OPTIMIZATION OF HYDROGENERATOR POLES FIXATION

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Abstract: Optimization of the vital parts of hydrogenerator takes increasing importance nowadays. Among the other one of the most important is the pole of the rotor. Sizing of its fixation, whether is made in form of "dove-tail", "hammer", or their combination, significantly affects the producing costs, the material price and the overall price of the product. The main goal of this work consists of presenting correct and quality modes dimensioning of the pole fixation and getting to know all involved with the same. Also to point out very important deficiencies that now exists in both design and tender/contractual documentation. All deficiencies are derived from the results of classical methods of stress calculation which has the widest use, thereof in the work is shown comparative analysis of stresses and a way of getting more accurate results and modern methods of calculation (theory of elasticity, photoelasticiometry, the finite element method - FEM) in order to overcome these shortcomings .

Key words: optimization, calculation, poles, hydrogenerator, FEM, fixation

1. INTRODUCTION

Strengthening of gender is one of the most strained parts of the rotor of synchronous Hydrogenerator especially when it comes to fast high-power generators and construction considerations can be done in several forms such as in a dove-tail, hammer, or their combination (Fig. 1 and 2). In the case of extremely high load performance of the work fixation is donne in a several dove-tails, hammers or their combinations.

Scaling fixation is necessary to spend for all regimes of Hydrogenerator of which most important are nominal mode, working and theoretical overspeed engine. Load on pole fixation results from its own gravity, centrifugal force and radial magnetic force of attraction of the stator and rotor.

The results of the calculation of stress depend on the choice methods of calculation and so the criteria of safety can not be unequivocally determined, although in the long-standing practice of using criteria based on the classic method of calculation (method of force reduction, section method, etc.). As the tender/contract documents define the criteria for safety without the associated methods of calculation often leads to misunderstandings between the manufacturer and the customer so that the manufacturer must act strictly according to the contract even though the contractual requirements are meaningful only in the traditional method of sizing. Such discrepancies are often known to cause significant cost hydrogenerator manufacturer.

Besides these shortcomings, there are significant inaccuracies of the methods of calculation, and not taking into account fatigue is often the case in the design, so structural solution is almost always far from optimal.

In this paper is shown the necessity of choosing the finite element method as relevant for good design and the accuracy of calculation and selection methods also represents the first step in the process of optimization.

The results of this study are the first and basic step to redefine the existing safety criteria to establish the minimum cost producers of hydrogenerator. Of course, that the optimum structural solution is closely related to the geometrical parameters of fixation (width, height, radius curves) and their mutual relations and in the paper is shown one of geometry optimization for the "hammer" pole fixation.

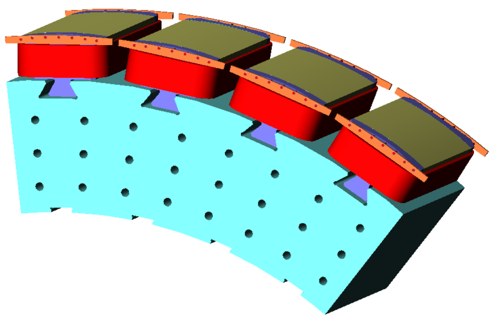

Figure 1. Segment of hydrogenerator rotor showing "dove-tail" form fixation

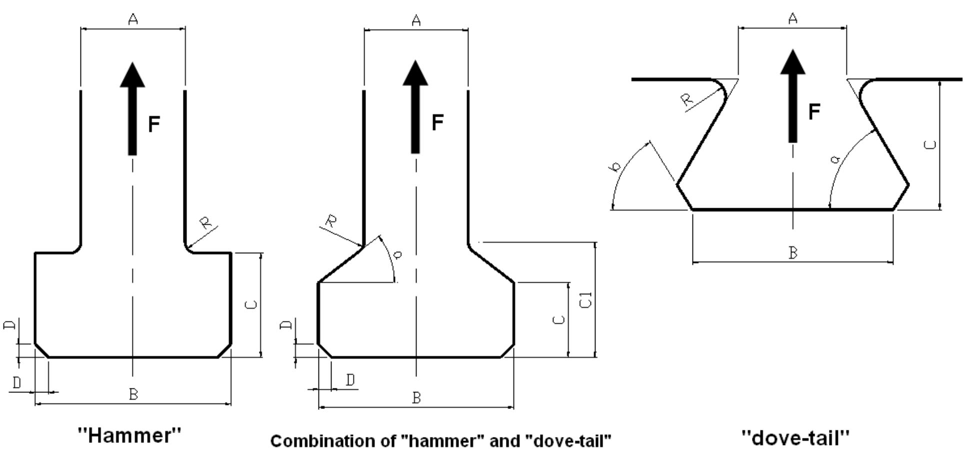

Figure 2. Typical forms of fastening poles

2. METHOD OF CALCULATION

For a realistic view of differences in the results of different methods, a comparative analysis of stresses was carried out in the case of dove-tail shape pole fixation. Below is a list of tags used in this paper, all related to the considered critical section s-s (Fig. 3).

