UNIVERSITY OF NIŠ
The scientific journal **FACTA UNIVERSITATIS**
Series: **Mechanical Engineering** Vol.1, N° 6, 1999 pp. 665 - 674
Editor of series: Nenad Radojković, e-mail: radojkovic@ni.ac.yu
Address: Univerzitetski trg 2, 18000 Niš, YU
Tel: +381 18 547-095, Fax: +381 18 547-950
http://ni.ac.yu/Facta

STIFFNESS OF MACHINE TOOL SPINDLE AS A MAIN FACTOR FOR TREATMENT ACCURACY

UDC 62-113, 624.046

Momir Šarenac

Mechanical Faculty University of Srpsko Sarajevo

Abstract. *This paper presents study of basic constructional parameters that determine stiffness of main machine tool spindle. The possibility of optimal choice of bearings, inter-distances relations and console length as well as other constructional parameters is also emphasized. The main purpose is to obtain rigid spindle with acceptable cross sections.*

Key words: *stiffness of main spindle, bearing stiffness.*

1. INTRODUCTION

The main. purpose of usage of machine tools is to obtain high accuracy of part dimensions and shape in shorter time interval. Therefore, accuracy and productivity are closely related. Today's demands are pointed towards very narrow part tolerances: IT5-IT2 for dimensions, IT3-IT0 for shape, R_a = 0,3 to 0,003 μm for surfaces and $t_p \geq 60\%$ for carrying capacity.

In method of treatment centers with single part grip, complete treatment is achieved: scraping, milling, classifying, drilling and finishing, so the stiffness of the main spindle treatment center is very important. Moving the spindle end where cutting force is applied, is directly related with treatment accuracy, and together with part deformation defines accuracy of work-piece dimensions and shape.

Following the development of static model of machine tool main spindle, it is clear that the first model is being proposed by Schenk in 1939 (Fig. 1) [1].

This model represents elastic beams leaned on rigid fulcrums, loaded on the middle of span with concentric force. Displacement under cutting force is the determining factor for treatment accuracy, so Schenk and Pittroff gave new model of main spindle [2] again as an elastic beam on rigid fulcrums (Fig. 2).

Received January 14, 2000

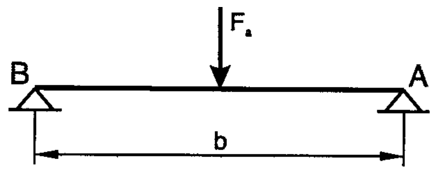

Fig. 1. Schenk's model

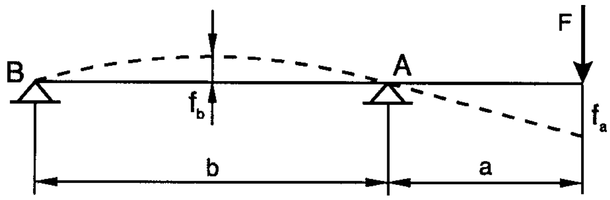

Fig. 2. Schenk's and Pittroff's model

Stiffnes of main spindle is defined by relation:

$$c = \frac{F}{f_a} \tag{1}$$

Influence of constructive elements placed on main spindle, on static model are given in [3], and model is presented on Fig. 3.

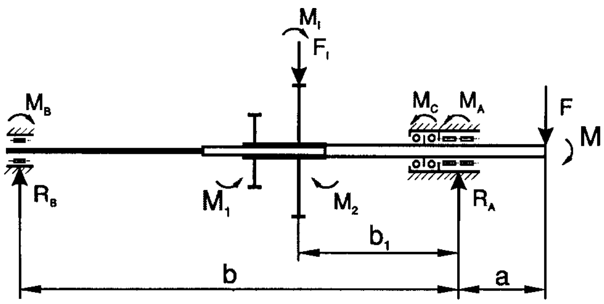

Fig. 3. Main spindle model with moments of reaction

All known stiffness criteria for machine tool main spindle can be divided into two groups: A - Constructional, B - Tecnological.

Constructional criteria enables normal working of gear pair placed in the middle of span or normal working of bearings in fulcrums. The most important criteria are Achercan, Decker, Angelov, Niemann, Reshetov, Fronins, Mamet and Pittrof criteria. Howevere, none of these criteria provides explanation of proposed values.

Among technological criteria, the most famous are Achecan and Zdenkovic criteria, where the highest accuracy demands stiffness of 400 N/μm.

