2

Basic Concepts and Theory

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2.1 Introduction

This chapter starts with an introduction of the machine-tool-workpiece loop stiffness and deformation, and then fundamentals of vibrations and followed by the definition and categories of machining chatter. It is not the purpose of this chapter to present the general theory of vibration and chatter in depth as there are a number of excellent books and papers available on these subjects. It is intended from the machining system's viewpoint to provide the basic concept and formulations and the necessary theory background for the following up chapters. Furthermore, the generic concept and classification of machining instability are proposed based on the analysis of various machining instable behaviors and their features.

2.2 Loop Stiffness within the Machine-tool-workpiece System

2.2.1 Machine-tool-workpiece Loop Concept

From the machining point of view, the main function of a machine tool is to accurately and repeatedly control the contact point between the cutting tool and the uncut material - the 'machining interface'. Figure 2.1 shows a typical machine-tool-workpiece loop. The machine-tool-workpiece loop is a sophisticated system which includes the cutting tool, the tool holder, the slideways and stages used to move the tool and/or the workpiece, the spindle holding the workpiece or the tool, the chuck/collet, and fixtures, etc. If the machine tool is being taken as a dynamic loop, the internal and external vibrations, and machining processes should be also integrated into this loop as shown in Figure 2.2.

Stiffness can normally be defined as the capability of the structure to resist deformation or hold position under the applied loads. Whilst the stiffness of individual components such as spindle and slideway is important, it is the loop stiffness in the machine-tool system that determines machining performance and dimensional and forming accuracy of the surface being machined, *i.e.*, the relative position between the workpiece and the cutting tool directly contributes to the precision of a machine tool and correspondingly leads to the machining errors.

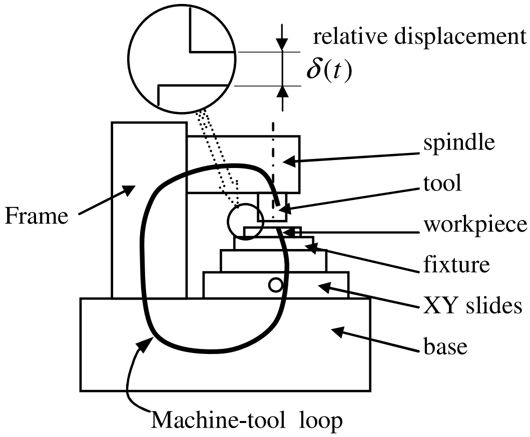

Figure 2.1. A typical machine-tool loop

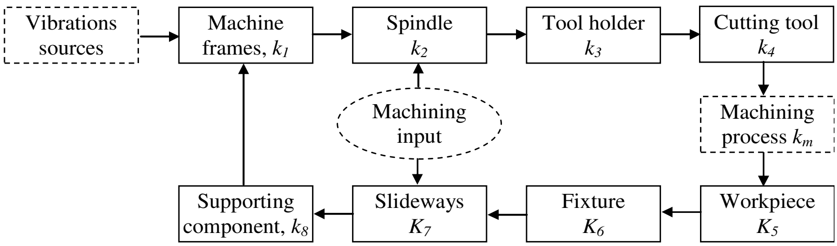

Figure 2.2. The machine-tool-workpiece loop taking account of machining processes and dynamic effects

2.2.2 Static Loop Stiffness

Static loop stiffness in machine tools refers to the performance of the whole machine-tool loop under the static or quasi-static loads which normally come from gravity and cutting forces in machine tools.

A simplified analogous approach to obtaining the static loop stiffness is to regard the machine tool individual elements as a number of springs connected to each other in series or in parallel, so that the static loop stiffness can be derived based on the stiffness of each individual element [1]:

$$x_{static\_loop} = \frac{F}{k_{static\_loop}} = \underbrace{\frac{1}{k_{s1}} + \frac{1}{k_{s2}} + \ldots + \frac{1}{k_{sn}}}_{\textit{connected in series}} + \underbrace{\frac{1}{k_{p1} + k_{p2} + \ldots + k_{pn}}}_{\textit{connected in parallel}} \tag{2.1}$$

Typically, a well designed machine-tool-workpiece system may have a static loop stiffness of around 50N/µm; a figure of 500 N/µm is well desired for heavy cutting machine tools in particular. While a loop stiffness of about 10N/µm seems not rigid enough, it is quite common in precision machines. Static loop stiffness can be predicted at the early design stage by analytical or numerical methods and thus design optimization and improvement are essential; also, a continuous process because of the increasing demands from the various applications.

