizes have always tended to increase as a consequence of the scale cost advantages, and especially in the ethylene applications the threshold of 50 MW has largely been exceeded. In order to cover this wide application range the steam turbine product line requires the design of an extensive set of components of different size and rotating speed capability. To generate it in an economic and robust way, different philosophies have been developed in steam turbine technology during the years. For what concerns the LP stages this problem has been historically addressed in two ways:

1) Perfect Scaling method: generate families of different size designs scaled from a unique master design (This is a traditional approach (see Brandt), that is also extensively used Gas Turbine design to scale different section of existing GT's. Perfect scaling means that the parts are scaled of a constant scale factor "f" in the three dimensions and the rotating speed of the LP section is scaled of a factor 1/f. Same method was also applied to scale the LP sections families traditionally utilized by this author company.

2) Traditional impulse design method: generate different designs applying the same last stage blade at different standard base diameters. For example in traditional impulse technology, a 12.5" last stage bucket could be applied at 31", 34" and 38" base diameter increasing the blade count to compensate the different root diameter and keep almost constant the pitch line chord over pitch ratio.

Advantage of the perfect scaling method resides in the consistencies related to the similitude of aero and mechanical design of all the components of the family. From the mechanical design point the three most important are:

- Campbell diagram apart from small frequency shift related to non-perfect scalability of damping effects are the same.
- Static stresses are the same and Goodman diagram of the bucket are affected just by a small variation of buckets vibratory effect that can be introduced by the non-scalability of damping and Reynolds number.
- There is a significant experience and history of aeromechanic validation based on scaled model. Therefore the test of one single component can cover the validation of all the family

From the aero standpoint the most important relations are the followings:

- Stage velocity triangles are the same
- Section and stage efficiency are the same apart from Reynolds effect
- Flow function (or Stodola coefficient of the LP section) is the same

Another significant advantage is the fact that a very large family of components can be generated from a single design.
The only disadvantage of this method when compared to the traditional impulse one is related to the production process. All the components of the family have different parts and this does not allow any production simplification.
Advantages of the traditional impulse method are related to the simplification of the manufacturing and sourcing process. There is one forging size, one drawing (with multiple set of tabulated dimensions), one set of dovetails cutters, and one set of interchangeable jigs and fixtures. However, since the designer needs to set a whole part count, an exact scale part is never obtained and hence it is necessary to generate a multiple set of tabulated dimensions and slightly different machining codes.
The disadvantages of this method are related the fact that the aero and mechanical operating conditions of the single components of the family are different. This has two important effects:

- There is a limit to the number of components that can be generated with the same bucket. This is because moving away from the optimum design, bucket frequencies, and static stress change and cannot easily be compensated by speed variation and aero performance deteriorates.
- Aeromechanic validation of all the family cannot be obtained with the test of just one component. The change in nozzle and bucket counts from one component of the family to another and the related change in wakes interaction is a serious concern for aeromechanics. Therefore the validation of all the components of the family would be strongly recommended.

For the HS8 design, the perfect scaling solution was chosen and in terms of granularity a scaling ratio of 12% was chosen to generate the family of 9 LP sections represented in Figure 24 where the number represent the approximate height of the last stage blade expressed in inches.

