NAPONSKO STANJE U NOSEĆIM VALJCIMA ROTACIONE PEĆI

STRESS STATE IN ROTARY KILN SUPPORT ROLLERS

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Originalni naučni rad

Ključne riječi:
noseći valjak, obodni napon

Keywords:
kiln roller, circumference stress

Paper received:
23.10.2016.
Paper accepted:
16.01.2017.

REZIME
U radu je sprovedena analiza napona koji se javljaju u nosećim valjcima rotacione peći. Naponi u nosećem valjku izazvani su temperaturnim gradijentom, steznim spojem s osovinom i kontaktom s nosećim prstenom. Analiza napona je urađena analitički i numerički, te su dobiveni rezultati uspoređeni.

Original scientific papar

SUMMARY
In the paper stress analysis of rotary kiln support rollers is conducted. Stress state in the support rollers is caused by temperature gradient, shrink-fit connection between shaft and roller and by the contact with kiln ring. The stress analysis is done analytically and numerically, and the results obtained are compared.

1. UVOD

Noseći prstenovi i valjci su elementi na koje se oslanja rotaciona peć i uslijed toga su izloženi različitim dinamičkim opterećenjima. Opterećenja potiču od same težine peći, zasipa, rotacije peći i temperaturnih gradijenata. Da bi se predvidjelo ponašanje valjaka i procjenio njihov radni vijek neophodno je izračunati ove napone.

Postoji ograničen broj studija u literaturi koji ispituju stanje napona u nosećim prstenovima i valjcima. Takvu analizu su proveli Xiao i oslali [1], koji su istraživali raspodjelu kontaknog pritiska i optimizaciju ugla koga valjci formiraju s centrom peći, ali nisu uzeli u obzir uticaj temperature. Dodatno, dosta studija istražuje napone u rotacionoj peći, napone u prstenovima, prenos toplote u peći, itd. [2-7], ali prema znanju autora, ne postoje objavljena studije s detaljnom teoretskom analizom i numeričkom simulacijom napona u nosećim valjcima.

Ovaj rad se fokusira na detaljnoj analizi napona u nosećem valjku koji su izazvani kontaktom sa prstenom, termičkim gradijentom i steznim spojem između osovine i valjka. Dodatno, analiza napona u valjku bit će provedena pomoću numeričkih simulacija.

1. INTRODUCTION

Riding rings and rollers are supporting elements of rotary kilns and therefore are subjected to various dynamic stresses. These stresses are caused by loads from the kiln weight and row-mix feed, rotation of the kiln and by temperature gradients. In order to predict the rollers behaviour and estimate their service life it is essential to calculate these stresses.

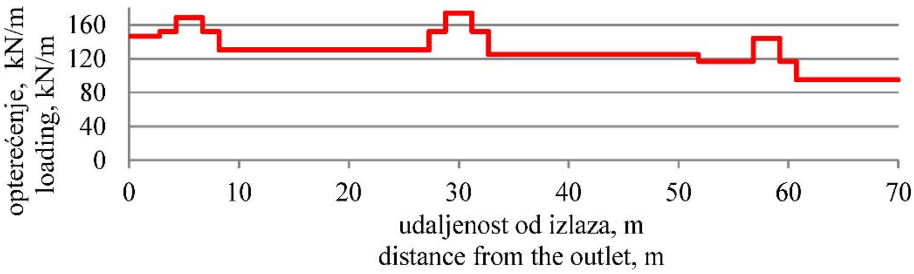
There is a limited number of studies in the literature that investigate stress state of the rotary kiln rings and rollers in more details. Such analysis was carried out by Xiao et al. [1], who investigated contact pressure distribution and support angle optimisation of kiln rings and rollers; but they did not include temperature effects. In addition, there are numerous studies investigating the rotary kiln's stresses, stresses in rings, heat transfer in kilns, etc [2-7], but to authors' knowledge, there are no published studies with the detailed theoretical investigation and numerical simulations of the stresses in kiln rollers.

This paper focuses on detailed analytical analysis of the kiln roller stresses caused by contact with the ring, temperature gradients and shrink-fit between shaft and roller. Additionally, stress analysis in the roller will be conducted by numerical simulations.

