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THE ANALYSIS OF THE STATIC BEHAVIOR OF SPECIAL ROLLER BEARINGS WITH FORGED OUTER RING

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ABSTRACT

This paper presents an analysis of stiffness and stress on the Hub unit bearing (HUB). The mathematical model consists of two parts. The first part of the developed software solution to determine the contact stiffness for each rolling elements, while the second part on the basis of contact stiffness determines the stiffness and stress of bearing based on the finite element method. Software solution is developed based on Hertz's contact load and John Harris of the quasi-static equilibrium equations. Mathematical model analyzes the influence of the cornering acceleration of the automotive wheel on the bearnig stiffness and stress. The research in this paper shows that the cornering acceleration of the automotive wheel significantly affects the axial, radial and angular stiffness from one side. On the other hand, lateral acceleration wheel significantly affectes the increase in stress on the flanged outer ring which are made by forging.
KEYWORDS: *Hub unit bearing, finite element method, static behavior, forging.*

1. INTRODUCTION

In today's automotive industry, the development of wheel bearings based on the integration of vehicles curtian componenet axis ("Hub Unit Bearing - HUB") is besoming more common, in order reduce the weight and dimensions as well as to improve overall vehicle performance.
Based on of present knowledge we can conclude that there are three directions of development of constructive solutions wheel bearing. One of the directions of the development of HUB unit bearing is advanced technology in making axle. This technology is based on a compact module HHM (Halfshaft Hub Module), which reduces the weight of the vehicle and increases the flexibility of the system for the suspension and steering systems. Another direction of development of constructive solutions wheel bearing vehicle is its integration with the brake drum. This solution provides a longer bearing life and rigidity while reducing weight, easy installation and lower costs. The third line of development of constructive solutions wheel bearing vehicles with integrated brake discs. Application of this solution is primarily expected in sports cars.

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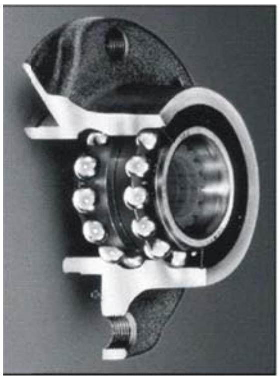
This paper analyzes the influence of lateral acceleration on stiffness and stress of the HUB unit beraing second generation. Hub unit bearing II is most often used for front or rear wheels, whereby the rotation of the outer ring (Fig. 1) that binds to the wheel and brake disc brake system, while the shaft is fixed with screws in the inner ring. The flanged outer ring is made by forging.The analysis was conducted using the developed mathematical model with five degrees of freedom and the finite element method. The mathematical model takes into account inertia forces, clearance in the bearing and the bearing temperature elements.



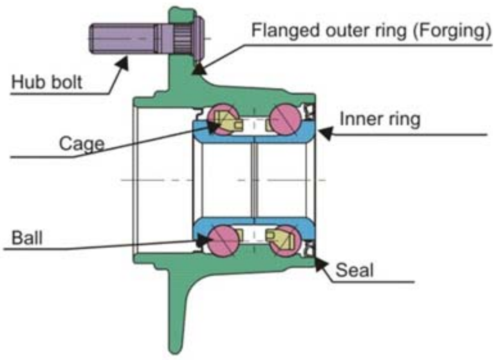


Fig.1 *- Hub unit bearing II solution with ball bearing and basic elements*

2. MATHEMATICAL MODEL

When defining the mathematical model, two coordinate system was used as shown in Figure 2. The first is a general coordinate system $\{x,y,z,\theta_y,\theta_z\}^T$, which is associated with the degrees of freedom of rotating ring and is located in the center of the bearing. The second is a local coordinate system $\{r, x, \phi\}^T$, which defines the position of the center rotating raceways for each rolling element and is set in the center of curvature of the outer raceways of rolling bearings. It is assumed that the center of curvature of the inner raceways is stationary and used as a fixed point when the outer ring is rotary.

