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Milling machine tool stability and performance

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Abstract

Machine tool performance can be strongly limited by the dynamic properties of a machine structure. There is a well-known problem of chatter occurrence. An analysis of a particular case is not always easy because of the modal complexity of such a structure. This paper reports on the analysis carried out at a few milling centers looking at tools and spindles and their influence on stability levels during cutting. Different kinds of work-piece material are considered as well as low and high cutting speeds. The synthesis of machine structure properties and tool stability properties can be used either to improve a spindle unit design or to choose and fit tools to an existing spindle, both with regard to improving their use. The approach is shown in the examples discussed below.

1 Introduction

A significant chatter can be detected during milling processes of any kind under certain cutting conditions. The chatter is, actually, self-excited vibration between tool and work-piece. The cutting process is unstable during the chatter. To avoid this instability, a machine tool designer must first consider the stiffness and damping of the spindle unit, including the tool. The user of a machine tool must know how to choose cutting parameters in order to avoid the chatter. Normally, a feed rate, cutting speed, radial depth of cut and axial immersion are chosen according to the tool technical guide. But these parameters are strongly dependent on the dynamic properties of the tool, as well as the spindle. It is really difficult to calculate all the dynamic properties of a spindle. On the other hand, it is not so difficult to measure these properties on a machine tool at the time of creating the part programme.

A predictive approach can be applied to the testing of a machine tool and its tooling. The specialized SW package and device Metalmax, developed by

Manufacture Laboratories Incorporation, USA, is available now. This measurement is based on frequency transfer function pick-up and further calculations. For the full theoretical studies, first of all the book [1] by J.Tlusty is recommended as well as all works by S.Smith, Th.Delio and the others, [2], [3]. The Pulse system and associated software of the company Brüel & Kjaer were used for advanced modal analysis.

2 Dynamic flexibility of a milling machine structure and a tool

The milling machine structure has a different contribution to the relative flexibility between a tool and a work-piece. There are generally two parts of a whole structure: a main frame and the parts associated with the spindle. This dynamic investigation will be focused on the second part. An example of transfer function of such a structure is known in Fig. 1. Among the parts of the shown structure, a spindle stock, a slider and a ram can be involved. Some of the latest models of machine tools use parallel structures for spindle positioning. These structures also have very similar characteristics to those shown in Fig. 1. Generally, the modes of spindle can be identified at the upper frequency range, let us say, above 500Hz.

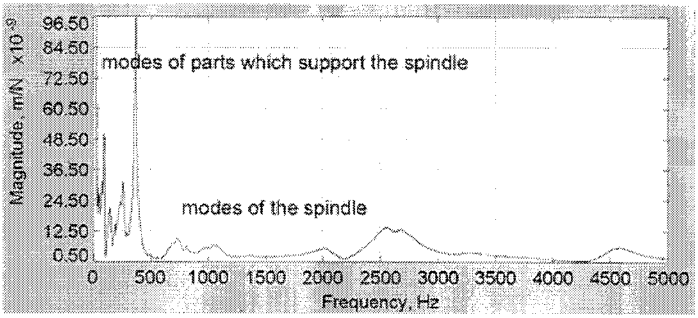

Fig. 1: Example of dynamic flexibility of a milling spindle and its nearest associated parts like the spindle stock, slider, ram etc. Typically, the spindle has higher natural frequencies than the spindle stock or ram. No tool is clamped in the spindle.

As far as the dynamic stiffness is concerned, its value depends on purpose of the milling spindle and the machine. But, comparing the dynamic stiffness of the spindle and its supporting parts in Fig. 1, the spindle modes are much stiffer. This relation could be changed if a tool is clamped into the spindle as it is shown in Fig. 2. We can see that the tool modes are almost as flexible as the structure ones. Now, there is a questiono f applying the structure modes to stability calculation. From the theoretical point of view, [3], they should be applied. Sometimes they must be considered. But the next example shows that a

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contribution of these low-frequency modes to the instability of the milling process is low. See Fig. 3 and Fig. 4.