$\sigma_n = \sigma_s + \sigma_t$	ï normal strain (σ_s ï bent, σ_t ï pressure ; KM)
τ_s	ï tangential stress (KM+TR)
σ_x, σ_y	ï normal stresses in a coordinate system xy (KM)
τ_{xy}	ï tangential stress in a coordinate system xy (KM)
$\sigma_1 = \sigma_{11bt}$, $\sigma_2 = \sigma_{22bt}$	ï main stresses (KM)
σ_{11}, σ_{22}	ï main stresses (KM+TR)
σ_{ekvbt}	ï equivalent stress according to the theory of HMH (KM)
σ_{ekv}	ï equivalent stress according to the theory of HMH (KM+TR)
σ_r, σ_φ	ï radial and circular stresses (TE)
$\tau_{r\varphi}$	ï tangential stress (TE)
σ_{xte}, σ_{yte}	ï normal stresses in a coordinate system xy (TE)
τ_{xyte}	ï tangential stress in a coordinate system xy (TE)
σ_{1te}, σ_{2te}	ï main stresses (TE)
σ_{1tr}, σ_{2tr}	ï main stresses (TE+TR)
σ_{ekvte}	ï equivalent stress according to the theory of HMH (TE)
σ_{ekvtr}	ï equivalent stress according to the theory of HMH (TE+TR)
σ_{1eksp}, σ_{2eksp}	ï main stresses (EKSP)
$\sigma_{ekveksp}$	ï equivalent stress according to the theory of HMH (EKSP)
σ_{ekvmke}	ï equivalent stress according to the theory of HMH (MKE)
KM	ï calculation made with classical method
TE	ï calculation made using the theory of elasticity
MKE	ï calculation made by finite element method
EKSP	ï experimental determination of stress photoelasticity
+TR	ï calculation made by considering the influence of friction
HMH	ï energy theory of strength (Huber, Mises, Hencky)

2.1. Analytical calculation - Classical methods

The conventional method of calculating the stresses using the static method (a method of force reduction, method of sections, etc. - Figure 3), and elementary theory of bending of prismatic bars

[5], [6]. Only about geometry of dove-tail, we can provide a significant number of shortcomings to this method. Critical section s-s (or x_0 ï x_1) is defined experimentally as the coefficient of friction on the contact surface. The results are presented for calculating the amount of unit pressure q = 1 MPa, and are shown directly in the intersection of s-s and the transformation of the xy coordinate system. Stresses in the section ss consist of normal σ_n (bent σ_s i pressure σ_t) and tangential τ_s. Inaccuracy of calculation method can be seen in not satisfying boundary conditions because in the transformed coordinate system xy stresses σ_y i τ_{xy} must be zero (for x = x_1) which, however, this theory does not provide. Main stresses σ_1 i σ_2 are also shown in Figure 4.

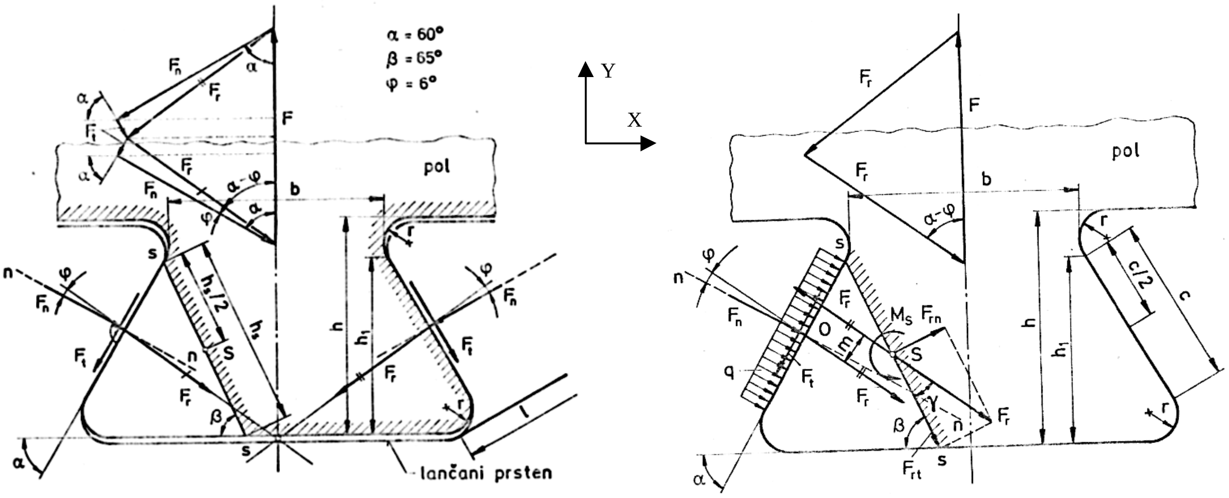

Figure 3. Preview of geometry and load of pole fixation and reduction of forces in critical section