Overall radial deformations at the end of console are given as a sum of deformations of elastic beam on rigid fulcrums (f_1) and deformations of rigid beam on elastic fulcrums (f_2), that is:

$$f_a = f_1 + f_2 \tag{2}$$

Fig. 4. represents a static model of main spindle without radial force in interspan and it can be used for spindle calculations for the majority of modern machine tools.

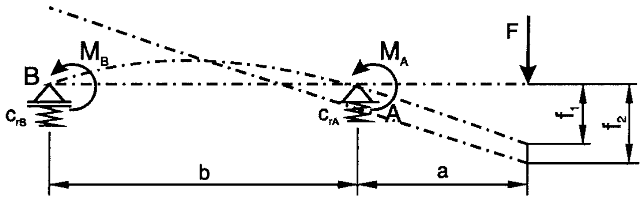

Fig. 4. Complex static model

2. Stiffnes of main spindle

Machine tool main spindle is the most important assembly that defines treatment accuracy and it should fulfill following criteria:

– high stiffness
– high rotation numbers
– wide rotation diapason
– minimum radial displacement
– low working temperature
– high reliability
– constructional compactness
– easy assembling
– acceptable price.

In construction of main spindle, the most important factors are the choice of bearings and basic geometry.

Some of the famous bearing manufacturers, such as FAG, has recently produced modular main spindles as a single assembly and place it on market. Depending on their purpose, these assemblies can be used for drills, lathes and mills. Every module is being made with roller-ball, hydrodynamic or hydrostatic bearing.

Table 1. lists basic characteristics of mill spindle with the most exact basic criteria.

Having in mind the basic purpose of mills, it can be said that high stiffness is most important, so double roller bearings are being used both on main and supplemental fulcrum.

In main spindle design, outer forces must be analyzed together with intensity and direction of each force.

Fig. 5 shows force schematic in scraping treatment.

When outer load is known, mechanic calculation model can be made.

Fig. 6 represents static model for calculation of stiffness of main lathe spindle.

Table 1. Mill spindless

Spindle head DIN55021 DIN2079	Roller bearing					Hydrodynamic bearing					Hydrostatic bearing				
	Label	Spindle diameter on cutting point	Radial stiffness on the end of spindle	Rotation/Displace-ment accuracy		Label	Spindle diameter on cutting point	Radial stiffness on the end of spindle	Rotation/Displace-ment accuracy		Label	Spindle diameter on cutting point	Radial stiffness on the end of spindle	Rotation/Displace-ment accuracy	
				rad	axi				rad	axi				rad	axi
	FAG	mm	N/μm	μm	μm	FAG	mm	N/μm	μm	μm	FAG	mm	N/μm	μm	μm
3	WZA3	50	300	4	3	HDK03	60	200	1	1	HSK03	55	130	0,5	0,5
4	WZA4	65	600	4	3	HDK04	70	280	1	1	HSK04	70	200	0,5	0,5
5	WZA5	85	900	5	4	HDK05	100	350	1	1	HSK05	100	330	0,5	0,5
6	WZA6	120	1000	6	4	HDK06	110	420	1	1	HSK06	120	550	0,6	0,6
8	WZA8	150	1800	6	5	HDK08	150	550	1	1	HSK08	150	750	0,8	0,8
11	WZA11	190	2150	7	5	HDK011	200	700	1	1	HSK011	190	1000	1	1

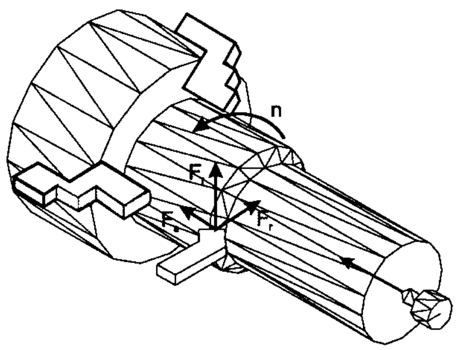

Fig. 5. Cutting force components in scraping treatment.

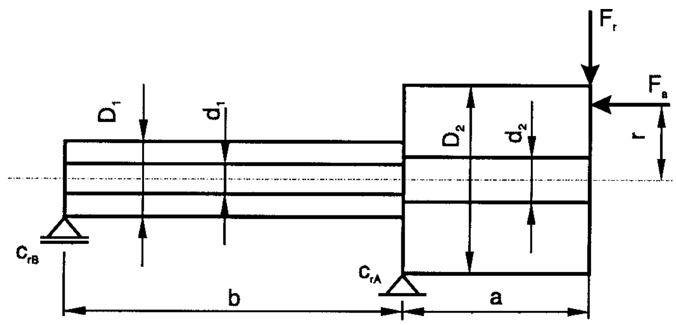

Fig. 6. Calculation model.

