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SELF – EXCITED VIBRATION IN ORTHOGONAL CUTTING WITH TIME LAG+

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Abstract

Tests have been conducted with a newly designed force dynamometer which has a natural frequency 500 Hz and cross sensitivity (to measure the forces accurately and separately) is less than 2.5 %. The calibrated static forces of the dynamometer in x-axis direction is about 2500N and in y-axis direction is about 3000N with 50% safety factor.
The dynamic cutting coefficients are evaluated from the steady- state cutting parameters. Modifications are made in the feed/penetration rate factor in order to determine the dynamic chip thickness coefficients. An mathematical approach was also made for investigating of the dynamic cutting forces.

Dynamic instability occurs as a result of the chip thickness variations. The use of simple damper reduces the chatter and the cutting forces when increasing the feed .The simple damper has active vibration damping to dynamically stabilize the cutting process is the presented in this paper .

By using computer program , the desired stability charts of the machine tool structure for different cutting conditions is determined .

المستخلص

ان الاختبارات التي عملت في هذا البحث قد تحققت بواسطة استخدام جهاز جديد لقياس قوى القطع في المحورين وبشكل دقيق . ومواصفات هذا الجهاز هي :ان التردد الطبيعي لهذا الجهاز كانت 500 Hz ، وحساسية التقاطع (لقياس قوى القطع بشكل دقيق ومنفصل عن بعضها البعض) كانت 2,5% . علما بان هذا الجهاز صمم عند التعبير لقياس قوى القطع في محور x-axis كانت 3000N ، ومعامل الامان لهذا الجهاز كانت 50% .

ان معاملات القطع الديناميكية قد تم حسابها على التغذية / معامل معدل الاختراق لكي نحسب بواسطتها معاملات سمك الرايش الديناميكية .

وقد استخدمت طريقة رياضية مناسبة في حساب قوى القطع الديناميكية. وكان الاهتزاز الديناميكي يحدث كنتيجة لتغييرات سمك الرايش .

عند استعمال مخفض الصدمة البسيط قد اعطى نتيجة واضحة في تقليل الاهتزاز وقوى القطع عند زيادة التغذية .
وان مخفض الصدمة البسيط له تاثير فعال على الاتزان في تخفيض الاهتزاز الديناميكي عند عملية القطع .

وبواسطة استخدام برنامج في الحاسوب قد حصلنا على منحنيات اتزان عديدة عند حالات القطع المختلفة.

Introduction

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Modern manufacturing technology is characterized by high demands on accuracy and productivity in an automated and flexible manufacturing environment. High cutting velocities, high spindle speeds and high feedrates that could lead to increased accuracy and productivity, are, however, frequently limited by machine tool chatter [1].
To control chatter most researches have concentrated on the following approaches :

1- Alleviating the regenerative effect by changing the phase shift between cutting waves through spindle speed variation [2,3].
2- Altering the tool – workpiece relation by following desired tool positions to reduce tracking errors in turning [4].
3- By actively changing the tool geometry, e.g. rake and clearance angle variation in turning [5].
4- Reducing the net energy into the machining system by using active dampers [6,7].

In order to improve accuracy, decrease surface roughness, and reduce cost, it is necessary to take suitable corrective actions to suppress machine tool chatter [1].
In turning operations, vibrations are a common problem as they affect productivity and performance of the machine tool. Due to the occurring vibrations, the geometrical accuracy as well as the surface quality are reduced [8].
Active vibration control has been investigated in many applications and much research has been performed to control the tool position in machining operation for the purpose of process improvement [4].

When the machine slides are moved during cutting operations, the distribution of masses, springs and levers changes. At the same time, the mechanical guiding are being pre – stressed when the tool is in contact with the work. This leads to increased natural frequencies compared to machine tool standstill [8].