2.2.3 Dynamic Loop Stiffness and Deformation

Apart from the static loads, machine tools are subjected to constantly changing dynamic forces and the machine tool structure will deform according to the amplitude and frequency of the dynamic excitation loads, which is termed dynamic stiffness. Dynamic stiffness of the system can be measured using an excitation load with a frequency equal to the damped natural frequency of the structure.

Equations 2.2-2.5 provide a rough approximation of dynamic stiffness k_{dyn} and deformation x_{dyn}:

$$x_{dyn} = \frac{\tilde{F}}{k_{dyn}} \tag{2.2}$$

$$k_{dyn} = \frac{k_{static}}{Q} \tag{2.3}$$

where $\tilde{F}$ is the dynamic load applied to the machine tool, k_{static} is the static stiffness of the machine tool, and Q is the amplification factor which can be calculated from:

$$Q = \frac{1}{2\zeta} = \frac{1}{2\dfrac{c}{2M\omega_0}} = \frac{M\omega_0}{c} \tag{2.4}$$

where M and c is the mass and damping:

$\omega_0 = \sqrt{\dfrac{k_{static}}{M}}$ is the natural frequency

$\zeta = \dfrac{c}{2M\omega_0}$ is the damping ratio

Therefore,

$$x_{dyn} = \frac{\tilde{F}}{k_{dyn}} = \tilde{F}\frac{1}{c\omega_0} = \tilde{F}\frac{1}{c}\sqrt{\frac{M}{k_{static}}} \tag{2.5}$$

In order to accurately predict and calculate dynamic loop stiffness or the behaviour of a whole machine-tool system, a dynamic model including all elements in the machine-tool loop needs to be developed. The finite element method has been widely used to establish the machine tool dynamics model and provide the solution with reasonable accuracy, but it would take more computational time because of the complexity of the machine tool system. On the other hand, some alternative analysis techniques to predict dynamics of machines have been proposed. For example, Zhang *et al.* proposed a receptance synthesis method-based approach to predict the dynamic behaviours of the whole machine-tool system [2], although the approach has the limitation of modelling accuracy.

2.3 Vibrations in the Machine-tool System

Vibrations in the machine-tool system are a well-known fact in causing a number of machining problems, including tool wear, tool breakage, machine spindle bearings wear and failure, poor surface finish, inferior product quality and higher energy consumption.

Vibrations can be classified in a number of ways according to a number of possible factors. For instance, vibrations can be classified as free vibrations, forced vibrations and self-excited vibrations based on external energy sources. It is useful to identify vibrations types in machine tools. The basic principles of the three vibrations above can be found in most textbooks in the subject area [3-4], but the contents discussed below are a formulation in the context of machine tools and provide fundamental concepts for the following up chapters.

2.3.1 Free Vibrations in the Machine-tool System

If an external energy source is applied to initiate vibrations and then removed, the resulting vibrations are free vibrations. In the absence of non-conservative forces, free vibrations sustain themselves and are periodic.

The vibrations of machine tools under pulsating excitations can be regarded as free vibrations. The origins of pulsating excitations in machine tools include:

- Cutter-contact forces when milling or flying cutting
- Inertia forces of reciprocating motion parts
- Vibrations transmitting from foundations
- Imperfects of materials

For instance, taking a single-point diamond turning a part as an example, the part has some material defects such as cavities, as shown in Figure 2.3a. If the cutting tool is taken as the object to be investigated, it can be simplified as a single DOF mass-spring free vibration system as shown in Figure 2.3b, although this is an idealized model and the real system is far more complicated.

