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THERMAL FATIGUE PROBLEMS OF TURBINE CASING

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Abstract

This paper presents numerical stress analysis of the turbine casing of an aero-engine. To solve the problem, the geometrically complicated numerical model was created. The finite element method was used in computations. In results of nonlinear static analyses performed for both mechanical and thermal load occurred under operating condition of engine, the stress and deformation contours were generated. High thermal stress gradients were found at the region of casing where fatigue cracks were detected during engine operation.

1. INTRODUCTION

The turbine casing, which joins the power turbine section with an exhaust manifold and a transmission box (Fig. 1), is one of many parts of engine in which the complex thermal and mechanical loads occurs. On the internal surface of casing the bearings of power turbine are mounted, thus in this area the dynamic forces derived from turbine shaft are imposed. The second mechanical load is the elevated pressure of the combustions gases acting on the internal surface of the casing. Additionally the non-uniform temperature field contributes to high stress gradient appearing in turbine casing. Moreover, a corrosion environment of hot exhaust gases has also negative influence on its longevity.

Fig. 1. View of a turbine casing for small helicopter turbo-engine.

This paper describes a case of premature fatigue failure of turbine casing of the aero-engine used for drive of PZL-W3 "Sokół" helicopter. In this engine, some problems related with cracks appearance and premature fatigue fracture of the connection between deflectors and outer cylindrical casing were occurred (Fig. 2-4) [4]. Explanation of this problem is not easy on the experimental way. An answer on the question about fracture of casing can gives numerical stress analysis.

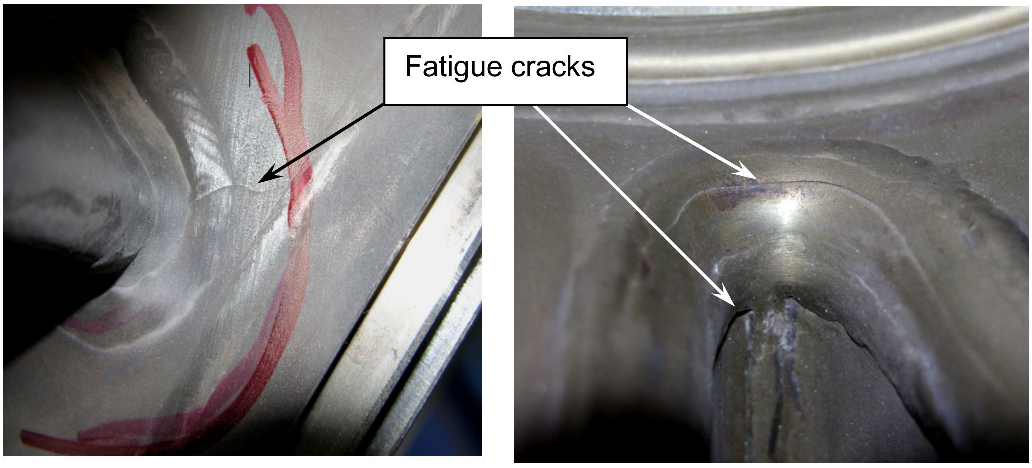


Fig. 2. Fatigue cracks detected under obligatory inspection of engine [4].