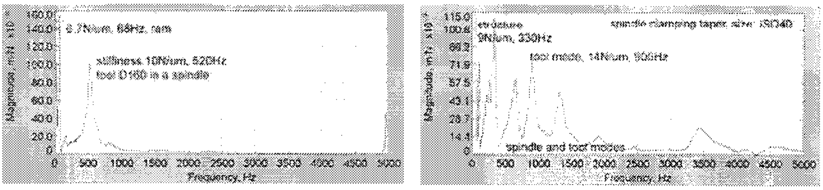

Fig. 2 Flexibility of a milling spindle attached to a ram. A tool of diameter 160 mm (D160) is clamped in the ISO50 spindle taper (left). A slightly stiffer tool D63 was clamped to another spindle, taper size ISO40 (right).

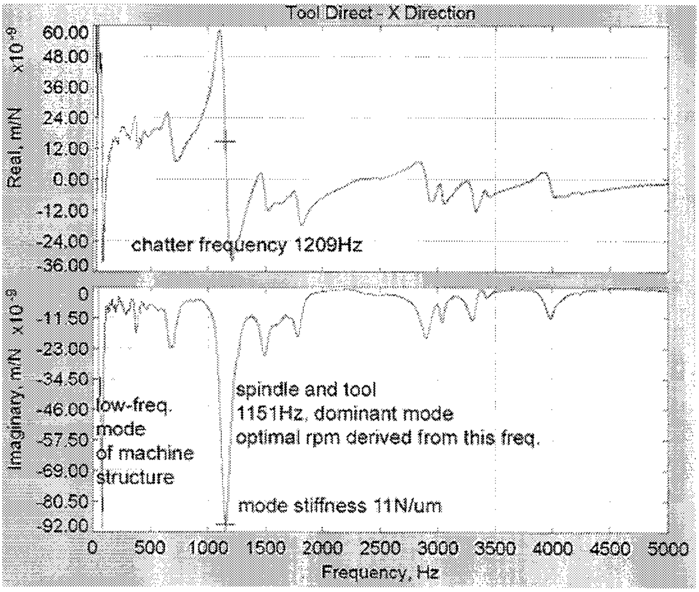

Fig. 3 Typical transfer functions of a spindle with clamped tool. The taper size: ISO40, tool diameter: 63 mm. The low-frequency mode has the same negative extreme as the dominant mode, namely tool and spindle mode. The low-frequency structure modes could mostly cause some resonance problems, but normally not chatter. See Fig. 4.

Let us consider other types of tools that create chatter. The tools with higher L/D ratio particularly create such the problems. They show very high natural frequencies as well as low stiffness. Some examples are depicted in Fig. 5. As we can see, the slender tools decrease the total dynamic, as well as static stiffness of a milling machine. This fact, naturally, affects stability level. Stable

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depth will be considerably decreased. The results of the stability calculation are shown in Fig. 6.

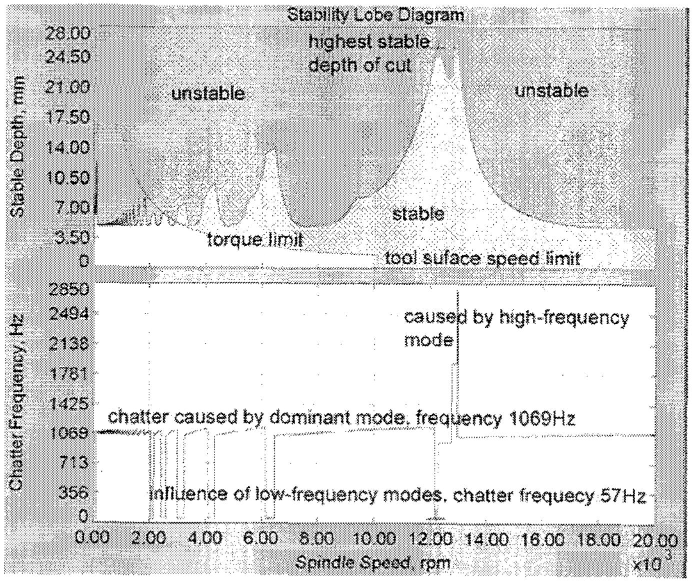

Fig. 4 Stability lob diagram calculated from the curves in Fig. 3. The diagram includes low-frequency modes 57Hz of FRF. These modes belong to a spindle stock. In spite of that, a dominant tool mode prevails over the others. So, in this case the low-frequency modes can be neglected.