According to this model, deformations under cutting force are:

$$f_a = \frac{F_r \cdot a - F_a \cdot r}{3E}\left[\frac{ab}{I_1} + \frac{6v_1(m+1)a}{mA_1b}\right] + \frac{F_r a}{3E}\left[\frac{a^2}{I_2} + \frac{6v_2(m+1)}{mA_2}\right] - \frac{F_a \cdot r \cdot a^2}{2EI_2} +$$
$$+ [F_r(a+b) - F_a \cdot r]\frac{a+b}{b^2 \cdot C_{rA}} + (F_r \cdot a - F_a \cdot r)\frac{a}{b^2 C_{rB}}$$

where

f_a – displacement of console end	a – console length
F_r – radial force	b – inter-distance
F_a – axial force	I – moment of inertia
r – radius of F_a force	A – cross section
E – elasticity module	m – Poisson's coefficient
d – hole diameter	v – cavity factor
D – outer diameter	C_r – radial stiffness

Indexes:

1 – inter-distance	A – main bearing
2 – console	B – supplemental bearing.

Analyzing equation (3), it can be concluded that deformations are smaller with shorter console (a), but this distance has constructional minimum. It can be also concluded that deformations are smaller for larger cross section A_1 and A_2, that is moments of inertia I_1 and I_2. Cross section is also limited with top-heavy construction, so the cross section is being determined by bearing diameter, that are already standardized for given construction.

Having in mind these limitations, designer can enlarge main spindle stiffness on by:

– enlarging the bearing stiffness,

– choosing optimal b/a relation (distance factor kb = b/a).

3. Bearing stiffness

Bearing deformations (Fig. 7) can be calculated according to [7]:

$$\delta = kF^p \tag{4}$$

where: k – proportional coefficient
F – bearing force
p – exponent: 2/3 for ball bearings
2/3 – 9/10 for roller bearings.

However, bearing stiffness is:

$$c = \frac{F}{\delta} = F^{1-p} \tag{5}$$

Ball bearing deformation is a function of external force:

$$\delta_r = \frac{k}{\cos\alpha}\cdot\sqrt[3]{\frac{Q^2}{D_W}}, \quad \text{where} \quad k = 0{,}436 \tag{6}$$

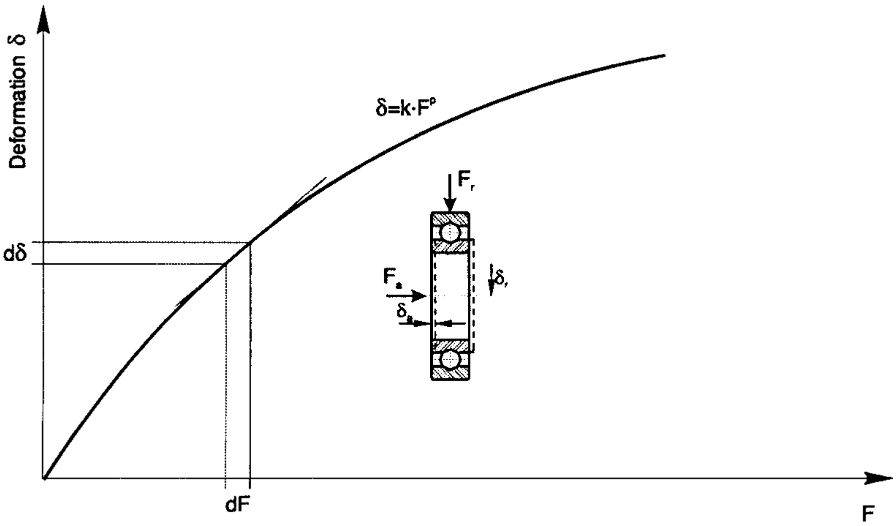

Fig. 7. Bearing deformation

$$\delta_r = \frac{k}{\sin\alpha}\cdot\sqrt[3]{\frac{Q^2}{D_W}} \tag{7}$$

Roller bearing deformations are:

$$\delta_r = \frac{k}{\cos\alpha}\cdot\frac{Q^{0,9}}{l_a^{0,8}}, \quad \text{where} \quad k = 0{,}077 \tag{8}$$

Value Q for radial bearings can be calculated according to:

$$Q = \frac{5F_r}{i\cdot z\cdot\cos\alpha} \tag{9}$$

where: δ_r – radial deformations, μm
δ_a – axial deformations, μm
D_W – ball (roller) diameter, mm
l_a – effective roller length, mm
α^0 – contact angle
z – number of balls
Q – single element load, N
F_r – radial force, N
F_a – axial force, N

Bearing stiffness is not proportional to bearing load, and therefore, is not constant value. It depends on force, bearing value and type.