In the event that there is no external load, the raceway groove curvature center is located on distance A, as shown in the Fig. 2. After effects of static load, the distance between the center of curvature of the inner and outer raceways A increases the size of the contact deformation δ_u and δ_s. The line between the load centers are collinear with distance A. However, the action of the centrifugal force and gyroscopic moment of rolling elements, due to the different angles of contact between the rolling body and the outer and inner raceways, the line of the load will not be colinear with the distance between the centers. In Figure 3 the positions of the center of rolling bodies are shown, as well as the positions of the inner raceway groove curvature center, both with and without the action of centrifugal force and gyroscopic moment in rolling body.

In Figure 3, X_{1j}, X_{2j}, α_{ij}, α_{oj}, Δ_{ij}, Δ_{oj} are the axial and radial components of the position of the center of the rolling body, angle of contact with the inner and outer raceways and rolling distance between the center of the body and the center of curvature of the inner and outer raceways. Bearing elements deform under the action of loads. On heating elements bearing leads to shrinkage of rolling bodies and expanding the rings. Due to these changes, there is a changing distance between the center of curvature of the inner and outer raceways and new position the center of the rolling body.

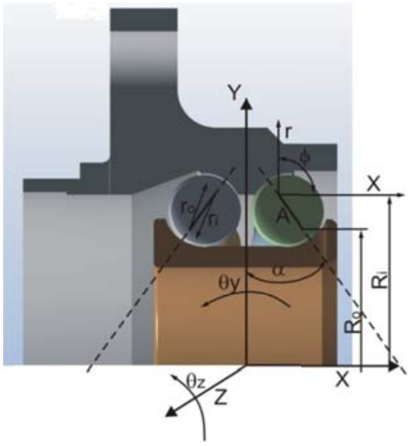


Fig.2 *- Coordinate system of model with five degrees of freedom*

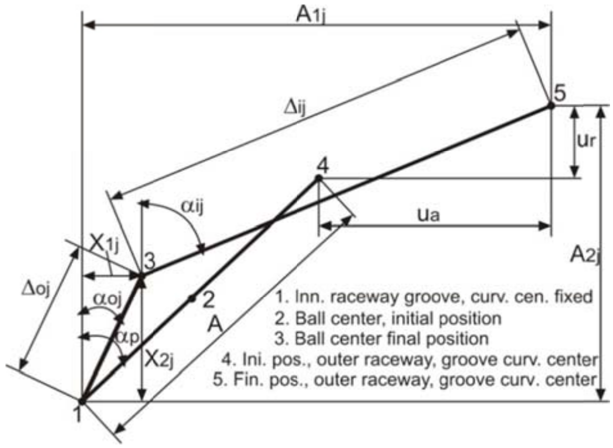


Fig.3 *- Positions of ball center and raceway groove curvature centers at angular positions ψ_i with and without applied load*

In order to account for the influence of clearence and temperature, the aformentioned id defined as:

$$\Delta_{ij} = \left(f_i - 0.5\right) d_b + \delta_{ij} \pm G_r - \varepsilon_b$$
$$\Delta_{oj} = \left(f_o - 0.5\right) d_b + \delta_{oj} \pm G_r - \varepsilon_b \qquad (1)$$

In the equations (1): *fi=d_b/di, fo=d_b/do,* d_b-rolling element diameter*, di, do*-diameter of the inner and outer raceways, δ_i, δ_o-deformations of the inner and outer raceways, G_r- positive clearance / negative clearance, ε_b- thermal expansion of the rolling element. Applying the Pythagorean theorem, according to Figure 2, we get the following equation of dispalcemenet [7]:

$$\left(A_{1j} - X_{1j}\right)^2 + \left(A_{2j} - X_{2j}\right)^2 + \left(l_{ij} + \delta_{ij} \pm G_r - \varepsilon_b\right)^2 = 0$$
$$X_{1j}^2 + \left(X_{2j} - \varepsilon_o\right)^2 + \left(l_{oj} + \delta_{oj} \pm G_r - \varepsilon_b\right)^2 = 0 \qquad (2)$$

The equation of the forces equilibrium on the roller element is:

$$Q_{oj}\cos\alpha_{oj} - Q_{ij}\cos\alpha_{ij} - \frac{M_{gj}}{d_k}\left(\lambda_{ij}\sin\alpha_{oj} - \lambda_{oj}\sin\alpha_{uj}\right) - F_{cj} = 0$$
$$Q_{oj}\sin\alpha_{oj} - Q_{ij}\sin\alpha_{ij} + \frac{M_{gj}}{d_k}\left(\lambda_{oj}\cos\alpha_{oj} - \lambda_{ij}\cos\alpha_{ij}\right) = 0 \qquad (3)$$

where: Q_i i Q_o- Hertz's contact forces, M_g- gyroscopic moment, F_c- centrifugal force, λ- constant depending of the raceway control.

