

SOME SELECTED EXAMPLES OF FEM MODELLING OF STRUCTURE-AFFECTING MATERIAL AND MANUFACTURING FAULTS

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Abstract

In design practice, in most of cases, stresses and deformations of structures with nominal dimensions and average material properties were analyzed. It has been concluded that such procedure is most often sufficient. However, there are design solutions, especially in high-effort structures for example aircraft and turbine engines, where scatter of geometrical dimensions consistent with assumed tolerances cause significant scatter of stresses and strains, differ from values coming from nominal dimensions. Scatter of these values has essential influence on service life of presented structures. Example of such effect is described in the paper. There were used FEM analyses to determination of scatter of stresses and stress concentration coefficient in herringbone lock joint of K-15 engine's turbine. Another example of fault impact on stresses in structure is analysis of undercarriage element with material fault. Both these examples indicate on great sensitivity of stress distribution on manufacturing imperfections. The impact of these faults and imperfections reaches hundreds of percent, and in some cases could be serious menace of operating safety. In structures with the multiple 3D stress state, non-linear geometric and material phenomena, with contact between friction-affected elements, any geometric imperfections can cause considerable local changes in the stress and strain.

Keywords: *FEM modelling, combustion engine, turbine engine, landing gear, imperfection*

1. Introduction

Material faults and the scatter of production-induced features prove of exceptional importance while analysing work of various structures when loaded. The reason is that they considerably affect life and reliability of these structures. The presence of material faults and geometric imperfections (tolerance ranges, wear-and-tear effected material losses, etc. – see Fig. 1) is inevitable in any real load-bearing structure. The intended aim of any design engineer is to minimise effects of the presence of such faults called imperfections. The ways to reach this goal are partial elimination thereof and determination of susceptibility of the structures under consideration to the presence of imperfections. Efforts to examine the effects of material and manufacturing faults upon the structure's condition with numerical methods is a narrow-ranged issue and relatively poorly described [1-4, 11]. The Authors are, in fact, precursors of the presented approach used in the design practice [1-8, 10-13]. What have been presented in the paper are some selected examples of how the faults under discussion affect operating characteristics of a structure. Our considerations have been based on the effects of work carried out under the projects on: K-15 turbine engine, aircraft undercarriage, and – under way nowadays – on some other

aeronautical structures (e.g. aircraft brakes). Many of these examples have been analysed numerically and verified experimentally. Findings of both some selected numerical analyses and some experiments are presented further on. Fig. 1 shows due to what factors geometric imperfections in mechanical components occur.

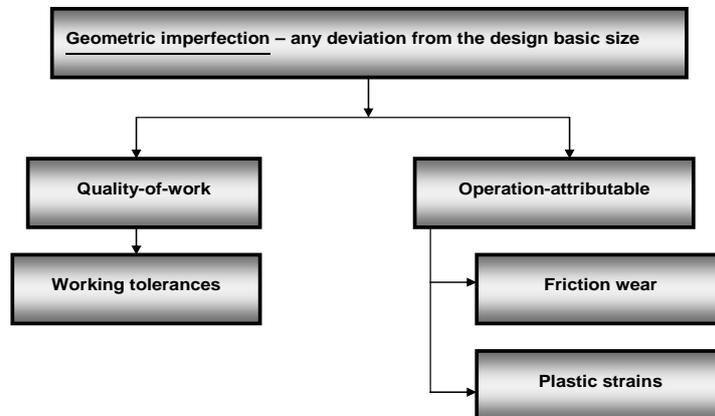


Fig. 1. Causes of geometric imperfections

2. The effect of geometric imperfections upon stress in a herringbone lock joint

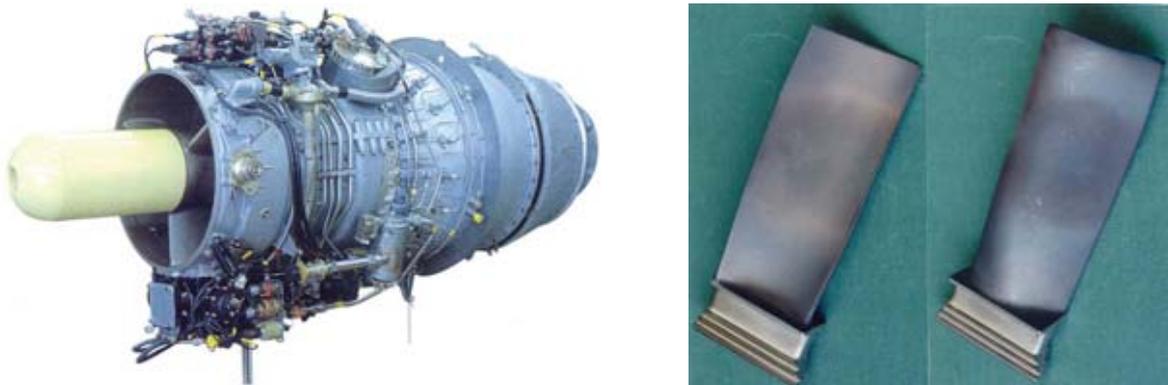


Fig. 2. Turbine engine K-15. To the left: general view. To the right: turbine blades, convex and concave surfaces

