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Modeling approach to calculate redistributions of HPT-shroud cooling channels including some turbine blade tip effects

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Executive summary

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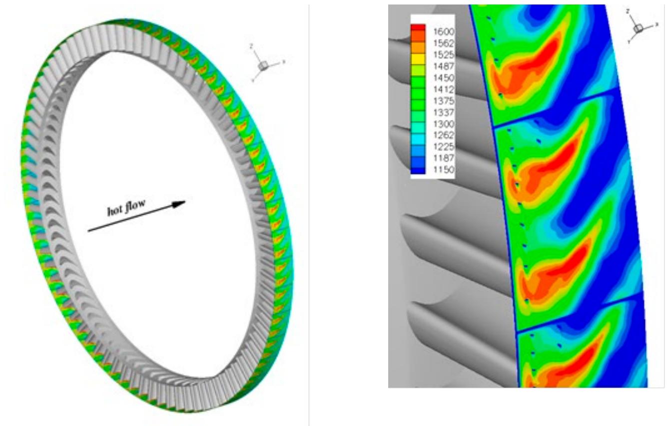

Instantaneous pattern of the gas near surface temperature (K)

Problem area
Chromalloy Holland BV maintains and manufactures solid shrouds, which are applied in the high pressure turbine of various civil turbofans. To better understand the shroud failure mechanisms, both an investigation of maintenance and in-service experiences as a numerical study have been carried out. This paper summarizes the results of the numerical study on shroud thermal loading.

Description of work
A numerical case study on a HPT-shroud of a medium-sized commercial engine has been carried out to investigate the heat loading and the possible redistribution of shroud cooling channels facing the turbine blade tip. A combination of modeling vehicles was used to quantify the aerodynamics, the thermodynamics and resulting heat loads on the shroud. This includes a 1-D gas turbine performance simulation model, engineering models for cooling flow distributions and heat loads, CFD modeling of the HPT flow including some tip flow effects and the finite element modeling to calculate the temperature and stress distribution in the solid shroud. Regions with high temperatures and/or maximum thermal stresses and the potential for reduction by relocating the cooling channels at equal amounts of cooling flow were identified.

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Thermal load

This report is based on a presentation held at the ASME Turbine Blade Tip Symposium (ASME-TBST2013), Hamburg, Germany, September 30th - October 3rd, 2013.

Results and conclusions
Though the physics involved in the processes are much more complicated than modeled, the parametric studies gave valuable insight and quantitative results in terms of differences in shroud temperatures and thermal stresses.

The main shroud damage mechanism seems to be blade tip rub. Furthermore, damage pattern observed on ex-service shrouds confirms the numerical findings.

Applicability
This study for Chromalloy Hollland BV gave design guidelines for further improvements in cooling measures, i.e. the relocation of cooling channels, to lower the shroud thermal load and stresses.

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Summary

A numerical case study on a HPT-shroud of a medium-sized commercial engine has been carried out to investigate the heat loading and the possible redistribution (number of channels, position and exit angle) of shroud cooling channels facing the turbine blade tip. A combination of modeling vehicles was used to quantify the aerodynamics, the thermodynamics and resulting heat loads on the shroud. This includes a 1-D gas turbine performance simulation model, engineering models for cooling flow distributions and heat loads, CFD modeling of the HPT flow including some tip flow effects and the finite element modeling to calculate the temperature and stress distribution in the solid shroud. Regions with high temperatures and/or maximum thermal stresses and the potential for reduction by relocating the cooling channels at equal amounts of cooling flow were identified. Although the physics involved in the processes is much more complicated than modeled, the parametric studies gave valuable insight and quantitative results in terms of differences in shroud temperatures and thermal stresses. A complementary experimental study on shroud maintenance and service experiences (not published yet) has delivered data for model input support and comparison with the numerical results.

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Contents

Nomenclature

c	=	tip clearance	[m]
c_p	=	Heat capacity	[J/kg/K]
E	=	Elastic modulus	[MPa]
E_w	=	windage heating	[W/m^3]
G	=	shear modulus	[MPa]
h	=	heat transfer coefficient	[W/m^2/K]
k	=	thermal conductivity coefficient	[W/m/K]
n	=	integer in velocity equation	[-]
N	=	shaft rotational speed	[rpm]
PR	=	pressure ratio HPT	[-]
PT_{in}	=	HPT total inlet pressure	[bar]
PT_{out}	=	HPT total outlet pressure	[bar]
r	=	reduction factor thermal load	[-]
T	=	total or static temperature	[K]
TT_{in}	=	HPT total inlet temperature	[K]
TT_{out}	=	HPT total outlet temperature	[K]
U	=	air velocity	[m/s]
W	=	mass flow through HPT	[kg/s]
W_t	=	mass flow in blade tip region	[kg/s]
x	=	time fraction	[-]
y	=	boundary layer co-ordinate	[m]
α	=	thermal expansion coefficient	[1/K]
δ	=	boundary layer thickness	[m]
μ	=	dynamic viscosity	[kg/s/m]
v	=	Poisson's ratio	[-]

1 Introduction

The present investigation relates to the High Pressure Turbine (HPT) Blade Outer Air Seal (BOAS) assemblies, also known as turbine shroud assemblies and called in this paper 'solid shrouds'. An example of a solid shroud is given in Fig. 1.