Table 2 lists relative stiffness values for some bearing types for same inner cross section.

Table 2. Bearing stiffness comparison

Bearing type	Radial stiffness	Axial stiffness
NN30 K	1	–
NNU49 BK	1,3	–
N10 AK	0,5	–
N19 AK	0,65	–
719 CD/DBA	0,21	0,11
70 CD/DBA	0,22	0,1
72 CD/DBA	0,25	0,1
719 ACD/TB TB	0,31	0,39
70 ACD/TB TB	0,31	0,36
72 ACD/TB TB	0,37	0,37
320 X	0,47	0,53
2344 (00)	–	1
BTA-B	–	0,56
BTA-A	–	0,36

Bearing stiffness depends also on clearance. Fig. 8 represents bearing deformation as a function of clearance (Δ) and bearing force.

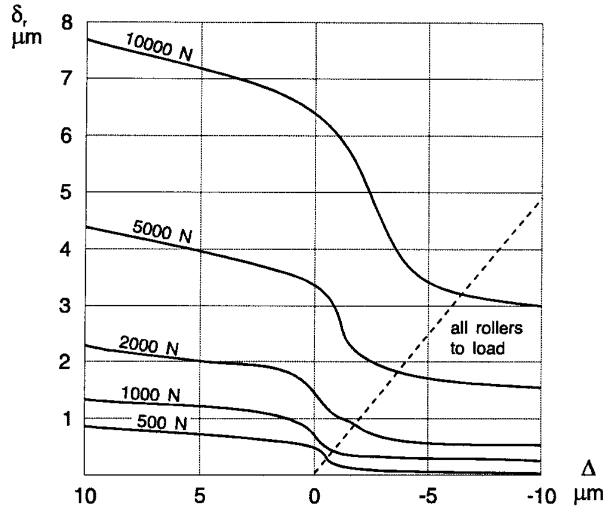

Fig. 8. Bearing deformation as a function of bearing clearance and force.

Bearing stiffness is also influenced by ball diameter together with way of ball tracking along inner and outer bearing ring, and ball material, because the ball carries the supreme load.

4. Factor k_b choice

Analyzing the equation (3), distance b can be determined (with presumption of other constant values) with the condition of the smallest console end deformation. According to DIN 2079 and DIN 55021, shaft diameter under main bearing is standardized and can be 50, 65, 85, 120, 150 and 190 mm.

For example, if console length is 90 mm, external load is 2000 N in radial direction and moment of 600 Nm for given bearing type, distance *b* can be determined so that stiffness is highest and inclination angle Φ_A is lowest.

Fig. 9 [6] represents stiffness and inclination angle function in main bearing for two given constructional bearing solutions.

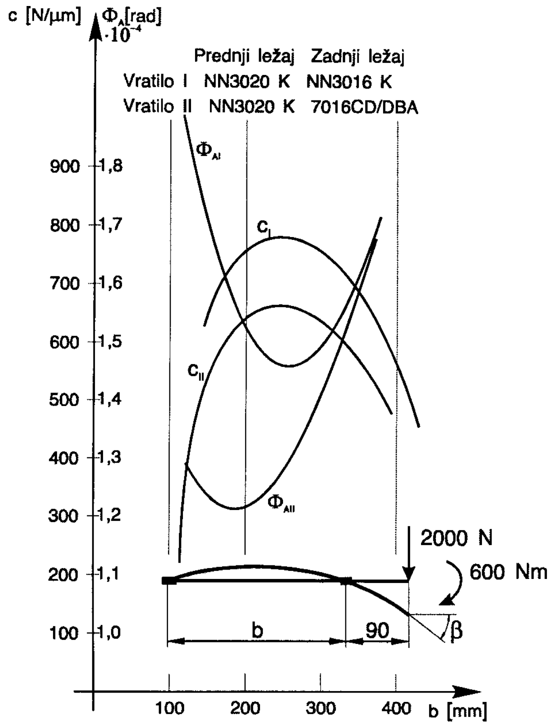

Fig. 9. Shaft stiffness as a function of distance and bearing type.

For further discussion of main shaft constructional solution, constructional coefficients are given as follows:

$k_a = \dfrac{a}{D_A}$ – console coefficient, $k_b = \dfrac{b}{a}$ – distance coefficient,

$k_d = \dfrac{d_0}{D}$ – shaft antrum coefficient, $k_{CL} = \dfrac{C_{rA}}{C_{rB}}$ – bearing stiffness coefficient,

$k_I = \dfrac{I_b}{I_a}$ – moment of inertia coefficient.