While analysing the structure, models of basic sizes are usually examined. Most often such an approach is justified, since the scatter of dimensions is in practice of no significant effect to results of strength analysis. However, this approach cannot be considered ‘an axiom’ (as it used to be quite lately) in the design-engineering practice. Accuracy of work is often of critical importance to the ability to mount mating parts/components and – maybe above all – to stress that is to be found in a given structure. Accuracy of the geometric shape or changes thereto in the course of operation can also, apart from the stress, affect the amounts and directions of loads upon the structural components (e.g. aerodynamic profiles of blades of various rotors or aircraft wings). A herringbone lock joint between the turbine disc and a blade in an aircraft turbine engine is another good example (Fig. 2). The blade root and the turbine-disc lip belong to two different structural components; the mating thereof is contact in nature. Durability of the whole compressor-turbine rotating unit strongly depends on the durability of this joint. The blade root (Fig. 3) is of very complicated herringbone shape. To have the mating of both elements correct, the blade root has to precisely correspond to holes cut out in the turbine disc. The assumed dimensional tolerances, within which the blade root’s cross-section should be contained, have been intended to satisfy this demand.

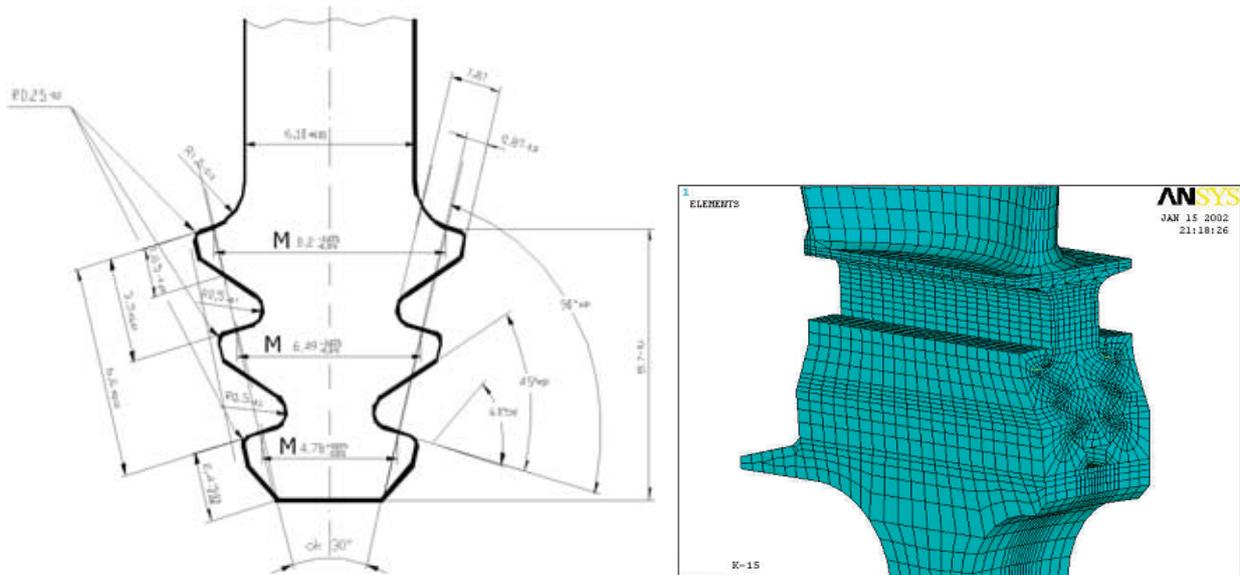


Fig. 3. Cross-section of the blade root and FEM model thereof

Most common inaccuracies (within the assumed tolerances) in the herringbone joint's workmanship are as follows:

- error in the so-called pitch of teeth, i.e. distance between subsequent teeth,
- error in parallelism of contact surfaces (along the blade root's axis),
- error in shape of the contact surfaces,
- error in the form of waviness of the contact surfaces,
- scatter of radii of tooth-space roundings and first-tooth fillet.

The above-mentioned inaccuracies in the workmanship refer to the turbine-disc lip as well.

What occurs after the blade is mounted in the disc is a chance match of tolerated dimensions, which are usually fit according to the normal (Gaussian) distribution. The errors mentioned result in:

- excessive loading of one or two teeth, e.g. if two lower teeth remain in distance given with – positive tolerance, whereas in the disc – with negative tolerance,
- irregular contact between mating surfaces (they deflect from each other by a certain angle),
- irregular contact between mating surfaces, one that consists in a local (point or linear) contact.

Given below are calculations-orientated assumptions [1, 4, 5]:

Calculations were a multi-stage effort. At the first stage, 2_D models were analysed.

- a) Initially, 2_D models were analysed with no account taken of a heat load; then, calculations were made for a model with the effect of temperature taken into account.
- b) At the second stage, 3_D analyses were conducted (Figs 3-7).
- c) Linear and bilinear material models with strain hardening areas of variable properties against temperature.
- d) Accepted were simplified temperature distributions gained from measurements taken on an engine for steady-state conditions reached thermally and mechanically.
- e) Analyses were carried out using nominal values (Fig. 3), and then for some selected boundary values.

The calculations were performed with the following numerical codes: the MSC/Patran, MSC/Nastran, ANSYS and ANDINA.