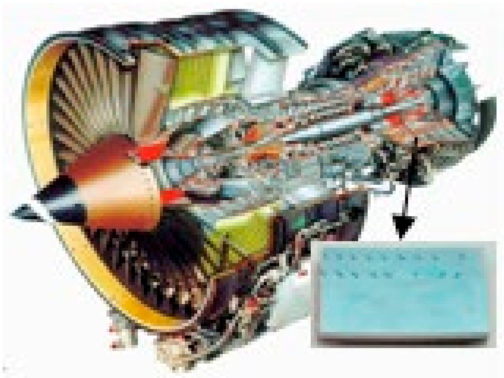


Figure 1: Solid shroud in the HPT

During engine operation, the turbine shroud components are exposed to the high temperature gas flow which may result in damage or ultimately in failure caused by oxidation/corrosion or thermal-mechanical fatigue. The difference in thermal expansion between the rotor and the turbine shroud assembly components and hard landings may also result in contact between the turbine shroud assembly and the blade tips of the rotor resulting in shroud wear and blade tip rub. This will give a subsequent increase in turbine blade tip clearance that significantly increases fuel consumption at equal thrust settings. A direct consequence is the rise of the turbine inlet temperature, which further distresses the hardware. Consequently, airlines are forced to replace these shrouds at regular intervals by repaired or new parts.

A research project has been carried out to better understand the deterioration of HPT solid shrouds of a medium-sized commercial engine and to quantitatively investigate whether relocation of shroud cooling channels could reduce thermal loads. The aims of the project were:

1. Identification of typical load conditions on solid shrouds, understanding existing designs and their damage modes.
2. Setting up a set of functional and design requirements for solid shrouds.
3. Development of the modelling of the (internal) cooling schemes for solid shrouds.
4. Development of the modelling of solid shrouds emphasizing operational conditions, to calculate temperature and stress distributions.\

5. To validate the calculated temperatures and stresses by comparing the FEM-results with the damages and their locations on actual hardware.

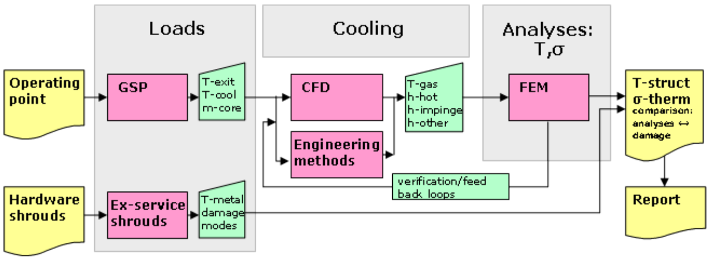
Thermal loading and cooling of the shroud consists of various components. Main loading components are the forced convection by the hot primary flow and the windage heating by the rotor blades. Main cooling components are the impingement cooling on the back side of the shroud, the forced convection cooling in the cooling channels and film cooling at the grazing flow surface. This paper will focus on the additional shroud loading effects caused by the moving turbine blades. In the next section entitled “Steady Heat Load on HPT Shrouds” an outline is given of the modelling approach in the project.

2 Modeling Techniques

A combination of modeling techniques with different levels of physical fidelity has been used to determine the thermal load on the shrouds (Fig. 2):

1. The NLR gas turbine performance simulation program GSP [1] has been used to quantify the station averaged thermodynamic data of the engine. For the HPT turbine the model delivered the upstream and downstream total temperatures and pressures. A parabolic HPT inlet temperature profile has been used with a radial temperature distribution factor of 0.1. The approximate cooling scheme was derived from guidelines given in [2].
2. Engineering models for temperature profile of primary flow, secondary (cooling) flows (amounts, flow division and pressure drops) and heat transfer (impingement cooling [3]) on the back side of the shrouds, forced convection cooling in the channels and film cooling [4].
3. NLR’s CFD method ENFLOW [5] for the simulation of the primary flow through the turbine blade, resolving the secondary flow region between the blade tip and the shrouds.

Abaqus as finite element code for the calculation of the temperature distribution and the thermal stresses in the shroud based on either the calculated results from the engineering methods or from the CFD code. The results of both methods differing in flow detail could be compared.



