Vibration Analysis of Lathe Machine Tool

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Abstract
Today, machine tools play a vital role in the production of parts in any manufacturing industry. The dimensional accuracy and surface finish of the work piece depends mainly on the condition of the machine. Vibration occurring on machine tools has been being a serious problem for engineers for many decades. Undesired relative vibrations between the tool and the work-piece jeopardize the quality of the machine surfaces during cutting.
Many Condition Monitoring Techniques are available to monitor the machine tool experimentally. Among these techniques, vibration monitoring is the most widely used technique because most of the failures in the machine tool could be due to increased vibration level.
In the present study, the vibration analysis of a lathe machine component has been investigated. The governing equation of motion of a lathe machine component is formulated using modal analysis approach. Given lathe machine tool is discretised into equivalent six lumped mass system which is having six degrees of freedom and the equivalent model is considered for the development of equation of motion of the machine tool. The natural frequencies and respective mode shapes are estimated using modal analysis. Furthermore, the displacement at each lumped mass is evaluated to investigate the transmission of steady state response to the machine tool. The precautionary measures are also suggested to reduce the transmission of the steady state response.
Keywords- Modal analysis of lathe, vibration monitoring of machine tool.

Introduction

The performance characteristics of machine tools depend on the dynamic properties of their structure. All operating machines, having rotary and/or reciprocating parts give rise to vibration. Machine tools are liable to deterioration in their performance level with respect to time due to various causes such as wear and tear, ageing, unbalance, looseness of parts etc., and produce a corresponding increase of the vibration level. Machine tool vibration, if uncontrolled, can adversely affect the surface finish, dimensional accuracy and tool life. About 70% of the failures in the machine tool could be due to increased vibrationlevel of the machine.

Simple Lathe machine Tool

The major causes of vibration known to us are as follows:

1. Axial deformation and / or radial deflection of spindle shaft under the action of cutting forces.
2. During machining process under the influence of cutting forces and the drive system dynamics, the machine tools components such as the guide and bed / column, carriage, tool post and tool shank, etc. are elastically deformed and deflected.
3. The jig-fixture may also be subjected to the similar type of deformations. other causes.

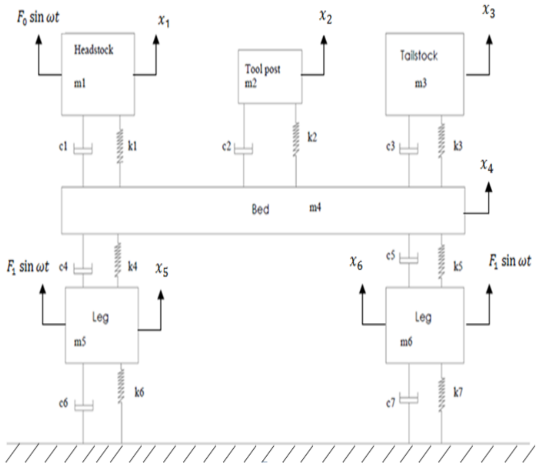
An unbalance forces in a lathe will induce more vibration, which result in deterioration of the dimensional accuracy and surface finish of the work piece. Different methodologies are presented for vibration analysis of a machine tool. Modal Analysis has become one of the most powerful and popular tool for dynamic analysis of machine tools.

Problem statement -Dynamic analysis of lathe machine tool.

Solution methodology-To solve given problem **Modal Analysis** is used and response at each point mass is plotted by using **MATLAB 7.0**

Modal Analysis

When external forces act on multi degree freedom system, the system undergoes forced vibrations for a system with n coordinates or degrees of freedom, the governing equation of motion are a set of n coupled ordinary D.E. of 2nd order .The solution of these equation becomes more complex when the degrees of freedom of system is large and /or when the forcing functions are non periodic in such a cases more convenient method known as modal analysis is used to solve the problem. In this method the expansion theorem is used and the displacements of the masses are expressed as a linear combination of normal mode of a system. These linear equations of motion so that we can obtain the set of n uncoupled differential equation of 2nd order. The solution of these equations which is equivalent to the solution of the equations of n single degree of freedom system can be readily obtained



