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Effects of upgraded cooling system and new blade materials on a real gas turbine performance

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Abstract

The aim of this paper is to study the effects on the performance of a real heavy-duty gas turbine of two different solutions for enhancing turbine blades thermal resistance. An upgrade of the first stator cooling system and the adoption of improved blade materials are simulated exploiting an in-house simulation tool (ESMS). The changes are studied separately in order to point out the positive effects as well as the related risks, such as the side effect of temperature increase on downstream blade rows, to be considered in service operations.

Keywords: Cooling system; gas turbines, performance analysis; modular tool.

1. Introduction

The Turbine Inlet Temperature (TIT) increase represents a direct drive to achieve higher thermal efficiency and higher power output of a gas turbine [1]. Consequently, combustor cooling and pollutant emission [2]–[4] as well as blade cooling [5], [6] issues arise. The constant development of materials as well as efficient blade cooling systems allowed to reach TIT values higher than metal melting temperature.

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Specific manufacturing techniques, like directionally-solidification and single-crystal castings developed appositely for increasing gas turbines parts life, use single crystal super-alloys (titanium, or super-alloys of nickel and cobalt) guaranteeing satisfying thermo-physical properties [7], [8]. A further increase of the engine operating temperature it is possible only developing new materials, resistant to higher temperatures, and by efficiently cooled gas turbine components, ensuring them an adequate lifetime. Despite all the efforts performed by material's designers, it would not be possible to reach TIT values higher than metal melting temperature without the presence of an accurate cooling system. Most of the recent improvements come from a greater understanding of the heat transfer and the three-dimensional temperature distribution in the turbine passage [6]. In [9] five main elements are identified for the conventional cooling technology applied on gas turbines: internal convective cooling, external surface film cooling, materials selection, thermo-mechanical design at both component and system levels, selection pretreatment of the coolant fluid. These aspects define the thermal management capabilities and limitations in modern gas turbines that nowadays can reach very advanced requirements. Any further increases in efficiency and lifespan may be obtained by small incremental improvements or by completely changing cooling systems' manufacturing process [10], [11].

Actual research focuses on the development of completely new cooling systems exploiting modern manufacturing technologies and processes such as additive manufacturing techniques like Direct Metal Laser Sintering. Additive manufacturing draws the path for innovative gas turbine cooling, allowing it the realization of the component and of its cooling system at the same time. New perspectives are therefore offered for servicing and performance upgrading [5].

In the present work, new materials and high-efficient innovative cooling systems are studied evaluating their effects on the performance of a real gas turbine used for energy production. A numerical model of the engine is realized and simulations are performed exploiting an internally developed modular tool ESMS (Equation Solver Modular System). The code has been extensively tested over the years on thermodynamic, design and off-design problems concerning gas turbines [12], [13], gas-steam [14] and ORC [15] cycles for power generation or mechanical drive.

In particular, the design of an existing gas turbine is performed at first. Successively, off-design simulations are conducted modelling a redesign of the first stator cooling system. A more efficient cooling systems is introduced by an increase of the blade cooling efficiency parameter within the code. As a second investigation, a replacement of the original blades with upgraded ones through increased admissible blade temperature is modelled. Sensitivities are carried out to the blade temperature and cooling efficiency on a range of operating conditions reducing the cooling mass flow and increasing the TIT, starting from their design values.

A complete map is drown for the turbine performance in terms of power output and system efficiency. Considerations are made on the advantages introduced, design options as well as on undesirable side-effects due to gas temperature increase.

2. Numerical Model and Methodology

2.1. ESMS modular code

Gas turbines preliminary design analysis and global performance estimation can be conveniently carried out with 0-D modular tools where the entire process is modeled inside black box units. Within each unit, thermodynamic, chemical and mechanical processes are treated as input/output relations that modify the characteristic parameters between inlet and outlet ports. These units represent the elementary components (such as compressors, turbines, combustors, heat exchangers etc.) whose combination made up the gas turbine engine. Mathematical equations characterizing the physical transformations are described inside each unit and give rise to a numerical system that is solved by appropriate procedures mainly based on Newton-Raphson or Gauss-Jordan methods. This solving approach is essentially fully implicit since all the equations are solved simultaneously and the problem coherence is granted by the reliability of the given set of boundary conditions.

