NUMERICAL CALCULATION OF THERMAL STRESSES IN A MICRO GAS TURBINE

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RESUMEN

Una microturbina de gas (MTG) es una turbomáquina de tamaño reducido que trabaja con gases a altas temperaturas. Esto causa diferencias de temperaturas y esfuerzos térmicos en sus componentes y son un parámetro de diseño crítico. La MTG se encuentra en fase de diseño y en este trabajo se calculan numéricamente los campos de temperaturas y de esfuerzos térmicos para un material usado para construir la MTG y para una curva de arranque propuesta. Los cálculos se realizaron usando programas comerciales de CFD y FEA (Fluent y ANSYS). Los esfuerzos térmicos más altos se localizan en el borde de salida de las toberas y en el borde de ataque de los álabes del rotor, respectivamente. Las magnitudes de las temperaturas y de los esfuerzos térmicos estuvieron por debajo de los límites de fluencia del material, por lo tanto, el material y la curva de arranque propuestas para la manufactura de la microturbina son viables.

ABSTRACT

A micro gas turbine (MGT) is a small turbomachine that works with high temperature gases. That leads to the development of temperature differences and thermal stresses in their components. In this work, temperature fields and thermal stresses in a micro gas turbine during at start-up transient cycle are numerically calculated. Due to the MGT is in design process, the material and the start-up curve were proposed. Calculations were carried out using CFD and FEA commercial codes, Fluent and ANSYS were performed to calculate temperature fields and thermal stresses, respectively. The highest thermal stresses were located on the nozzle trailing edge and on the rotor leading edge, respectively. The highest temperature value and the highest thermal stress reached were under the yield strength of the material, therefore, the material and the start-up curve proposed for the MGT manufacture are suitable.

Keywords: Thermal stress, Micro gas turbine, Conjugated heat transfer, CFD, FEA.

INTRODUCTION

In the last decades, clean and renewable energy sources have been studied due to issues like the global warming and the air pollution by exhaust gases. Micro gas turbine results a feasible option due to its low emission of polluting gases, however, it works with high temperature gases that impacts on the resistance and the remaining life of their components. Furthermore, temperature gradients in the turbine components raise as the turbine temperature inlet raises, and thermal damages are generated by high thermal stresses.

Thermal stresses are the result of the temperature gradients and the constraints in the materials. They reach their highest values during transient cycles as the turbine start-up, shutdown or load change. Thermal stresses produce Low-cycle fatigue (LCF) and they can induce cracks or lead to the material fatigue. During the start-up, the turbine components are subjected to several thermal stresses as result of the thermal elongations which are more intense for a shorter start-up [5]. When the start-up ends, the heating period ends and the temperatures and the thermal stresses reach a stationary stage [13]. In the present study, temperature fields and thermal stresses calculations were conducted using numerical methods.

Numerical methods have been used for heat transfer and thermal stresses calculations in turbomachinery...
due to their capacity to resolve complex problems. Almaral S. et al. [1] and Min K. et al. [8] calculated and analyzed thermal stresses in gas turbine blades, using both FEA and CFD calculations for temperature distribution and thermal stresses, respectively. They found that the highest temperature values and the highest thermal stresses were located on the blade trailing edge and on the blade root, respectively. On the other hand, Son, Bumshin and Sungho [13] calculated thermal stresses in the rotor of the high pressure turbine during start-up cycle. They performed the turbine start-up simulations where thermal stresses curves were obtained. Similar results were obtained by Kim T. S. et al [4] who analyzed transient thermal stresses of the heat recovery steam generator. Finally, Joanna Wilk [14] carried out temperature distribution calculations in a rotary heat exchanger’s rotor model during the start-up. She showed that the temperature was increased linearly during the start-up cycle. The results have been qualitatively compared with the investigations above mentioned.

