Mechanical Engineering Research; Vol. 2, No. 2; 2012
ISSN 1927-0607 E-ISSN 1927-0615
Published by Canadian Center of Science and Education

Surface Grinding Machine Stability Characteristics Limited Prediction

Yuting Yang[1], Jianzhong Lin[1] & Shilong Xu[1]

[1] College of Mechanical Engineering, University of Shanghai for Science and Technology, Shanghai, China

Correspondence: Yuting Yang, College of Mechanical Engineering, University of Shanghai for Science andTechnology, 516 Jun Gong Road, Shanghai 200093, China. Tel: 86-188-0192-8470. E-mail: yangyuting080101@126.com

Received: October 22, 2012 Accepted: November 15, 2012 Online Published: November 25, 2012

doi:10.5539/mer.v2n2p114 URL: http://dx.doi.org/10.5539/mer.v2n2p114

Abstract

The chatter in the grinding process has a great influence in improving work piece surface quality and production efficiency. The formula of flutter system limit grinding depth and the rotating speed of the grinding wheel are induced based on the chatter theory and the chatter dynamitic model of the grinding system. The computer modeling and simulation are carried out to get flutter stability predicted picture. Finally the reliability and validity of the predicted picture are verified by the experiments. Flutter stability prediction method provides a theoretical basis in selecting the grinding process parameters for the machine processing operators and it also has an important meaning to the work piece surface quality and processing efficiency.

Keywords: chatter, dynamitic model, simulation, stability prediction, grinding process parameters

1. Introduction

In the grinding process, the vibration of the machine is inevitable. The chatter caused by unstable grinding is an important factor influencing the processing quality. The regenerative chatter caused by grinding wheel and work piece not only affects the precision and surface roughness of the work piece but also affects grinding efficiency. The regenerative chatter is chatter phase difference caused by chatter mark of last time and chatter mark of this time. The depth of cutting will be changed and the chatter will be caused. The production mechanism is very complex and it belongs to nonlinear vibration. It is very difficult to decrease or eliminate the vibration.

According to stability theory analysis of cybernetics and prediction of stability domain of the regenerative chatter of the grinding system, this paper will establish the dynamic model, through the computer simulation to get stable prediction diagramthus to analyze the practicality of the prediction diagram by lots of experiments. It is of great importance to optimize the grinding parameters and improve the processing quality and production efficiency.

2. Method

2.1 The Establishment and Solution of Regenerative Chatter Model

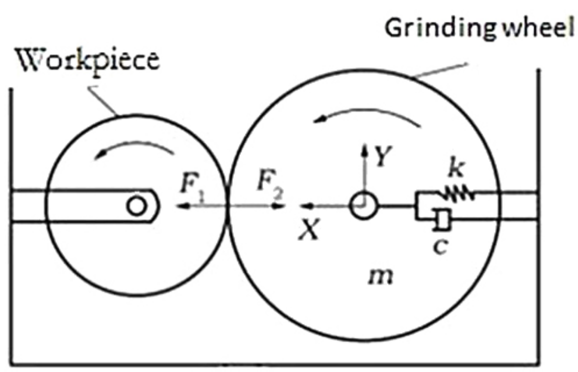

Figure 1. Grinding dynamic model

The equation of motion for grinding dynamic model:

$$m\ddot{y}(t)+c\dot{y}(t)+ky(t)=F(t) \tag{1}$$

m refers to equivalent mass of the machine vibration system, Kg;

c refers to the equivalent damping of machine vibration system, Ns/mm;

k refers to the equivalent stiffness of machine vibration system, N/mm.

Dynamic grinding force caused by the change of material removing rate

$$F(t)=k_m bhS(t) \tag{2}$$

$$S(t)= S_0-[\mu y(x-T)-y(t)] \tag{3}$$

k_m is the coefficient of wheel grinding force, N/mm^3;

b is wheel grinding contact width, mm;

h is wheel grinding depth, mm;

S(t) is the surface chatter mark of workpiece, mm;

y(x-T) refers to grinding vibration displacement of the last time;

y(t) refers to grinding vibration displacement of this time;

μ is the overlap coefficient of two times before and after grinding;

T is period of rotation of grinding wheel.

