THERMAL MODELLING OF HEAT TRANSFER IN A ROTATING DISC DUE TO A SUPERPOSED FLOW OF COOLING AIR

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Abstract

Temperature variations around the cooling air receiver holes in gas turbine rotor discs may give rise to large thermal stresses around the edges of the holes, and so cause fatigue that could affect component life. In the present work, a finite element thermal model is used to simulate a simplified cooling air delivery system that has previously been studied experimentally. In these experiments, heated air flowed over a plane rotating disc made from transparent polycarbonate and painted with thermochromic liquid crystal (TLC), and heat transfer coefficients were deduced using the surface temperature information recorded by the TLC and a one-dimensional transient analysis technique. These measured heat transfer coefficients have been used here to supply boundary conditions to the finite element model, which is found to reproduce qualitatively the temperature changes with time observed experimentally. The validity of the one-dimensional assumptions made in the analysis of the experiments is also demonstrated.

The model is then adapted to study conditions representative of those occurring in engines, by scaling the measured heat transfer coefficients to typical engine operating conditions using a correlation based on rotational Reynolds number. These results show significant variations in the temperature of the disc around the receiver holes. Finally, the same boundary conditions are mapped to the surface of a generic 3D geometry representative of a turbine disc in an engine. The computed temperature distributions agree with information derived from other sources, suggesting that the detailed results from the simplified experiments should be useful to designers of gas turbine engines. The regions of highest thermal stress occur around the receiver holes and contribute to an estimated fatigue life for the simulated turbine rotor of around 15,000 hours.

Nomenclature

a	inner radius of disc
b	outer radius of disc
h	heat transfer coefficient ($=q/(T_{aw}-T_w)$)
k	thermal conductivity of air
Nu	Nusselt number ($=hr/k$)
q	rotor surface heat flux
r	radius
Re	Rotational Reynolds number ($=\rho\Omega b^2/\mu$)
s	clearance between rotor and stator
T	temperature
x	nondimensional radius ($= r/b$)
μ	dynamic viscosity
ρ	density
Ω	angular speed of disc

Subscripts

aw	adiabatic wall value
b	at "blade-cooling" receiver hole radius
p	at pre-swirl nozzle radius
w	wall (rotating disc surface)

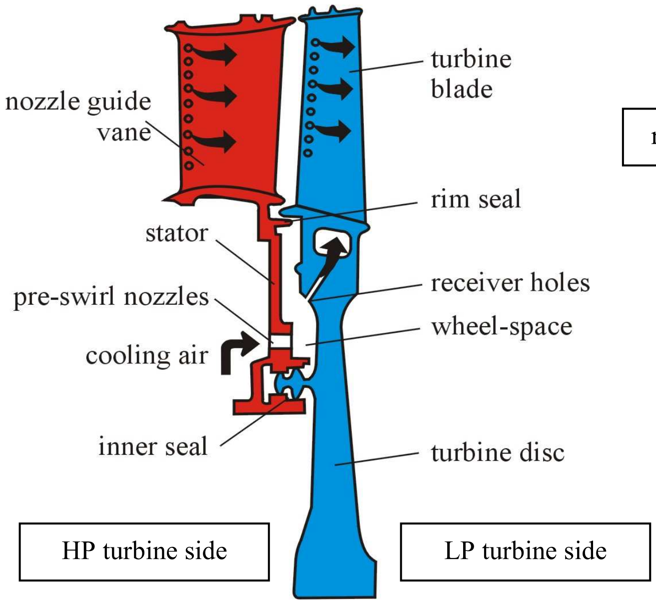

Figure 1 diagrammatic representation of a pre-swirl cooling system in a gas-turbine engine (from Lock et al, 2005)