In this work, temperature distribution and thermal stresses in a micro gas turbine of axial flow are calculated. Figure 1 shows the full MGT geometry that is in design process. For thermal stresses calculations, temperature distributions on the blades are required. Computational fluid dynamics (CFD) was employed for temperature distribution calculations solving a conjugated heat transfer (CHT) problem where the blades are heated by the interaction with combustion gases. After, FEA was used for thermal stresses calculations using temperature distribution obtained by CFD as boundary condition. Thermal stresses analysis on that kind of turbines is sparse and this analysis could provide useful information for the design and the operation of that turbomachines. The aim of this work was to show that thermal stresses and maximum temperatures were not higher than the yield strength limit of the material proposed for the MGT manufacture.

GOVERNING EQUATIONS

The governing equations of the flow dynamics are the follow:

- Continuity equation
  \[ \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho u) = 0 \]
  where
  \[ \nabla = \left[ \frac{\partial}{\partial x}, \frac{\partial}{\partial y}, \frac{\partial}{\partial z} \right] \]

- Momentum conservation equation or Navier-Stokes equation (N-S)
  \[ \frac{\partial}{\partial t} (\rho u_i) + \frac{\partial}{\partial x_i} (\rho u_i u_j) = \]
  \[ \rho g_i - \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_i} \left( \mu \frac{\partial u_i}{\partial x_j} \right) + \frac{1}{3} \frac{\partial}{\partial x_i} \left( \mu \frac{\partial u}{\partial x_j} \right) \]

- Energy equation
  \[ \frac{\partial}{\partial t} (\rho e_0) + \frac{\partial}{\partial x_j} (\rho u_j e_0 + u_j p + q_j - u_i T_{ij}) = 0 \]

where \( \rho \) is the density, \( u_i \) is the velocity in the \( i \) direction, \( p \) is the static pressure, \( \mu \) is the molecular viscosity and \( g_i \) is the gravitational acceleration. These equations describe the mass, momentum and energy conservation on a continuous flow in laminar or turbulent conditions.

The N-S equations were transform in the RANS (Reynolds Averaged Navier-Stokes) equations, which means that N-S equations are divided into their average and fluctuating components of velocity, pressure and other scalar quantities such as energy. Then, these components and the averaged time are introduced in the instantaneous momentum and continuity equations giving rise to the RANS. Reyn-
olds stresses are included and are modeled by Boussinesq hypothesis. Boussinesq hypothesis uses one or two additional equations to solve the turbulent viscosity and was used to calculate the eddy viscosity.

The Spalart-Allmaras model for the resolution of the turbulence was used. It is a model of one equation that solves transport equations with the modified form of the turbulent kinematic viscosity. The transport variable \( \tilde{\nu} \) is identical to the kinematic viscosity except in turbulent regions close to the wall. The transport equation for \( \tilde{\nu} \) is

\[
\frac{\partial}{\partial t} (\rho \tilde{\nu}) + \frac{\partial}{\partial x_i} (\rho \tilde{\nu} u_i) = G_\nu + \frac{1}{\sigma_\nu} \left[ \frac{\partial}{\partial x_i} \left( \mu + \rho \tilde{\nu} \frac{\partial \tilde{\nu}}{\partial x_j} \right) \right] + C_{b2} \left( \frac{\partial \tilde{\nu}}{\partial x_j} \right)^2 - Y_\nu + S_\nu
\]

where \( Y_\nu \) is the destruction of turbulent viscosity that occurs in the near-wall region due to wall blocking and viscous damping, \( \sigma_\tilde{\nu} \) and \( C_{b2} \) are constants and \( \nu \) is the molecular kinematic viscosity. \( S_\nu \) is a user-defined source term. \( G_\nu \) is the production of turbulent viscosity and is modeled as

\[
G_\nu = C_{b1} \rho S_\nu
\]

where

\[
\tilde{S} = S + \frac{\tilde{\nu}}{d^2 \tilde{\nu}} f_{\nu^2}
\]

and

\[
f_{\nu^2} = 1 - \frac{X}{1 + X f_{\nu^2}}
\]

where

\[
X = \frac{\tilde{\nu}}{\nu}
\]

\( C_{b1} \) and \( k \) are constants, \( d \) the distance from the wall, and \( S \) is a scalar measured from the strain tensor and based on the magnitude of vorticity.