Discretised Lathe Model

Equations Of Motions

General equations Equation of motion for single degree of freedom system is written as,

$$m\ddot{x} + c\dot{x} + kx = f(t)$$

For multi degrees of freedom system, the equations of motions is expressed in matrix form as,

$$[m]\{\ddot{x}\} + [c]\{\dot{x}\} + [k]\{x\} = \{f(t)\}$$

given system can be constructed in matrix form as

$$\begin{bmatrix} m_1 & 0 & 0 & 0 & 0 & 0 \\ 0 & m_2 & 0 & 0 & 0 & 0 \\ 0 & 0 & m_3 & 0 & 0 & 0 \\ 0 & 0 & 0 & m_4 & 0 & 0 \\ 0 & 0 & 0 & 0 & m_5 & 0 \\ 0 & 0 & 0 & 0 & 0 & m_6 \end{bmatrix} \begin{Bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \\ \ddot{x}_3 \\ \ddot{x}_4 \\ \ddot{x}_5 \\ \ddot{x}_6 \end{Bmatrix} +$$

$$\begin{bmatrix} c1 & 0 & 0 & -c1 & 0 & 0 \\ 0 & c2 & 0 & -c2 & 0 & 0 \\ 0 & 0 & c3 & -c3 & 0 & 0 \\ -c1 & -c2 & -c3 & c1+c2+c3+c4+c5 & -c4 & -c5 \\ 0 & 0 & 0 & -c4 & c4+c6 & 0 \\ 0 & 0 & 0 & -c5 & 0 & c5+c7 \end{bmatrix} \begin{Bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \\ \dot{x}_5 \\ \dot{x}_6 \end{Bmatrix} +$$

$$\begin{bmatrix} k1 & 0 & 0 & -k1 & 0 & 0 \\ 0 & k2 & 0 & -k2 & 0 & 0 \\ 0 & 0 & k3 & -k3 & 0 & 0 \\ -k1 & -k2 & -k3 & k1+k2+k3+k4+k5 & -k4 & -k5 \\ 0 & 0 & 0 & -k4 & k4+k6 & 0 \\ 0 & 0 & 0 & -k5 & 0 & k5+k7 \end{bmatrix} \begin{Bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \\ x_5 \\ x_6 \end{Bmatrix} = \begin{Bmatrix} F_0 \sin \omega t \\ 0 \\ 0 \\ 0 \\ F_1 \sin \omega t \\ F_1 \sin \omega t \end{Bmatrix}$$

By referring standard lathe machine specifications,

m_1=50 kg, m_2=25 kg, m_3=25 kg, m_4=100 kg, $m_5 = m_6 = 200$

k_1=2x10^5 N/m, k_2=1x10^5, k_3=1x10^5N/m, k_4= k_5=5 x10^5N/m, k_6=k_7 =3x10^5

Mass and stiffness matrix are given as,

$$M=\begin{bmatrix} 50 & 0 & 0 & 0 & 0 & 0 \\ 0 & 25 & 0 & 0 & 0 & 0 \\ 0 & 0 & 25 & 0 & 0 & 0 \\ 0 & 0 & 0 & 100 & 0 & 0 \\ 0 & 0 & 0 & 0 & 200 & 0 \\ 0 & 0 & 0 & 0 & 0 & 200 \end{bmatrix}$$

$$[K]=\begin{bmatrix} 2 & 0 & 0 & -2 & 0 & 0 \\ 0 & 1 & 0 & -1 & 0 & 0 \\ 0 & 0 & 1 & -1 & 0 & 0 \\ -2 & -1 & -1 & 20 & -5 & -5 \\ 0 & 0 & 0 & -5 & 8 & 0 \\ 0 & 0 & 0 & -5 & 0 & 8 \end{bmatrix} \times 10^5$$

From MATLAB Eigen values i.e. natural frequency and Eigen vectors i.e. modal matrix can be found as

