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# DOBÓR OPTYMALNYCH CECH KONSTRUKCYJNYCH KOŁPAKÓW WIRNIKA TURBOGENERATORA TWW-200-2

Streszczenie. W referacie przedstawiono sposób modelowania i metodę obliczeń stanu naprężenia i odkształcenia kołpaka turbogeneratora. Przedstawiona analiza może znaleźć szersze zastosowanie przy projektowaniu innych układów elektromechanicznych.

# SELECTION OF OPTIMUM DESIGN FEATURES FOR ROTOR CAPS OF A TURBOGENERATOR TWW-200-2

Summary. A method of modelling as well as a method for calculations of the state of stress and strain of the turbogenerator's cap have been presented. The presented analysis is liable to find a winder application when designing other electromechanical systems

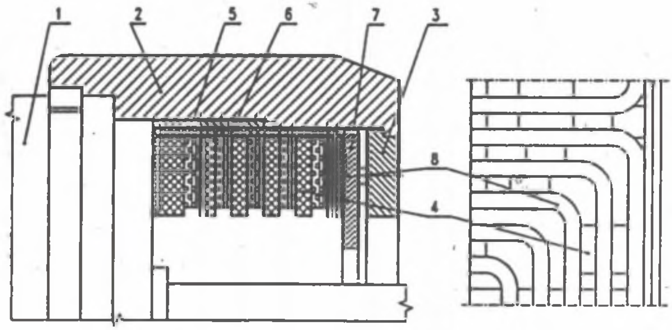
# ВЫБОР ОПТИМАЛЬНЫХ КОНСТРУКЦИОННЫХ СВОЙСТВ КОЛПАКА РОТОРА ТУРБОГЕНЕРАТОРА

Резюме. В статье приводится способ моделирования и метод вычисления состояния напряжения и деформации колпака турбогенератора. Приведённый анализ может найти более широкое применение в проектировании других электромеханических систем.

## 1. INTRODUCTION

End excitation windings of a turbogenerator placed in grooves of a rotor are fastened radially and axially by means of solid shrouds called caps (fig. 1)

A band of a cap 2, protecting and excitation windings from radial displacements and a thrust ring 3 which is joined with the band of the cap by interference and protects and excitation windings from axial displacements constitute basic elements of the cap.




1 - beczka wirnika
2 - obręcz kołpaka
3 - pierścień oporowy
4 - przekładki dystansowe
5 - ekran tłumiący
6 - izolacja główna
7 - wkładka oporowa
8 - uzwojenia wzbudzenia

Fig. 1. Sketch of a cap and of end winding of the rotor
Rys. 1. Szkic kołpaka i kół uzwojenia wirnika

As regards the connection of a cap to the rotor there are two types of design solutions as applied most often:

- interference joint of a cap band in teeth of the rotor barrel 1 and interference joint of a thrust ring in the rotor shalf,
- interference joint of a cap band in teeth of the rotor barrel only- these are the so called console- caps.

The first of the design solutions is characterized in that a deflection of the rotor shaft of the turbogenerator has an effect on stresses occurring in places of interference joints.

The rotors of turbogenerators which embody this design solution are being modernized at present so thet console - caps may be applied. These caps are protected from axial displacement by means of a nut or through bayonet joint.

The caps band belongs to those elements of the rotor of the turbogenerator which are exposed to the greatest mechanical loads.

In the band there are stresses arising from centrifugal forces of the mass of end excitation windings and their stiffenings as well as from centrifugal forces of proper mass of the cap. Besides, there are stresses resulting from the interference joint of the cap band in teeth of the rotor barrel.

The interference value must be so selected that no play should occur in the area of the interference joint at rotational speed applied during overspeeding of the rotor (20% above the rated speed).

Caps are made of non-ferromagnetic nickel steel or chromium-manganese--nickel steel. The steel X8CrMnN1818 with a required proof stress $R_{e0.2}$ = 900 to 1000 MPa is used most often for this purpose recently.

## 2. PHYSICAL MODEL OF JOINING A CAP BAND WITH THE ROTOR BARREL

Components of the state of stress, strain and displacement have been determined at the assumption of linear theory of elastic [4] with an axially symmetrical stress being assumed at the same time.

The programme "kolo pc" [2] elaborated on the basic of the finite elements method has been used for numerical calculations [3,5].

The assumption of an axially symmetrical stress in the band is abvious. However, some deviations arise here in the course of modelling the rotor barrel. This results both from the existence of grooves on excitation windings of the turbogenerator and from axial asymmetry of the arrangement of grooves (fig. 1). The asymmetry of the arrangement of grooves on the rotor barrel is mainly a consequence of the existence of the so called "great teeth".

In view of the fact, that the influence of asymmetry of the arrangement of grooves on the state of stresses in the cap band is inconsiderable it has been assumed, when calculating the state of stress, that the grooves are arranged uniformely on the perimeter of the rotor barrel.

The influence of grooves of the rotor barrel on interference joints has been taken into account when modelling a barrel by means of material with a substitute modulus of elasticity $E_z$ and substitute density $\rho_z$.