By the way of example, an instance of three most disadvantageous dimensional matches due to the pitch error for the blade and the turbine disc has been discussed. These are as follows:

1. Excessive load that affects the first pair of teeth, clearance between the contact surfaces of the second and third pairs of teeth is 0.02 mm (Figs 5, 6, 7).

2. Excessive load that affects the second pair of teeth, clearance of 0.02 mm occurs in the first and third pairs of teeth.
3. Excessive load that affects the third pair of teeth, clearances of 0.02 mm occur in the first and second pairs of teeth.

Furthermore, errors in shape of the contact surfaces of teeth were also analysed. They were the „barrel shape” errors along the tooth. It is evident that values of maximum radial forces in contact may vary from nominal values by approximately $\pm 30\%$ (Fig. 8).

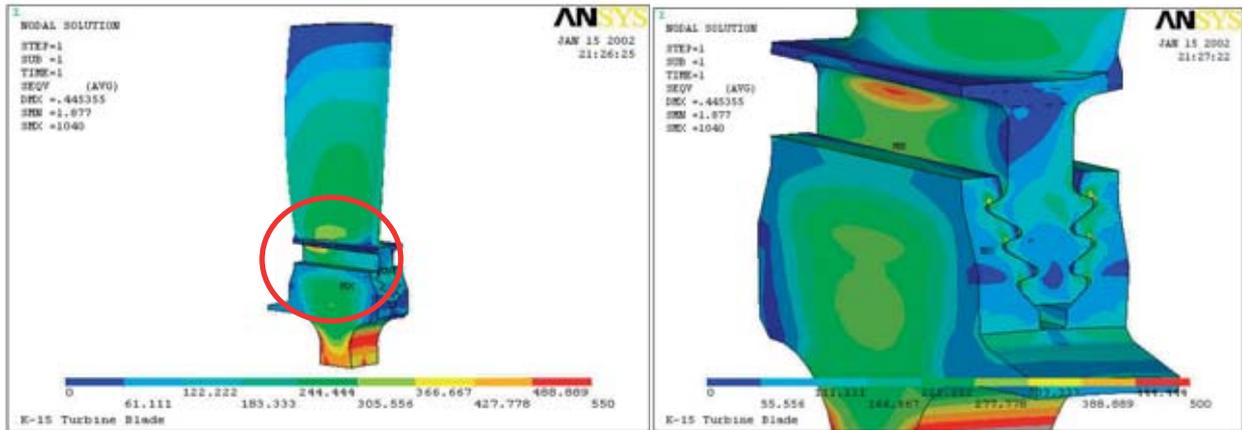


Fig. 4. A model of a lock joint of basic dimensions – stress distributions, boundary conditions show cyclic repeatability

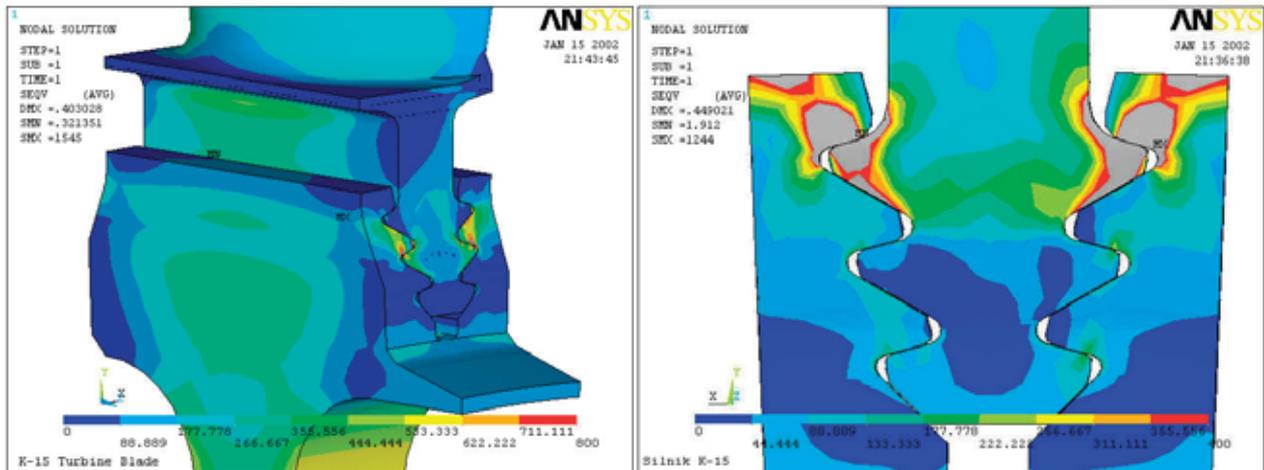


Fig. 5. A model of a lock joint with the first pair of teeth overloaded - cyclic repeatability – stress fringe