T-exit = exit temperature of the combustor
T-cool = compressor exit temperature (= temperature of the cooling air)
T-gas = gas temperature on the hot side of the shroud
h-hot = heat transfer coefficient on the hot side of the shroud
h-impingement = heat transfer coefficient on the impingement cooled pocket
h-other = heat transfer coefficient on other locations
m-core = mass flow through engine core
T-metal = structural temperature as derived from cross-sectioned ex-service shrouds
damage modes = damage mechanisms dominating part life; as derived from cross-sectioned ex-service shrouds
T-struct = structural temperature calculated by FEM
σ-therm = thermal stresses
GSP = Gas turbine Simulation Program, CFD = Computational Fluid Dynamics, FEM = Finite Element Method

Figure 2: Outline of modeling approach

In the first phase of the project only steady analysis was performed. Shroud gas surface temperature distributions for both methods (engineering and CFD) using only steady analysis are given in Fig. 3. The engine operating condition is maximum power at take-off, which is regarded as the most severe condition in terms of thermal loads to the shroud. The results are very similar except near the film cooling holes which in the engineering model are treated in a laterally averaged sense. The highest gas surface temperatures are found at the shroud leading edge (T=1230 K). The temperature decay in axial direction to a value of T=970 K is mainly caused by the flow work performed by the HPT rotor blades. The reduction of temperature at the right edge (right figure) is caused by the outflow of impingement cooling on the shroud side walls. These and similar (steady) results were used to investigate the effects of relocation of cooling channels on temperature and thermal stress distributions using the finite element code Abaqus.

The results of the steady state analysis raise the question what the additional unsteady effects of the HPT rotor would be, in particular the windage heating by the rotor blades and the possible cooling effects by the radial outflow of cooling flow near the rotor blade squealer tips. A simple engineering approach has been applied to estimate the possible effects of windage heating and cooling by the squealer tip. Furthermore CFD analysis has been applied to only quantify the effect of windage heating on the solid shroud by rigid HPT rotor blades without squealer tips and (blade) cooling flows. One objective is to investigate whether the results of the steady heat load analysis on the shrouds of the first phase of the project would still be valid or should be adapted due to the unsteady effects of the HPT rotor.

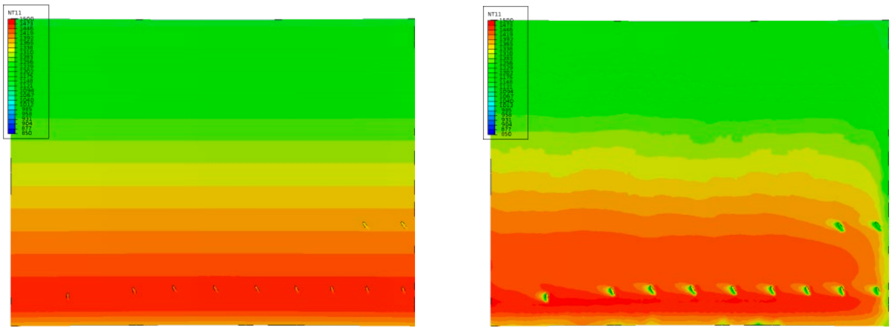


Figure 3: Gas steady state temperature distribution at the proximity of the shroud surface by engineering model (left) and by CFD (right)

2.1 Blade tip effects

In the sequel the additional unsteady effects of the HPT rotor, the main subject of study of this paper, are discussed. The added value of an engineering model is limited due to the highly complex flow and heat transfer phenomena near the blade tips. Focus therefore will be on the CFD modeling. The engineering model is only used to estimate the order of magnitude (important/can be neglected) of the effects of windage heating due to the shear flow between the blade tips and the shroud surface and cooling flow from the squealer blade tips, which were not taken into account by the CFD model.

2.2 Engineering estimate for additional shroud heat loads caused by blade tip effects

Two aspects related to the moving blades were examined with an engineering approach: the windage heating by friction of the flow with the shrouds and the possible effect of cooling by the squealer tips. The windage heating can be much better predicted by unsteady CFD calculations due to the complexity of flow and heat transfer phenomena near the blade tips. However, squealer geometry and radial tip cooling outflow were not modeled in the CFD calculations.

2.2.1 Engineering estimate of windage heating

The amount of windage heating is derived from a linear shear flow between two parallel plates (one steady and one moving with velocity U_θ) at small distance c from each other. The shear stress τ

$$\tau = \mu \frac{U_\theta}{c} \qquad \{1\}$$

multiplied by the velocity U_θ and divided by the flow volume gives the energy production per time and unit volume:

$$E = \mu \left(\frac{U_\theta}{c}\right)^2 \qquad \{2\}$$

The total energy production $E_{windage}$ therefore is the energy production per unit volume and time multiplied by the volume V_o occupied by the air between the blade tips and shrouds

$$E_{windage} = V_0 \mu (\frac{U_\theta}{c})^2 \qquad \{3\}$$

The over the circumference averaged flow temperature rise however depends strongly on the axial mass flow through the tip boundary layer region:

