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NEW CHALLENGES AND DESIGN FOR HIGH MACH HIGH FLOW COEFFICIENT IMPELLER FOR LARGE SIZE LNG PLANT

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ABSTRACT

The large scale of Liquefied Natural Gas plant, has always meant large size of compression train for refrigeration duty, most of the time driven by high power driver in the range of 40-60 MW. The new generation of LNG plant are moving toward a larger size, that mean larger compression stations, driven by higher power gas turbine that can arrive and exceed 100 MW. This increase of specific power means also increase in compression gas flow and so the need to have compressors that are able to handle it in efficient way. This is true in particular for impeller stages equipping the Propane Compressors that will be selected at higher specific flow coefficient and Mach number. The higher the flow coefficient and the peripheral Mach number the higher are the losses and so it is difficult to continue to keep a good level of stage efficiency and operating range, that are the main key performance parameter for the refrigerant compressor for LNG. The present paper illustrates an improved impeller stage designed in particular to fit this duty, the need of a multidisciplinary optimization, from aerodynamic, structural mechanic, aeromechanic and rotordynamic. The paper illustrates the main design challenges for this type of impeller design, the validation done by the OEM and the benefits of their usage by mean of dedicated Case studied.

INTRODUCTION

<u>Market overview</u>
LNG global trade has grown from 155 mtpa (million tonnes per annum) in 2006 to 258 mtpa in 2016 and is expected to grow to 350 mtpa by 2020, resulting in a 15-year average annual growth rate of 5.5% (from Houston LNG summit 2017). The surge in LNG export facilities is foreseen between 2020 and 2025 and so the upcoming need of new LNG plants, being a typical LNG export facility requiring at least 5 years to develop, construct, and commission.

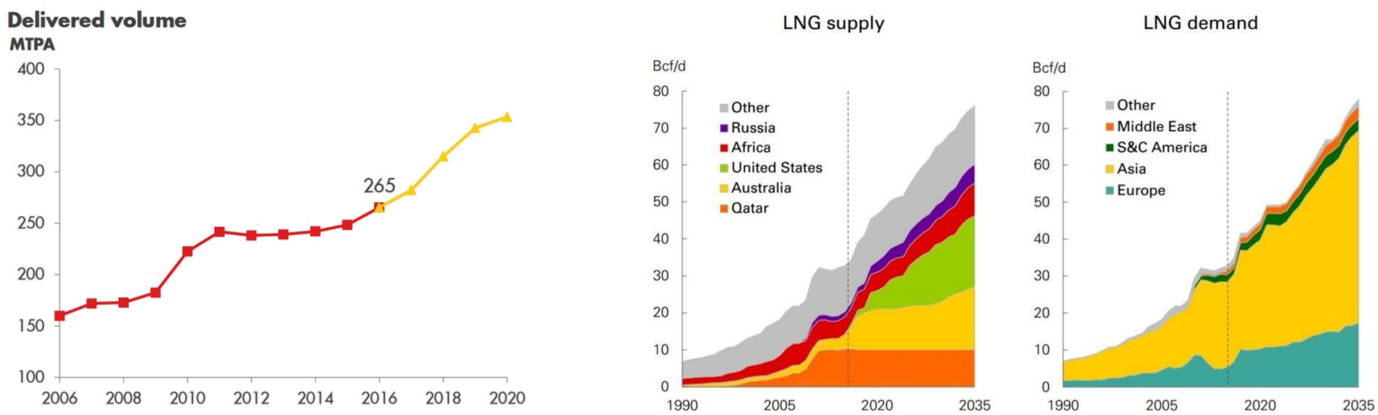


Figure 1 – a) LNG delivery volume forecast (Source: Shell LNG Outlook 2017);
b) LNG supply and demand forecast (Source: BP Statistical Review of World Energy 2017)

The emerging markets show considerable challenges to convert from existing sources, such as fuel oil or coal. LNG is increasing in importance, as countries look to diversify their energy mix while lowering carbon emissions.

Even though global prices for LNG have dropped considerably in recent years, LNG gas remains more expensive and volatile than coal for power generation. With a lower LNG market price, it is imperative that project developments be executed at a minimal cost. The average projects cost escalates from $1,000/tpa of decade ago to more than $3,000/tpa in the most recent projects *(from JPT Agust 2017)*. The most competitive project developments are looking to target a cost of approximately $500-700/tpa using design standardization, and larger individual train capacity.
The impact foreseen for the Turbomachinery trains of next LNG plant is then to move toward:

- larger individual train capacity;
- referenced and standardized configurations;
- larger gas turbine power, with even more attention of today to the efficiency and emissions;
- larger compressor size and specific flow, with growing focus on efficiency and productivity.

<u>LNG Liquefaction process and turbomachinery</u>
The turbomachinery design of LNG plants, in particular the main refrigeration units, is strongly connected also to the selected liquefaction process.
From *IGU 2017 World LNG Report*, Air Products' liquefaction processes accounted for nearly 80% of existing plants in 2016: the AP-

C3MR™ process held the greatest share at 47%, followed by the AP-X® (15%) and AP-C3MR/SplitMR® (14%) processes. Air Products processes account for 68.2 MTPA (59%) of the 114.6 MTPA of capacity under construction as of January 2017. Air Products is therefore expected to retain its leading position. However, the ConocoPhillips Optimized Cascade® process will see strong growth with eight trains (35.9 MTPA of capacity) under construction as of January 2017. Sixty percent of the 35.3 MTPA of new capacity that came online since January 2016 utilizes the Optimized Cascade® process.

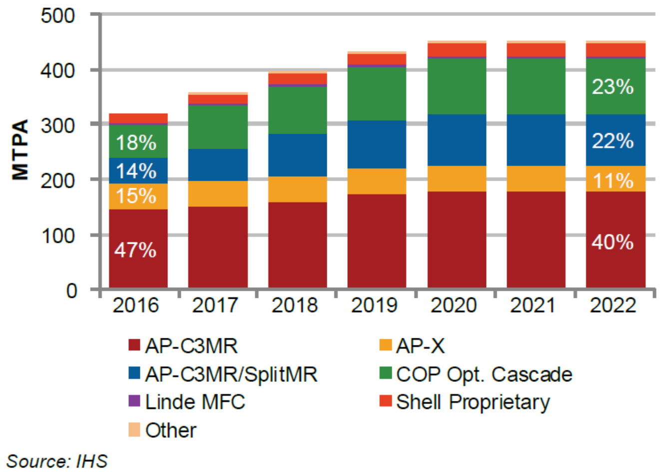


Figure 2 – LNG process market shares.

For the large majority of the cases (>90%), the refrigeration process implies the use of multistage and side stream compressors processing high Molecular weight gas (mol weight > 28 g/mol). This provides to the compressor OEM a specific challenge, related to the high specific Mach number of the impeller that can makes difficult to target large machines operating range and high efficiency. This is particularly true for Propane compressors, and this task get even harder if associated to the above described trend of train capacity increase.

The design of large flow coefficient, high specific Mach number impeller is a constant area of OEM investment. The design optimization for these impeller families has historically benefit of the increasing usage of CFD tool and advance validation and measurements methods. Since early 2000s, when the higher value of CFD support in designing transonic impeller blades was proven [1], different developments have been brought. Mixed Flow impeller design has been developed and proven to be the best choice for very large flow coefficient [2] and today with more sophisticated Aerodynamic and Mechanical design optimization it is possible to design and release impeller stages with an improved level of efficiency, operating range and mechanical reliability.

This document address mainly the compressor impeller stage optimization for LNG service, but it is worth to reminding that the above-mentioned LNG market trends has dictated also a new direction for gas turbine development. The OEM has moved to design larger gas turbine aero derivative frames, to keep at pace of the increased train power at highest possible efficiency and lower possible emissions.

AERODYNAMIC DESIGN OF STAGES FOR LNG APPLICATIONS

Centrifugal compressors for LNG applications usually require stages able to operate at high flow coefficient and peripheral Mach number values. Moreover, rotordynamic assessment on this type of multistage machines often suggests a high stiff design for impellers, in order to compensate the large axial span due to typical stage stacking and presence of large sidestream injections. All these requirements lead to high value of relative Mach number at impeller eye, with transonic or supersonic conditions within rotor inducer and consequent shock phenomena, even at design flow coefficient.

Iso-curves for relative Mach number at impeller eye are represented in Figure 3 as function of machine design flow coefficient φ and peripheral Mach number *Mu*; different zones are roughly depicted in the plot with dashed lines, based on flow at impeller inlet, with respect to sonic conditions. It can be observed as transonic or supersonic impeller design is required in most of the cases, considering that large size LNG applications often require flow coefficient above 0.12 and peripheral Mach number greater than 1.0 at design point. It is worth to note that actual iso-curves position and shape in φ-*Mu* plane depend on specific design parameters and mainly on (see nomenclature section for symbols meaning):

- Rotor stiffness $\frac{D_2}{D_{1h}}$

- Shroud over hub diameter ratio at impeller eye $\frac{D_{1s}}{D_2}$
- Impeller inlet flow distortion $\nu_1 = \frac{c_{m1s}}{c_{m1}}$
- Blockage at impeller leading edge

The curves shown in Figure 3 are based on reasonable values for the parameters listed above, kept constant through the whole flow coefficient and peripheral Mach number ranges. Proper choices during design phase can reshape the iso-curves in Figure 3 and move them in position. Boundaries among the three zones (subsonic, transonic and supersonic) can move in turn.