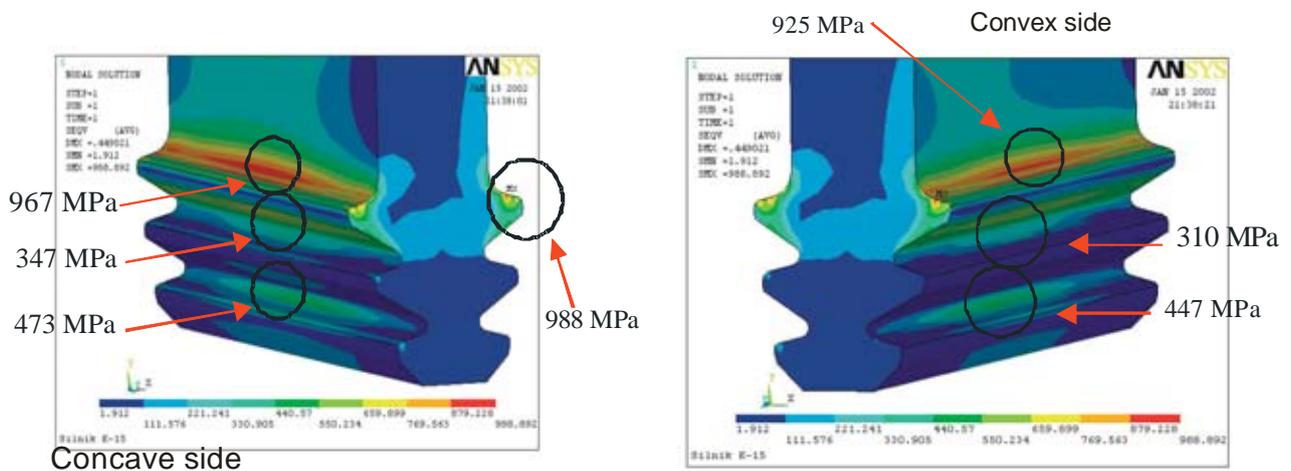


Fig. 6. A model of a blade root with the first pair of teeth overloaded – stress fringe

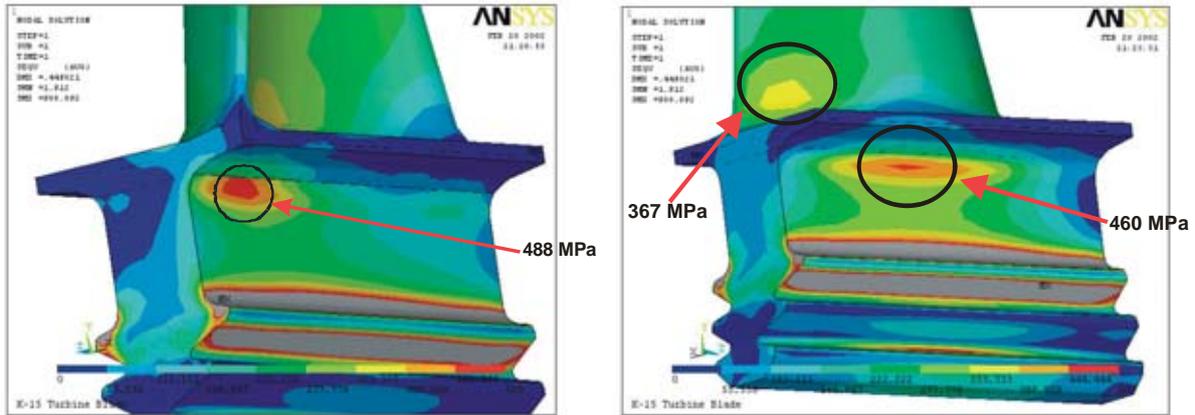


Fig. 7. A model of a blade root with the first pair of teeth overloaded – stress fringe

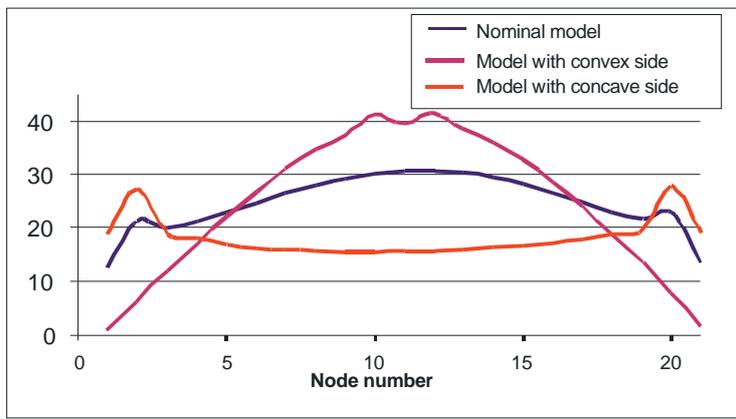


Fig. 8. Radial forces in nodes over the contact surfaces of a tooth along its length. Flat surface – basic model; other ones – convexity and concavity of the surface within tolerance limits

Fig. 9 shows maximum stresses plotted against values of pitch clearances upon two lower teeth. It is evident that in the case under consideration, for possible maximum clearances in pitch of 0.02 mm, values of maximum stresses may vary by more than 100% [5]. This is of crucial importance to fatigue life of the joint.