With this problem proposal, by changing each coefficient, stiffness (*c*), shaft end displacement (f_a) and inclination angle of main bearing (Φ_A) can be determined for every combination of basic shaft geometry values. Fig. 10 represents stiffness diagram of main spindle with NU222 bearings as a function of constructional coefficients.

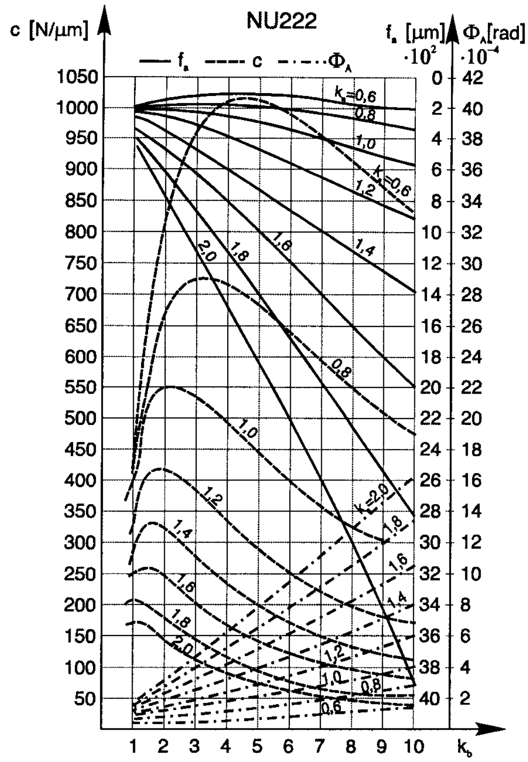

Fig. 10. Stiffness diagram of main spindle.

5. Conclusion

With assessment of length b or ratio b/a, stiffness can be easily determined. Stiffness criterion is directly opposed to velocity criterion, and that fact must be considered when designing machine tool main spindle. Final solution is always a compromise between these two basic criteria.

Higher spindle stiffness is achieved by choosing higher bearing stiffness and larger spindle cross section.

Dimensions a and b have direct influence on the accuracy of machine tool. Dimension a should be as small as possible, and dimension b (inter-distance) is being determined from condition of minimum sag under radial component of cutting force.

Higher radial forces is to be expected on machine tool spindles, so roller bearings should be used. Spindles with high rotation speed should be followed with ball bearings.

Stiffness diagram of main spindle enables choice of geometry-constructional characteristics necessary for spindle design.

REFERENCES

1. Zdenković, *Osnovi projektovanja i konstruisanja alatnih strojeva s aspekta tačnosti obrade i izvornih grešaka*, Strojniški vestnik XIII, Ljubljana, 1967-3.
2. Šarenac, M., Cvjetanović, M., *Uticaj momenta uklještenja na statički model glavnog vretena alatne mašine*, Zbornik radova, Znanstveno-stručni skup "Nauka o konstruisanju i konstruisanje pomoću računala", Zagreb, 1981.
3. Koch, J., *Die Spindelkonstruktion und ihre Steifigkeit*, Werkzeugmachine international, Ig. 1973, Nr. 1, 1973.
4. Šarenac, M., *Doprinos određivanju kriterija dozvoljenog nagiba alatnog stroja pri valjčastim uležištenjima*, Disertacija, Fakultet strojarstva i brodogradnje, Zagreb, 1984.
5. FAG, *Spindeleinheiten für das Bohren Drechen Frösen*, ubl. Nr. 02102 /2 DA
6. Šarenac, M., *Projektovanje vratila alatnih mašina po kriterijumu krutosti*, Zbornik radova IRMES'98, Beograd, 1998.
7. SKF, *Bearing systems for high performance machine tools*, 1992.

KRUTOST GLAVNOG VRATILA ALATNE MAŠINE ODLUČUJUĆI FAKTOR TAČNOSTI OBRADE

Momir Šarenac

U radu se obrađuju osnovni konstruktivni parametri koji određuju krutost glavnog vratila alatne mašine. Ukazuje se na mogućnost optimizacije izbora ležaja i odnosa dužine međurazmaka i dužine konzole i drugih konstruktivnih parametara s osnovnim ciljem da se dobije kruto vratilo uz prihvatljive poprečne presjeke.

Ključne reči: *Krutost glavnog vratila, krutost ležaja.*