EXPERIMENTAL SETUP FOR FAST STABILITY CHART RECONSTRUCTION OF MILLING PROCESSES

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Abstract: The self-excited vibrations during the milling process cause poor surface quality and shorten the tool life expectancy. In order to choose proper technological parameters, the so-called stability chart must be created. In the stable regions, large amplitude forced vibration can occur, which also leads to poor surface quality. Our calculation shows that both the good surface properties and the large material removal rate can be achieved by using appropriate axial immersion in case of helical fluted tools. A test rig has been designed and manufactured according to the analytic model. The measurements confirmed the theoretical results.

Keywords: *Surface quality, Milling, Stability chart*

1. INTRODUCTION

Large amplitude vibrations should be avoided during the milling process, because they lead to poor surface quality, and shorten the lifetime of the machine. Vibrations can be caused by the well known regenerative effect. This type of vibration is often called chatter. Most investigations in this field focus on the construction of the stability chart that defines those regions of the technological parameters (feed rate, depth of cut, axial immersion, cutting speed, etc.) where chatter does not occur [1,2,3,4,5].

In those regions of the stable technological parameters where large material removal rate can be achieved, forced vibrations occur which are excited by the periodic cutting force [3,4,6,7]. In [8,9], we proved theoretically that in case of helical tools both the good surface properties and the efficient milling process can be realized by using proper axial immersion w given by

$$w = \begin{cases} \dfrac{\Omega p}{\omega_n} n \\ \dfrac{p}{N} n \end{cases} \qquad n \in \aleph , \tag{1.1}$$

where N is the number of teeth of the helical tool, p is the helix pitch of the tool, Ω is the angular spindle speed, ω_n is the angular natural frequency of the system. To show this phenomenon, the surface properties were calculated above the stable domain of the technological parameters (see Figure 1.). The optimal axial immersions (1.1) are denoted by dashed and dotted lines.

To confirm our analytical investigations, we established an experimental setup.

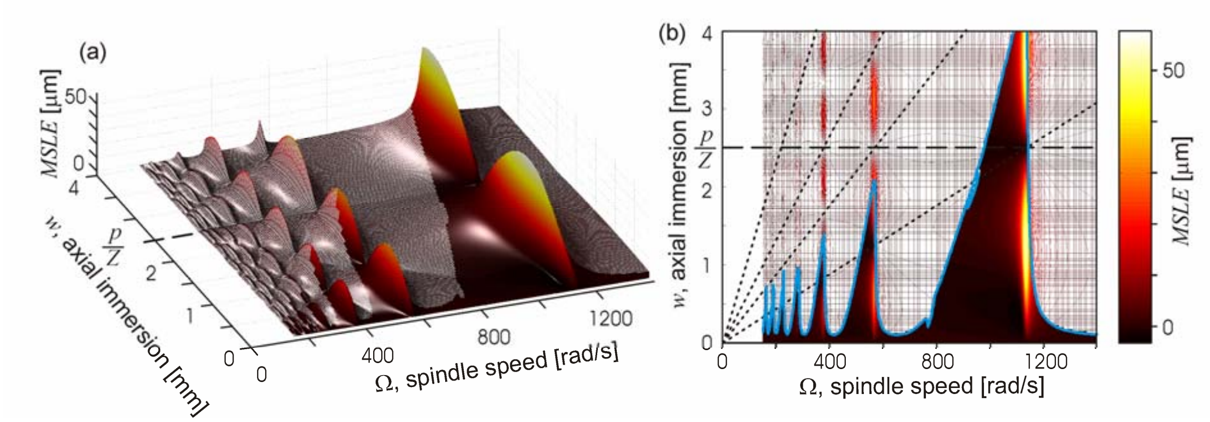

Fig.1. Maximum surface location error (a) and its top view (b). Dark areas denote the stable region (N=4, τv=0.1[mm], p=10 [mm], a/D=0.05)

