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Numerical Study of Machine Tool Spindle Design with Focus On Blank Dimensional Accuracy

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Abstract

This paper examines the potential for a locally made lathe machine to produce machined parts with high and acceptable dimensional accuracy with particular focus on the technical status of the rigidity, vibration, and strength properties of the spindle unit. The spindle unit design is very highly important because this unit actually controls the operations of the lathe machine, therefore the operating performance in obtaining machined products with high precision dimensions is investigated. The models used by Acherkan et al (1973), for rigidity, vibration proof and strength properties were applied in this study. The deflection at the nose of the spindle was found to be 2.0mm, the vibration dimensionless property was determined to be 0.58, and the factor of safety was calculated to be 0.375. These values are at great variance with standard values. They indicate that the spindle unit has been under designed and is capable of producing blanks with marked dimensional inaccuracy. Therefore the spindle unit has been recommended for redesign. The design procedure adopted for this study successfully determined the technical status of the spindle unit.

Keywords: machine tool, rigidity, spindle unit, strength, vibration proof property.

1.0 Introduction

Multipurpose lathe machines are used for several machining and work operation purposes such as milling, drilling, boring, cutting and facing. Chapman [2] described a multipurpose or standard machine as the one which is able to deal with a variety of work, and permits a reasonably wide range of operations to be performed. The multipurpose machine, such as the lathe machine is used for the production of several machine tools/parts. It is uneconomical, for instance, to produce a unique screw bolt for joining a particular part to another one, because if that bolt breaks, the technician would have to visit the jobshop to order another one. This limitation is usually eliminated by producing a bolt that can fit into so many similar holes used to join the required parts together. A multipurpose bolt is essential, and the ability of the lathe machine to produce this bolt within the range of tolerances of the required size is crucial. Dieter [3] said that the use of standards provide a definitive solution to a repetitive problem with the best technical means available at the time. He also added that the greatest cost saving comes from re-using existing parts in design. The main savings come from eliminating the need for new tooling in production and from a significant reduction in the parts that must be stocked to provide service over the lifetime of the product. Investing in standardization is therefore an economically sound focus leading to substantial cost savings in inventory and manufacturing.

Thus, the accuracy of the tool dimension is of great importance and this study intends to investigate the rigidity status of the spindle unit of a locally made lathe machine. Kung and Huang [4] said that in machining systems, the spindle is the most critical element that affects the dynamic performance and capabilities of the system in the machining process.

Chapman [2] said that the stiffness and rigidity of machine tools are important factors in the success or otherwise with which they perform their functions. If, when under the load of the cut, an undue amount of distortion or deflection takes place in several portions of the machine, the accuracy of the work will suffer, perhaps to the extent of rendering it unfit for service. A machine which is not solid or rigid enough to absorb and damp out the vibrations promoted by the cutting operation, but allows them to be transmitted to the work or cutter, will produce work of a poor finish, and the vibrations (or chattering) may be serious enough to prevent the operation from proceeding except under very reduced conditions of speed and efficiency. Chapman [2] also said that an additional factor which has necessitated increased rigidity has been the increase in speed and power made possible by the development of high efficiency cutting alloys, the use of which to their full capacity demands the highest possible properties of stiffness and rigidity in machine tools.

Majou et al [5] were of the opinion that heavy moving parts require from the machine structure high stiffness to limit bending problems that lower machine accuracy, and limit the dynamic performances of the feed axes. Symens et al [6] said

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that there is growing competition as international markets generate a demand for faster machine tools that can reduce machining time, while preserving or improving the final accuracy.

High accelerations of these machines excite the machine structure up to high frequencies leading to marked vibrations. These structural vibrations need to be damped if accurate positioning and trajectory tracking are to be achieved. In this study, the importance of rigidity, vibration, and machine strength were studied to know if the locally produced lathe machine has the expected damping structure to sustain the accurate or precise production of machine parts.