In $\omega_n=\sqrt{\frac{k}{m}}$, $\zeta=\frac{c}{2m\omega_n}$, ω_n is natural angular frequency of the grinding wheel system, ζ is the damping ratio of the grinding wheel system.

Plug this into (1) we can get

$$\ddot{y}(t)+2\zeta\omega_n\dot{y}(t)+\omega_n^2 y(t)=\frac{F(t)}{m} \tag{4}$$

After Laplace transformation of the above equations, we can get

$$s^2 y(s)+2\zeta\omega_n y(s)+\omega_n^2 y(s)=\frac{F(s)}{m} \tag{5}$$

In the end we can get the transfer function

$$G(s)=\frac{y(s)}{F(s)}=\frac{1}{k}\frac{1}{(\frac{s}{\omega_n})^2+2\zeta\frac{s}{\omega_n}+1} \tag{6}$$

From Engineering Control Theory, we know that time domain feature of the vibration system output depends on the property of the root s in the equation of transfer function. If s=δ+$i\omega$,

δ>0 the system is in unstable state;

δ=0 the system is in critical stable state;

δ<0 the system is in stable state;

If δ=0 the system is in critical stable state, we plug it into (5)

$$G(i\omega)=\frac{1}{k}\frac{1}{(\frac{i\omega}{\omega_n})^2+2\zeta\frac{i\omega}{\omega_n}+1} \tag{7}$$

Grinding force

$$F(t)=\Delta Fe^{i\omega t} \tag{8}$$

Periodic excitation response

$$y(t)= G(i\omega)\,\Delta Fe^{i\omega t} \tag{9}$$

$$y(t\text{-}T)= e^{-i\omega T}G(i\omega)\Delta Fe^{i\omega t} \tag{10}$$

If we plug (8) (9) (10) into (2) (3) we can get:

$$\Delta Fe^{i\omega t}=k_m bh[\mu y(x\text{-}T)\text{-}y(t)]$$

After finishing we can get the condition of chattering:

$$1+k_m bh_{lim}(1-\mu e^{-i\omega T})\ G(i\omega)\ =0 \tag{11}$$

In the formula $e^{-i\omega T}=\cos\omega T - i\sin\omega T$, $\lambda=\frac{\omega}{\omega_n}$, we plug it into (7)

$$1+k_m bh_{lim}[1-\mu(\cos\omega T - i\sin\omega T)]\frac{1}{k}\frac{1}{1-\lambda^2+2\zeta\lambda i}=0 \tag{12}$$

We can get this formula: $1\text{-}\lambda^2+2\varsigma\lambda i = \text{-}k_m bh_{\lim}[1-\mu(\cos\omega T - i\sin\omega T)/k$, according to the real part equals the imaginary part in complex number, we can get:

$$1\text{-}\lambda^2=\text{-}\frac{k_m bh_{\lim}}{k}(1-\mu\cos\omega T)$$

$\frac{2\varsigma\lambda}{1-\lambda^2}=\frac{\mu\sin\omega T}{1-\mu\cos\omega T}$ we can get the following formula after combing the above two:

$$N=\frac{60\lambda\omega_n}{2j\pi+\sin^{-1}\frac{2\varsigma\lambda}{\sqrt{(1-\lambda^2)^2+(2\varsigma\lambda)^2}}-\tan^{-1}(\frac{2\varsigma\lambda}{1-\lambda^2})} \tag{13}$$

$$h_{\lim}=\text{-}\frac{k(1-\mu\cos\omega T)[(1-\lambda^2)^2+(2\varsigma\lambda)^2]}{k_m b(1-\mu\cos\omega T+\mu^2)(1-\lambda^2)} \tag{14}$$

In the formula (j=0,1,2……m), (λ=1,2,3……..n).