$$\Delta T = \frac{E_{windage}}{c_p W_{tip}} \qquad \{4\}$$

with c_p=specific heat coefficient and W_{tip}=mass flow in the tip region effected by windage heating. This mass flow is not well defined (or even arbitrary), but here defined as the mass flow in the initial boundary layer just upstream of the blades in a layer with the height of the rotor tip clearance. The boundary layer thickness using the equation of turbulent flow development along a flat plate is estimated to be about 1.5 mm. By describing the velocity distribution by a power law with n=7

$$\frac{U_y}{U_0} = (\frac{y}{\delta})^{\frac{1}{n}} \qquad \{5\}$$

the boundary layer mass flow up to the height of the rotor clearance (with the assumption of absorbing the windage energy) can be calculated. The resulting estimated increase in near shroud surface temperature is given in table 1. For tip clearances larger than 0.5 mm the effect of windage heating on the shroud surface is found to be small, which has to be confirmed by CFD. In practice the amount of shear is even less due to the application of squealer tips, increasing the averaged distance between the moving blade tip and shroud surfaces.

Table 1 Dissipated energy and gas temperature rise near HPT blade tips at various gaps

Blade tip clearance c [mm]	Estimated annular mass flow in tip region [kg/s]	Dissipated energy [W]	Mean temperature rise [K]	Local temperature rise [K]
0.1	0.07	2400	30	180
0.25	0.22	960	4.4	26
0.5	0.48	480	1	6
1.0	1.08	240	0.22	1.3

2.2.2 Engineering estimate of impingement cooling from the tips squealer tip (not modeled in CFD calculations

The radial outflow of the cooling air will be at (or very near) sonic conditions. The method of Kercher and Tabakoff [2] is used to estimate the heat transfer coefficient for steady impingement cooling. With a cooling flow Reynolds number of 46,000, 10 radial cooling holes with a pitch of 5 mm and distance to the shroud of 4 mm, the steady impingement heat transfer coefficient would have a value of 14,000 W/(m^{2}K). The reduction factor in thermal loading to the shroud surface covered by the blades (5/6th part of the time heat transfer to the shroud with heat transfer coefficient of 7000 W/(m^{2}K) and shroud surface temperature of 1250 K) can be calculated with the following equation:

$$r = \frac{(1-x)h_{flow}\Delta T_{flow} - xh_{imp}\Delta T_{imp}}{h_{flow}\Delta T_{flow}} \qquad \{6\}$$

with x the time fraction for which the shroud is covered by the blades. ΔT's in the equation have positive values and the minus sign before $xh_{imp}\Delta T_{imp}$ denotes cooling. The reduction factor is 0.4, which means that the thermal loading to the shroud covered by the blades has been reduced by 60%. Even with a lower cooling flow Reynolds (30,000), the reduction of thermal loading to the shroud would be substantial (h_{imp} =8,000 W/(m^{2}K), r = 0.58 leading to 42% reduction). If the effect of impingement cooling would be small due to high cross tip flow velocities (but only relatively cool "spots") would travel near the blade tips, the thermal loading would be reduced by 16 % (1/6th part of the travelling time).

The results from the engineering model indicate that (1) the effect of windage heating is expected to be small and (2) that the radial outflow of relatively cool air from the squealer tips (causing relatively cool air pockets covering about 15% of the shroud surface and moving along that) may significantly reduce the thermal loads on the shrouds, as film cooling does. The first effect, windage heating, is subject to CFD calculations discussed hereafter.

2.3 CFD simulation through the HPT blade-shroud configuration

The objectives of the CFD simulation are:

1. To provide the hot gas temperature distribution on the surface of the shroud, that will be used by the FEM method to determine the heat load in the solid shroud.
2. To mutually verify the temperature estimate based on engine-cycle model using GSP.
3. To determine the additional unsteady effect of rotating HPT blades.

The flow condition through the HPT rotor is estimated based on the engine cycle model using GSP. This also gives a radial parabolic profile of the combustor exit total temperature distribution with a maximum temperature difference of 165 K.

The available CAD model describes a one-row geometry consisting of the shroud and the rotor blade, which includes the definition of the cooling holes on the shroud. Unfortunately, no nozzle vane geometry was available for this study. An estimate must be made concerning the exit angle from the nozzle vane trailing edge entering the rotor domain.