[v,d]=eig (k, m)

$$v=[X]=\begin{bmatrix} -0.589 & .03080 & .09020 & -0.0695 & -0.04670 & 0.02070 \\ -0.05890 & 0.1100 & -.1407 & -.0567 & -.0313 & 0.0207 \\ -0.0589 & -.1604 & -0.0620 & -.0567 & -.05676 & .0207 \\ -.0331 & 00 & 0 & 00 & 0 & -.0944 \\ -.0368 & -.0051 & .0019 & -0.0040 & 0.586 & 0.01290 \\ -0.368 & 0.0029 & 0.0026 & 0.545 & -.0221 & 0.0129 \end{bmatrix}$$ …………..Modal matrix

$$d=\omega_n=\begin{Bmatrix} \omega_1 \\ \omega_2 \\ \omega_3 \\ \omega_4 \\ \omega_5 \\ \omega_6 \end{Bmatrix}=\begin{Bmatrix} 41.8694 \\ 63.2456 \\ 63.2456 \\ 63.2456 \\ 63.2456 \\ 149.1541 \end{Bmatrix}$$ ………….Natural Frequency

$$\overline{\boldsymbol{F}(\boldsymbol{t})}=\begin{Bmatrix} F_o \sin\omega t \\ 0 \\ 0 \\ 0 \\ F_1 \sin\omega t \\ F_1 \sin\omega t \end{Bmatrix}=\begin{Bmatrix} 50\sin\omega t \\ 0 \\ 0 \\ 0 \\ 20\sin\omega t \\ 20\sin\omega t \end{Bmatrix}$$

By considering single phase 60 Hz induction motor of 1750 rpm

Hence = $\omega = \frac{2\pi N}{60} = 183.26\ rad/s$

$$\overline{\boldsymbol{F}(\boldsymbol{t})}=\begin{Bmatrix} 50 \\ 0 \\ 0 \\ 0 \\ 20 \\ 20 \end{Bmatrix}\sin 183.26t$$

Damping coefficient ζ= 0.05

The equations of motion of multi degrees of freedom models given by Equation 2 generally have n coupled differential equations. To solve these n coupled equations for harmonic $\{F(t)\}$, they must be uncoupled by using modal transformation. The uncoupled equations of motions in modal coordinates are expressed as

$\ddot{q}_i(t)+2\xi\omega_i\dot{q}_i(t)+\omega_i{}^2 q_i(t) = Q_i(t)$, i=1, 2, 3……..n

where $\bar{Q}$(t) are associated generalized forces which are expressed in vector form as

$$\bar{Q}(t)=[X]^{T}\bar{F}(t)$$

$$\bar{Q}(t)=\begin{Bmatrix} Q_{10} \\ Q_{20} \\ Q_{30} \\ Q_{40} \\ Q_{50} \\ Q_{60} \end{Bmatrix}=\begin{Bmatrix} 4.4209 \\ -5.2983 \\ -.04784 \\ 1.8270 \\ 0.6403 \\ 1.5514 \end{Bmatrix}\sin 183.26t$$

The steady-state solution of eqn (3) can be written as,

$$q_i(t)=q_{i0}\cos(\omega_b t-\emptyset_i), \quad i=1, 2, 3\ldots.n$$

where,

$$q_{i0}=\frac{Q_{io}}{\omega_i{}^2}\times\frac{1}{\sqrt{\left[\left\{1-\left(\frac{\omega_b{}^2}{\omega_i{}^2}\right)\right\}^2+\left(2\xi\frac{\omega_b}{\omega_i}\right)^2\right]}} \text{ and } \emptyset_i=\tan^{-1}\left\{\frac{2\xi_i\frac{\omega_b}{\omega_i}}{1-\left(\frac{\omega_b}{\omega_i}\right)^2}\right\}$$

$$\begin{Bmatrix} q_{10} \\ q_{20} \\ q_{30} \\ q_{40} \\ q_{50} \\ q_{60} \end{Bmatrix}=\begin{Bmatrix} 1.385\text{x}10^{-4} \\ -1.788\text{x}10^{-4} \\ -1.6157\text{x}10^{-5} \\ 6.1707\text{x}10^{-5} \\ 2.1626\text{x}10^{-5} \\ 1.3302\text{x}10^{-4} \end{Bmatrix} \text{ and } \begin{matrix} \emptyset_1 = -1.412 \\ \emptyset_2 = -2.243 \\ \emptyset_3 = -2.243 \\ \emptyset_4 = -2.243 \\ \emptyset_5 = -2.243 \\ \emptyset_6 = -13.555 \end{matrix}$$

