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Vibration Analysis for Lathe Machine Gearbox Train

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Abstract:

A vibration analysis of the lathe machine condition monitoring is presented. The machine gear train was modelled in SolidWorks and modal analysis was used to determine the vibration characteristics of the lathe machine gearbox, by determining the mode shapes and natural frequencies with the intention of estimating the resonance frequency below which the machine should be operated to prevent its failure. The vibration data were analysed using ANSYS and the gear train parameters such as its stiffness, mass and damping parameters were obtained from the analyses. First six natural frequencies were obtained between the range of 0-800 Hz. The six mode shapes were obtained at various natural frequencies and maximum deformation of (432.91 Hz, 402.52 mm), (482.75 Hz, 428.98 mm), (507.83 Hz, 526.07 mm), (544.16 Hz, 445.71 mm), (671.98 Hz, 944.32 mm), (735.11 Hz, 856.58 mm). The modal analysis showed that the system natural frequency is greater in the 6th mode. The system's resonance frequency occurs at 126.01 dB at a natural frequency of 479.89 Hz. The resonance magnitude of the system limits the operating frequencies of the machine above which the system is considered not reliable for safe operation.

Keywords: Modelling, mode shape, harmonic, failure, frequency, lathe- machine

1. Introduction

The lathe machine is used extensively in metal removal process such as turning. The operation of the machine over a long period of time often required predictive maintenance procedure to avoid failure especially in the machine gearbox that could cost downtime of operation which could affect productivity. Several studies concerned with the vibration analysis of the machine gearbox had been presented (Aherwar and Khalid, 2012; Vijaykumar *et al* 2014; Zuber *et al* 2014; Chaari *et al* 2012; Zimroz *et al* 2010).

Predictive maintenance techniques is an effective strategy used for the reduction unexpected machinery failure and vibration monitoring is an essential predictive maintenance technology. Mathew and Afredson, 1984 presented a review on techniques of machine condition monitoring. The study discussed the monitoring of vibration signatures of several rolling element bearings with a view to detecting incipient failure. Vibration data were obtained and analyzed for several parameters so as to determine their effectiveness in the monitoring of bearing condition. The study established that the parameters associated with the system frequency were more consistent in the detection of damage when compared with the time domain parameters. Montalvão e Silva, 1990 presents a review upon which vibration analysis could be based and the procedure for such analysis data collection and interpretation. Vibration monitoring reveals the condition of the machine. Sutar *et al* 2018 established that every machinery problem produces specific pattern of spectrum and the problem could be identified using frequency and phase analysis. The study present cases of machine failures such as the balancing problems, issues of misalignment and Resonance.

Machine failure is the inability of a machine to function in the way it was intended as dictated by the design. The cost of running a machine could be reduced by reducing the downtime of the machine which usually arises from the machine failure and machinery often suffers from the effect of vibration of the rotating components (Djaidir *et al* 2017). Recent developments in the field of machine fault analysis using vibration analysis were discussed in Khadersab abd Shivakumar, 2018. The study identified the different types of faults detected using the vibration signature analysis with emphasize on the rolling element bearing fault signature analysis. Vibration analysis can help predict the failure of such components so that predictive maintenance can be carried out to avoid such failure.

This study focus on determining the reliability status of the general purpose lathe machine model AJL180-325VS gap bed obtained at the Mechanical Engineering department workshop of Lagos State University, Nigeria for purpose of research work. Ebersbach and Peng, 2008 adapted expert system for machine condition monitoring. The system interpreted the associated data due to the ability to identify systematic reasoning processes. The use of expert systems allows for flexible analysis which was easier to approach. The expert system incorporates algorithms for high accuracy fault detection. The study confirms the potential value of the expert system manufacturing and mineral processing plants.

Lathe machine play a vital role in the production of parts in any manufacturing industry. The dimensional accuracy and surface finish of the work piece could depend mainly on the condition of the machine. Vibration occurring on machine tools has been a serious problem for engineers for many decades. An undesired relative vibration between the tool and the work-piece reduces the quality of the machine surfaces during cutting (Nilesh, 2017). Vibration monitoring is the most widely used technique because most of the failures in the machine tool could be due to increased vibration level. The level and pattern of the vibration could expose the condition of the machine especially its rotating component. Lebold *et al* 2000 and Thompson and Fenger, 2001 discussed extensively the use of vibration analysis for fault detection in machines. The study developed a time dependent procedure which could be use for assessing faults in motors and the possible maximum time that such motor could be operated before it is scheduled for maintenance. This will tend to reduce the cost of maintenance of the motor. Zhao *et al* 2017 presents an investigation of the failure of a low-pressure steam turbine blade. The dynamical behaviour of the blade was analyzed using finite element model. The model was used to predict the blade resonances for specific speed range. The natural frequencies and the mode shapes of the blade at static condition were determined. It was found that the 2nd natural frequency of the blade is very close to the 9th rotor speed harmonic. Nabby et al 2018 discusses the use of vibration analysis for machine health monitoring. The analysis was based on Impact demodulation technique by exchanging severity faulty bearing within planned short down. It was concluded that the machine availability and health can be achieved by reducing vibration impact through continuous condition monitoring. It is important for a designer to understand the nature of the frequencies of a machine to ensure that these frequencies do not reach the excitation frequencies in order to avoid failure of the machine.