If the mesh is fine enough to resolve the sublayer dominated by viscosity, wall shear stress is obtained from the laminar stress-strain relationship:

\[
\frac{u}{u_*} = \frac{\rho u_* y}{\mu}
\]

Otherwise, if the mesh is coarse, it is assumed that the centroid of the cell adjacent to the wall falls within the logarithmic region of the boundary layer and the law of the wall is used:

\[
\frac{u}{u_*} = \frac{1}{k} \ln \left( \frac{\rho u_* y}{\mu} \right)
\]

where \( u \) is the velocity parallel to the wall, \( y \) is the distance from the wall, \( u_* \) is the shear velocity, \( k = 0.4187 \) is the von Karman constant and \( E = 9,793 \).

Turbulent heat transport equation is modeled using the concept of the Reynolds analogy to turbulent momentum transfer. The energy equation is modeled as follows:

\[
\frac{\partial}{\partial t} (\rho E) + \frac{\partial}{\partial x_i} [u_i (\rho E + p)] =
\]

\[
\frac{\partial}{\partial x_j} \left[ k + \frac{c_p \mu_t}{Pr_t} \frac{\partial T}{\partial x_j} + u_i (\tau_{ij})_{eff} \right] + S_h
\]

where \( k \) is the thermal conductivity, \( E \) is the total energy and \( (\tau_{ij})_{eff} \) is the deviatoric stress tensor defined as:

\[
(\tau_{ij})_{eff} = \mu_{eff} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \mu_{eff} \frac{\partial u_i}{\partial x_j} \delta_{ij}
\]

This equation represents the viscous heating.

To solve the heat transfer, a CHT problem where take place the conduction and the convection was solved. Heat transfer by conduction is represented by Fourier’s law:

\[
q = -\lambda \nabla T
\]

where \( \lambda \) is the conduction coefficient of the material [W/mK] and \( q \) is the heat flow [W/m²]. The term \( \nabla T \) indicates the temperature gradient in the three Cartesian directions. However, transient and three dimensional heat conduction includes terms such as enthalpy, so that the energy equation for solid regions that Fluent solves is as follow:
\[ \frac{\partial}{\partial t}(\rho h) + \nabla \cdot (\bar{v}p H) = \nabla \cdot (k\nabla T) + S_h \]

where \( \rho \) is the density, \( k \) is the thermal conductivity, \( T \) is the temperature, \( S_h \) is the volumetric heat source and \( H \) is the enthalpy calculated as \( \int T_{ref} \cdot \rho_p dT \). The \( T_{ref} \) value is 298.15 K. Furthermore, the convection heat transfer is represented by Newton’s Law of Cooling:

\[ q = hA(T_s - T_f) \]

where \( h \) is the heat transfer coefficient [W/m²K], \( T_s \) is the surface temperature and \( T_f \) is the fluid temperature.

Once temperature distribution was obtained, thermal stresses were calculated through FEA. Thermal stresses occur when there are constraints against the thermal elongation of a material; if the material is not constrained thermal stresses are present [6]. Then thermal stresses for a structural material are proportional to the material properties (elastic modulus \( E \), the coefficient of thermal expansion (\( \alpha \)) and a temperature difference (\( \Delta T \)):

\[ \sigma = E\alpha\Delta T \]

Equivalent stresses or Von Mises stresses are shown in the results because they are used to predict the yield strength of a ductile material. In terms of stress, Von Mises’ yield criterion takes the form:

\[ \sigma_y = \left[ \frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2} \right]^{1/2} \]

**COMPUTATIONAL MODEL**

In this work, part of the MGT geometry (Figure 1) designed by Martinez [7] was taken to carry out the numerical calculations. The geometry consists of a single nozzle passage and a single rotor blade passage (Figure 2).