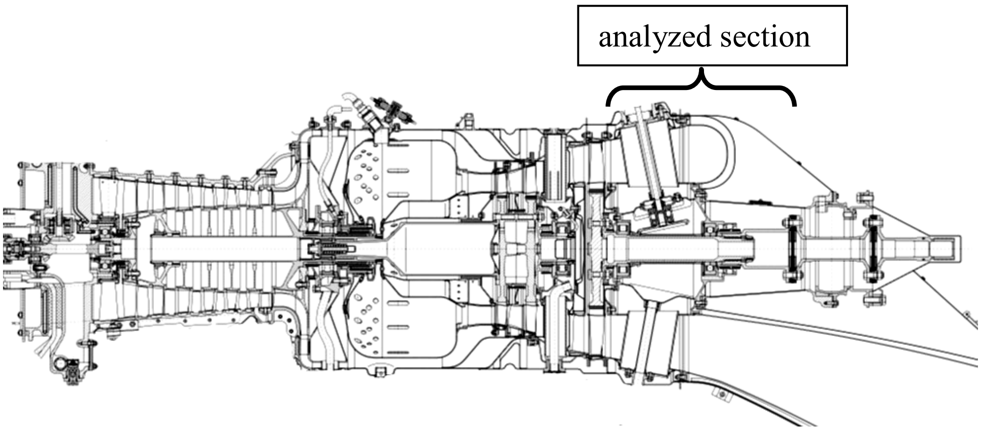


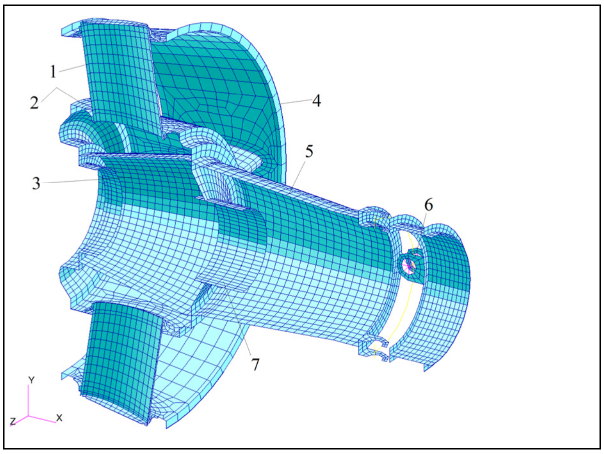
Fig. 3. Cross-section of engine with marked the turbine casing [4].

The component of engine (turbine casing) in which the fatigue cracks were detected is located between power turbine section and the exhaust manifold/transmission box section (Fig.3).

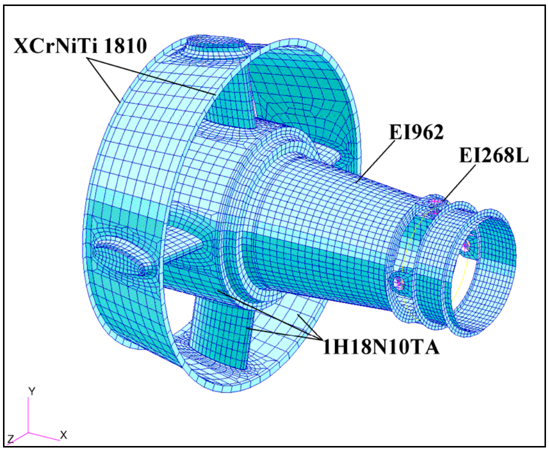
The failure analysis of the aero-engine components has received the attention of several investigations [4-8]. Attention of this work is mainly devoted to explain the reasons of the fatigue fracture of the aero-engine turbine casing.

2. DESCRIPTION OF THE NUMERICAL MODEL

In the numerical model three kinds of finite elements were used. A thin-walled (sheet) components were modeled with use Quad-4 (four node shell) elements. The universal couple and screws were replaced by linear bar elements (Bar-2). The bearings are modeled as explicit multiple point constraints (MPC) elements. The roller bearing was modeled with use MPC elements converged on the bearing casing. The ball bearing is modeled using MPC elements which bind the translation of the center point of the bearing with the bearing casing on the x, y and z coordinates. The model consists of 12889 nodes, 12690 shell elements, 28 bar elements and 20 MPC's (inclusive five on the bearings). A commercial finite element program Patran 2005r2 [2] was used for creation of geometrical model of the casing.



a)



b)

Fig. 4. Longitudinal cross-section of the turbine casing (a) and materials used in different parts of the numerical model (b).

The turbine casing consists of following parts (Fig 4a):

1. Deflector.
2. Cylindrical casing.
3. Inner conical casing.
4. Flange.
5. Rear conical casing.
6. Universal couple joint.
7. Bearing casing.

The turbine casing was made of four different types of alloys which are presented in Fig. 4b. In the numerical model appropriate material constants (i.e. Young modulus in function of temperature) were defined for modeling of mentioned alloys. All materials are defined as linear-elastic.