Firstly, consider the case of an undamped free vibration system. The general form of the differential equation for undamped free vibrations is:

$$M\ddot{x} + Kx = 0 \tag{2.6}$$

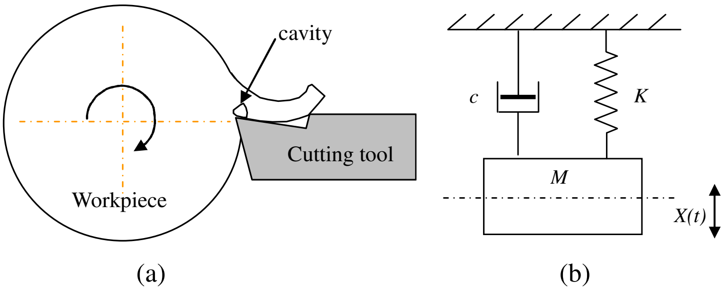

Figure 2.3. a Turning process with material defects **b** Single DOF free vibration system

Where M and K are the mass and stiffness which are determined during the derivation of the differential equation. Equation 2.6 is subject to the following initial conditions of the form:

$$x(0) = x_0$$
$$\dot{x}(0) = \dot{x}_0$$

The solution of Equation 2.6 is:

$$x(t) = x_0 \cos \omega_n t + \frac{\dot{x}_0}{\omega_n} \sin \omega_n t \tag{2.7}$$

where x is displacement at time t:

x_0 is the initial displacement of the mass

$\omega_n = \sqrt{\dfrac{K}{M}}$ is the undamped natural frequency

There is a slight increase in system complexity while a damping element is introduced to the spring-mass system. Here only viscous damping is taken into account. The general form of the differential equation for the displacement of damped free vibrations becomes:

$$M\ddot{x} + c\dot{x} + Kx = 0 \tag{2.8}$$

where c is the damping of the system. Dividing Equation 2.8 by M gives:

$$\ddot{x} + \frac{c}{M}\dot{x} + \frac{K}{M}x = 0 \tag{2.9}$$

The general solution of Equation 2.9 is obtained by assuming:

$$x(t) = Be^{\alpha t} \tag{2.10}$$

The substitution of Equation 2.10 into Equation 2.9 gives the following quadratic equation for α:

$$\alpha^2 + \frac{c}{M}\alpha + \frac{K}{M} = 0 \tag{2.11}$$

The quadratic formula is used to obtain the roots of Equation 2.11:

$$\alpha_{1,2} = -\frac{c}{2M} \pm \sqrt{\left(\frac{c}{2M}\right)^2 - \frac{K}{M}} \tag{2.12}$$

The mathematical form of the solution of Equation 2.9 and the physical behaviour of the system depend on the sign of the discriminant of Equation 2.12. The case when the discriminant is zero is a special case and occurs only for a certain combination of parameters. When this occurs the system is to be critically damped. For fixed values of K and M, the value of c which causes critical damping is called the critical damping coefficient, c_c:

$$c_c = 2\sqrt{KM} \tag{2.13}$$

The non-dimensional damping ratio, ζ, is defined as the ratio of the actual value of c, to the critical damping coefficient:

$$\zeta = \frac{c}{c_c} = \frac{c}{2\sqrt{KM}} \tag{2.14}$$

The damping ratio is an inherent property of the system parameters. Using Equations 2.13 and 2.14, Equation 2.12 is rewritten in terms of ζ and ω_n as:

$$\alpha_{1,2} = -\zeta\omega_n \pm \omega_n\sqrt{\zeta^2 - 1} \tag{2.15}$$

Therefore, the general solution of Equation 2.9 is:

$$x(t) = e^{-\zeta\omega_n t}(C_1 e^{\omega_n \sqrt{\zeta^2 - 1}t} + C_2 e^{-\omega_n \sqrt{\zeta^2 - 1}t}) \tag{2.16}$$

where C_1 and C_2 are the arbitary constants of integration. From Equation 2.16, it is evident that the nature of the motion depends on the value of ζ; Equation 2.9 then becomes:

$$\ddot{x} + 2\zeta\omega_n\dot{x} + \omega_n^2 x = 0 \tag{2.17}$$

This is the standard form of the differential equation governing the free vibrations with damping.

There are different conditions of damping: critical, overdamping, and underdamping. Detailed discussions of these three cases can be found in most of the subject textbooks [3, 4].