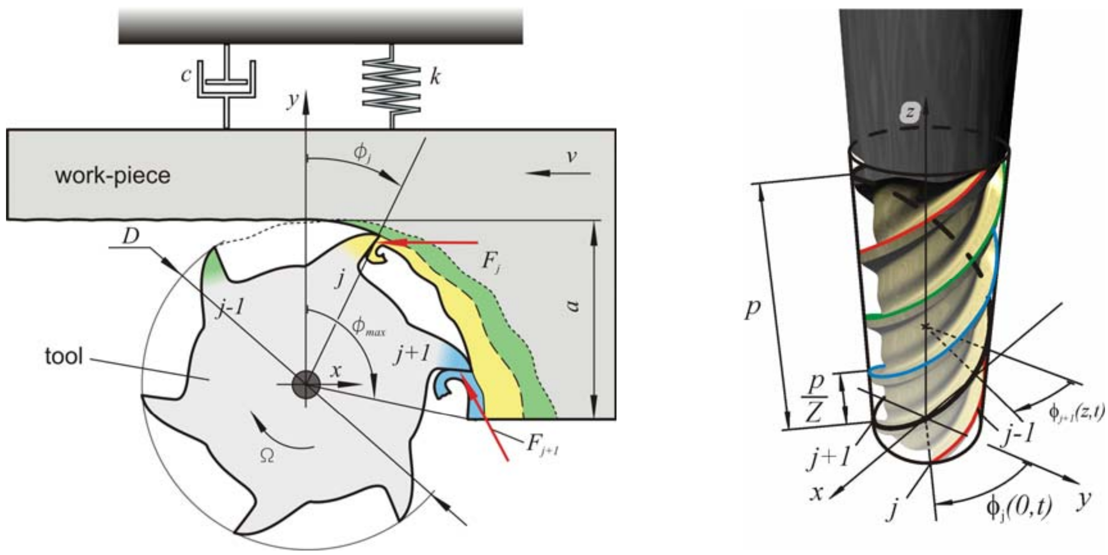

Fig.2. Schematic representation of the flexible work-piece holder and the helical tool

2. DESIGN OF THE EXPERIMENTAL RIG

In the above presented calculations, we used a one degree of freedom model with helical tool shown in Figure 2, which is rigid in the feed direction x and flexible in the perpendicular (radial depth of cut) direction y. To realize this model, we manufactured a test rig with high flexibility (see Figure 3) based on [5]. In order to measure the above mentioned effects, we had to satisfy some basic requirements:

- The natural frequency of the test rig $f_n=\omega_n/2\pi$ has to be set in the region of tooth-pass frequency $1/\tau = N\Omega/2\pi$. In case of a 5-fluted tool and the available spindle speed of the machine-tool, it can be 12-122 [Hz]. Thus the desired natural frequency of the test rig must be at about 70 [Hz].
- During the machining process, the mass of the work-piece is decreasing, therefore the mass of the upper part has to be sufficiently large in order to neglect this effect.
- In the calculations, we supposed that the path of the cutting edge is approximately a circle. This approximation is satisfactory if the vibration amplitude $X_{\max}$ is smaller than ~5% of the diameter of the tool D even for resonant excitation. In case of D=20[mm] we obtain $X_{\max}$ =1[mm].

