Modelling and simulation of transient thermal loading in a gas turbine

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ABSTRACT

The nozzle guide vane of the gas turbine is considered to be one of the most critical engine components from a thermal stress point of view. Hot gases from the combustor exhaust heat the outer surfaces of the vane, and because the turbine vane is cooled internally by air diverted from the compressor, a temperature gradient results between the inner and outer surfaces of the vane. These high gradients often make the nozzle guide vane the most likely engine component to fail due to low cycle fatigue. During engine transients each engine acceleration and deceleration induces a cycle of thermal stress, which can eventually cause component failure.

Therefore it is very important to predict the thermal loading of turbine components and assess how that can affect component life. For that purpose a heat transfer model for a cooled turbine component has been developed taking advantage of simplifications in heat flow analysis.

Numerical simulations have been carried out using the generic gas turbine simulation tool GasTurboLib. This Simulink library enables 0-D modelling, which is the simplest level of modelling, but most widely used in industry. The developed model of a cooled turbine has been implemented into turbine block of GasTurboLib library, enabling simulation and analysis of heat transfer effects within turbine components in more details.

The transient behaviour of a gas turbine during full load acceptance has been simulated to validate the proposed model against engine test data, and good agreement with test results has been observed. Analysis of thermal loading for a turbine component during engine rapid transients has been carried out and numerical results are presented in this paper.

NOMENCLATURE

Variables

A area [m^2]
C specific heat [J/kg k]
D diameter [m]
h heat transfer coeff [W/Km^2]
k thermal conductivity [W/Km]
L length [m]
$\dot{m}$ mass flow [kg/s]
n rotational speed [rpm]
P pressure [Pa]
q heat flow [J/s]
R gas constant [J/kg K]
T temperature [K]
V volume [m^3]
γ specific heat ratio [-]
η efficiency [-]
θ heat [J]
π pressure ratio [-]
ρ density [kg/m^3]
τ time constant [s]

Indices

c cold side
cl coolant
d design point
h hot side
hg hot gas
in inlet
out outlet
s surface
t turbine
w wall
∞ free stream

1. INTRODUCTION

Gas turbine engine hot section components operate under cyclic loading conditions and harsh environments, and hence deterioration of these components is accelerated. Due to elevated temperatures and operational stresses, the design life of a hot section component is significantly reduced compared to that of a cold section component. Deterioration is generally described by damage evolution, and with uses of new materials, previously unseen damage mechanisms are being discovered. The most common failure modes for a gas turbine engine include: low cycle fatigue (LCF), high cycle fatigue (HCF), creep, rupture, corrosion and oxidation [1].

Generally speaking the performance of current industrial gas turbines is limited by the temperature and strength capabilities of the materials used. Since industrial gas turbines are designed to run continuously for extended periods of time, the primary failure modes for the blades are HCF and creep.
HCF can result from a combination of steady stress, vibratory stress, and material imperfections that initiate the formation of small microscopic cracks. Since today's advanced turbomachinery blading is designed to have high steady stress levels, HCF occurs because of high mean stress – low amplitude vibratory loading of the airfoils.
On the other hand, some engines perform more cycles between start-up, maximum load and shut down and therefore in these cases LCF is the dominant failure mechanism.

The nozzle guide vane of the gas turbine is considered to be one of the most critical engine components from a thermal stress point of view. High temperature gradients often make the nozzle guide vane the most likely engine component to fail due to low cycle Thermal Fatigue (TF). During engine transients each engine acceleration and deceleration induces a cycle of thermal stress, which can eventually cause component failure.
Hot section components also develop so-called Thermo Mechanical Fatigue (TMF), which is a unique type of fatigue because the material is simultaneously subjected to fluctuating loads and temperatures. Isothermal life prediction techniques are often not applicable to TMF because different damage mechanisms can arise under extreme temperature conditions. We can recognize two different TMF cycles: in-phase which occurs when the maximum strain and peak cycle temperature are present, and out-phase, where the maximum strain and lowest cycle temperature coincide. One example of out-phase cycles is the TMF damage occurred on the gas turbine blade from repeated turbine starts and stops.

It is obvious that temperature is the key factor in hot section component life. Heat transfer models are usually used to model the distribution of component temperatures. The metal temperatures are then used in conjunction with the pressure and centrifugal loads, as well as the material models to determine the hours of life used or remaining, typically through finite element analysis [2].
In this work a turbine heat transfer model has been developed and implemented into a transient performance simulation tool, which has been used for the evaluation of thermal loading of engine hot section components in a high pressure turbine.