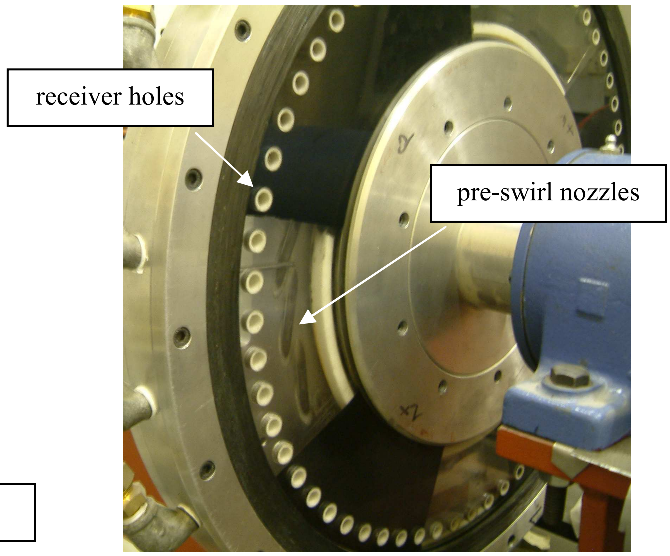

Figure 2 photograph of the rotating disc rig used by Lock et al

1. Introduction

Some of the air in the internal air system of a gas-turbine engine is used to supply air to cooling passages inside the blades of the high-pressure (HP) turbine stage downstream of the combustor. The blades are protected from the extreme temperatures in the gas path (that can exceed the melting point of the material from which the blades are made) by film cooling, convection cooling and impingement cooling, and the "blade cooling" air required enters the internal turbine blade passages through receiver holes located near the periphery of the turbine disc, see Figure 1. The air is often swirled in the direction of rotation of the turbine disc (by angled "pre-swirl" nozzles in an adjacent stationary casing as illustrated in Figure 1), in order to reduce the relative temperature of the air entering the rotating receiver holes. A review of fundamental aspects of pre-swirl cooling air systems is given by Owen and Rogers (1989).

The flow in pre-swirl systems has been studied both experimentally and computationally by, among others, Yan *et al* (2002), Geis *et al* (2003), Chew *et al* (2005) and Lewis *et al* (2006), and detailed heat transfer measurements have been made by Lock et al (2005a,b). Lewis *et al* showed that the nonuniform flow around the receiver holes gives rise to local variations in heat transfer; this will affect the temperature of the cooling air entering the receiver holes and may also give rise to undesirable thermal stresses in the disc.

Monico and Chew (1992) applied a finite element thermal model to the region around the cooling holes of a turbine disc and identified the importance of representing three-dimensional effects. Verdicchio *et al* (2001) and Okita and Yamawaki (2005) described results from conjugate (coupled fluid and solid) models of the flow and heat transfer in components of gas turbines. While studies such as these demonstrate the potential of general-purpose 3D computational procedures, computing times can be very long (due partly to the different thermal response characteristics of the fluid and solid regions), also limited information is often available for detailed verification of predictions, due to the difficulties involved in obtaining measurements in engines.

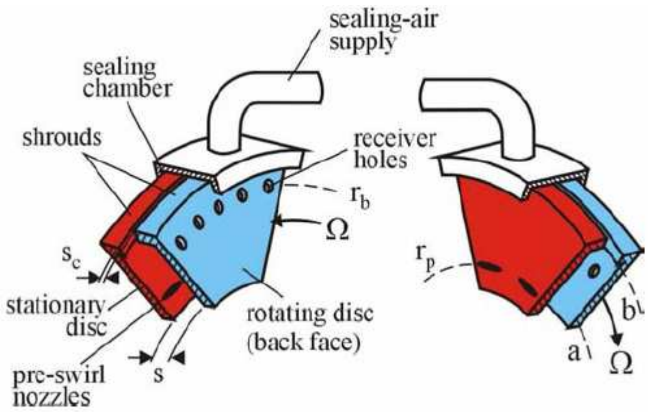

Figure 3 Schematic of the experimental rig (from Lock et al, 2005a)

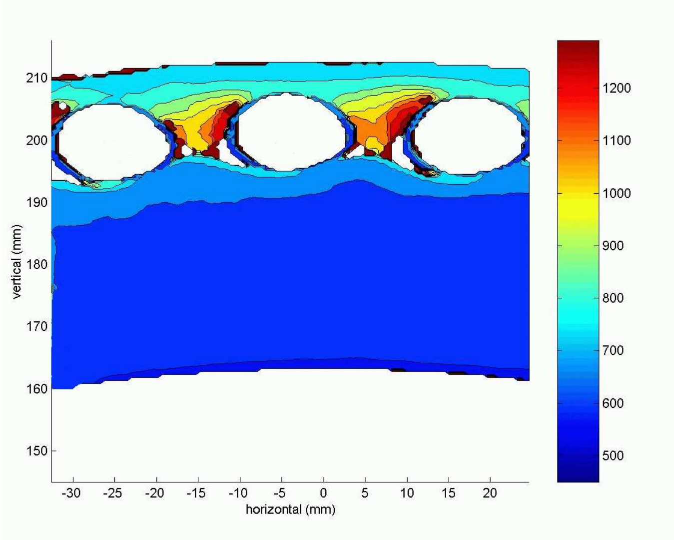

Figure 4 Measured radial variation of Nu (from Lock et al, 2005a)

Lock *et al* (2005a,b) used thermochromic liquid crystal and a simplified experimental model of a pre-swirl rotor-stator system, as shown in Figure 2, to make detailed measurements of heat transfer around receiver holes in a transparent polycarbonate rotating disc. The non-dimensional pre-swirl flow-rates used were representative of the practical situation; the experimental conditions however did not match the high temperatures and rotational speeds encountered in gas-turbine engines.