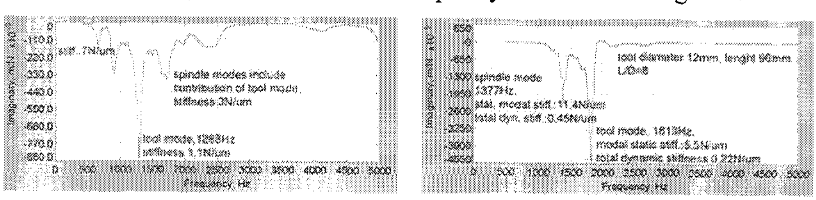

Fig. 5 Stiffness of slender tools is very low. Their natural frequencies, on the contrary, are very high. The tool modes are mostly dominant modes. In the left diagram: horizontal spindle, taper ISO40, tool diameter 20mm, L/D3. In the diagram on the right: vertical spindle, taper ISO40, tool diam.12mm, L/D8.

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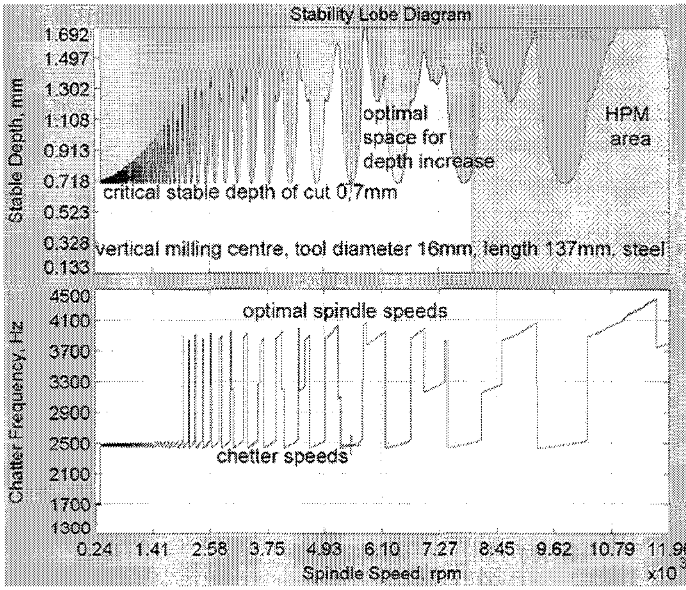

Fig. 6 Slotting of steel with slender tool of diameter 16mm, L/D8,5. The critical depth of cut is only 0,7mm. An improvement is possible up to 1,2mm within conventional cutting speeds 400m/min. A higher performance is accessible in HPM area. It is associated with high spindle speeds, not with higher chip sections. See the upper right corner in diagram.

3 Work-piece material influence

One of the key expressions in the theory of stability available in [1], [3] is the formula

$$a_p = -\frac{1}{2.K_c.G(f)_{neg}}, \tag{1}$$

where a_p is depth of cut, K_c is specific cutting force of the work-piece material and $G(f)_{neg}$ is real part of the transfer function within the frequency range, where the function is negative. Specific cutting force is to be assumed approximately to 2000 N/mm^2 for steel, 800 N/mm^2 for aluminium alloys, 1400 N/mm^2 for cast iron grey and 2400 N/mm^2 for titanium. To predict stability diagram for various work-piece material, the cutting data like tool diameter, number of tool teeth, width of cut, cutting mode and feed per tooth, are required. An example can be seen in Fig. 7 (left: titanium, right: aluminium). Because of different cutting speed applied, the charts look different. With regard to the tool live, the conventional values of cutting speeds in the range 40-70m/min have been used for titanium. For aluminium alloys the speed limit 1000m/min has been chosen.