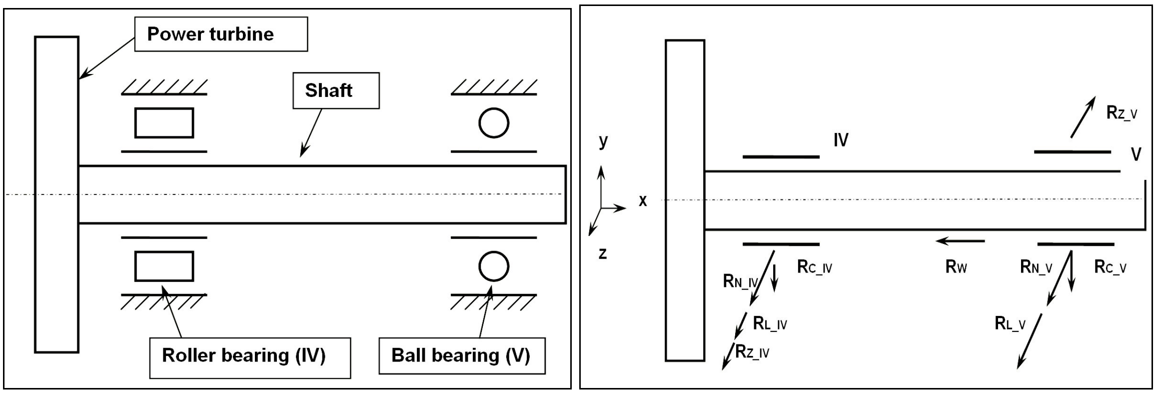
3. MECHANICAL LOADS

The stresses occurring in the turbine casing arise in results both mechanical and the thermal loads. From the mechanical loads, the following forces can be distinguish: weight force (R_C), force related with bearing clearance (R_L), force related with gyroscopic moment (R_Z), longitudinal force (R_W) and imbalance force (R_N). Values of these forces used in computations (Tab. 1) [4], were defined on the base of technical project of the engine. The mechanical load resulting from pressure of the combustions gases acting on internal surface of the casing was neglected because of fact that this load are not meaningful (the pressure of combustion gas after turbine is not high).

The mechanical forces were defined as a continuous load (with use of the MPC elements), imposed to the surface of model where the bearings are located. The forces transferred from shaft (presented in Tab. 1) can appear in different directions in space. These directions are related with different factors i.e. position of helicopter in space (different direction of the gravity and centrifugal acceleration vectors under aerobatics). In full report [4], four load conditions were defined to model of the few basic cases of engine work. For limit of size of this paper, results for only one load condition (Fig. 5b) are presented.

Tabl. 1. Mechanical forces acting on the turbine casing (transferred from shaft).

Force	**R_{IV} [N]**	**R_{V} [N]**
Weight R_C	98.2	49.1
Bearing clearance R_L	982	982
Gyroscopic R_Z	491	491
Longitudinal R_W	-	1178.4
Imbalance/ Geometrical irregularity R_N	687.4	196.4



a) b)

Fig. 5. Location of the bearings inside the casing (a) and forces acting from shaft on the turbine casing for one specified load condition (b).

4. THERMAL LOAD

In this analysis, the thermal load was also considered. The temperature field was defined as a two parametric function in the cylindrical coordinates. The temperature values, obtained from technical project of engine are presented in Tab. 2 and Fig. 6.

Table.2. Thermal load defined for specified points of the numerical model [4].

Units [mm]	Z_1 =-73	Z_2 = 45	Z_3 = 342
R_1 = 44	200 °C	200 °C	120 °C
R_2 = 103	300 °C	300 °C	120 °C
R_3 = 170	120 °C	170 °C	120 °C

Description of specified points (related with coordinates) of the casing (Tab. 2):

R_1 is the outer radius of the bearing casing.
R_2 is the inner radius of the cylindrical casing.
R_3 is the radius of the flange on the outer conical casing.
Z_1 is the coordinate of the flange at the entrance of the turbine housing.
Z_2 is the coordinate of the flange in the rear conical housing.
Z_3 is the coordinate at the rear part of component where the turbine casing is connected to the transmission box.

