ZESZYTY NAUKOWE POLITECHNIKI ŚLĄSKIEJ 1992
Seria: MECHANIKA z. 108 Nr kol. 1161

International Conference on
COMPUTER INTEGRATED MANUFACTURING
Internationale Konferenz über
RECHNERINTEGRIERTE FERTIGUNGSSYSTEME
Zakopane, March 24-27 1992

Ladislav JAVORČÍK, Peter JAVORČÍK

The Faculty of Mechanical Engineering
Slovak Technical University, Bratislava, Czechoslovakia

STATICS AND DYNAMICS OF MACHINE TOOL SPINDLES

Summary. The paper presents some results showing the difference between static and dynamic analysis and the model constraints to the number of element used, the number of load cases of the order and bandwidth of the stiffness matrix and mass matrix. The housing of the spindle for SUI 32 CNC has been investigated and some design for assembly recommended.

1. Introduction

In the field of machine tool building, instrument making and many other engineering activities ever increasing demands are placed on the spindle housing with respect to bearing performance, material spindle and an assembly system. These issues have been involving into the design for assembly, bearing arrangement, shape of the spindle, the speed limit, working accuracy and the other requirements such as lubrication, life adjustments, the type of load, heat generation, static deflections and dynamic responses, installation and replacement of bearings, etc.

In general, due considerations in the spindle design depend upon a number of factors to be sometimes in contradictory actions. some of them are:
- the magnitude and direction of the load vector,
- the range of speed - maximum and minimum of revolutions,
- the deflections limited corresponding to accuracy,
- the housing space available or mass configuration in,
- the type of technology used, etc.

More detailed information can be obtained in the SKF, GMN, FAG, TIMKEN Co. or the ZVL Catalougues or in the specialized papers [1], [2], [3] and others.

2. Theoretical Approach

According to the main demands, the designing process can be divided on: designing for - accuracy, stiffness i.e. pre-loading of bearings, spindle stiffness, speed limit, heat generation, cooling conditions, component accuracy, lubricant and lubricant system, etc. An important role makes, when aiming for high speed and stability of the cutting process, the dynamic analysis of the structural system in terms of mass, damping and stiffness configuration.

An approach based on the equilibrium equations for complex structural systems reflecting a linear system of structural elements can be derived by several different approaches [4], [5], [6]. All methods yields a set of linear equations of the following expression

$$\mathbf{M}.\ddot{u} + \mathbf{C}.\dot{u} + \mathbf{K}.u = \mathbf{R} \qquad (1)$$

Where **M** - is the mass matrix, **C** - is the damping matrix and **K** - is the stiffness matrix of the elements assemblage, u, $\dot{u}$, $\ddot{u}$ - and **R** are the nodal displacement, velocities, accelerations and generalized external loads, respectively. The structure matrices are formed by direct addition of the element matrices - for example

$$\mathbf{K} = \mathrm{SUM}\ (Km) \qquad (2)$$

Where Km is the stiffness matrix of m-th element. Although Km is formally of the same order as K, only those terms in Km (which pertain to the element matrices) can therefore be performed by using the element degree of freedom is zero. In the programme used, the stiffness matrix and the diagonal matrix are assembled.

Therefore, a lumped mass analysis is assumed where the mass structure is the sum of the individual element mass matrices plus additional concentrated mass are specified at selected degree of freedom.

The damping is assumed to be proportional and this quantity is specified in the form of a modal damping factor. The assumptions may be discussable. But generally the modal damping factor for a housing of ball bearing with angular contact used makes less than 0.2 and so a simplification on linearity can be accepted.

A solution through the Rayleigh-Ritz method is available rather for an elementary case and the using this approach brings also only estimated results.

Boundary conditions can be expressed such as

a/ if a displacement component is zero, the corresponding equation is not retained in the structure equilibrium equations and the corresponding to element stiffness and mass terms are disregarded.

b/ if a displacement is non-zero then the equation

$$ui \cdot k = kx \tag{3}$$

It should be specified at a degree of freedom (i), giving

$$ui = u$$

and the eq. (3) is added to the eq. (1), where $K \gg ki$.

This mathematical operation can be interpreted as an adding at the degree of freedom (i) to a spring of high stiffness (k) and specifying a load (R) which is consequenting to the required displacement (x).