In the present work, a finite element model of the rotating disc studied by Lock *et al* is used to scale the results obtained in the experiments to typical engine operating conditions. In section 2, the methods and findings of the previous experimental study are summarised, and the computational model used for the simulations in the present work is described in section 3. Qualitative comparison between the results of simulations and the experiments is made in section 4, and in section 5 simulations corresponding to the engine operating environment are carried out using boundary conditions devised by suitable scaling of the heat transfer coefficients measured in the experiments. The additional effects of features of typical gas-turbine engine geometry are investigated in section 6.

2. Results of previous work

Figure 3 shows schematically the rotor-stator configuration studied experimentally by Lock *et al* (2005a,b). Pre-swirl air (which was passed through an upstream mesh heater in order to create a sudden increase in its temperature) entered the system through 24 pre-swirl nozzles in the stationary disc. The nozzles were angled at 20° to the tangential in the direction of rotation of the adjacent rotating disc, in which there were 60 axial receiver holes at a higher radius (r = 200 mm, given non-dimensionally by $x_b = 0.93$) than that of the inlet nozzles (for which $x_p = 0.74$). The heated pre-swirl air flowed radially outward from the nozzles to the holes, and the transient temperature response of the polycarbonate rotating disc was measured using thermochromic liquid crystal (TLC). Other details of the experiments, and of the calibration of the TLC, are given by Lock *et al* (2005a).

The time-varying colour change observed for the TLC over the disc surface provided instantaneous surface temperatures at known times, from which the heat transfer coefficient was found by solving Fourier's equation for a semi-infinite plate assuming one-dimensional conduction. Results were presented by Lock *et al* (2005a,b) in the form of two-dimensional contours of local Nusselt number Nu and heat transfer coefficient h over the disc surface. These revealed that the highest rates of heat transfer occurred around the receiver holes, and that high values also occurred at lower radii for cases involving high pre-swirl flow rates. This was attributed to inertial “impingement” of the superposed pre-swirl flow on the rotating disc, while at lower flow rates heat transfer was governed by the development of the viscous boundary layer on the rotating disc. An example of results obtained experimentally for conditions in the low flow-rate “viscous regime” is shown in Fig. 4. The variation in heat transfer rates around the receiver holes is of interest to engine designers, in view of the possible implications for fatigue and crack propagation.

3. Finite element computational model

Both transient and steady state simulations of the heat transfer through the rotating disc studied by Lock *et al* (2005a,b) have been carried out using the commercial finite element software ANSYS. The properties used for the model in section 4, for the polycarbonate material from which the disc in the experiments was made, are given in Table 1. The properties used for the models in sections 5 and 6, for the Nickel Chromium alloy Inconell frequently used for the manufacture of turbine discs in engines, are also shown in Table 1.

Table 1 Material Properties

Material Property	Polycarbonate	Inconell
Density (kg/m^3)	1200	8440
Poisson’s Ratio	-	0.289
Specific Heat Capacity (J/kg K)	1200	410
Young’s Modulus (GPa)	-	207
Thermal Expansion Coefficient (μm/m K)	-	12.8 (at 600 K)
Thermal Conductivity (W/m K)	0.22	9.8

The non-uniform distribution of heat transfer coefficient h, used in section 4 for the “impingement” surface of the rotating disc (i.e. the “HP turbine side”, see Figure 1) is illustrated in Figure 5a, and was obtained from the measured values given by Lock *et al* (2005b) for a case in the high superposed flow rate “inertial regime” (see section 2) that matches typical non-dimensional pre-swirl flow rates used in engines. A smooth distribution of values for h, fitted to around twenty-five digitised measurements (both radially and tangentially), was applied over the impingement surface. The heat transfer coefficient on the back (unheated) surface of the disc was given a uniform value h = 80 W/m^2 K. (The use of a correlation from Owen and Rogers (1989) applicable to heat transfer from an unconfined rotating disc was also tested, but did not alter significantly the results obtained.) As the heat transfer coefficients were measured for the rotating disc at a specific value of rotational speed (this being around 5000 rpm), additional conditions to represent rotation in the thermal model were not required. The bulk fluid temperature at the unheated (back) face of the disc, and the initial temperature of the disc, was taken as 293 K. The bulk fluid temperature for the impingement face was taken as the inlet air total temperature measured in the experiment, 333 K. (The heat transfer coefficients given by Lock *et al* (2005b) were based on an “adiabatic wall temperature” for the rotating disc surface, estimated using the inlet air total temperature and a fluid dynamics termed that varied with radius. The influence of the simplified treatment used here is believed to be small. Details of the adiabatic wall temperature are given by Newton *et al*, 2003.)