Hertz's contact force between the inner raceways and rolling elements and the outer raceways and rolling elements are determined by the [3,4]:

$$Q_{i/o(i)} = \begin{cases} K_i \delta_{i(i)}^{3/2} \\ K_o \delta_{o(i)}^{3/2} \end{cases} \qquad (4)$$

Whether there will be any inner or outer raceways rolling will depend on conditions set out in the table T.1 [1].

Table T1. Conditions in which there is pure rolling

Outer ring	$Q_{sj} a_{sj} E'_{sj} \cos\left(\alpha_{uj} - \alpha_{sj}\right) > Q_{uj} a_{uj} E'_{uj}$
Inner ring	$Q_{uj} a_{uj} E'_{uj} \cos\left(\alpha_{uj} - \alpha_{sj}\right) > Q_{sj} a_{sj} E'_{sj}$

Under different conditions of rolling the body loses contact with the inner ring. In other words, the outer raceway generates the force of reaction between rolling elements and the centripetal acceleration. As a result, the external angle of contact α_{oj} j and gyroscopic moment becomes equal to 0. Contact force with the external ring Q_{oj} in this case, is equal to the centrifugal force F_{cj}, respectively, with the external deformation path is $d_{oj}=(F_{oj}/K_o)^{2/3}$. In order for the loss of contact with the inner racewazs to occur, the following condition must be satisfied [1]:

$$A_{1j}^2 + \left[A_{2j} - \left(f_o - 0.5\right)d_b + K_o^{-2/3} F_{cj}^{2/3}\right]^2 \le \left[\left(f_i - 0.5\right)d_b\right]^2 \tag{5}$$

Nonlinear motion equation (2) and the equilibrium equation (3) can be solved simultaneously using the Newton-Raphson's method of iteration to determine unknown X_{1j}, X_{2j}, δ_{ij} i δ_{oj}.

If the relative displacement bearing marked with δ_k^m, where k = x, y, z, θy, θz and *m= i, o*, then the contact stiffness of bearing can be expressed by:

$$K_{i,k}^L = \frac{\partial F_k^u}{\partial \delta_k^u} = \left(-\sum_{j=1}^{Z} [T]_j^T \frac{\partial Q_j}{\partial u_j} [T]_j\right)_i$$
$$K_{o,k}^L = \frac{\partial F_k^s}{\partial \delta_k^s} = \left(-\sum_{j=1}^{Z} [T]_j^T \frac{\partial Q_j}{\partial u_j} [T]_j\right)_o \tag{6}$$

where [Tj] transformation matrix of the form:

$$\left[T_j\right] = \begin{bmatrix} \cos\psi_j & \sin\psi_j & 0 & -\sin\psi_j & \cos\psi_j \\ 0 & 0 & 1 & r_u\cos\psi_j & -r_u\cos\psi_j \\ 0 & 0 & 0 & -\sin\psi_j & \cos\psi_j \end{bmatrix} \tag{7}$$

The total contact stiffness of bearing based on [5] calculated as stiffness inner and outer ring:

$$K^L = K_i + K_o \tag{8}$$

By changing the vehicle's movement from linear to non-linear, a significant increase of moment around the Z axis occurs, which in turn increases the angular displacement. Due to all of that, for the HUB unit bearing, an additional important parameter is the angular stiffness, as well as the stresses on the flanged outer ring. However, the angular stiffness can not be determined using standard methods for the outer ring, which is significantly larger than the inner ring. The outer ring represents the hub and wheel vehicles. The most reliable way of determining the angular stiffness of the HUB unit bearing is by using finite element method. Based on the drawings of the bearing, defined a 3D model of the HUB unit bearing is defined, as shown in Figure 4. Axial and radial effect of the rolling elements in rolling raceways (contact load) is defined in terms of contact pairs. Hence particular attention was given in order to ensure that raceways discretization remained the same.

