Int. Journal of Refractory Metals and Hard Materials 29 (2011) 163–169

Contents lists available at ScienceDirect

Int. Journal of Refractory Metals and Hard Materials

journal homepage: www.elsevier.com/locate/IJRMHM

Linear analysis of chatter vibration and stability for orthogonal cutting in turning

Erol Turkes [a], Sezan Orak [b], Suleyman Neseli [c,*], Suleyman Yaldiz [c]

[a] Department of Mechanical Engineering, Dumlupinar University, 43100 Kütahya, Turkey
[b] Eskişehir Osmangazi University, Department of Mechanical Engineering, Batı meşelik, Eskişehir/Turkey
[c] Mechanical Department, Technical Education Faculty, Selçuk University, 42031 Konya, Turkey

ARTICLE INFO

Article history:
Received 25 June 2010
Accepted 14 October 2010

Keywords:
Turning
Chatter vibration
Stability analysis

ABSTRACT

The productivity of high speed milling operations is limited by the onset of self-excited vibrations known as chatter. Unless avoided, chatter vibrations may cause large dynamic loads damaging the machine spindle, cutting tool, or workpiece and leave behind a poor surface finish. The cutting force magnitude is proportional to the thickness of the chip removed from the workpiece. Many researchers focused on the development of analytical and numerical methods for the prediction of chatter. However, the applicability of these methods in industrial conditions is limited, since they require accurate modelling of machining system dynamics and of cutting forces. In this study, chatter prediction was investigated for orthogonal cutting in turning operations. Therefore, the linear analysis of the single degree of freedom (SDOF) model was performed by applying oriented transfer function (OTF) and \tau decomposition form to Nyquist criteria. Machine chatter frequency predictions obtained from both forms were compared with modal analysis and cutting tests.

1. Introduction

Machine tool chatter is a self-excited vibration problem occurring in large rates of material removal, resulting from the unavoidable flexibility between the cutting tool and workpiece. When uncontrolled, chatter causes the rough surface end and dimensional inaccuracy of the workpiece, along with unacceptably loud noise levels and accelerated tool wear. In general, chatter is one of the most critical limiting factors, which is considered in designing a manufacturing process. The cutting force becomes periodically variable, reaching considerable amplitudes and the machined surface becomes undulated. There are two main sources of self-excitation in metal cutting. These are mode coupling and regeneration of waviness on machined surface [1,2]. Studies of chatter date back over half a century such as those by Merchant [3] and Arnold [4]. Basic understanding of the problem, including the mechanism of surface regeneration, mode coupling and association with structural dynamics, was postulated more recently by Tobias [5] and Koenigsberger and Tlusty [6], Tlusty [7], and Anderson et al. [8]. Their studies with numerous respective collaborators provided the frame of the current chatter research.

Nowadays, researchers brought forward analysis and control techniques through various models for the prediction of chatter vibrations. But since the mechanics and dynamics of cutting could not be put forward satisfactorily, a complicated model that is capable of expressing metal removal dynamics does not exist properly yet. Analysis of chatter vibrations is realized by the process of linear and nonlinear forces. All chatter analysis techniques begin with a model of the machining force process and a model of the tool-workpiece structure. These two models are combined to form a closed-loop dynamical model of the machining operation. Analysis techniques are used to generate so-called stability lobe diagrams (SLDs): plots of the stable and unstable regions in the cutting parameter space. There are two techniques that may be used to generate SLDs. These are Nyquist criterion and Time Domain Simulation (TDS) techniques. The Nyquist criterion is applied to determine if the cutting conditions are stable. The depth of cut is adjusted and the procedure is repeated until the critical value is determined. In TDS technique, the closed-loop dynamical model of the machining operation is simulated for a particular set of cutting conditions. Steady state tool/workpiece displacement and machining force signals are examined to determine system stability. Displacement of the tool according to the workpiece and signals of cutting force is investigated for the determination of system stability. Thus, effort is given to determine the critical depth of cut which provides the best stability [9–13]. One of the most important superiority of TDS is that it covers nonlinear characteristics concerned in chatter analysis. Its disadvantage, however, are those simulations involving numerous runs that make the calculation difficult, and even there may be excessive loss of time. Due to these disadvantages, efforts to develop analytical methods in plotting SLD are still made by researchers, and new analytical methods have been devised [14–17]. In this study, chatter prediction was investigated for orthogonal cutting in turning operations. Machine chatter vibrations were predicted in orthogonal cutting with SDOF turning system by two analytical forms. Firstly, this cutting process was modeled according to OTF and τ decomposition forms. Then, stability of this system was investigated by

* Corresponding author. Tel.: +90 332 2232353; fax: +90 332 2410185.
E-mail addresses: eturkes@dpu.edu.tr (E. Turkes), sorak@ogu.edu.tr (S. Orak), sneseli@selcuk.edu.tr (S. Neseli), syaldiz@selcuk.edu.tr (S. Yaldiz).

0263-4368/$ – see front matter © 2010 Elsevier Ltd. All rights reserved.
doi:10.1016/j.ijrmhm.2010.10.002

applying OTF and τ decomposition form to Nyquist criteria. Finally, results obtained from both forms were compared with the modal analysis conducted and the results of cutting tests in [18].

2. Modeling of the cutting system

Machine chatter prediction and stability analysis are conducted for a turning process with SDOF. Since the dynamic cutting operations are very complicated, the turning system which is investigated in this study was simplified by modeling mass-spring-damper as it is seen in Fig. 1.

