Structural and Aerodynamical Parametric Study of Truss-Core Gas Turbine Rotor Blade

Dulat Akzhigitov1, Tamerlan Symbetov2, Abilkhairkhan Aldabergen3, Christos Spitas4

1 Department of Mechanical and Aerospace Engineering, Nazarbayev University, 53 Kabanbay Batyr Ave., Nur-Sultan, 01000, Kazakhstan, Email: dulat.akzhigitov@nu.edu.kz
2 Department of Mechanical and Aerospace Engineering, Nazarbayev University, 53 Kabanbay Batyr Ave., Nur-Sultan, 01000, Kazakhstan, Email: tamerlan.symbetov@nu.edu.kz
3 Department of Mechanical and Aerospace Engineering, Nazarbayev University, 53 Kabanbay Batyr Ave., Nur-Sultan, 01000, Kazakhstan, Email: abilkhairkhan.aldehyben@gmail.com
4 Department of Mechanical and Aerospace Engineering, Nazarbayev University, 53 Kabanbay Batyr Ave., Nur-Sultan, 01000, Kazakhstan, Email: christos.spitas@nu.edu.kz

Received October 2020; Revised December 27 2020; Accepted for publication December 27 2020.
Corresponding author: T. Symbetov (tamerlan.symbetov@nu.edu.kz)
© 2020 Published by Shahid Chamran University of Ahvaz

Abstract. Improvement of turbine blades is currently the prime area of research dedicated to the development of more efficient gas turbines. This study examines the structural performance of the gas turbine rotor and stator blades with the implementation of Kagome truss-core structure as inner topology. The truss-core structure was hypothesised to improve the stress behaviour of the blade by reducing the mass and, hence, the centrifugal force induced by rotation, while remaining robust enough to withstand bending stress induced by the flow. In order to analyse the stress state of the truss-core model, fluid flow analysis of transonic turbomachinery was performed via the Frozen Rotor technique in ANSYS CFX and then coupled with ANSYS Mechanical. As a result, the combined surface load of the rotor was obtained and used to estimate the structural performance. By examining the obtained complex stress state of the rotor blades, the truss-core density-dependent structural performance was derived for the given initial and boundary conditions.

Keywords: Gas turbine, Rotor blade, Kagome truss core, Structural analysis, Numerical simulation.

1. Introduction

Beginning in the 20th century, the development of gas turbine technology made continuous improvements in various aspects such as outer design, inner topology, materials, aerodynamics and structural, among others. Generally, operational conditions expose a blade to interactions with pressurized gases during combustion process as well as being rotated at high speeds [1]; therefore, the blade is to be designed mechanical and thermal stress-resistant. As it is known, associated high temperatures considerably decrease mechanical characteristics of blades [2], e.g. fatigue and creep resistance is reduced [3].

Historically, the first gas turbines were limited to 800 °C operational temperature [4]. After this limit was increased to roughly 1250 °C for industrial use and to a little smaller than 1600 °C for military use at the end of the 20th century [5]. Contemporary turbines work with up to 1700 °C in the combustor outlet and 90% efficiency [6]. This leap took place due to the development of superalloys capable of coping with both thermal and structural stresses as well as having high creep resistance [7]; structural stresses and creep of gas turbine blades are mainly associated with centrifugal forces and high rotational speeds. Most common components of superalloys are nickel and titanium [8]. Ni-based alloys have advantages in creep resistance, as well as other mechanical properties depending on a manufacturing process used [9-11]. Meanwhile, Ti-based alloys have advantages in strength-to-weight ratio, thermal and corrosion resistance and metallurgical stability [12]. In terms of conventional materials for gas turbine blades, steel was always in demand due to its low cost, good manufacturability, sufficient mechanical properties and thermal, wear, corrosion and oxidation resistance [13], although its higher density leads to increase in centrifugal tensile forces, which make it less applicable to conventional high speed rotating machinery. Properly designed metal-matrix composite materials are also known to have superior properties than their base alloys in terms of strength, toughness and creep, extending the utility of the aforementioned solutions [14] and even allowing otherwise less suited alloys, such as aluminium, to be of use [15]. The choice can be further enriched with the emergence of new types of stiff materials exemplified by shape-memory structures proposed by Yu et al. [16].