2. TURBINE MODELLING FOR DYNAMIC PERFORMANCE ANALYSIS

Generic gas turbine simulation tool GasTurboLib [3] has been used as a numerical test bed in this work. GasTurboLib is Simulink blockset library for gas turbine engine performance analysis, which enables both, steady state and transient simulations for different gas turbine configurations. The engine model is created by arranging different library blocks, i.e. components, representing a configuration similar to the gas turbine type to be simulated. This tool offers building 0-D models, where averaged fluid characteristics of engine components are computed at discrete positions inside the engine – at the inlet and outlet of each engine component.

The fully open architecture of this tool supports a “zooming” concept, where high order mathematical models can be implemented, allowing high-fidelity simulations of local phenomena in the engine, together with less complex models of the rest of the gas turbine. Applying the “zooming” philosophy to the engine transient simulation tool GasTurboLib has expanded the capability to simulate heat transfer effects within turbine components in more details. A description of the implemented turbine heat transfer model is given in this section.

The turbine is modelled with the turbine body representing the steady state turbine model and interconnecting volumes. The steady performance characteristics are not able to handle transient events, but if they are coupled with volume dynamics, which continuously feed the steady state maps with unsteady data, satisfactory dynamic responses are achievable [4].
Heat transfer between the gas in the volume and the metal components is modelled to represent heat soak effects, and mixing of compressor cooling air at the turbine inlet is also considered.

The turbine is represented by two steady state maps, which are based on the turbine pressure ratio $\left(\pi_t = P_{in\_t} / P_{out\_t}\right)$ and the turbine corrected speed $\left(\overline{n}_t = n_t / \sqrt{T_{in\_t}}\right)$:

- turbine corrected mass flow map

$$\frac{\dot{m}_t \sqrt{T_{in\_t}}}{P_{in\_t}} = tfmap\left(\pi_t, \overline{n}_t\right) \tag{1}$$

- turbine normalized efficiency

$$\eta_t = temap\left(\pi_t, \overline{n}_t\right) \tag{2}$$

For a given pressure ratio and corrected speed, using turbine steady state characteristics, values for the turbine corrected mass flow and efficiency were evaluated through interpolation on digitized maps. Turbine temperature drop is given by:

$$T_{out\_t} = T_{in\_t} \pi_t^{\frac{\eta_t \left(1-\gamma_{hg}\right)}{\gamma_{hg}}} \tag{3}$$

where γ - specific heat ratio varies with temperature, and is determined at the arithmetic average between T_{in_t} evaluated at the current time step and T_{out_t} obtained at the previous time step.

In the adopted modelling approach the turbine is represented with a volume and flow model. Two volume models are considered, one for the hot gas path and second for the coolant fluid (Fig. 1.). A flow model is the result of a modelling abstraction, where the volume is neglected. These flow models contain algebraic equations that describe the flow of mass and energy. To model the dynamic behaviour of the turbine components, the following generic equations for pressure and temperature have been used:

$$
\begin{aligned}
\frac{dP_{out}}{dt} &= \left(\frac{\partial P_{out}}{\partial t}\right)_v + \sum_j \frac{\partial P_{out}}{\partial \varsigma_j}\frac{\partial \varsigma_j}{\partial t} \\
&= \left[\frac{RT_{out}}{V}\left(\sum_i \dot{m}_{in}^i - \dot{m}_{out}\right) + \frac{P_{out}}{T_{out}}\frac{\partial T_{out}}{\partial t}\right] + \sum_j \frac{\partial P_{out}}{\partial \varsigma_j}\frac{\partial \varsigma_j}{\partial t} \\
\frac{dT_{out}}{dt} &= \left(\frac{\partial T_{out}}{\partial t}\right)_v + \sum_l \frac{\partial T_{out}}{\partial \xi_l}\frac{\partial \xi_l}{\partial t} \\
&= \frac{1}{C_v \rho V}\left[\left(\sum_i \dot{m}_{in}^i C_p^i T_{in}^i - \dot{m}_{out} C_p T_{out}\right) + q - C_v T_{out}\left(\sum_i \dot{m}_{in}^i - \dot{m}_{out}\right)\right] \\
&+ \sum_l \frac{\partial T_{out}}{\partial \xi_l}\frac{\partial \xi_l}{\partial t}
\end{aligned}
\tag{4}
$$

where C_p and C_v are specific heats and q is heat flow. The flow model variables, ς_j and ξ_l are internal thermodynamic state (pressure, temperature) and mechanical variables (rotational speed, geometrical data) used to model irreversible or dissipative processes. Variables with no subscripts refer to properties within the gas volume. Variables with subscript X_{out} refer to the outlet stream and subscript X_{in}^i refers to inlet streams i=1 to n.