Compared to more complex 3-D tools, modular 0-D codes can achieve accurate details of the whole engine with short computational time. They are also flexible and need no modification to the source code when changing plant layout or cycle scheme, as dedicated codes do. Moreover, by simply adding new elementary components, modular tools can be expanded without modifying their global architecture (see Figure 1 for an example of scheme).

In order to carry out design and off-design simulations showed in this paper, the in-house numerical code ESMS has been used. ESMS is based on descriptive equations, derived from the principles of thermodynamics, heat transfer and conservation of momentum equations that are coupled with empirical correlations to allow, in design mode, the geometric characterization of the components. In off-design mode, assuming as boundary conditions the geometric parameters just defined, it is possible to assess the components response over different operating conditions. During the calculations, thermodynamic properties of real gases are made available with the use of an appropriate thermodynamic library.

2.2. Blade cooling theory

The main parameters for describing the performance of a cascade cooling system are the internal cooling efficiency, ε_h, and the global film cooling effectiveness, η_f defined as follows [16] and [12]:

$$\varepsilon_h = \frac{T_{c,e} - T_c}{T_b - T_c} \quad \text{Eq. 1} \qquad\qquad \eta_f = \frac{T_g - T_{a,b}}{T_g - T_{c,e}} \quad \text{Eq. 2}$$

Where $T_{c,e}$ is the exhaust coolant temperature, T_b the blade temperature, $T_{a,b}$ the adiabatic blade temperature and T_g the gas total temperature.

ε_h shows the capability of the cooling gas to remove the heat within the blade inner ducts: the closer the exhaust coolant temperature is to the blade temperature, the better is the internal cooling system.

η_f: estimates the level of coolant protection of the blade outer surface from the hot gases: the closer the adiabatic blade temperature is to the film coolant temperature, the higher is the achieved external protection.

Total energy conservation implies that the heat transferred by the hot gas to the outer surface of the blade is equal to the heat received by the cold gas inside the internal cooling system; this is true in steady state conditions, when the blade temperature is assumed to be constant. Introducing the definition of Stanton number, *St*, the coolant mass flow rate can thus be determined:

$$\frac{m_c}{m_g} = \mathrm{St}_g \frac{c_{p,g}}{c_{p,c}} \frac{A_b}{A_g} \frac{1}{\varepsilon_h} \frac{(T_{a,b} - T_b)}{(T_b - T_c)} \quad \text{Eq. 3}$$

Where the ratio of the specific heat for coolant and hot gases, $c_{p,g}/c_{p,c}$, and the equivalent blade to gas path area ratio, A_b/A_g, are introduced. This expression is valid for a generic cooled blade without restriction on the type of cooling system. A different form of equation can be obtained, introducing two dimensionless parameters, in order to better emphasize the contribution of internal and external cooling:

$$\frac{m_c}{m_g} = \mathrm{St}_g \frac{c_{p,g}}{c_{p,c}} \frac{A_b}{A_g} \frac{1}{\varepsilon_h} \frac{\varphi}{(1-\varphi)} \left(1 - \chi_f\right) \quad \text{Eq. 4} \qquad \varphi = \frac{T_g - T_b}{T_g - T_c} \quad \text{Eq. 5} \qquad \chi_f = \frac{T_g - T_{a,b}}{T_g - T_b} \quad \text{Eq. 6}$$

φ is called cooling effectiveness and indicates the overall blade protection. χ_f is the equivalent film cooling effectiveness and represents the weight of film cooling respect to the internal one; if the blade is only internally cooled then $T_{a,b}$ is equal to T_g. When film cooling is present, its effect decreases the adiabatic blade temperature, thus, assuming same hot gas conditions and internal cooling system, a lower amount of coolant is needed.

In design analysis, when the cooling system efficiency parameters are selected for the chosen application (see Section 2.3), this expression is used to evaluate the coolant mass flow that is needed to keep the blade temperature below the imposed admissible limit. This is done by putting T_b equal to the desired admissible temperature, according to the blade material.

Once the cooling system has been properly sized, different remarks must be made on this formulation to take into account the variation of gas turbine operating conditions. Introducing the Stanton number for the internal cooling gas and scaling all the parameters that depends on the modified hot gas and coolant conditions with simple off-design concerns, it is possible to write the off-design form of equation Eq. 4.

