Survey on stiffness and damping of machine tool elements

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SURVEY ON STIFFNESS AND DAMPING OF MACHINE TOOL ELEMENTS

by

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SUMMARY

The stiffness and damping of elements as bearings, slideways, joints etc. has to be known in order to predict completely the behaviour of a machine tool by means of Computer-Aided-Design. This paper aims to draw up an inventory of the work which has been done on this stiffness and damping in the several institutes.
1. INTRODUCTION

In Computer-Aided-Design of machine tool structures it is almost common practice to consider the machine tool as consisting of a number of beam elements. In this way of thinking it is assumed that the connection between two elements of the model of the machine tool is rigid for every degree of freedom, or in case of a hinge is rigid for some degrees of freedom. As a result the division of a machine tool in elements is often more directed to "easy modelling" than to "reality".

The input for the computer program consists of the geometrical properties of the elements and the material properties such as modulus of elasticity, shear modulus and specific mass. The damping in the material can be neglected.

Also the co-operative work in Computer-Aided-Design of machine tools within CIRP Technical Committee Ma follows the procedure mentioned above *)).

This way of modelling passes the reality of the machine tool which is built up out of a number of elements connected by means of elements as joints, bearings and slideways. In a correct model these connecting elements should be represented by springs and dampers.

Although the modelling mentioned above is basically wrong, the results of the analysis are not always useless. Up to now, there are roughly three reasons for leaving out of the model springs and dampers representing the connecting elements:

1st there are rather few - reliable - numerical values of these springs and dampers available,

2nd the way in which these data are available, is largely unsuitable for direct use in CAD,

*) COWLEY, A : Co-operative work in Computer-Aided-Design in the CIRP.
The introduction of damping in the existing programs makes these programs more complicated - and last but not least - more expensive.

The aim of this publication is to draw up an inventory of what has been done on stiffness and damping of the connecting parts of machine tools. The inventory concerns mainly the work of the following members and their co-workers:

Professor J.G. Bollinger,
University of Wisconsin, Madison, U.S.A.,

Professor F. Koenigsberger and Dr. R. Bell,
University of Manchester, Great Britain,

Professor H. Opitz,
T.H. Aachen, West Germany,

Professor J. Peters and Mr. P. Vanherck,
University of Louvain, Belgium,

Professor A.C.H. van der Wolf and Mr. J.A.W. Hijink,
Eindhoven University of Technology, The Netherlands.
2. CATEGORIES OF CONNECTING ELEMENTS

2.1 Machine tool joints

On the static and dynamic behaviour of constructions joints have an important influence. A very good overall view on the work done until this moment can be found in the survey paper on joints by BACK, BURDEKIN and COWLEY (1). They describe how almost all the work on joints was concentrated upon the surface stiffness, which depends upon: the materials in contact, hardness, machining process, surface roughness, relative orientation of the surface layers, size of the contact area, flatness deviation and the contact pressure. DOLBEY, BELL (2) et. al. give for a number of materials and surface conditions values with which the surface stiffness can be calculated. Experiments in order to obtain these data were often done on small testpieces which were rather rigid and up to now there is no model in which all the factors influencing the surface stiffness are considered. BACK (3) however did incorporate the surface stiffness characteristics into non rigid finite elements models of a number of different joints. He describes three methods to enclose the surface stiffness between the two surfaces of the joint, particularly the hydrostatic-, plate- and spring method. Using the plate- or spring method the surface is considered as a separate non linear element and with an iterative way the deflections and pressure distribution are calculated. A very good correlation was found between the theoretical and experimental deflections. A very important conclusion of Back is that for most of the joints the surface deflections contribute only for approximately 10 percent to the overall deflection (see Fig.1). This means that it is almost impossible to predict the stiffness of a joint only by taking into account the surface stiffness. However calculating every joint with the finite elements method will be expensive.

To calculate bolted joints PLOCK (4) gives a method starting from the stiffness of a flange-bolt combination and the surface stiffness of the flanges. Knowing the outline of the joint and the distribution of the bolts, it is possible to calculate the total contact area the tension in the bolts and the local deformations. In this method the global deformation
of the flanges can not be calculated and the results as shown in Fig. 2, are only satisfactory for rather stiff flanges and bolts placed almost in line with the walls.

BACK, BURDEKIN, COWLEY (1) and GROTH (5) drew the following conclusions about the damping in joints:
- the dynamic and static stiffness for dry surfaces in contact is the same, the damping is negligible,
- the introduction of a lubricant adds damping to the structure, the damping coefficient is independent of the normal load,
- the damping coefficient for joints with metallic contact depends on the pressure distribution of the surface, surface condition and viscosity of the lubricant,
- the mechanism of the normal damping is analogous to squeeze film damping of a lubricant film.