2.3.2 Forced Vibrations

If vibrations occur during the presence of an external energy source, the vibrations are called forced vibrations. The behaviour of a system undergoing forced vibrations is dependent on the type of external excitation. There are a few types of external forces including harmonic, periodic but not harmonic, step, impulse and arbitrary force, etc. If the excitation is periodic, the forced vibrations of a linear system are also periodic.

Considering the internal grinding process as shown in Figure 2.4a in which the spindle is out of balance, the resulted unbalance force is assumed in a harmonic form, $F\sin(\omega t+\varphi)$. This force will vibrate the grinder relative to the workpiece and result in forced vibrations.

Again, an undamped mass-spring system under harmonic forces is considered as shown in Figure 2.4b. The differential equation for undamped forced vibrations subjected to an excitation of harmonic force is:

$$\ddot{x} + \omega_n^2 x = \frac{F}{M}\sin(\omega t + \varphi) \tag{2.18}$$

If excitation frequency ω is not equal to ω_n the following equation is used to obtain the particular solution of Equation 2.18:

$$x_p(t) = \frac{F}{M(\omega_n^2 - \omega^2)}\sin(\omega t + \varphi) \tag{2.19}$$

The homogeneous solution is added to the particular solution with the initial conditions applied, yielding:

$$x(t)=\left[x_0-\frac{F\sin\varphi}{M(\omega_n^2-\omega^2)}\right]\cos(\omega_n t)$$
$$+\frac{1}{\omega_n}\left[\dot{x}_0-\frac{F\omega\cos\varphi}{M(\omega_n^2-\omega^2)}\right]\sin(\omega_n t)+\frac{F}{M(\omega_n^2-\omega^2)}\sin(\omega t+\varphi) \tag{2.20}$$

In a damped forced vibration system with harmonic excitation the standard form of the differential equation is:

$$\ddot{x}+2\zeta\omega_n\dot{x}+\omega_n^2 x=\frac{F}{M}\sin(\omega t+\varphi) \tag{2.21}$$

The particular solution of Equation 2.21 is:

$$x_p(t)=\frac{F}{M[(\omega_n^2-\omega^2)+(2\zeta\omega\omega_n)^2]}[-2\zeta\omega\omega_n\cos(\omega t+\varphi)$$
$$+(\omega_n^2-\omega^2)\sin(\omega t+\varphi)] \tag{2.22}$$

Equation 2.22 can be rewritten in the following alternative form:

$$x_p(t)=A\sin(\omega t+\varphi-\phi) \tag{2.23}$$

where
$$A=\frac{F}{M\sqrt{(\omega_n^2-\omega^2)+(2\zeta\omega\omega_n)^2}}$$

$$\phi=\tan^{-1}\left(\frac{2\zeta\omega\omega_n}{\omega_n^2-\omega^2}\right)$$

A is the amplitude of the forced response and ϕ is the phase angle between the response and the excitation.

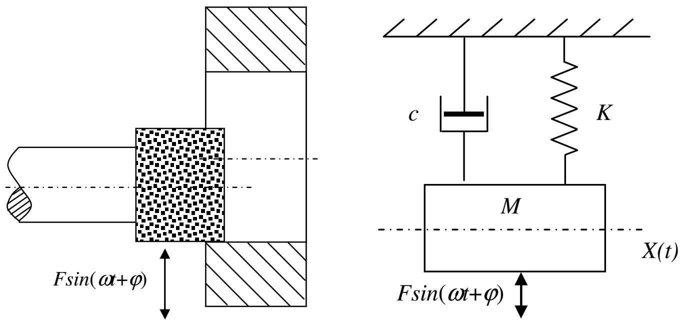

Figure 2.4. a Internal grinding process **b** Single DOF forced vibration system

Forced vibrations in machine tools can be generated from two kinds of energy sources, which are internal and external vibration sources. External vibration sources, such as seismic waves, usually transfer vibrations to the machine tool structure via the machine base. The design and use of effective vibration isolators will be able to eliminate or minimize forced vibrations caused by external vibration sources. There are many internal vibration sources which cause forced vibrations. For instance, an unbalanced high speed spindle, an impact force in machining processes, and inertia force caused by a reciprocal motion component such as slideways, etc.