The equation of motion of the cutting system in the feed direction (y) was formed as follows,

$$m_y\ddot{y}(t) + c_y\dot{y}(t) + k_y y(t) = F_y(t) \tag{1}$$

where $F_y(t)$, $F_y(t) = -F(t)\cos\beta$ can be written in terms of the cutting force and this is;

$$F_y(t) = a K_f h(t) \tag{2}$$

can be expressed by a, chip width (mm); K_f, specific cutting energy of the material (N/m^2) and $h(t)$, instantaneous chip thickness. Instantaneous chip thickness, however;

$$h(t) = h_0 - y(t) + y(t-\tau) \tag{3}$$

can be written clearly according to the geometry in Fig. 1. Term $[y(t)-y(t-\tau)]$ is the dynamic chip thickness produced owing to vibrations at the present time (t) and one spindle revolution period (T) before. With the help of these equations, chatter prediction of the cutting system was investigated analytically by OTF and τ decomposition, respectively.

2.1. Oriented transfer function

Oriented transfer function of the SDOF in Fig. 1 is obtained from Eqs. (1) to (3) [1].

$$m_y\ddot{y}(t) + c_y\dot{y}(t) + k_y y(t) = F_y(t) = aK_f h(t) \tag{4}$$

If equation of motion is written in Laplace domain, in order for the OTF between chip depth of the system $y(t)$, which is being removed now, and the force in the feed direction will be written,

$$m_y s^2 y(s) + c_y s\, y(s) + k_y y(s) = F_y(s) \tag{5}$$

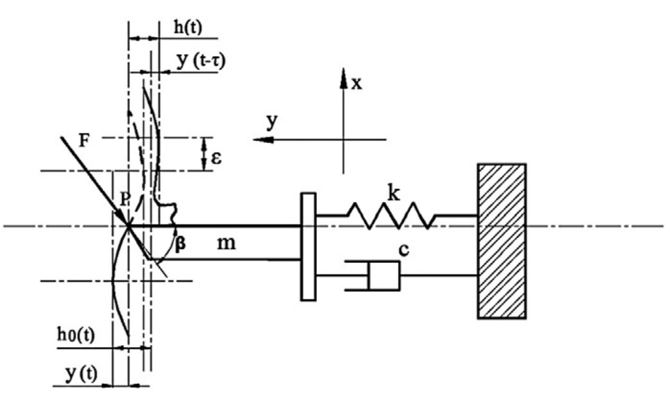

Fig. 1. Modelling of SDOF system in turning.

Where, as seen in Fig. 2, transfer function of the open-loop between $y(s)$ and $F_y(s)$, can be written as follows,

$$G(s) = \frac{y(s)}{F_y(s)} = \frac{1}{ms^2 + cs + k} \tag{6}$$

on the other hand, closed-loop transfer function between dynamic and reference chip loads is obtained by taking Laplace transformation of Eq. (3) and substitution of $y(s) = G(s)\cdot F_y(s)$ into $h(s)$;

$$\frac{h(s)}{h_0(s)} = \frac{1}{1 + aK_f G(s)(1 - e^{-\tau s})} \tag{7}$$

Closed-loop transfer function between $y(s)$ and $h_0(s)$ is written as Eq. (4)

$$m_y\ddot{y}(t) + c_y\dot{y}(t) + k_y y(t) = aK_f[h_0 - y(t) + y(t-\tau)] \tag{8}$$

and the formula applying Laplace transformation on both sides can be written as follows,

$$\frac{y(s)}{h_0(s)} = \frac{aK_f G(s)}{1 + aK_f G(s)(1 - e^{-\tau s})} \tag{9}$$

Denominators of Eqs. (7) and (9) are the characteristic equation of the system.

2.2. τ-Decomposition form

The cutting force, which changes in the course of time, proportionally with the surface area of the chip removed from the surface, is a general acceptance of linear modeling. For this reason, the constant component of the cutting force is neglected but variable component produced by dynamic chip load is taken into account. According to this calculation, the variable force can be written as,

$$F_v(t) = aK_f[y(t) - y(t-\tau)] \tag{10}$$

This is shown in the block diagram in Fig. 3.

In the diagram, F_m is the average cutting force, H_m is average chip load. Where, Eq. (1) can be written as follows;

$$m_y\ddot{y}(t) + c_y\dot{y}(t) + k_y y(t) = -F(t)\cos\beta \tag{11}$$

if the equation for τ decomposition form is written independent of time and according to the course taken by the relative movement of tool tip to the workpiece,

$$l = Vt + x \tag{12}$$

where, V is the linear speed in the rotation direction of the workpiece (m/s), x is the displacement of tool tip independently from workpiece on the axis x during cutting (m). When the derivative of Eq. (10) is taken as time related,

$$\frac{dl}{dt} = V + \dot{x} \approx V \tag{13}$$

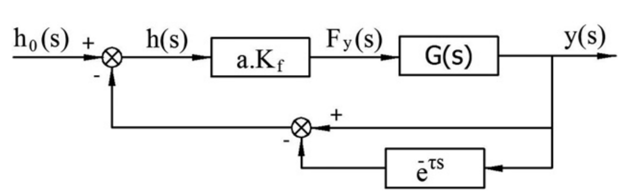

Fig. 2. Block diagram of chip depth for the turning system with SDOF.