3. Programme Architecture for Calculation

Calculation of the structure stiffness matrix and mass matrix can be provided by the Programme: A Structural Analysis Programme - SAP [7] in three sections:

The 1st section: the modal point input data is red up and generated by the programe. The equation number for the active degrees of freedom at each nodal points are established.

The 2nd section: The element stiffness and mass matrix are calculated together with their connection array

The 3rd section: The structure stiffness and mass matrix are formed by addition of the element matrices.

This programme architecture is being independent of the element type used and is the same for either a static or dynamic analysis. As for the static analysis the programme "TUHOST-STIFFNESS" [8] has been applied and the results compared to the SAP-exc. gained.

Effort to develop improved interfaces are currently under way at a host FEA (Finite Element Analysis) vendors. Although one-way coupling is sufficient for many analyses nature of the problem should be in interfacing FEA software two-way coupling because of greater interaction between forces such as when a deformation caused by heat could generate enough heat in itself to cause more deformation.

It should be pointed out that thermal loads affect a total displacement itself may or need not have a negligible impact on temperature. The used method is a one-way coupling.

4. A Case Study - The Housing of the Spindle of SUI 32 CNC

The machine tool SUI 32 CNC (made TOS Galanta - Czechoslovakia) is a modular conception lathe made for higher accuracy of for very high accuracy if needed. This machine is comparable to MASTER 2500 COLCHESTER - UK or FARTEL 2D 160 (Swiss).

The Headstock body has a box shaped dimension made of gray alloy. The spindle is supported on the ball bearings with angular contact, Fig. 1. The front bearing node arrangement consists of the bearings SEB 110 - Tandem plus face-to-face and the rear node represent the bearing SEB 90 in face-to-face arrangement. The bore of spindle makes 80 mm.

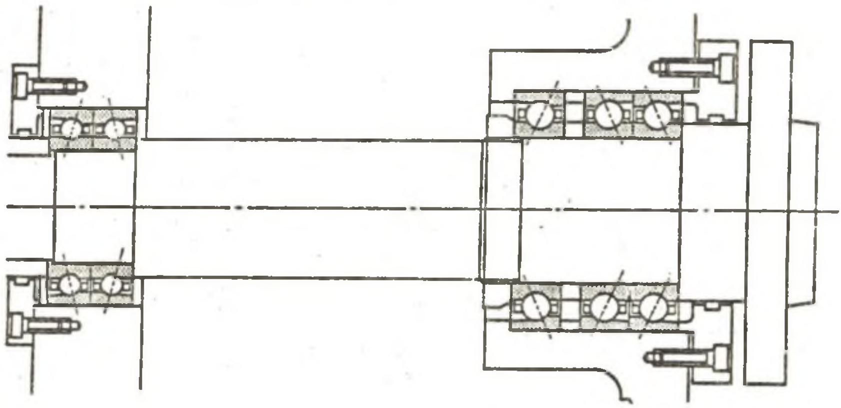


Fig. 1 Bearing Arrangement of the Spindle Housing

A geometrical configuration of the spindle and bearing arrangement can be shown in Fig. 2.