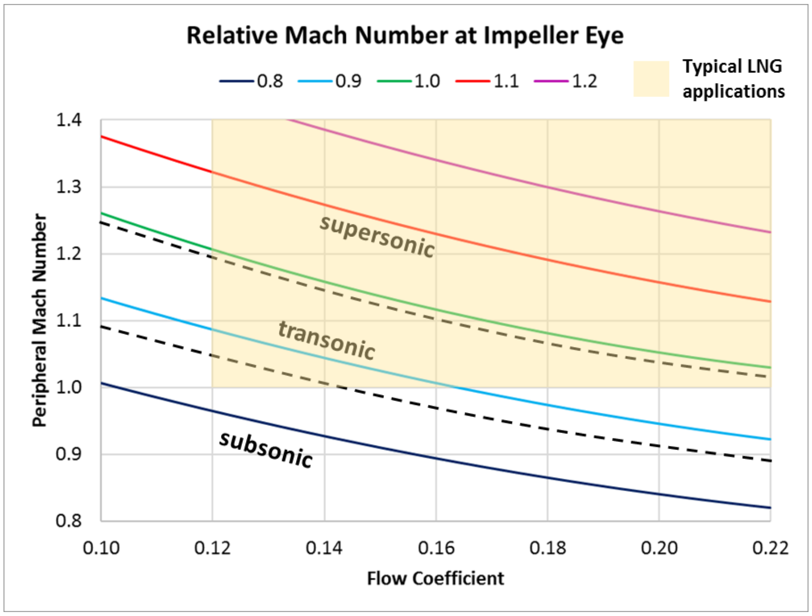


Figure 3 - Iso-curves of relative Mach number at impeller eye

Impeller Design

Referring to Figure 3, stages working in transonic or supersonic regions consist of impellers which show heavily non-uniform flow conditions in span-wise direction near the leading edge. This is mainly because of the transition from subsonic flow in the relative frame at the hub to supersonic one nearby the blade tip and furthermore intensified by the high flow coefficient (greater than 0.12). Moreover, shock phenomena involving the upper spans of inducer part of blade, cause significant non-uniformity also in the aft part of the blade.

For these reasons, strong pressure gradients exist within rotor channel in both stream-wise and span-wise directions: the first is a potential trigger for boundary layer separation, especially downstream the shock (shock-boundary layer interaction), and the second fosters the onset and increase of secondary flows. Both of them affect impeller efficiency.

It deserves to be underlined that off-design conditions are even more critical on this type of stages: stonewall characterizes the right limit of the operating range, where the inlet condition usually becomes supersonic and a normal shock occurs at the impeller throat area; high incidence and increased flow diffusion (adverse pressure gradient) at left limit of the operating range enable the shock to be stronger, move forward on the blade and potentially interact with the boundary layer, causing wide flow separation.

Impeller design is therefore mainly focused on the mitigation of shock effects on rotor efficiency as well as on flow field distortion up to the diffuser inlet. This implies the need for a local control of blade geometry at different span positions, in order to optimize blade loading consistently with flow regime (subsonic or supersonic) and balance its hub-to-shroud distribution. Furthermore, the control of shock strength on the suction side of the blade, along with its stream-wise position, plays a key role in determining impeller behavior at design flow coefficient as well as at right and left limits of operating range.

Lastly, shroud curvature on meridional flow-path needs to be carefully considered during design phase. being a critical parameter which impacts on shock strength and position in the fore part of the blade; it contributes as well to diffusion control on the aft part.

To all these aims, the following features have been employed and optimized during the design of a new impeller for high flow coefficient, high peripheral Mach number stages:

- Fully three-dimensional design, with blade geometry defined by several constant-span profiles
- Different blade bowing shape from inducer to impeller exit
- Control on local surface curvature, especially on suction side of the blade
- Proper leading-edge shape at each constant-span profile
- Proper hub-to-shroud profile of leading edge projection on meridional view
-

The optimized stage design needs also to be compliant with the same geometrical constraints of the baseline stage, in terms of

- Overall axial span (same rotor bearing span)
- Maximum radius at top of U-bend over impeller external diameter (same compressor casing)
-

In addition, the stage has been designed with a higher impeller stiffness (higher $\frac{D_2}{D_{1h}}$ ratio), about 10% above the baseline. This choice gives benefit in rotordynamic behavior; on the other hand, it implies higher relative Mach number at impeller eye with the effects described above on both efficiency and operating range.
New stage impeller, as well as the baseline, is a mixed-flow type. The advantage of this choice with respect to the purely radial design has been already proved [2] and it is justified by the high flow coefficient target (greater than 0.12) along with the high relative Mach number at shroud.

The details provided in the subsequent paragraphs about design, expected performance and test results refer to the following non-dimensional design parameters, for both new and baseline stage:

- Flow coefficient φ 0.16
- Peripheral Mach number Mu 1.05

Baseline represents the best-in-class geometry available for OEM, for the same flow coefficient and Mach number (same application).
As it will be described in other sections of the present paper, specific features implemented to optimize aerodynamic behavior, make impeller design challenging also from mechanical standpoint. Impeller design has been therefore carried out with a multidisciplinary approach, ensuring the best compromise between aerodynamic and mechanical performance.

Stator Components Design

Within centrifugal compressor stages, also the design of stator components is dependent on the specific combination of flow coefficient and peripheral Mach number, primarily because of the flow profile delivered by the impeller to the diffuser. High flow coefficients mean indeed large channels width; along with this, high relative Mach number at impeller eye (with subsequent shock at blade tip) stresses even more the difference in flow conditions between hub and shroud, affecting span-wise uniformity up to impeller exit and diffuser inlet. High efficiency stages require therefore properly designed stator components able to handle potentially non-uniform flows, in a wide range of operating conditions. The following features have been selected for the stator of the new stage:

- Vaneless diffuser
- Three-dimensional return channel blades.
-

Nevertheless, best stator behavior cannot be achieved without a proper impeller design, also aimed to the optimization of flow conditions at diffuser inlet.

CFD ANALYSIS

High relative Mach number represents a challenge for CFD predictability on centrifugal compressors stage. This is primarily because of the existence of supersonic and subsonic regions in the same domain and the shock occurring within the rotor blade row. Numerical simulation setup as well as mesh grid generation have been properly carried on to correctly capture the strong gradients which characterize the flow field across the shock and their effect on overall performance prediction.

CFD analyses have been performed by means of a proprietary code for computational fluid dynamic, based on a 3D multiblock, multigrid, structured and unstructured, non-linear and linear Euler/Navier-Stokes solver for turbomachinery blade rows. TACOMA (Turbine And COmpressor Analysis) is a cell-centered explicit flow solver based on the so-called JST scheme; the solution is obtained via a multi-step Runge-Kutta explicit time marching scheme with convergence acceleration via local time step, residual averaging and V-cycle or W-cycle multigrid. The results shown in the present analysis are based on 3D Reynolds-Averaged Navier-Stokes (RANS) equations, coupled with the two equations k-ω turbulence model developed by Wilcox [3]. The production modification of Launder and Kato [4] has been used, instead of the original one.

All CFD analyses here presented have been run in steady state conditions, on single vane, periodic domains. A mixing-plane has been used at the interface between rotor and stator domains. Simulation setup has been done according to last available best practices.
The OEM considers this type of CFD analysis reliable for stage performance prediction within an acceptable uncertainty, leveraging its own experience on numerical simulations and data match with experimental results.
Expected performance is shown in Figure 4, in terms polytropic efficiency and head coefficient. Data are normalized on design point of the baseline stage. The overall performance parameters are summarized below, with respect to baseline stage:

- Polytropic design efficiency 3% higher
- Left and Right limits aligned
- Design head coefficient 4% lower