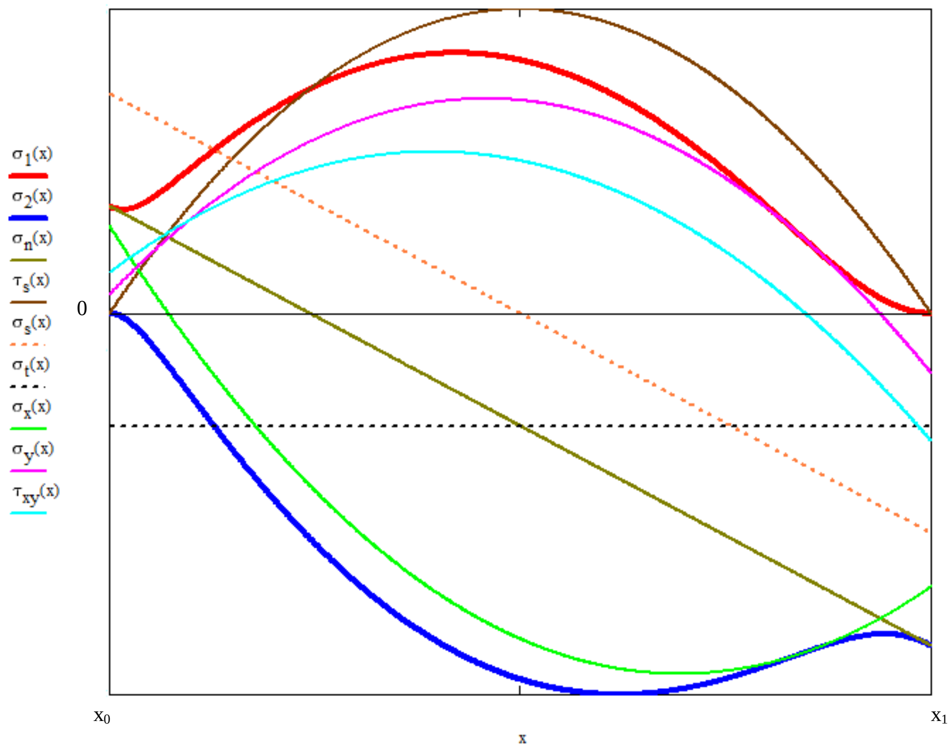

Figure 4. Qualitative view of the stress distribution in the critical section - a classic method

2.2. Analytical calculation - theory of elasticity

Introducing the Airy stress function and the observation of pole root as pin is possible to determine the distribution of stresses in the dove-tail pole fixation. Stresses that arise due to lateral pressure q depends only on the coordinates φ while friction further affects the distribution of radial stress and it is dependent on both the coordinates in the polar coordinate system rφ [1], [5]. The influence of friction is shown in the distribution of principal stresses σ_{1tr} and σ_{2tr}, while all other stress distribution showing the distribution of stresses neglecting the friction on the side of the contact surface. Unlike the conventional method, the theory of elasticity satisfies the boundary conditions on the lower surface of the dove-tail and has thus can be classified in a much more accurate method of calculation [7].

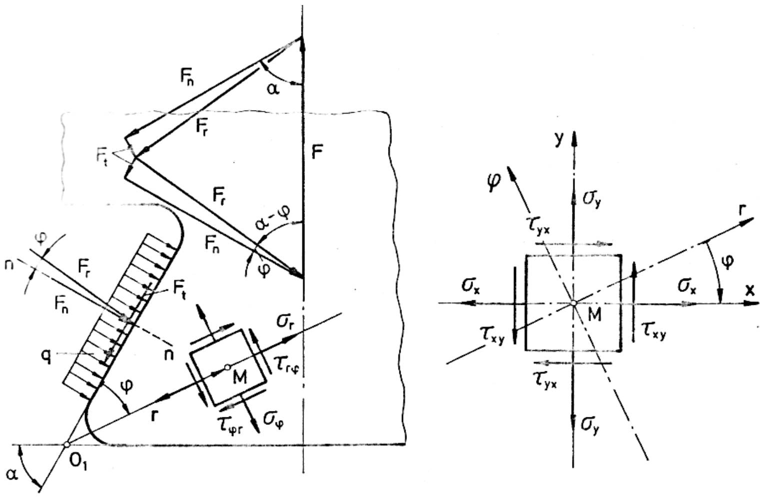

Figure 5. Display of fixation under load in the polar rφ and xy coordinate system