Apart from the change of materials, alterations and studies of blades design have taken place as well; an eminent inner topology study was the analysis of a hollow blade concept in the 1970s [17]. Inner voids were designed in a way to improve inner heat transfer to increase mechanical and heat stresses resistance [18]. Furthermore, these improvements increased strength-to-weight ratio, which in its turn decreased the centrifugal forces, i.e. the major source of structural stresses. The study of Quinn and Picard showed graphically the effect of a rotor blade mass alteration on a mechanical performance [19]. Among prospective designs of the inner blade geometry, the truss-core sandwich structure was the best candidate due to having not only all of the abovementioned advantages but also high tensile stiffness [20-21]. Specifically, Kagome truss-core structure is of interest due to...
its higher softening and plastic buckling impedance than that of other structures [22]. A similar study was done by Sierra et al. [23] regarding the dependencies between cooling rate and holes’ positions and between heat transfer and mechanical stress in the blade surface.

The main objectives were to perform a numerical study the ways of gas turbine rotor blades’ performance enhancement through the use of ASTM type 310 stainless steel and implementation of Kagome truss-core as inner topology. These design alterations were hypothesised to decrease structural stresses, blade surface strain and increase blades’ strength-to-weight ratio, stiffness and mechanical stress resistance. The numerical study was based on the use of finite element solvers (ANSYS CFX and ANSYS Mechanical). The structural parametric analysis of truss core density was also performed in order to derive an optimal truss-core configuration. Regarding a stator blade’ structure, its inner design was not changed in the scope of this work.

2. Methodology

Finite element solvers used in this paper were based on the ANSYS Workbench platform, namely CFX and Mechanical, which includes a Static Structural solver. The latter was used to simulate centrifugal and mechanical loads on a central rotor blade. The simulations were inter-dependent on the basis of 1-way fluid-structure interaction (FSI): heated flow analysis provided data of pressure distribution with respect to rotor blade surface, and then mechanical loads were deducted from resultant pressure distribution. In order to attain physically justified results with reduced computational power, frozen rotor technique was used during CFX analysis.

The studied rotor and stator blades’ models were based on that in a transonic turbine stage [24]. The fluid box for the corresponding 3D model was built using the TurboGrid in-built tool. The choice of this gas turbine model was supported by the availability of experimental and numerical results from a work of Lastiwka [25]. The benchmark considered in this work is the original solid blade model. The rotor model was designed to contain Kagome truss-core inner structure with uniform truss diameter, seen in Fig. 1. The trusses were evenly distributed with respect to a surface perpendicular to an airfoil suction side with their centres coinciding with the mean camber line. The first and the second matrix numbers define the number of truss-cores along the mean camber line and along the principal edges, respectively.

In the next step, design Degrees of Freedom (dDOF) had to be stated in order to define the scope of the study. For proposed rotor blade design, the number of Kagome-type truss cores, their diameter, airfoil span and material had to be identified. The geometry for the rotor blade and its material properties can be seen in Table 1, and these properties are shared between the benchmark and the proposed truss-core model. The truss-core rotor blades have a shell thickness of 1.5 mm. The centres of Kagome trusses were distributed along the camber line at equal distance from each other. Considering these terms, the single dDOF for the parametric study was chosen to be the truss volume ratio, \( V_r \), of the blade, i.e. the ratio of the model’s inner volume to the benchmark aside from the shell volume. The ratio \( V_r \) was controlled by the number of the trusses inside the rotor, ranging from 18 to 66 trusses, and variation of the diameter of each truss, ranging from 1mm to 3mm. The above description summarized in Table 2.

Fig. 1. Proposed design of the truss-core gas turbine rotor blade model with 18 (6×3 matrix) Kagome truss elements.

<table>
<thead>
<tr>
<th>Geometric Parameter</th>
<th>Value</th>
<th>Material Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotor chord [mm]</td>
<td>55.9</td>
<td>Grade 310 stainless steel [26]</td>
<td></td>
</tr>
<tr>
<td>Rotor axial chord [mm]</td>
<td>46.8</td>
<td>Density [kg.m(^{-3})]</td>
<td>8000</td>
</tr>
<tr>
<td>Rotor span [mm]</td>
<td>66.5</td>
<td>Poisson’s Ratio</td>
<td>0.3</td>
</tr>
<tr>
<td>Rotor tip clearance/span</td>
<td>1.4%</td>
<td>Shear Modulus [GPa]</td>
<td>77</td>
</tr>
<tr>
<td>Stage diameter [mm]</td>
<td>800</td>
<td>Tensile Strength [MPa]</td>
<td>520</td>
</tr>
<tr>
<td>Rotor to stator ratio:</td>
<td>2.3</td>
<td>Yield Strength [MPa]</td>
<td>245</td>
</tr>
</tbody>
</table>

Table 2. Table of values of \( V_r \), depending on the inner truss topology of the turbine blade.