The substitute modulus of elasticity has been determined from eqality condition of radial displacements from equality condition of radial displacements on the surface of the interference joint with grooves and that of a "solid" rotor with the substitute modulus E on the assumption that the outside diameter and axially symmetrical load intensity p are the same in both cases. This is a from of the condition:

$$\Delta_{wz} = \Delta_{uz} + \Delta_{urw} \qquad (1)$$

where acc. to fig. 2a

$\Delta_{wz}$ - radial displacement on a surface of the rotor with a substitute modulus,

$\Delta_{uz}$ - change in length of a substitute tooth,

$\Delta_{urw}$ - radial displacement of a rotor core and of a tooth base.

The substitute density p has been determined in the same way ehen assuming the equality of axial displacements on the surface of a real rotor and of a substitute one. These displacements are caused by centrifugal forces for a constant angular velocity of the rotor.

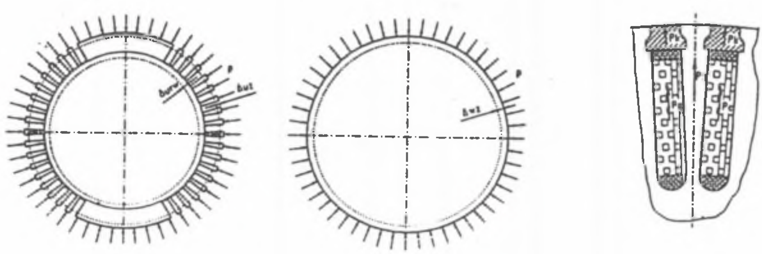


Fig. 2. Model for determining of:
a) substitute modulus $E_z$ of the rotor barrel ,
b) substitution density $\rho_z$ of the rotor barrel

Rys. 2. Model do wyznaczenia:
a) zastępczego modułu $E_z$ beczki wirnika,
b) zastępczej gęstości $\rho_z$ beczki wirnika

The equality condition of displacements is represented by the expression:

$$\Delta_{zast} = \Delta_z + \Delta_{zck} + \Delta_r \tag{2}$$

where:

$\Delta_{zast}$ - radial displacement on the surface of a rotor with modulus $E_z$ and density $\rho_z$,

$\Delta_z$ - elongation of a tooth due to interial forces $P_z$(fig. 2b) of proper mass of the tooth,

$\Delta_{zck}$ - elongation of a tooth due to interial forces $P_c + P_k$ (fig. 2b) of masses of the rotor windings with grooved keys,

$\Delta_r$ - radial displacements of the rotor core at the base of teeth due to interial forces.

Because of a non-uniform section of the rotor tooth- a change in its length due to inertial forces has been calculated when dividing it into segments.

## 3. NUMERICAL CALCULATIONS

Calculations of the state of stresses and strains have been made for normal operating conditions of the turbogenerator [1] ($\omega$ = 314 1/s) and for operation at angular velocity increased by 20% ($\omega$ = 377 1/s). When carrying out calculations these loads of a cap have been taken into account.

- due to inertial forces of proper mass of the cap,
- due to inertial forces of end winding of the rotor and their stiffenings, of insulation of the cap and damping screens,
- due to operating interference occuring during rotation of the rotor.

Charts of the distribution of reduced stresses and diagrams of their values on the inner periphery of the cap band (for $\omega$ = 377 1/s) are presented: in fig. 3a for stresses due to inertial forces of the cap mass and in fig. 3b - for stresses due to inertial forces of end windings of the rotor.

The assumption covers an operating interference of 0.15 mm occuring in the interference joint of the cap band with the rotor barrel at a speed as well as an operating interference of 0.1 mm in the joint of the cap band with a thrust ring. Compinent stresses occuring in the cap band have been calculated for these values of the operating interferences. The superposition of component stresses has been made according to Huber's hypothesis. Charts of the distribution of reduced stresses and of their values on the inner and outer periphery of the cap band are presented in fig. 3c - for stresses due to the operating interference and in fig. 3d - for total reduced stresses occurring in the cap band.

The calculations of radial strains of the cap band due to inertial forces of its proper mass and due to inertial forces of masses of end windings of the rotor as well as the calculations of radial strains of the outside surface of the rotor barrel and of the outside surface of the thrust ring have indicated that the necessary radial interference in the joint of the cap band and the rotor barrel is of 1.1 mm.