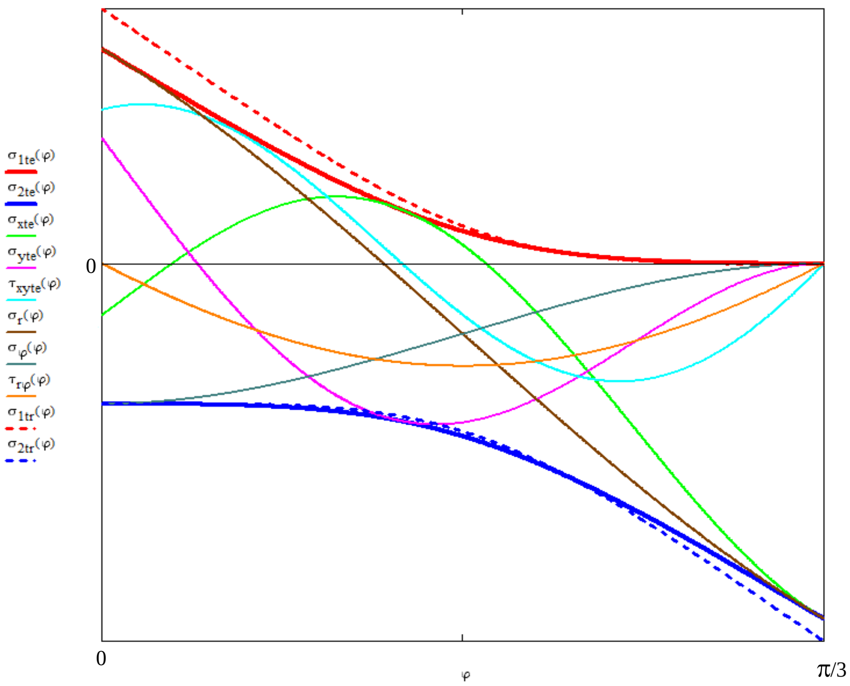

Figure 6. Qualitative view of the stress distribution in the critical section - the theory of elasticity

2.3. The experimental method - Photoelasticity

Until the advent of the finite element method, this experimental method has had great significance in determining the accurate distribution of stresses in the elastic deformation of material. Specifically, the time required for the experiment, high price of the model and the problems that often arise during simulation of the load on the model caused the abandonment of this method by engineers and replacing with more modern and more accurate methods of calculation. In this paper are shown the test results (literature) in order to indicate the inaccuracy of the conventional method and the theory of elasticity, but also the possible inaccuracies in the study. The figure below shows a comparison of stresses for the three methods of calculation (classical theory, the theory of elasticity and the experimental method), where dashed curves show the effect of friction. It is evident that photoelasticity includes the effect of stress concentration (σ_{1eksp}).

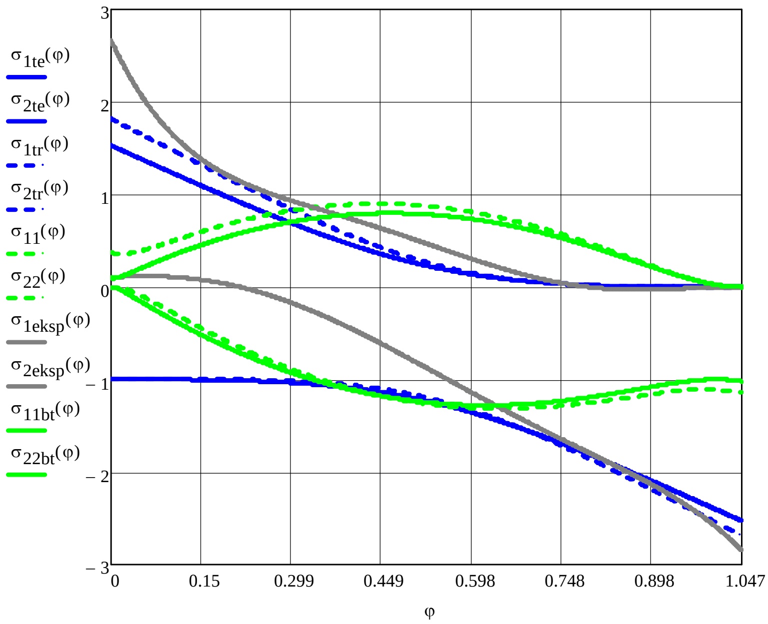

Figure 7. Comparative study on the distribution of main stresses in the critical section

2.4. Numerical calculation - Finite element method (FEM)

The finite element method (FEM) gives results that closely match the actual which can be checked with tensionmetar measurements on a real model [8]. Schedule of equivalent strain and comparison of the same for all previously mentioned methods shows inaccuracies that may appear on the determination of experimental stress - photoelasticity. Namely, the equivalent stresses are certainly higher in the final point x_0, which is located in the zone of influence of stress concentration, but in the second end point of the study of the critical section, which itself is not confirmed by experiment.