Process damping is a phenomenon that helps in the low-speed range to achieve acceptable performance of a tool [1]. For machining of aluminium alloys, we have the large area of stable depth in the right part of the diagram.

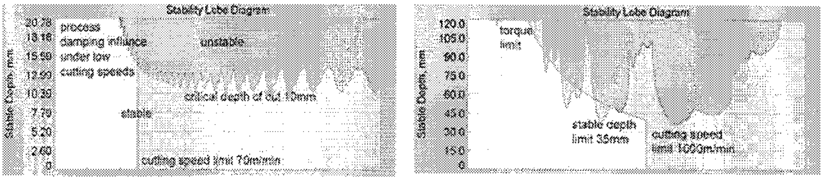

Fig. 7 Down milling of titanium (left), K_c =2400 N/mm^2 and aluminium (right), Kc=810 N/mm^2 , tool diameter 100 mm, number of tool teeth z=5, feed per tooth 0,15 mm,wi dth of cut 25 mm.

4 Synthesis of machine structure properties and tool stability properties

Stability diagrams are, in general, created from the expression (1) and from the following formulas.

$$G(f) = u_1.G_1(f) + u_2.G_2(f) + ...u_n.G_n(f), \qquad (2)$$

where the G(f) is resulting cross transfer function, the $G_i(f)$ are individual direct transfer functions and u_i are directional factors. As we have seen in the above shown charts, the structure of spindle and tool can be in most cases reduced to a two degree of freedom system. For this structure can be written:

$$G(p) = \frac{u_1}{k_1}.\frac{1-p_1^2}{(1-p_1^2)^2+(2\zeta_1 p_1)^2} + \frac{u_2}{k_2}.\frac{1-p_2^2}{(1-p_2^2)^2+(2\zeta_2 p_2)^2}, \qquad (3)$$

where k_i is static stiffness, $p_i=f/f_{ni}$ is frequency ratio and ζ_i is damping ratio. All these parameters are the key parameters for design of machines. They can be influenced by many design or assembly approches. The equations (1), (2), (3) represent mechanical structure. To develop the whole diagram, the “structure equations” have to be completed with the following “drive equation”:

$$n(f) = \frac{60.f}{z.(\frac{\varepsilon}{2\pi} + N)}, \qquad (4)$$

where

$$\varepsilon = actg\frac{G(f)}{H(f)}, \qquad (5)$$

Here H(f) is imaginary part of the resulting complex cross transfer function. The parameter N with possible values of 1, 2, 3, 4, is a number of entire waves between two succeeding teeth of the milling cutter. Let us imagine now, we generate a stability curve for a single degree of freedom system starting theoretically from f_n in small steps of frequency, going

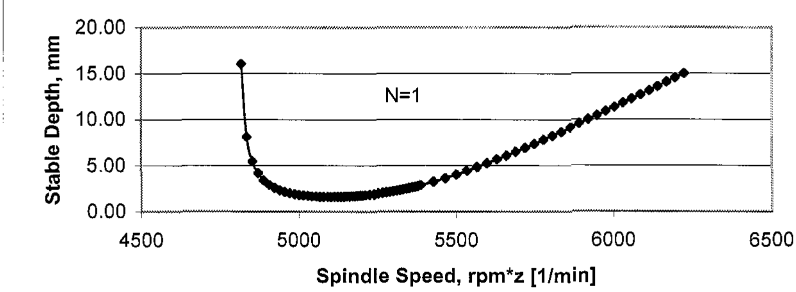

Fig. 8 Stability diagram chart for SDOF structure, for N=1.