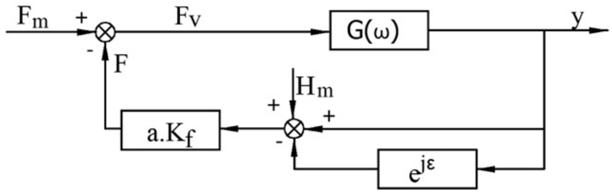

Fig. 3. Block diagram of cutting forces for the turning system with SDOF.

where, ($\dot{x}$) was neglected, since, according to linear speed, it has no significant value for linear modeling. Eq. (11) with these simplifications;

$$\dot{y}(t) = \frac{dy}{dt} = \frac{dy}{dl}\,\frac{dl}{dt} = (V + \dot{x})\,\frac{dy}{dl} \approx V\,\frac{dy}{dl} \rightarrow y'' = \frac{dy}{dl}$$

$$\ddot{y}(t) = \frac{d^2y}{dt^2} = \frac{d}{dt}\left[(V + \dot{x})\,\frac{dy}{dl}\right] = \ddot{x}\,\frac{dy}{dl} + (V + \dot{x})\,\frac{d^2y}{dl^2}\,\frac{dl}{dt} \approx V^2\,\frac{d^2y}{dl^2} \rightarrow y'' = \frac{d^2y}{dl^2}$$

the following equation is obtained,

$$m \cdot V^2 y'' + cVy' + k\,y = -aK_f\,[y(l) - y(l - \pi\,dl)]\,\cos\beta \tag{14}$$

thus, according to constant (l) equation of motion is obtained instead of time dependent one, τ. If both sides of Eq. (14) are reduced for simplification,

$$c_1 = \frac{c}{mV} \qquad k_1 = \frac{k}{mV^2} \qquad F_1 = \frac{-aK_f\,\cos\beta}{mV^2}$$

and if these values are placed and the equation is equaled to zero, and if Laplace transformation is applied, the characteristic equation of the system is obtained,

$$s^2 + c_1 s + (k_1 - F_1) + F_1 e^{-sd\pi} = 0 \tag{15}$$

with further simplification, $a_2 = {}^{1}\!/_{F_1}$, $a_1 = {}^{c_1}\!/_{F_1}$, $a_0 = {}^{(k_1 - F_1)}\!/_{F_1}$ are obtained.

$$e^{sd\pi} = \frac{-1}{a_2 s^2 + a_1 s + a_0} \tag{16}$$

According to the Nyquist criteria, the right side of this equation expresses Nyquist plane curve $U_2(s)$ and the left side expresses critical orbit $U_1(s)$. If $s = j\omega$ is taken in the equation, the roots of the characteristic Eq. (16) is found by equalizing the magnitude of the right side of the equation to 1,

$$\begin{gathered} \frac{1}{\left|(-a_2\,\omega^2 + a_0) + j\,a_1\,\omega\right|} = 1, \\ a_2^2 \cdot \omega^4 + \left(-2 \cdot a_2 \cdot a_0 + a_1^2\right) \cdot \omega^2 + a_0^2 - 1 = 0 \end{gathered} \tag{17}$$

Thus, the positive real root of this equation gives the chatter frequency of the system.

3. Investigating of the stability for cutting system

Two methods are used for the determination of stable areas for chatter-free spindle speed or the system of spindle/tool holder/cutting tool. The first method is the determination of the natural frequency of the system and mode shapes by measuring transfer functions by using an impact hammer and accelerometer. Analytical predictions of performance can be done by using this information. The second method is performed of cutting tests. The method gives the cutting ability of spindle/cutting tool in a better completeness, but requires a number of tests to be performed. The first analysis technique is based on the investigation of stability and plotting the SLD from the solution of the characteristic equation of the system depending on the critical parameters such as axial cutting depth of the system and spindle speed. Two chatter analysis techniques are used in plotting SLDs. The first technique is Nyquist technique, which has been used by many researchers so far. According to this technique, stability of the technique is investigated in accordance with cutting conditions that are taken as basis (i.e. depth of cut and spindle speed) by constructing the characteristic equation of the system, and the procedure is repeated until the critical value which provides stability is determined. The second technique is the TDS, in which cutting conditions of the cutting operation of the closed-loop dynamic model are stimulated for a chosen group. In this technique, tool/work piece displacement which continually change and signals of machining force are examined until marginal stability is obtained according to a chosen critical parameter (i.e. cutting depth). Since the technique involves most outstanding aspects like nonlinear characteristics of cutting operation, it is a more effective technique for analysis. But it also has disadvantages mentioned before. Also, in studies conducted so far in the field of SLD drawings, at least one difference has been observed between analytic predictions and TDS techniques [1,11]. Analytic prediction is realized by iterative analytical solution of time-changing force coefficient of mathematical model formed by applying it to the distribution of Fourier series. This analysis is made with the acceptance that the force process is linear according to feed and depth which doesn't depend clearly on cutting speed. Additionally, studies have been made recently to mount a sensor/actuator on the tool/tool holder system on the machine tool designed to suppress chatter vibration [13,19,20]. In the cutting process dealt with in this section, SLDs, which give stable and unstable cutting regions for chatter-free cutting, will be drawn according to two different forms explained in the previous section depending on cutting depth and spindle speed.