2. Modeling and Analysis

The lathe machine used for this study was the general purpose AJL180-325VS Gap Bed Lathes machine. The lathe machine has a 300 mm turning length capacity and is installed with 250 mm chuck diameter. The machine is available in the Mechanical Engineering Department of Lagos State University, Nigeria. A 3-dimensional (3D) model of the lathe machine gear train assembly is carried out using the SolidWorks software version 2014. The 3D model is exported to the ANSYS software for the vibration analysis which includes the modal and harmonic analysis. The natural frequencies of the system at varying speeds were obtained from the modal analysis of the gear train, and harmonic analysis was used to quantify the amplitude of the vibration. The ANSYS uses the principle of the Finite Element Analysis. The vibration of the lathe machine gear train was characterised and assessed by the amplitude and frequency of the gear train assembly. The parameters associated with the vibration analysis during operation of the lathe machine include the gear train displacement, velocity, acceleration, resonance, and natural frequency.

2.1. Modelling of the Gearbox Train

The machine gearbox is an assembly of twelve meshing gears mounted on three shafts. The gears are spur gears of defined pitch diameters and numbers of teeth. The model for the gear train is as shown in Figure 1. The modelling of the gear train was done on the SolidWorks software using the various dimensions as given in Table 1.



Figure 1: 2D Model for Lathe Machine Gear Train Assembly

The gears A, B, G, and H were modelled and assembled on shaft 1. The gears on the shaft 1 are designed to mesh with the gears C, D, E, and F on shaft 2 depending on the speed selection of the gear train during operation of the lathe machine. Two gears I and J are also assembled on shaft 2 to mesh with gears K and L which were carried on shaft 3. The shafts were modelled as shown in Figures 2.



Figure 2: (a) In-line shaft model for gears A, B, G, H (b) In-line shaft model for gears C, D, E, F, I, J (c) In-line shaft model for gears K, L

The dimensions of the individual gears making up the train are as given in Table 1. These parameters were used to develop the 3D model for the individual gears and then assembled to make the gear train.

	Gear A	Gear B	Gear G	Gear H
Pitch circle diameter (mm)	90	58	40	74
Number of teeth	40	25	18	33
Module	2.25	2.25	2.25	2.25
	Gear D	Gear C	Gear F	Gear E
Pitch circle diameter (mm)	108	74	90	123
Number of teeth	48	33	40	55
Module	2.25	2.25	2.25	2.25
	Gear I	Gear J	Gear L	Gear K
Pitch circle diameter (mm)	110	40	83	151
Number of teeth	40	16	30	55
Module	2.75	2.5	2.75	2.75

Table 1: Modelling Data for the Gear Train

2.2. Modal Analysis

Modal analysis was used to determine the dynamic properties of the lathe machine gearbox in the frequency domain by determining the mode shapes and natural frequencies. The modal analysis technique exposed the dynamic characteristics of the system under vibration excitation. The vibration characteristics such as the natural frequencies, modes shapes and the damping factors of the machine were then determined. The mode shape of the system is the deformed shape of the system at a specific natural frequency. The focus of the modal analysis is to determine the frequency at which resonance of the vibration is likely to occur so as to avoid operating the machine at such resonance level.

The analysis is carried out with the fixed shafts ends supports boundary conditions. The results of the simulation are as shown in Table 3. The numerical mode shapes of the system with respect to the frequencies are as obtained in Figures 3 to 8.

Mesh Sizing				
Relevance centre	Coarse			
Element size	Default			
Initial size seed	Active assembly			
Smoothing	Medium			
Transition	Fast			
Span angle centre	Coarse			
Minimum edge length	9.6587 x e ⁻⁶			

Table 2: Mesh Sizing For Gear Train System

2.3. Harmonic Analysis

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The harmonic analysis was used to develop the Frequency Response Function (FRF) of the natural frequencies. The frequency response is characterised by its magnitude in this case. The analysis was done using the ANSYS software. The harmonic analysis was linked with the modal analysis for purpose of sharing data. The input to the lathe machine was subjected to a range of frequencies 0-800 Hz and a solution interval of 10. The amplitude of the stress and deformation responses at each frequency was recorded. The mode superposition method was applied for the analysis. The simulation was done with an input acceleration of 100 m/s². The frequency response for stress distribution and deformation were obtained as presented in Figures 9 and 10.

