Monitoring The Machine Elements In Lathe Using Vibration Signals

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ABSTRACT:
In any manufacturing industry, machine tools play an important role in the production of parts. The dimensional accuracy and surface finish of the workpiece depends mainly on the condition of the machine. To analyse the condition of the existing machine tool numerically finite element method has been used. Here, an attempt has been made by modeling the lathe using finite element software to find out the response of the structure for defective spindle bearing and unbalance forces. 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. Experiments were carried out on a lathe using the Condition Monitoring instrument to measure vibration severity for different spindle speed. From the experimental and numerical analysis, it was found that the vibration velocity increases as the spindle speed and depth of the defect increases. Also the value of vibration velocity at front bearing due to unbalance forces was determined and it was compared with the experimental value.

INTRODUCTION:
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 produces 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 vibration level of the machine.

Lathe is one of the most important machine tool in manufacturing industries. The quality of the workpiece depends mainly on the condition of the lathe. A bearing is the most common critical component in a Lathe. Proper performance and functioning of bearings has always been a major concern in rotating machinery, since all the forces are transmitted to the bearings. It has been well established that 80% of the bearings failed to
attain their expected life. A defect in the spindle bearing and unbalance forces in a lathe will induce more vibration, which result in deterioration of the dimensional accuracy and surface finish of the workpiece. Finite Element Analysis has become one of the most powerful and popular tool for structural analysis of machine tools. Finite Element Modeling allows accurate modeling of the actual structure of the lathe through the use of variety of beam, plate and solid elements to determine the static and dynamic characteristics of the structure under these conditions. By choosing proper shapes, size and number of elements, the Finite Element Model of the Lathe structure can be made very close to the actual one. Finite Element Software provides the necessary tools to perform modeling as well as analysis.

The extraction of vibratory signatures can be a valuable diagnostic tool to predict impending failures of the rolling element bearing. A lot of research was carried out and is continuing in the field of condition monitoring of machine tools using finite element method. W.R. Wang and C.N. Chang [1] carried out simulation of a spindle-bearing system with a finite element model and then compared it with the experimental results. Radial and tilting springs and dashpots were considered in angular contact spindle ball bearings. This Finite Element model shows that additional tilting characteristics make significant effect on higher-order vibration modes. Braun S. and B. Datner [2] described a vibration based diagnostic method aimed at detecting localized defects developing in roller bearings. The feasibility of the technique has been tested for a roller bearing with artificially induced defects. Prashad H. et. al.[3] have discussed about the vibration signature of the bearings at different speeds of operation. K. Ono and Y. Okada [4] carried out an analysis of ball bearing vibrations caused by outer race waviness. Bearing vibration could be caused by a number of factors, such as defects occurring on the racetrack or on the rolling elements. Analytical study was carried out to evaluate the effect of waviness number, radial gap and shaft imbalance on the bearing vibration. An experimental investigation was carried out to confirm the analytical study. Juhn-HorngChen et. al.[5] proposes a new method of identifying linearized characteristics of rolling element bearings based on Finite Element Formulation.

The present work deals with the Finite Element Method to study the effect of defective spindle bearing and unbalance forces on the vibration characteristics of the lathe structure by performing eigen value, frequency response and transient dynamic analysis. Experiments were carried out to determine the vibration velocity level on the headstock both at the front and rear bearing. Experimental results were compared with the theoretical results.

FINITE ELEMENT MODELING & ANALYSIS PROCEDURE:

The Finite Element Modeling is described as the representation of the geometric model of lathe in terms of finite number of elements and nodes which are the building blocks of the numerical representation of the model for solution. In addition to
information about element and nodes, this model also contains information about material properties and boundary conditions. Fig.1 shows the finite element beam model of a lathe. Lathe structure was divided into a number of elements so that the behavior of the elements can be studied under the action of external and internal forces transmitted from adjacent elements. The finite element model of a lathe was developed using a pre-processor NISA DISPLAY V10.5. The lathe model was made up of 3D beam elements, 3D spring elements and shell/plate elements. In a model, left leg, right leg, tailstock, bed, spindle chuck and headstock are modeled as 3-D beam element, which has a six degrees of freedom at each node. 3-D beam element is uniaxial elements with tension, compression, torsion and bending capabilities. Spindle Bearings are represented by 3-D general spring elements. Tool post, Carriage and feed rod were modeled using 3-D solid elements, 3-D Shell elements and 3-D spring element respectively. Spindle, Bearings and Chuck were modeled as hollow circular beams whereas left leg, right leg gap bed were modeled as hollow rectangular beams. Complete finite element model was checked for duplicate nodes and duplicate elements for accurate results. The Finite Element Model of lathe has totally 125 elements with 118 nodes.