3.1. According to oriented transfer function

If the denominator of Eq. (7), which is the characteristic equation of the system, is equaled to zero,

$$1 + a_{lim} K_f G(s)\left(1 - e^{-\tau s}\right) = 0 \tag{18}$$

where, $a_{\lim}$, means chatter-free maximum cutting depth. The roots of this characteristic equation will give the chatter frequency of the system in the form of $s = \sigma + j\omega$. When the real part of the roots is zero ($s = j\omega$), the system is critically stable and workpiece oscillates with constant vibration amplitude at chatter frequency (ω). If $s = j\omega$ is placed in characteristic equation for critical borderline stability analysis, Eq. (18), can be written as,

$$1 + a_{lim}\,K_f\,G(j\omega)\left(1 - e^{-\tau j\omega}\right) = 0 \tag{19}$$

when it is placed in Eq. (19) in the form of real and imaginary parts $G(j\omega) = \mathrm{Re} + j\mathrm{Im}$, characteristic equation can be written in the form of real and imaginary parts, $\{1 + a_{lim}K_f[Re(1\text{-}\cos\omega\tau)\text{-}Im\sin\omega\tau]\} + j\{a_{lim}K_f[Re\sin\omega\tau + Im(1\text{-}\cos\omega\tau)]\} = 0$. For stability, both real and imaginary parts of the equation must be zero. If the imaginary part is equaled to zero first, $Re\sin\omega\tau + Im(1\text{-}\cos\omega\tau) = 0$, the ratio of real and imaginary parts gives the phase angle (ψ) of the root on Nyquist diagram,

$$\tan\psi = \frac{Im(\omega)}{Re(\omega)} = \frac{\sin\omega\tau}{\cos\omega\tau - 1} \tag{20}$$

This value is the phase delay of the frequency transfer function of the system. If the half-angle formula in trigonometry is applied [1],

$$\omega\tau = 3\pi + 2\psi \tag{21}$$

The phase effect in regenerative chatter vibration can be written as follows,

$$n + \frac{\varepsilon}{2\pi} = \frac{f}{\Omega} = f\tau \tag{22}$$

Where, f is frequency of the cutting tool (Hz); Ω is the spindle speed (1/s); $\varepsilon/2\pi$ the fractional number of waves formed on the surface. It can be seen here that there is a phase shift between the inner and outer waves $\varepsilon = 3\pi + 2\psi$. The corresponding spindle period and maximum spindle speed are found as,

$$\tau = \frac{2n\pi + \varepsilon}{2\pi f} \rightarrow \quad N = \frac{60}{\tau} \qquad n = 0, 1, 2, 3, \ldots \tag{23}$$

The critical axial depth of cut can be found by writing the reel part of the characteristic equation to zero. Hence,

$$a_{lim} = \frac{-1}{K_f Re[(1 - \cos\omega\tau) - (Im\,/\,Re)\sin\omega\tau]} \rightarrow \frac{Im(\omega)}{Re(\omega)} = \frac{\sin\omega\tau}{\cos\omega\tau - 1}$$
$$a_{lim} = \frac{-1}{2K_f Re(\omega)} \tag{24}$$

is calculated.

3.2. According to τ-decomposition form

According to τ-decomposition form, the characteristic equation of the system is Eq. (15). The roots of this equation are obtained from the solution of Eq. (17). Each positive real root ($\omega_i(j\omega)$) is substituted back into the right side of Eq. (16) to find$U_2(j\omega_i)$. The phase angle of the resulting number is computed as follows;

$$\psi_i = \tan^{-1}\frac{\mathrm{Im}(U_2(j\omega_i))}{\mathrm{Re}(U_2(j\omega_i))} \tag{25}$$

Again, according to Eq. (22), the value of time delay is found,

$$\tau_i = \frac{\psi_i + 2\pi n}{\omega_i} \qquad n = 0, 1, 2, 3, \ldots\ldots \tag{26}$$

Depending on this, maximum spindle speed to be gained in stable cutting is found as in Eq. (23). If critical axial depth of cut required for the borders of stable cutting is organized and as seen in Eq. (11) in Fig. 3 and according to Nyquist criterion, can be found in relation to point (-1,0j) of the unit circle.

$$aK_f G(s)(1 - e^{-\tau s}) = (-1, 0j) \tag{27}$$

If the equation is reorganized in a form of $s = j\omega$, it can be calculated by the following expression

$$aK_f\, G(\omega)\left(1 - e^{-j\omega\tau}\right) = -1 \tag{28}$$

Thus, the borders of stable cutting corresponding to spindle speeds given in Eq. (28), are calculated with the expression

$$a_{lim} = \frac{1}{K_f\, G(1 - e^{-j\varepsilon})} \tag{29}$$

Table 1
Data of cutting and modal analysis parameters.

Cutting data		Modal analysis data	
N (rpm)	1000	C (N/m^2)	1.67×10^9
s (mm/rev)	0.12	k (N/m)	1×10^6
a (mm)	1.2	ω_n (Hz)	773
β (°)	70	ξ (%)	2

Since K_f, is a real value, the real value of $G(1 - e^{-j\varepsilon})$ is included in the process and the equation becomes as follows,

$$a_{lim} = \frac{1}{2K_f Re(G(j\omega))} \tag{30}$$

4. A comparison of numerical calculation and test results

In the current study, AISI-1040 steel has been used as a workpiece material whose diameter is 64.9 mm. The workpiece is cut by Kennametal (SDJR-2525M11 NA3) inserts on universal lathe TOS SN50C. Tool holder dimensions are (b×h×l) = (25×25×110) mm. Data required for calculation, cutting and dynamic parameters determined as a result of model analysis performed by an impact hammer are given in Table 1. Dynamic parameters were determined by using a modal test, CutPro®MalTF software and CutPro®Modal software. The other samples and results related with this subject have been given by Dohner et al. [20].