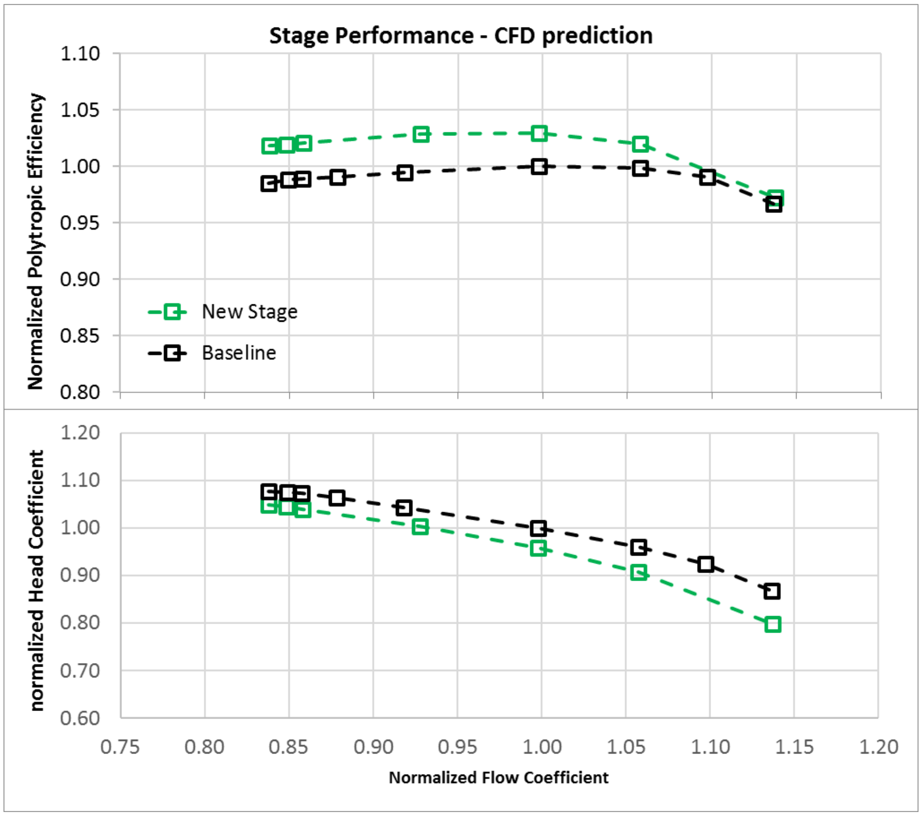


Figure 4 - Performance prediction by CFD analysis. Data are normalized with performance of baseline stage at design point. Flow coefficient on x-axis is normalized with design value (0.16 for both new and baseline stage)

The improvement of impeller behavior is shown in Figure 5 where the maps of total relative pressure loss coefficient are depicted at several stream-wise positions. Most of the losses inside the rotor channel are at suction side, nearby the shroud, where flow velocity is higher and shock in the relative frame occurs. New stage shows a smaller region of high losses, with respect to baseline, despite of the higher stiffness which implies higher relative Mach number.
In the new stage, the diffuser performance is improved as well, as it can be observed in Figure 6 and. Figure 7, where entropy and vorticity magnitude maps are shown. The pictures highlight also the improved flow field distortion at both impeller and diffuser exits. The return channel benefits from this and provides overall higher performances in terms of loss coefficient and static pressure recovery coefficient (normalized values in Figure 8).

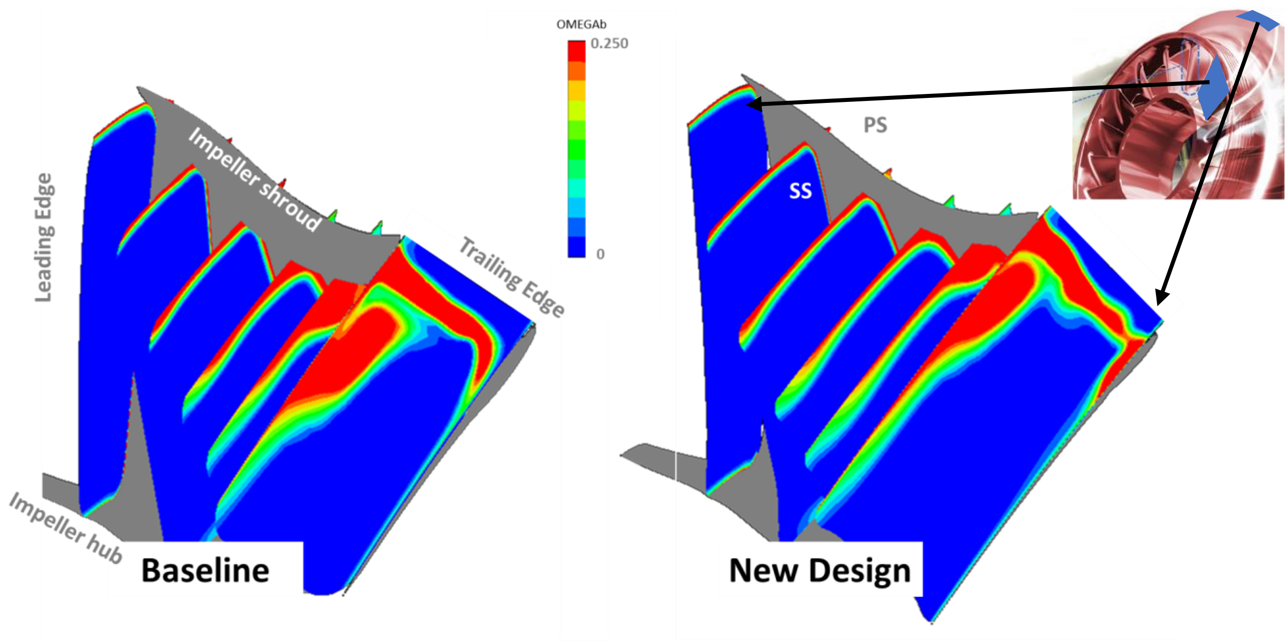


Figure 5 - Relative total pressure loss coefficient in the impeller; blue means isentropic, increasing losses towards red.

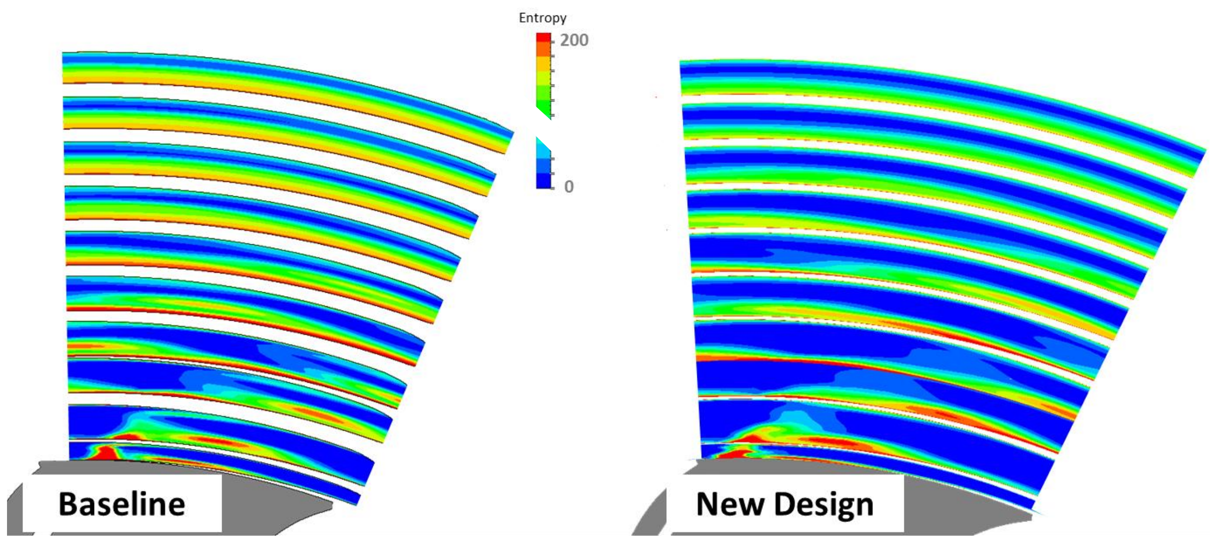


Figure 6 - Flow field at impeller exit and within diffuser channel, in terms of entropy

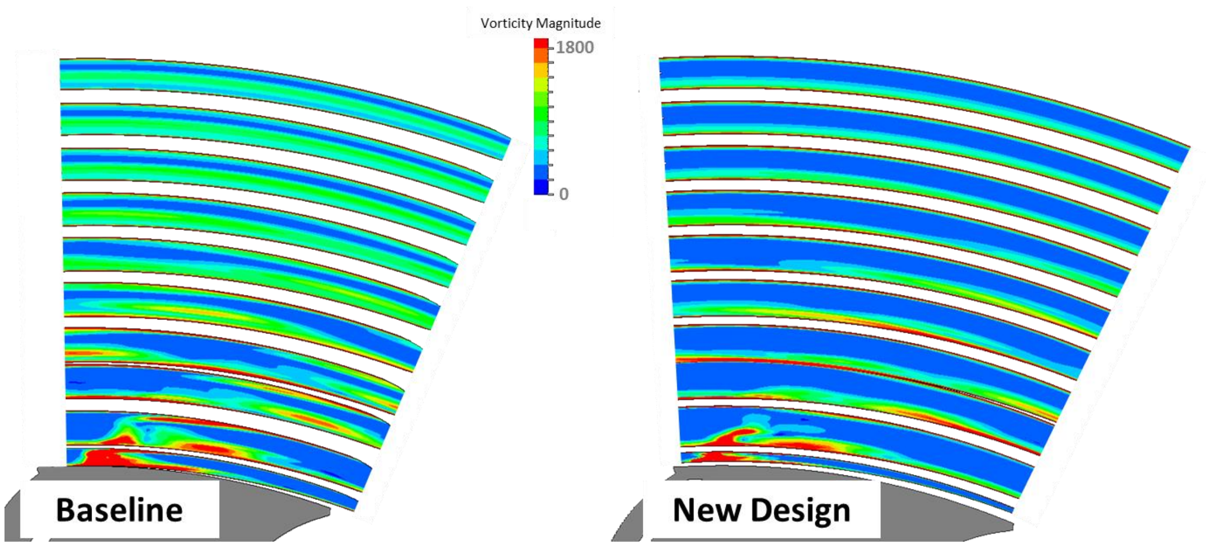


Figure 7 - Flow field at impeller exit and within diffuser channel, in terms of vorticity magnitude