In turning operations the tool and tool holder shank are subjected to dynamic excitation due to the deformation of work material during the cutting operation. The stochastic chip formation process usually induces vibrations in the machine tool system [4].
With machine tools, the demand free precision and speed has led to continuous improvement in dynamic behaviour [9]. The tool vibrations in a turning operation mainly comprise vibrations in two directions, the cutting speed direction and the feed direction. Usually, the vibrations in the cutting speed direction and the feed direction are linearly independent [10].

THEORY

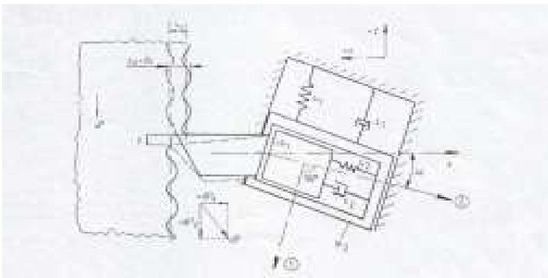

Fig.1. The mechanical model

Fig.1. shows the mechanical model, where the tool is assumed to vibrate relatively to the rigid workpiece in orthogonal cutting.

The dynamic cutting forces

Under steady- state cutting conditions the cutting force is defined as [11]

$$p = f(s_0, v_0)$$

The following are considered

$$v = R\Omega, \qquad \frac{\partial p}{\partial v_0} = k_v$$

$$dv = Rd\Omega, \qquad k_v = \frac{k\Omega}{R}$$

Therefore, the cutting force variation can be written as $dp = \frac{w}{s_0} k_s ds + k_v R d\Omega$ (1)

Under the dynamic cutting conditions the cutting force is a function of three independent factors.

$$p = f(s_0, r, v_0)$$

The cutting force variation for small changes in these factors can be written as :

$$dp = k_1 ds + kdr + k_3 dv \qquad (2)$$

Where k_1 , k_2 , and k_3 being the dynamic cutting coefficients

After calculating the dynamic cutting coefficients the dynamic cutting force becomes as:

$$dp = k_1 ds + (\frac{w}{s_0} k_s - k_1)Tdr + [k_v - (\frac{w}{s_0} k_s - k_1)T \frac{s_0}{2\Pi R}]dv \qquad (3)$$

Where ds, dr and dv are functions of time and are independent from each other,calculated as :

$$ds = (x - x_T) + v(y - y_T) \qquad \text{when,} \qquad x_T = x(t - T)$$

$$dr = x + vy \qquad y_T = y(t - T)$$

$V = 0.69\frac{h}{L}$ the displacement coefficient as Tobias [11]

The calculated dynamic cutting force in the x –axis could be writing from equ.(3)

$$dp_X = k_{1X}[(x - x_T)] + v\ (y - yT) + (\frac{w}{S_o} k_{sx} - k_{1x})T(x + vy) \qquad (4)$$

when $(\frac{w}{so} k_{sx} - k_{1x})$ is the penetration rate coefficient in the x-axis direction. The penetration rate factor C_0 considered by knight [13] is

$$C_0 = \frac{k_s - k_1}{k_1}$$

Similarly, the dynamic cutting force in the y-axis direction was determined, as :

$$dp_y = k1_y[(x - x_T) - v(y - y_T)] + [k_{vy} + (\frac{w}{s_o} k_{sy} - k_{1y})T \quad (1 - \frac{s_o}{2\Pi R})](x + vy) \qquad (5)$$

where , $(\frac{w}{so} k_{sy} - k_{1y})(1 - \frac{s0}{2\Pi R})$ is the penetration rate coefficient in y- axis direction .