The meshes were created in Gambit software. In order to carry out a mesh convergence analysis four meshes with different number of elements were created. The coarsest mesh contains 464,009 elements while the finest one contains 1,487,226 elements. The meshes were unstructured conformed of tetrahedral elements (top of Figure 2). The Grid Convergence Index (GCI) was calculated with the equations proposed by Roache P. [12] and used by Castro et al. [2]. With a mesh of 786,088 elements was obtained a GCI value less than 1%, hence, that one was chosen for carrying out the thermal and the structural calculations. From that mesh 109,283 elements correspond to the solids domain and 644,769 elements correspond to the fluid domain.

**Temperature distribution calculation**

The temperature distribution and the thermal stresses were calculated with CFD at the time points indicated in the start-up curve of Figure 3. The start-up curve comprises from cold start until the MGT reaches its rated speed at 240 seconds. Velocity and mass flow were increased linearly as in the start-up curves reported by Quinkertz et al. [11], Hernandez et al. [3], Kosman et al [5] and Wilk [14].
The working flow was treated as compressible. The flow was air at the temperature of 1035 K and a pressure of 2 atm (once reached the nominal speed). Multiple Reference Frame (MRF) model and Least Square cell-based method with discretization schemes of second order were used. Spalart-Allmaras turbulence model and PISO algorithm were used. Residuals were assigned to $1 \times 10^{-3}$ except for the energy which was assigned to $1 \times 10^{-6}$.

The CFD computational domain consists of a single flow passage corresponding to the stage of the turbine (bottom of Figure 2). On the side walls of the flow passage periodicity condition was used. It was developed an interface between the nozzle and the rotor sections in order to obtain a single passage. At the input and the output of the passage, mass flow and pressure were assigned as boundary condition, respectively.

The material of the MGT was set as NIMONIC 105 whose properties are shown in Table 1. This material is used to manufacture gas turbines blades, its melting temperature is 1573 K and its yield strength limit is 853 MPa. Thermal conditions at the walls were assigned as coupled.

<table>
<thead>
<tr>
<th>NIMONIC 105 alloy properties (at 600 °C)</th>
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<tbody>
<tr>
<td>Young’s modulus ($E$)</td>
</tr>
<tr>
<td>Density ($\rho$)</td>
</tr>
<tr>
<td>Specific heat ($C_p$)</td>
</tr>
<tr>
<td>Poisson’s ratio (\nu)</td>
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<tr>
<td>Thermal expansion coefficient ($\alpha$)</td>
</tr>
<tr>
<td>Thermal conductivity ($k$)</td>
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</table>

Thermal stresses calculation

Thermal stresses calculations were carried out with a Finite Element Analysis (FEA) using ANSYS Mechanical APDL software. The meshes used in the FEA are shown in Figure 4. Both meshes were comprised by tetrahedral elements. The model corresponding to the nozzle comprises 57,836 elements meanwhile the model of the rotor blade consists of 51,447 elements.

Figure 5 shows the contours of temperature through the MGT passage with a flat cut on a blade radial...
height of 50% for some stationary time points corresponding to the start-up curve of Figure 3.

The MGT temperature was increased linearly during the start-up curve. Rotor and nozzle started to heat up on the leading and the trailing edges, respectively, as is shown in Figure 5. The nozzle trailing edge was heated faster than the rest of the same, whereas in the case of the blade, it was further heated at the leading edge. In the nozzle the coldest section was located in the center while the one in the rotor was located on the trailing edge. The maximum temperatures of the nozzle and the rotor were equivalent to 1005 K and 935 K, respectively. Stabilization of the temperatures was reached at a simulated time of 300 seconds.

Temperature differences

As already mentioned, thermal stresses change in magnitude as much as the temperature differences change. These differences are bigger during the start-up cycle and consequently the largest thermal stresses are obtained. For this reason temperature differences behavior and their effects on the thermal stresses were analyzed. Figure 6 shows the behavior of the temperature differences during the start-up cycle on the nozzle and on the rotor.

Temperature differences \( \Delta T \) were obtained by taking to the highest and the lowest temperatures at each stationary time point, as follow:

\[
\Delta T = (T_{\text{max}} - T_{\text{min}})
\]

Temperature differences increased rapidly until they reached their highest values at a simulated time of one and ten seconds for the nozzle and the rotor, respectively. In the nozzle the largest difference was equivalent to 141.58 K while the one in the rotor was of 182 K. Once the nominal speed was reached, nozzle and rotor were subjected to a temperature difference of 17.4 K and 115.38 K, respectively.