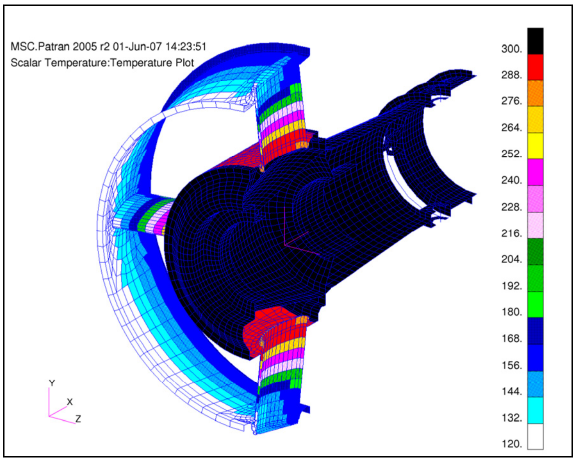


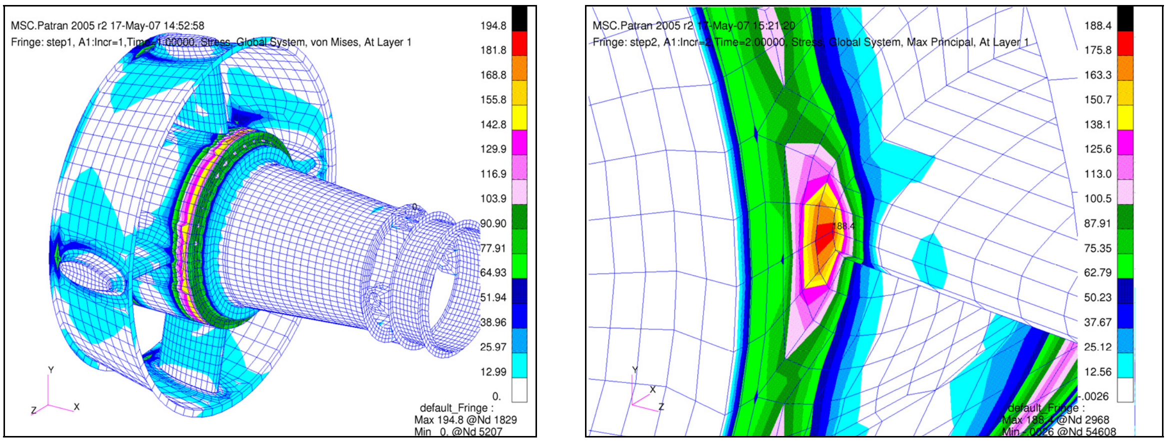
Fig. 6. Temperature field defined for the numerical model of the casing (Celsius scale).

5. RESULTS OF THE NUMERICAL CALCULATIONS

The MARC-2005 [1] program was used for stress analysis of engine casing. The nonlinear (incremental), Newton-Raphson method was applied. For all results, Megapascal (MPa) units were used to describe the fields of stresses.

The Von Mises stress distribution does not show if the material is tension or compressed. Because of this fact, in this paper also the maximum principal stresses were analyzed. This stress is particularly interesting from the point of view of the fatigue strength because just the tensile stresses contribute the most to the fatigue crack initiation and next to crack propagation.

Fig. 7 shows, that the tensile stress in the critical region where the cracks appeared (connection between deflectors and outer cylindrical casing), achieves 188 MPa (Fig. 7). Under loading, the outer cylindrical part of casing is strongly distorted (Fig. 8). By specify two load components it was possible to estimate which load has more impact on the premature fatigue failure of casing. As seen in Fig. 9 and 10, the thermal stresses observed in the model are about two times larger than the mechanical stresses.



a) b)

Fig. 7. Von Mises stress distribution for the casing (a) and maximum principal stress values in the critical region of component (b).

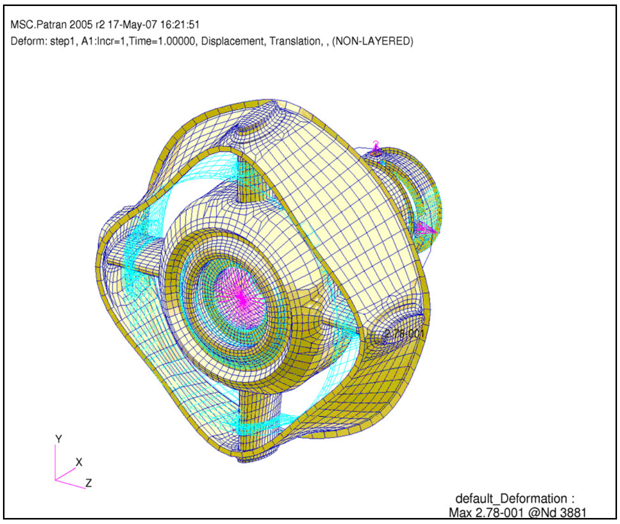


Fig.8. Visualization of the casing deformation (front view, scale of deformation 150:1).