2.4 Chatter Occurring in the Machine Tool System

2.4.1 Definition

Apart from free and forced vibrations, self-excited vibrations exist commonly in machine-tool system. A self-excited vibration is a kind of vibration in which the vibration resource lies inside the system. In machining self-excited vibrations usually result in machine tool chatter vibration. It should be noted that chatter vibration can also be caused by the forced vibration, but it is usually not a major problem in machining because the external force or the dynamic compliance of the machine structure can be reduced to reasonable levels when the external force causing the chatter is identified [5].

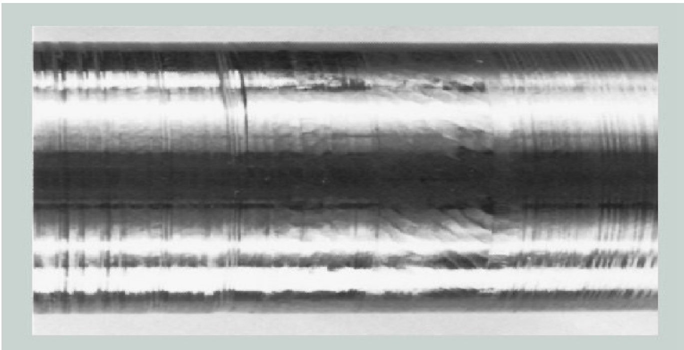

Figure 2.5. Poorly machined surface resulted from chatter (Courtesy: GE Company)

Chatter occurs mainly because one of the structural modes of the machine tool-workpiece system is initially excited by cutting forces. Chatter is a problem of instability in the machining process, characterized by unwanted excessive vibration between the tool and the workpiece, loud noise, and consequently a poor quality of surface finish. It also has a deteriorating effect on the machine and tool life, and the reliability and safety of machining operation [6]. The problem has affected the manufacturing community for quite some time and it is a popular topic for academic and industrial research. Therefore, it is very important to identify and to get a better understanding of the machine structural dynamic performance at both the machine design and production stage. Figure 2.5 shows a poorly machined surface resulting from chatters, and more information about chatters is available in Chapters 3 and 4 of this book.

2.4.2 Types of Chatters

There are mainly three forms of self-excited chatters. The first one is the velocity dependent chatter or Arnold-type chatter, named after the man who discovered it, which is due to a dependence on the variation of force with the cutting speed. The second form is known as the regenerative chatter, which occurs when the unevenness of the surface being cut is due to consequent variations in the cutting force when on the previous occasion the tool passed over that location, causing detrimental degeneration of the cutting force. Depending on the phase shift between the two successive wave surfaces, the maximum chip thickness may exponentially grow while oscillating at a chatter frequency that is close to but not equal to the dominant structural mode in the system. The growing vibrations increase the cutting forces and produce a poor and wavy surface finish [7]. The third form of chatter is due to mode coupling when forces acting in one direction on a machine-tool structure cause movements in another direction and vice versa. This results in simultaneous vibrations in two coupling directions. Physically it is caused by a number of sources, such as friction on the rake and clearance surfaces [8] and mathematically described by Wiercigroch [6].

Most of the chatters occurring in practical machining operations are regenerative chatter [9], although other chatters are also common in some cases. These forms of chatters are interdependent and can generate different types of chatter simultaneously. However, there is not a unified model capable of explaining all chatter phenomena observed in machining practice [10].

2.4.3 The Suppression of Chatters

After identifying chatters occurring in the machine-tool system, a number of approaches for reducing chatters have been proposed. Classical approaches usually use the stability diagrams to avoid the occurrence of chatters [9, 11-12]. The following approach formulates some general methods for the reduction of chatters both on the design and the production stage:

- Selecting the optimal cutting parameters
- Selecting the optimal tooling geometry
- Increasing the stiffness and damping of the machine tool system
- Using the vibration isolator as necessary
- Altering the cutting speed during the machining process
- Using a different coolant

More recently, modern control and on-line chatter detection techniques were applied to suppress chatters [13, 14, 15, 16]. Furthermore, a change of tool geometry is also an industrial feasible approach to chatter control [17], for instance, through the application of cutting tools with irregular spacing or variable pitch cutters [18].