3. Result and Discussions

3.1. Result of the Modal Frequencies

Table 3 shows the natural frequency obtained for the modal analysis

Mode	1	2	3	4	5	6
Natural frequency (Hz)	432.91	482.75	507.83	544.16	671.98	735.11
Table 3: Frequency of the Gear Train Analytical Model						

The characteristic displacement pattern for the frequency distribution is as shown in Figures 3-8 indicating the 1st to the 6th modes pattern of the vibrating gear train. The analysis exposed the likely portion of the gears and shaft where failure might occur during operation of the lathe machine at the respective frequencies.



Figure 3: The First Mode Shape

Figure 3 shows the mode shape of the first frequency of 432.91 Hz. At this frequency the characteristic of the system is the bending along the y axis with maximum deformation 402.52 mm occurring on gears D and E and minimum deformation observed on shaft 3. This implies that likely failure of the gear system at this frequency could occur on gears D and E at this operating frequency.

The 2nd mode shape is shown in Figure 4 where the maximum deformation occurs on gear B and shaft 1. The second order of the frequency of 482.75 Hz has maximum deformation of 428.98 mm acting on shaft 1 with the minimum deformation observed on shaft 3.



Figure 4: The 2nd mode shape

Figure 5 shows the mode shape of the third frequency. The vibration mode at this frequency of 507.83 Hz is bending along Y axis with the maximum deformation of 526.07 mm on the compound gears A, B with considerable portion of the deformation on gear B.



Figure 5: The 3rd mode shape



Figure 6: The 4th Mode Shape

Figure 6 shows the mode shape of the fourth natural frequency. The deformation in the system is pronounced on gears D and K with excessive deformation of the shaft 3.



Figure 7: The 5th Mode Shape

Figures 7 and 8 shows the mode shapes of the 5th and 6th natural frequencies respectively. Vibration modes of the 5th and 6th mode shapes both occur on the gears A and B. The likely failure of the system could occur in a similar manner as shown but the deformation magnitude of the two modes are distinct at 944.32 mm for the 5th mode and 856.58 mm for the 6th mode.



Figure 8: The 6th Mode Shape

4.2. Results of Frequency Response Analysis

The magnitude characteristics curves for the stress and deformation of the system are shown in Figures 10 and 11. The distinct feature of the response functions are the prominence of the resonance of the stress FRF and deformation FRF at a close range value of the system frequency, the system stiffness and mass.



Figure 9: Stress Modulus Log-Log FRF



Figure 10: Deformation Modulus Log-Log FRF

The resonant peak values of the response occur at the resonant frequencies of 479.89 Hz. At low frequency in vibration theory the FRF is dominated by the stiffness characteristic of the gear train and by the mass characteristic at high frequency. The estimate of the stiffness and mass of the system are obtained from the stiffness and mass line intercepts. The resonance magnitude characteristics are dependent on the damping ratio ξ according to the expression of equation (1)

$$M_r = \frac{1}{2\xi\sqrt{1-\xi^2}} \tag{1}$$

 M_r is the resonance peak value of the system. This damping characteristic described the function at the vicinity of the resonance. Table 4 shows the estimated parameters from the system simulation. The natural frequency and band width of the system is obtained from equation (2) and (3).

$$\omega_n = \frac{\omega_r}{\sqrt{1-\xi^2}}$$

BW = $\omega_n \left[1 - 2\xi^2 + \sqrt{2 - 4\xi^2 + 4\xi^4} \right]^{1/2}$

The transient response of the system is the bandwidth measure of the system.

Parameter Estimation				
Resonance peak value, M_r (dB)	126.01			
Resonance frequency, f_r (Hz)	479.89			
System stiffness, k (N/mm)	19.9526			
System mass, m (kg)	133.352			
Damping ratio	0.00394			
Bandwidth, BW	4684.98			

Table 4: Parameter Estimation for the Lathe Machine Gear Train

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(2)

(3)

The frequency response analysis could be used to verify the steady-state response of a lathe machine gear train, which enables engineers to determine whether the gear train of a lathe can withstand resonance or other problems related to vibration during its operating life.

4. Conclusion

This study presents the modal and harmonic analysis of lathe machine gear train, the harmonic analysis and modal analysis was done using ANSYS R15 work bench the material used was grey cast iron. The first six natural frequency was determined, it was seen that after performing the harmonic analysis the maximum amplitude occurred at a frequency of 480 Hz which was similar to the frequency between the first and second mode predicted by doing the modal analysis, from this analysis it could be concluded that resonance will occur at a frequency of 480Hz and the maximum failure will occur on the compound gear C, D, E, F and shaft 1. Modal analysis was used to determine the vibration characteristics of the lathe machine gear box, by determining the mode shapes and natural frequencies. The mode shape and FRF parameters could be use to assess the condition of the lathe machine for predictive maintenance to avoid sudden failure.

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