Since analytical methods do not consider the actual effect of stress concentration critical section can only be defined by numerical or experimental means. From the stress distribution obtained with FEM, it is clear that the actual critical section begins at a radius curves or in place of maximum stress concentration. It is clear that the values of equivalent strain obtained from FEM around so-called critical section is less than the value obtained by the theory of elasticity (Fig. 8), while the maximum value of stress is higher in comparison to all other listed methods (Fig. 9) because it most accurately describes the stress concentration. Namely in the so-called critical section the stress value obtained with the FEM in the starting point is 2.1 MPa, while the radius increases up to a maximum stress value of 3.1 MPa.

In Figure 8, dashed curves considering the influence of friction on the distribution of equivalent stresses in von Mises.

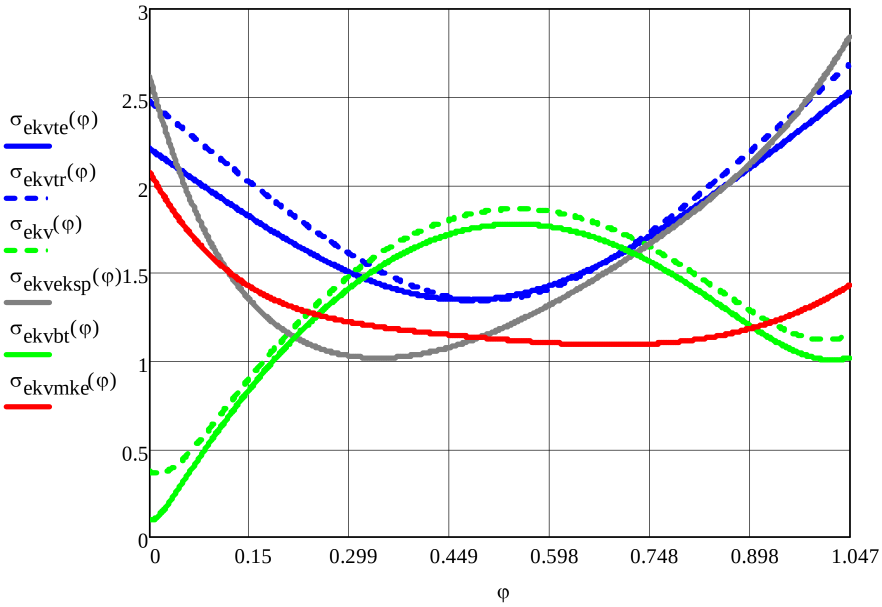

Figure 8. Comparative study on the distribution of equivalent stresses (HMH theory) in the so-called "critical" section

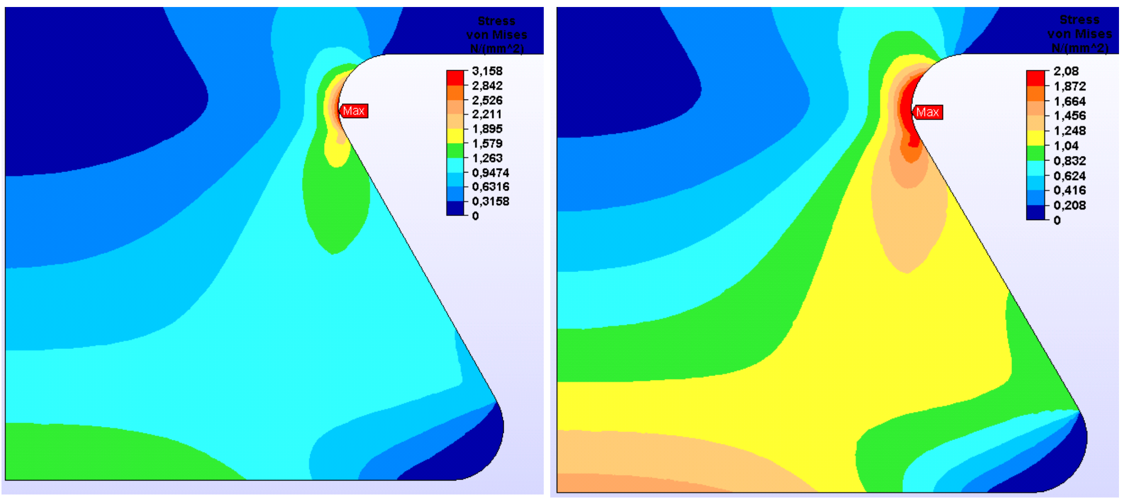

Figure 9. Showing the distribution of equivalent stresses (HMH theory) with the realistic view (right) due to the narrow category of local stress concentration

3. FINITE ELEMENT METHOD – TOOL FOR OPTIMIZATION

All previously presented methods of calculation related to the elastic area of material deformation, but in practice it is often fixation under load so that the material is in elastic-plastic deformation of the material. All previously presented methods other than the FEM can not describe the behavior of materials in the plastic deformation of materials, and it is clear how and why the FEM is the best way of determining the stress distribution. In addition, much better accuracy in the elastic deformation of material, and the possibility of considering the effects of fatigue FEM placed ahead of all other methods when selecting method, for both stress analysis and optimization and sizing the pole fixing. Therefore, only the FEM provides a more realistic picture "behavior of materials under load and is necessary as a selection method for the determination of the optimum structural fixation.