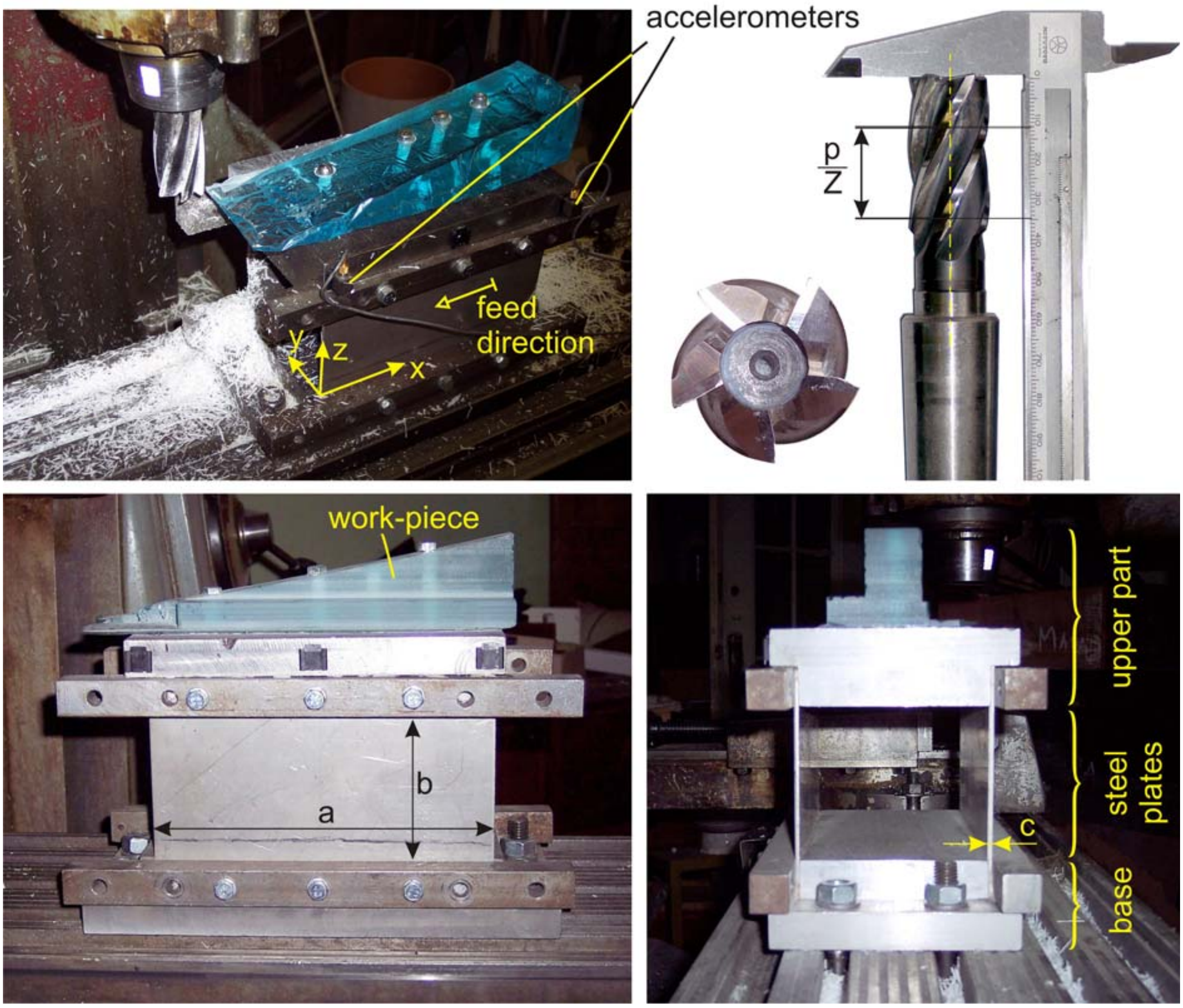

Fig.3. The flexible holder and the wedge-shaped polymer work-piece.

During the measurement, small radial immersion a_e=0.5[mm] was used, which is a usual value in case of surface finishing processes. The axial immersion is limited by the tool, so w_{max}=35[mm]. In the test measurements, we used epoxy resin material where the cutting coefficient was supposed to be ~100[N/mm^2]. Using these values, the maximal cutting force could be calculated (see Appendix 1): $F_{cut, max}$=164[N].

To satisfied our third requirement, the stiffness of the test rig must be larger than k_{min}=2.7·10^6[N/m] in case of the damping ratio ξ = 3% (see Appendix 2).