The computational mesh is illustrated in Figure 5b. Tests showed that a model containing about 66,000 thermal elements was required in order to obtain results that did not vary significantly with further refinement to the mesh. Although periodicity suggests that only one of the sixty receiver holes in the experimental rig need be modelled, meshing quality issues at the tangential boundaries motivated the use of an 18° sector involving three holes. Adiabatic boundary conditions were applied at the tangential faces and at the surfaces inside receiver holes, as the holes were insulated in the experiments. The thickness of the disc is 2.5 times the diameter of the receiver holes.

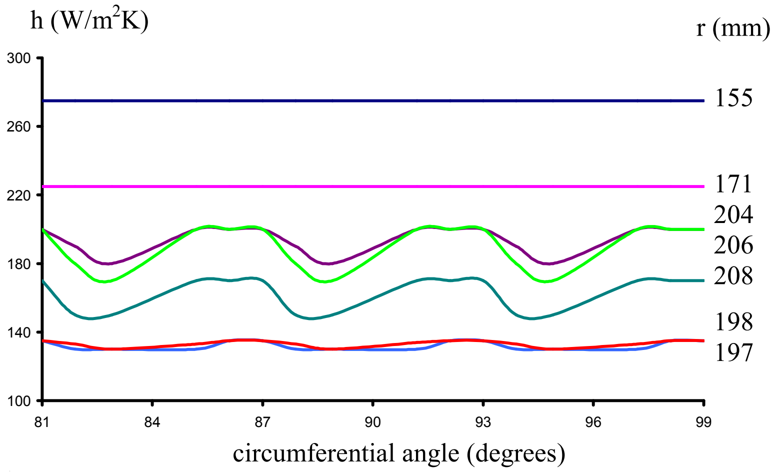

Figure 5a Impingement surface boundary conditions

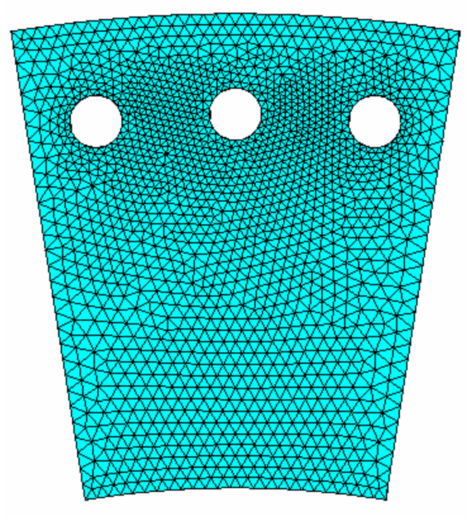

Figure 5b computational mesh

4. Comparisons between simulation and experiment

Due to the simplifications made in modelling the disc and the boundary conditions, qualitative comparisons were made between the results of the simulation and measurements available from the experimental rig. Figure 6 shows a computed radial distribution of temperature at the impingement surface at the end of the transient simulation. The peak temperature at r ≈ 160 mm (x ≈ 0.74) is due to the impingement of the heated superposed flow (which gives rise to the relatively high values of h used in this region). Similarly, increasing values of heat transfer coefficient with radius cause the slight rise in disc temperature around the receiver holes to up to r ≈ 205 mm.

The variations in temperature occurring between the receiver holes were computed to be very small, verifying that the one-dimensional analysis used in the processing of the experimental results is appropriate in this respect, while the modest variations in temperature shown in Figure 6 also supports the use by Lewis *et al* (2006) of an averaged uniform surface temperature for steady-state turbulent flow and heat transfer computations based on the transient experiments. The transient thermal simulation carried out here showed qualitative agreement with video recordings of the TLC colour variation made during the experiment (the duration of which is the order of one minute), indicating realistic behaviour of the computational model.

