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STRUCTURAL AND THERMAL STRESSES OCCURRING DURING PRESS-FIT MOUNTING PROCESS

Представлен метод расчета соединений с натягом, собираемых путем нагрева или охлаждения одного из элементов, основанный на температурном балансе. Отмечено, что при определенных условиях монтажные напряжения могут превысить номинальные напряжения. Предложен метод, с помощью которого можно с большой точностью рассчитать значения этих напряжений. Обоснованность предложенного метода подтверждена численным анализом.

The paper describes a method to calculate pressure coupling by means of cooling or heating one of the elements and based on the temperature balance. It is established that in some conditions the assembly stresses can exceed nominal stresses. A method to calculate these stresses with a pinpoint accuracy is offered. This method is justified by numerical analysis.

Stress calculations of contract press fit connections are usually conducted according to Lame`s task. Some simplifying assumptions are being taken. First is that the temperature field is homogeneous, static at working level. In real conditions, during mounting process there is a transient, gradiental temperature field, causing generation of thermal stresses in both of elements. Besides phenomenon provoking thermal stresses, quasi-static temperature field occurs in certain moment, when the temperature in both elements is more less the same and higher than the ambient temperature. Thus if the thermal expansion coefficient of pin material is higher then for hub, the pressure between connected elements will grow above the nominal pressure. This fact is not taken into account in conventional press-fit calculations method.

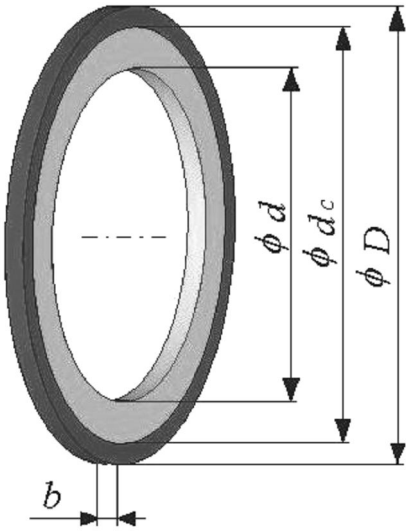


Fig.1. Press-fit sketch

As an example for contract press-fit analysis the cast iron hub and steel pin were chosen.

Dimensions: $b = 10$ mm; $d = 160$ mm; $d_c = 200$ mm; $D = 160$ mm.

Material properties:

Elastic modulus $E_1 = 2{,}05 \cdot 10^5$; $E_2 = 1 \cdot 10^5$ MPa;

Conductance coefficient $\lambda_1 = 0{,}04$, $\lambda_2 = 0{,}05 \frac{W}{\text{mmK}}$; Specific Heat $c_1 = 480$, $c_2 = 540 \frac{J}{\text{kgK}}$.

Density $\rho_1 = 7830$, $\rho_2 = 7250$ kg/m^3; Thermal Expansion coefficient $\alpha_1 = 16 \cdot 10^{-6}$ K^{-1}, $\alpha_2 = 9 \cdot 10^{-6}$ K^{-1}; Initial hub's temperature: $T_{20} = 673$ K, initial pin's temperature: $T_{10} = 293$ K.

Pressure at the contact surfaces of pin and hub [1]:

$$p = \frac{W_{nom} - (R_{z1} + R_{z2}) + (\alpha_1 \Delta T_1 - \alpha_2 \Delta T_2) 2r_s}{2r_s \left(\frac{k_1 - \nu_1}{E_1} + \frac{k_2 + \nu_2}{E} \right)}; \quad (1)$$

It was assumed for simplicity that, contact surfaces are dead smooth: $R_{z1} = R_{z2} = 0$ mm. Second assumption was that at the first moment of assembling process the outside diameter of pin has equal value to inner diameter of hub, thus nominal interference amounts to zero.

After cooling the assembly to ambient temperature, the value of interference totals:

$$(\alpha_1 \Delta T_1 - \alpha_2 \Delta T_2) 2 r_s = 0{,}684 \text{ mm}.$$

Substituting this to (1) the nominal pressure value was found: $p = 26{,}615$ MPa.