Based on the previously prepared data on loads and stiffness values for each rolling body, in the processing part mathematical model was established (Fig. 4). Its solution is defined by the angular displacement of points A and B. Based on certain shifts in the points A and B, it is possible to determine the angular stiffness outer ring, and thus the whole bearing as [6]:

$$\begin{aligned} &u=(\delta_A - \delta_B) \\ &\theta=tg^{-1}(u/D) \\ &k_\theta=M/\theta \end{aligned} \tag{9}$$

where: u -angular displacement; A and B - displacement the points A and B, M-moment, D-diameter of outer ring acorrding to Figure 4

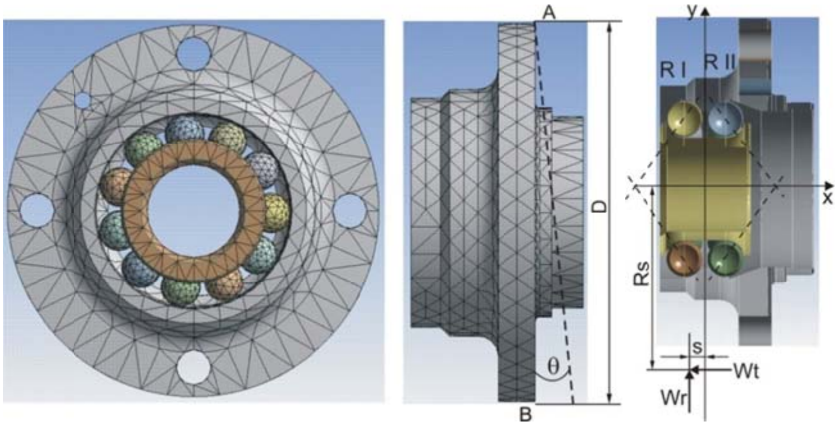


Fig.4. *Mathematical model for determining the angular stiffness*

3. RESULTS AND DISCUSSION

Analysis of the stiffness and stress of HUB unit bearng is made on the basis of reaction forces at the wheel of vehicles for a variety of conditions, lateral acceleration g = 0 to 1.2. The data in the table T.1 is the load acting on the wheel of the vehicle according to the values of lateral acceleration.

Table T.1 Load of bearing used in the analysis

Lateral acc. (g)	0	0,2	0,4	0,6	0,7	0,8	1,2
Wr [KN]	4.5	5.13	5.77	6.4	6.72	7.04	8.31
Wt [KN]	0	1.02	2.30	3.84	4.70	5.63	9.97
M [KNm]	-1.3	-1.4	-1.6	-1.8	-1.9	-2.0	-2.3

Due to the uneven distribution of the load and the contact angle on the rolling raceways, there is an unequal stiffness of raceways I and II. Since all the rolling elements on the raceway II are in the loading zone, greater stiffness is evident on the raceway II. Figure 5 shows the stiffness in the radial direction of the raceways I and II with lateral acceleration G=0.4. The sum of these two stiffness gives the overall stiffness depending on the lateral acceleration. Figure 6 shows the modified total axial and radial stiffness depending on acceleration. The axial stiffness of the HUB unit bearing decreased with increasing lateral acceleration from 0 to 1,2 with 595 to 234 [N / μm], which caused a decrease in the radial stiffness of 642 to 323 [N / μm]. The biggest drop in the axial and radial stiffness after a lateral acceleration is g = 0.4, after which it decreased by 56%. Therefore, for the inteval from g=0 to g=0.4, the stiffness is reduced by 9 %. Figure 7 shows the changes of angular displacements and angular stiffness depending on the lateral acceleration. Angular displacement of the HUB unit bearing growth by increasing lateral acceleration from 0 to 0.8 to 87%, with a reduction in angular stiffness by 46%. Angular stiffness to the lateral acceleration g = 0.4 decreases by 4%, followed by a sharp decline by 41%.