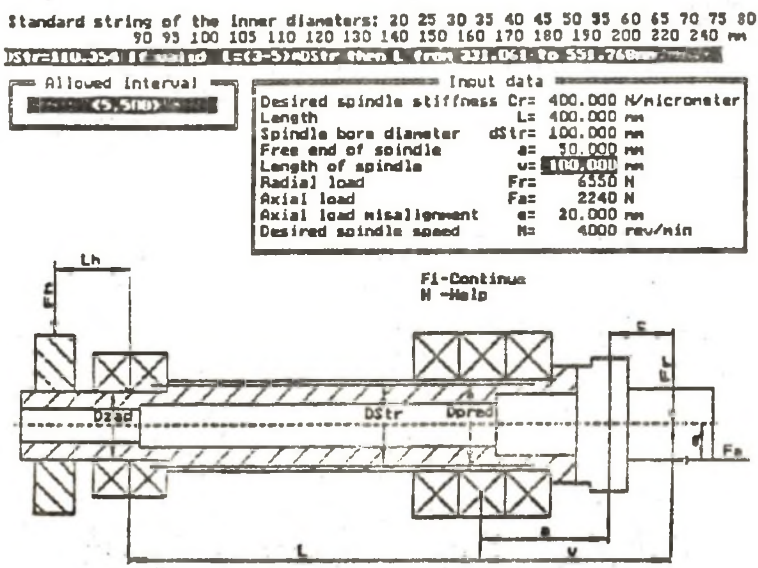
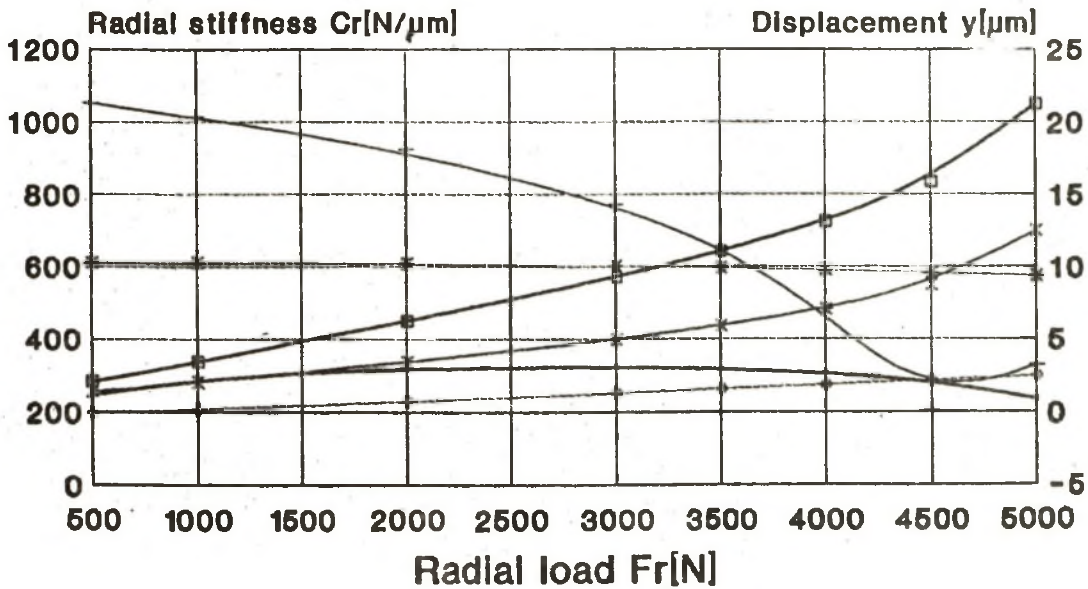


Fig. 2 A Geometrical Configuration

Radial stiffness\displacement = f[F]



Fh = 1000 N Fig.3

TND-360

4.1. Static Analysis

A static analysis can be taken as a special dynamic response under condition when frequency of the external acting forces is equal zero. The progamme TUHOST-STIFFNESS enables to calculate all geometrical data of the spindle and the deflections caused by the radial/axial forces and torque. The acting forces are determined through technology, the power and the place-point of the force vector. Some result are plotted out in the graph Fig. 3. The programme TUHOST-STIFFNESS, Release 2.0 is available for commercial purpose under specification [8].

4.2. Dynamic Responses

A Dynamic response for a mechanical system can be characterized by the natural mode and the first two modes of frequencies are usually dominant. According to the model based on the scheme of a beam supported upon two flexible nodes [9] with boundary conditions specifying two degree of freedom system, the result - the first two natural frequencies are

OMEGA (1) = 775 Hz
OMEGA (2) = 1787 Hz

In spite of the model simplification the results can be accepted such as the first approximation to the real system. Using the method of FEA after the appropriate spindle and bearing discreditation the first three eingenvalues or natural frequencies are

OMEGA (1) = 658 Hz
OMEGA (2) = 1133 Hz
OMEGA (3) = 1158 Hz

In both cases of calculation the input data for bearing node was the same values received such as the total stiffness of elements in the parallel connection.

4.3. Comparing the results

The results gained through two models of the spindle system arrangement allow to express the following conclusions:

1. All results are comparable each to other and natural frequencies/eigenvalues difference is not significant from the practical use.
2. Each natural frequency/eingenvalue is so far beyond the limit frequency - approximately 200 Hz. From this aspect the design of housing the spindle seems to be correct.
3. Frequency of spindle rotation thus has no impact on the process stability at machining.

The fact that the cutting forces/load has a dynamic character can be taken as more important. The time responce of the cutting force is going to be the input of the modified model.

Comparing the results obtained of the programme TUHOST-STIFFNESS and of the programme SAP-IV can be shown in TAB.1.

