**ISSN:** 1231-4005
**e-ISSN:** 2354-0133
**DOI:** 10.5604/12314005.1137848

# OPTIMIZING STATIC AND DYNAMIC STIFFNESS OF MACHINE TOOLS SPINDLE SHAFT, FOR IMPROVING MACHINING PRODUCT QUALITY

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***Abstract***

*Automotive industries in Indonesia have grown up fast, and in-line with it, machine tools population has also grown up. Nevertheless, most of these machines are imported, because there are only few machine tools industries in Indonesia. Based on this situation, Indonesia has started to build its own machine tools industry in order to support mostly, automotive industries. This paper deals with an activity in machine tools performance improvement researches in Indonesia, that is, improving spindle static and dynamic stiffness. Spindle shaft is the weakest point in machine tools structure, increasing its stiffness will further increase machine tools accuracy and machining product quality as well. Moreover, high productivity needs machine tools with high speed machining capability, which leads into unavoidable dynamic effects that occur in the machine tool spindle during production process such as regenerative chatter. A study has been conducted on developing mathematical model for calculating static and dynamic stiffness of machine tools spindle shaft and studying changes of spindle shaft static and dynamic stiffnesses, by varying bearings preload, number of bearings, bearings span and spindle length. The study was conducted theoretically and experimentally on a national-built CNC vertical milling machine. The study shows that the machine tools spindle shaft stiffness can be increased by optimizing the parameters aforementioned.*

***Keywords:*** *static stiffness, dynamic stiffness, preload*

## 1. Introduction

Ground transportation industry or automotive industry is one of the rapidly growing areas in Indonesia. In 2008, automotive industry has contributed 8.2% of gross domestic product and this is the biggest contributor to the manufacturing industry category, which reached 27.4%. That is why the automotive industry becomes one of the three industries that are expected to drive the growth of national industry and economy of Indonesia [1].

Machine tools are the main driving force of manufacturing industry, including automotive industry. Development in automotive sectors is always coupled with development in machine tools sectors. In Indonesia, development in machine tools sector can be analysed from its machine tools import and export data, in shown in Fig. 1.

Figure 1 shows that exported machine tools quantity in Indonesia is approximately one-tenth of the imported one. Due to the situation, the government to speed up machine tools industry in Indonesia has established a project, which involves government, university and machine tools industries. The first project is to build a 3-axis vertical milling machine, checking its accuracy and then improving it. The geometric accuracy of the machine has been successfully improved closed to commercial machine tools accuracy [3-7]. Nevertheless, the spindle speed is still limited to 3000 rpm, far beyond the target of 10 000 rpm. In order to solve the problem, a study has been conducted to build a high-speed spindle which has high static and dynamic stiffness but with moderate bearing temperature during operation. This paper deals with the research progress.

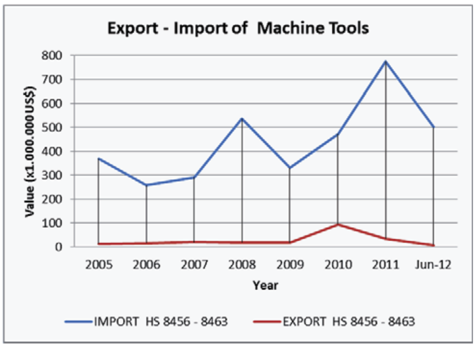


*Fig. 1. Indonesia Machine Tools Export and Import from 2005 to 2012[2]*

Machine tool has accuracy which will affect machining product quality. The higher the accuracy of the machine tools, the higher the quality of the machining product. Accuracy is affected by many factors; among them are geometrical accuracy, static stiffness, dynamic stiffness. Among others, machine tools spindle is the weakest point; therefore, more attention must be given to this part. Spindle static and dynamic stiffness are affected by bearing preload, distance between bearing and spindle dimensions.

This paper deals with optimizing static and dynamic stiffness of a machine tools spindle. In order to realize it, firstly, theoretical model of static stiffness, dynamic stiffness must be formulated or taken from literature. The models are then validated by an experimental study. The validated models are then utilized to get optimum spindle shaft static and dynamic stiffness.

## 2. Static Stiffness of a Spindle Shaft

### 2.1. Theoretical Static stiffness of spindle shaft

The spindle shaft is modelled as a stepped shaft, supported by two bearings. The smaller diameter part is called as spindle shaft body, while the bigger one is called as spindle shaft head. A force $P$ is exerted on the head part, as shown on Fig. 2.