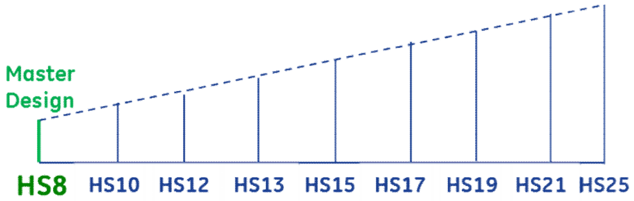

Figure 24: HS family

TIP TIMING FREQUENCY VALIDATION

Steam turbine industry has a long and positive experience in perfect scaling of LP sections.
Large power generation steam turbines manufacturers uses to scale a single design from 50hz to 60hz to cover both frequency grid markets. In many instances given the large size of the LP section (large Power generation blades have nowadays passed the 48" length) and the cost associated with a full load, full size laboratory tes a scaled version of the LP sections (scale factors of the order of 2-3) is tested at full load to prove aeromechanics behavior and performance.
As discussed in previous paragraph also mechanical drive steam turbines manufacturers have applied extensively perfect scaling to generate the family of LP sections.
Despite of the industry experience and tradition in blades scaling it was decided to measure the scalability of the family performing frequency and damping experimental testing of some of the scaled-up rotating blades.. This validation was intended as a risk mitigation of the overall scaling process (design + manufacturing).
Traditional method to experimentally determine the frequency and damping of continuously coupled blades at full speed is based upon strain gages technology. This method requires installing strain gages on blades and, route the strain gages wires to a telemetry mounted on the rotor. This technology (utilized to test the master design) although has proven high reliability requires building a dedicated test rotor and the overall instrumentation process (strain gage applications, wire routing and telemetry set-up) is typically time-consuming.
Another method named "tip timing" is being applied in the turbomachinery industry since several years. This method relies upon externally mounted sensors to determine the passing times of the blades. The passing times after conversion to deflections, can be used to measure each blade's vibratory response characteristics such as frequency and damping. Differently to what happen with strain gages method where just the instrumented blades are monitored with "tip timing" every blade is measured therefore also blade mistuning and nodal diameter can easily characterized.
Tip timing technique is very attractive as it allows avoiding rotor instrumentation with significant reduction of the test cost and lead time and with potential application in the field.
Major challenge with this technology is associated to the definition of the reading location of the probe in terms of axial

and radial position.
Radial distance from the tip of the blade influences the quality of the measurement, as amplitude of the signal is a function of the distance of the probe from the object subjected to measure. In theory would be optimal to position the probe few tenth of millimeter from the tip of the blade but this requirement clearly conflicts with the need of avoiding contact between the blade and the probe itself. Design engineer must concur with test engineer to define trade-off between these two requirements.
Axial position is also important as it drives the sensitivity of the probes to blade modes. In this case the goal is to position the probe at the axial location at which for a given vibratory mode blade is expected to show the highest displacement. Hence designer must trade off among the different modes and make sure the untwist of the blade and all the contributions to the axial displacement of the rotor are well captured in the analysis.
Accuracy in the determination of the axial position of the probe relative to the blade tip in running condition will have a strong influence in the accuracy of the response measurement.