The first term in the above equations, represents volume gas dynamics using an ordinary differential equation, and the second term describes the behaviour of components using nonlinear algebraic equations, which are based on steady state component characteristics.

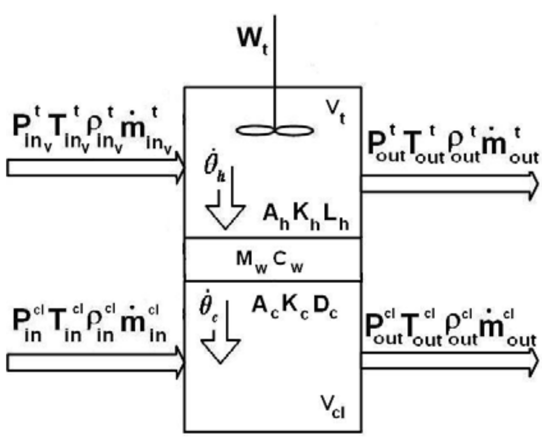

Fig. 1. Cooled turbine modelling framework

Generic equations for modelling of the temperature and pressure transients of the turbine working fluid (hot gas) volume take the following form:

Flow model

$$\sum_j \frac{\partial P_{out}}{\partial \varsigma_j}\frac{\partial \varsigma_j}{\partial t} = \frac{dP^f_{out\_t}}{dt} = \pi_t \dot{P}_{in\_t} + \prod \dot{P}_{in\_t}$$

$$\sum_l \frac{\partial T_{out}}{\partial \xi_l}\frac{\partial \xi_l}{\partial t} = \frac{dT^f_{out\_t}}{dt} = \pi_t^{\frac{\eta_t(1-\gamma_{hg})}{\gamma_{hg}}} \dot{T}_{in\_t} + Q\dot{T}_{in\_t} \tag{5}$$

In this modelling framework, the flow model is represented by pressure $\prod$ and temperature Q drop coefficients and empirical information derived from rig tests, i.e. turbine component maps $\pi_t = tfmap(\dot{\overline{m}}_t, \overline{n}_t)$ and $\eta_t = temap(\pi_t, \overline{n}_t)$.

Volume model

$$\frac{\partial P_{out\_t}}{\partial t} = \frac{R_t T_{out\_t}}{V_t}\left(\dot{m}_{in\_t} - \dot{m}_{out\_t}\right) + \frac{P_{out\_t}}{T_{out\_t}}\frac{\partial T_{out\_t}}{\partial t}$$

$$\frac{\partial T_{out\_t}}{\partial t} = \frac{1}{C_{v\_t}\rho_t V_t}\begin{bmatrix}\left(\dot{m}_{in\_t}C_{p\_in\_t}T^v_{in\_t} - \dot{m}_{out\_t}C_{p\_out\_t}T_{out\_t}\right) \\ + q_h - C_{v\_t}T_{out\_t}\left(\dot{m}_{in\_t} - \dot{m}_{out\_t}\right)\end{bmatrix} \tag{6}$$

To model coolant fluid volume of the turbine component, the following governing equations have been utilized:

$$\sum_j \frac{\partial P_{out}}{\partial \varsigma_j}\frac{\partial \varsigma_j}{\partial t} = 0$$

$$\sum_l \frac{\partial T_{out}}{\partial \xi_l}\frac{\partial \xi_l}{\partial t} = 0$$

$$\frac{\partial P_{out\_cl}}{\partial t} = 0 \tag{7}$$

$$\frac{\partial T_{out\_cl}}{\partial t} = \frac{1}{C_{v\_cl}\rho_{cl} V_{cl}}\begin{bmatrix}\left(\dot{m}_{in\_cl}C_{p\_in\_cl}T_{in\_cl} - \dot{m}_{out\_t}C_{p\_out\_cl}T_{out\_cl}\right) \\ + q_c - C_{v\_cl}T_{out\_cl}\left(\dot{m}_{in\_cl} - \dot{m}_{out\_cl}\right)\end{bmatrix}$$