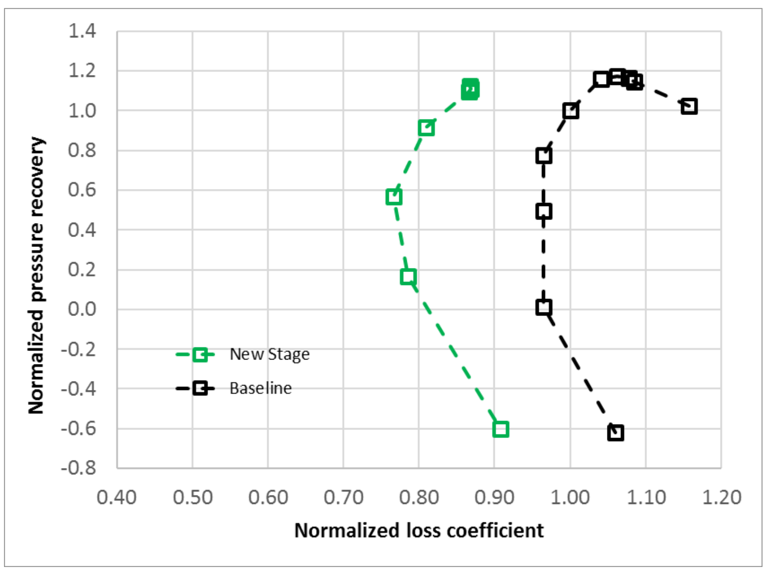


Figure 8 - New and old design are compared in terms of normalized performance of return channel

Left Limit prediction

Steady state condition is a realistic assumption to describe turbomachinery flow field as long as stall does not occur, neither in impeller nor in stator. This means it is valid through most of the stable operating range of the machine. In OEM experience, a reliable criterion to detect left limit of stage operating range is to watch at incipient unsteadiness. The most important effects which could be observed by decreasing the impeller flow rate (at constant speed) are:

- Numerical instability of the steady CFD simulation or decay of the quality on solution convergence
- Decreasing head coefficient curve
- Discontinuity in work coefficient curve

The discontinuity on work coefficient curve represents a good criterion to identify left limit when stall occurs in rotor channel rather than in stator. This is a typical situation for transonic or supersonic stages; it is identified calculating the local slope, according to Equation (1), where index "*i*" increases while flow coefficient φ decreases.

$$\left(\frac{\Delta\tau}{\Delta\varphi}\right)_i = \frac{\tau_i - \tau_{i+1}}{\varphi_{i+1} - \varphi_i} \qquad (1)$$

In Figure 9, the left limit is identified for the high flow coefficient, high peripheral Mach number stage by means of the criterion on work coefficient slope. Last three points on the left are considered outside the stable operating range, even though they are still numerically stable and characterized by a continuously rising head curve.

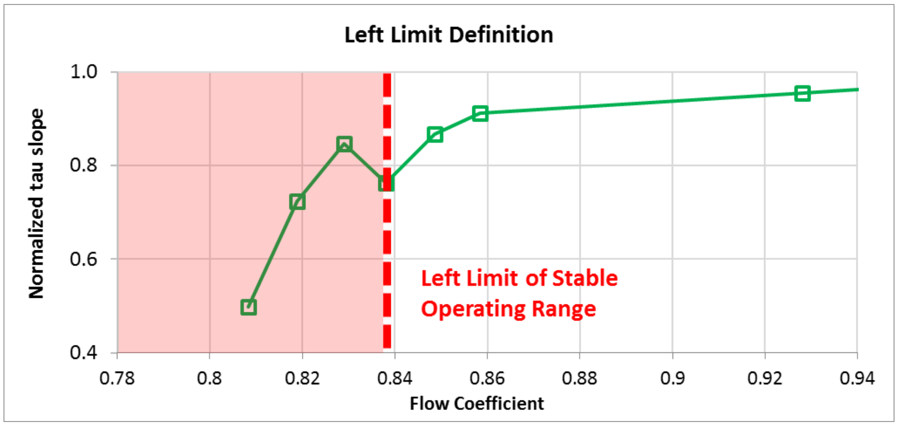


Figure 9 - tau-slope criterion for left limit detection on CFD analysis. Data are normalized on design point values. Flow coefficient on x-axis is normalized with design value.

STATIC DESIGN

The design of high flow high Mach and high stiffness impeller is a challenge for mechanical team requiring a lot of effort to contain the stress and the deformation during operating conditions.

Before going in the detail of design process it is worth to describe which is the mechanical behavior of this type of structures.

Generally, for high flow impeller the main driver of the static stress is the congruence of displacements between hub shroud and blades. To improve the aerodynamic performance blades are often twisted in backward direction at leading edge respect to a pure radial stacking of streamlines. This twist angle normally decreases going toward the trailing edge of the impeller resulting in pseaudo axial stacking. This concept is clearly depicted in Figure 10 where the design of high Mach high flow impeller is compared with a current high flow impeller. The leading edge of high Mach high flow impeller is tilted in backward direction and the it stacking axis is bended to reduce the aerodynamic load in the shroud area.

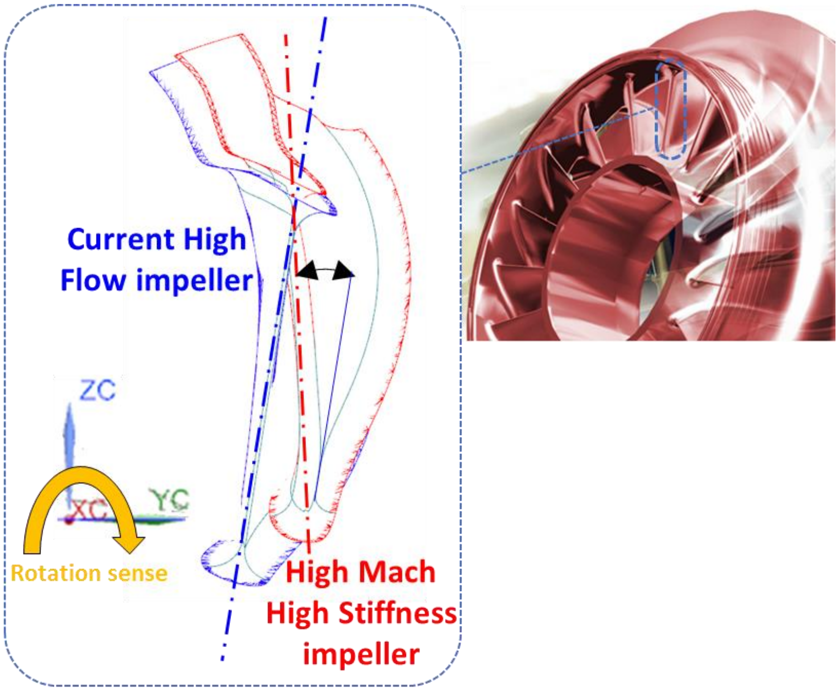


Figure 10 - 3D features of high Mach high flow impeller

Under the centrifugal load the blade tends to recover this twist angle with a deflection in hoop direction mainly at leading edge where the height span is higher, trying to become aligned to radial direction. The gradient of hoop deflection in axial direction generates a relative rotation of blade stacking axis that is in principle not allowed by hub and shroud creating a local stress imposed by the displacement compliance. Therefore, the blade is subjected to a distributed bending stress at its root whereas hub and shroud are loaded with a local bending component at junction with blade that is superimposed to the hoop stress generated by centrifugal load. This concentrated stress could in principle limit the centrifugal load capability of impeller dressing, creating the condition for local plasticization and, in case of very brittle material, a potential fracture. This effect is more severe for high Mach high stiffness impeller where the reduction of blade thickness and the increase of hub diameter create the condition for high deformations.

Being the stress generated by imposed congruence between two or more sub-structures, its reduction is not easy and a manual optimization of local stress playing with thickness distribution of blade and disks isn't a straightforward process. With the above scenario it is worth the use of an integrated optimization process that allows to find a good design with a reduced analysis time. The core of optimization process is the integration of CAD tool for the generation of slice model and automatic FE program for impeller static analysis and post processing of results. The impeller dressing is defined with a text file that contain the hub and shroud thickness distribution at different radial positions. All generated geometries employ the same aero design and the blade thickness can be changed only at specific locations and within the tolerance specified by aerodynamic team.