Hence, the mass of the upper part should be around 10.8 [kg] to satisfy the first and second requirements.

In the work-piece, holder the two steel plates serve as a spring. Its stiffness is given by [10]:

$$k = 2\frac{a}{E}\left(\frac{c}{b}\right)^3, \tag{2.1}$$

where the modulus of elasticity E=210[GPa], the geometry is defined by the sizes a, b and c as shown in Figure 3. We choose a=180[mm], b=70[mm] and c=2.5[mm], so the calculated stiffness is 3.4·10^6[N/m].

3. MODEL TEST AND MEASUREMENT

After manufacturing the test rig, we carried out its modal analysis to check the natural frequency and the damping ratio. We use Pulse Front-end and Pulse Labshop v11.0 software to detect the signal of the accelerometers (B&K4397) placed on the upper part. From the Impact test procedure, the measured natural frequency is f_n =67.5 [Hz], and the damping ratio is ξ =1.14%.

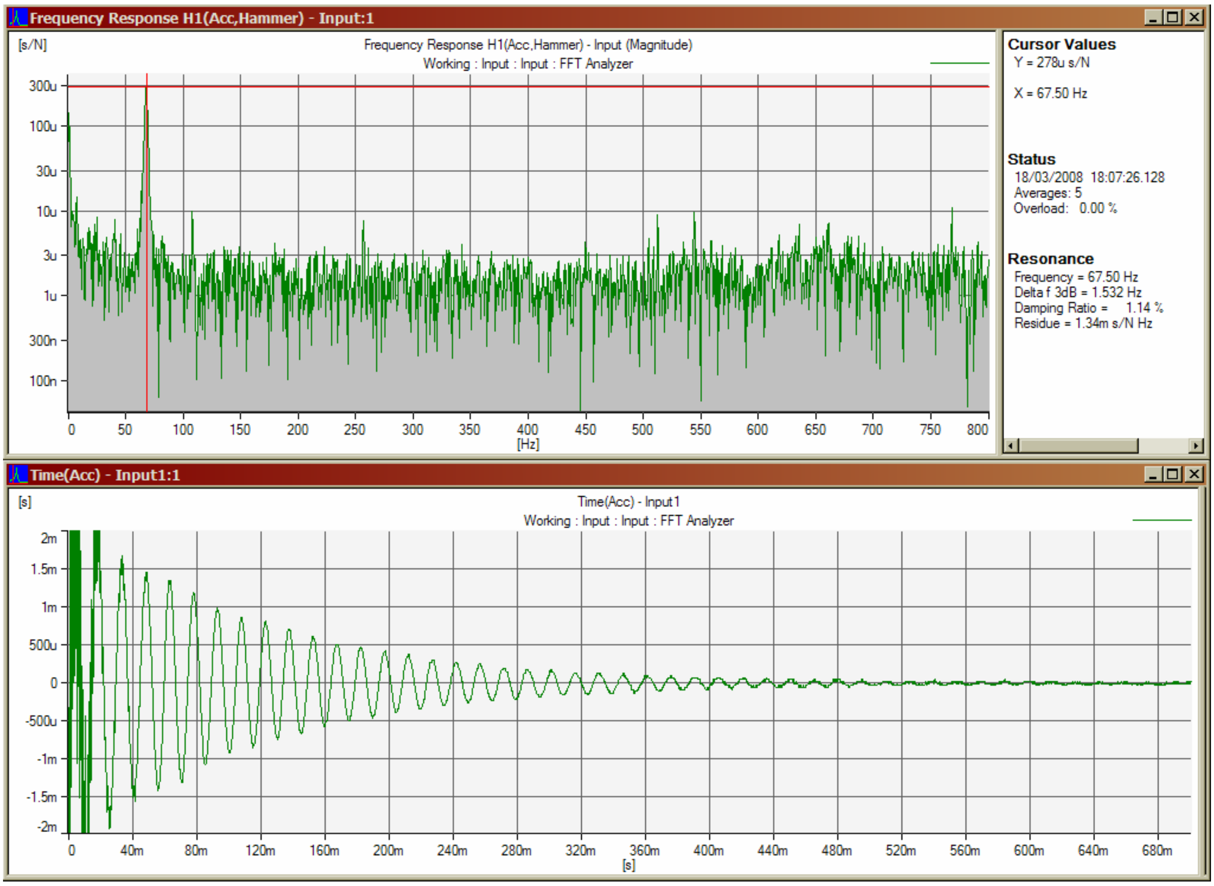

Fig.4. Impact test procedure. Time signal and its Fourier transform.

To confirm our analytical result (1.1), we had to carry out a series of measurements along the constant spindle speeds

$$\Omega = \frac{\omega_n}{N\,n} \qquad n \in \aleph \tag{1.1}$$

lines, where the higher Fourier harmonics of the cutting force excite the system just at its resonant frequency. Instead of making numerous measurements along each line, we used wedge-shaped work-piece to speed-up the lengthy experiments. In this way, the axial immersion was increasing continuously during the cutting process. So the stability boundary and the optimal axial immersions could be found with one single test along each line.

During the measurements a 5-fluted tool was used with helix pitch p=112.5[mm] and diameter D=Ø20[mm]. We used the radial immersion a_e=0.5[mm] as explained above, and the feed rate was set to τv=0.15[mm/tooth], where v is the feed velocity.

In Figure 5, the amplitudes of the vibrations are shown in case of different spindle speeds. It is clearly observed, that in the optimal axial immersion points denoted by red lines according to formula (1.1), the amplitudes of the vibrations get smaller. According to the theory, the amplitude should be close to zero in these regions, but due to the cylindrical asymmetry of the tool, the so called run-out effect appears that still causes some excitation of the system even at the optimal parameters.

5. CONCLUSION

We designed an experimental setup to validate our previous theoretical results for the values of the expected surface location errors in case of milling with helical tools. The properties of the test rig were appropriate for the test measurements. The tests were carried out for 3 different spindle speeds. We found that the vibration amplitude was smaller in case of the optimal axial immersion, which confirmed our theoretical predictions.