Assuming that the following condition is satisfied $\dot{m}_{in\_cl} - \dot{m}_{out\_cl} = 0$ the above equation reduces to:

$$\frac{\partial T_{out\_cl}}{\partial t} = \frac{1}{C_{v\_cl}\rho_{cl} V_{cl}}\left[\left(\dot{m}_{in\_cl}C_{p\_in\_cl}T_{in\_cl} - \dot{m}_{out\_t}C_{p\_out\_cl}T_{out\_cl}\right) + q_c\right] \tag{8}$$

3. HEAT TRANSFER CLOSURE EQUATIONS

Two distinct mechanisms of heat transfer can be recognized in the cooled turbine configuration. The heat transfer analysis for cooled turbine is shown schematically in figure Fig.2. In considering cooled turbine component (vane or blade), it can be seen that the hot gas transfers certain amounts of thermal energy flow or heat flow to metal. To maintain a desired metal temperature, the cooling mass flow through the component internal channels has to remove the heat flow from the material.

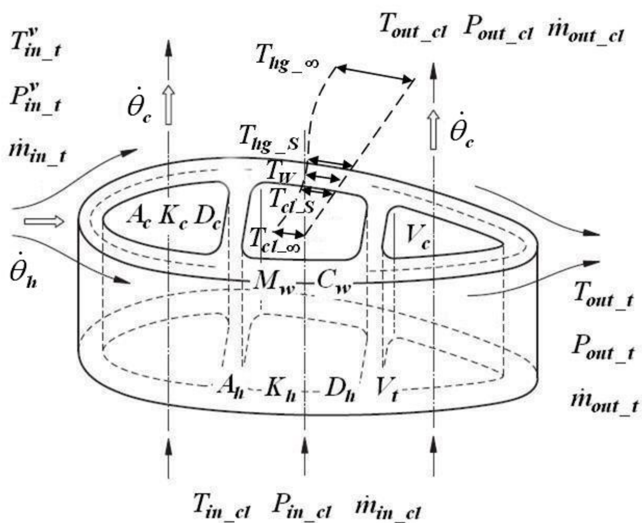

Fig. 2. Scheme of convective cooling of turbine component

The first mode that represents heat flow transferred from the working medium – hot gas to metal $\dot{\theta}_h$ can be described as [5]:

$$\dot{\theta}_h = h_h A_h \left(T_{hg\_s} - T_w\right) \tag{9}$$

where h_h is averaged heat transfer coefficient for hot side, A_h is the contact surface for hot side, T_{hg_S} is hot gas temperature on the hot side surface and T_w is wall temperature.

The second mode represents heat flow, which is removed from the metal by the cooling medium – coolant, and is given by [5]:

$$\dot{\theta}_c = h_c A_c \left(T_w - T_{cl\_s}\right) \tag{10}$$

where h_c is averaged heat transfer coefficient for cold side, A_c is area exposed to coolant, T_{cl_S} is coolant temperature on the cold side and T_w is wall temperature.

To evaluate the convective heat transfer coefficient for the calculation of transferred thermal energy on hot side:

$$h_h = N_{u_h} \frac{k_h}{L_h} \quad (11)$$

we use k_h thermal conductivity, L_h characteristic length – blade \ vane cord length and a Nusselt number correlation:

$$N_{u_h} = f\left(R_e, P_r, T_i, S_{tr}, R_s, A_r\right) \quad (12)$$

where the major parameters affecting the turbine heat transfer are Reynolds number R_e, Prandtl number P_r, Strouhal number S_{tr}, turbulence intensity T_i, surface roughness R_s and acceleration ratio A_r. In this model a simplified approach has been utilized, and the heat transfer coefficient has been related to the design point hot gas temperature and mass flow values using following empirical formulation:

$$h_h = \left(\frac{T_{hg\_\infty}}{T_{d\_hg}}\right)^x \left(\frac{\dot{m}_{hg}}{\dot{m}_{d\_hg}}\right)^y \frac{k_{eff\_h}}{L_h} \quad (13)$$

where T_{d_hg} and $\dot{m}_{d\_hg}$ are design point hot gas temperature and mass flow respectively. T_{hg_∞} is the hot gas temperature of the free stream, $\dot{m}_{hg}$ is mass flow rate, and $K_{eff\_h} = N_{u_h} K_h$ effective thermal conductivity on the hot side.