over a chatter frequency further to very high values of frequency. For N=1 a curve shown in Fig.8 will be created. The minimum of the curve represents critical depth of cut. The whole lob curve can be moved vertically if the structure parameters k_i - static stiffness, and ζ_i - damping ratio are changed. Simultaneously, the natural frequency varies. These changes move the curve horizontally. All these changes should be related to the spindle drive characteristic that is horizontally fixed to the spindle speed. If we compare the curves in the Fig. 9, Fig. 10 and Fig. 11, we can see their mutual horizontal and vertical. A considerable difference can be observed between critical power values. Diagrams in the Fig. 10 and Fig. 11 belong to the same spindle. The different tool modal frequencies move the lob curves along spindle speed coordinate. If we accept the rule, that the area of instability is unusable for machining, then the relative penetration of the stability power lobs and the drive power characteristic qualifies a degree of utilization of the drive characteristic area and this can be measured simply as ratio of both areas. Degree of the utilization can be also expressed as a power ratio, defined by (6) as the ratio of power of the tool with the critical depth of cut a_p^{crit} and of the power installed in the spindle drive.

$$\upsilon = \frac{P_{crit}}{P_{inst}}, \tag{6}$$

where

$$P_{crit} = \frac{V.K_c}{60.\eta.10^6} = \frac{a_p^{crit}.a_e.f_{\min}.K_c}{60.\eta.10^6} = \frac{a_p^{crit}.a_e.f_z.n.z.K_c}{60.\eta.10^6}. \tag{7}$$

Here V is metal removal rate in cm^3/min, and η is drive efficiency.

The values of critical power are designated in the figures at the bottom of curves. These are 19,7kW, 14,6kW and 5,2kW. The utilization degree of 52%, 38% and 23% has been achieved. This is the minimal degree utilization of the

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spindle drivewi th above specified tool that can be achieved over the entire spindle speed range.

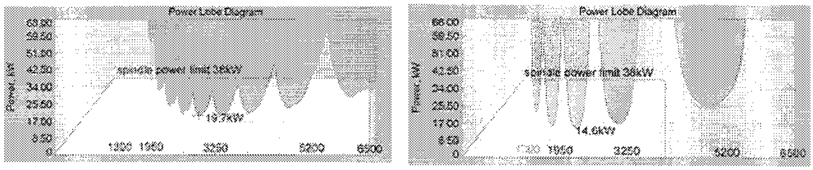

Fig. 9 Spindle drive characteristic related to the spindle and a tool structure. (Spindle taper ISO50, tool diameter of D50mm). This is an example of the relatively good relation between the spindle design and spindle drive for cast iron grey milling (left). The change of structure characteristic is caused by tool parameters (right). There is the same spindle, but another tool with D80, z5, feed 0,18, installed at machine tool. The tool surface speed limit is kept at the value of 1000m/min.

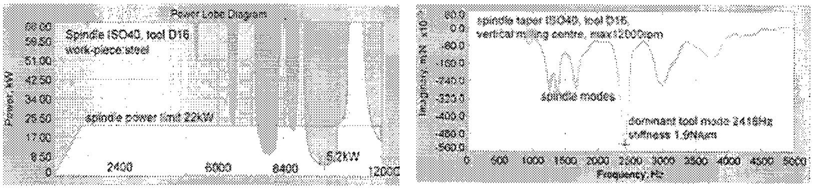

Fig. 10 The power lobe diagram is generated from two modes with frequencies 2400 and 3500 Hz. The high frequencies shift the lobes towards higher spindle speed (compare the rpm scale with Fig. 10, where the diagram is based on transfer function modes with frequencies of 580 and 624Hz only.

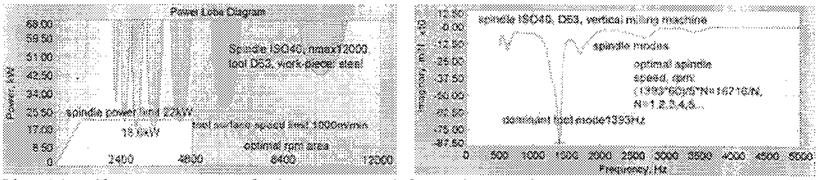

Fig. 11 The next example is generated from lower frequency modes. The mode 1293Hz appears in another direction. The second mode has 1393Hz. The corresponding optimal spindle rpm range is from 5500 up to 16716 rpm. The used spindle has a speed limit 12000 rpm. Tool dynamic stiffness: 12N/um.