About damping in joints almost no data are available, and up to now it is impossible to predict the damping coefficient when geometry, pressure and viscosity are known.

2.2 Plain Slideways

The static and dynamic properties normal to the sliding motion of plain slideways are almost the same as those of joints. The same difficulties and possibilities in establishing values for the normal stiffness occur.

In the direction of sliding the dynamic characteristics are mainly determined by the damping caused by friction, the mass of the carriage and the stiffness and damping of the drive. The damping caused by slideway friction is nonlinear and may be either positive or negative, which means stable sliding or self-excited oscillations caused by stick-slip. BELL and BURDEKIN (6), (7), (8), (9) have published the dynamic behaviour of a testrig representative of a machine tool. They concluded that dynamic measurements are essential to obtain values for the damping. Besides the mass of the carriage, the stiffness of the drive and the coefficient of friction, the carriage speed has an important effect on the stability and damping of the carriage (see Fig. (3)). Because the stability and damping is a complex function of a number of parameters BELL and BURDEKIN (8) thought that there is little merit in
the development of a complex analytical model. The influence of time dependent variables are difficult to accommodate in definite terms.

BRITTON and BELL (10) examined the influence of design parameters on the stability of feed drives. Their principal conclusions were:
- the drive stiffness is a dominant factor,
- the influence of lubricant viscosity is important,
- the drive natural frequency has no primary influence on the stability of sliding motion. The drive stiffness and table mass must be considered as separate parameters.
- Effective design solutions may be achieved when the critical velocity is below the minimum feed speed or when the amplitude of vibration is not detrimental to machine performance.

To determine the resonance curves of machine tool feed drives for a number of coefficients of friction and sliding speeds KALS (11) and HOOGENBOOM (12) give analogue models. The only difficulty here is to obtain proper values for the coefficient of friction.

2.3 Roller bearings and roller guideways

Based on the Hertz theory Palmgren developed a set of formulae to calculate the deflections of all common types of bearings subjected to radial or axial loading. In these formulae the only unknowns are the number of rolling elements, the dimensions of the rolling elements and their contact angle. The formulae give only a fair result for bearings without clearance and no preload.

GÜNTER (13) did research on cylindrical-roller bearings. He examined the influence of radial clearance, accuracy and preload on the radial stiffness of the bearings and made nomograms for two types of bearings.

Thrustbearings, which are an important part of the feeddrives of NC machines, have been examined by BELL and KIMBER (14), (15). They concluded that the use of high preloads ensures higher stiffness of the bearing. Bearings with a relatively large number of rolling elements give a higher stiffness but a reduction of the load carrying capacity. In the case of double thrustbearings the influence of temperature is a secondary factor.

On the clamping stiffness of taperroller bearings some work has been done by ELSERMAN (16) DEBRABANDERE and VAES (17). Tests were carried out
with a rotating and nonrotating bearing loaded by a static moment. Experiments showed that, due to frictional resistance, the clamping stiffness of a nonrotating bearing is 20-30 times higher than the clamping stiffness of a rotating bearing (see Fig.4). Between 300 and 1100 cycl/min the clamping stiffness does not change. A theoretical model is developed to calculate the clamping stiffness as a function of the preload.

To provide an effective antifriction guideway, rolling element guideways can be used. DE FRaine (18), (19) studied the static stiffness of ballslideways and its relation with the load eccentricity. He showed that the load-capacity decreases very quickly with the eccentricity; the decrease in the stiffness is not as critical.

Because of the fact that below a certain frequency the dynamic stiffness of roller guideways without dampers is lower than the static stiffness, Hallowes and Bell (20) added fluid film dampers to the guideways and became good dynamic results. The results depend on the thickness of the squeeze film and the viscosity of the oil but a general increase in the dynamic stiffness is always attained.

2.4 Hydrostatic bearings and -guideways.

In the past decades many constructions for hydrostatic thrust- and journal bearings have been developed. However, the design was almost only concerned with the static behaviour i.e. stiffness, maximum load and flowrates.

Böttcher, Effenberger and Opitz (21), (22) and also Cowley and Kher (23) investigated the dynamic behaviour of hydrostatically supported spindle-bearing systems. Böttcher used a more complicated spring-damper model for the bearing (Fig.5), while Cowley used a one spring-damper model. They showed that it is possible to optimize the stiffness and damping for the spindle-bearing system (Figs. 6 and 7). By only stiffening the bearings the system can get a very bad dynamic response.

The damping and stiffness of a hydrostatic bearing can be changed in a wide range by varying the pressure and/or dynamic viscosity of the oil. In this respect the hydrostatic bearing has much advantage on the roller bearing.