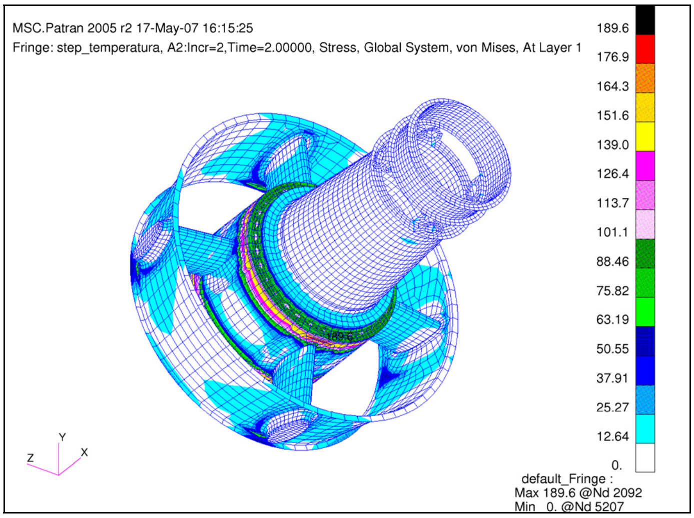


Fig. 9. Von Mises stress resulting from thermal load.

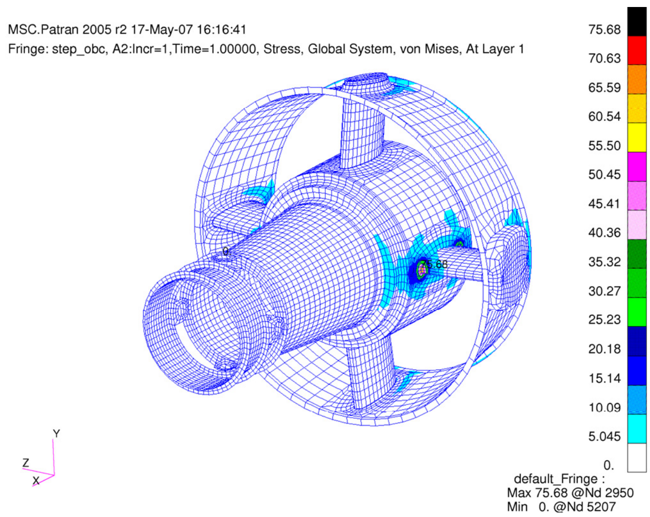


Fig. 10. Von Mises stress as a result of the mechanical load.

6. CONCLUSIONS

In this work results of the stress analysis performed for the turbine casing of the aero-engine were presented. To solve the problem a geometrically complicated numerical model was created. In results of the calculations both stress and deformation contours were obtained. The maximum stress areas overlap the zones on the casing where the first fatigue cracks were detected. By specify two load components (thermal and mechanical), and analysis of numerical results for these separated loads, the complex fatigue failure process of engine casing is better understanding.

During the work preparation the following conclusions can be formulated:

- The stress in the critical zone where the first fatigue crack occurs (Fig. 7) was at about 180-190 MPa. This value is close to the yield stress of the material (220 MPa for temperature 200^{O}C). The cyclic variation of stress in this critical zone was reason for fatigue crack arising.
- The thermal stresses observed in the model were about two times larger than the mechanical stresses (Fig. 9 and 10). Based on this information, the next conclusion is that low cycle thermal fatigue (LCTF) was the reason for premature fatigue failure.
- It can also be concluded that the additional stresses as a result of vibrations of the engine were reason for decreasing the fatigue life of the casing. However, the dynamic analysis was not performed in this work.

The following recommendations to avoid early fatigue cracks appearing can be formulated:

a) Increasing the radius between the deflector and the casing surfaces in the critical region.
b) Weld residual stress may occur under welding. Different heat treatment method can be used in order to reduce the residual stress.
c) The material used in the critical component could be changed. The alloy 1H18N10TA used for this part has a yield stress of 240 MPa (for 0 ^{O}C). It was proposed that the 1H18N10TA alloy can be replaced by Inconel 625 alloy, which has a much higher yield stress (414 MPa) and is often used in aerospace applications.

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