Similarly, the convective heat transfer coefficient for the calculation of transferred thermal energy on the cold side is defined as:

$$h_c = \left(\frac{T_{cl\_\infty}}{T_{d\_cl}}\right)^z \left(\frac{\dot{m}_{cl}}{\dot{m}_{d\_cl}}\right)^v \frac{k_{eff\_c}}{D_c} \quad (14)$$

where T_{d_cl} and $\dot{m}_{d\_cl}$ are the design point temperature and mass flow rate of the coolant respectively. Effective thermal conductivity is defined with $k_{eff\_c} = N_{u_c} K_c$ and D_c represents the diameter of coolant hole. T_{cl_∞} is bulk temperature of the coolant and $\dot{m}_{cl}$ is the actual mass flow rate.

The coupling condition between the cold and the hot side is provided by the material temperature differential equation. This equation can be derived from the energy balance equating the heat flux to the temperature rise of the component:

$$\dot{\theta} = M_w C_w \frac{dT_w}{dt} \quad (15)$$

where C_w and M_w are heat capacity and mass of the component respectively.

Heat flux equation for the wall is defined as follows:

$$\dot{\theta} = \dot{\theta}_c + \dot{\theta}_h, \tag{16}$$

assuming that the heat flow on the hot side $\dot{\theta}_h$ can have a positive or negative value. Heat flow on the cold side $\dot{\theta}_c$ is always negative, since heat is rejected, and for the case of no cooling $\dot{\theta}_c = 0$. For the steady state or dynamic equilibrium case where the blade temperature change approaches zero, following condition is satisfied:

$$\dot{\theta}_c + \dot{\theta}_h = 0 \tag{17}$$

Hence, equating the above expressions, one can write the following dynamic equation of the material temperature:

$$\frac{dT_w}{dt} = \frac{\left(T_{hg\_S} - T_w\right)}{\tau_{wh}} + \frac{\left(T_w - T_{cl\_S}\right)}{\tau_{wc}} \tag{18}$$

In the above equation τ_{wh} and τ_{wc} are component thermal time constants for the hot and cold side respectively, and they are given by:

$$\tau_{wh} = \frac{M_w C_w}{A_h \, k_{eff\_h} / L_h} \left(\frac{T_{hg\_\infty}}{T_{d\_hg}}\right)^{-x} \left(\frac{\dot{m}_{hg}}{\dot{m}_{d\_hg}}\right)^{-y}$$

$$\tau_{wc} = \frac{M_w C_w}{A_c \, k_{eff\_c} / D_c} \left(\frac{T_{cl\_\infty}}{T_{d\_cl}}\right)^{-z} \left(\frac{\dot{m}_{cl}}{\dot{m}_{d\_cl}}\right)^{-v} \tag{19}$$

4. GOVERNING EQUATIONS FOR MODELLING OF COOLED TURBINE COMPONENT

Assuming that the inlet stream temperatures of the control volumes correspond to the free stream temperatures of the hot gas and coolant fluid:

$$T_{hg\_\infty} = T^{v}_{in\_t} = T^{f}_{out\_t} \;\; \& \;\; T_{cool\_\infty} = T_{in\_cl}, \tag{20}$$

and equating the surface temperatures of the hot and cold side of the turbine material with the outlet stream temperatures for hot gas and coolant control volumes:

$$T_{hg\_S} = T_{out\_t} \;\; \& \;\; T_{cl\_S} = T_{out\_cl}, \tag{21}$$

Finally introducing the following substitutions:

$$q_h = \dot{\theta}_h \;\; \& \;\; q_c = \dot{\theta}_c, \tag{22}$$

the complete set of governing equations for the cooled turbine model takes the following form:

$$\dot{P}_{out\_t} = \frac{R_t T_{out\_t}}{V_t}\left(\dot{m}_{in\_t} - \dot{m}_{out\_t}\right) + \frac{P_{out\_t}}{T_{out\_t}}\dot{T}_{out\_t} + \dot{P}^{f}_{out\_t}$$

$$\dot{P}^{f}_{out\_t} = \pi_t \dot{P}_{in} + \prod \dot{P}_{in}$$

$$\dot{T}_{out\_t} = \frac{1}{C_{v\_t}\rho_t V_t}\begin{bmatrix}\left(\dot{m}_{in\_t} C_{p\_in\_t} T^{v}_{in\_t} - \dot{m}_{out\_t} C_{p\_out\_t} T_{out\_t}\right) \\ + \dot{\theta}_h - C_{v\_t} T_{out\_t}\left(\dot{m}_{in\_t} - \dot{m}_{out\_t}\right)\end{bmatrix} + \dot{T}^{f}_{out\_t}$$

$$\dot{T}^{f}_{out\_t} = \dot{T}^{v}_{in\_t} = \pi_t^{\frac{\eta_t\left(1-\gamma_{hg}\right)}{\gamma_{hg}}} \dot{T}_{in} + Q\dot{T}_{in}$$