Figure 8a shows the distribution of the maximum equivalent stress in the flanged outer ring (forging) for g= 0,4 and figure 8b and 8c shows the distribution of the maximum equivalent stress in the flanged outer ring for g= 0,7 and g=1,2 respectively. The maximum equivalent stress occurs on the raceway of the flanged outer ring and is approximately 201 [MPa] for g= 0,4 or 332 and 401 [MPa] for g= 0,7 and g=1,2 respectively. By changing the lateral acceleration g = 0 to g =0.4 a slight increase in the maximum equivalent stresses on the flanged outer ring occurs, going from 112 to 180 [MPa]. In the area where lateral acceleration (g>0,4) are present, a significant increase of maximum equivalent stresses on the flanged outer ring occurs, going from 180 to 401 [MPa], as shown in figure 9.

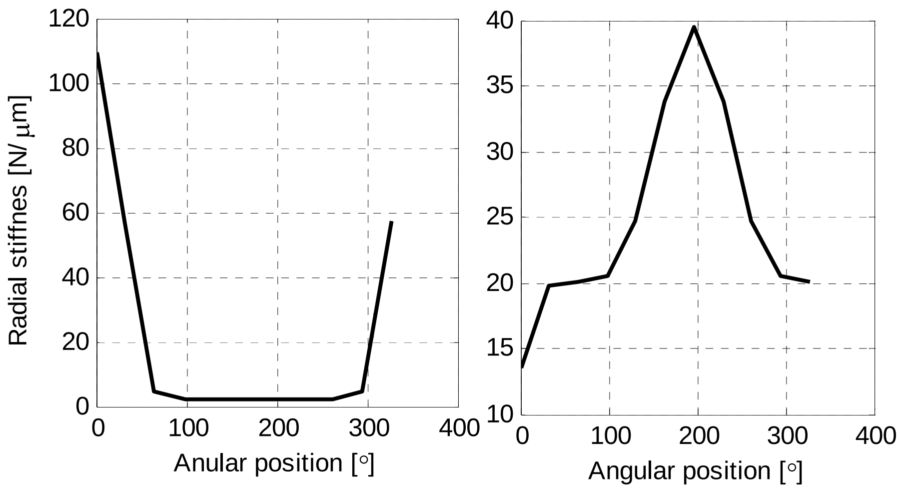


Fig.5. *Changing the radial stiffness of each rolling body with G = 0.4::a) raceway I (RI); b) raceway II (R II) according to Fig 4.*

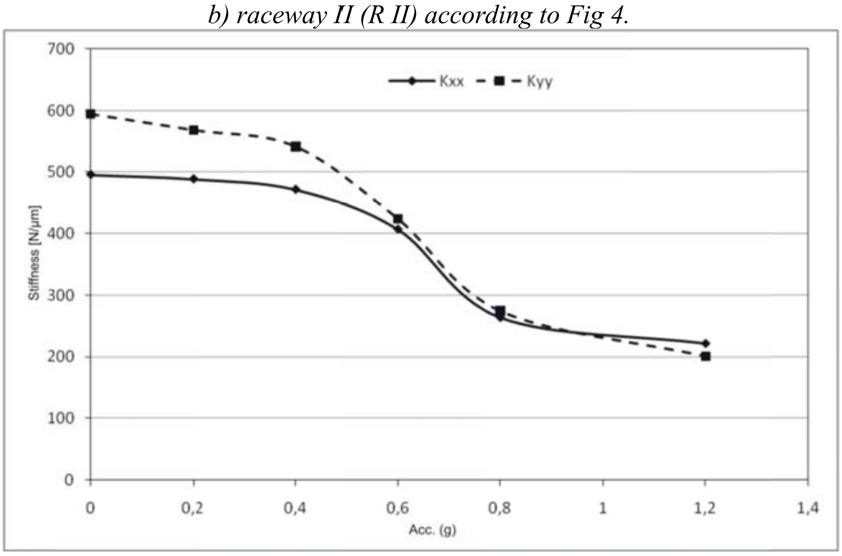


Fig.6. Changing the axial and radial stiffness depending on the lateral acceleration