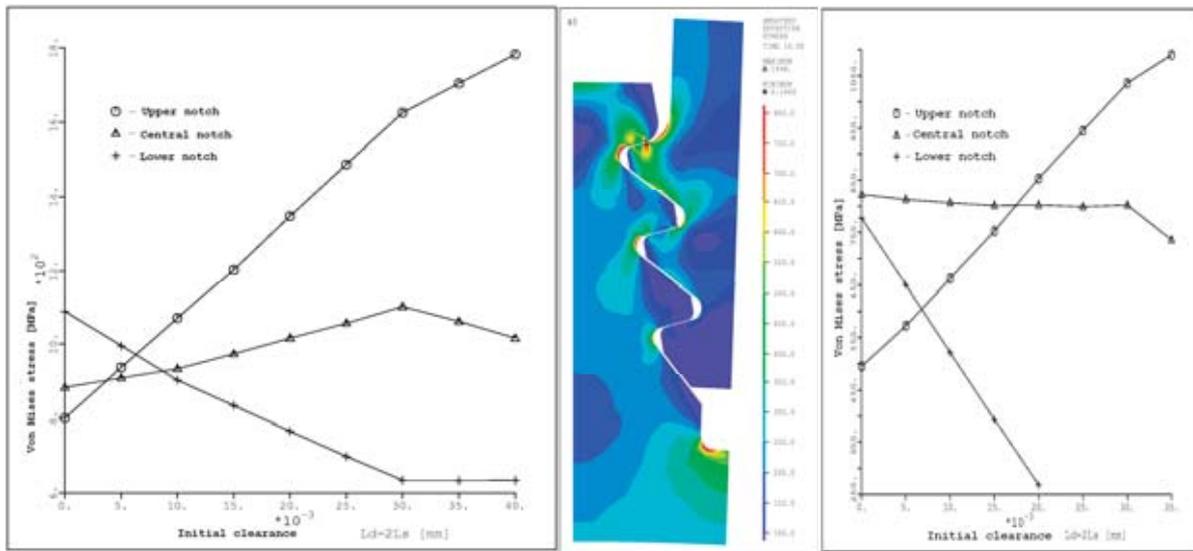


Fig. 9. The effect of initial clearances on lower contact areas upon reduced stresses at the bottom of notches of the turbine-disc lip (to the left) and the blade root (to the right)

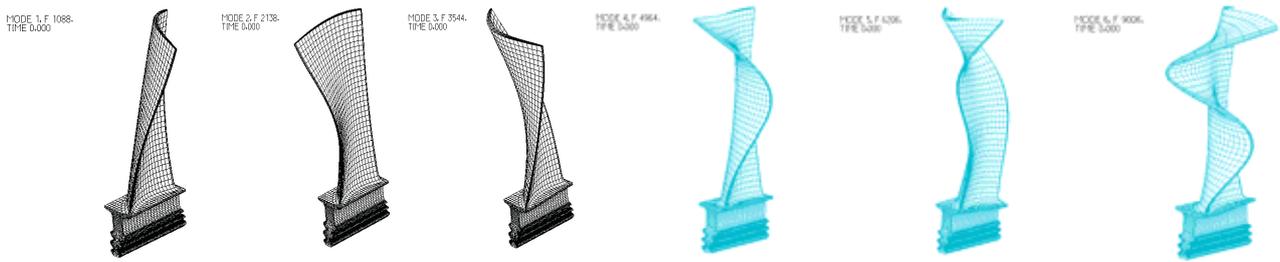


Fig. 10. First six forms of vibration of turbine blades built in the herringbone lock joint, exposed to centrifugal forces (15900 rpm)

Figures 10 and 11 illustrate the effect of imperfections upon frequencies of free vibration of blades fixed in the locking pieces. The former presents six first forms of vibration, whereas the latter-free vibration of turbine blades in the herringbone lock joint (Fig. 2). The blades are within the area of centrifugal forces resulting from the turbine rotating at the rate of 15 900 rpm. Below (Fig. 11), percentage difference between free vibration frequencies of the system ‘lock joint–blade of basic dimensions’, and the system with clearances 0.02 mm upon the first pair of upper teeth (case no. 3). The differences shown refer to first ten frequencies of free vibration. It is evident that the differences may reach the level of 27% of the first frequency.