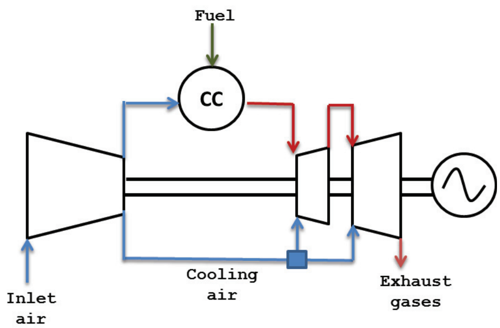
$$\frac{m_c}{m_g} = \mathrm{St}_{g,0}\left(\frac{m_g}{m_{g,0}}\right)^{n_g-1}\frac{c_{p,g}}{c_{p,c}}\frac{A_b}{A_g}\frac{A_{cp}}{A_g}\frac{1}{St_{c,0}}\left(\frac{m_{c,0}}{m_c}\right)^{n_c-1}\frac{\varphi}{(1-\varphi)}\left(1-\frac{\eta_{f,0}\left(\frac{m_c}{m_g}/\frac{m_{c,0}}{m_{g,0}}\right)^{n_1}}{\varphi}\right) \qquad \text{Eq. 7}$$

Where the pedex 0 refers to design values and n_g, n_c, n_1 are usually assumed to be respectively to 0.8, 0.8 and 1. Eq. 7 can be written in for the blade temperature as follows:

$$T_b = \frac{T_g \mathrm{St}_{g,0}\left(\frac{m_g}{m_{g,0}}\right)^{n_g-1}\frac{c_{p,g}}{c_{p,c}}\frac{A_b}{A_g}\frac{A_{cp}}{A_g}\frac{1}{St_{c,0}}\left(\frac{m_{c,0}}{m_c}\right)^{n_c-1}\left(1-\eta_{f,0}\left(\frac{m_c}{m_g}/\frac{m_{c,0}}{m_{g,0}}\right)^{n_1}\right)+T_c\eta_{f,0}\left(\frac{m_c}{m_g}/\frac{m_{c,0}}{m_{g,0}}\right)^{n_1}}{\frac{m_c}{m_g}+\frac{c_{p,g}}{c_{p,c}}\frac{A_b}{A_g}\frac{A_{cp}}{A_g}\frac{1}{St_{c,0}}\left(\frac{m_{c,0}}{m_c}\right)^{n_c-1}} \qquad \text{Eq.8}$$

2.3. Gas turbine design and off-design solution strategy

The investigated gas turbine is the GE MS9001E gas turbine at 50 Hz [17]. The chosen configuration is a GE E-class units burning natural gas. It is widely used in simple-cycle, combined-cycle, and industrial and mechanical drive applications. The studied version features 3-stage turbine module and a 17-stage compressor. It is fired at lower E-class temperatures for hot gas path.



Performance Parameter	Data sheet data	Model result	
Output (MW)	126.1	126.1	Input
Pressure ratio	12.6:1	12.6:1	
Mass flow (kg/s)	418	418	
Turbine speed (rpm)	3000	3000	
Exhaust temperature (K)	816	816	
Efficiency (%)	33.8	34.2	Out
Coolant mass flow (kg/s)	-	67	
Max cycle temperature (K)	-	1455	
Firing temperature (K)		1306	

Figure 1 Scheme of the simulated MS9001E, main input parameters and obtained results

The model realized in ESMS is schematically shown in Figure 1. Part of the flow from the axial compressor enters the combustion chamber, where CH_4 is injected and the combustion takes place. The remaining part of the compressor air feeds the turbine cooling system. In order to have a direct control on the first stage cooling system, in the built model the 3-stage gas turbine is divided in a first 1-stage module and a second 2-stage one. To carry out the design analysis few data needs to be assumed for the cooling system performances. In particular the following values are imposed in the simulation:

- Maximum blade temperature before the cooling system activates: 970 K
- Blade/gas path area ratio : 4
- Stanton number : 0.005
- ε_h, for both stator and rotor blades: 0.30
- Equivalent film cooling effectiveness χ_f: 0.35

Input data shown in grey in the table in Figure 1 are assigned as boundary conditions in the model. The obtained results, shown in the same table in light blue, are compared with the available data in literature. A firing temperature of 1306 K, downstream the first cooled nozzle, is predicted. Considering that the latest "E" technology uprate packages of GE lead the firing temperature up to 1397 K from the base design of 1277 K, the obtained configuration is retained a reasonable one to assess the proposed upgrades [17], [18].