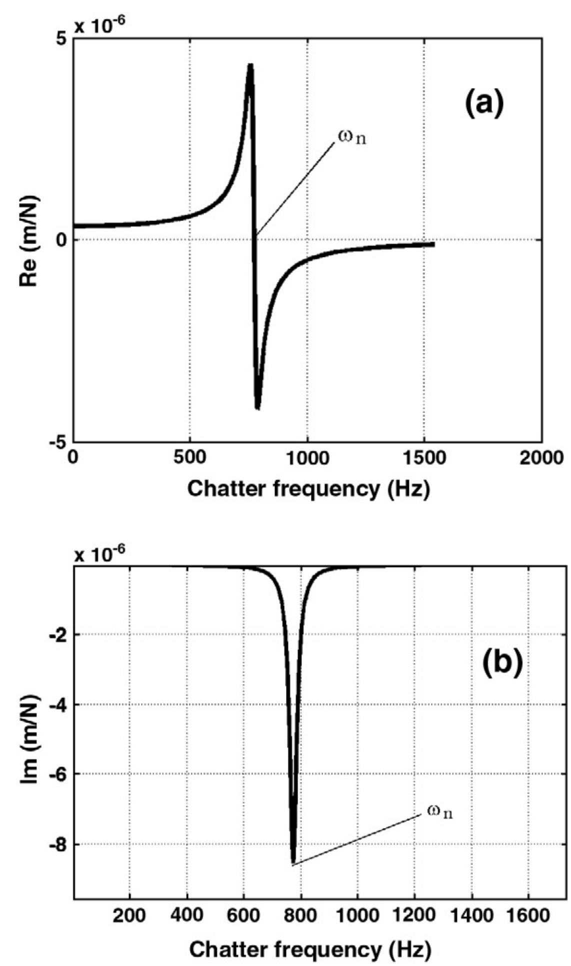

Fig. 4. Transfer function of a SDOF system represented by its (a) real and (b) imaginary parts.

In this study, a system of orthogonal cutting was dealt with as a SDOF. Analytical modeling of this system and investigation of its stability were conducted in two different forms. The movement of the tool in (x) direction for SDOF system was neglected because the natural frequency of the tool in this direction was very low in relation to the other direction. For this reason, it was accepted that the movement in this direction had no effect on the stability limit of orthogonal cutting. Some simplifications were made in order to obtain a useful system model. These were observed to be effective on the results. According to OTF form, which was dealt with first in the study, curves of real and imaginary parts of the transfer function of SDOF system are given in Fig. 4. Thus, the frequency corresponding to the minimum negative real part of the transfer function can be predicted as chatter frequency [2]. This value, according to the form calculated here, was predicted to be 789 Hz. Prediction chatter frequency in both calculation forms are explained in this study, as expected [1,2], is greater than the natural frequency that was found as a result of the modal analysis of the system.

Later, equations of motion of the system were organized in τ-decomposition form. For this reason, equations were expressed in terms of relative displacement on the workpiece of the tool (l) instead of time. Stability of the system was predicted according to Nyquist criterion. By Eq. (16), critical orbital curve (unit circle) was plotted as $s=j\cdot\omega_i$, $U_1(s)=e^{sd\pi}$ and Nyquist place curve was plotted as $U_2(s)=\frac{-1}{a_2\cdot s^2+a_1\cdot s+a_0}$ as it can be seen in Fig. 5(a). Subscript ($i=1, 2$) is the number of positive real root of characteristic equation of the system (Eq. (17)).

The plot of this curve has been dawn by increasing and decreasing the real roots of characteristic equation of the system (17) in a way that they would enter or leave into the unit circle. The system is stable only in intervals where the value of this root is zero. In order to screen the frequency intervals in simulation, the roots were increased in such a way that they would be around of the natural frequency. Thus, while the curve $U_2(j\omega_i)$enters into the unit circle $U_1(j\omega)$for an increase of one root, the other would leave. The value of frequency at the points where the curve $U_2(j\omega_i)$ intersects the unit circle $U_1(j\omega)$is the value of natural frequency(ω_n). These points are in the regions 3rd and 4th of the unit circle, respectively. This situation can be observed from magnitude graphics $|U_2(j\omega_i)|$ in Fig. 5(b) plotted for the increase of both root values. Again the frequency value at the point where the two graphics coincide corresponds to the natural frequency of the system. This situation is natural. Because at this point of coincidence, as it can be seen from Eq. (16), the term is $|U_1(j\omega)|=|U_2(j\omega_i)|=1$. Lines and dashed lines in Fig. 5(b) and in all graphics plotted with this method represent the increases related with the curve $U_2(j\omega_i)$ for each root value coinciding with the circles U_2 and U_1 in the 3 and 4 complex planes respectively. Real and imaginary graphics of the expression $U_2(j\omega)$ are given in Fig. 6. The prediction of chatter in this method was made according to Nyquist criteria [2]. Since the term C in Eq. (29) was chosen to be real and $a_{\lim}$ was found to be real, the term $G(1-e^{-j\varepsilon})$should also provide a real value. So the equation in Fig. 5(a) should be $|U_2(j\omega_i)|=|U_2(j\omega_i)e^{-j\varepsilon}|$. Thus, $(U_2(j\omega_i)-U_2(j\omega_i)e^{-j\varepsilon})$ becomes parallel to real axis and the expression $U_2(j\omega_i)-U_2(j\omega_i)e^{-j\varepsilon}=2\cdot\mathrm{Re}(U_2(j\omega_i))$ is obtained. According to all said, frequency value corresponding to the point equal to the root value on Nyquist diagram for each imaginary value of the curve $U_2(j\omega_i)$ can be taken as the value of chatter frequency (see Fig. 6(b)). According to this calculation, the value was predicted as 779.5 Hz.