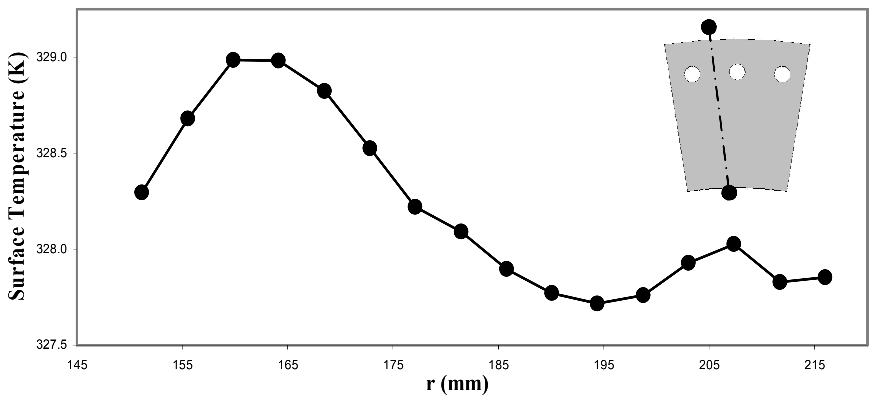

Figure 6 Computed radial surface temperature distribution for the polycarbonate disc

5. Simulations at engine operating conditions

Owen and Rogers (1989) showed that Nusselt numbers due to a turbulent boundary layer flow over an unconfined rotating disc may be expected to be proportional to $Re^{0.8}$, where $Re = \rho\Omega b^2/\mu$ is the rotational Reynolds number. Lewis *et al* (2006) showed that Nusselt numbers measured by Lock et al for the pre-swirl system correlated well with $Re^{0.8}$ for cases in the viscous (boundary layer) regime, though this was a less satisfactory correlating parameter for cases in the inertial regime.

For the present study the heat transfer coefficients measured on the experimental rig, for which $Re \approx 1 \times 10^6$, and used for the simulation described in section 4, were scaled to a typical engine operating condition $Re = 1 \times 10^7$ using $Re^{0.8}$ as the scaling parameter (giving an increase in the values for Nu by a factor of around six). The finite element model of the disc was adapted to use these higher values for h over the impingement surface and the material properties for the metal alloy Inconell given in Table 1. For these simulations, the boundary conditions represent the heat transfer due to cooling air (at a bulk temperature of 800 K, based on modern gas turbine HP compressor operating conditions) flowing over a heated disc. For the back face of the disc, a uniform value h = 900 W/m^2 K (and bulk temperature 950 K) was applied.

As described in section 4 the receiver holes in the experimental rig were insulated, resulting in the use of an adiabatic boundary condition at the surfaces inside the holes. For simulations at engine conditions it was assumed (following Al-aqal, 2003) that the heat transfer coefficient at the surface inside a hole is similar to that at the adjacent disc surface. A value h = 1000 W/m^2 K (and bulk temperature 800 K) was therefore used for the surfaces inside receiver holes. A further convective boundary condition, with h = 900 W/m^2 K at bulk temperature 1100 K, was applied at the outer rim of the disc, in order to represent the effects of the external hot gas stream present in an engine.

The computed steady-state results are illustrated in Figure 7. In comparison with the results at rig conditions, Figure 6, the inertial impingement effects here give rise to a relatively cool region of the disc at low radius, while temperature variations around the receiver holes are evident in spite of the additional effects of conduction from the simulated hot gas stream at the periphery of the disc. Due to the elevated heat transfer rates in this simulation compared with the experiment, more significant temperature variations are computed around the holes than for the simulation described in section 4. Two-dimensional conduction effects may occur inside the disc as a result, and these were not accounted for in the analysis leading to the measured values of h. Kingsley-Rowe *et al* (2005) considered the effect of lateral conduction on heat transfer coefficients calculated using one-dimensional assumptions, and the method they described could in principle be used to adjust the scaled measured of h used here.

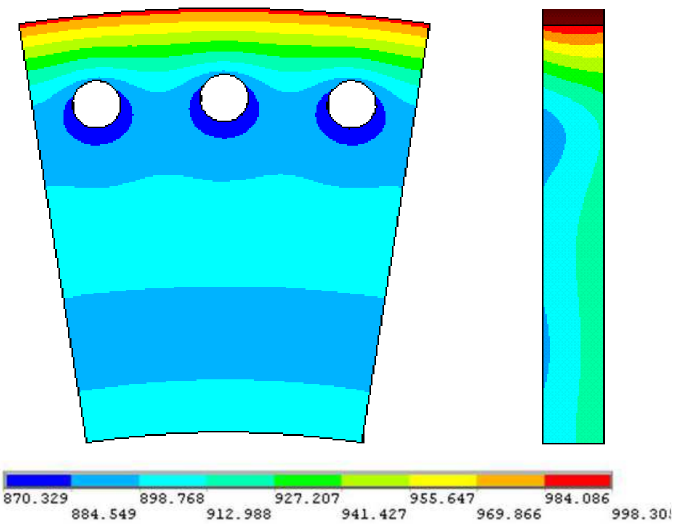

Figure 7 Disc temperature distribution (K) computed at engine operating conditions