The performance variations that can be obtained with the first stator cooling system and blade material upgrade will be evaluated relatively to the obtained design.

2.4. Off-design simulation of upgraded internal cooling system

The first analysis carried out investigates the effect of an improvement in the internal cooling system of the first nozzle on the global performances, a frequent after-sale engine upgrade (see [17], [18]).

Advanced cooling system design having innovative film cooling techniques and coolant flow paths in the interior blade lead to increased efficiency that can be simulated adjusting the ε_h .

Reasonable values for ε_h lie between 0.15 and 0.6. Considering current cooling systems and that DMLS can make it possible to build cooling system geometries unfeasible with classical manufacturing techniques, the higher part of the range is chosen varying ε_h within 0.3-0.65.

Off-design simulations are carried out considering the same maximum cycle temperature and the same blade material of the original configuration (admissible temperature 970 K). The amount of cooling air at the first stator is progressively reduced. As a consequence, the blade temperature increases.

In a post processing routine the internal cooling efficiency ε_h is enhanced to bring the blade temperature back to 970 K. A more efficient cooling of the first stator allows the achievement of the same blade temperature with less coolant. More air from the compressor enters the combustor and constitutes the main gas flow. Therefore, the gases leaving the first stator have higher temperature levels and the blades rows that follow undergo a metal temperature increase leading their temperature above the admissible one.

The analysis wants to show the entity of such a heating effect and to point out which blade row along the engine axial direction is the most stressed one.

2.5. Off-design simulation of innovative blade materials

Substituting the turbine blades with new ones with more thermal-resistant materials allows reaching higher blade temperatures than the design one of 970 K. As a consequence, it is possible to reduce the cooling air mass flow to cool down the blades with a direct impact on the gas turbine performance.

Generally the engine control system logic is based on temperature control. The measured exhaust temperature is correlated to the maximum cycle temperature. The control system restricts the ranges of exhaust temperature not to injure the gas turbine. In this case, the same control logic (same T_{max}) is maintained and therefore off-design simulations have been conducted at the design maximum temperature level. Off-design design analyses are performed reducing of a constant factor the total coolant mass flow. A maximum admissible blade temperature of 1150 K has been chosen. Since it is assumed that all the blades are of the same material, the highest blade temperature reached within the 6 cascades denotes the thermal limit for the given material and no further coolant reduction is possible.

The operation is then repeated at various maximum cycle temperature.

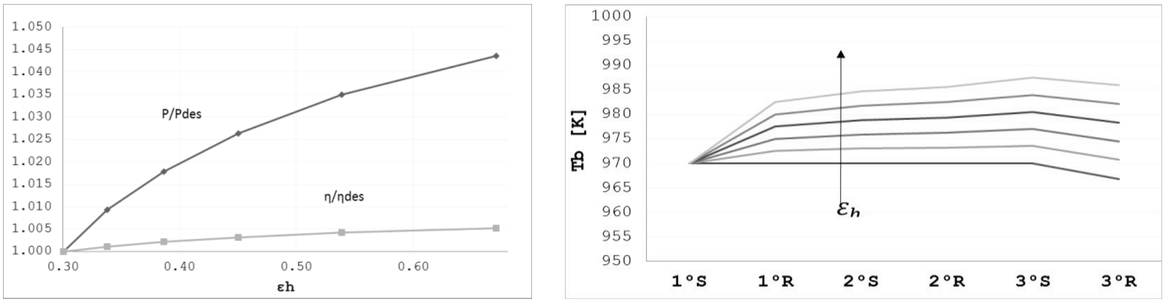
3. Results

3.1. Cooling system upgrade

Generally, the actions to upgrade the cooling system involve mainly the first stator and rotor. Therefore, it is of interest the study of the effect on the engine performance of an upgraded of the first stator only. Simulations are carried out as described in Section 2.4.

Enhancing the cooling system efficiency, ε_h, leads to a reduction of the coolant-hot gases mass flows ratio, as seen in Eq. 7, for fixed admissible blade temperature. The same cooling effect is reached with less coolant mass flow and the saved mass flow can participate to the reactive process in the combustion chamber. As a consequence the produced power and the global efficiency increase as shown in Figure 2 (a). The system efficiency variation is limited as at the increased air flow rate corresponds an increase of the fuel mass flow rate of the same entity.