4. OPTIMIZATION

The main parameters of the optimization:

a) the geometry of the fixation (fixation dimensions)
b) structural constraints (pole geometry, geometry of the laminated rotor/pole wheel, etc.)
c) technological constraints (tools for making the radius, etc.)
d) material (mechanical properties of materials, availability of materials, cost of materials, etc.)
e) etc.

The first parameter that must be taken in optimizing is the geometry of the fixation and optimization of the geometry. Below is considered that parameter. In other words, here are presented some results of geometry optimization. Already from previous presentation we can draw the following conclusions important for optimizing the geometry:

a) finite element method is the best tool for the optimization (comprehensive calculation method, which covers both elastic and plastic deformation area of the material and the calculation of lifetime),
b) optimization must cover the area of elastic-plastic deformation of materials (without this an appropriate optimization can not be implemented),
c) optimization is several parametric,
d) defining the criteria of security (without this an appropriate optimization can not be implemented).

4.1. Security Criteria

As already mentioned preconditions for appropriate and quality optimization is defining the criteria of security, which will include the elastic-plastic deformation area of the material.

Common security criteria in the contractual documentation is defined so that maximum stresses in the theoretical overspeed should not be larger than 2/3 or 3/4 of conventional yield limit of material, while in normal mode does not exceed one third of conventional yield limit of material. When the safety criteria is defined in such manner it is not clear what is meant by the concept of maximum stress. Is it the maximum average stress?, Is it the maximum concentration of stress?, etc. All this entails that there are different interpretations, which usually goes to the detriment of producers who often must meet unrealistic demands. On the other hand a way of defining the safety factor does not say anything about the method of calculation, as can be seen from the preceding considerations significant differences appear both in values and in the distribution of stress for different methods of calculation. Using finite element method calculation each pole fixation (dove-tail, hammer, or their combination) will give a certain concentration of stresses that will be significantly higher than the average stress and dimensioning of such high values of stress result in unnecessarily oversized part. Calculation, carried out by the finite element method with nonlinear material properties easily come up with indicators that illustrate the nature of stress concentrations. Also the implementation of the fatigue calculation can show how high is the actual impact of stress concentrations.

All this tells us that we need to seriously consider the safety factor in the redefinition of contractual documentation, of which will benefit all, and the client and the generator manufacturer.

4.2. Geometry optimization

The very specific fixation geometry is defined by the number of parameters, and all parameters can be defined to be mutually interdependent, and for optimization of geometry, they must be. Below is given an example of geometry dependence of hammer pole fixation. We analyzed the hammer with the geometry shown in Figure 2. From a series of calculations performed with the FEM computer applications (one example shown in Figure 10) is evident dependence of the coefficient of stress concentration on the change of geometry (changes of the width of the hammer, hammer height, neck width hammer, the radius of curvature, etc.). All analysis were carried out in the field of elastic deformation of material as one of the first steps in the process of optimization.