The stability condition

The equations of motion of a two – degree of freedom system are written as [11]

$$(6)\begin{bmatrix} m_x & o \\ o & m_y \end{bmatrix}\begin{bmatrix} x \\ y \end{bmatrix} + \begin{bmatrix} c_x & c_{xy} \\ c_{xy} & c_y \end{bmatrix}\begin{bmatrix} x \\ y \end{bmatrix} + \begin{bmatrix} s_x & s_{xy} \\ s_{xy} & s_y \end{bmatrix}\begin{bmatrix} x \\ y \end{bmatrix} + \begin{bmatrix} dp_x \\ dp_y \end{bmatrix} = \underline{0}$$

Let us solve the matrix equation (6) by considering the following

$x(t) = \alpha e^{\lambda t}$

$x(t-T) = \alpha e^{\lambda(t-T)}$

$y(t-T) = \beta e^{\lambda(t-T)}$ and $y(t) = \beta e^{\lambda(t)}$

Substituting these in Eq. (6) and for trivial solution of matrix equation , if :

$\begin{bmatrix} a \\ \beta \end{bmatrix} \neq \underline{O}$ but $\begin{bmatrix} a \\ \beta \end{bmatrix} = \underline{O}$

The characteristic equation is written as

$$D(\lambda) = \begin{vmatrix} a_{11}\lambda^2 + b_{11}\lambda + d_{11} + e_{11}e^{-\lambda T} & b_{12}\lambda + d_{12} + e_{12}e^{-\lambda T} \\ b_{21}\lambda + d_{21} + e_{21}e^{-\lambda T} & a_{22}\lambda^2 + b_{22}\lambda + d_{22} + e_{22}e^{-\lambda T} \end{vmatrix} = 0 \qquad (7)$$

$D(\lambda)= Re\ D\ (\lambda) + Im\ D(\lambda)$ (8)

when,

$\sqrt{-1}$ $\iota=$ $\lambda = \iota\psi$,

$e^{-\lambda T}= \cos(\psi T) - \iota \sin(\psi T)$

let

real part $M(\psi) = Re\ D(\iota\psi)$

imaginary part $S(\psi) = Im\ D(\iota\psi)$

Eq. (8) can be rewritten as :

$D(\iota\psi) = M(\psi) + \iota S(\psi)$ (9)

The soluation of Eq. (7) is asymptotically stable if the roots have negative real parts. It can be proved [12] that the system is stable if and only if

$$\sum_{k=1}^{mt} (-1)^{k+1} sign(\psi_k) = -2 \qquad (10)$$

A computer program was written with the aid of the algorithm from the theorem of G. stepan for computation of the positive real parts of Eq. (9) and to check whether Eq. (10) is true or not .

The program has been written in WANG 2200 computer with the BASIC language . The program and the flowchart are attached in appendix (A, B, C, D, E, F).

Results And Discussions

The stiffness, natural frequency , damping coefficients

It is calculated that the natural frequencies, the stiffness and damping coefficients increase as the tool shanks decrease and their values in x-axis direction are more than that in y – axis direction.

The average main cutting force Py as a function of the cutting velocity v was plotted in Fig.2. So that to get different values of coefficients at different main cutting forces. The figure shows non linear relationship between the average main cutting force Py and the cutting velocity v.

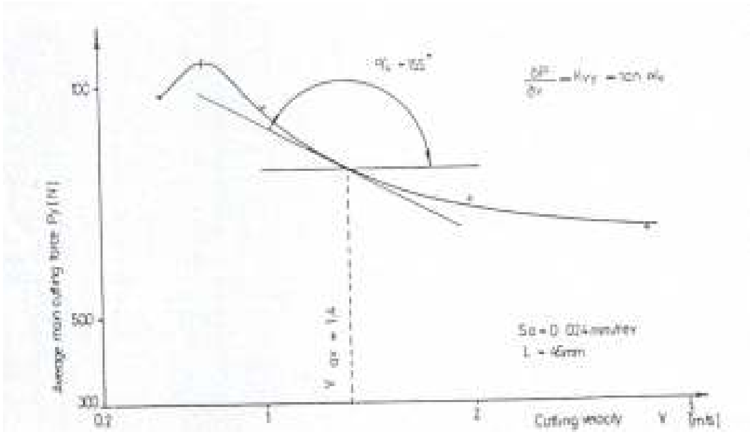

Fig.2. The cutting velocity coefficient

The average main cutting force and the average thrust force are plotted against the feed for, each cutting speed as shown in Fig.3.