The boundary conditions applied on the inlet are the momentum, a total temperature profile based on the combustor exit total temperature distribution, and an estimate of the exit flow angle from the nozzle vane. The same type of boundary condition is applied on each cooling hole and on the slit in front and sides of the shroud. On the outlet, a static pressure is specified. The inlet and outlet boundary conditions are tuned to match as close as possible the turbine stage properties estimated by GSP:

- Rotational speed = 14250 rpm
- Inlet total pressure = 36.6 bar
- Inlet total temperature = 1650 K
- Mass flow = 57.5 kg/s
- Outlet total pressure = 7 bar
- Outlet total temperature = 1177 K
- Pressure Ratio = 0.19 (5.26)

The adiabatic wall boundary condition is applied throughout all the solid surfaces. In terms of heat loads, this implies a conservative estimate because heat is kept inside the flow.
NLR's CFD method ENFLOW [5] was employed for the flow simulation, based on a finite volume scheme on a multi-block structured grid. A time-accurate unsteady flow simulation is performed by means of a dual-time stepping scheme. A k-ω turbulence model in the TNT formulation is used [6].

A typical arrangement for turbo machinery CFD modeling is one, in which advantage is taken of the geometric periodicity. The number of blades has been slightly increased to get a blade/shroud ratio of 2 (84 blades and 42 shrouds), giving a moderate domain scaling factor of 5% to allow an affordable number of grid cells of 1.5 million. The flow domain is depicted in Fig. 4.

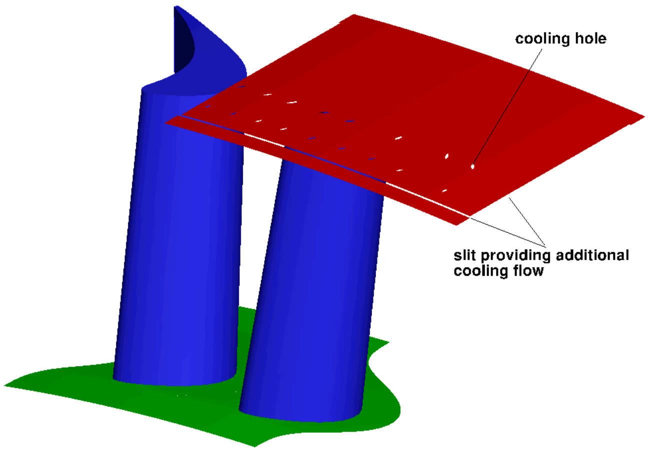


Figure 4: Flow domain consisting of rotor blade and shroud segment

A usual best practice for CFD simulation was followed regarding grid convergence of a second-order finite volume scheme and turbulent boundary layer resolution giving a $y^{+}\approx 1$.

The flow domain is divided into a rotating part containing the rotor blade and a stationary part containing the shroud. An unsteady time-accurate flow simulation is performed with a sliding grid technique on the interface between these two domains.

For the purpose of the present study, that is to produce the surface temperature distribution, two criteria are used for an acceptable flow solution:

1. it should produce results close to the estimation given by GSP above,
2. no adverse flow properties present in the flow field,
3. no adverse variation in the force and moment coefficients.

2.3.1 Computational results

A comparison between GSP estimate and CFD simulation is presented in Table 2, showing a good agreement. The streamline patterns in Figure 5 show no evidence of adverse flow properties. A minor flow separation observed in the picture at 50%-span does not suggest an adverse condition. This indicates that a correct estimate of the exit flow angle from the nozzle vane has been applied on the inlet to the rotor domain.

Table 2 Comparison of GSP and CFD HPT performance parameters

	GSP	Time averaged	Unit	Percent difference [%]
N	14,250	14,250	rpm	0.0
PTin	36.6	34.6	bar	-5.5
TTin	1650	1636	K	-0.8
W	57.5	57.5	kg/s	0.0
PTout	7.0	6.92	bar	-1.1
TTout	1177	1140	K	-3.1
PR	5.3	5.0	-	-4.9

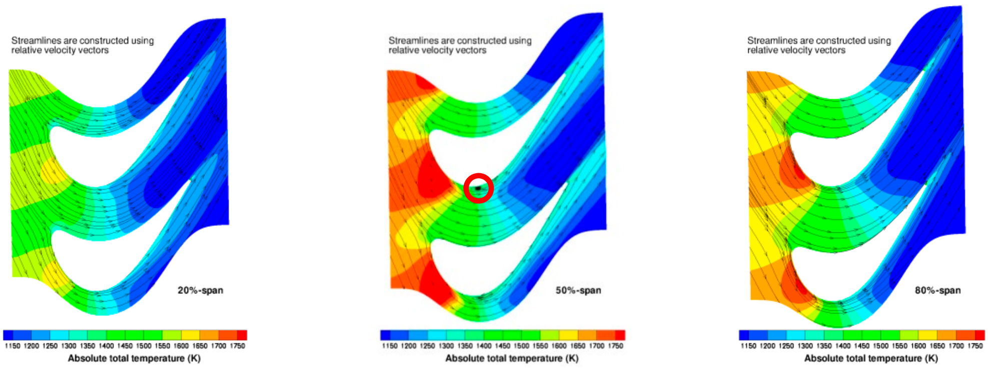


Figure 5: Blade temperature distribution at various span wise locations (20, 50 and 80%)

Figure 6 shows a sample of the time signals in terms of temperature averaged over one segment of the shroud, plotted against the rotation angle of a blade. The rotation angle equal to 100 degrees means that the shroud segment experiences about 24 blade passings.