Istraživanje je sprovedeno za slučaj rotacione peći Fabrike cementa u Kaknju, prikazane na slici 1. To je čelična cijev dužine 70 m, unutrašnjeg prečnika 4,4 m, nagiba 3,5° i brzine rotacije 2 obr/min. Masa peći, uključujući oblogu i zasip iznosi oko 1000 t. Oslanja se na tri oslonca koje čine prsten i valjci, raspoređeni duž peći. Osnovne dimenzije i podaci o opterećenju peći, neophodni za dalju analizu, dati su u tabeli 1 i na slici 2.

The investigation is conducted for the case of the rotary kiln in the Cement plant in Kakanj, shown in Fig.1. It is a 70 m long steel tube with inner diameter of 4,4 m, slope of 3.5° and rotation speed of 2 rpm. The mass of the kiln, including refractory line and feed, is around 1000 t. It is supported by three ring-roller stations, spaced along the length of the kiln. The main dimensions and data for the kiln loading, necessary for subsequent analysis, are given in Tab. 1 and in Fig 2.

Slika 1. *Rotaciona peć Fabrike cementa u Kaknju*
Figure 1 *Rotary kiln in the cement plant in Kakanj*



Slika 2. *Opterećenje rotacione peći*
Figure 2 *Rotary kiln loading*

Tabela 1. *Dimenzije rotacione peći*
Table 1 *Rotary kiln dimensions*

Ring	Inner radius, R_1	Outer radius, R_2	Width, B	Roller Width, B_r	Inner radius, R_u	Outer radius, R_v
Ring 1	2318	2700	750	980	310	800
Ring 2, 3	2323	2700	880			

2. REAKCIJE OSLONACA ROTACIONE PEĆI

Kako bi se izračunale reakcije oslonaca, peć je podijeljena na 17 segmenata usljed različite krutosti omotača, položaja oslonaca i pogonskog zupčanika, te kontinuiranih opterećenja, kako je prikazano na slici 3.

2. ROTARY KILN SUPORT REACTIONS

In order to calculate support reactions, the kiln is divided into 17 segments, due to different shell rigidity, supports and drive-gear positions, and distributed loads, as shown in Fig.3.

Peć je modelirana u programu MDSolids-u 3.2 sa sljedećim pretpostavkama:

- zasip je ravnomjerno raspoređen duž peći sa specifičnom težinom od 18,58 kN/m,
- zasip je simetrično raspoređen oko vertikalne ose peći (slika 4),
- temperatura ne utiče na krutost omotača.

Specifična težina opeke na ulaznoj strani (dužine 18,2 m), tj. na desnoj strani na slici 3. je 44 kN/m. Ostali dio peći ima opeke sa specifičnom težinom od 74 kN/m. Kontinuirano opterećenje na slici 3. odgovara dijagramu iz slike 2. Težina pogonskog zupčanika je F_G=353 kN. Težine prstenova su: F_{R1}=353 kN, F_{R2}=406 kN, F_{R3}=410 kN.

Kao rezultat analize, dobivene su reakcije u osloncima (čvorovi 4, 9 i 15): F_4=2183,7 kN, F_9=4013,86 kN and F_{15}=3154 kN. Koristeći najveću vrijednost, na srednjem osloncu, može se dobiti maksimalno opterećenje Q koje djeluje na valjke (slika 4) kao:

$$Q = \frac{F_9}{2\cos 30°} \quad (1)$$

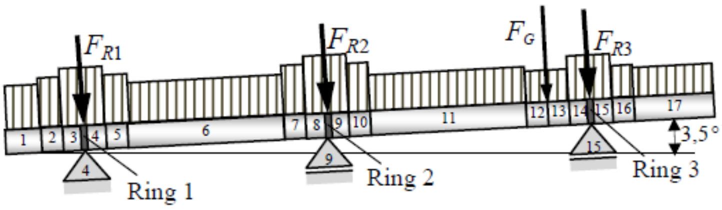
The kiln is modelled in MDSolids3.2 software with following assumptions:

- raw-mix is evenly distributed along the kiln length with specific weight of 18.58 kN/m,
- raw-mix is symmetrically distributed around the vertical axis of kiln (Fig. 4),
- temperature does not affect shell rigidity.