(23)

$$\dot{P}_{out\_cl} = 0$$

$$\dot{T}_{out\_cl} = \frac{1}{C_{v\_cl}\rho_{cl}V_{cl}}\left[\left(\dot{m}_{in\_cl} C_{p\_in\_cl} T_{in\_cl} - \dot{m}_{out\_t} C_{p\_out\_cl} T_{out\_cl}\right) + \dot{\theta}_c\right]$$

$$\dot{\theta}_h = h_h A_h \left(T_{out\_t} - T_w\right) \qquad \dot{\theta}_c = h_c A_c \left(T_w - T_{out\_cl}\right)$$

$$\dot{T}_w = \frac{\left(T_{out\_t} - T_w\right)}{\tau_{wh}} + \frac{\left(T_w - T_{out\_cl}\right)}{\tau_{wc}}$$

$$\tau_{wh} = \frac{M_w C_w}{A_h \, {k_{eff\_h}}/{L_h}}\left(\frac{T^{v}_{in\_t}}{T_{d\_hg}}\right)^{-x}\left(\frac{\dot{m}_{hg}}{\dot{m}_{d\_hg}}\right)^{-y} \quad \tau_{wc} = \frac{M_w C_w}{A_c \, {k_{eff\_c}}/{D_c}}\left(\frac{T_{in\_cl}}{T_{d\_cl}}\right)^{-z}\left(\frac{\dot{m}_{cl}}{\dot{m}_{d\_cl}}\right)^{-v}$$

5. NUMERICAL SIMULATION

Simulation model generated using GasTurboLib tool can be used to calculate gas temperatures, pressures, mass flow and gas properties at relevant engine stations. This particularly applies to stations for which no measured data are available such as the critical high-pressure turbine entry temperature. Using this tool it is possible to accurately calculate the dynamic responses of parameters which are critical from the perspective of engine component life, but where measured data are not available or data are affected with high measurement time lags or low update frequencies.

The previously described model of the cooled turbine has been implemented into turbine component of GasTurboLib Simulink library. A simulation model has been built for a single shaft industrial gas turbine using GasTurboLib library components. The transient behaviour of the engine during full load acceptance has been simulated. Engine transient simulation has been carried out with an integration step size of 1 ms, which is sufficient to accurately calculate the critical effects such as typical severe acceleration / deceleration temperature transients in the hot section.

The validation test data was gained from a load acceptance test performed at site, where the engine was driving an alternator into resistive island load using local load banks, and good agreement with test results has been observed.

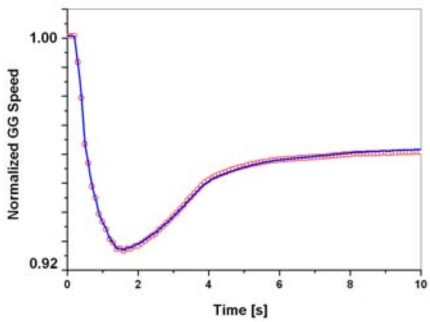

Fig. 3.a. Gas Generator speed – full load acceptance test

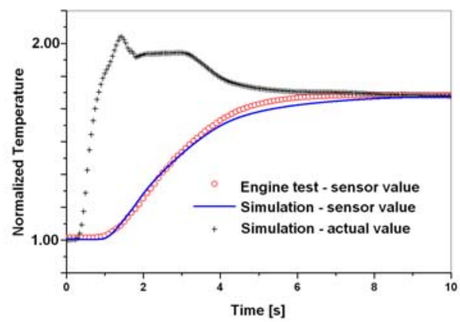

Fig. 3.b. Power Turbine Exit temperature – full load acceptance test

Following simulation values have been compared with engine test data: gas generator speed and power turbine exit temperature. These engine parameters are directly measured engine variables, and they are essential for the engine control system. The simulation results presented in Fig.3.a. and Fig.3.b., show that the predicted numerical results are in good agreement with real engine data.