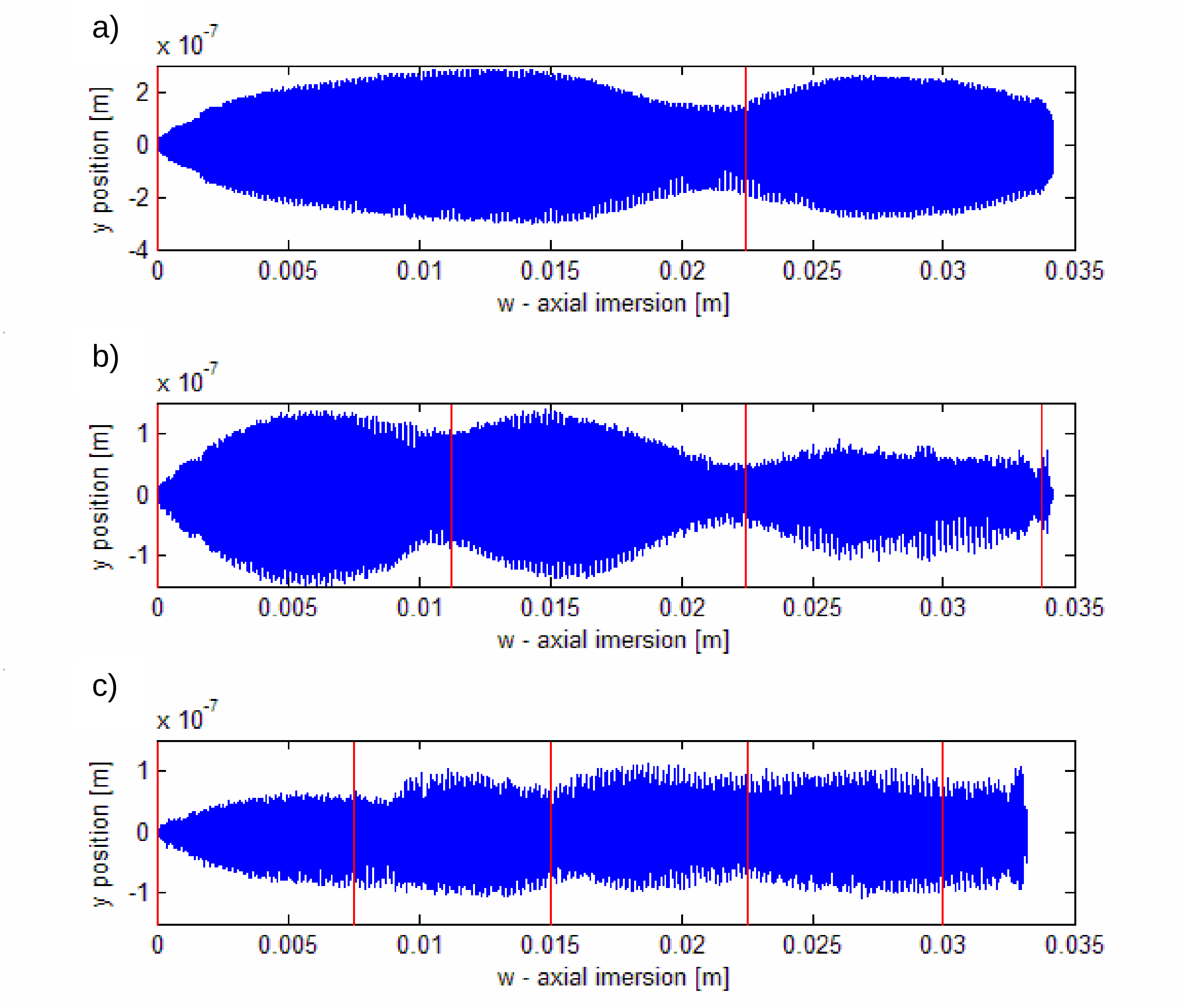

Fig.5. Vibration amplitude along the wedge shaped work-pice. Red lines denote the optimal axial immersion. a) $\Omega=\omega_n/(3N)$ b) $\Omega=\omega_n/(2N)$ c) $\Omega=\omega_n/N$ (N=5, τv=0.149[mm], p/N=22.5 [mm], a_e/D=0.025)

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APPENDIX 1 - *Maximal predicted cutting force*

The cutting force F_{cut} is supposed to be a linear function of the chip thickness h, the chip width w and the cutting coefficient K:

$$F_{cut} = Kwh \tag{I.1}$$

If the path of the cutting edges is supposed to be circular, the maximal chip thickness is given by:

$$h_{\max} = f_z \sin\phi_{\max} = f_z \sin\phi_{\max}\,, \tag{I.2}$$

where the maximal angular position of the cutting edge during the cutting phase is

$$\phi_{\max} = \arcsin\frac{D - 2a_e}{d} \tag{I.3}$$

(see Figure 2).

APPENDIX 2 - *Minimal stiffness and natural frequency*

For the equation of motion

$$m\ddot{y} + c\dot{y} + ky = F_{\max}\sin(\omega t) \tag{II.1}$$

of the test rig, the largest vibration amplitude X_{max} can be determined in the resonant case $\omega^2 \cong \omega_n^{\,2} = k/m$ and small damping ratio ξ in the following way:

$$X_{\max} = N_{\max}\frac{F_{\max}}{k} \cong \frac{1}{2\xi}\frac{F_{\max}}{k}\,. \tag{II.2}$$

Thus the minimal stiffness can be calculated from

$$k_{\min} = \frac{1}{2\xi}\frac{F_{\max}}{X_{\max}}\,. \tag{II.3}$$

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