<table>
<thead>
<tr>
<th>Number of trusses</th>
<th>d=1 mm</th>
<th>d=1.5 mm</th>
<th>d=2 mm</th>
<th>d=3 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>18</td>
<td>0.016</td>
<td>0.035</td>
<td>0.064</td>
<td>0.100</td>
</tr>
<tr>
<td>30</td>
<td>0.027</td>
<td>0.060</td>
<td>0.107</td>
<td>0.197</td>
</tr>
<tr>
<td>42</td>
<td>0.037</td>
<td>0.084</td>
<td>0.150</td>
<td>0.293</td>
</tr>
<tr>
<td>54</td>
<td>0.048</td>
<td>0.108</td>
<td>0.193</td>
<td>0.389</td>
</tr>
<tr>
<td>66</td>
<td>0.059</td>
<td>0.132</td>
<td>0.235</td>
<td>0.486</td>
</tr>
</tbody>
</table>

Hollow blade
Benchmark (solid)
Table 3. Initial and Boundary conditions.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet total temperature [K]</td>
<td>402.7</td>
</tr>
<tr>
<td>Outlet static pressure [bar]</td>
<td>0.9</td>
</tr>
<tr>
<td>Inlet mass flow rate [kg/s]</td>
<td>22</td>
</tr>
<tr>
<td>Rotational Speed [rpm]</td>
<td>3000</td>
</tr>
</tbody>
</table>

2.1. Modelling

As it was mentioned before, for the fluid analysis, the special ANSYS feature, TurboGrid, was used. The feature provides the opportunity to automatically set up the fluid box and structured mesh around the blades. The generated fluid boxes around the stator and rotor blades account the periodicity of the analysis of flow in the gas turbine, thus allowing to simulate the flow through the whole gas turbine by considering only two stator and three rotor blades. To further increase the accuracy of the analysis, the clearance between the blades and the shroud walls is accounted for in the model. This allows modelling the shroud wall in the rotor fluid domain as stationary, which is the case in real gas turbine machines. Figure 2 shows a sample model for stator and rotor configuration.

The choice of the correct turbulent model is critical for the accuracy of fluid dynamics analysis. Thus, two turbulence models, k-ε and SST were considered for the calculation. Both numerical models were accurately described by Zeidi and Mahdi [27]. Based on the preliminary study, the SST model displayed better convergence and smoother behaviour in comparison with the k-ε model. The residuals of k-ε were seen to pass only 10-5 threshold accuracy value, and SST model residuals were seen to pass the value of 10-6. Therefore, this study employed SST as its turbulence model [28].

2.2. Initial and Boundary Conditions

Initial and boundary conditions for outer flow analysis were set according to the work of Jiang, Zheng, Dong, Yue and Gao [29]; this work contains experimental data as well, which was used for further model validation. In order to simultaneously decrease computational power and receive credible data for further structural analysis in ANSYS Mechanical, supporting tools were needed to substantiate the change of a rotational speed of the turbine from an original value of 10617 rpm to a lower value. The gas turbine total efficiency \(\eta\) was used for this purpose and Eq. (1) shows the procedure to compute the corresponding value based on the methodology in the work of Sheldrake [30]. As a result, the value of \(\eta\) for the modelled turbine had to be equal to that for the original turbine to verify the correctness of our numerical model. Other implemented initial and boundary conditions were not changed and can be found in Table 3. Figure 2 graphically shows the boundary conditions for the gas turbine blade.

\[
\eta = \frac{T_{0,\text{in}} - T_{0,\text{out}}}{T_{0,\text{in}} \left(1 - \frac{P_{0,\text{out}}}{P_{0,\text{in}}}ight)^{\gamma-1}}
\]

2.3. Static Structural Conditions

The results of the CFX analysis were coupled to the static structural solver via the ANSYS Workbench framework. The resulting pressure distribution on the rotor blade’s surface was imported into the structural simulations as initial preload condition. Because the preload could be imported separately from the body of the blade, the rotor could be discretized to meshes differently from the fluid solver. This was used to optimize the speed of static structural simulations by decreasing the number of meshes through mesh verification. As for other boundary conditions, the end of the rotor blade connecting to the rest of the turbine was considered as a Fixed Support. The centrifugal force was applied in the form of a proportional imposed rotational velocity, ranging from 10,000 rpm to 18,600 rpm. The pressure field was imported separately for each case and was dependent on the Reynolds number similarity [31].