As it issues from Lame`s task the biggest radial stresses appear at contact surfaces and the biggest circular stresses appear at the inner surfaces of rings:

– radial stresses:

$$\sigma_{r1} = \sigma_{r2} = -p = -26{,}615 \text{ MPa}; \quad (2)$$

– circular stresses:

$$\sigma_{\varphi 1} = p(1 + k_1) = -147{,}846 \text{ MPa}; \quad (3)$$

$$\sigma_{\varphi 2} = p k_2 = 280{,}096 \text{ MPa}; \quad (4)$$

– Reduced stresses according to Huber-Mises-Hencky hypothesis for hub's inner surface [3]:

$$\sigma_{red} = \frac{1}{\sqrt{2}} \sqrt{(\sigma_r - \sigma_\varphi)^2 + (\sigma_\varphi - \sigma_z)^2 + (\sigma_z - \sigma_r)^2}; \quad (5)$$

$$\sigma_{red} = 294{,}31 \text{ MPa}.$$

In real conditions there are two phenomenon taking place at the same time. First is equalizing temperatures between both elements of joint and the second is cooling the joint until it has the same temperature as ambient air. From stress point of view a particularly unfavourable situation takes place when losses of heat from joint to environment are inconsiderably small comparing to heat exchange between hub and pin. Then energy balance has the following form:

$$T_1 m_1 c_1 + T_2 m_2 c_2 = T_k (m_1 c_1 + m_2 c_2).$$

The highest equalized temperature:

$$T_k = 437 \text{ K}; \text{ K} = 164 \text{ °C}.$$

Pin`s temperature increment:

$$\Delta T_1 = T_k - T_{10} = 164 - 20 = 144 \text{ K}.$$

Hub`s temperature descent:

$$\Delta T_2 = T_k - T_{20} = 164 - 400 = -236 \text{ K}.$$

Free contact diameters changes would be as follows:

$$(\alpha_1 \Delta T_1 - \alpha_2 \Delta T_2) 2 r_s = 0{,}8856 \text{ mm}$$

Thus after (1)

$$p = 34{,}460 \text{ MPa};$$

– radial stresses:

$$\sigma_{r1} = \sigma_{r2} = -p = -34{,}460 \text{ MPa};$$

– circular stresses:

$$\sigma_{\varphi 1} = p(1 + k_1) = -191{,}4253 \text{ MPa};$$

$$\sigma_{\varphi 2} = p k_2 = 362{,}657 \text{ MPa}.$$

– Reduced stresses according to Huber-Mises-Hencky hypothesis for hub's inner surface:

$$\sigma_{red} = 381{,}06 \text{ MPa}.$$

Comparing stresses calculated conventionally and stress calculated using energy balance the results are considerably different. Stresses occurring during mounting process are 30 % higher then nominal stresses in mounted joint.

As a verification of preceding calculations the finite elements method simulation was conducted. It was assumed that mounting process takes place in closed space, where the air temperature is 20 °C. For such conditions convection coefficient can be estimated at value:

$$\alpha_{wnik} = 20 \frac{W}{m^2 K}.$$

Thanks to symmetry only quarter of joint was modelled as it is shown below.

Analysis was conducted by two stages: thermal and structural. The results from first thermal, transient analysis became loads in series of structural, static analysis. In joint there are two sources of stresses. First results from different values of thermal expansion coefficients – these are mechanical stresses. The sec-

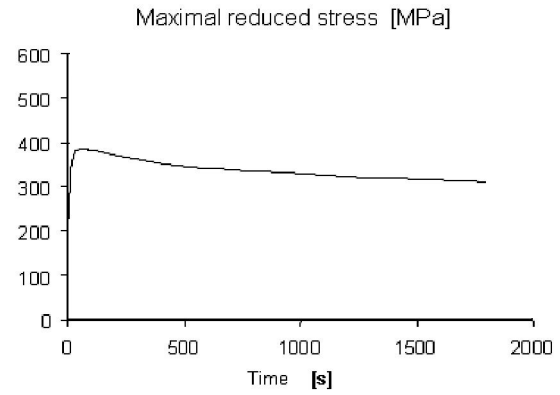


Fig.2. Maximal reduced according to HMH stresses vs. time

ond one is connected with gradient field of temperature in solid materials.

There were conducted several thermal analysis with different time at the end of load steps and different lengths of time step to achieve optimal accuracy. The highest, homogenous temperature of whole joint with accuracy of 1 K appeared about 74th second and approximated about T_k = 152 °C.

In set of structural analysis the state of reduced stresses (HMH) was investigated for certain time steps.

The highest reduced stresses appeared at inner surface of hub and approximated σ_{red} = 384 MPa. These stresses are 30 % higher than stresses calculated with conventional method and almost identical (difference 0,8 %) with these ones calculated using energy balance.

Whereas in example presented above maximal stresses during mounting process where higher then in steady conditions, it is reasonable in some cases to calculate energy balance. Stresses calculated using this method may be higher then the real ones, but surely they will not be smaller. It is particularly relevant in highly loaded contract or expansion press-fits.

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