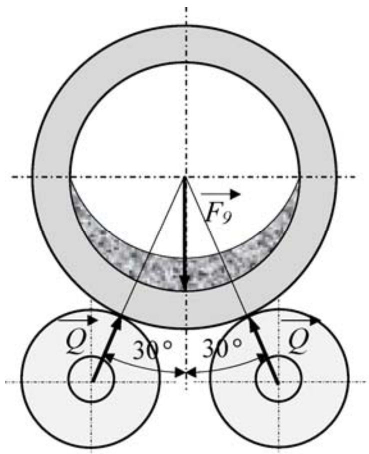
The specific weight of line bricks on the inlet side (18.2 m length), ie. right-hand side in Fig. 3 is 44 kN/m. The remaining part of the kiln has line bricks with specific weight of 74 kN/m. The distributed loads in Fig.3 correspond to the graph in Fig. 2. The weight of the gear ring is F_G= 353 kN. The weights of the rings are: F_{R1}=353 kN, F_{R2}=406 kN, F_{R3}=410 kN.

As a result of the analysis, roller support reactions are obtained (nodes 4, 9 and 15): F_4=2183.7 kN, F_9=4013.86 kN and F_{15}=3154 kN. Using the highest value, in the middle roller station, the maximum load, Q, acting on the rollers (Fig.4) can be obtained as:

$$Q = \frac{F_9}{2\cos 30°} \quad (1)$$



Slika 3. *Model peći*
Figure 3 *Kiln model*



Slika 4. *Reakcije na valjcima*
Figure 4 *Roller reactions*

3. NAPONI U NOSEĆIM VALJCIMA

Kod nosećeg valjka, pritisna sila po jedinici dužine iznosi P=2635,88 N/mm i dobiva se iz izraza:

$$P = \frac{Q}{B} \quad (2)$$

gdje je Q sila opterećenja valjka (1), a B=880 mm je širina prstena.

3. STRESSES IN KILN ROLLERS

Pressure force per unit length of support roller has a value P=2635.88 N/mm and it is given by:

$$P = \frac{Q}{B} \quad (2)$$

where Q is the force acting on the roller (1), and B=880 mm is the ring width.

Širina kontakte $2a$=8,48 mm je proračunata pomoću izraza [8]:

$$a=\sqrt{\frac{4PR}{\pi E^*}}, \tag{3}$$

gdje je R =617,1 mm ekvivalentni radijus, a E^* = 15,385 GPa je ekvivalentni modul elastičnosti.

Ekvivalentni radijus se računa prema [8]:

$$R=\left(\frac{1}{R_2}+\frac{1}{R_v}\right)^{-1}, \tag{4}$$

gdje je R_v = 800 mm radijus valjka, a R_2 =2700 mm radijus prstena.

Ekvivalentni modul elastičnosti je definiran s jednačinom [8]:

$$\frac{1}{E^*}=\frac{1-\nu^2}{E_1}+\frac{1-\nu^2}{E_2}, \tag{5}$$

gdje su $\nu_1=\nu_2$=0,3 Poissonovi koeficijenti, a $E_1=E_2$=210 GPa moduli elastičnosti materijala valjka i prstena. Oznake 1 i 2 odnose se na valjak i prsten, slijedom.

Maksimalna vrijednost pritiska je p_0= 396 MPa i locirana je na sredini kontaktne zone, a dobivena je na osnovu izraza [8]:

$$p_0=\frac{2P}{\pi a} \tag{6}$$

Pored pritisnog kontaktnog napona, u valjku se javljaju termički naponi usljed polja temperature i naponi usljed steznog spoja osovina-valjak.

Raspodjela termičkog napona u valjku dobivena je na osnovu izraza (7), [9] i prikazana je na slici 5. U proračunu su korištene vrijednosti temperature unutrašnje i vanjske strane valjka: t_u=40 ºC, t_v =100 ºC, te unutrašnjeg i vanjskog radijusa valjka: R_u=310 mm i R_v=800 mm.

The width of contact $2a$=8.48 mm is calculated using [8]:

$$a=\sqrt{\frac{4PR}{\pi E^*}}, \tag{3}$$

where R =617.1 mm is the equivalent radius and E^* = 15.385 GPa is the equivalent modulus of elasticity.