2.4. Discretisation and Mesh Verification Parameters

Meshing for the fluid dynamic simulations in CFX was performed using adaptive mesh settings of the ANSYS Turbomesh package. The produced mesh was checked for connectivity and geometric suitability until every cell met appropriate requirements, namely connectivity number of 12, an element volume ratio of 20, maximum and minimum face angles of 165° and 15° respectively. The boundary layer was based on \(y^+\) offset calculated for referenced Reynolds number. At each mesh iteration, the produced model was examined during simulation to satisfy the maximal limit of residuals of 10-5. Eventually, the produced number of mesh elements for the simulation of flow over the studied gas turbine rotor was 3.9x10^6, and 1x10^6 for the flow over the stator blades.
As was mentioned in section 2.3, the result of aerodynamic pressure distribution was imported into structural solver as an initial preload separately from the blade’s mesh, so that the rotor could be discretized anew. This was used to optimize the structural simulations’ accuracy and computational power requirement by additional mesh verification. As a result, several reiterations of structural analysis had been performed to achieve a necessary convergence of simulation solutions, which was investigated through the maximal results of equivalent Von-Mises elastic strain $\varepsilon_v$ and normal elastic strain $\varepsilon_n$ in the direction of the rotor’s span for the case of the blade with 18 truss elements inside. As can be seen in Fig. 4, the necessary convergence was reached when the number of mesh elements reached 1,336,535. Therefore, further calculations were performed according to respective mesh settings to achieve the appropriate results.

3. Results

3.1. Fluid Dynamics Analysis

There are two main functions of fluid dynamics analysis: (1) to validate the use of a correctly established and industrially applicable gas turbine model, and (2) to produce results for subsequent structural analysis of Kagome truss-core structure implemented in the rotor blade.

First, the pressure distribution across the gas turbine section shows that the pressure decreases across both stator and rotor, as seen in Fig. 5. At the first glance, the results comply with expectations: stator section of the turbine increases the velocity of the flow and reduce its pressure, while the rotor section of turbine converts flow energy to mechanical energy, thus reducing both velocity and pressure. These trends comply with those reported by Wacks, Nakhchi and Rahmati [32].

The contour of pressure on the rotor surface is another descriptive parameter. The rotor rotates the turbine by creating lift induced by the pressure difference across its suction and pressure surfaces. Figure 6 presents the pressure contours for both turbulence models. As expected, the pressure at the top surface (suction side) is lower than that of the pressure side, and the pressure concentration point agrees with the literature. The pressure values at these points are nearly equal to the total pressure, meaning that these points are stagnation points.
To validate the results of the frozen rotor turbine analysis, the comparison of the turbine efficiency was made as discussed in the Methodology section using Eq. (1). This value was compared with the original experimental paper by Lastiwka [25] because the rotor and stator profiles of that work are similar to the ones used in the study. By using equation 1, the results for the simulations is $\eta = 0.847$. Comparing with the value obtained by the referenced experimental research, the divergence is 3.5%, which is a comparably valid result.

3.2. Structural Analysis

As described in the Methodology section, structural analysis was based on a 1-way FSI procedure. The input values were the (1) pressure on the rotor blade surface from CFX simulation resultant pressure distribution and (2) centrifugal forces from centripetal acceleration, which was calculated using the reference turbine rotation value. The resultant Von-Mises stresses can be viewed in Fig. 8.
An important fact observable from the stress distributions visible in Fig. 8 is that the stress distribution is not uniform across the turbine blade. The highest stress values are located on the interface between the airfoil and the blade root, and the stress gradually dissipates in the direction of the blade’s edge. According to simulation results and figures, there is a consistent point of stress concentration in the trailing edge of the blade connected to the hub, i.e. the fixed support. This point attributes to the concentration of the tension and bending stress that form from the fluid pressure and rotation of the turbine respectively.

Fig. 8. Equivalent stresses generated on a rotor blade surface based on ANSYS CFX SST model results.

Fig. 9. Combined total stress values for different truss element diameters at 10000 rpm.