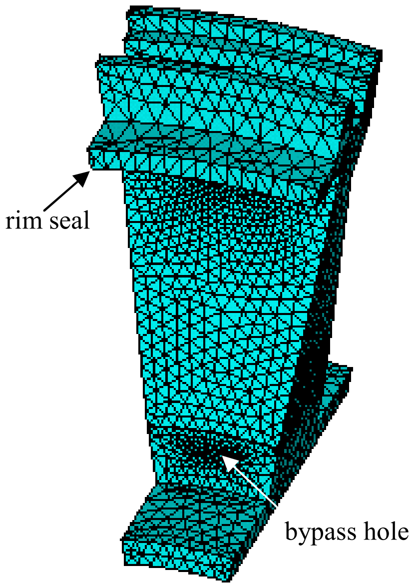

Figure 8a Computational model of a hypothetical turbine rotor

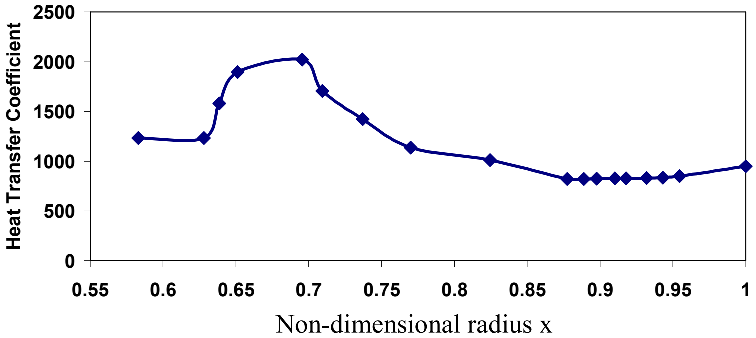

Figure 8b Assumed radial distribution of h (mid-way between blade cooling holes)

6. Application to a gas turbine rotor disc configuration

A new finite element model was constructed based on general dimensions for a generic high pressure (HP) turbine rotor disc (Varah, 2005). An 18° sector of the disc was modelled (again involving three equi-spaced blade cooling passages), and some geometrical simplifications were made for modelling convenience. The mesh used for the computations is shown in Figure 8a. Measured heat transfer coefficients, scaled to engine conditions as described in section 5, were again used for the impingement surface of the disc (the HP turbine side, see Figure 1) for this configuration. Values were adapted and mapped to this different geometry as a function of non-dimensional radius x, see Figure 8b. The model includes a bypass hole at low radius, and the blade cooling passages are straight and circular with entrances at a radius close to that of the rim seal. The disc outer radius, and the rotation rate assumed (in order to scale the measured values for Nu) corresponded to a rotational Reynolds number of $Re \approx 1 \times 10^7$ for this hypothetical application.

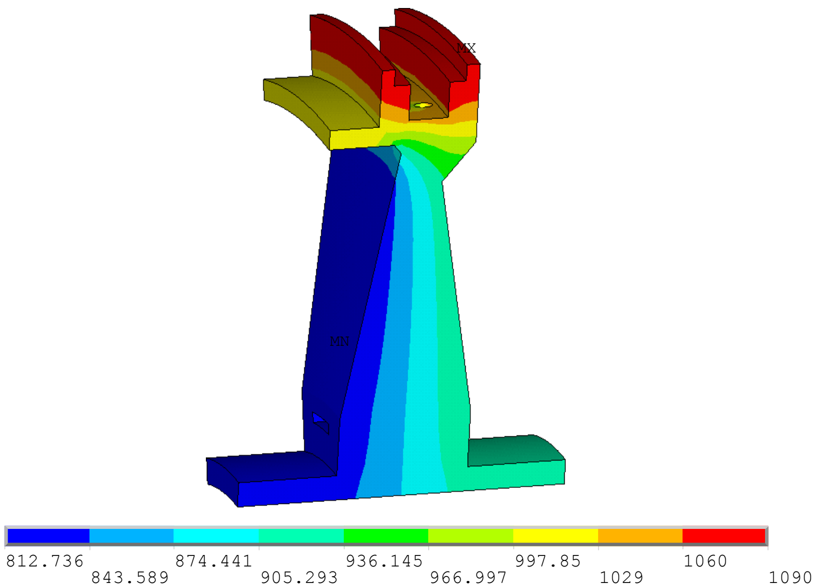

Figure 9 Computed turbine disc temperature distribution (K)