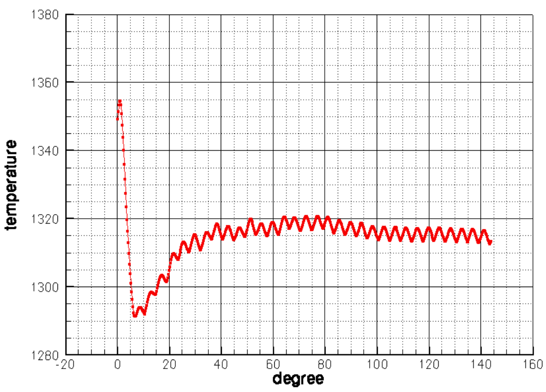


Figure 6: Time history of the surface-averaged temperature (K), including the initial transient

The figure shows that after about 24 blade passages, the transient has been sufficiently decayed. This is also illustrated in Figure 7 by the instantaneous pattern of the temperature distribution, which is quite periodic from one shroud segment to another. After about 24 blade passages, samples can be collected for the time-averaging of the surface-averaged temperature.

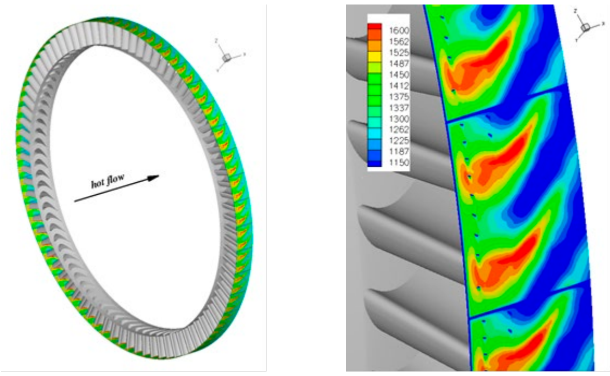


Figure 7: Instantaneous pattern of the gas near surface temperature (K)

The pattern of the time-averaged surface gas temperature is shown in Figure 8, having an average temperature of 1311 K over the shroud segment. The cooling holes on the top surface are taken as provided in the CAD model, but the cooling holes on the side surface are modeled by a slit with a uniformly distributed mass flow through the side holes. The results show that the highest surface gas temperature is just in front of the first row of cooling holes. The location of the maximum peak temperature on the surface is marked by black arrow in right-bottom side of figure 8, which has a temperature of 1500 K. The shown surface temperature distribution is used for verifying the engineering estimate of the surface temperature gradient, and passed to the FEM method for the analysis of the heat loads.

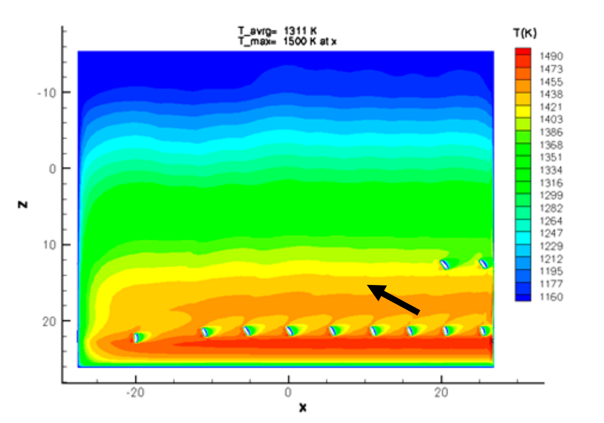


Figure 8: Gas near surface temperature distribution for a tip clearance of 0.85 mm

2.3.2 Effect of tip clearance variation

To investigate the effect of variation of the tip clearance, e.g. due to operation, a hypothetical blade is generated by cutting the blade tip to get a clearance of 1.60 mm from the nominal value of 0.85 mm. A comparison of the time history of the surface-averaged temperature is presented in Figure 9, showing that the clearance of 1.60 mm elevates the average temperature by 3 K with respect to that of 0.85 mm.