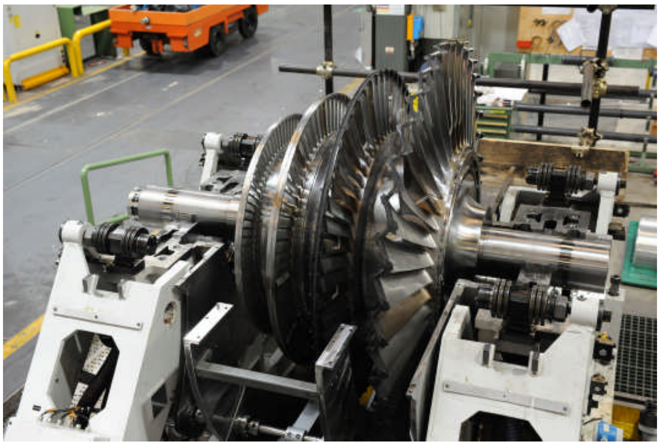

Figure 25: HS15 tip timing and strain gages WBT rotor

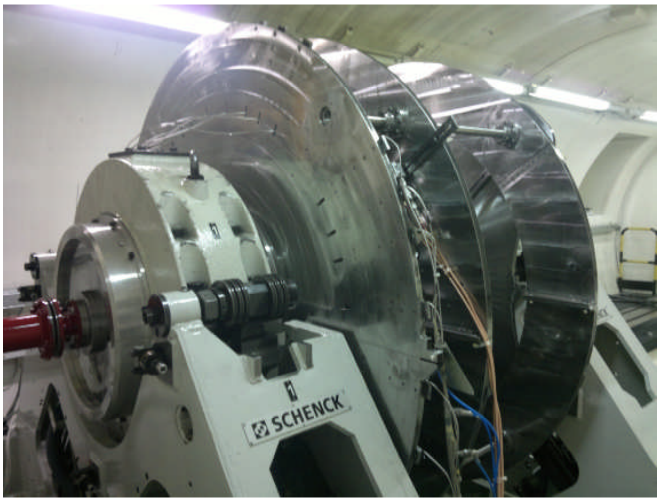

Figure 26: HS15 tip timing and strain gages WBT rotor

In order to be able to perform the frequency validation direct on fleet leaders production rotors it was decided to establish tip timing technology building another test rotor. A back to back comparison Wheel Box Test on HS15 (see Figure 25 and 26) was designed as a mean to select the best tip timing sensors, tune the excitation system and calibrate the tip timing frequency measurement against strain gages.
Three types of tip timing sensors were tested: eddy-current, laser and microwave. Two type of excitations were tested: oil jets and permanent magnets.
Blades were also heavily instrumented with strain gages (55) and thermocouples (4) an illustration of L0 blades instrumentation is shown in Figure 27.
A back to back comparison between the strain gages signal (reference) and the different tip timing probes was performed. Eventually eddy-current probes (with oil jet excitation) were selected as optimal solution demonstrating very good accuracy and fidelity to the strain gages data. The signal back to back comparison illustrated in Figure 28 shows how tip timing sensors are able to detect accurately the response peak.
Also statistical data (over 20 strain gages) taking as reference L0 blade 1st mode frequency show a very good consistency between strain gages and tip timing measurements (Figure 29).

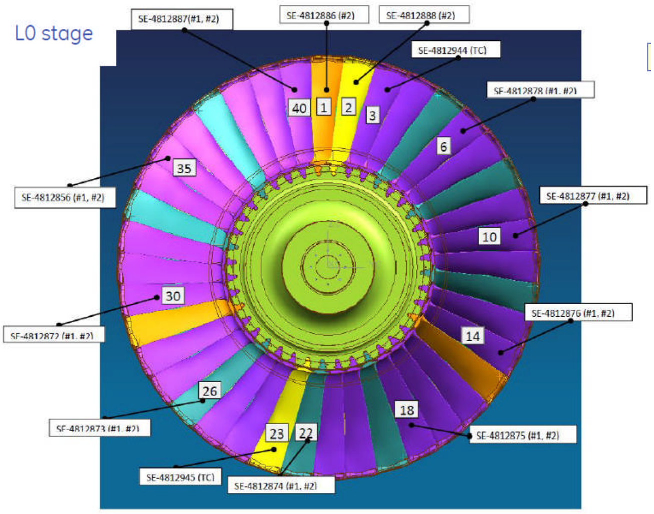

Figure 27: HS15 WBT L0 stage instrumentation

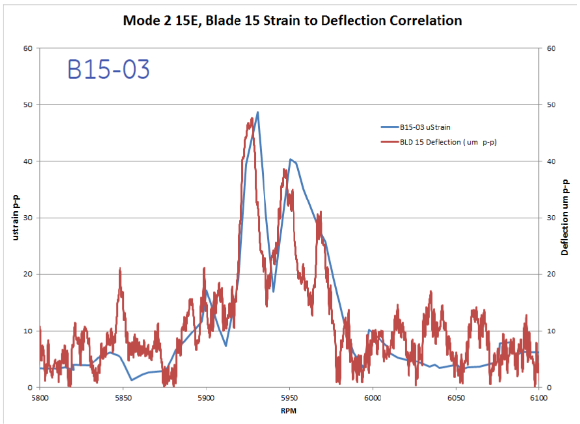

Figure 28: Tip timing vs strain gages signal

frequency [Hz]

SG	Tip Timing
AVG	AVG
418.4	417.8
STD	STD
0.59	0.45

Figure 29: Tip timing vs strain gages measured frequency

Once tip timing technology was established on HS15 it has been applied on fleet leader production rotors of HS10, HS13, and HS17. A picture of the tip timing test set-up performed on an HS10 ammonia syngas driver reaction unit is shown in Figure 30.