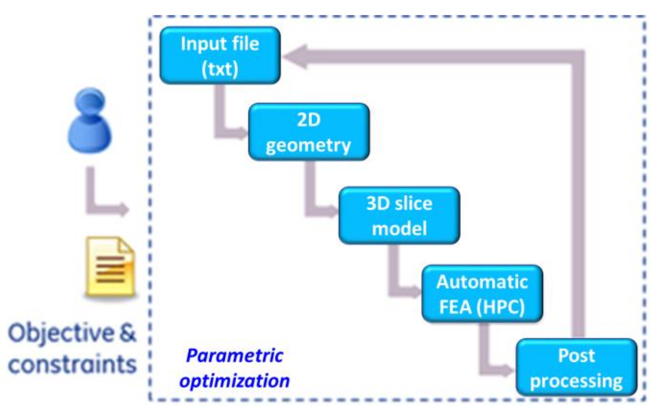


Figure 11 - Parametric optimization process for static design

The FE analysis are done with linear material properties and a constraint of peak stress is imposed to make coherent the stiffness between the involved substructures and reduce the stress due to compliance. Additional constraints are the impeller axial span and the eyes seal diameter. The target given to optimization toll is the reduction of impeller weight, finding the best impeller design once yielding material limit and peripheral speed are imposed.
In the optimization phase the linear peak stress is limited to "design" value order to have a partial local yielding area during over speed test, that improves the fatigue behavior of impeller. The cross of yielding limit by peak stress creates a local increase of material yielding value and on the other hands generates beneficial compressive stress that reduce the tensile stress during operative conditions.
The design process is completed with non-linear analysis of optimized dressing to verify the robustness of previous design phase and the entity of total deformation and the relief of shrinkage that can't be captured with a simple linear analysis.

DYNAMIC DESIGN

The dynamic analysis of impeller behavior is already introduced in standard design process of new stages to predict any potential High Cycle Fatigue failure due to unavailable resonance during operative conditions. This design step becomes mandatory for high flow and high Mach impellers, where the high blade radial span together with reduced thickness move down the frequency of many basic blade modes. Moreover, the use of optimized disk thickness creates the condition for the presence of coupled disk-blades modes that can be critical for the unlucky matching between static and dynamic stress distribution.
Figure 12 shows a comparison between the modes of two different solutions of the static optimization process, obtained with the same aero blade design and changing the shroud thickness. Both configurations satisfy the static criteria.
The first mode is a pure leading-edge mode whereas the second is a coupled disk-blade mode with leading edge participation. The former shows higher amplitude at blade and therefore is more excitable by upstream wakes typical of high Mach applications. The latter, despite the modal amplitude at blades is reduced, shows a shape with shroud participation and a potential limited vibration capability due to the matching between static and dynamic stress distribution. The tradeoff between two solutions is based on a forced response analysis and fatigue safety factor evaluation.

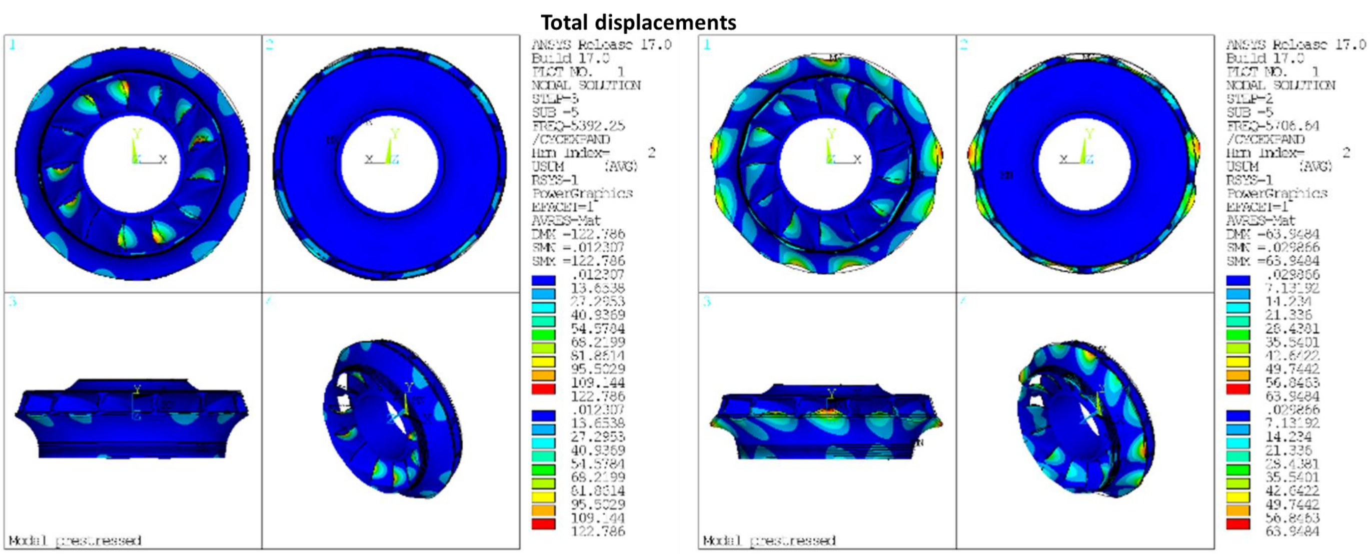


Figure 12 - Comparison between pure LE mode and coupled blade-disk mode for high Mach high flow impeller

The methodology and the predictability level of forced response analysis and aerodynamic damping computation is well described in [5]. Nevertheless, the introduction of this type of computation in the design loop is too time consuming and doesn't fit with the requirements of typical optimization tools, limiting the integration of those two types of analysis.
To fulfill the requirement of dynamic validation of each solution proposed by optimization process and the limitation of computation time, a simplified version of forced response analysis has been developed by author's OEM making effective and efficient the integration of those types of analysis.
The simplified approach is manly based on the availability of an extensive database of modal force and aero damping values obtained with several forced response analyses and aeromechanic test post processing carried out during last years within the impeller developing programs.
The database is organized on the base of type of rotor-stator interaction, i.e. inlet plenum vs return channel or vane diffuser, application type, i.e. high Mach vs low Mach, and impeller modes. In addition, to generalize and extend the use of the proprietary database some

physic rules have been defined to scale the modal force and modal damping value for different operating conditions or different design. In this way, the dynamic verification can be easily introduced in optimization loop because it is reduced to a mathematical scaling of modal stress and the Goodman diagram construction, incrementing the computation time of each loop just for the time of an additional modal analysis. Doing so, each modification of dressing thickness or blade design can be made effective and easily quantified in term of fatigue strength of impeller.
An example of this approach is reported in Figure 13. The two designs differ for the extension of central streamline at leading edge and for shroud thickness at trailing edge. These differences affect the frequency of the modes and the associated modal amplitude and modal stress. It is possible to note that, also if the mode type and the position of critical point of Goodmam diagram are the same, and the same is the level of static stress, the design reported in the right side of Figure 13 shows a fatigue safety factor increased of 50%.
Once the final impeller design is chosen a complete aeromechanical analysis with unsteady CFD and modal force calculation is done to quantify the fatigue safety factor in the worst operating conditions.

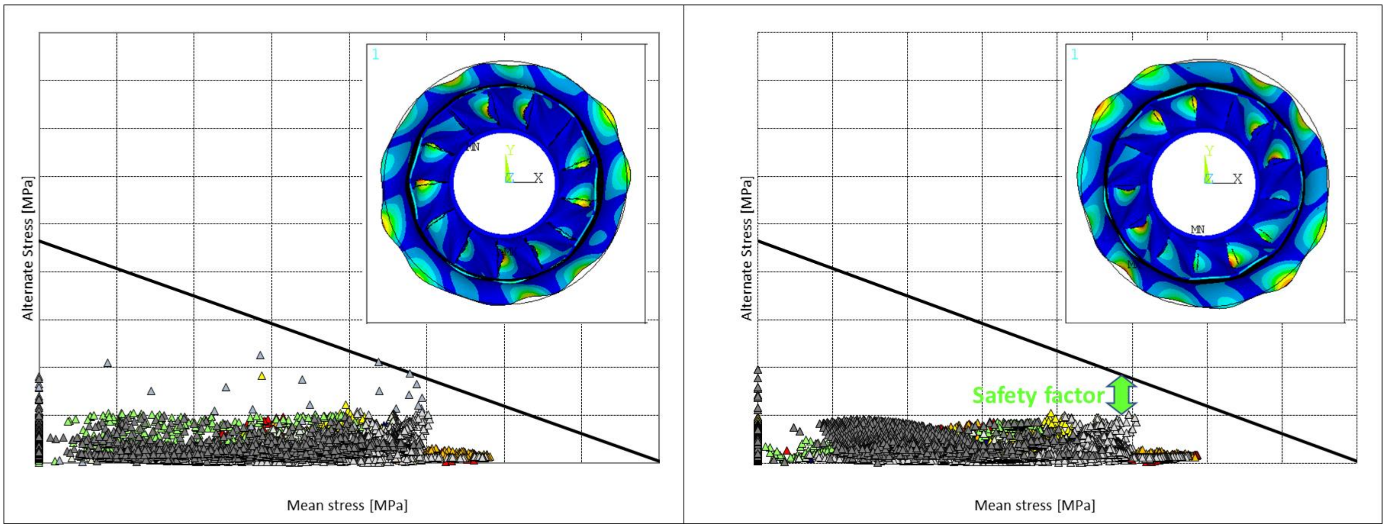


Figure 13 - Simplified forced response analysis for two different designs of high Mach high flow impeller

TEST DESCRIPTION

The performance of new stage design has been validated through a model test campaign. Test rig have been equipped with dynamic and static probes at different measure locations (see Figure 14) so to be able to evaluate flange-to-flange as well as single component performance. Pressure, temperature, flow angle and velocity measurements have been performed at each section; dynamic probes have been properly installed to detect occurrence of stall and surge.