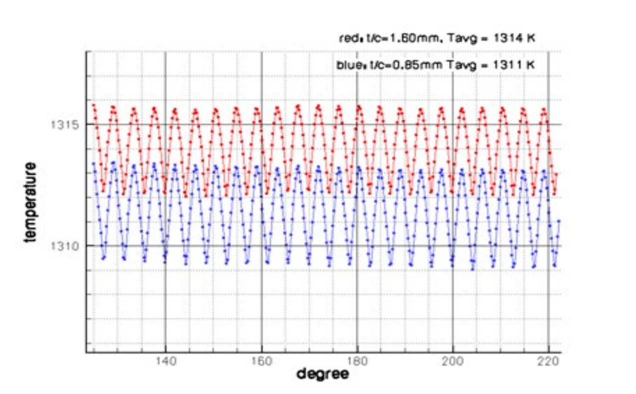


Figure 9: Effect of the tip clearance on the gas near surface temperature

The effect on the surface temperature distribution is indicated in Figure 10 in comparison with Figure 8 showing a smoother distribution. The location of the maximum temperature on the surface stays identical (at the right-bottom side in Figure 10) and is negligibly different for the given variation of the clearance.

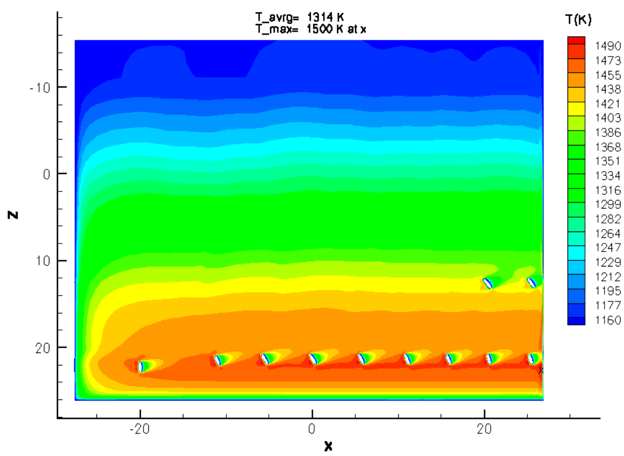


Figure 10: Gas near surface temperature distribution for a tip clearance of 1.60 mm

2.4 FEM calculations on shroud heat load

Two approaches were developed in the first phase of the project to determine the steady thermal loading (the gas temperatures and heat transfer coefficients) on the shroud. The first approach was based on engineering methods and the second one was based on computational fluid dynamics (CFD). The finite element method (Abaqus) was used to determine metal temperature and stress levels using these two boundary conditions. Both approaches were compared to determine whether the use of an engineering approximation can be justified when evaluating new designs to allow a short turn-around time in a preliminary design phase for various geometries.

The FEM calculations included the following features:

1. Heat transfer coefficients and gas temperatures were applied on the hot side, the impingement cooled back side, inside the convectively cooled cooling holes and on the sides.
2. Temperature dependent anisotropic (cubic) material properties (E, G, ν, α and k) for a second generation single crystal super alloy were applied.
3. A thermal model was used to determine the metal temperature. These temperatures were subsequently used in the mechanical model to determine the stress distribution.
4. The shroud is radially supported at the support rail location. Displacement in axial and tangential direction is suppressed at the pin support.
5. Pressure loading with an axial gradient is applied for the difference in pressure between the hot side and the impingement cooled pocket on the rear.
6. A grinding operation if often applied to match the shrouds to the rotor. Additional calculation was made with a thinner shroud caused by grinding.

Figure 11 shows the calculated metal temperature on the hot surface for two different cooling hole patterns. The impingement holes cool the front of the shroud and therefore the strong temperature gradient from the CFD results (Figure 3) is not found in the metal temperature.

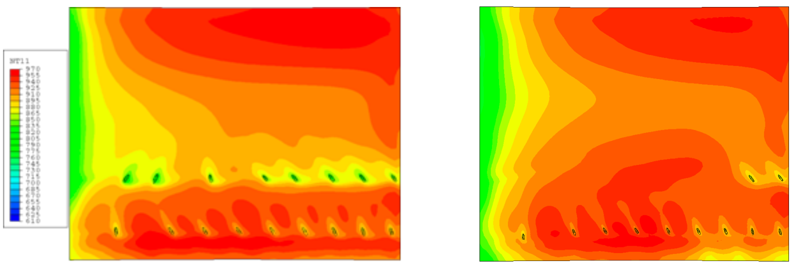


Figure 11: Metal temperature using CFD loads for two different cooling hole patterns

The design on the left has a cool area in the center caused by the many cooling holes in the second row combined with the somewhat lower gas temperatures. Cooling holes could therefore be removed in this area, leading to a more constant metal temperature on the surface.