6. TURBINE THERMAL RESPONSE

Numerical results of the thermal analysis for turbine components during block load acceptance are presented in this section. Free stream temperature profiles for hot gas and turbine metal temperatures are compared to the thermocouple measurements, which have been taken during engine transients (Fig. 4.).

The gas turbine has been tested with instrumented Nozzle Guide Vanes (NGV's), and temperature measurements have been taken during a full load acceptance test. The objective of the test was to measure NGV temperatures under realistic operating conditions in the engine. NGV's were fitted with 10 thermocouples, and positions of the instrumentation are shown in the inset of figure Fig.4. The thermocouples were positioned at the radial mid-span section of the NGV, where postions 1,10,9 and 8 are on the pressure side, and positions 2,3,4,5,6 and7 are on the suction side of the vane.

Figure Fig. 4. shows the numerical simulation prediction for the turbine transient temperature response for full load acceptance. One can see that the temperature profiles of the NGV's compare well with the numerical simulation results.

The heat transfer process during load acceptance is shown in Fig. 5. Heat energy flow from the hot gas to the metal and from the metal to the coolant is given in this graph. It can be seen that because the hot gases heat the outer portion of the turbine, heat flow q_h has positive value on the hot side, but on the other side - cold side, where turbine is cooled internally by air diverted from the compressor, heat flow q_c is negative since heat is rejected. As a consequence of this heat transfer process, a temperature gradient results between the inner and outer portions of the turbine. The higher the temperature gradients, the greater become the thermal stresses.

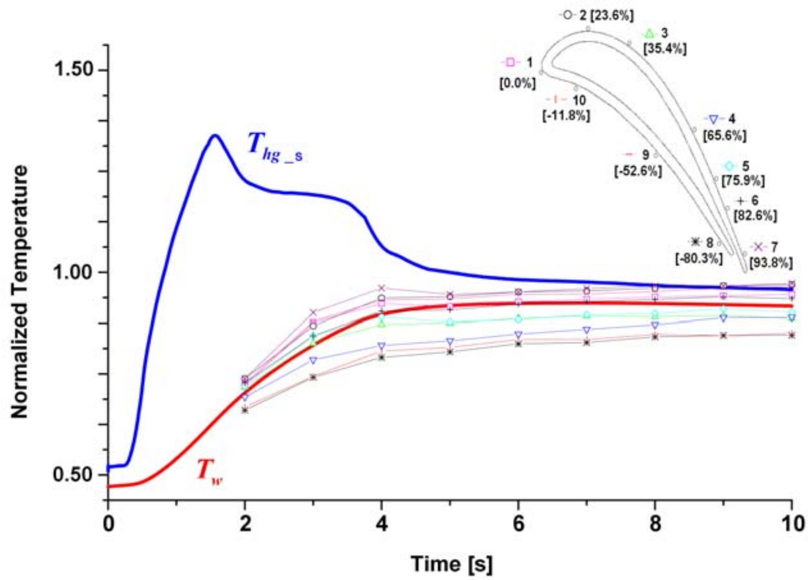

Fig. 4. Turbine component thermal profiles during load acceptance

The same figure shows time history of thermal loading q and total heat stored in the turbine component θ, which is obtained by integrating surface under thermal loading curve. This stored heat increases internal energy of material and hence component temperature, but it is also dissipated through deformation and damage processes.

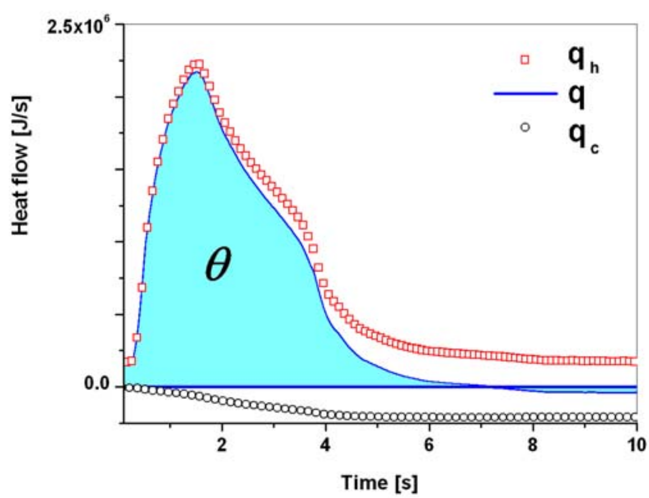

Fig. 5. Heat transfer from hot gas to metal and metal to coolant

One can observe that during engine transients the turbine component can be subjected to very rapid thermal loadings. These high temperature gradients and rapid thermal loadings are the main contributors to LCF damage, and they should be used as a basis of damage assessment of hot section components.