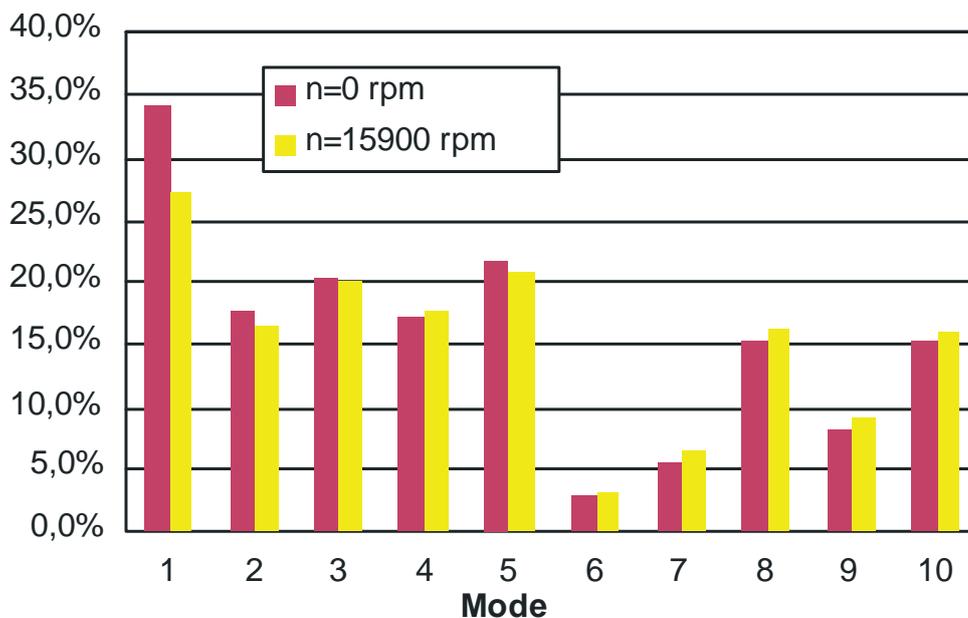


Fig. 11. Percentage difference in free vibration frequency for first ten forms of vibration between the blade of ideal nominal matching and the blade with clearance on the upper teeth

3. The effect of material and production-induced imperfections upon stress

Fractographic examination [9] of fractures allow of identification of imperfections in the weld. The data were used to model imperfections in the local model of the weld [6, 7, 8, 11]. Some imperfections were selected for mapping in the FEM models. These were the following imperfections: material discontinuity (Fig. 13), positional welds, and aligning sleeve (Fig. 12). The sleeve proved to be a significant component that affects work of the weld. It has been intended to axially align two structural components of aircraft undercarriage (Fig. 12) before they are welded. First, it was joined with the upper part of the undercarriage leg by means of positioning welds. The sleeve and the welded joints (13 positioning welds) were mapped then in the local model. The positioning welds were modelled by means of connecting corresponding nodes of the sleeve and

the undercarriage leg. Fractographic examination proved one of these welds was the primary source of fatigue.

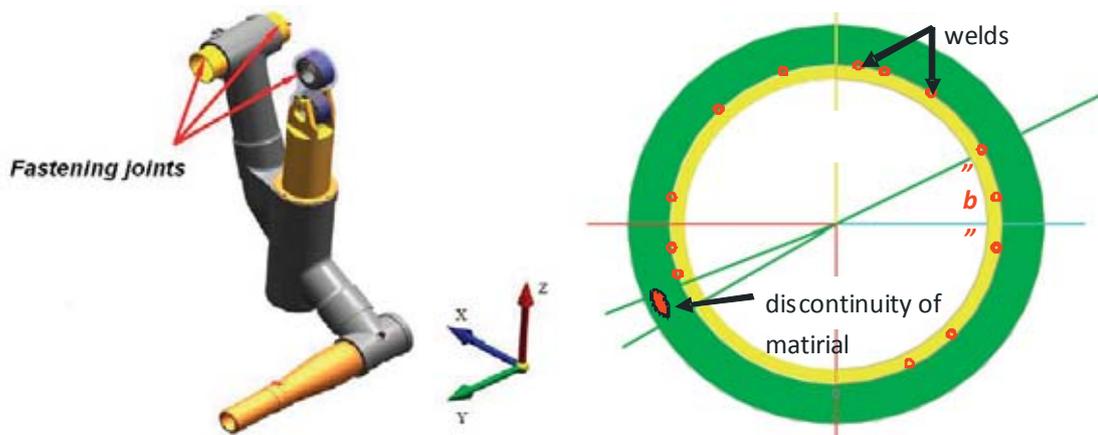


Fig. 12. General view of aircraft undercarriage (to the left). Localizations of positioning welds and material discontinuity (to the right)

The modelled imperfection took a very irregular shape in a real structure (Fig. 13). In the FEM model the area was slightly simplified because of limitations resulting from the software and hardware engaged. Fig. 13 shows – to the right – the shape of this discontinuity assumed in the FEM model. The area was suitably refined in order to obtain more accurate stress distributions. Very high gradients of stresses were expected in the neighbourhood of this discontinuity; therefore, the FEM mesh was refined to reach internodal distances of 0.2 mm. What resulted in the area of interest were 40 elements across the thickness of the wall of the model under analysis. Cubic elements were used in this area. The major part of the model was built of these elements; only areas of changes in the mesh density were modelled with tetrahedral elements.

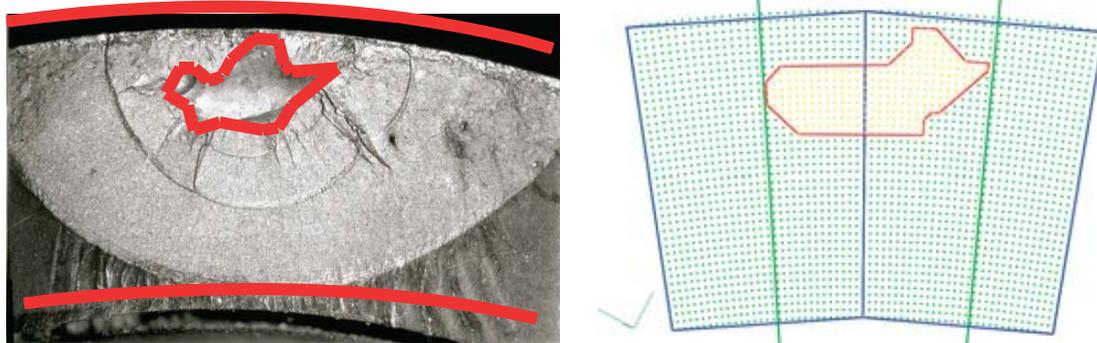


Fig. 13. The area of material discontinuity – its real shape to the left, and to the right – the FEM mapping thereof

FE models reached the number of nodes as high as, approximately, 60 thousand. Greater refinement of elements would require transition to a subsequent level of local modelling. After FEM models were ready, they were subjected to numerous geometrically and materially non-linear analyses. The phenomena of contact and friction were taken into account as well. Every model was analysed using eight (8) instances of loading. While determining boundary conditions for local models, account was also taken of the varying stiffness of the system due to the crack propagation.