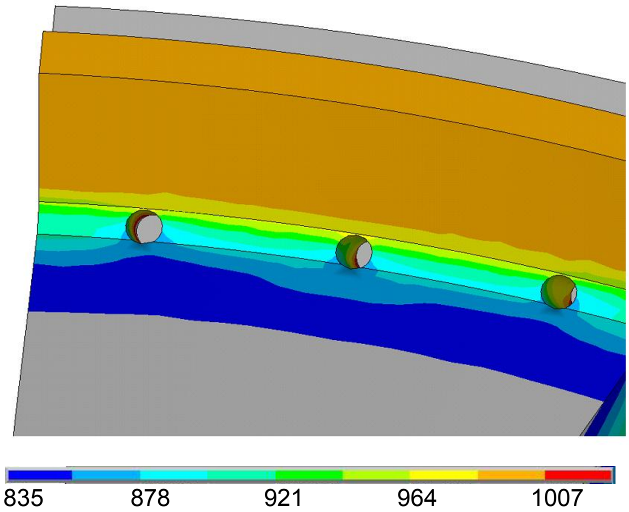

Figure 10a Computed rotor temperatures (K)

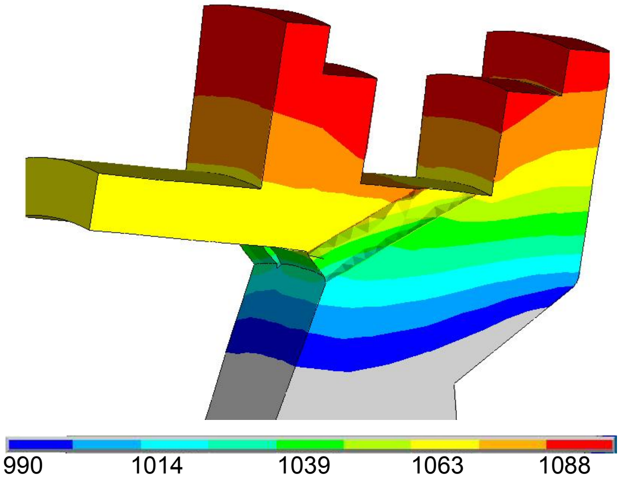

Figure 10b Computed rotor temperatures (K) in a section through a cooling passage

A fixed heat transfer coefficient of 1000 W/m^2 K (with an associated bulk temperature of 1100 K) was used at the blade attachment surfaces at the periphery of the model. This allowed the plane at the blade-cooling hole exit to reach an equilibrium condition rather than remaining at a prescribed temperature. (This condition had negligible effects inboard of the outer rim seal but allowed for cooling around the exit of the holes.) A heat transfer coefficient of 1050 W/m^2 K (assuming 800 K bulk temperature for the cooling air as in section 5) was applied to the internal surfaces of the blade-cooling holes, allowing the effect of the cooling flow to be represented. Cooling flow through the bypass hole at low radius was modelled similarly. A uniform heat transfer coefficient of 900 W/m^2 K (at 1000 K bulk temperature) was used on the back (low pressure turbine) face of the disc.

The steady state temperature distribution computed using this model is illustrated in Figure 9. In comparison with the results shown in Figure 7, there is in general a more uniform radial distribution of temperature on the tapered face of the impingement surface, but in other respects the distributions are very similar. The lower temperature shown at the exit of the hole in Figure 9 is due to the simulated effect of the cooling air flow in the blade cooling passage.

Figure 10a and Figure 10b show, respectively, the computed metal temperatures at the rotor surface around the blade-cooling holes and at a section passing through one of the blade cooling passages. The effects of the pre-swirl cooling flow and the simulated conduction from the hot external gas stream result in significant temperature variations around the entrances to the receiver holes, Figure 10a. In comparison with the results for the plane disc discussed in section 5, temperature variations are concentrated closer to the edges of the blade-cooling holes as a result of the more complicated local geometry and different hole size.

This computed temperature distribution was used to produce the corresponding prediction of thermal stresses within the rotor. The results are illustrated in Figure 11. Slightly higher stresses occur on the back (LP) turbine side of the disc, due to the higher temperatures on this side. (The cooling flow through the bypass hole also had a significant local effect on the thermal stress, as the stresses are comparatively low in adjacent areas.) The computed thermal stresses are small on the tapered HP rotor surface but increase around the receiver holes.