The equivalent radius is given by [8]:

$$R=\left(\frac{1}{R_2}+\frac{1}{R_v}\right)^{-1}, \tag{4}$$

where R_v =800 mm is the radius of roller, and R_2 =2700 mm is the radius of ring,

The equivalent modulus of elasticity is defined with the equation [8]:

$$\frac{1}{E^*}=\frac{1-\nu^2}{E_1}+\frac{1-\nu^2}{E_2}, \tag{5}$$

where $\nu_1=\nu_2$=0.3 are Poisson' ratios, and $E_1=E_2$=210 GPa are the modules of elasticity of roller and ring materials. The subscripts 1 and 2 relate to the roller and the ring, respectively.

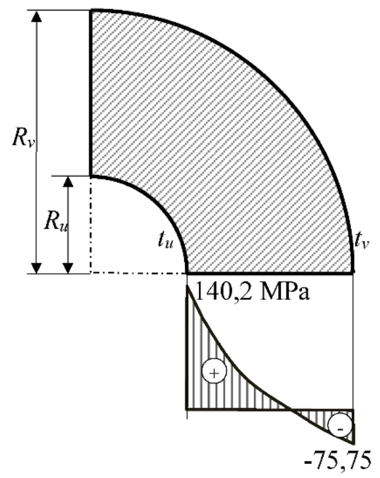
Maximum contact pressure is p_0= 396 MPa and it is located in the middle of contact area, and it was calculated using [8]:

$$p_0=\frac{2P}{\pi a} \tag{6}$$

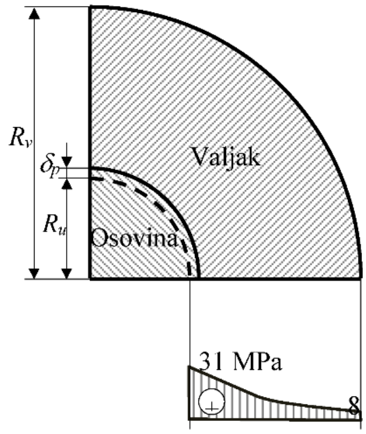
In addition to the contact pressure, there are thermal stresses in roller caused by temperature gradient and stresses caused by shrink-fit between shaft and roller.

Thermal stress distribution in roller is obtained using the equation (7), [9] and is shown in Fig. 5. In the calculation values for the inside and outside temperatures of the roller: t_u=40 ºC, t_v =100 ºC, and the inside and outside radius of roller: R_u=310 mm and R_v=800 mm, are used.

$$\sigma_t=\frac{E}{1-\nu}\left[\frac{1}{r^2}\int_{R_u}^{r}\alpha t\cdot rdr+\frac{r^2+R_u{}^2}{r^2\left(R_v{}^2-R_u{}^2\right)}\int_{R_u}^{R_v}\alpha t\cdot rdr-\alpha t\right]$$
$$t=\frac{\Delta T}{\ln\frac{R_v}{R_u}}\ln\frac{R_v}{r}+t_v \tag{7}$$



Slika 5. *Raspodjela termičkog napona kroz debljinu valjka*
Figure 5 *Thermal stress distribution through roller thickness*



Slika 6. *Raspodjela napona usljed preklopa kroz debljinu valjka*
Figure 6 *Stress distribution caused by shrink-fit through roller thickness*

Usljed temperature, proizvedene unutar peći, doći će do širenja osovine i valjka. Vrijednosti pomjeranja vanjskog vlakna osovine (0,1936 mm) i unutrašnjeg vlakna valjka (0,381) usljed širenja mogu se dobiti na osnovu izraza za pomjeranje $u(r)$ (8), [9]. Razlika pomjeranja daje zazor između ova dva elementa u vrijednosti od 0,1874 mm.

Due to temperature, produced inside the kiln, elongation of shaft and roller will occur. The values of elongation of the outer radius (0.1936 mm) and the inner radius of roller (0.381) can be obtained by equation for elongation $u(r)$, (8), [9]. The difference between the elongations gives a gap between these two elements which has a value of 0.1874 mm.