Fig.1 Finite Element Beam Model of a Lathe
After modeling, eigen value analysis was carried out to find the dynamic characteristics of a lathe structure in terms of its natural frequency and mode shapes. Mode shapes can be defined by the amplitude of displacements of all the mass points during the vibration of the structure at natural frequencies. Lumped mass formulation and conventional subspace iteration methods were employed for Eigen value extraction. The analysis was carried out in the absence of damping and load. Thirty modes and its natural frequencies were determined. The modal data and results were used for the computation of responses in the presence of dynamic loads.

The unbalanced centrifugal force due to spindle weight is calculated as follows. The weight of the spindle is 8.67 Kg.

Calculation of Angular Velocity [$\omega$]

<table>
<thead>
<tr>
<th>Speed of Spindle</th>
<th>= 1200 rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\omega$</td>
<td>= 1200/60 = 20 Hz</td>
</tr>
<tr>
<td>Therefore, $\omega$</td>
<td>= 20 x 2$\pi$ rad/sec</td>
</tr>
<tr>
<td></td>
<td>= 125.66 rad/sec</td>
</tr>
</tbody>
</table>

Calculation of Radius of Eccentricity [r]

According to Indian Standard 11723, the balanced quality grade for machine tool spindle is G6.3 mm/sec (vibration level) [6].

We know that, $V = \omega \times r$

Therefore, $r = \frac{V}{\omega} = \frac{6.3}{125.66}$

$= 0.05$ mm

Centrifugal force is given by $m \times r \times \omega^2$

Therefore, Spindle unbalance force

$= \frac{8.67}{9810} \times 0.05 \times 125.66^2 = 0.698$ Kgf

Similarly, unbalanced force due to pulley $= \frac{14}{9810} \times 0.05 \times 83.72 = 0.75$ Kgf

Unbalanced force due to chuck

$= \frac{10.096}{9810} \times 0.05 \times 125.62 = 0.811$ Kgf

Unbalanced force due to Work Piece $= \frac{5.1}{9810} \times 0.05 \times 125.62 = 0.412$ Kgf

The unbalanced force due to spindle runout were also considered. Head stock assembly consists of two sets of shafts including gears and bearings. The unbalanced force due to shaft 1 and shaft 2 are 0.38 Kgf and 0.51 Kgf respectively. This unbalanced force was applied on the front and rear side of the headstock i.e. at node 28 and 26 along X direction. The unbalanced force due to spindle, chuck and pulley are applied at node 88, 29 and 26 respectively. Frequency range of 0 – 150 Hz was considered. The analysis was carried out considering first ten modes and 3% damping. The response in terms of vibration velocities at node 28 and 26 along x and y direction were determined. Fig.2 shows the rolling element bearing with the assumption of the defect in the outer race along horizontal direction. Transient dynamic analysis was carried out to determine the
response of structure subjected to time varying loads. The variation of loads can be represented in terms of amplitude versus time. A defect in the outer race of a rolling element bearing generates an impact every time a roller rolls over it. If the bearing rotates with constant speed, each defect generates a regular set of impacts with an impact repetition rate that depends upon bearing speed and location of the defect. The assumed defect was 0.1 mm depth and 3 mm length. The eigen value analysis must precede the transient dynamic analysis to extract natural frequencies and its mode shapes. The starting time and ending time for integration in the analysis were 0.0 sec and 0.05 sec respectively. The time taken by the bearing for one revolution is 0.05 sec at 1200 rpm. As each roller rolls over the defect, it produces a triangular pulse. The variation of loads can be represented in terms of amplitude versus time. A smaller time step of 6E-7 was chosen when the load was present, and a time step of 6E-5 was chosen when the load was absent. A concentrated nodal force (stiffness of the bearing) was applied on the front bearing. The transient dynamic analysis was carried out to determine the response of structure subjected to time varying loads by assuming the defect of 0.1, 0.2 and 0.3mm depth in horizontal direction as well as in vertical direction.

