Analysis of Thermomechanical Load of GT Turbine

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Abstract: The published article describes the thermal and mechanical loading analysis of a generator turbine assembly of turboprop aircraft engine. The methodology of determining the critical point with maximum equivalent stress is presented. The input information of the generator turbine stress determination methodology is the loading conditions and the geometry of the assembly. The paper presents a mechanical analysis, where the detailed temperature analysis is already described in the previous article.

Keywords: generator turbine; mechanical analysis turbine loading; turbomachinery.

1 Introduction

The aim of mechanical analysis is to define the inputs and then determine the maximum stress in the component. Since the external time-varying input values are essentially non-stationary, it is necessary to determine a steady response to these inputs. The article therefore deals with the analysis of stress in the steady state of the engine on defined power modes.

The overall load analysis is divided into two steps. In the first step, the response of the system to the thermal load was determined. This response is read as a boundary condition in the stress analysis calculation presented in previous article [1].

For these boundary conditions, the problem of stress field distribution in the GT turbine assembly is solved. The problem assumes the solution of an elastic problem without considering the mechanism of plasticity, which would be calculated using, for example, Neuber's solution presented in [2] or in Nagode research [3].

2 Loadings

The GT turbine assembly is a highly thermally and mechanically stressed part with a whole range of loads, which can be divided into the following categories:

- Temperature loading,
- Rotation (centrifugal loading),
- Aerodynamical forces,
- Contact loading.

2.1 Temperature field

Temperature analysis is used as an input boundary condition for subsequent stress (mechanical) analysis. The load cycle was calculated based on the proposed speed modes theoretically calculated based on the procedures set out in [4].

Steady state is solved by linear temperature analysis in steady state engine operation. This article considers the temperature field, which has already been determined in the article mentioned in [1] and are also described in [5, 6] and [7].

Dividing the task into two steps is for practical reasons. The distribution consists in the type of calculation of the steady response. For each task, it is necessary to set different calculation conditions with regard to networking and other FEM calculation settings.

Furthermore, the possible use of advanced analysis involving the conditions of variable temperature inputs along the blade length, which would be calculated using CFD software and subsequently serve as the input temperature load to the temperature analysis, is also envisaged. The temperature distribution on the components is read from the temperature analysis. In Fig. 1 shows the distribution of the temperature field on a GT turbine assembly.

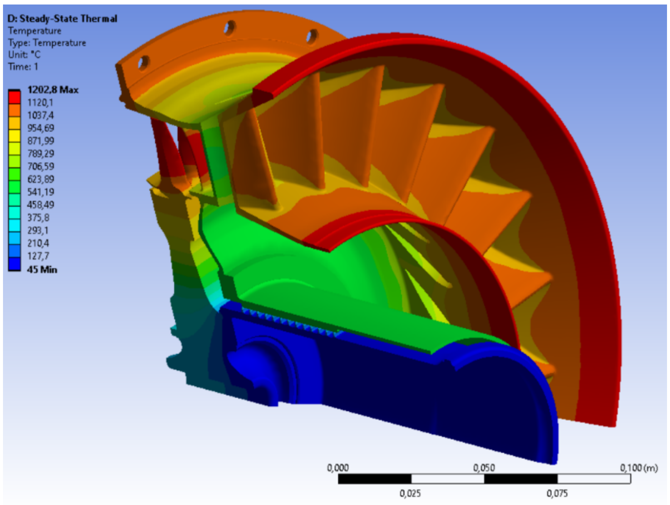

Fig. 1: Temperature field distribution on generator turbine assembly.

2.2 Centrifugal forces

The second significant load is the load from centrifugal forces. The general formula for calculating the centrifugal stresses is given in Eq. 1.

$$\sigma_{od} = r m_r \omega^2, \tag{1}$$

where r is the radius of curvature, m_r is the mass of the centrifugal part as a function of the radius of curvature and ω is the angular velocity for the given flight mode.

The centrifugal force is implemented in the ANSYS calculation program. The input parameters for this calculation are the material density, geometry and considered speed of the GT wheel disc. In Fig. 2 shows the above load. A fixed bearing is the bearing is selected. Axial displacement is ensured by binding D and C. Further, in Fig. 1 shows the speed of the GT disc and shaft.