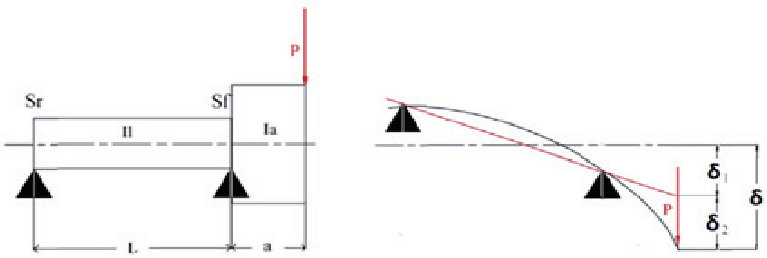


*Fig. 2. Spindle shaft static analysis*

Based on Fig. 2., displacement δ can be calculated by using equation (1):

$$\delta = P\left(\left(\frac{L+a}{L}\right)^2\left(\frac{1}{S_f}\right)+\left(\frac{a}{L}\right)^2\left(\frac{1}{S_r}\right)+\frac{a^2}{3E}\left(\frac{L}{I_L}+\frac{a}{I_a}\right)\right). \tag{1}$$

where:

$S_f$ = front bearing stiffness (close to the tool),
$S_r$ = rear bearing stiffness (far from the tool),
$I_l$ = Inertia moment spindle shaft body,
$I_a$ = Inertia moment spindle shaft head,
L = Spindle shaft body length (m),
a = Spindle shaft head length (m),
P = External force (N).

### 2.2. Model Validation

The static stiffness model validation was conducted experimentally, on a spindle of national built machine tools [3]. The experimental setup is shown on Fig. 3. The procedures are as follow:

– static load was actuated at the spindle shaft head by using pneumatic actuator,
– the corresponding deflection was measured by using a dial indicator.

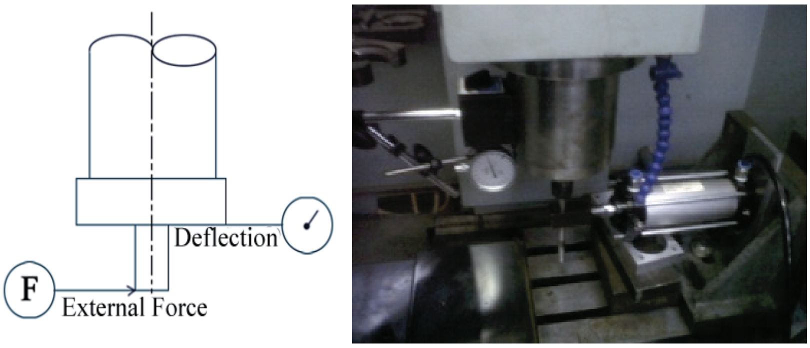


*Fig. 3. Experimental setup for model validation*

The experimental result is then compared to the theoretical model, and the result is shown on Fig. 4.

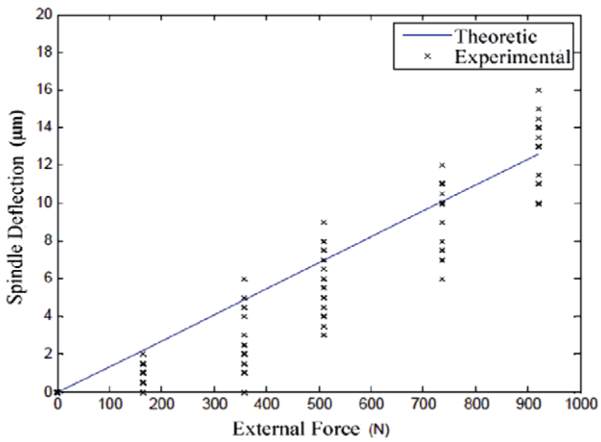


*Fig. 4. Validation Result*

Based on Fig. 4, maximum deviation of the experimental result from its theoretical model is about 2 μm. Therefore, the theoretical model is valid for predicting spindle shaft static stiffness behaviour.

## 3. Dynamic Stiffness of a Spindle Shaft

### 3.1. Mathematical Modelling

Unlike static stiffness, dynamic stiffness is frequency dependent. Nevertheless, except for high speed machine tools, when the lowest resonance frequency is kept higher than the highest operational speed than the dynamic stiffness magnitude is no longer taken into account. Therefore, concerning the dynamic stiffness, the aim is to shift the lowest resonance frequency as high as possible.