Thermal stresses

The highest thermal stresses were obtained at a simulation time of one and ten seconds in the nozzle and the in rotor, respectively, and they agreed with the highest temperature differences obtained. In the nozzle, a thermal stress value equivalent to 239.48 MPa was obtained while in the rotor it was of 791.85 MPa as Figure 7 shows. Once the start-up cycle ended and the nominal speed was reached, the stabilization period was achieved so that the nozzle was subjected to a stress of 68 MPa and the rotor at 212 MPa. Thermal stresses start-up curves behavior agree qualitatively with the curves reported by Nagpure et al., Nowak et al., Quinkertz et al. and Song et al. [9, 10, 11, 13].

![Figure 7. Maximum thermal stresses behavior during the start-up.](image-url)

Figure 8 shows the location of the highest thermal stresses contours on the nozzle with the representation of the Von Mises stresses. The stresses on the pressure side were larger than at the suction side. The maximum stress of 239.48 MPa was located on the trailing edge near to the base. There was a coincidence with the location of the highest temperature. High thermal stress values were developed on leading edge base too.
Figure 9 shows the location of highest thermal stresses contours in the rotor. Stresses on the base are larger than at the tip. The maximum stress of 791.85 MPa was generated on the leading edge base coinciding with the highest temperature location.

Furthermore, the behavior and the magnitude of thermal stresses at different locations of the MGT were analyzed. Figure 10 and Table 2 show the localization of each point and the meaning of each acronym used.

For the nozzle, the NTET point corresponds to the maximum stresses and it was located on the trailing edge base. It reached a maximum stress value at a time of one second, after; it was reduced and stabilized at 22 MPa as is shown in Figure 11. Meanwhile, the NLEH point was located to determine the thermal stresses behavior on the leading edge. Thermal stresses increased during the first few seconds of the start-up and then they were reduced and stabilized at 9 MPa. There was a similar behavior of stresses on NTEH point. It was stabilized at 4 MPa. Hence, thermal stresses behaviors along the nozzle were similar but with different magnitudes as Figure 11 shows and they were higher on the trailing edge base and on the leading edge base.

In the case of the rotor, the RLEB point was located on the leading edge base where the highest stresses occurred. As is shown in Figure 12, the maximum value was obtained at a time of 10 seconds and once the nominal speed was reached it was stabilized at a stress value of 200 MPa. It was result of highest temperature difference. The RTEH was subjected at increment and decrement variations until it reached its highest value of 97 MPa once the nominal speed was reached. Moreover, the RLEH point and the RTET point located on the leading edge and on the trailing edge, respectively, were raised slightly obtaining a similar behavior between them. After that,
they were stabilized at stresses values of 57 and 31 MPa respectively. Results showed that the highest stresses were obtained on the blade base while the ones on the tip were smaller.

**CONCLUSIONS**

In order to predict if the material is suitable to support the high thermal stresses, the calculations of temperature fields and thermal stresses is required. Thermal stresses greatly depend on the temperature fields at the turbine components. The present paper calculated temperature fields and stress distributions on a MGT blades and nozzle which is under design process. Calculations were carried out using the material and the turbine start-up cycle simulation. The maximum temperature and the temperature stabilization were reached one minute after the start-up cycle finished. The highest temperatures occur on the nozzle trailing edge and on the stagnations point of the blade leading edge. The maximum temperature value achieved was equivalent to 1005 K. This value was far away of the material melting temperature.

On the other hand, the highest temperature differences and the highest thermal stresses were detected at the times of one and ten seconds for the nozzle and for the rotor, respectively. The highest thermal stresses were located on the nozzle trailing edge and on the rotor leading edge, and they rose in the constrained zones. The highest thermal stress value reached in the turbine was equivalent to 791 MPa. This value was under the yield strength limit of the material. In summary, the material and the start-up curve proposed are suitable for the MGT operation.

**REFERENCES**