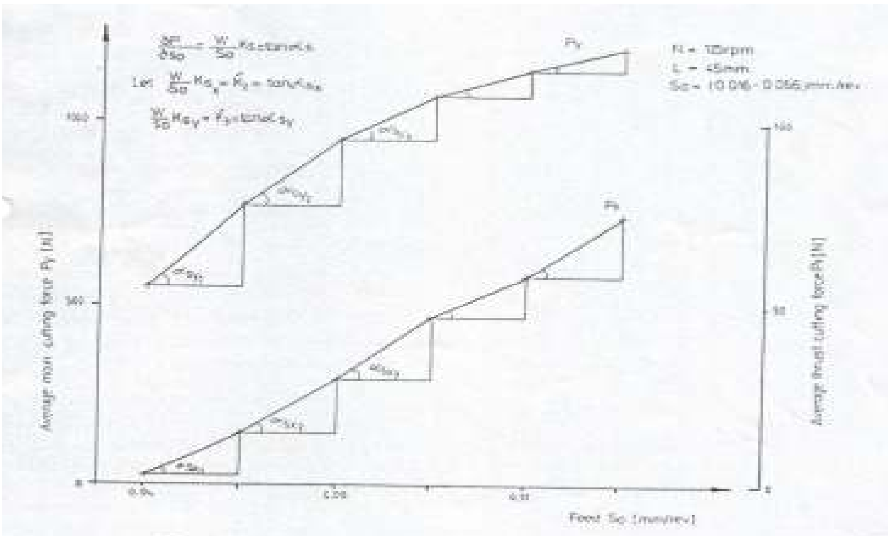

Fig.3. The chip thickness coefficient

The method is used in calculating the chip thickness coefficients in two axis direction by considering the slope between each two feed points. From the results it is concluded that the chip thickness coefficients in the y-axis direction are bigger than in the x-axis direction.

In determining the stability conditions, the coefficients in the x-axis direction have more effects on stability than those in the y-axis direction.

The stability charts

There are different stability charts at different tool shanks. Fig.4. in appendix (G) shows the stability chart for the tool shank L= 25 mm at feed between 0.0084 mm/rev and 0.14 mm /rev and the number of revolution between 63 rpm and 355 rpm.

The penetration rate factor C_0 used was 0.0005 which means that the dynamic chip thickness coefficients K_{1x} and K_{1y} are closely equal to the chip thickness coefficients

$\frac{W}{S0}K_{sx}$ and $\frac{W}{S0}K_{sy}$ as shown in Fig .3.

The predicted stability chart showed 80% agreement with the measurements.Fig.5. in appendix (H) represents the stability chart for tool shank L=45 mm which has two zones. The first zone shows approximately a small area of stability at speeds from 63 rpm to 80 rpm and feeds between 0.016 mm/rev and 0.056 mm/rev.

The second zone shows the stable areas in the unstable region . To get reasonable stability chart, it was urgently necessary to increase or decrease the coefficients to some percentage in programming.

The unstable region has the stable zone too which gives the possibility of diagnosing and reducing the charter by increasing the stable region in the unstable zone and the predicted chart showed 70% of agreement.

Fig.6. in appendix (I) shows the stability chart for the tool shank L=35 mm at feed 0.012mm/rev and 0.042 mm/rev. The chart shows the stability zones changing due to the decrease of dynamic chip thickness coefficients and without changing the values of the damping coefficients. The unstable region contains a stable zone at high cutting speed and feed , the predicted agreement with the experimental values was 70%.

Simple method of reducing chatter

There are several methods used in preventing chatter, one of them is the using a simple damper as shown in Fig.7. for the tool shank L=35 mm .