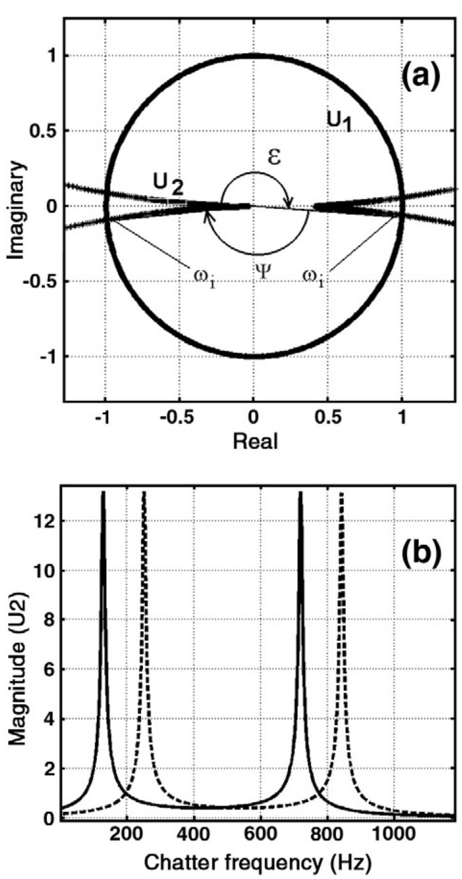

Fig. 5. Plot of (a) U_1, unit circle and roots in the complex plane (b) magnitude of (U_2).

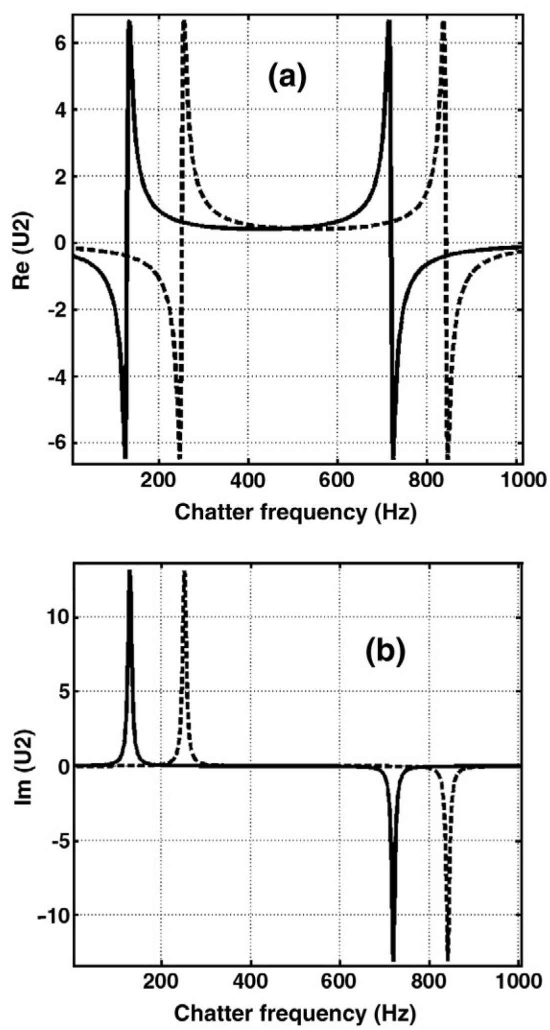

Fig. 6. Plot of U_2, (a) real part and (b) imaginary parts.

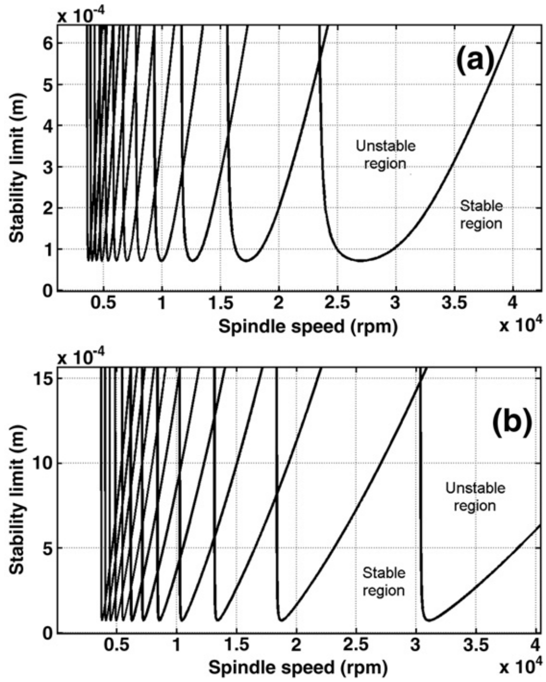

Fig. 7. Plot of SLDs respect to (a) OTF Form and (b) τ-decomposition form.

SLDs, which were plotted depending on the depth of cut and spindle speed according to both forms dealt with in the current study, are given in Fig. 7. As it can be seen in Fig. 7, due to some calculus differences between the two forms, stable depth of cut presents differences at high cutting speeds. This difference between the borders of stable depth of cut gradually narrows at cutting speeds that are appropriate for working in general. This situation is shown in Fig. 8. Besides, for spindle speed of N = 1000 rpm, stable depths of cut,