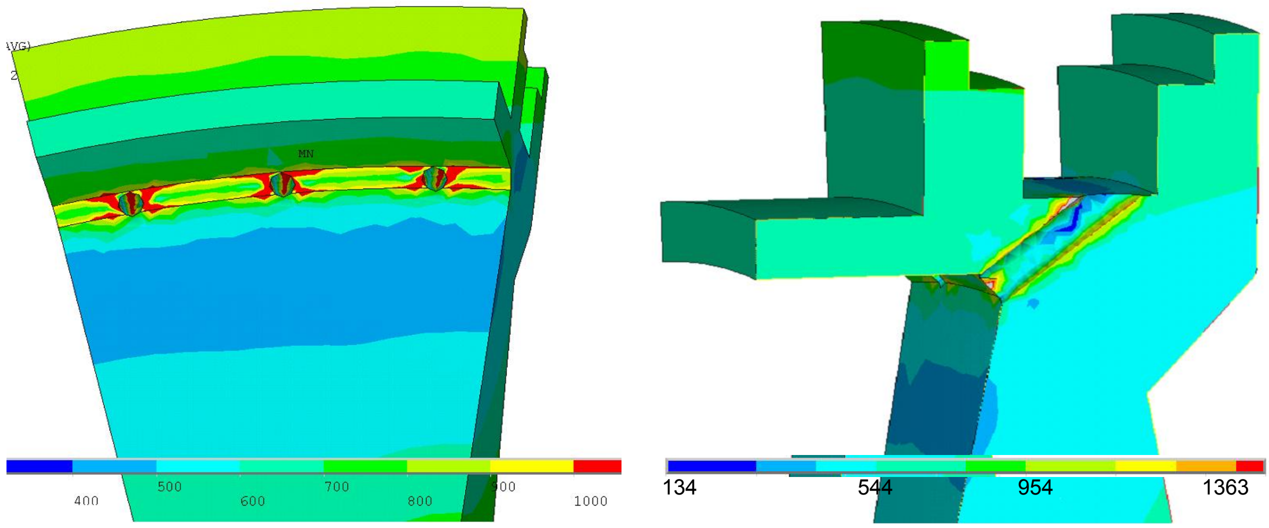

Figure 11 Computed thermal stresses (KPa) in the hypothetical turbine rotor

The levels and distribution of thermal stress around the blade-cooling holes, due mainly to the thermal effects of the cooling flow (see Lewis et al, 2006), would be likely to encourage the propagation of cracks from the edges of the holes. Figure 11 shows highest levels of stress around the blade-cooling hole inlets, specifically where the tapered disc merges with the outer rim seal. (The stresses occurring at the sharp corner between the disc and the rim seal would be reduced by using a suitable fillet radius.) Figure 11 also shows increased thermal stress at the surface of the blade-cooling passage inside the disc. There is a large temperature difference at the sharp points of the elliptical hole section at the blade attachment surface (see Figure 9), resulting in highest stresses in excess of 1000 KPa. The stresses on the blunt edges of the ellipse are around 550 KPa. (A subsequent preliminary mechanical stress simulation, at a hypothetical rotational speed of 15000 rpm, suggested that the thermal stresses are at most around 3% of the combined stress for the disc. Although the peak stress values are associated with simplifications made to the geometry of the model, the combined stress levels computed here suggest a fatigue life of around 15,000 hours, which is consistent with the usual service intervals for the type of engines considered.)

The temperature distribution illustrated in Figure 9 agrees qualitatively with results obtained by Monico and Chew (1992), which benefited from the availability of temperatures and heat transfer coefficients taken from real engine information. (Lower metal temperatures were indicated however, due in part to the use of different materials.) The computed temperature distributions around the rim seal in the present study also appear to show broad similarity with this earlier work. As the computed temperature distribution is due to the imposed heat transfer coefficients, this suggests that the scaled measurements of Lock *et al* (2005a,b) are representative of the engine situation. Although more detailed investigation is required, the detailed measurements of Lock *et al* should therefore be relevant to the validation of computational models for flow and heat transfer intended for application to engines. The generic geometry and boundary conditions described here have also been used, by Kim and Fenton (2006), to investigate the application of computational techniques for topology optimisation to turbine rotors based on temperature distribution and thermal stress considerations.

7. Conclusions

A finite element model has been developed to simulate a transient heat transfer experiment involving a rotating disc and a superposed flow of air over the disc surface. Measured values of heat transfer coefficient were used to devise boundary conditions for one surface of the disc, so that the temperature distribution within the disc could be computed. The results of the simulation were consistent with observations of thermochromic liquid crystal transient experiments.

The finite element model was also used to investigate the temperature variations that are likely to occur around the blade-cooling holes in a rotating disc in a gas-turbine engine, using values of heat transfer coefficient scaled from the measurements to typical engine operating conditions with the rotational Reynolds number as the scaling parameter. The simulations show that significant variations of metal temperature may be expected to occur around the blade-cooling holes in turbine discs, and that the consequent thermal stresses could affect the integrity of the discs should cracks occur around the surfaces of the holes. The computed results for a hypothetical turbine disc geometry agree qualitatively with findings from previous studies, suggesting that detailed heat transfer measurements made on a generic rotating-disc research rig can be used to develop and verify computational models of flow and heat transfer for gas turbine applications.

8. Acknowledgement

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