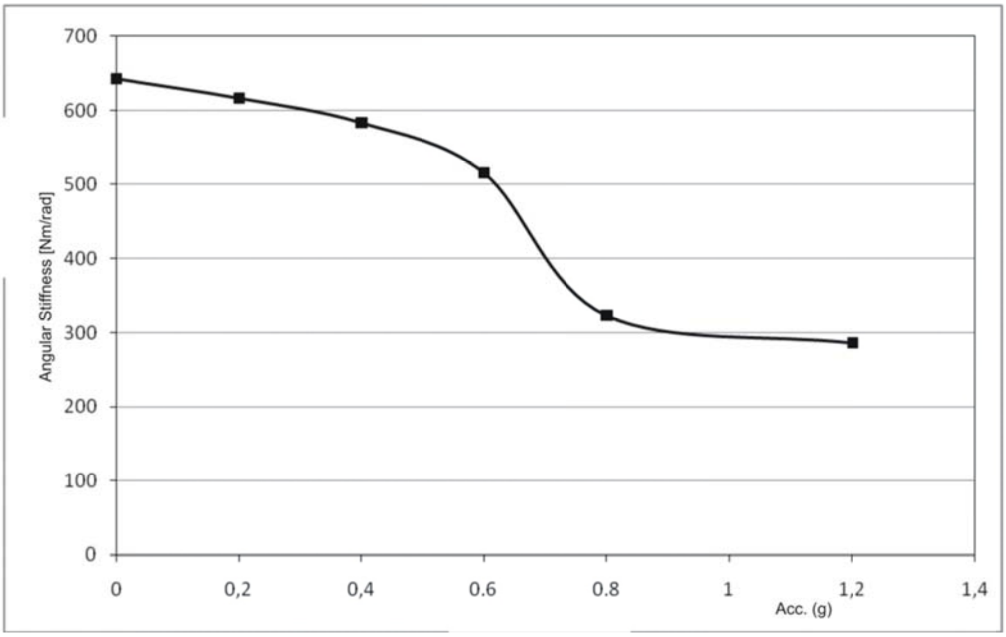
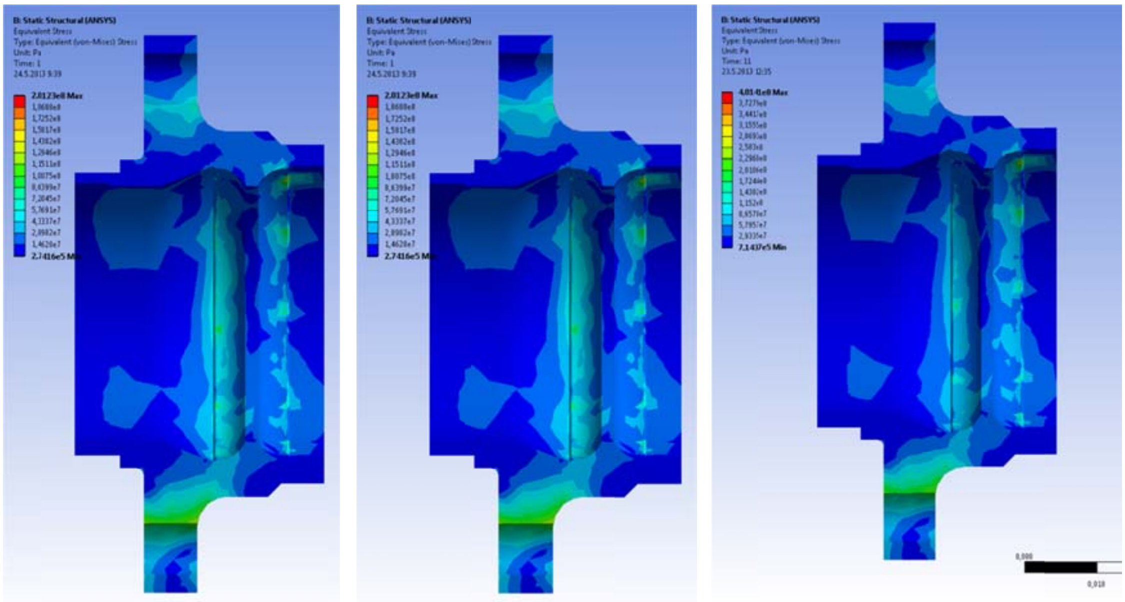