Furthermore, Koenigsberger and Cowley (24) gave a critical review of
the work done on the dynamic behaviour of thrust and journal bearings including squeeze film damping, compressibility and inertia effects.

On hydrostatic guideways PORSCH (25) did some research on the static stiffness of guideways with different frames.

2.5 Aerostatic bearings and guideways.

WARNECKE (26) has given a very good survey on the static- and dynamic characteristics of all kinds of aerostatic bearings. Aerostatic bearings have advantage when used for very low friction and high accuracy. Attaining a sufficient load capacity and stiffness are the most difficult problems, caused in particular by the low dynamic stiffness and pneumatic instability.
3. DAMPERS

In machine tool structures dampers have a special function. Only when the dynamic behaviour of the machine tool leaves a certain range, one will try to change this dynamic behaviour with the help of a damper. So in CAD a damper never can be implemented in the first stage of a design. Only when the dynamic behaviour is known one can give the proper dimensions to the damper.

Two kinds of dampers can be distinguished: the passive damper and active damper.

3.1 Passive dampers.

The passive dampers can be subdivided in the following systems:

a) a viscous damper between two machine parts,
b) damped added mass,
c) Lanchester damper,
d) Impact damper.

In the cases a and b the fluid film damper is often used as damping element. VANHERCK (27) gives a description of the dimensioning of flat and circular fluid film dampers. Also plastics can be used as element with damping and spring characteristics. With the help of special dynamic CAD programs these dampers can be implemented in the machine tool structures in order to see how the results are.

The impact damper can only be calculated with help of an analogue computer.

3.2 Active dampers.

BECKENBAUER (28) developed an active damper for machine tools. When using an active damper, energy is added to the total system. Therefore it will be impossible to treat this damper with CAD.
4. CONCLUSIONS

Much work has been done on the static and some on the dynamic behaviour of "connecting" elements in machine tool structures. In spite of this work, only for the static stiffness of roller bearings, and hydrostatic- and aerostatic bearings data are available for direct use in CAD. For the normal stiffness of joints and plain slideways every joint has to be calculated with a special purpose program. On damping there is some knowledge on hydrostatic and aerostatic bearings and guideways which could be used in CAD.

The knowledge of the damping of joints and plain slideways is very specific and there is no general model available for all kinds of joints and guideways.

In table 1 a survey of the usefulness of the different categories is given.

As fields of research are recommended investigations into:

1st the stiffness of joints and guideways, in a general way,
2nd the dynamic stiffness and damping of joints and plain guideways,
3rd the dynamic stiffness and damping of roller bearings.

Remark: The references are only from the institutes mentioned in the introduction. Of course there has been done much more research on this subject within other institutes. Particularly in ref. (1), (24) and (26) one can find a survey on the work done in some special fields.
REFERENCES


<table>
<thead>
<tr>
<th>Category</th>
<th>Stiffness</th>
<th>Usefullness in CAD</th>
<th>Damping</th>
<th>Usefullness in CAD</th>
</tr>
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<td>Machine tool joints</td>
<td>known, but only specific knowledge of static stiffness</td>
<td>by means of special finite element programs</td>
<td>something known</td>
<td>not usefull</td>
</tr>
<tr>
<td>Plain slide ways</td>
<td>some specific knowledge</td>
<td>by means of special finite element programs</td>
<td>something known by means of analogue models</td>
<td>not yet, to specific</td>
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<td>Roller bearings and guideways</td>
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<td>usefull</td>
<td>unknown</td>
<td>-</td>
</tr>
<tr>
<td>Hydrostatic bearings and guideways</td>
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<td>Dampers</td>
<td>-</td>
<td>-</td>
<td>known</td>
<td>only passive dampers</td>
</tr>
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</table>

**TABLE 1. Survey of the categories**
Fig. 1. Deformations (1000x) of a joint using the spring method (after BACK (3))

Fig. 2. Comparison between calculated and measured deflections, for different preloads $F_v$ on the bolts (after PLOCK (4))
Fig. 3. Slideway damping characteristics for different slideway materials and Tonna 27 as lubricant (after BELL (9))

Fig. 4. The clamping stiffness of a taperroller bearing (after DEBRABANDERE (17))
Fig. 5. Hydrostatically supported spindle with its equivalent calculable model (after BÖTTCHER (21))

Fig. 6. Damping ratio and resonance amplitude of a system as a function of the compressibility stiffness of the front journal bearing (after BÖTTCHER (21))

Fig. 7. Variation of damping ratio with damping constant for various normal flexibilities (after COWLEY (23))