Frequency scalability has been validated within 1% proving the success of the design and manufacturing scale-up process and confirming once again the soundness of the perfect scaling method.

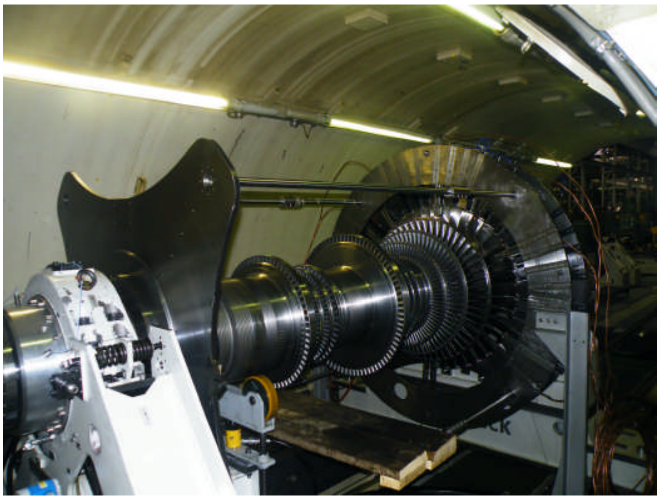

Figure 30: Tip timing test on an HS10 of an ammonia syngas turbine

BLADES EROSION RISK MANAGEMENT

In the last stages of a condensing steam turbine, steam is expanded below the saturation line. Steam quality and therefore the amount of water that is impacting rotating blades is strongly varying as a function of condensing pressure, inlet pressure and inlet temperature.

The part of steam that turns into liquid water forms a fog of small droplets whose typical radius is in the order of 0.01÷1 µm. The size and the relative velocity of these droplets is such that they do not represent a risk for rotating blades. A fraction of these droplets deposits on the stationary blades where it combines with the spray of coarse water coming from the previous rotating blades. The water film attached to the stationary blades moves towards the trailing edges where it becomes unstable due to aerodynamic forces and finally is dragged by surrounding stream in the main flow as a spray of large droplets.

The radius of the droplets that detach from the trailing edge is in the order of 1000 µm. The large droplets enter a region of higher steam velocity where they are broken into stable smaller droplets, known as coarse water droplets, of the order of 100 µm. The size of the stable droplets is a function of the Weber number therefore it is strongly influenced by the steam density (hence the pressure) at the stator trailing edge. The lower the pressure the larger will be the size of the stable droplet. Larger droplets will cause higher erosion damages.

$$\mathrm{We} = \frac{\rho v^2 l}{\sigma}$$

Where:

ρ = steam pressure
v = steam velocity
σ = droplet surface tension

The re-entrained coarse water accelerates gradually with the steam and eventually hits the downstream rotating blades. As a function of the distance between the stationary blades trailing edge and rotating blades leading edge the droplets absolute velocity remains a fraction of steam velocity, typically 10%. Consequently droplets strike suction side of trailing edge with a relative velocity (see Wf in Figure 31) that is close to rotating blade tangential velocity (U).

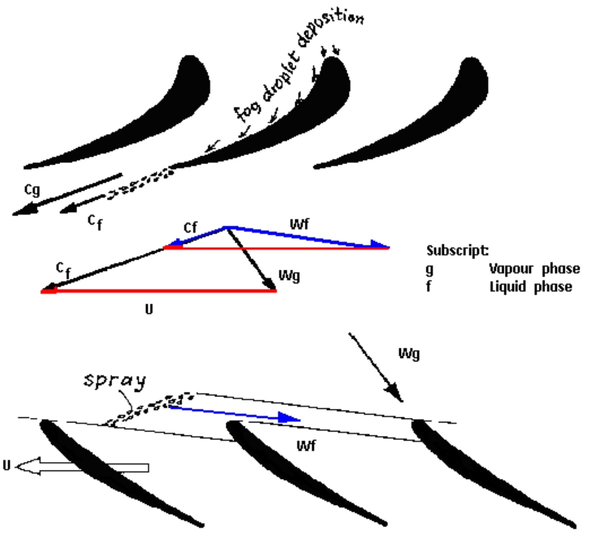

Figure 31: Water droplets velocity triangle

At impact, the formation of a shock front in the liquid of the drop is accompanied by corresponding stress waves propagating into the blade material. At the same time the liquid jets out laterally.
The erosion damage will be then also a strong function (cubic) of the impact velocity.