## 4. ANALYSIS OF RESULTS OF THE NUMERICAL CALCULATIONS

The assumed method for modelling the system - the cap and the applied mathod of finite elements- makes it possible to evaluate the state of stress and strain in these elements for any state of load and for any design features. Thus, it is possible to determine optimum design features of an

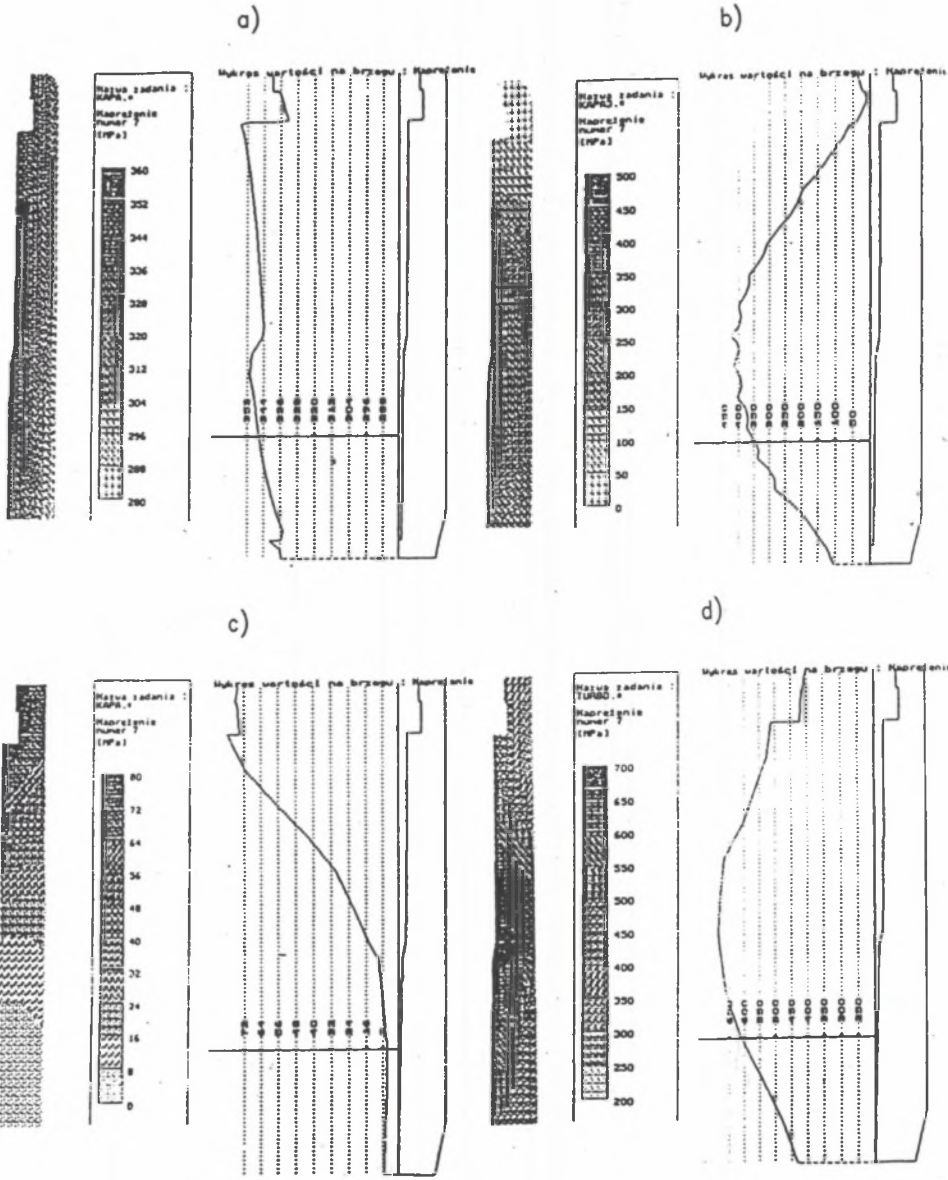



Fig. 3. Charts of stresses and their values on the inner periphery of the cap band:
a) due to centrifugal forces of the cap band, b) due to centrifugal forces of end windings of the rotor, c) due to the operating interference, d) resultant stresses

Rys. 3. Mapy naprężeń oraz ich wartość na brzegu wewnętrznym obręczy kołpaka
a) od sił odśrodkowych obręczy kołpaka, b) od sił odśrodkowych czół uzwojenia wirnika, c) od wcisku eksploatacyjnego, d) naprężenia wypadkowe

interference joint providing for durability and reliability of the joint during the operation by a numerical method.

A value of the interference the upper limit of which results from the requirement concerding the strength of the material whereas the lower limit of which is conditioned by life of the joint can be definad on the basic of numerical examinations.

The analysis of the effect of geometrical characteristics in places of stress concentration has served as the basic for design and construction work on the cap of the turbogenerator TWW-200-2

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Streszczenie

Przyjęty sposób modelowania układu - kołpak oraz zastosowana metoda elementów skończonych pozwala na ocenę stanu naprężenia i odkształcenia w tych elementach dla dowolnego stanu naprężenia oraz dowolnych cech konstrukcyjnych. Można więc na drodze numerycznej określić optymalne cechy konstrukcyjne połączenia wciskowego, zapewniające jego trwałość i niezawodność w czasie eksploatacji.

Na podstawie badań numerycznych można określić wartość wcisku, którego górna granica wynika z warunku wytrzymałości materiału, natomiast dolna jest określona trwałością połączenia.

Analiza wpływu zmian cech geometrycznych w miejscach koncentracji naprężeń była podstawą w procesie projektowo-konstrukcyjnym kołpaka turbogeneratora.

Przedstawiony sposób modelowania i metoda obliczeń mogą znaleźć szersze zastosowanie w innych układach elektromechanicznych.