To predict the thermal stresses and induced damage, a general form of the three-dimensional equations should be obtained first and then the appropriate conditions relevant to particular problem could be applied. The governing three-dimensional partial differential equations for heat flow in any solid can be written as [6]:

$$k_x \frac{\partial^2 T}{\partial x^2} + k_y \frac{\partial^2 T}{\partial y^2} + k_z \frac{\partial^2 T}{\partial z^2} = \rho c \frac{\partial T}{\partial t} \tag{24}$$

and its boundary conditions as:

$$k_x \frac{\partial T}{\partial x} l_x + k_y \frac{\partial T}{\partial y} l_y + k_z \frac{\partial T}{\partial z} l_z + q = 0 \tag{25}$$

where q is determined thermal load defined with:

$$q = q_c + q_h \tag{26}$$

7. DISCUSSION

To prevent turbine damage induced by excessive, prolonged combustor outlet gas temperature, the engine is operated at a turbine peak temperature that is several degrees below the vane's critical life cycle fatigue temperature. Traditionally, the turbine is protected by the engine control unit based on the measured gas temperature (MGT).

With the engine operating at steady-state operating condition, adequate engine life can be assessed by limiting peak temperature based on the MGT. During transient operation, however, peak temperature may be exceeded because the response of the MGT is inadequate and does not reflect the true critical turbine temperature.

The developed heat transfer model has been used for calculation of the temperature transient loading in the turbine component, based on the heat input from hot gas stream. It has been demonstrated that the thermal loading of the turbine component can be determined accurately using Model Based Temperature (MBT) generated by transient performance simulation tool. Presented model based approach offers potential improvement in several areas:

- Life prediction analysis Because fatigue of gas turbine components is dependant on numerous factors, it is very difficult to predict the onset of fatigue in most of the cases. One of the important influencing factors is certainly the thermal loading of components. Studies in the past had not quantitatively calculated the transient heat loading in the component for the determination of fatigue life, and introduction of model based thermal loading, could lead to better prediction of component life.

- Over-temperature protection The MGT does not reflect the true critical temperature because the thermocouple probes in the hot section gas path are constructed for accuracy and durability, not quick response. With engines capable of full load acceptance in just over a couple of seconds, the transient gas temperature quickly increases. Although consideration of this signal lag is not critical for engine accelerations of long duration, the delay becomes most significant

when attempting to accurately compensate for thermocouple dynamics during rapid accelerations of short duration. In order to mitigate this problem the MBT limiting control parameter could be employed, which can adequately reflect the true critical turbine temperature loading during rapid engine transients.
- Health monitoring The thermal load history determined using the model based temperature of every individual component could be tracked down and could be used to determine the inspection interval or actual life limit of that specific component.

8. CONCLUSIONS

A method to predict the thermal loading of the engine hot section based on engine performance transient analysis is demonstrated on a gas turbine component. The thermal loading of the high-pressure turbine of a single shaft industrial gas turbine, one of the most severely loaded components in the engine, is simulated and numerical results are presented.

A heat transfer model for a cooled turbine component has been developed taking advantage of simplifications in the heat flow analysis, and it has been implemented into generic gas turbine simulation tool GasTurboLib. This simulation tool has been used to simulate heat transfer effects within a turbine component in more detail, and the numerical simulation of rapid transients during full load acceptance has been carried out to evaluate thermal loading of the turbine component.

To validate the proposed model, the gas turbine has been tested with instrumented Nozzle Guide Vane (NGV's). Temperature measurements taken during full load acceptance test have been compared with simulation results, and it has been shown that the temperature profiles of the NGV's compare well with numerical prediction.

It has been demonstrated that the gas path thermocouple probes have slow response characteristics as compared to that of the critical turbine hardware and hence the measured gas temperature can not reflect the true component thermal loading during rapid transients. The present study has shown that the model based thermal loading generated using the transient simulation tool can adequately reflect the critical temperature during transient operations, and it has potential to contribute towards better life prediction analysis, over-temperature protection and engine health monitoring.

9. REFERENCES

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