2. Materials and Methods

2.1: Materials

A multipurpose lathe machine made by Technodrill Company in Nigeria, is made from steel and cast iron available locally was used to machine a 40mm rectangular blank. The spindle revolves at a speed of 1400rpm. The measurements taken and recorded are shown in Tables 1 - 3. These measurement values were used for rigidity, vibration and strength design evaluations.

2.2: Methods

Rigidity Design

Spindle design is usually based on rigidity calculations. Rigidity determination usually involves the calculation of deflection in bending, γ, in some cases twist in torsion, vibration proof and strength calculations. Generally, spindle is replaced by a beam on hinged supports as illustrated by [1]. Acherkan et al. [1] gave the deflection, γ at the spindle nose and slope, θ in the front support, respectively, as

$$\gamma = \frac{1}{3EI}\left[P_1 a^2\left(a+l\right) - 0.5P_2 abl\left(1-\frac{b^2}{l^2}\right) - Mal\right] \tag{1}$$

and

$$\theta = \frac{1}{3EI}\left[P_1 al - 0.5P_2 al\left(1-\frac{b^2}{l^2}\right) - Ml\right] \tag{2}$$

where

I = average value of the moment of inertia of the sections of the spindle, M = reactive moment in the support ≤ 0.35 P_1a, P = peripheral force, a = overhanging part of the length of the spindle, b = width of the spindle part.

Considering the elastic strain of the support, the deflection at the nose of the two supports of the spindle which was acted upon by the drive system, becomes

$$\gamma = \frac{P}{j} = P\left[\frac{1}{kj_o} + \frac{1}{jo_{oh}} + \frac{\left(1+k\right)^2}{j_B} + \frac{k^2}{j_A}\right] \tag{3}$$

Where

j = rigidity of the spindle unit

$j_o = \frac{3EI_1}{a^3}$ = conditional rigidity of the spindle in the length between the supports.

$jo_{oh} = \frac{3EI_2}{a^3}$ = conditional rigidity of the overhanging part of the spindle,

$$k = \frac{a}{l} = \textit{ratio of the overhanging part to the length between} \sup \textit{ports}$$

I_1 and I_2= average moments of inertia of the section of the spindle between the supports and in the overhanging part, respectively.

j_A and j_B = rigidity of the front and rear spindle supports, respectively.

E = young's modulus of the spindle material.

Equation (3) which depicts the formula for obtaining the deflection at the spindle nose is simplified, for ease of the application, by taking the reciprocal of rigidity j as a unit deflection, C, that is,

$$C = \frac{1}{j} \tag{4}$$

substituting Eq (4) into Eq (3), leads to

$$\gamma = P\left[\frac{1}{k}C_o + Co_{oh} + (1+k)^2 C_B + k^2 C_A\right] \qquad (5)$$

Vibration Behaviour Calculation

Theory of vibration behavioral pattern in machine tools is applied to determine the natural frequency of the spindle in order to avoid resonance vibration, which is usually carried out for high speed spindles [1].

$$P_r = A\rho\omega_o^2\gamma \qquad (6)$$

Where, A is the cross sectional area of the spindle, ρ is the mass density of the spindle material and Equation (6) is used to determine the centrifugal force (in kg), Pr when considering the random angular velocity, ω_o

The critical angular velocity, ω_{cr} is given as

$$\omega_{cr} = \omega_o\sqrt{\frac{Y_{II}}{Y_I}} \qquad (7)$$

Where Y_I, Y_{II} are elastic lines drawn with some degree of accuracy by deflective wavelengths. This is constructed representing the deflection due to the weight of the spindle. Figure 1 shows the determination of elastic lines illustrated by [1].