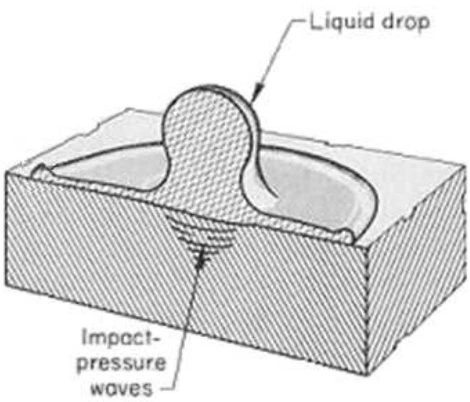

Figure 32: Droplet Impact Illustation

The solid surface is subjected to a multitude of sharp pressure pulses and jetting outflows, each of very short time duration and acting on a very small area.
Blade material damage depends on whether the material is ductile or brittle and on its microstructure. In general randomly disposed dimples gradually develop and the surface undergoes gradual deformation.

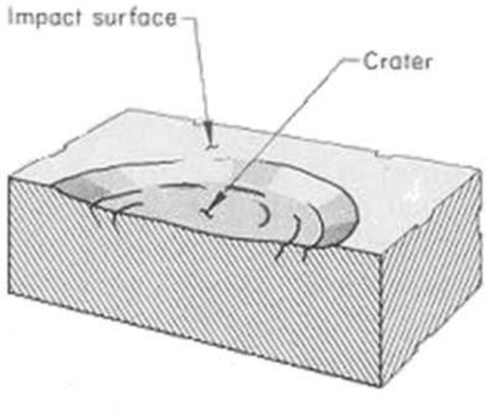

Figure 33: Erosion Damage

Erosion is a phenomenon that develops with time. Accurate long-term predictions are impossible due to the non-linear progress of erosion with time under constant impingement conditions but, from a qualitative standpoint, a standard trend can be identified.
A general erosion time-pattern is divided in the stages described in Figure 34.

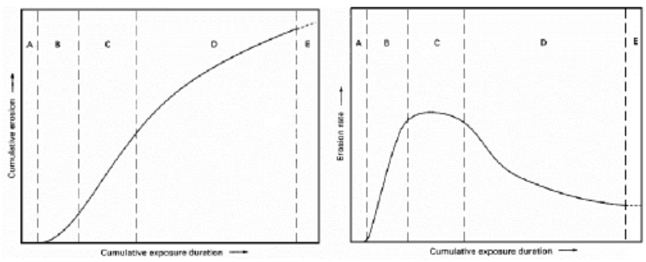

Figure 34: Erosion trend over time

(A) Incubation stage: little or no material loss; metallurgical changes take place in the surface; incubation period may not appear if impact conditions are severe enough to cause single impact material loss.
(B) Acceleration stage: erosion rate accelerates rapidly to a maximum.
(C) Maximum: erosion rate remains nearly constant.
(D) Deceleration stage: the erosion rate declines to some fraction (25÷50%) of the maximum rate.
(E) Terminal Stage: the erosion rate remains constant indefinitely.
Practical implication of the erosion development nature is that turbine blades will show some erosion after a limited number of hours of operation at erosion risk conditions and then erosion damage will progress slowly. To give an example: the amount of erosion which is detected after the 1 year of operation will most likely double not in two years of operation but in 8-12 years.