TAB. 1

LOAD / DEFLECTION	Fr = 1 kW		Fr = 2 kW		Fr = 3 kN	
	STF	SAP	STF	SAP	STF	SAP
yz [μm]	-0.55	-0.64	-0.77	-0.81	-1.66	-3.44
yp [μm]	1.26	1.40	2.46	2.81	4.02	4.74
yTOT [μm]	2.67	2.47	4.57	5.07	8.31	8.35

Remark: STF - Pgm TUHOST-STIFFNESS
SAP - Pgm A Structural Analysis

Where : yz - the deflection under the rear support point
yp - the deflection under the front support point
yTOT - the deflection at the spindle nose

Comparing the results in relative deviations all values can be accepted but the mentioned one-way interfacing FEA software is a certain simplification because for example temperature of the spindle and bearings may affect the total deflection at the spindle nose. From this aspect it is necessary to take into account two-way coupling concept of calculation if the comparing to experimental data should be correct.

5. Consulsion

The static and dynamic stiffness of the spindle with ball bearing arrangement play an important role in the working accuracy and performance and surface quality of the workpiece.

The computer programme TUHOST-STIFFNESS and SAP have been used and some static and dynamic responces of SUI 32 CNC compared. The deformation of the spindle housing can be ignored. The relative proportion of the other parts depends mainly on the type of bearing node arrangement and the spindle design. The conditions such as pre-loading of bearings, the limit speed, lubrication and assembly are also important and involved into the programmes. The programme SAP based on the FEA method has been used as a one-way coupling and the results obtained can be implemented using translators that convert the output data from the FEA on the force deformation into properly formatted input for the second analysis for example the FEA on the termal influence and vice versa.

On the base of the results some recommandations for the spindle design and bearing arrangement has been made [9]. The programme SAP is going to be adapted in meshing on p-code (polymonial order up to 8 or 9, typically) because so called h-ellements (bricks, wedges and three or four-sides plates or shells) are analyzed only to linear order although most h-codes offer parabolic element.

REFERENCES

[1] Fröhlich J.: Technika uložení s valivými ložisky (Housing techniques with rolling bearing). SNTL/ALFA Praha 1980, p. 445.

[2] Šooš Ľ.: Statika uloženia vretien obrábacích strojov. (Statics of the Noded Spindle Housing of Machine Tools). Disertation - STU Bratislava 1990, p. 140.

[3] Javorčík L. and Co.: Applied Software Technology for Designing a Spindle Housing for the Lathes made in TOS Trenčín - Czechoslovakia. Research Report HZ-33/90-30/66, STU Bratislava, Part I.-1990, Part II.-1991, dissc-VRETENNÍK - Release 2.0.

[4] Algor Interactive Systems Inc., Pittsburg - VisiCAD Plus Supper Draw II.

[5] PDA Engineering - The PARTAN MCAE

[6] MacNeal-Schwendler Corp. - MSc./Dyna.

[7] Bathe K. and Co.: SAP-IV. A Structural analysis Programme for Static and Dynamic Responce of a Linear System. Berkeley, California 1973.

[8] Šooš Ľ.: Applied Software - TUHOST/STIFFNESS, Release 2.0.

[9] Javorčík P.: Dynamika vretien obrábacích strojov. (Dynamics of Machine Tool Spindles). M.Sc.Dissertation 1991.

STATISCHE UND DYNAMISCHE VERHALTENS DES SPINDEL DER WERKZEUGMASHINEN

__Zusammenfassung__

Der Beitrag bescheftigt sich mit Problem der Dynamik von Bearbeitungsmaschine - Spindellagerung und der Dynamische festtigkeit mit Methode FEA (Programm SAP) und Methode Übertragungs Matrizen (Programm TUHOST) berechnet. Auf Grund gewonenen Ergebnisen sind einige aktuälle Änderungen von Konstruktions konception empfohlen.

STATYCZNE I DYNAMICZNE ZACHOWANIE SIĘ WRZECION W OBRABIARKACH

Streszczenie

Artykuł zajmuje się problematyką dynamiki obrabiarki, łożyskowania wrzeciona i dynamiczną wytrzymałością z zastosowaniem metody elementów skończonych (program SAP) i metodą przetwarzania macierzy (program TUHOST). Na podstawie uzyskanych wyników wprowadzono aktualizację poprzez zmiany konstrukcyjne.

Wpłynęło do redakcji w styczniu 1992 r. Recenzent: Jan Kosmol