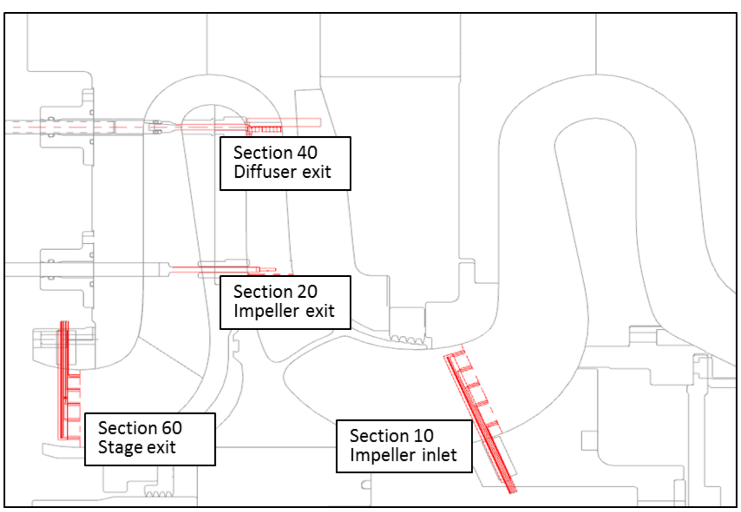


Figure 14 – Schematic of test rig configuration

The test rig setup allows a complete characterization of stage and all its components at standard impeller diameter D_2.(390 mm) in many different layouts, such as first stage (with inlet plenum upstream), intermediate stage (also with side-stream injection), intermediate stage and axial inlet. Well established process is followed to leverage similitude rules and obtain stage performance at actual machine conditions.

STAGES PERFORMANCE

Performance shown in the following figures refers to the new stage design at high flow coefficient and high peripheral Mach number All the results are at design peripheral Mach number, with performance coefficients normalized over baseline stage at design point. Experimental data are substantially aligned to expected performance:

- New stage provides overall efficiency increase above 2.5% (with respect to baseline) in a wide portion of operating range.
- Higher stiffness affects the right limit of operating range, where chock occurs at higher flow coefficient than baseline; anyway, efficiency curve is above the baseline up to 114% of design flow coefficient, where the baseline itself gives an efficiency decay of 10% with respect to peak efficiency
- Left part of the operating range is 1% wider than the baseline stage
- Head coefficient is lower than baseline by 2% at design point

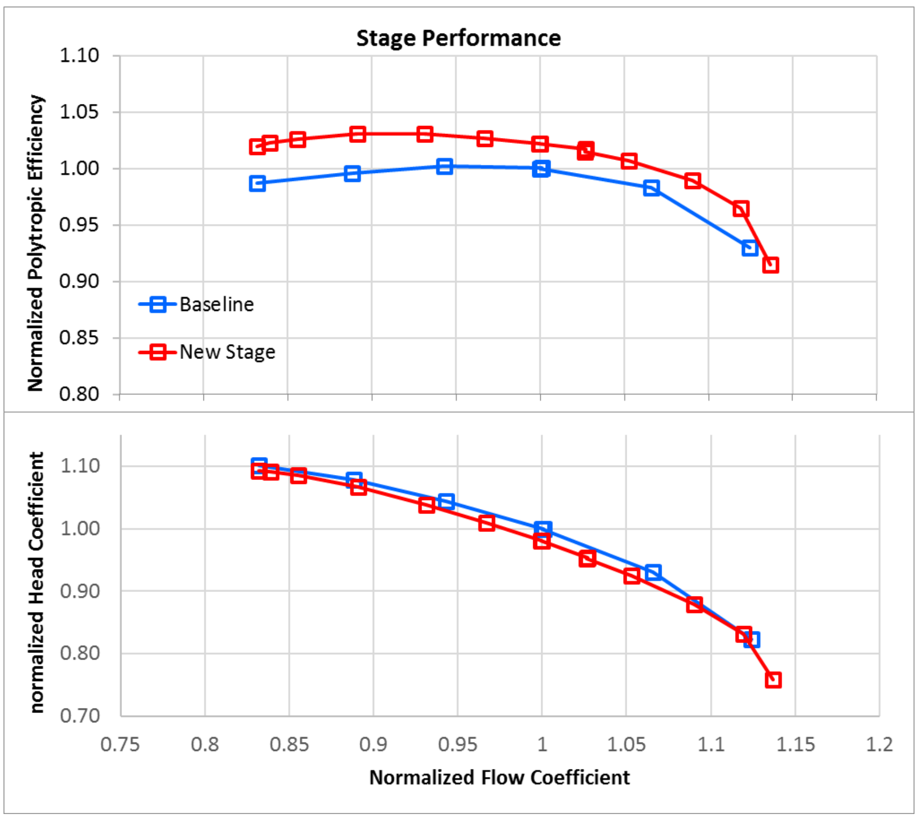


Figure 15 - Performance curves of the new stage compared to the baseline – Data are normalized on the baseline design point values Flow coefficient on x-axis is normalized with design value (0.16 for both new and baseline stage)

CASE STUDY

Global liquefaction capacity in 2016 and 2015 grew at a similar rate, reaching a global nominal capacity of 340 MTPA in January 2017 (IGU World LNG Report — 2017 Edition). In 2016, global LNG trade reached a new record of 258.0 MT, the third consecutive year of incremental growth. The majority of supply growth was supported by commercial production at multiple new liquefaction plants in East and West Australia, and commissioning production at a new train in Malaysia. The average liquefaction capacity of the new plants under construction is around 5 MTPA, aligned with the trend of the last projects, and several proposals with capacity above 10 MTPA are

ongoing.
For the above reason, the LNG trend is seen to be stable and with continues development of new plants. To improve the production, the end users tend to increase the compressor flow capacity in order to maximize the liquified gas output per single plant.
The OEM is developing and proposing to the market new products to accommodate these requirements, improving at the same time the efficiency and the available power.

The OEM has developed a series of new turbines with the aim to provide reliable and efficient drivers for the different LNG plant sizes and configurations. The power and the operating range of the turbines has been optimized to cover all the sizes of the future plants, ranging from 52 MW (LM6000PF+) to 85MW (MS7001EA), with in between the new LM9000 with 75 MW of rated power (mature phase development). This continuous range of power supply perfectly fits the different combinations of train arrangements and the multiple LNG process compression configurations.

The authors have applied the new stages in a feasibility study for a new LNG plant having the capacity of 6.3 million tons of LNG per annum, studying the different combination of trains and turbines. The selected LNG process is the AP-C3MR, as it represents the majority of the processes; in this process, the compressors are used to boost propane gas and a mixed refrigerant gas.
The different combination of compressors and turbines are described in the following figures.

In the case with LM6000PF+, the following combination of trains have been studied (Figure 16):

- 2x [LM6000PF+ + Propane compressor] + 4x [LM6000PF+ Mixed Refrigerant 1 + Mixed Refrigerant 2];
- 3x [LM6000PF+ + Propane compressor + Mixed Refrigerant 1] + 3x [LM6000PF+ + Mixed Refrigerant 2] ;
- 6x [LM6000PF+ + Propane compressor + Mixed Refrigerant 1 + Mixed Refrigerant 2].

N	Configuration	N° turbines	N° Compressors
1	2x [PF+ + C3] + 4x [PF+ + MR1 + MR2]	6	10
2	3x [PF+ + C3 + MR2] + 3x [PF+ + MR1]	6	9
3	6x [PF+ + C3 + MR 2 + MR 1]	6	18

Table 1 – LM6000 PF+ Train Configurations.