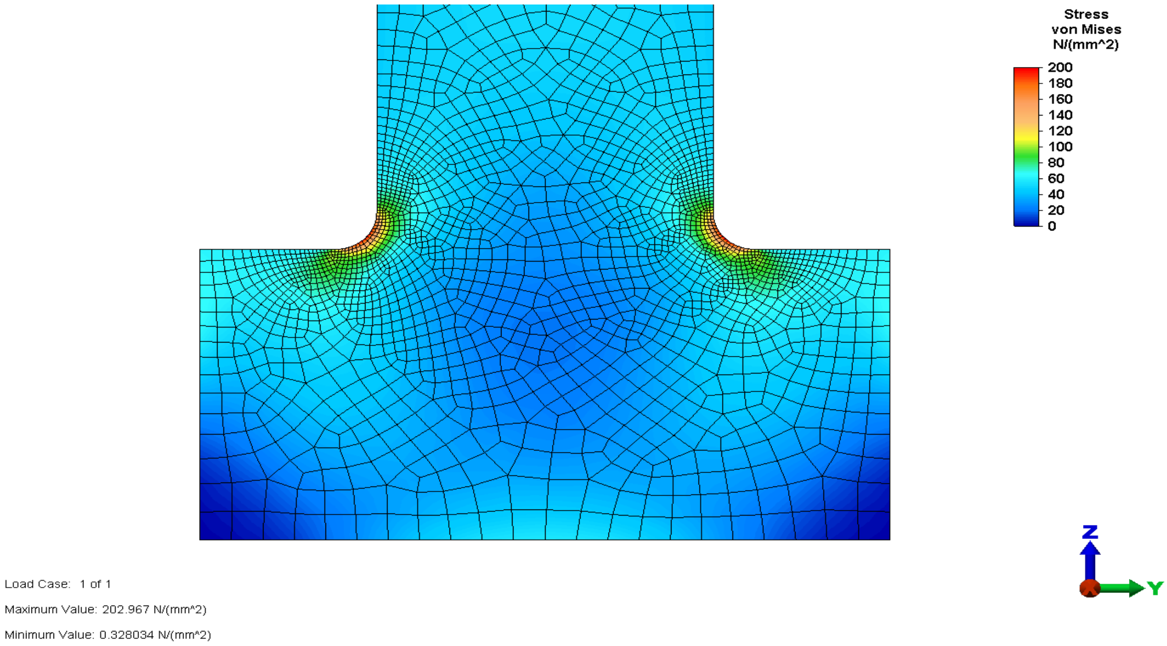

Figure 10. Distribution of stresses in the "hammer" pole fixation

Considering latitude of "hammer" pole it is clear that for any size radius curves there is a certain value width of the body in which the stress concentration are minimal (Fig. 11). Nominal stress to calculate the concentration coefficient is defined as a normal tensile stress in the hammer neck.

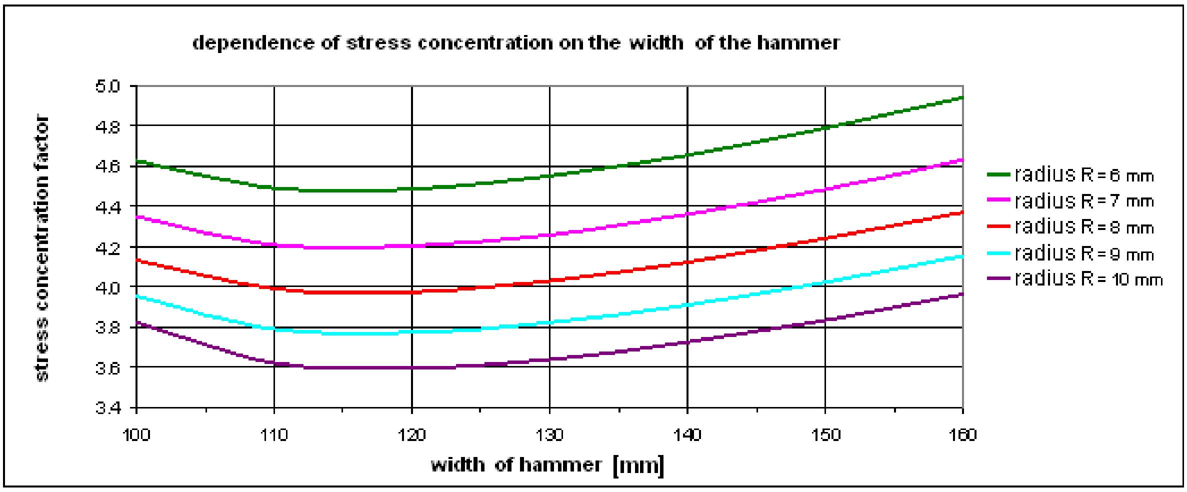

Figure 11. Dependence of coefficient of stress concentration on the width of the body, neck and „hammer" pole radius curves

Considering the dependence of the coefficient of stress concentration on the height, width of hammer neck and the radius of curvature, it is evident that for any radius of curvature with increasing hammer body height stress concentration is falling and asymptotically approaching a minimum value (Fig. 12).