Principle co-ordinates in modal analysis

$q_1(t) = 1.385\text{x}10^{-4}\cos(183.26t+1.412)$

$q_2(t) = -1.788\text{x}10^{-4}\cos(183.26t+2.243)$

$q_3(t) = -1.6157\text{x}10^{-5}\cos(183.26t+2.243)$

$q_4(t) = 6.1707\text{x}10^{-5}\cos(183.26t+2.243)$

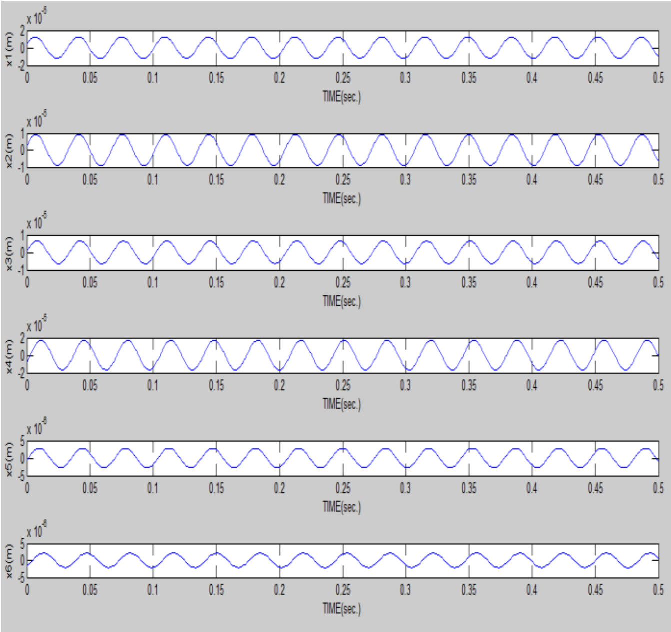
$q_5(t) = 2.1626\text{x}10^{-5}\cos(183.26t+2.243)$

$q_6(t) = 1.3302\text{x}10^{-4}\cos(183.26t+13.555)$

By expressing solution vector $\vec{x}$ as a linear solution of natural modes

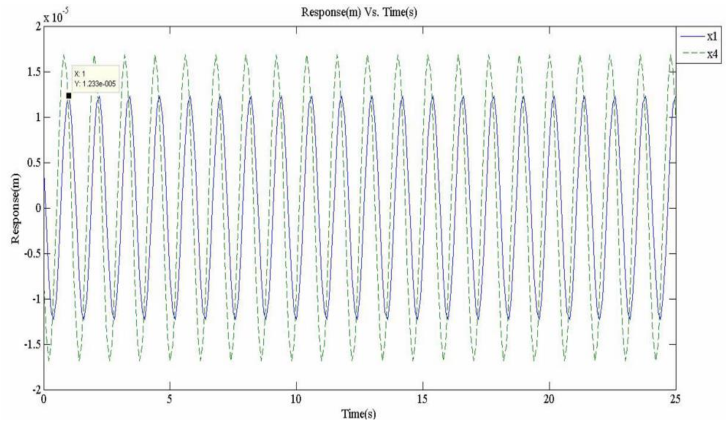
{ x(t)}=[X]*q{t}

Above equation represents 6 response equations in this system .These responses are directly plotted vs. time in MATLAB.



Graph of Response vs. time

From the above response curves it can be seen that m_1 and m_4 are subjected to more vibrations. Graph of x_1 and x_4 vs. time is exploded as follows



Results and conclusions

From the above graphs it is concluded that among all considered masses Head stock and Bed is subjected to maximum vibrations. As motor here is considered as source of vibrations in lathe machine, directly connected to head stock, is also subjected to vibrations with maximum amplitude. Also the vibrations created at the interface of tool and work piece will be transferred to head stock. The other forcing function mentioned at the legs of lathe represents the base excitation. This base excitation here can be considered as the vibrations of floor or base due to other machine vibrations. The bed is subjected to vibrations due to base excitation. Also the vibrations in headstock are transferred to bed .Hence it can be seen that bed vibrates with maximum amplitude.

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