Initial FE results from the local models, the effect of imperfections that occur within a given structure upon the operation of undercarriage legs proved significant influence. Comparison between stress distributions gained from the analysis of geometrical and material imperfections with those obtained from the basic (i.e. with no disturbances) model reveals the effect of these imperfections on work of the cross-section. Fig. 14 shows a basic model (to the left); evident are

stress distributions and a model with the aligning sleeve and material discontinuity (to the right). There are local stress concentrations around these disturbances; furthermore, the aligning sleeve together with the positioning welds diametrically changes the stress distributions as compared to the nominal model. On the grounds of this analysis one can determine the effect of the aligning sleeve and positioning welds upon the work of the structure as negative. Locally, more than twice as large increase in the reduced stress levels as compared to the nominal model was found.

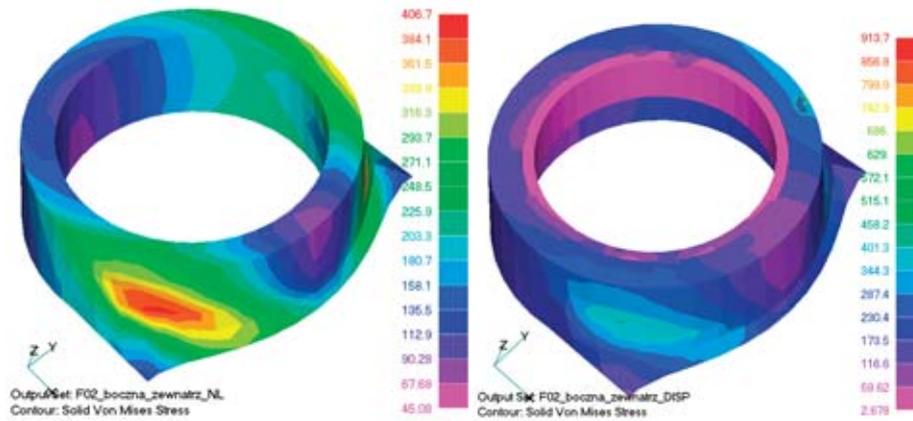


Fig. 14. To the left, the basic weld with no imperfections; to the right, analysis with account taken of the aligning sleeve and material discontinuity

The material discontinuity shown in Fig. 14 has proved extremely dangerous from the strength point of view. Fig. 15 presents results for the region of discontinuity. A white arrow to the left indicates where the discontinuity exists. The hazard that may be provoked by this kind of fault consists in that it is located very close to the surface. That is why this kind of fault is so difficult to find. Results of analyses presented in the left portion of Fig. 15 show of how little effect this discontinuity is to stress distributions on the surface of the undercarriage lever. Only slight disturbances of the stress field are evident, not very much different in values from those throughout the neighbouring area. In spite of that, the reduced stress on the fault-surrounding surface represents value of approximately 550 MPa, just under the surface, at the crack tip, this value increases to more than 1000 MPa (Fig. 16), to finally reach nearly the yield point, i.e. 1400 MPa. The loading with a lateral force to the centre in the area shown in Fig. 15 generates compressive forces; hence, crack initiation should not happen. However, the loading with a lateral force directed outwards as well as the loading with vertical forces result in the spreading of the area of the material discontinuity. Fractographic examination has confirmed that this, in turn, resulted in the initiation of a crack in this area. Figures 15 and 16 present results for the loading with lateral force directed outwards, which causes the spreading of the area with the material discontinuity.

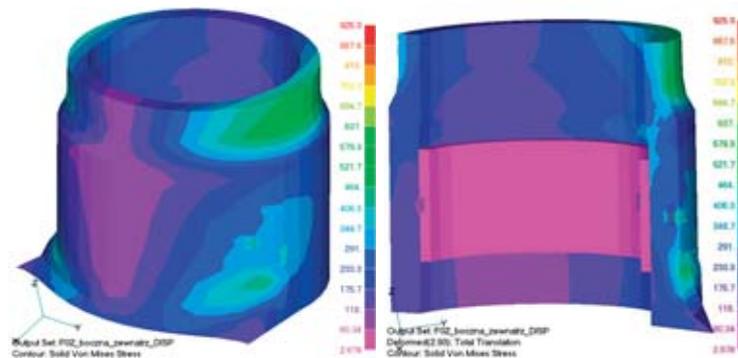


Fig. 15. Charts of reduced stress, the loading with a lateral force directed outwards, a general view of the whole model, and a vertical section through the material discontinuity