2.2 The Stability Limit Diagram Establishment and Experimental Verification

We can know from (13) (14): when all kinds of dynamic parameters (k, k_m, ω_n, ζ, μ, b) in machine grinding system are known, h_{lim} is the function of λ (that is ω) when *j* is indifferent nonnegative inter conditions and the rotate speed N. Therefore, when *j*=0, according to different *λ*, we can get the corresponding spindle speed N and the limit grinding depth h_{lim}, then we can draw the curve h_{lim}-N when *j* = 0 plotted by N on the horizontal axis and h_{lim} on the vertical. In the end, we can get the complete curve of h_{lim}-N when we put all curves when j = 0,1,2,3, 4 … in one diagram.

By using vibration analyzer in surface grinding machine and by doing model experiments we can get all order natural frequency and relevant modal parameters. In grinding wheel rack we set the response point of vibration test and use the force-hammer to hammer just as the following diagram demonstrated.

Figure 2. Experimental measurement

The instrument used in photo above is signal analyzer, in the process of modal experiment, the influence of the random error and noise on the result of measurement should be reduced as far as possible. The grinding parameters get from modal experiment analysis and calculation are shown below:

Table 1. The grinding parameters

$k(N/mm)$	$k_m(N/mm^3)$	$\omega_n(Hz)$	b/mm	ς	μ
3068.2	2745.8	312.58	7.8	0.03	0.98

If we plug the parameters from the above experiment into the formula (13) and (14), we can get the stability limit diagram:

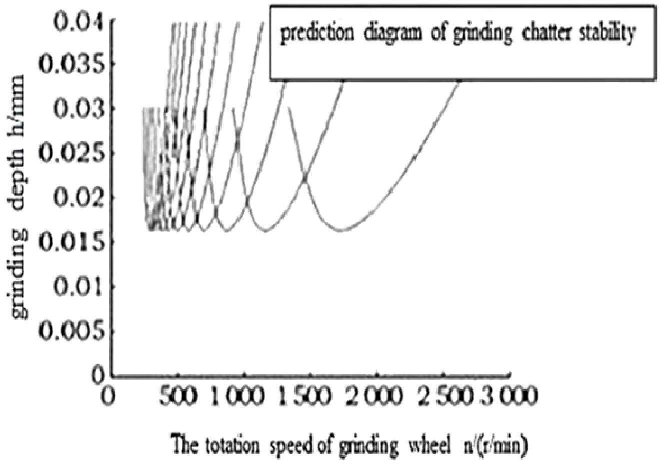

Figure 3. Stability limit diagram

In the curve h_{lim}-N, above the earlobe line it is "the unstable region" of the cutting system and below the earlobe line it is "Absolute stability region", and it is also called "the unconditionally stability region". Between the adjacent earlobe lines it is "conditional stability region". Apparently, h_{lim} that has the "root" of the earlobe line is minimum limit of grinding depth $(h_{lim})_{min}$. As the increase of the grinding wheel speed, the limit grinding depth also has the trend of becoming increase. Therefore, by increasing the grinding wheel speed and keeping the grinding depth less than the limit grinding depth, we can make the process in a stable state.

In order to validate the reliability of the prediction diagram of grinding chatter stability limits, the wheel speed 900 r/min, 1100 r/min, 1300 r/min and 1500 r/min are selected respectively for grinding experiment. The diagram of grinding depth of each wheel below the wheel speed and the prediction diagram are obtained as follows:

Table 2. The grinding depth of each wheel below the wheel speed

Wheel speed/(r/min)	900	1100	1300	1500
Grindingdepth prediction/μm	17.78	17.16	15.85	20.08
Experimental grinding depth/μm	18	17	16	20
Δ/μm	0.02	0.16	0.15	0.08

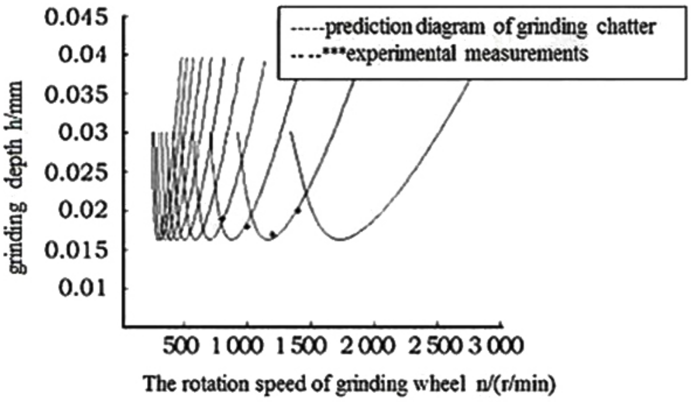

Figure 4. Predicting outcomes of stability limits

As the diagrams shown above, the absolute value between the critical grinding depth of the experiment measurement and the limit prediction diagram of grinding depth are $\Delta \leq 0.17$. It indicates the limit prediction diagram curve is consistent with the experimental measurement results. Therefore, the prediction method is proved to be effective and reliable.