2.3 Aerodynamical forces

The load from aerodynamic forces is prescribed on the blades of the stator and the rotor. It's based on the assumption set out in [8]. The specific forces are determined based on the relations for the tangential specific load Eq. 2 and axial specific load Eq. 3.

$$q_y \approx \frac{P_\omega}{\omega z} \frac{2}{D_1^2 - D_2^2}, \tag{2}$$

where ω is the power of the turbine stage as a function of speed, z is the number of disc blades and D_1, D_2 are the head and foot diameter of the blade.

$$q_x \approx (p_{1(\omega)} - p_{2(\omega)}) \frac{2\pi D}{z}, \tag{3}$$

where p_1, p_2 are the inlet and outlet pressures of the blade grid stage for the given flight mode, z is the number of disc blades and D is the diameter at which the specific load is calculated.

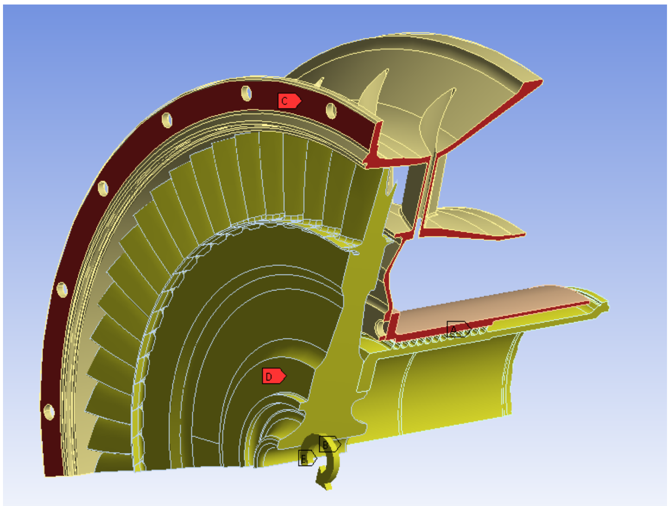

Fig. 2: Centrifugal forces on rotating parts.

In Fig. 3 shows the above load. Further, in Fig. 3 is shown individual components of aerodynamic forces on the blades from the flue gas stream.

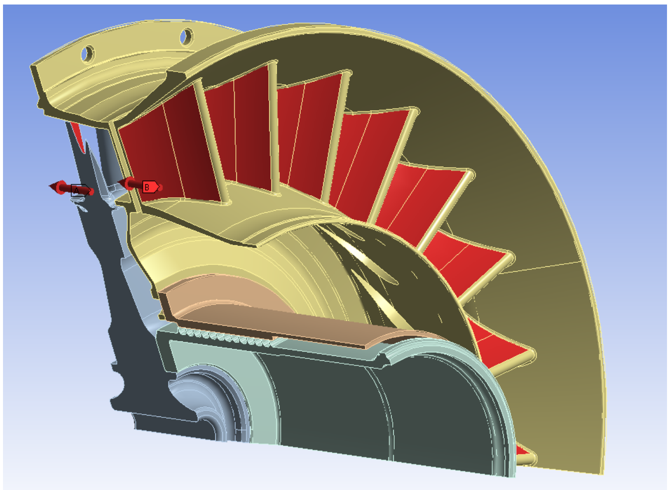

Fig. 3: Aerodynamical forces on stator and rotor blades.

2.4 Contact loading

The generator turbine assembly is simplified for the requirement of mechanical analysis of the system. The shaft, which transmits the torque from the turbine part to the compressor part, is guided by a cut at the bearing location. At this point, the bearing under consideration is then the corresponding bearing in the bearing. Another contact surface is the connection of the stator part in the motor assembly. It is fixed by a structural connection of screws on the outer edge of the stator. Next, a section is made through the stator cover of the shaft.

In the analysis, a system without internal additional stresses is considered. For this reason, there is zero external force on all contact surfaces. The system considers only the internal load of the system. For the analysis of contact surfaces, it is necessary to know all the following structural elements solved within a large system.