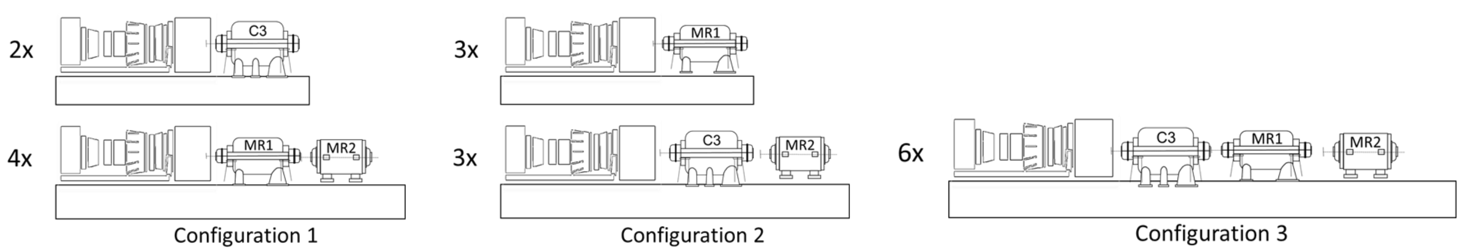


Figure 16 – AP-C3MR train configurations with LM6000PF+

The number of turbines is the same for all three configurations of compressor arrangement, but the number of compressors can vary from 9 to 18. The machine number ratio may be slightly different if we consider also the spare trains but, as this is a choice that depends on the end-user philosophy, we don't consider in this analysis any spare train.
Configuration n°1 and 2 of Table 1 have a significant improvement in capex compared to configuration n°3. In fact, the number of compressors is half and the balance of plant may benefit of the lower number of pipes and connections needed to bring the gas to the compressors and again to the plant. However, configuration n° 3, has much higher flexibility, thanks to the higher number of trains in parallel.
If one train of propane of case n°1 would have an unexpected stop, the whole production would suffer a reduction by half, whilst if the same failure would happen in the case n°3, the production would be reduced just by 1/6.

In the case with LM9000, the following combination of trains have been studied (Figure 17):

- 1x [LM9000 + Propane compressor + Helper] + 2÷3 x [LM9000 + Mixed Refrigerant 1 + Mixed Refrigerant 2];
- 2x [LM9000 + Mixed Refrigerant 1] + 2x [LM9000 + Propane compressor + Mixed Refrigerant 2];
- 4x [LM9000 + Propane compressor + Mixed Refrigerant 1 + Mixed Refrigerant 2];

N	Configuration	N° turbines	N° Compressors	Helper (EM)
1a	1x [LM9000 + C3 +H] + 2 x [LM9000 + MR1 + MR2+ H]	3	5	2
1b	1x [LM9000 + C3 +H] + 3 x [LM9000 + MR1 + MR2]	4	7	1
2	2x [LM9000 + MR1] + 2 x [LM9000 + C3 + MR2]	4	6	0
3	4 x [LM9000 + C3 + MR1 + MR2]	4	12	0

Table 2 – LM9000 Train Configurations.

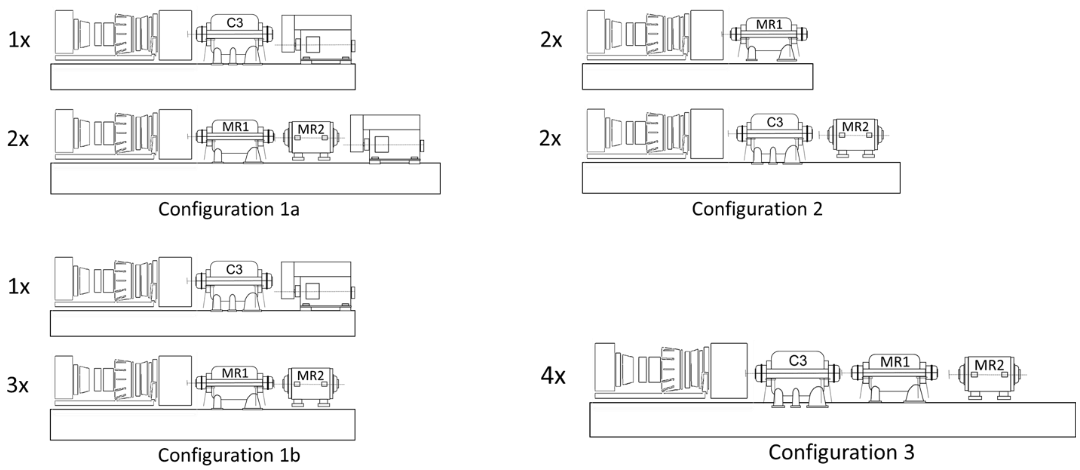


Figure 17 - AP-C3MR train configurations with LM9000

Also in this case, number of turbine is the same, expect if for the 1st configuration the addition of a helper is allowed. As the power required by the propane compression is above 80 MW, for the configuration 1x LM9000 + C3, a helper is needed to match that size of power.
Reducing the number of trains is a way to reduce the number of compressors, keeping the CAPEX lower. However, the flexibility of the plant may be compromised, exposing the production to an higher risk in case of failure.

In the case with MS7001EA, the following combination of trains have been studied (Figure 18):

- 1x [MS7001EA + Propane compressor + Helper] + 2 x [MS7001EA + Mixed Refrigerant 1+ Mixed Refrigerant 2+ Helper];
- 2x [MS7001EA + Mixed Refrigerant 1] + 2x [MS7001EA + Propane compressor + Mixed Refrigerant 2];
- 4x [MS7001EA + Propane compressor + Mixed Refrigerant 1 + Mixed Refrigerant 2];

N	Configuration	N° turbines	N° Compressors	Helper (EM)
1	1x [MS7001EA + C3 +H] + 2 x [MS7001EA + MR1 + MR2 + H]	3	5	2
2	2x [MS7001EA + MR1] + 2 x [MS7001EA + C3 + MR2]	4	6	0
3	4 x [MS7001EA + C3 + MR1 + MR2]	4	12	0

Table 3 – MS7001EA Train Configurations.

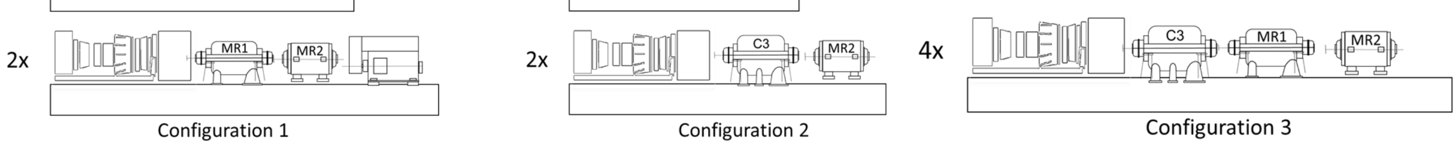


Figure 18 - AP-C3MR train configurations with MS7001EA

The results are the same already discussed for the cases with LM9000, except for the fact that the electric motors used as helper are of lower size, as the difference between the maximum turbine output and the compressor requirements is much lower.

Each configuration described above would bring its own selection of compressor and impeller stages. As the rotational speed of the power shaft of each turbine is different, to provide the same head, the impeller diameter is changed to match the peripheral speed. Some adjustments are then necessary, as the flow coefficient and in turn the stage efficiency are different, to provide the same final work.

The configurations selected with the same driver have in most of the cases the same impeller size and peripheral Mach number, as the same pressure ratio has to be provided, with different flow coefficients, as the elaborated mass is differently split among the trains. In the cases where the single propane saturates all the power of the turbine (1x C3), the stages have the highest flow coefficients (see Figure 20). This leads to a very large stage span and very long rotors, making the rotordynamics very challenging; due to the slenderness of the rotors, the separation between the operating range and the second critical speed is significantly reduced. This has been cured developing the new stages with higher stiffness compared to the traditional used ones, increasing the shaft diameter and improving the rotor stiffness.

In Figure 19 the flow paths of the different configurations of Propane compressor are reported. The rotating speed is fixed by the turbine; hence the impeller diameter is imposed by the head required by each section, with the result of having the same impeller size among all the configurations. Increasing the compressor mass flow has the only effect of increasing the impeller flow coefficient and in turn the stage span, keeping the same shaft diameter, resulting in a very slender shaft and challenging rotordynamics. For this reason, the new design of impeller has been developed increasing by 10% the rotor stiffness compared to traditional stages used in LNG applications.

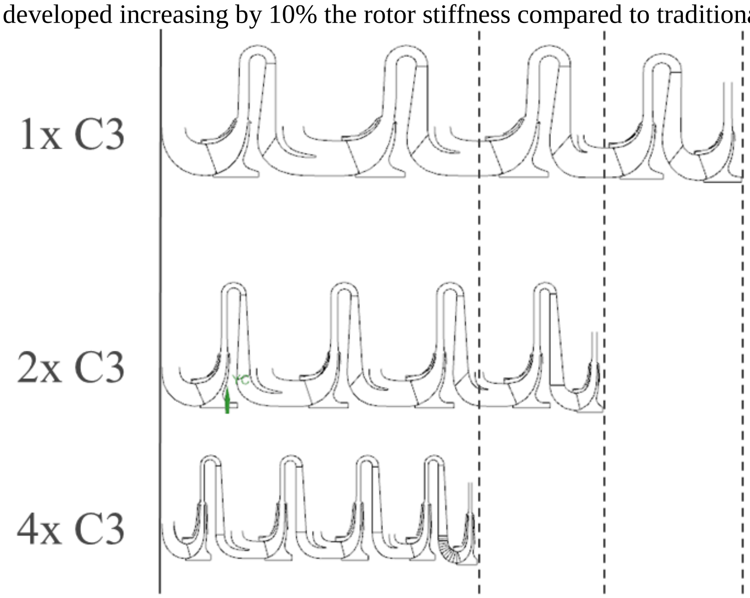


Figure 19 – Rotor configuration for 1x, 2x and 4x propane trains.