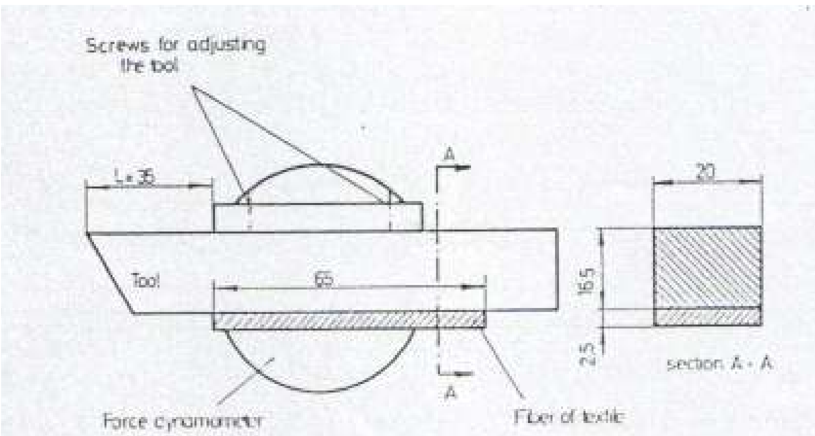

Fig.7. The simple damper during chatter

Experimentally it was noticed that it was impossible to cut at feed more than 0.1 mm/rev because of very fast tool wear and breakage which is noisy and dangerous. The experiment and the calculations showed good reduction of chatter at a feed between 0.04 mm/rev and 0.1 mm/rev .

The stability chart related to this condition of reducing charter is shown in Fig.8. in appendix (J). When using the simple damper , the damping coefficients were increased to 1.5 times and the dynamic chip thickness coefficients were decreased slightly in programming.

These affected the form of stability chart and the agreement between experiment and theory was 70%.

Experimental Method

A CNC turret lathe was used, a workpiece (Ø160X160)mm of A50 – U steel was used as shown in Fig.9. in appendix (K).

The tests were made at different number of revolution, feeds and tool shank, but at constant width of cut and tool geometries.

The recorded signals of cutting forces from the designed force dynamometer in the x-axis direction and acceleration vibration in y-axis direction were recorded on a tape-recorder . The recorded signals were evaluated by an FFT program in the digital analyzer system for evaluating the vibration amplitude, cutting forces and their relations with time and spectrum .A tipped carbide tool of type DA/20 was used with the following cutting conditions:

- tool geometries : relief angle $\alpha = 5^\circ$, rake angle $\gamma = 10^\circ$ and approach angle $\kappa = 0^\circ$
- tool shank : 19 x 20 mm
- tool overhang : 25 , 35 , 45 mm
- feed (eleven levels) : 0.0084 , 0.012 , 0.0166 , 0.024 , 0.033 , 0.04, 0.06, 0.08, 0.1, 0.12, 0.14 [mm/rev]
- number of revolutions (six levels) : 63 , 90 , 125, 180, 250, 355 rpm

A special force dynamometer was designed which is capable of measuring forces in the x and y directions independently is shown in Fig.10.

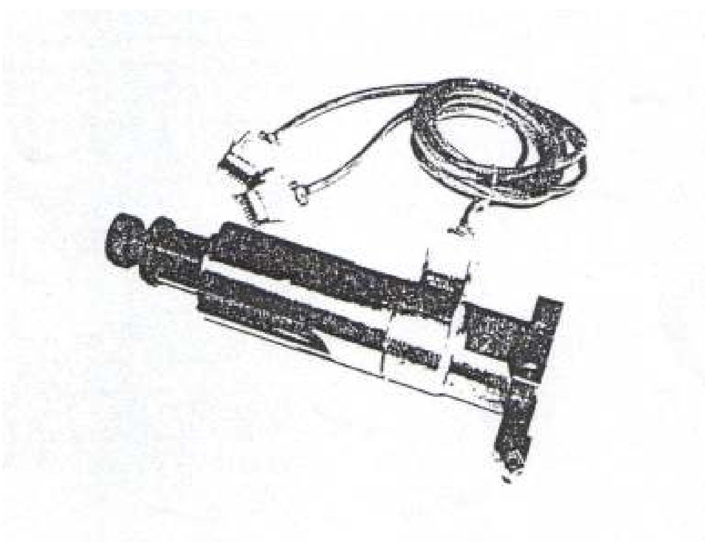

Fig.10. The Force Dynamometer

The designed force dynamometer has a natural frequency about 500 Hz and the cross sensitivity is less than 2.5%. The natural frequency of the cutting tool which was clamped in the force dynamometer was obtained by hitting the rubber put on the tool with different types of hammers in two directions, x and y , at different tool shanks as shown is Fig.11. and .12.