$$\begin{aligned} u(r) &= \frac{1}{r}\frac{1+\nu}{1-\nu}\int_{R_u}^{r} \alpha \cdot t(r) \cdot r dr + C_1(r)\cdot r + C_2(r)\cdot \frac{1}{r} \\ C_2(r) &= \frac{1+\nu}{1-\nu}\frac{R_u^2}{R_v^2-R_u^2}\int_{R_u}^{R_v} \alpha \cdot t(r)\cdot r dr \\ C_1(r) &= (1-2\nu)\frac{1+\nu}{1-\nu}\frac{1}{R_v^2-R_u^2}\int_{R_u}^{R_v} \alpha \cdot t(r)\cdot r dr \end{aligned} \tag{8}$$

Prvobitan preklop od 0,26 mm (jednak je maksimalnoj vrijednosti preklopa), usljed djelovanja temperature, smanjit će se i iznositi δ_p=0,0726 mm. Vrijednosti obodnih napona usljed steznog spoja osovina-valjaka izračunate su na osnovu izraza (9), [9] i date su na slici 6.

$$\begin{aligned} \sigma_\theta(R_u) &= p_{12}\frac{R_v^2+R_u^2}{R_v^2-R_u^2} \\ \sigma_\theta(R_v) &= p_{12}\frac{2R_v^2}{R_v^2-R_u^2} \end{aligned} \tag{9}$$

pri čemu je pritisak između osovine i valjka:

$$p_{12} = \frac{E\delta_p}{R_u}\frac{R_v^2-R_u^2}{2R_v^2} \tag{10}$$

The original interference of 0.26 mm (equal to maximal interference value), due to temperature, will be lowered and has a new value of δ_p=0.0726 mm. Circumference stresses due to shrink-fit between shaft and roller are calculated by expression (9), [9] and shown on Fig. 6.

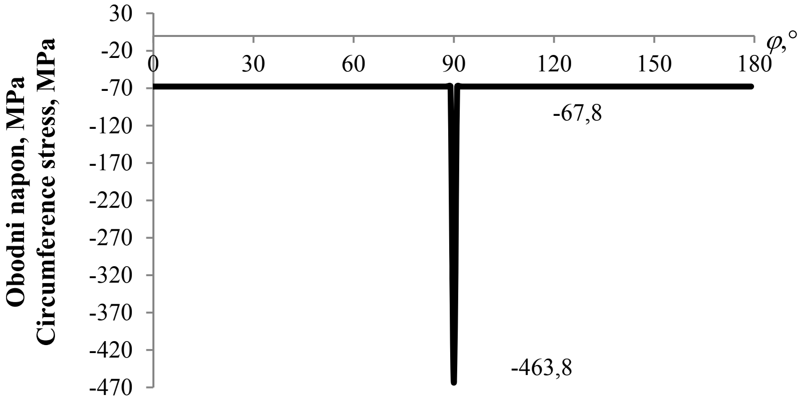
$$\begin{aligned} \sigma_\theta(R_u) &= p_{12}\frac{R_v^2+R_u^2}{R_v^2-R_u^2} \\ \sigma_\theta(R_v) &= p_{12}\frac{2R_v^2}{R_v^2-R_u^2} \end{aligned} \tag{9}$$

wherein, the pressure between shaft and roller is:

$$p_{12} = \frac{E\delta_p}{R_u}\frac{R_v^2-R_u^2}{2R_v^2} \tag{10}$$

Zbirni obodni napon u vanjskom vlaknu valjka prikazan je na slici 7. Maksimalni pritisni napon u sredini zone kontakta superponira se s termičkim naponom i naponom usljed preklopa osovina-valjak i iznosi -463,8 MPa.

The total circumferential stress on the outside radius of roller is shown in Figure 7. Maximal pressure stress, in the middle of contact area, is superimposed with thermal and shrink-fit stresses and it has a value of -463.8 MPa.



Slika 7. *Obodni napon u vanjskom vlaknu valjka*
***Figure** 7 Circumference stress on the outside raduis of roller*

5. NUMERIČKA ANALIZA

Numerička analiza je provedena koristeći program ABAQUS [10]. U svim simulacijama, prsten i valjak modelirani su u ravni, sa samo polovinom domena zbog simetrije, te su primjenjeni uslovi za ravno stanje deformacija. Simulacije su statičke, tj. inercijalni efekti se zanemareni zbog spore rotacije peći. Osobine materijala za oba dijela su osobine čelika.