![Fig. 2 Rolling Element Bearing showing the defect in the outer race along horizontal direction](image)

**EXPERIMENTAL PROCEDURE**

The experiment was carried out on Enterprise 1330 lathe by selecting four measuring points i.e. horizontal and vertical measuring points on both front and rear spindle bearings. The instrument used to carry out the experiment is shock pulse tester T2000. The condition monitoring functions of the T2000 are based on two widely used measuring techniques, namely Shock Pulse Method for bearing monitoring and Vibration Velocity measurement according to ISO 2372. Machine class number was set to Class I since the power of the lathe motor is 2.25 K.W. According to ISO 2372 Machine Classification, production electrical motors of up to 15 K.W. are come under Class I. The
sensitivity of the transducer was set to 10.5 pc/m/s² [7]. The vibration velocity was measured by placing the vibration velocity transducer in vertical direction and horizontal direction both at the front and rear bearing housing. The experiments were repeated for different spindle speed.

RESULTS AND DISCUSSION

Eigen value analysis was carried out to obtain the natural frequencies and its mode shapes. The first five natural frequencies and its mode shapes were computed. Fig 3 shows the first eigen mode of a lathe structure and it was observed that resonance is occurring at the first natural frequency i.e. at the 86.6 Hz. The second, third, fourth and fifth natural frequencies are 91.7hz, 98.8hz, 129.7hz and 171.1hz respectively.

![Fig. 3 First Eigen mode of a lathe structure](image)

Fig. 4 shows the frequency response analysis of the vibration signals in terms of velocity. The frequency spectrum gives information about the vibration level caused by the unbalance forces during operation. From Fig. 4, it was found that the value of vibration velocity at 20 Hz is 0.2 mm/sec. The experimental vibration velocity along horizontal direction at the front bearing at 1200 rpm was found to be 1.4 mm/sec. The value of vibration velocity obtained from frequency response analysis considering unbalance force was lower compared with the experimental value because in actual practice, the various effects like vibration due to belt tension fluctuations, gear tooth
frequencies at the mating gears, defects in bearing and due to unbalance rotors were also included.

**Fig. 4 Frequency response analysis of a lathe considering unbalance forces**

The vibration pattern or signature which results from a single local defect in a bearing outer race were shown Fig.5 and Fig.6. The transient waveform depends on the structural response due to the impacts between rolling element and defect. The Fig.5 shows the vibration level at front bearing at 1200 rpm when the defect is 0.1 mm depth located in horizontal direction. Here, the value of vibration velocity at node 28 and 26 along X direction (horizontal direction) was more compared to the values along Y direction. The Fig.6 shows the vibration velocity level when the outer race defect is 0.1mm depth which was located in vertical direction. Here, the value of vibration velocity at node 28 and 26 along Y direction (vertical direction) was more compared to the values along X direction. Thus the response depends upon the location of the defect. If the location of the defect was along X direction, the vibration level along X is more compared to Y direction. The analysis were carried out for 0.2mm, 0.3mm depth and it was observed that as the depth of the defect increases, the impact force between rolling element and outer race also increases which in turn increases the vibration velocity level.
Fig. 5 Transient dynamic analysis due to 0.1mm depth (horizontal direction)

Fig. 6 Transient dynamic analysis due to 0.1mm depth (vertical direction)
From Fig.7 and Fig.8, it was clear that vibration velocity measured in horizontal direction is higher compared to vertical direction and also vibration velocity increases with the increase of spindle speed. Comparing Fig.7 and Fig.8, it was observed that vibration velocity level at front bearing is higher compared to rear bearing. Because front bearing is nearer to the source of vibration since more load is acting on the front bearing i.e. due to chuck, spindle, etc.

**Fig. 7 Vibration velocity at front bearing for different spindle speed**

**Fig. 8 Vibration velocity at rear bearing for different spindle speed**

**CONCLUSIONS**

Transient dynamic analysis showed that the vibration velocity level increases as the damage in the outer race increases. It was also found that the vibration velocity level depends upon the location of the defect. The value of vibration velocity obtained from frequency response analysis considering unbalance force is lower compared with the experimental value because in actual practice, the various effects like vibration due to belt tension fluctuations, gear tooth frequencies at the mating gears, defects in bearing
and due to unbalance rotors were also included. From the experimental analysis, it was found that vibration velocity increases with the increase of spindle speed. Hence the theoretical results are agreement with experimental results. The present work shows that the FEM is a valuable tool in finding sources of increased or undesirable vibrations from various defects. Thus Finite Element Analysis is not only helpful during the development of new structures but also helpful to analyze the existing lathe structure under given circumstances.

REFERENCES