3. Real-time Measures of Inhibiting Chatter

The existence of real-time chatter can lead to the poor quality of surface in the process and the redundant concentrated force will effect in grinding wheel and spindle. So far there are lots of measures that can effectively control chatter such as to use the drive to improve the dynamic stiffness of the grinding machine system and to reduce the regenerative phase.

Maximum grinding may appear in high speed grinding. Continuous curve peak is equal to the integralvalue of the chatter frequency. When it is minimum cutting depth i.e. regenerative phase (-180°), the vibration waveform can be produced at the end of the grinding cycle. In this grinding speed the internal and the external waveform phase are opposite and the maximum regenerative chatter will be produced. The dynamic grinding thickness will be increase and achieve maximum along with the chatter process. So when grinding wheel vibration frequency is equal to a multiple of spindle speed, the frequency can be used as integral value of the chatter frequency and realize chatter minimum and stability best.

Most of the measures cannot be applied from the experiment to real production. The reasons are as follows: grinding chatter happens in low frequency structure. When the frequency of low speed grinding machine can reach 100Hz or the frequency of high speed grinding machine can reach up to 400Hz above, then spindle and grinding wheel and the grinding carriage are greatly affected. Low frequencies are from large machine and various spare parts. The driver must provide large power to make the output stroke to achieve 5-10μm 100Hz bandwidth. High speed machine chatter frequency control need output 10-20μm displacement and bandwidth higher than 1000Hz.

However, the existing driver cannot achieve the above bandwidth and dynamic displacement requirements. Chatter dynamic closed loop system has regenerative feedback delay and it makes the real-time control of nonlinear of the system difficult.

The current problems and challenges lie in nonlinear research in the process of grinding. Grinding wheel and workpiece contact and the vibrating coefficient that influence grinding wheel condition are the main factors that restrict the frequency domain control chatter.

4. Conclusion

According to the dynamic model the prediction diagram is consistent with the experiment results. The prediction method of the grinding regenerative chatter stability limits provides a favorable theoretical basis for machine operators to choose grinding process parameters. It is of great significance to improve the quality of the surface of workpiece and efficiency in the process of grinding machining.

References

Altintas, Y. (2001). Analytical Prediction of Three Dimensional Chatter Stability in Milling. *Society of Mechaincal Engineers, International, 44*(3), 717-723.

Guillem, Q., & Joaquim, C. (2011). Chatter in machining processes: A review. *International Journal of Machine Tools & Manufacture, 51*(5), 363-376. http://dx.doi.org/10.1016/j.ijmachtools.2011.01.001

Han, X. R. (1993). Research on the Construction Dynamic Model of Cutting Vibration of Machine Tool. *Machine Tool & Hydraulics*.

Kong, F. S., & Yu, J. Y. (1998). Fuzzy Stability Analysis of Regenerative Chatter in Cutting Process. *Journal of Vibration Engineering, 11*(1), 106-109.

Ma, H. L., Chen, Q. K., & Duan, H. (2009). Study on the Stability Models of Regenerative Chatter. *Tool Engineering, 2*, 43-45.

Shi, H. M. (1984). Effects of Some Non-Linear Factors on Machine Tool Chattering and Their Mathematical Models. *China Academic Journal Electronic Publishing House, 12*(6), 101-112.

Wu, Y. (1993). *Machine Tool Cutting System Chatter and Control*. China: Science Press.

Yang, L. (2003). *Machine Tool Dynamics*. China Machine Press.

Zhang, S. (1992). *Vibration Test and Analysis Techniques* (Chapter 4). Peking: Tsinghua University Press.