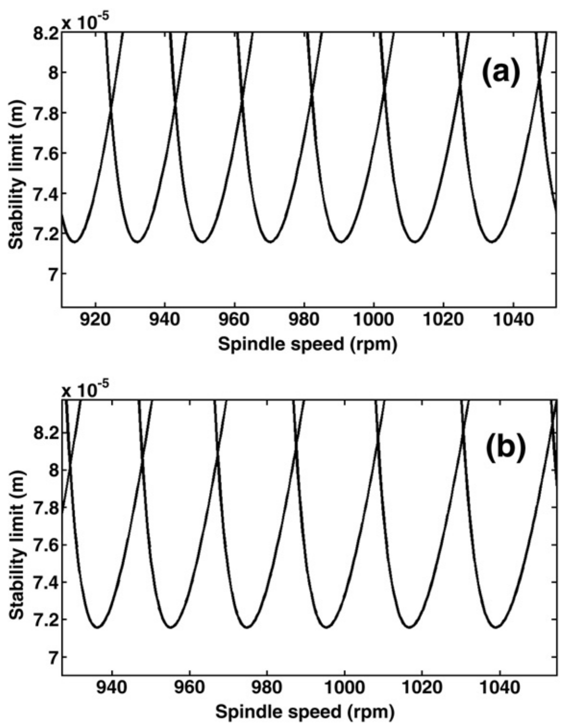

Fig. 8. Plot of SLDs at about 1000 rpm respect to (a) OTF Form and (b) τ-decomposition form.

according to OTF and τ-decomposition forms, were determined as 7.6×10^{-5} m and 7.3×10^{-5} , respectively.

After attaching an accelerometer with a sensitivity of 10.43 mV/g on the surface of the tool holder in the feed direction, modal analysis test of SDOF was conducted by applying small impacts by impact hammer at appropriate point again in the same direction. Time response of the accelerator was measured and transformed into data frequency domain. For this transformation of time data, Fast Fourier Transform (FFT) was utilised. At all steps of modal analysis, a portable computer was used for collecting data, calculating modal parameters and presenting the results. All data were collected by using CutPro®MalTF software, and modal analysis was performed by using CutPro®MalTF software [1]. Real imaginary graphics of the transfer function of the system obtained from the current study are given in Fig. 9 and modal parameter values are given in Table 1.

Cutting test for SDOF system was performed under the conditions given in Table 1 and not using cooling fluid. Data of cutting test were processed by LabVIEV 7 software that is loaded into the same computer. Noise produced during this process was recorded by a microphone attached to power supply. Noise data recorded was recorded to time domain via LabVIEV 7 software as seen in Fig. 10. Then this noise recording was transformed into frequency domain by the software and spectrum graphic was formed. The frequency corresponding to the highest amplitude in the spectrum graphic can be determined as current chatter frequency of the tool performing the process of cutting under these valid conditions. The chatter frequency determined here is 763 Hz.

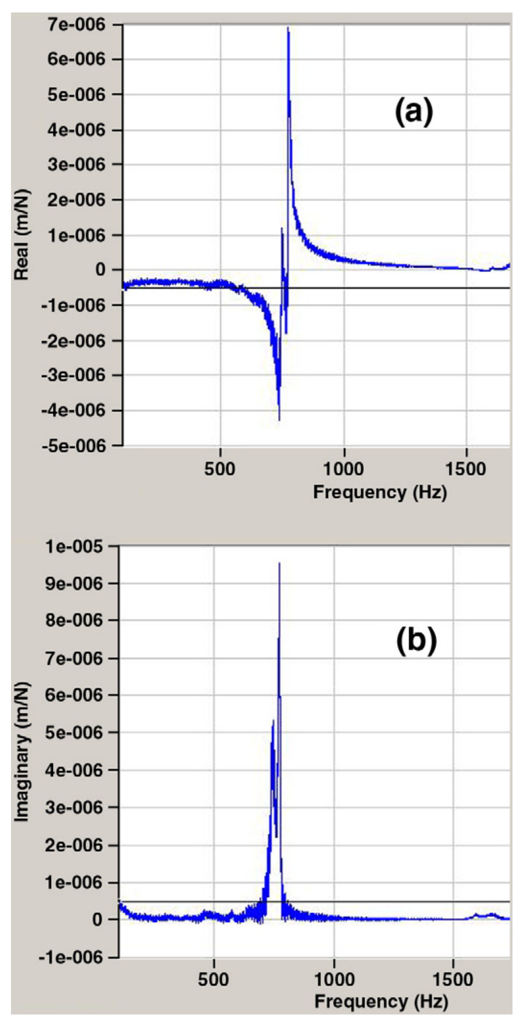

Fig. 9. Real and imaginary graphics obtained by modal analysis of the SDOF system.

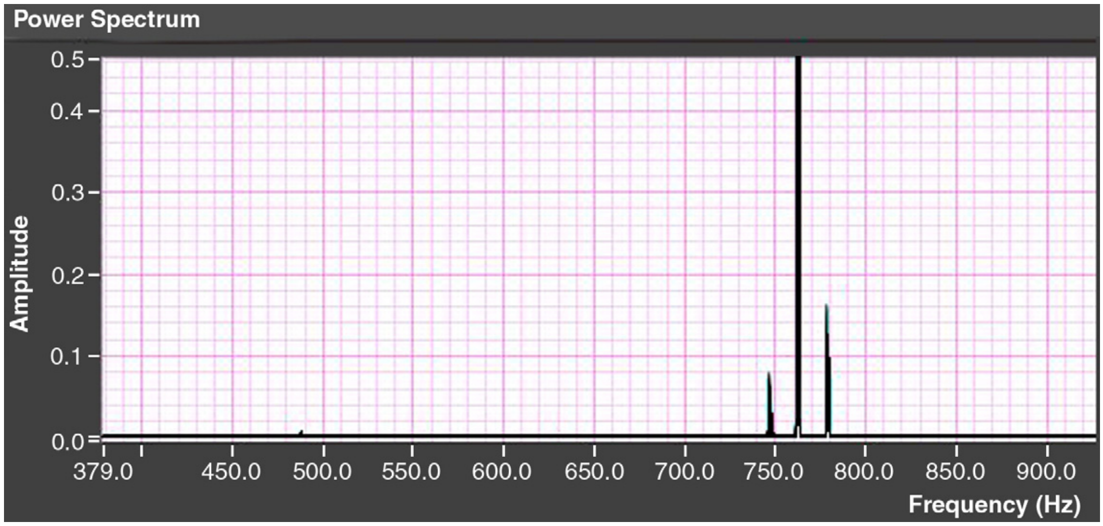

Fig. 10. Graphics of power spectrum for the SDOF system.