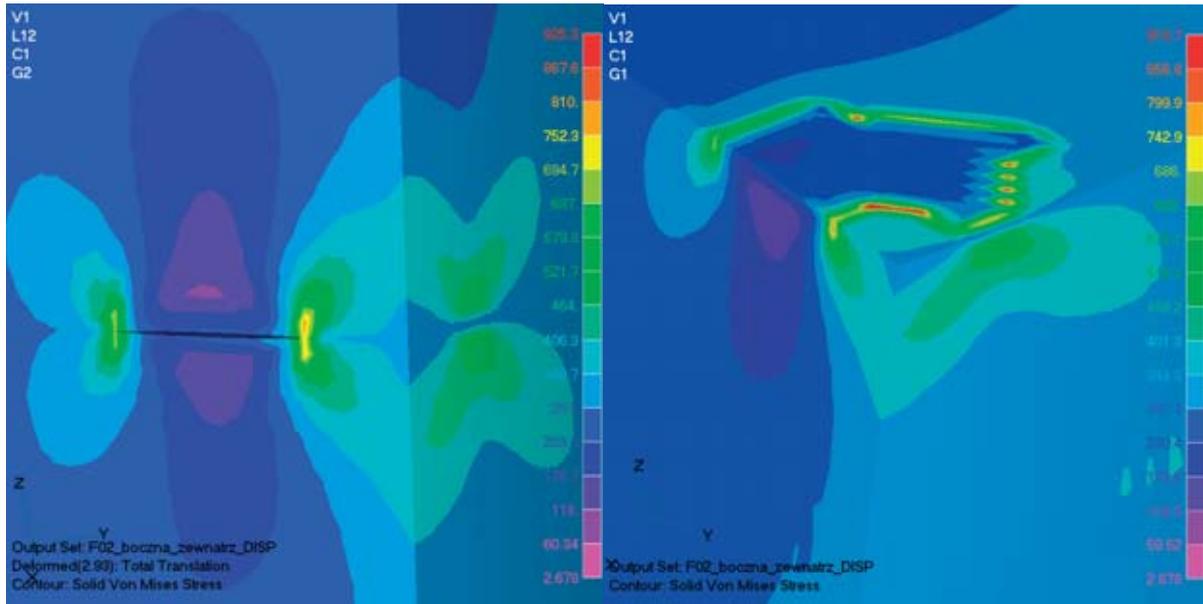


Fig. 16. The magnified area with the material discontinuity shown in Fig. 11; to the left – vertical section; to the right – a $\frac{1}{4}$ sector of the discontinuity, the loading with a lateral force directed outwards

Further two stages were focused on the modelling of the crack growth and propagation. The cracks were assumed to reach 18% and 50% of the basic section area. Another significant part of these analyses was the additional modelling of the contact within the crack. This allowed of considering pressure between the surfaces (the crack closing) when compression of the crack-containing area occurs. The change in the stiffness of a global model due to a failure to the cross-section considerably affected the way the whole system operated. Introduction of a simplified crack to the global model, analysis thereof, and re-determination of boundary conditions of local models proved, therefore, indispensable.

4. Summary

The presented examples prove that in structures with the multiple 3D stress state and numerous others geometrically and materially non-linear phenomena and where contact may occur between friction-affected elements, any geometric imperfections can cause considerable local changes in the stress and strain, and may significantly affect behaviour of the structure throughout the operational phase. As far as strength-related phenomena are concerned, they often become the source of dangerous states of the structure due to, first and foremost, fatigue; also, although not so often, to immediate strength.

Therefore:

- If, due to imperfect introduction, stress in structural notches in the turbine's herringbone lock joint (see Section 2) increases by 100%, and even by 250-300%, the joint may suffer premature low-cycle failure [1, 2, 4, 5]. Similar phenomena may also occur in structural and lock joints of different types, e.g. in pin or riveted joints.
- The wear-and-tear and losses of material as well as plastic strains that occur locally throughout the structure may locally provoke some increase in stress followed by accelerated degradation and fatigue of materials. Examples are numerous, just to mention clearances in joints such as: locking pieces of turbine rotor blades, compressor and fan blades, pin joints (fittings) in aircraft and helicopters, riveted joints, etc. As with imperfections in workmanship, the material's wear and tear resulting in imperfections and losses in surface layers of friction cladding considerably affect operation of brakes, since they both contribute to the origination of the so-called 'spinning spots' [10, 12, 13] and significantly reduce the braking effectiveness.

- In the areas with material imperfections such as foreign-matter inclusions, voids, cracks, or local residual stresses (see: Section 3), local stress concentrations occur, which increase stresses by approximately 100%-200%, and more. They are initiators of fatigue cracking in the structure [6-9, 11].
- While compressing structural components, geometric imperfections provoke – depending on values thereof – the structure's buckling (instability) [3] (e.g. of coatings under compressive loads). Greater values of these imperfections reduce critical loads, what disadvantageously affects work of the structure, since it reduces immediate strength [15].
- Geometric imperfections have their effects upon loads that affect structural components. In the case of mechanisms they may produce additional assembly-induced loads; environmental effects upon the structure while in the air [14] or water may considerably change aerodynamic or hydrodynamic forces (e.g. due to the icing).

To conclude, nowadays the above-presented issue is a matter of major importance that needs thorough studies.

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