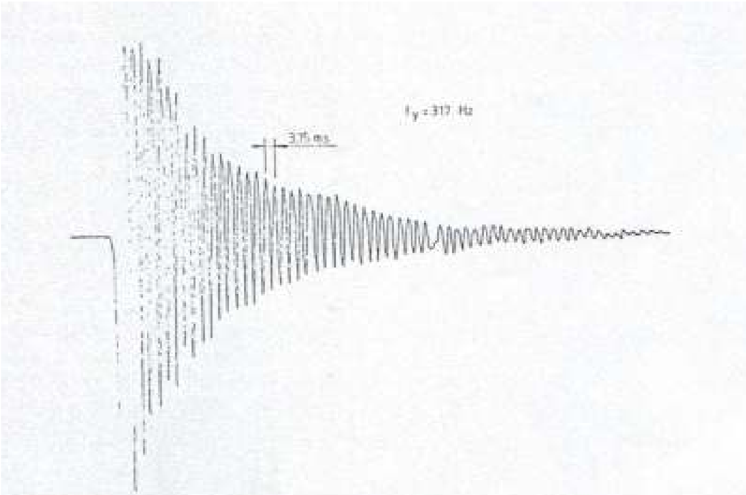

Fig.11. Free vibration of the tool holder in y-axis for the tool shank length L=25mm

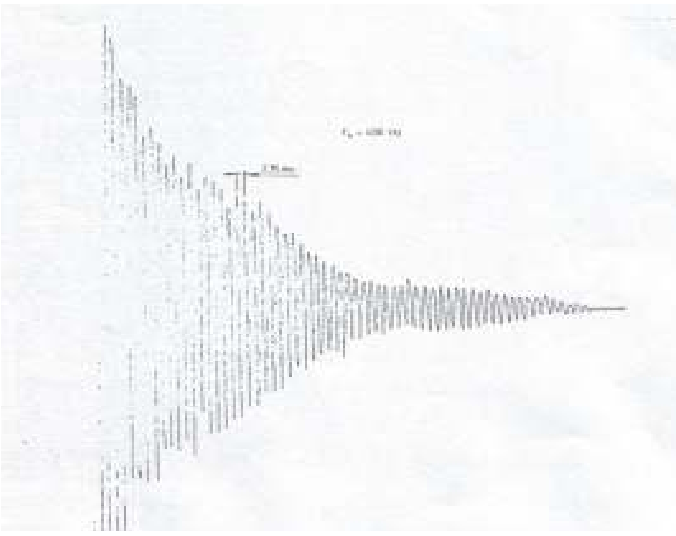

Fig.12. Free vibration of the tool holder in x-axis for the tool shank length L=25mm

The evaluation of stiffness of the tool which was clamped in the tool – holder of the force dynamometer.

The experimentally stiffness is calculated by :

$$S = \frac{Load}{Displacement} \quad [N/M]$$

Conclusions

A simple method was used in determining the chip thickness coefficients. The chip thickness coefficients in x-axis have more effects than in y-axis direction on the stability in orthogonal-cutting .

The use of simple damper reduces the cutting forces as the feed is increased and also gives three times better result to reduce the chatter when increasing the feed.

Increasing the feed produces an increase in the level of stability at high speed and decrease at low speed when the tool shank L=25. These variations correlate with changes in the cutting forces. These findings are important from the general workshop point of view, as by increasing the feed rate to some extent to eliminate chatter.

A new mathematical investigation model had been developed for the prediction of vibrations. The suitable experimental methods were used for measuring the stiffness and damping coefficients in the free vibrations.

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