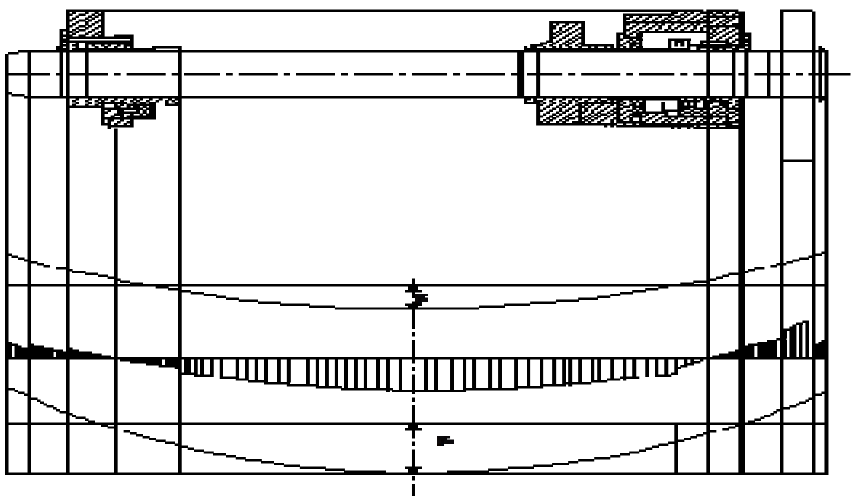

Fig. 1: Vibration Behavior of a Spindle showing Elastic Lines drawn with some Degree of Accuracy by Deflective Wavelengths [1]

The vibration behavior of the spindle unit is defined by the level of resonance that has occurred. To eliminate the dangers of resonance, the condition posed by Eq. (8) should be met. Resonance vibration for high speed spindles is the major cause of dimensional inaccuracies of workpiece.

$$\left[\frac{\omega_{cr} - \omega}{\omega}\right] \geq 0.25 \; to \; 0.3 \qquad (8)$$

Where ω = maximum angular velocity of the spindle rotation

Strength Calculation

Strength calculations are used for checking heavily loaded spindles. This involves checking the factor of safety, n for alternating stresses.

$$n = \frac{\left(1-\xi^4\right) d_o^3 \sigma}{10\sqrt{\left(aM\right)^2 + \left(a_k M_k\right)^2}} \tag{9}$$

Where,
d_o = outside diameter of the spindle,
$\xi = \frac{d_o}{d}$ = ratio of the inside to the outside diameter of the spindle
σ = endurance limit for bending with a symmetrical cycle of stresses
M and M_t are average values of the bending moment and torque
a and a_k are coefficients that take into accounts stress concentration and the degree of variation of the moments (and torque) which are determined by using the expressions in
Eqs (10) and (11)

$$a = k_\sigma \left(1 + C\right) \tag{10}$$

and

$$a_k = \frac{\sigma}{\sigma_T} + k_\tau C_k \tag{11}$$

Where k_σ and k_τ = dynamic coefficients of stress concentration for normal and shearing stresses ($k_\sigma \approx k_\tau \approx 1.7$ to 2.0). σ_T = yield stress, and the coefficients,

$$C = \frac{M_a}{M} \tag{12}$$

$$C_k = \frac{M_{ta}}{M_t} \tag{13}$$

Where M_a is the amplitude of the moment, M_{ta} is the amplitude of the torque and M, M_t are the average values of the bending moment and torque.
However, [1] suggested that for finished turning and drilling $C \approx C_k \approx 0.1$ to 0.2. For milling and roughing operations in which an extremely non-uniform allowance is removed $C \approx C_k \approx 0.3$ to 0.5. Safety factor is usually considered to be within 1.3 to 1.5.

Discussion Of Result

Table 1 contains the measured parameters and parameters obtained from manufactured specifications.

Table 1: Measured Parameters for determining the Rigidity Design

Spindle support length, l	15mm
Overhanging part of the spindle length, a	3mm
Load applied to the spindle nose influenced by the centrifugal force	0.73kg
The rigidity of the front spindle support, j_B	4kg/mm
The rigidity of the rear spindle support, j_A	5kg/mm
The average moments of inertia of the sections of the spindle between the supports, I_1	12kgmm
The average moment of inertia of the sections of the spindle between the overhanging parts, I_2	9kgmm
The Young's Modulus of the spindle material, E	2.0N/m^2