5. Conclusions

The present study focused on a turning system with SDOF in orthogonal cutting process. Prediction of the chatter frequency of the system and the investigation of its stability were conducted in two separate forms; one was in the form OTF and the other was in the form of applying τ-decomposition form to the Nyquist criterion. Results obtained from both forms were compared with the modal analysis conducted and the results of cutting test. Prediction chatter frequency in both calculation forms explained in this study is greater than the natural frequency that was found as a result of the modal analysis of the system. But according to the result of this cutting test, it was found that the value of this frequency is, in fact, under the natural frequency of the system. The fact that both of the calculation forms have analytical and linear structure may be a factor that affects these results. Besides, in predicting chatter frequency, the clamping situation of the tool holder is fairly important. During various modal analysis made, it was observed that a second mode appeared in real and imaginary graphics when the attachment of the tool holder was not rigid enough. Another factor affecting these results is that the damping in process and cutting process were not taken into account. The stability of the system is affected by dynamic factors such as nose radius of the tool holder, sharpness or bluntness and built-up edge. Although it's written in literature how these factors affect the stability of the system, there is no complicated modeling that takes into consideration all of dynamic factors yet. Although all these effects were considered in this study, a deviation of 3.3% in OTF form and 2.12% deviation in τ-decomposition form were observed in chatter frequency prediction compared with current chatter frequency. Under current working conditions, according to both forms of calculus, the results in the prediction of the stability of the system were quite similar. However, it was observed that this difference is larger in high cutting speeds.

Acknowledgment

This work was supported by Science Research Projects Commission of the Eskisehir Osmangazi University. The supported project name is “Stability Analysis of Chatter Vibration on Machine Tools”, and its number is 200615012.

References

[1] Altintas Y. Manufacturing automation; metal cutting mechanics, machine tool vibrations and CNC design. first ed. New York: Cambridge University Press; 2000.
[2] Tlusty G. Manufacturing processes and equipment. first ed. New Jersey: Prentice Hall; 2000.
[3] Merchant ME. Basic mechanics of the metal-cutting process. J App Mech II Trans ASME 1944;66 A-168.
[4] Arnold RN. The mechanism of tool vibration in the cutting of steel. Proc Inst Mech Eng 1946;154:261–76.
[5] Tobias SA. Machine tool vibration. first ed. London: Glasgow; 1965.
[6] Koenigsberger F, Tlusty J. Machine tool structures-stability against chatter. Stability against chatter, vol. 1. Oxford: Pergamon Press; 1967.
[7] Tlusty J. Machine dynamics. In: King RI, editor. Handbook of high speed machining technology. New York: Chapman & Hall; 1985.
[8] Anderson CS, Semercigil SE, Turan ÖFA. Passive adaptor to enhance chatter stability for end mills. Int J Mach Tools Manufactures 2007;47:1777–85.
[9] Landers RG, Ulsoy AG. Chatter analysis of machining systems with nonlinear force processes, 58. Atlanta, Georgia: ASME International Mechanical Engineering Congress and Exposition; 1996. p. 183–90. DSC.
[10] Nosyreva EP, Molinari A. Nonlinear analysis of chatter in orthogonal cutting, dynamics of rigid or flexible systems. C R Acad Sci Paris 1997:435–42 t. 325 Serie II b.
[11] O'Reilly TB. The effect of variable delay length on stability in orthogonal cutting. Machine Dynamics Research Lab. Master of Science. The Pennsylvania State University Mechanical Engineering; 1998. p. 63–142.
[12] Xiao M, Karube S, Soutome T, Sato K. Analysis of chatter suppression in vibration cutting. Int J Mach Tools Manuf 2002;42:1677–85.
[13] Lagö TL, Olsson S, Håkansson L, Claesson I. Performance of a chatter control system for turning and boring applications. 4th GRACM Congress on Computational Mechanics, Patras, Greece, 27–29 June 2002; 2002.
[14] Altintas Y, Budak E. Analytical prediction of stability lobes in milling. Ann CIRP 1995;44(1):357–62.
[15] Olgac N, Hosek M. A new perspective and analysis for regenerative machine tool chatter. Int J Mach Tools Manuf 1998;38:783–98.
[16] Budak E. An analytical design method for milling cutters with nonconstant pitch to increase stability Part I: theory. J Manuf Sci Eng Trans ASME 2003;125:29–34.
[17] Budak E. An analytical design method for milling cutters with nonconstant pitch to increase stability Part II: application. J Manuf Sci Eng Trans ASME 2003;125:35–8.
[18] Turkes E. Theoretical and experimental analysis of process damping in machine tool chatter vibration. Doctoral dissertation, Eskişehir Osmangazi University Applied Science Institude, Eskişehir, Turkey; 2007.
[19] Pan J, Su CY. Chatter suppression with adaptive control in turning metal via application of piezoactuator. Proceedings of the 40th IEEE Conference on Decision and Control, Orlando, Florida, USA; 2001.
[20] Dohner JL, et al. Mitigation of chatter instabilities in milling by active structural control. J Sound Vib 2004;269:197–211.