Spindle resonance frequencies are estimated by using Dunkerley method for rotating shaft radial vibration [8]. If a shaft has $i$ number of element and a force $P$ works on element a, then every element will undergo deflection $y_n$ as follow:

$$y_n = \eta_{na} P_a. \tag{2}$$

If there are other forces working on other elements b, c and so on, then the deflection of every element is:

$$y_n = \eta_{na} P_a + \eta_{nb} P_b + \eta_{nc} P_c + \dots, \tag{3}$$

where:

$y_n$ = deflection of the n$^{th}$ element,

$\eta_{na}$ = deflection characteristic of the n$^{th}$ element due to force working on element $a$ (so are b, c, and so on),

$P_n$ = force working at the n$^{th}$ element which consists external force ($F_n$), mass centrifugal force ($M_n$) of the element and bearing reaction force ($K_n$).

Equation (3) for all elements can be expressed in matrix form:

$$\begin{bmatrix} y_1 \\ y_2 \\ y_3 \\ \vdots \\ y_n \end{bmatrix} = \begin{bmatrix} \eta_{11} & \eta_{12} & \eta_{13} & \cdots & \eta_{1n} \\ \eta_{21} & \eta_{22} & \eta_{23} & \cdots & \eta_{2n} \\ \eta_{31} & \eta_{32} & \eta_{33} & \cdots & \eta_{3n} \\ \vdots & \vdots & \vdots & \ddots & \vdots \\ \eta_{n1} & \eta_{n2} & \eta_{n3} & \cdots & \eta_{nn} \end{bmatrix} \begin{bmatrix} P_1 \\ P_2 \\ P_3 \\ \vdots \\ P_n \end{bmatrix}. \tag{4}$$

Each value of $\eta_{xa}$ is calculated as follow:

$$\eta_{xa} = \frac{A_{m1}\left(\frac{x^3}{6}\right) + A_{m2}\left(\frac{x^2}{2}\right) + C_{m1}x + C_{m2}}{EI}, \tag{5}$$

where m is spindle element number (1, 2, 3, 4 and 5).

The geometry and material data of the spindle are listed in Tab. 1.

*Tab. 1. Geometry and material data of the national-built machine tools spindle*

| Item | Unit | Symbol | Sub-structure | | | |
|---|---|---|---|---|---|---|
| | | | A | B | C | D |
| Length | mm | L | 24 | 134 | 76 | 136 |
| Outer dia. | mm | $D_0$ | 60 | 45 | 45 | 40 |
| Inner dia. | mm | $D_i$ | 26 | | | |
| Moment of Inertia | mm$^4$ | I | 613741 | 178857 | 178857 | 103232 |
| Area | mm$^2$ | A | 9186 | 4238 | 4238 | 2903 |
| Density | Kg/m$^3$ | ρ | $8.06 \times 10^{-6}$ | | | |
| Elastic moduli | MPa | E | 207000 | | | |

The result is a compliance or flexibility and is shown in Fig. 5.

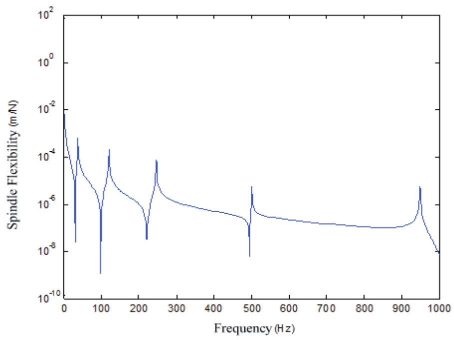


*Fig. 5. Compliance/flexibility of the spindle shaft model*

### 3.2. Dynamic Model Validation

The model is then validated by conducted an FRF experiment. For this purpose, accelerometer, impact hammer and spectrum analyser are needed. The spindle model during experiment is shown in Fig. 6.