Erosion risk is higher in power generation application because as illustrated in Figure 1 the PG steam cycles are characterized by lower condensing pressure which is associated by larger droplets and higher moisture content. On the contrary the traditional mechanical drive applications (e.g. fertilizers, ethylene and refinery) are characterized by higher condensing pressure and hence very low erosion risk.
Considering that the HS Low Pressure blades feature a higher rotating (and peripheral speed) compared to similar size traditional LP stages the erosion risk has been addressed since the initial phase of the development.
A high erosion resistance last stage blade material (M152) has been selected. There is a long experience with this material in large power generation units (33” last stage blades).
Also a stellite based coating has been developed to further enhance the erosion resistance of the blade.
In addition moisture removal systems have been designed for the larger sizes of the family. The most effective moisture removal system is the so-called hollow stationary blade (illustrated in figure 35). In this system utilized the internal volume of the last stage stationary blade is connected to the condensing pressure. Radial slots are then machined on the surface of the blades and the pressure ratio across the surface and the internal part of the blade is utilized to create a suction effect that extract the water film that is flowing over the surface of the blade reducing significantly the erosion.

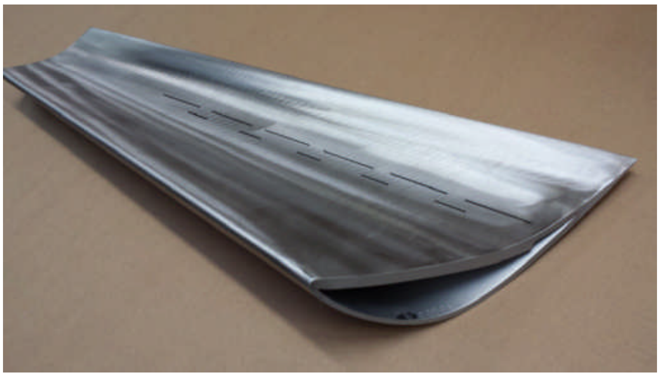

Figure 35: Hollow stationary blade

.

FIELD APPLICATION EXPERIENCE

First unit with HS Low Pressure Stages has been commissioned in 2009. This is a power generation turbine located in a Waste-to-Energy plant in Italy (see Figure 20). The turbine, with a nominal power of 10 MW, is a typical example of a geared, small power generation application. Unit passed positively the performance test in July 2009 and has been continuously running since then.
At the moment of this writing 22 units with HS LP stages has been applied for geared Power Generation units (combined cycles, O&G plants and renewable) and Mechanical Drive (Ethylene, Fertilizers and Refineries) applications that required high rotational speed.

Figure 36: Ethylene charge gas driver turbine

A 3 stages version of the family has been also developed and applied for PG applications to allow injecting or extracting steam at lower pressure. Also a solution with an interstage extraction between next to the last (L1) and next to the next to the last stage (L2) has been developed to support even lower extraction pressure need in PG regenerative cycles.
All the mechanical drive applications have been designed with radial exhaust configuration whereas in Power Generation applications both axial and radial exhaust configurations have been applied.
All MD and PG rotors performed successfully the factory 120% overspeed test in the vacuum cell.
All MD units performed and passed the API mechanical running test.
All power generation units and one of the mechanical drive units have performed a contractual performance test with positive results.
A field performance test of an industrial steam turbine is based on torque or generated power measurement. Therefore it assesses the overall turbine performance including control stage, HP/IP sections LP section, secondary flows, bearing losses and gear losses (when present). For this reason the LP section efficiency is not directly measured, it is calculated and it is affected by the following uncertainties:

- Mass flow measurements
- Torque or electric power measurement
- Bearing loss estimation
- Gear loss estimation
- Prediction of the performance of other sections of the turbine (control stage, HP/IP section and secondary losses)

To illustrate the typical industrial steam turbine field performance test execution the data relative to the ethylene charge gas driver unit of Figure 36 are reported here below. That unit was equipped with a torquemeter therefore the produced shaft power was directly measured. Performance was assessed measuring the mass flow, pressure and temperature conditions as shown in Figure 37. Once the steam conditions are stabilized instruments data are acquired for 30 minutes and then averaged. Comparison between expected power and measured one shows a discrepancy of 0.3% (measured power is higher than expected). This result is interpreted as the confirmation of the good accuracy of the overall performance prediction model and considering that LP stages in this unit are producing 25% of the overall power an implicit confirmation of the accuracy of LP stages model.