Fig. 10. Max. stress values $\sigma_{v,\max}$ for different proposed designs with a varying of rotational speed.
3.3. Parametric study

Based on parametric studies, the graphs of the structural trends were plotted. Following is the discussion of visible parametric dependencies based on the observation of the graphs.

Results indicate that the hollow inner topology of a rotor blade has the highest deformation and induced mechanical stresses. Comparison with other simulation models reveals that the hollow benchmark model possesses the least desirable mechanical characteristics in the study. This model might have higher cooling effectiveness, thus showing better heat strength at high temperatures than the solid (massive) benchmark model. However, this aspect is not studied in the scope of this work.

Figure 9 is aimed to illustrate the maximum combined stress values, \( \sigma_{\text{v,max}} \), depending on the variation of the truss element diameter at 10000 rpm. Three diameters of a truss were analysed: (a) 1 mm, (b) 2 mm and (c) 3 mm. Based on stress values for each truss diameter case, it can be noticed that the use of a truss element diameter of 1 mm results in the highest stress values. Therefore, a further comparison was made between truss diameters of 2 and 3 mm. It is also worthy of notice that as a truss diameter becomes larger, the model becomes closer to the benchmark, i.e. a massive gas turbine blade.

Figure 10 is the combination of all separately retrieved results on proposed designs by comparing them with the benchmark of a massive solid rotor blade. The main finding is that the proposed designs are becoming advantageous with an increase of rotational speed, and subsequently, centrifugal acceleration. Therefore, the further increase in the speed value is significantly \( \sigma_{\text{v,max}} \) thereby providing better stress-resistance compared to the benchmark. The graph in Fig. 10 graphically shows this trend with \( y \)-axis corresponds to a value relative to a result of a massive solid benchmark. To add, the implementation of Kagome truss-core structure increased the performance and decreased the bulk mass of transonic gas turbine rotor blades. That means the truss elements localize the stress deformations inside the rotor blade structure, hence protecting the surface and theoretically increasing the lifetime.

4. Conclusions

In spite of its relative maturity, gas turbine technology is open to further improvements. One of these improvements were analysed in this work: Implementation of Kagome truss-core structure as an inner topology for rotor blades. The operating gas turbine was modeled in 3D and then simulated using ANSYS CFX solver and 1-way FSI analysis. Based on prior research, experimental and numerical results were available in the literature for comparison, which was used to validate the fluid dynamics model. The results of the numerical study indicate that the reduction in the number of truss-core elements and the corresponding increase in the void ratio resulted in reduced maximum stresses on the rotor blade surface. This effect becomes more noticeable with an increased rotational speed of the gas turbine. Among the studied design configurations for the particular turbine blade geometry and operating conditions, the optimal number of truss cores was found to be 18, while the gains compared to the solid and hollow benchmark designs were shown to vary with rotating speed. Therefore, the implementation of Kagome truss-core structure was proved to increase the strength of gas turbine rotor blades, together with a decrease in total mass, and no consequent drawbacks. Improved cooling properties of the proposed design may further allow the use of otherwise unconventional materials, such as stainless steel, which due to their density are currently not used in gas turbine blades in favour of Nickel superalloys.

Author Contributions

D. Akzhigitov built the 3D models and developed the simulation setup for the structural analysis; T. Srymbetov prepared the manuscript and made the literature review; A. Aldbergen developed the simulation setup for the aerodynamics analysis; C. Spitas initiated the project and suggested the original design of inner rotor blade topology. All authors discussed the results, reviewed, and approved the final version of the manuscript.

Conflict of Interest

The authors declared no potential conflicts of interest with respect to the research, authorship, and publication of this article.

Acknowledgements

The authors acknowledge funding support from grants (HYST) 50E2017005 and (UltraSat) 091019CRP2115 by Nazarbayev University.

References

[15] Spitas, V., Besterci, M., Micheli, P., Spitas, C., Shear Testing of Al and Al-Ni-Cu Materials at Elevated Temperatures, High Temperature Materials and

ORCID iD
Dulat Akzhigitov https://orcid.org/0000-0002-1078-9629
Tamerlan Symbetov https://orcid.org/0000-0002-5180-8005
Abilkhairkhan Aldabergen https://orcid.org/0000-0002-2039-3054

© 2020 by the authors. Licensee SCU, Ahvaz, Iran. This article is an open access article distributed under the terms and conditions of the Creative Commons Attribution-NonCommercial 4.0 International (CC BY-NC 4.0) license (http://creativecommons.org/licenses/by-nc/4.0/).