5 Spindle unit utilization

Naturally, the degree of utilization, mentioned above, can be increased. Using the optimal speeds, which lie in each gap between lobs, we can raise metal removal rate as well as the drive power utilization. This solution suits better to the part programmers.

From a designer point of view, stiffness and damping are the most important parameters that should be considered. If we theoretically accept for SDOF system that

$$G_{neg}^{\max} = -\frac{1}{4k\zeta} = \frac{1}{2}.\left|\Phi_{dom}\right|, \tag{8}$$

then the expression (1) can be rewritten as

$$a_p^{\lim} = \frac{2.k.\zeta}{K_c} = \frac{1}{K_c.\left|\Phi_{dom}\right|}, \tag{9}$$

where Φ_{dom} is the dynamic flexibility of a dominant tool mode. So, it can be written for material removal rate:

$$V = \frac{2.a_e.k.\zeta.f_z.z.n}{K_c} = \frac{a_e.f_z.z.n}{K_c.\left|\Phi_{dom}\right|}. \tag{10}$$

It means, the static stiffness and the damping ratio influence in the same way the performance of machine tool. Spindles, designed for HSC, need small diameters of bearings, which limits their stiffness. The increase of the damping ratio is therefore more promising for the HSC spindle design. Nevertheless, the stiffness of the tool holder is important parameter for the stability of machining.

Let us now compare a MRR of tools used in the Fig. 10 and Fig. 11. The cutter D16 for slotting of steel, has the following parameters: D=16mm, dynamic flexibility 500.10^{-6} mm/N, z=4, f_z=0.2mm. It gives MRR=13,4 cm^3/min at 1000 rpm. The second cutter D63, again for slotting steel has Φ_{dom}=70, z=5, fz=0,2. This cutter gives 440cm^3/min at 1000rpm as compared to the Fig. 12. In reality, the calculated value of the MRR depends on a stability curve influenced by more then one mode. We can see more possibilities to improve performance of the machine in the right part of the chart around the spindle speed 1496 rpm with the corresponding tool surface speed 300m/min. With the acceptable spindle drive overloading of 20% almost 200% of MRR can be reached.

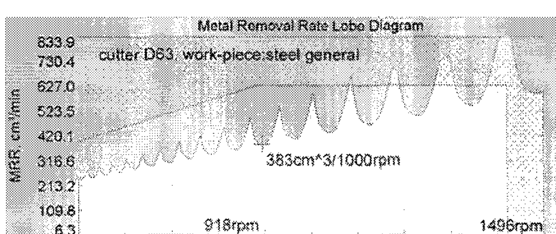

Fig. 12 Metal removal rate lobe diagram. Horizontal center, spindle taper ISO40, steel slotting, D63mm

6 Conclusions

Dynamic stiffness is decisive parameter for MRR and a power drive ratio. To avoid machining process instability, stiffness and damping ratio of the spindle and tool has to be considered.
Static stiffness and a damping ratio act in the same way on performance of a machine tool.
Tool modes are dynamically almost as flexible as the structure modes (not spindle modes).
Spindles and tools with increased damping ratio should be developed.
Tool holders of higher stiffness should be designed.
Stiffness of slender tools is too low. Theirs natural frequencies, on the contrary, are quite high. Therefore, they need high-speed spindles for a good performance.
Higher tool surface speeds and smaller chip sections are recommended to achieve HPM.
Stiffness and damping ratio should be related to a spindle drive characteristic.
For HSC applications spindles need as high natural frequencies as possible.
In general, modes of current spindles have already natural frequencies above 500 Hz.
Relative penetration of the stability power curve and the drive power characteristic qualify degree of utilization of the drive characteristic area.
Machine parts, which are close to a spindle, can cause rather resonant vibration then chatter.
Process damping is a phenomenon that helps in the low-speed range to achieve acceptable performance of a tool.
Stability diagrams of depth of cut, MRR and power can improve design and performance of spindle units.

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