From the calculations done applying Eqs. (1 – 4), the conditional rigidity of the spindle in the length between the supports and in the overhanging part of the spindle were determined to be 0.375kg/mm and 0.50kg/mm respectively. These rigidity values are sufficient to keep the spindle supports and overhanging parts in a non-deflecting position. However, a spindle nose deflection of 2.0mm was obtained. This deflection is large compared to the permissible deflection at the spindle nose which is limited to 0.33mm of the tolerance for runout at the spindle nose. It is also expected to have a maximum deflection of $\leq$ 0.002L, where L is the distance between supports [1]. The maximum deflection in this case is 0.003mm. The deflection at the spindle nose was simulated by vibrations induced by the electric motor. This deflection value is 99.85 times that of the standard specified value.

The vibration proof property of the spindle nose was also investigated using the values in Table 2.

Table 2: Measured Parameters and Parameters from Secondary Data used for Determining the Vibration Behaviour of a Milling Machine.

Centrifugal Force	60kg
The cross sectional area of the spindle material	$4mm^2$
Spindle deflection, γ	2mm
Vibration causing a wavelength with elastic lines drawn with some degree of accuracy of $\pm$1.5mm and $\pm$2.6mm	
Angular velocity, ω	1.2m/s
Mass density of spindle material, ρ	$3.6kg/mm^2$

Using Eqs. 5 – 7, the random angular velocity of 1.44mm/s was obtained. This velocity is responsible for the fluctuating movement of the vibration of the moving parts of the spindle unit, but when this velocity reached 1.90mm/s, which is the critical angular velocity, chattering may occur. When the chattering increased steeply, the danger of resonance is visibly observed. The vibration proof property, tends to reveal the fact that if the rigidity structure of the machine is stable and adequate, there would be no resonance or vibration, and dimensional accuracy will be certain. Based on the criteria specified for eliminating the danger of resonance, when $\left|\frac{\omega_{cr}-\omega}{\omega}\right| \geq 0.25\ to\ 0.3$ the danger of resonance has been eliminated [1]. In this study, 0.58 was obtained. This value does not lie between the specified values, therefore, the danger of resonance is not eliminated and the locally made lathe machine is highly prone to vibrations.

The strength used to evaluate the load bearing capacity of the spindle unit was determined. The strength property was assessed by determining the factor of safety. The factor of safety here means the design factor which allows for variability in materials, variability in construction practices, and uncertainties in in-service conditions (Hopkins, 2002). Table 3 was used to determine the factor of safety.

Table 3: Measured Parameters used for the determination of the factor of safety.

Outside diameter of the spindle shaft material, d_o	1.9mm
Inner diameter of the spindle shaft material, d_i	1.0mm
The endurance limit for bending with symmetric cycle of stresses, δ	$220N/mm^2$
Average value of the bending moment, M	120Nmm
Torque generated, M_t	180Nmm
Dynamic coefficient of stress concentration for shearing stress, K_δ	1.7
Yield stress, δ_τ	$210N/mm^2$
Amplitude of the moment, M_a	45mm
Amplitude of the torque, M_{ta}	40mm

The factor of safety was determined to be 0.375 through Eq (9). This value is not within the specified range of 1.3 to 1.5 [1]. Therefore the spindle unit was under designed and should be recommended for redesign to attain the recommended range.

Conclusion

The rigidity, vibration and strength status of the locally made lathe machine was studied. From the findings, it was found that the deflection at the spindle nose is large and capable of causing great danger of resonance. This indicates that the vibrations that took place in the spindle unit are steep, with very high angular velocity. These conditions make it difficult to produce a blank material with acceptable dimensional accuracy. This is further buttressed by the factor of safety which value shows that the spindle unit has been grossly under designed.

Summarily, the lathe machine is not fit for use because it is outside the quality control scope required to design a conventional lathe machine. Therefore, the locally produced lathe machine is recommended for redesign taking into consideration the variations in design standards and the primary objective of producing a product with high dimension accuracy by ensuring the stable rigidity position, vibration damping capability and strength status of the locally made lathe machine.

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