Sistem je izložen djelovanju temperaturnog gradijenta i opterećen kao na slici 4. Osobine analize su: *coupled temperature-displacement step, steady-state response*. Sistem prsten-valjak je izmrežen, s odgovarajućim osobinama kontakta između dijelova. Za mrežu su korišteni pravougaoni elementi s osam nodova, tipa CPE8T. Ukupan broj elemenata sistema prsten-valjak je 9 444 (slika 8).

Mreža je u području kontakta usitnjena kako bi se postigla bolja tačnost rezultata. Osobine kontakta su: *penalty friction formulation* (koeficijent trenja je 0.1) *i surface-to-surface interaction with finite sliding formulation.*

Nanesene su vrijednosti temperatura unutrašnje i vanjske strane valjka: t_u=40 ºC, t_v =100 ºC. Da bi nastao preklop od 0,26 mm, osovini je povećan radijus za vrijednost preklopa. U prvom koraku (*General, Coupled temp-displacement*), istovremeno s djelovanjem temperature, preklop se rješava uklanjanjem čvorova u preklopu.

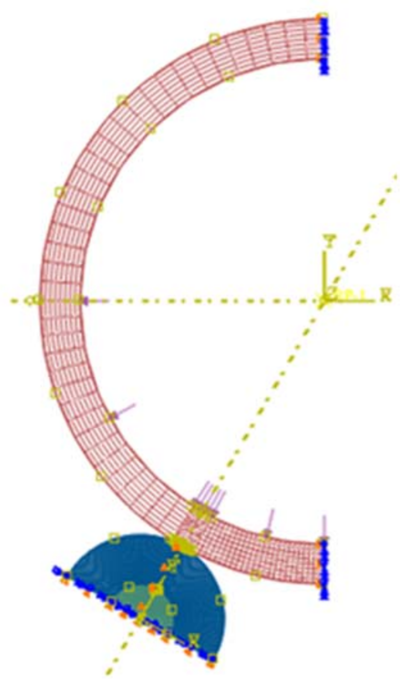
5. NUMERICAL ANALYSIS

Numerical analysis is carried out using ABAQUS software [10]. In all simulations the ring and rollers are modelled in 2D, with only a half of the domain modeled due to symmetry and a plain strain condition is applied. The simulations are static, i.e. inertia effects are neglected due to slow rotation of the kiln. Material properties for both parts are those of steel.

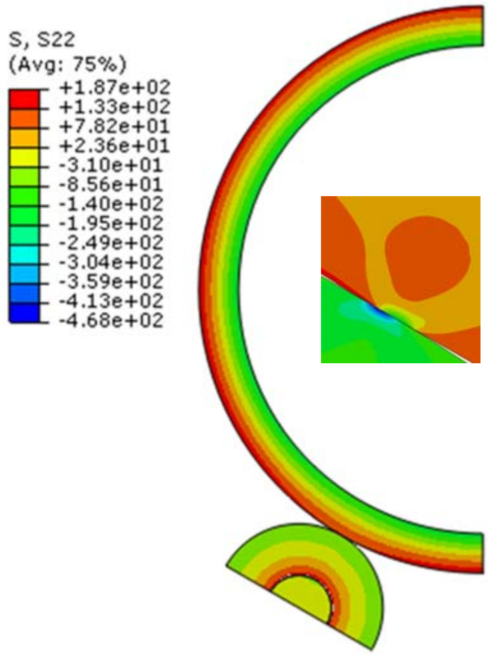
Ring-roller system is exposed to temperature gradient and loaded as in figure 4. Analysis features are: *coupled temperature-displacement step, steady-state response*. Ring-roller system is meshed, with appropriate contact properties between parts. For the mesh, quadrilateral elements with eight nodes, CPE8T type, are used. The total number of elements of the ring-roller system is 9 444 (Fig. 8).

The mesh in contact area is refined in order to achive better result accuracy. The contact properties are: penalty friction formulation (coefficient of friction is 0.1) and surface-to-surface interaction with finite sliding formulation.