2.5 Machining Instability and Control

2.5.1 The Conception of Machining Instability

In the previous sections, many aspects of self-excited machine tool vibrations or chatters have been briefly discussed. In practice, however, many problems of poor work surface finish are due to forced vibrations and the methods of reducing forced vibrations should thus well be understood. Forced vibrations are usually caused by an out-of-balance force associated with a component integrated with, or external to, the machine tool, whereas a self-excited vibration is spontaneous and increases rapidly from a low vibratory amplitude to a large one; the forced vibration results in an oscillation of constant amplitude. An exploration into chatter vibrations enables a better understanding of machining instability in practice.

From the machining point of view, with the designed machining conditions, a desired surface finish will be produced under a stable machining process. But as a complicated dynamic system, various mechanisms inherent in the machining process may lead the innately stable machining system to work at a dynamically unstable status which invariably results in unsatisfactory workpiece surface quality [19]. The machining instability coined here is a new generalized concept, which includes all phenomena making the machining process departure from what it should be. For instance, a variety of disturbances affect the machining system such as self-excited vibration [20], thermomechanical oscillations in material flow [21], and feed drive hysteresis [10], but the most important is self-excited vibrations resulting from the dynamic instability of the overall machine-tool/machining-process system [22-23]. However, sometimes the machining process is carried out with a relative vibration between the workpiece and the cutting tool, especially in heavy cutting and rough machining, in order to obtain high material removal rates. The relative vibration is not necessarily a sign of the machining instability for the designed machining conditions and prescribed surface finish. In another extreme case, such as in ultra-precision machining or micro/nano machining, the relative vibration between the workpiece and the cutting tool is too small to be measured, but the machining is sensitive to environmental disturbances. The surface generated may be unsatisfactory because of the disturbance, even though the machining system itself operates in the stable state. Therefore, the machining instability is related to the level of the surface quality required and the designed machining conditions.

Table 2.1 The classification of machining instability [25]

Machining Instability							
	Chatter vibrations			Random and free vibrations			Forced Vibration
	Regenerative (Dominate)	Frictional	Mode coupling	Tool dependent	Workpiece dependent	Environment dependent	Machine tool component dependent
Location	Between cutting edge and workpiece	Tool flank-workpiece; chip-tool rake face	In cutting and thrust force directions	Tool flank-workpiece; chip - rake face	Cutting zone	Whole cutting process	Whole cutting process
Causes	Overlapping cut	Rubbing on the flank face and the rake face	Friction on the rake and clearance faces; chip thickness variation, shear angle oscillation.	Tool wear and breakage; BUE, etc.	Material softening and hardening; hard grain and other kinds of flaws	Environmental disturbances	Off-balance of moving components, such as the spindle
Features	Self–excited vibration; left a wavy surface on workpiece	Self-excited vibration; amplitude depends on the system damping	Mode coupling vibration; Simultaneous vibration in two directions	Random and chaotic; depends on cutting conditions	Random and chaotic; depends on material property and its heat treatment	Radom and chaotic; depends on work environment	Forced vibration
Suppression method	Select proper depth of cut and spindle speed according to regenerative stability chart	Select proper clearance and rake angles	Change the tool path; Select proper cutting variables	Select high quality tool materials and proper cutting parameters	Select proper cutting tool and cutting parameters	If needed, isolate the machine tool	Well balance moving component in machine tools

2.5.2 The Classification of Machining Instability

Based on the conception above, Cheng *et al.* summarize all kinds of machining instability and their features as listed in Table 2.1 [24-25]. The instability is classified as the chatter vibration, the random or free vibration and forced vibration. The random or free vibration usually includes any shock or impulsive loading on the machine tool. A typical random vibration is the tool vibration, for instance, when the tool strikes at a hard spot during the cutting process. The tool will bounce or vibrate relative to the workpiece, which is the beginning of the phenomenon of a self-excited vibration. The initial vibration instigated by the hard spot is heavily influenced by the dynamic characteristics of the machine tool structure which must be included in any rational chatter analysis.

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