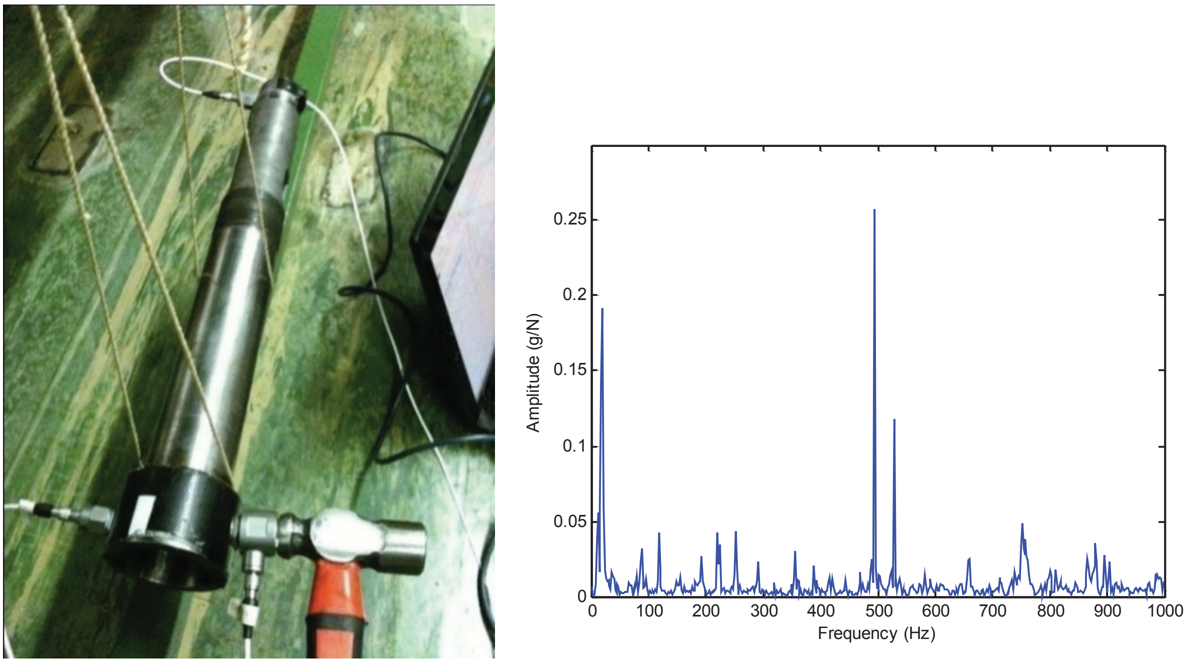


*Fig. 6. Spindle shaft FRF receptance testing*

The resonance frequencies from the model and from the experiment are compared in Tab. 2.

*Tab. 2. Natural Frequencies comparison between model and experiment*

| **Resonance Frequency (Hz)** | |
|---|---|
| Model | Experiment |
| 39 | 36 |
| 122 | 118 |
| 248 | 252 |
| 500 | 494 |
| 950 | 902 |

Based on the result, the theoretical model is valid and can be used to predict the resonance frequencies of the spindle.

## 4. Optimizing spindle shaft stiffness

### 4.1. Preload effect

By using the theoretical model, effect of preload on spindle static stiffness and 1$^{st}$ resonance frequency can be examined, and the results are shown in Fig. 7.

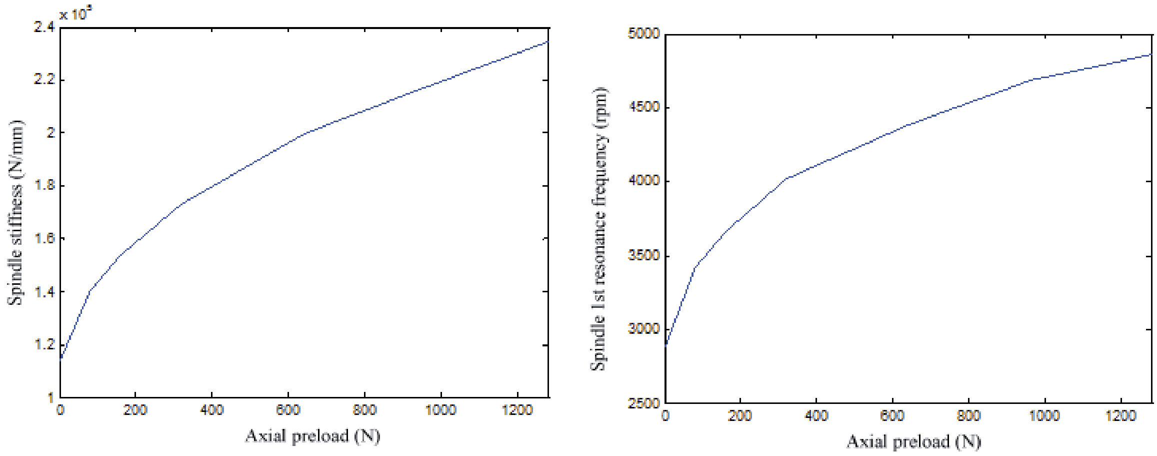


*Fig. 7. Preload effect on static stiffness and 1st resonance frequency*

### 4.2. Bearing span effect

Bearing span affects spindle static stiffness and 1st resonance frequency as shown in Fig. 8. The figure shows that there is an optimum-bearing span at which the static stiffness reaches its maximum value.