3 Boundary conditions

The assumptions for mechanical analysis are based on thermal analysis stated in a previously published article. This article provides a calculation for a steady flight mode at maximum engine speed. The rotor shaft is embedded in the bearing location. The forces were prescribed on the surface according to Fig. 3.

3.1 Contacts

Contact problems are crucial for the choice of bonding conditions. The problem of contacts is their correct definition, which requires sufficient computing power and it is necessary to understand the physics of the problem.

The bonds of the individual parts were created by a combination of contact and kinematic boundary conditions. The contact was selected as the contact between two “Flexible-to-Flexible” bodies. In terms of discretization of contact surfaces, the so-called “Surface-to-Surface” method is chosen for the solution. Due to the time-consuming solution of the contact condition, the contact was defined only on the parts immediately in contact with the target solved part – the turbine wheel.

A so-called “Bonded” bond was chosen for the contact solution, as a small mutual movement of the contact pairs relative to each other is assumed during the solution. The contact is therefore evaluated on the same (small) limited number of nodes, which of course speeds up the calculation itself.

Other contact conditions are “No Separation” and “Frictionless”. The difference between these bonds is the normal direction of the force, where a non-friction bond can be caused to settle.

Contact definitions are prescribed in the following sections:

- GT disc - Rotor shaft (No Separation area),
- GT disc – Stator Case (Frictionless surface),
- Disc Seal – Rotor Shaft (Frictionless surface),
- Disc Seal – Stator Case (No Separation).

In the mentioned ANSYS software, it is possible to specify the areas between which a binding condition is created, and the software itself finds the nearest pairs of nodes to which it prescribes identical DOF. The two surfaces are therefore firmly connected, and the contact algorithm is not solved between them.

3.2 FEM model

Finally, the element model follows the thermal analysis for the given assembly. The following model is chosen for the requirement of mechanical analysis. A necessary prerequisite for correct analysis is the creation of finite-element model mesh. The mesh option is created with respect to the assembly geometry. Defining the meshing correctly and its size is a key step in achieving correct results.

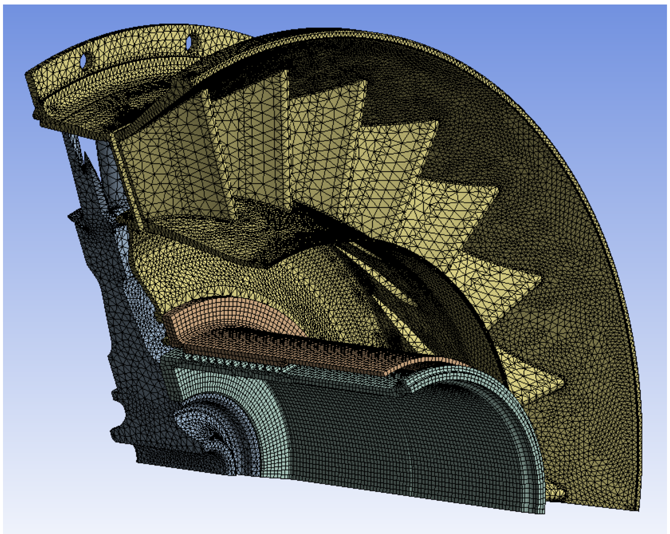

Fig. 4: Mesh of generator turbine assembly.

The mesh was chosen by Automatic elements method due to the rotational strength task in the size according to Fig. 4. As a critical point, a blade corners were assumed, as already mentioned. The mesh has been refined around this section. Excessively refined mesh is also not a suitable solution, so refinement was chosen by the number of elements around the perimeter of the hole. The size of the mesh was different from the mesh size at the edge of the hole due to the computational software that calculates based on the “beam” structure.

3.3 Loading cycle

The course of the load corresponds to steady values and is shown in Fig. 5. It therefore assumes a cycle between zero values at the maximum values of thermal and mechanical load. The resulting load is then a superposition of partial loads.

In Fig. 5 is shown the resulting reduced (Misses) stress distributed in the GT disc assembly. In Fig. 6 represent critical point of the assembly, connection of blade and case.