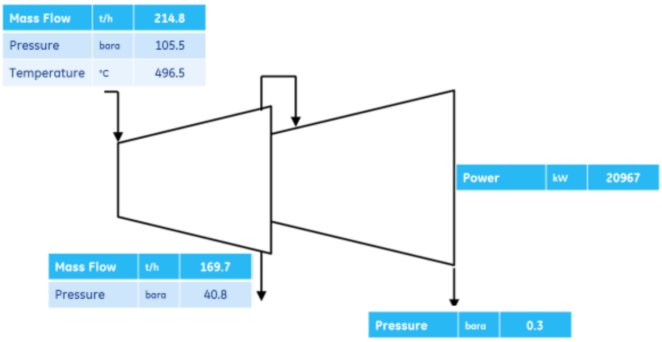

Figure 37: Measured performance data

CONCLUSIONS

The design, manufacturing, validation and field application of a new family of steam turbine Low Pressure stages have been described focusing on the key conceptual choices and the main technology aspects. An overview of the parts manufacturing critical aspects and the experimental validation technology that are available to test the LP section components behavior has also been provided.
LP stages technology is still a key research area for the Steam Turbine industry due to the complexity of the physics involved with wet steam modeling, high Mach airfoil design and long blades aeromechanics.
Manufacturers continue to invest in the development of LP stages to improve its efficiency and rotating speed capability as the design characteristics of this section of the turbine have a great influence on the performance of the entire train.

NOMENCLATURE

ST = Steam Turbine
LP = Low Pressure
HP = High Pressure
IP = Intermediate Pressure
MD = Mechanical Drive
PG = Power Generation
A = Blades row annulus area
N = turbine rotating speed
VAN = Last stage exit annulus axial velocity
WBT = Wheel Box Test
SCC = Stress Corrosion Cracking
We = Weber number
ρ = steam pressure
v = steam velocity
σ = droplet surface tension
Wf = droplet relative velocity
U = blade peripheral velocity

REFERENCES

Mujezinovic, A., Hofer, D., Barb, K., Kaneko, J, Tanuma, T. and Okuno, K, Introduction of 40/48 Inch Steel Steam Turbine Low Pressure Section Stages, Proceeding of the Power-GEN Asia, (2002), CD-ROM.

Denton, J. D., Spurr, A., Aerodynamic factors in the design of the final stage of large high speed steam turbines, Paper C193/79, I Mech E Conference Publication 1979-12,.,

Piraccini F., Cosi L., Biondi R., “Aeromechanical Validation of a New Steam Turbine LP Section: Test Major Outcomes”, Proceedings of ASME Turbo Expo 2010, Glasgow, UK.

Gyarmathy, “On the design limits of steam turbine last stages”, Proceedings of Technology of turbine plants operatings with wet steam, 1988

Mitaritonna N., Piraccini F., Cosi L., Cioncolini S., Cortese C., “Advanced Experimental Techniques for Turbine Blades Dynamic Characterization”. Proceedings of International Operational Modal Analysis Conference 2007.

Verstraete, T., Priniser, J., Di sante, A. Della Gatta, S. and Cosi, L., “Design optimization of a low pressure steam turbine radial diffuser using an evolutianary algorhytm and 3D CFD”. Proceedings of ASME Turbo Expo 2012

Verstraete, T., Priniser, J., and Cosi, L., “Design and off-design optimization of a low pressure steam turbine radial diffuser using an evolutianary algorhytm and 3D CFD”. Proceedings of ASME Turbo Expo 2014.

Brandt D.E., GER-3434D “GE Gas Turbine design Philosophy”,

ACKNOWLEDGEMENTS

This author wants to thank of all the professionals in Italy, USA, France, India and Poland that contributed to the development of this family of Low Pressure Stages.