In Figure 20 the stages selected for the propane compressors are reported. Reducing the number of trains, the flow coefficient of the stages is significantly increased; for instance, in the case of 1x C3, all the mass flow is elaborate by the single Propane. Considering the cases with the same driver, although the rotational speed is the same, the peripheral Mach is not kept constant among the selections. This is due to the different stage performances at the different design flow coefficients; in the very high flow coefficient range, the work coefficient and the efficiency are lower compared to the middle flow coefficient design. To recover the reduction in head, it is necessary to change the impeller diameter, increasing the peripheral speed and so the peripheral Mach number.
In the medium-high area, the performances are aligned with the medium-low flow coefficient region, as the new stages designed for LNG application are optimized for this area, considering since from the initial steps the range of flow coefficient and Mu where the stages will be operated. This has been possible, as the OEM has focused its efforts on the joined design of turbines and compressor stages, to find the optimum solution.

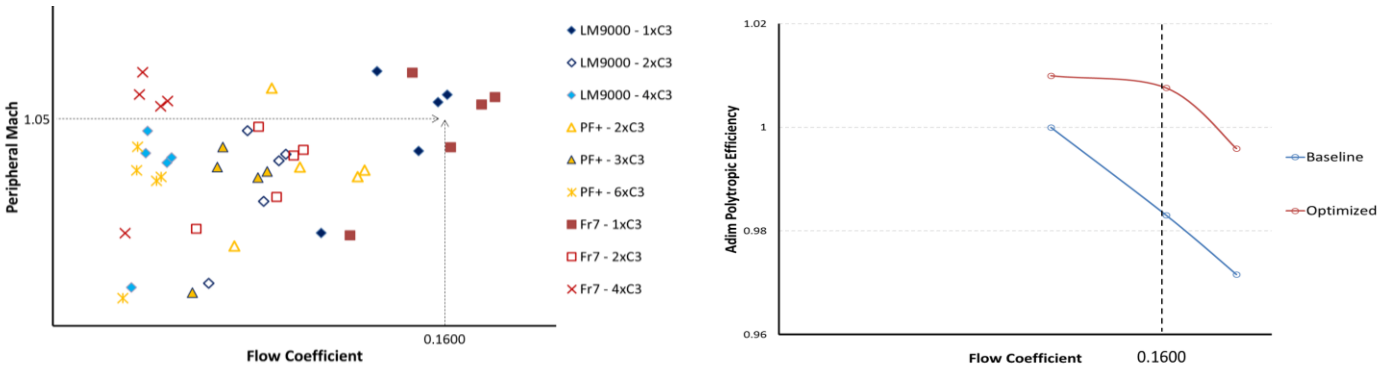


Figure 20 – Selected stages for the different Propane train configurations (left) and performance difference between baseline stage and optimized one (right). The symbols on the legend represent the train configurations described in this chapter.

In Figure 20 the difference in performance between the optimized stages and the baseline stages is reported for different flow coefficients. It is evident from the chart that the optimization has been directed to improve the efficiency in the medium-high flow coefficient ranges, to follow the market trend of increasing the mass flow elaborate by the compressor.

The power consumption of the Propane compressors for the different configurations of train is reported in Figure 21. The most promising configurations are the intermediate ones, the so-called *configuration 2* of the previous tables. This is due to the best combination of stage flow coefficients in the compressors, in the middle-high range of Figure 20, in the area of highest polytrophic efficiency. The additional variation among the configurations with different drivers is due to the different rotational speeds of each driver, that bring the selections in separate flow coefficient areas. On the chart on the right of Figure 21, the difference between the compressors equipped with the baseline stages and the optimized ones is reported. The intermediate configuration is still the optimal one from the power consumption standpoint, but thanks to the new stages the variation among the cases has changed. As highlighted above, the new stages have been optimized to be employed in the high and medium flow coefficient area (Figure 20), as those stages are able to keep a satisfactory level of efficiency in a wider range.
Thanks to the new stages, the compressors of configuration 1 (high flow coefficients) show a significant improvement in efficiency, more evident compared to configuration 2 and even more compared to configuration 3. The difference between configuration 2 and configuration 1 in terms of power consumption is now less than 2%, while in the old case it was more than 3% (left chart).
Considering the improvement brought by the new stages, configurations 1 are now more attractive, considering the power consumption almost aligned with the other configurations and the reduction in capex thanks to the reduction of the number of compressors and turbines.

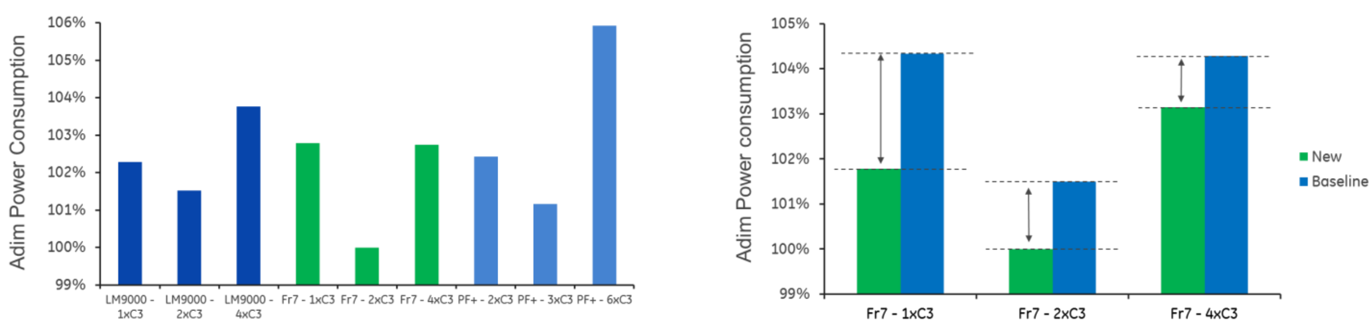


Figure 21 – Adimensional power consumption for the different train configurations (left) and comparison between the baseline stages and the new one (right) in the case of MS7001EA

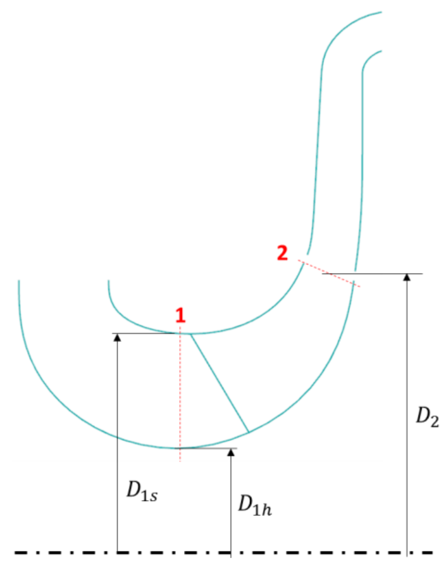
CONCLUSIONS

LNG production is expected to grow in the next years to keep pace with the increase of Energy Demand and expected conversion from fuel oil and coal to gas. This change is possible only if the gas and so also LNG remains in a competitive price range. This land on the need to reduce the cost of LNG plant infrastructures and one of them is the design standardization, and larger individual train capacity.

For this reason, the OEM has invested significant efforts to improve large size gas turbines and the driven refrigerant compressors. In particular for those dealing with high molecular weight, the challenge is to keep a good level of efficiency and operating range even if increasing the flow coefficient and peripheral Mach number. This challenge becomes even more difficult if associated to the mechanical constraints, dictated to the need of a robust rotodynamic and so a high rotor stiffness. This paper illustrates the possibility to improve the present centrifugal compressor efficiency, even if facing these new challenges, and how detailed CFD and multidisciplinary design optimization can support on this duty. The paper also illustrates, for a specific case study of 6.6 MTPA of LNG production, how different Gas Turbine and compressor arrangements can change the overall train absorbed power and reliability. Finally it will be clear how the improved stage design can help in making all the train solution more efficient, helping in particular the larger capacity configuration.

NOMENCLATURE

Mu	Peripheral Mach number $\frac{u_2}{a_1}$
φ	flow coefficient defined as $\frac{4\,Q_1}{D_2^2\,u_2\pi}$
η	efficiency
τ	work coefficient defined as $\frac{H_{02}-H_{01}}{u_2^2}$
ψ	head coefficient defined as $\eta\ \tau$
ω_b	relative total pressure loss coefficient, defined as $\frac{p_{0,rel,2is}-p_{0,rel,2}}{p_{0,rel,1}-p_1}$
H	enthalpy
Q	volumetric flowrate
D	diameter
a	speed of sound
c	absolute flow velocity
u	peripheral speed
w	relative flow velocity
p	pressure
P	power
Mrel	Mach number in relative frame
v	flow velocity distortion
Subsripts	
m	meridional component
h	hub
s	shroud
rel	relative to rotating frame of reference
0	total condition
1	impeller inlet
2	impeller exit
p	*polytropic*
is	isentropic



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