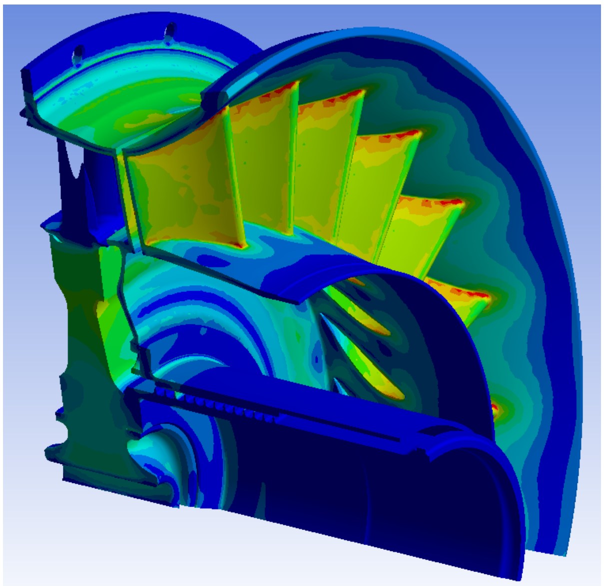

Fig. 5: Equivalent Misses stress distribution.

In the following Fig. 6, the critical point with the largest accumulated reduced stress is indicated. The calculation corresponds to a steady state at maximum speed and maximum temperature.

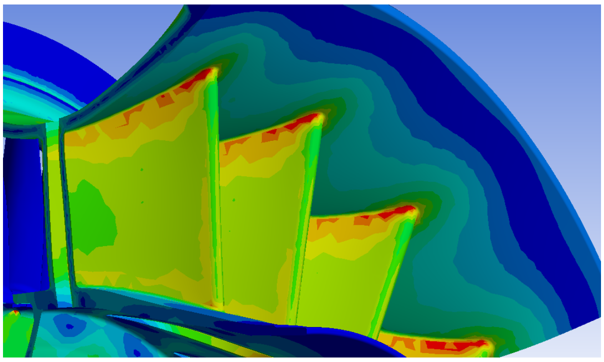

Fig. 6: Maximum equivalent stress zone.

3.4 Preparation of experimental measurement

After a detailed numerical analysis of the generator turbine behavior, in order to proceed with the experimental investigation of the stress and strain field, there is preparing the configuration of the test rig. The turbine engine with the analyzed generator turbine section, tested and studied during the research activity of CASR [9], will be first integrated with additional measuring components and forthcoming experiments. Then will be stator generator turbine instrumented with the strain gauges for the strain measuring in the radial direction. In terms of functionality and operational reasons, the location of the strain gauges will be inside the stator cooling blades and will be taken out of the motor's flow path.

Finally, the temperature field will be measured for the clarification of assumptions presented in previous research [1]. Static temperature on stator blades of the generator turbine section during the experimental tests will be measured with NiCr-NiAl thermocouples type K. After completing the installation and the calibration of all the sensors and of the equipment necessary to conduct the experiments, the experimental tests on the test rig will be carried out.

4 Conclusion

The mechanical analysis consisted in calculating the stress response to a steady state field determined by a linear steady state temperature analysis. Stress analysis considers the transfer of heat from the flue gas and the solution of the contact problem between the parts. The result is a linear elastic stress behavior of the GT wheel assembly. The critical point in the figures was determined - the place with the largest accumulation of reduced Misses stress.

Different temperature distribution has different system stiffness due to temperature-dependent parameters (modulus of elasticity is considered depending on temperature). Due to the distribution and size of this temperature field, the stress field varies depending on the speed modes.

Due to the high temperature, at a stable temperature field, determined by linear temperature analysis, there is a greater redistribution of deformation due to lower stiffness in different parts. The course of temperatures and stresses was marked in the figures.

The area of stator blades and the rotor disc area were determined as critical points from the analysis. The accumulation of the maximum reduced stress occurs at the connection of the blades and the case, due mainly to the different thermal stress and expansion of the blades. The connection area of the labyrinth seal and the case stator could not be analyzed, and a more detailed analysis is necessary in case of fatigue behavior.

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