Fig.7. Change of angular stiffness depending on the lateral acceleration



a) *b)* *c)*

Fig 8. Disposition of maximum equivalent stress

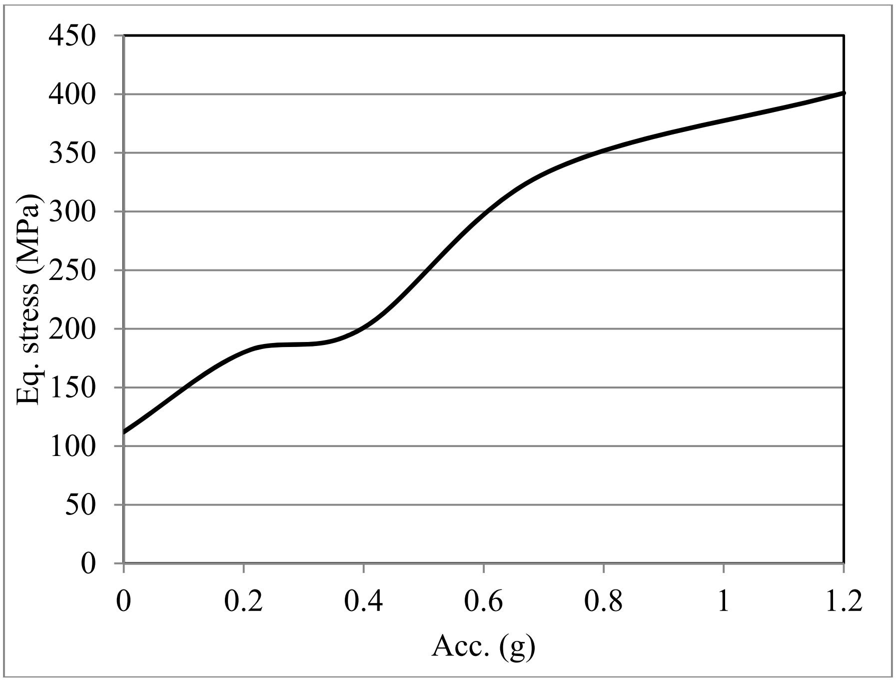


Fig.9. Change of maximum equivalent stress depending on the lateral acceleration

4. CONCLUSION

For of the HUB unit bearing, the centrifudal force will be smaller due to lower vechicle speed, so the difference between the contact loads on the raceways is below 3 % thus, the influence of inertial force can be negleted. Due to the uneven distribution of the load and the contact angle on the rolling slopes, there is an unequal stiffness bearings. The axial, radial and angular stiffness of the HUB unit bearing decrease with increasing lateral acceleration. On the other hand, an increase in lateral acceleration causes a significant increase in the stress on the flanged outer ring, which is made by forging.

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ANALZA STATIČKOG PONAŠANJA SPECIJALNIH KOTRLJAJNIH LEŽAJA SA KOVANIM SPOLJAŠNJIM PRSTENOM

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REZIME

U radu je prikazana analiza krutosti i napona integrisanog ležaja točka. Matematički model se sastoji iz dva dela. U prvom delu je razvijeno programsko rešenje za određivanje kontaktne krutosti za svako kotrljajno telo, U drugom delu je na osnovu kontaktnih krutosti određena promena ugaone krutosti ležaja i napona na spoljašnjem prstenu ležaja koji je izrađen kovanjem, primenom metode konačnih elemenata. Programsko rešenje je razvijeno na bazi Hertz-ove kontaktne krutosti i John-Harris-ovih kvazistatičkih jednačina ravnoteže. Matematički model je iskorišćen za analizu uticaja bočnog ubrzanja točka vozila na krutost ležaja i napone na spoljašnjem prstenu ležaja. Na osnovu prikazanih rezultata može se zaključiti da promena bočnog ubrzanja značajno utiče na smanjivanje aksijlane, radijalne i ugaone krutosti ležaja, dok povećanje bočnog ubrzanja, u velikoj meri povećava napone na spoljašnjem prstenu ležaja.
Ključne reči: *Integrisani ležaj točka, metoda konačnih elemenata, statičko ponašanje, otkovak*