While the new cooling system guarantees the first stator to be at the same design temperature, it leads to blade heating in the following rows, accordingly to Eq.8. If the blade temperature reaches the maximum admissible for the blade material, these are going to be damaged. Otherwise, if after service operations the blades are seen to work at under-firing conditions, they can sustain the increased temperature levels.



(a) (b)

Figure 2 Power and efficiency against ε_h (a) and temperature inside the turbine (b) obtained in off-design simulation.

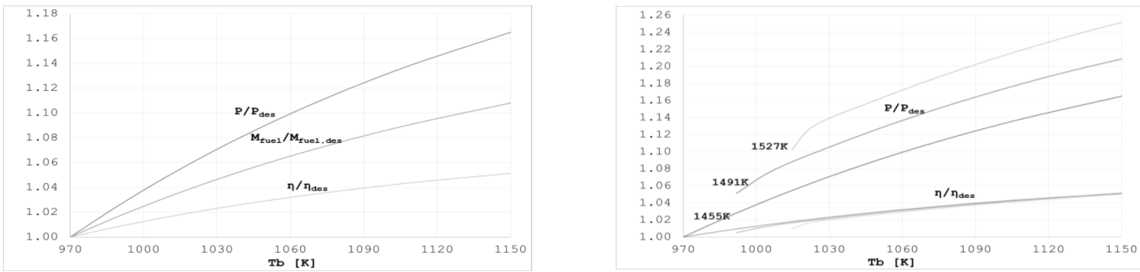
An evaluation of the temperature trend within the turbine can be of interest to understand the temperature variation that each blade needs to face and to understand which row becomes the most critical from a thermal point of view.

In Figure 2 (b) the temperature along the turbine is shown for the tested ε_h. Following the increased cooling efficiency of the first nozzle, the temperature of the gas leaving such a row increases and consequently all the turbine undergoes a global heating. As it is possible to see, the most critical row becomes the third nozzle. The presented effect can limit the redesign of only a single blade row which needs to be combined with the redesign of other blade rows in terms of cooling system or material. From another point of view, if the blade temperature can be raised without risks, thanks to available margins on blade admissible temperature or thanks to upgraded blade materials, the re-design of the first stator only leads to satisfying performance upgrades.

3.2. Blade material upgrade

Simulations have been carried out as reported in Section 2.5. The cooling system has been assumed not to vary with respect to the design one. The use of higher thermal resistant blade materials allows having more fresh air into the combustor and a less amount of cooling air. A higher T_b in Eq. 5 leads to a decrease of the cooling effectiveness φ and, consequently to a reduction of m_c/m_g in Eq. 7.

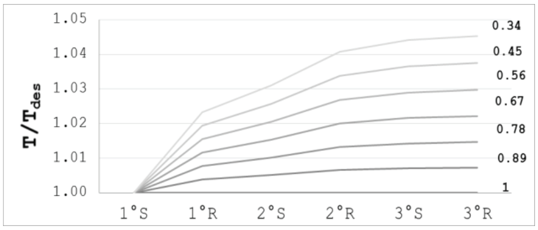
In Figure 3 (a) it can be seen that higher energy content expanding gases produce a net power output that constantly increases from the design value, as well as the overall efficiency.



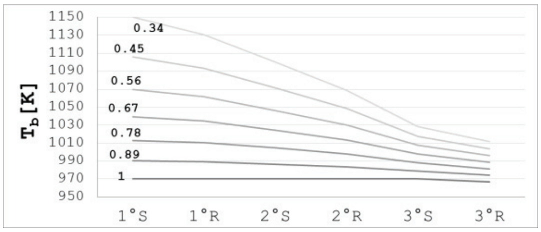
(a) (b)

Figure 3 Normalized power and efficiency curves for the design (a) and different maximum cycle temperatures (b) against blade temperature.

Figure 5 (a) shows the hot gas variation along the turbine. Starting from the same T_{max} at the first stator, the last stages undergo the higher temperature variations. On the other hand, in the Figure 5 (b) it is seen that the most stressed row is the first one while only a contained blade temperature increase is observed for the last rows.