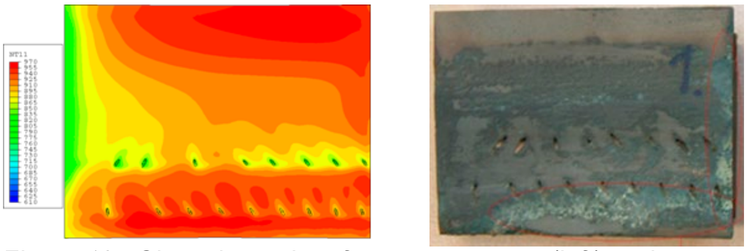


Figure 12: Shroud metal surface temperature (left) and an ex-service shroud (qualitative comparison)

Figure 12 shows the shroud temperature after grinding that corresponds to the un-grinded temperature shows in Figure 11 on the right. The overall temperatures are lower when the shroud is grinded. The reason is that there is less material between the hot front surface and the impingement cooled back side, creating more effective impingement cooling. The high temperatures at the front and right of the shroud are confirmed by common damage patterns in ex-service shrouds (Figure 12).

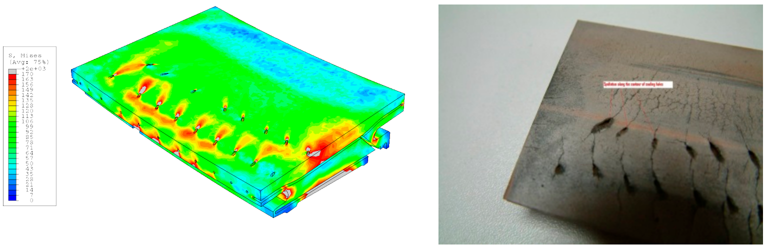


Figure 13: Von Mises stress and an ex-service shroud (qualitative comparison)

The stress levels in the shroud are shown in Figure 13. High stress levels are found where the cooling tubes exit the shroud. This is also a typical location for crack in ex-service shrouds.
A typical radial shroud deformation caused by the combination of thermal and pressure load is shown Figure 14. The tip clearance will be somewhat smaller in the center of the shroud compared to the sides. This is confirmed by rub often being most severe in the center of the shroud.

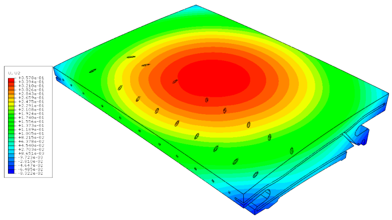


Figure 14: Calculated radial deformation (left) confirmed by common damage pattern (right) (qualitative comparison)

3 Conclusions

A numerical case study on a HPT-shroud of a medium-sized commercial engine has been carried out to investigate the heat loading and the possible redistribution of shroud cooling channels facing the turbine blade tip. The study has demonstrated the viability of a combination of modeling vehicles to quantify the aerodynamics, the thermodynamics and resulting heat loads on the shroud. It involves modeling at different levels of physical fidelity, encompassing a 1-D gas turbine performance simulation model, engineering models for cooling flow distributions and heat loads, the NLR CFD code ENFLOW for the three-dimensional modeling of the HPT flow including some tip flow effects, and a finite element code Abaqus to calculate the temperature and stress distribution in the three-dimensional solid shroud. Regions with high temperatures and/or maximum thermal stresses and the potential for reduction by the relocation of cooling channels at equal amounts of cooling flow were identified. The study has made use of a complimentary study on shroud maintenance and service experiments to obtain input for the modeling and to allow a comparison with numerical results. The numerical study has led to the following conclusions:

1. Both methods, i.e. the engineering approach and CFD, give comparable results with respect to the steady heat load on the shrouds, the first however in less detail near the film cooling holes.
2. The engineering approach for the unsteady effects (the passing of the blade tips) is less suited due to the complexity of the flow phenomena. These models indicate that radial cooling from the squealer tips have a larger effect on shroud thermal loading than the effect of windage heating by the rotating blades.
3. CFD calculations have been carried out for the steady and unsteady flow. The number of blades was adjusted to maximize the periodicity resulting in grid domain of 1.5 million points. The unsteady CFD calculations with adiabatic shroud wall conditions revealed the temperature rise (windage heating) due to the moving blades.

4. CFD results on solid blades (without squealer tip and cooling flows) show that increasing the blade tip clearance from 0.85 to 1.6 mm has only a small effect of increased gas shroud surface temperatures. No CFD calculations were done to quantify the effects of a squealer tip and blade cooling flows.
5. The FEM analysis revealed the maximum temperature and stress distributions. Maximal tensile stresses were found near the surface and inner cooling holes.
6. The main shroud damage mechanism is blade tip rub, which follows from a parallel study on shroud maintenance and service experiences (not published yet, see Fig. 14).
7. Damage pattern observed on ex-service shrouds confirms qualitatively the numerical findings.

Recommendations are (1) to further sophisticate the modeling by applying conjugate analysis (coupled CFD/FEM) on the thermal loading of the shroud service and (2) to include squealer geometry and radial outflow of cooling flow in the CFD calculations. The main challenge however is to find a balance between the applied methods and the (many) uncertainties in the complex flow and heat transfer phenomena in the tip region of the HPT blades. Furthermore a FEM rub model has to be developed since blade tip rub is the main damage mechanism.

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