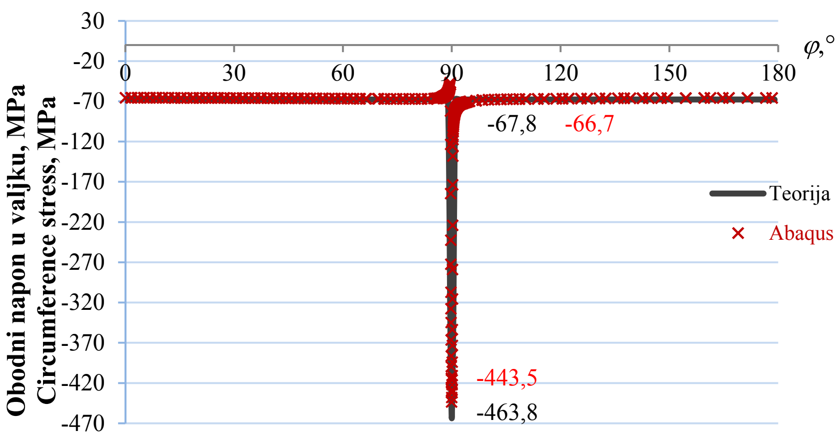
Temperature values on inside and outside radius of roller are applied: t_u=40 ºC, t_v =100 ºC. In order to get interference of 0.26 mm, the radius of shaft is increased by this value. In the first step (General, Coupled temp-displacement), simultaneously with action of temperature, the interference is resolved by node removal from the interference.



Slika 8. *Granični uslovi i opterećenje sistema prsten-valjak*
Figure 8 *Boundary conditions and the load on the ring-roller system*



Slika 9. *Obodni napon u prstenu i valjku – lijevo, detalj kontakta –desno*
Figure 9 *Circumference stress in the roller and the ring, detail of contact-right*



Slika 10. *Obodni napon u vanjskom vlaknu valjka-numerički i analitički*
Figure 10 *Circumference stress on outside radius of the roller- numerical and analitical*

Obodni napon sistema prsten-valjak prikazan je na slici 9, s istaknutim detaljem napona u području kontakta. Obodni napon na vanjskoj površini valjka dat je na slici 10. U vanjskim vlaknima valjka djeluje skoro konstantan napon pritiska od -66,7 MPa, sa naglim skokom do vrijednosti od -443,5 MPa u području kontakta. Ukupno slaganje numeričkog i analitičkog rješenja je dobro.

The circumference stress of the ring-roller system is shown in Fig. 9, with the highlighted detail of stress in the contact. The circumference stress on the outside radius of roller is shown in the Fig. 10. On the outside radius of roller, there is almost constant pressure in the amount of -66.7 MPa with the peak value of -443.5 MPa in the middle of contact area. Total agreement between analytical and numerical solutions is good.

5. ZAKLJUČAK

Noseći valjak preuzima dio opterećenja rotacione peći preko kontakta dvije zakrivljene površine. Kontakt se ostvaruje po pravougaonoj površini vrlo male širine, manje od 1 cm. Dužina površine kontakta, u idealnom slučaju, jednaka je širini prstena, ali može biti smanjena zbog trošenja i formiranja valovitosti površine. Osim ovog kontaknog napona pažnju treba obratiti i na napone koji su izazvani nejednakim zagrijavanjem valjka od strane peći. To stvara značajne, obodne, termičke napone. Zagrijavanje valjka utiče i na njegovo širenje. Dolazi do širenja unutrašnjeg vlakna valjka i vanjskog vlakna osovine. Ova širenja utiču na smanjenje vrijednosti projektovanog preklopa steznog spoja osovina-valjak. Ovo može biti toliko izraženo da valjak sklizne s osovine, što je često zabilježeno u praksi. Analitička i numerička analiza daju mogućnost da se ovi naponi u fazi projektovanja predvide i uzmu u obzir prilikom izbora tolerancije steznog spoja.

5. CONCLUSION

The support roller takes one part of the rotary kiln loads over the contact of two curved surfaces. The contact is established on the rectangular area of a very small width, less than 1 cm. A length of contact area is, ideally, equal to a width of the ring, but can be lowered due to wear and formation of surface waviness. In addition to this contact stress, attention should be paid to the stresses caused by unequal roller warming-up by the kiln. This creates significant, circumference, thermal stresses. Warming up the roller also causes its elongation. There is an elongation of the inside radius of roller and the outside radius of shaft. These elongations affects the decrease in value of projected interference of shaft-roller shrink fit. It can be so significant that the roller slides off the shaft, which is often observed in practice. The analytical and numerical analysis provide the option to anticipate and take into account these stresses in a stage of design when tolerance of the shrink-fit is selected.

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