(a)

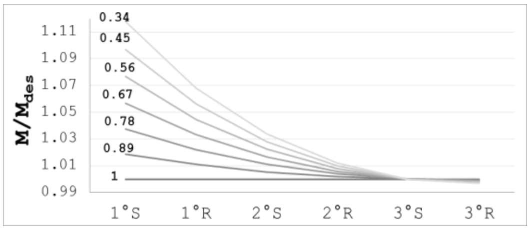


(b)

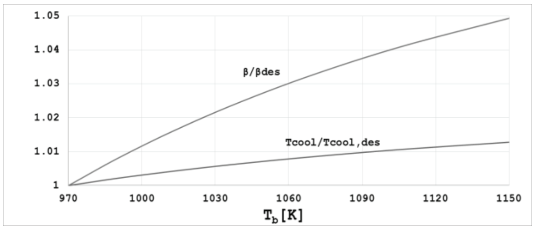
Figure 4 Normalized hot gases temperature profile (a) along the cascades and blade temperature (b)

The explanation is that the amount of hot gases flowing through the cascades is higher for the first stages than the last ones, as seen in Figure 5 (a). The higher the coolant reduction the higher the turbine entry hot gases mass flow.

In Figure 5 (a) it is also observed that the gases mass flow tends to values near the design one moving to the turbine outlet. Therefore, the last stage blade temperature increase seen in Figure 4 (b) is due only to the different gas temperature. Another factor that influences the first cascades blade temperature rise is explained considering the warming of the coolant air, as shown in Figure 4 (b). Reducing the coolant mass flow rate, the turbine inlet flow increases and consequently increased pressure rations are required to ensure the passage of hot gases (Figure 4 (b)). A new operating point is established between compressor and turbine at a higher compression ratio. Consequently, the cooling air from the compressor reaches a higher temperature.



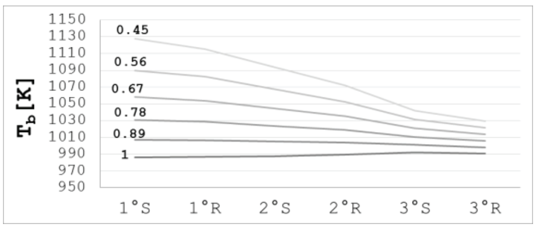
(a)



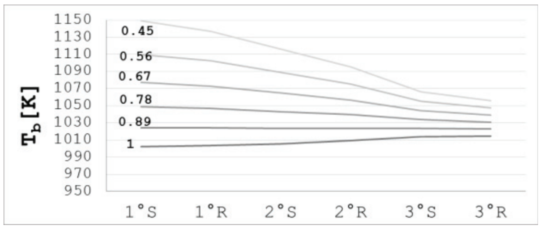
(b)

Figure 5 Normalized mass flow rate along the cascades (a); normalized pressure ratio and coolant temperature (b) against blade temperature.

With a more resistant material, a further option could be to raise the maximum cycle temperature without coolant reduction or a combination of the two actions. In this case, the proposed action would need the control system to adapt to the new maximum cycle temperature and therefore represents a more complicated operation. Without giving a detailed analysis, in Figure 3 (b) are shown the results for two increased maximum cycle temperatures, where it is possible to denote the great advantages in term of delivered output power.



(a)



(b)

Figure 6 Blade temperature profile along the cascades for simulations at maximum cycle temperature of 1491K (a) and 1527 K (b)

Looking at these curves a change in the slope is observed at lower blade temperatures. This is due to the fact that at low coolant reduction the most critical stage is the last one while reducing further the coolant the critical stage becomes the first stator as seen in Figure 6 (a) and (b) for two levels of simulated maximum temperature.

4. Conclusions

The effects on performance of upgrading the internal cooling system of the first stator and blade material on the GE MS9001E gas turbine are investigated with an in-house simulation tool. The improvement of the first cascade internal cooling system leads beneficial effects on the cycle performance since a lower amount of coolant is needed to cool down the stator. As a drawback, the gas path temperature downstream the first stator increases so that service operations need to evaluate alternative solutions: maintain the same blades exploiting possible existing margin on blade admissible temperature, adopt advanced materials and/or re-design the cooling system for all the blades.

A more resistant thermal blade material upgrade introduce a beneficial effect on the overall performance as a consequence of the coolant reduction can be realized. A further design option could be the raise of the maximum cycle temperature without coolant reduction as well as a combination of the two actions.

The analysis carried out constitutes the first important step to define practical margins and limitations for cooling system and blade material upgrades. In future step a complete map will be drawn, considering also possible combinations of the actions, to point out the optimal design condition.

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