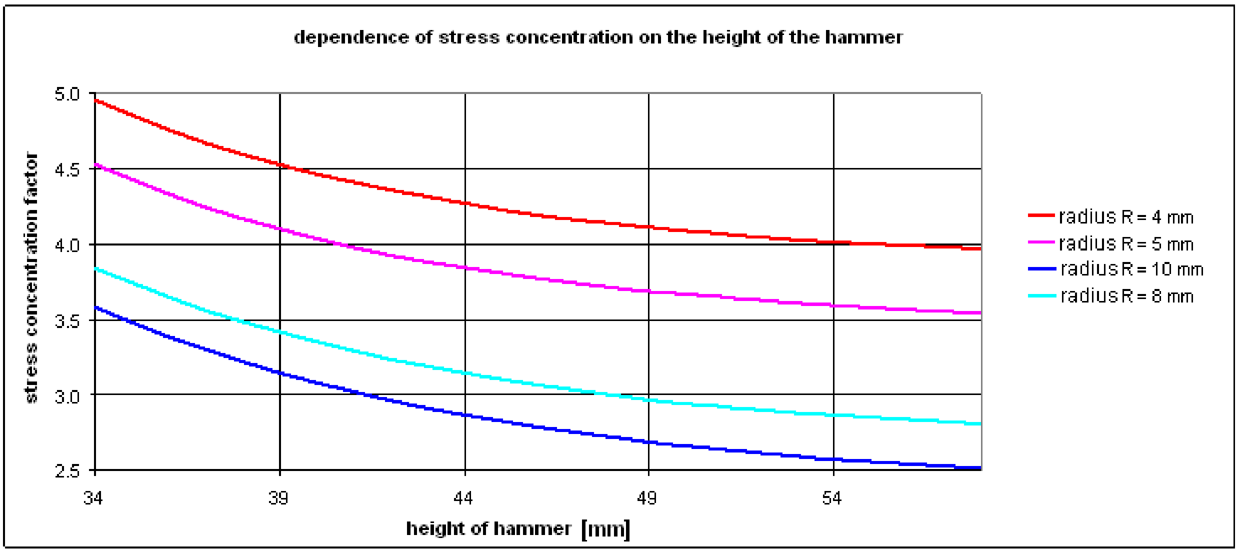

Figure 12. Dependence of coefficient of stress concentration on the height, width of the hammer neck and radius curves

Figure 13 shows the dependence of the coefficient of stress concentration on radius curves. The results represent the minimum factors of stress concentration arising on the basis of the results shown in Figure 11 (optimization by all parameters). It can be seen a significant reduction in stress concentration with increasing radius from 4 mm to 12 mm. Further increase shows no significant impact on reducing stress concentrations.

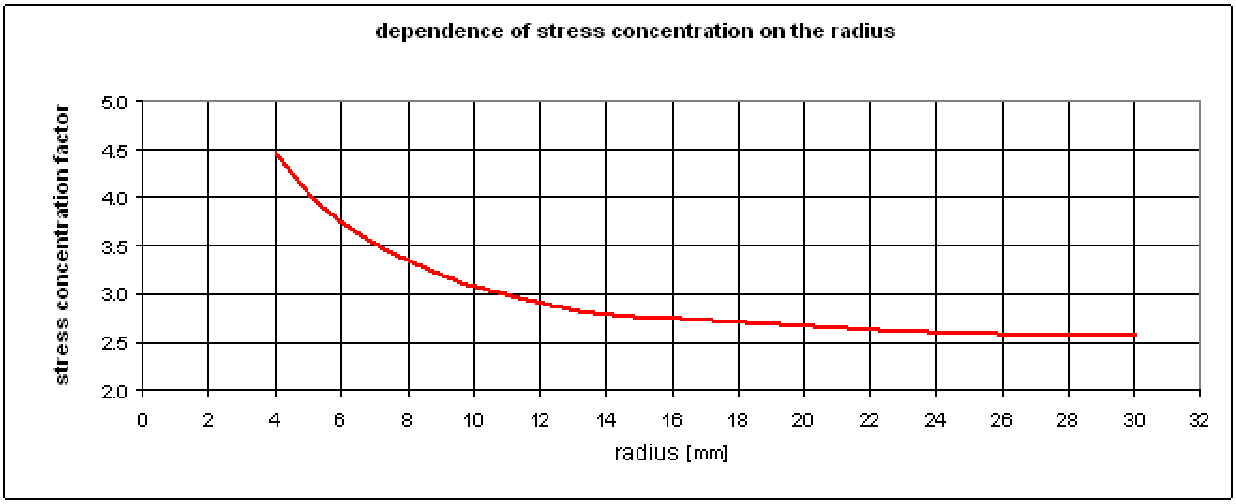

Figure 13 Dependence of concentration of stresses on the radius curves

From this short display of the analysis results, conducted for one type of fixing and certain geometry can be noted that the optimization certainly makes sense even if it is not necessary. It should be noted that here is only shown the effect of changing geometry on the coefficient of stress concentration and thus the maximum concentration of stress.

5. CONCLUSION

Comparative analysis of stress emphasizes the finite element method as relevant for determining the quality distribution of stresses in the pole fixation. In fact, all other methods have significant deficiencies (described in detail in this paper), and besides, not all of other methods have possibilities such as a FEM stress analysis of elasto-plastic state of deformation of materials or the analysis of lifetime due to fatigue effects, etc.

As the most accurate stress analysis, is a basic and initial step in the process of optimization, so the selection of the FEM is imposed as necessary in selecting method to optimize the geometry of the pole fixation.

To optimize the geometry of the hydrogenerator rotor pole fixation very important is redefinition of existing safety criteria. Safety criteria needs to be redefined in a way that they are defined in relation to the prescribed lifetime of the generator and all the other vital parts of the calculations should be based on lifetime, not just theoretical overspeed engine which is almost always the case. Of course, then in the contractual documentation shall prescribe the predicted number of cycles (number of start/stop, the expected number of working overspeed and the expected number of theoretical overspeed, which is usually only one or statistically speaking, happens almost exclusively when the flourishing hydrogenerator rotor factory is tested).

Almost all manufacturers base geometry on outdated methods of fixation, and in a certain way it is imposed to them by a contractual requirements, and this work shows them the possibility of significant cost reduction, but only if optimizations was well implemented to the vital parts of the generator rotor.

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