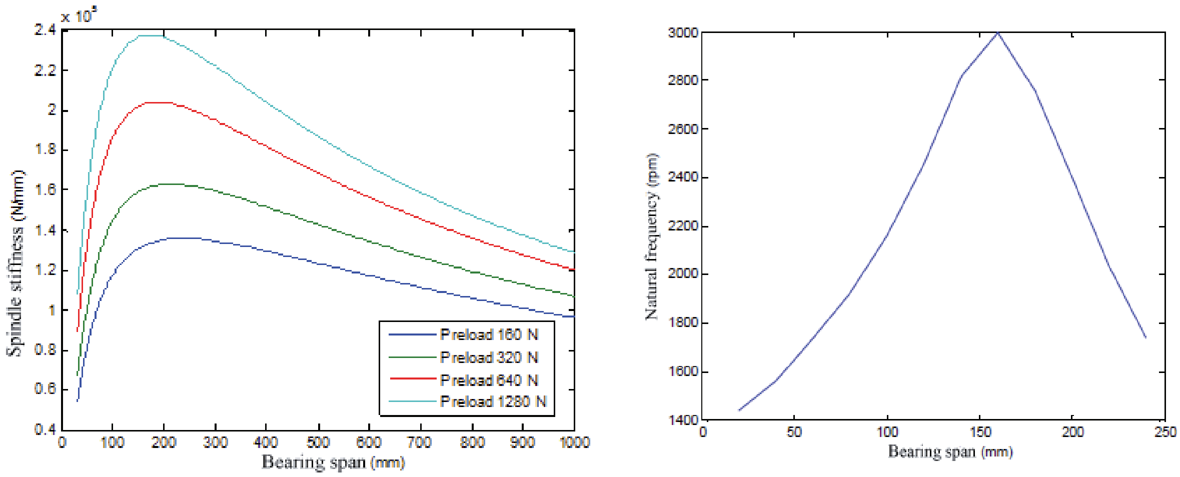


*Fig. 8. Bearing span effect on spindle static stiffness and 1st resonance frequency*

### 4.3. Number of bearing effect

In order to find number of bearing effect, three types of bearing configuration are analysed, as shown in Fig. 9.

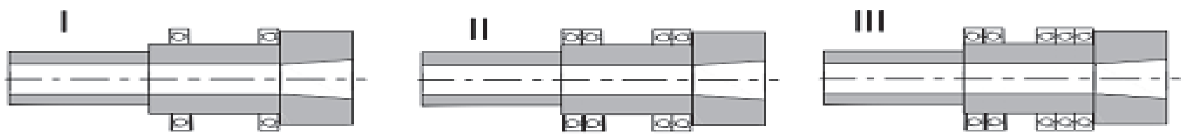


*Fig. 9. Bearing distance effect on spindle shaft static stiffness*

Static stiffness and 1st resonance frequency of the three configurations are shown in Fig. 10.

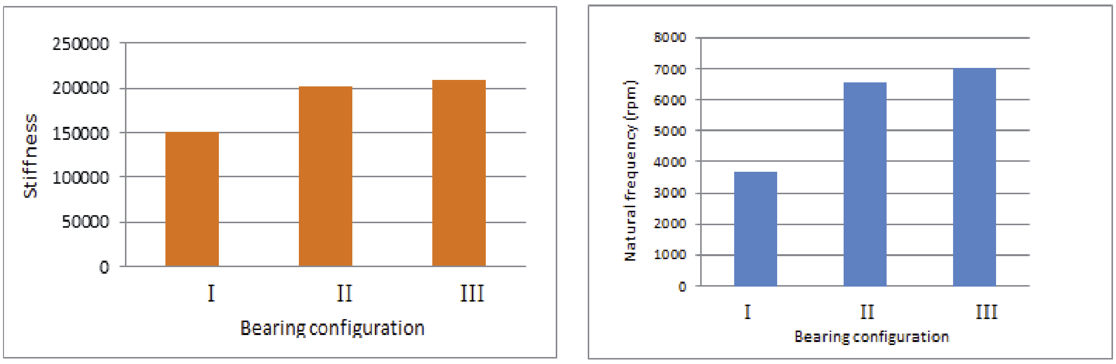


*Fig. 10. Bearing distance effect on spindle shaft static stiffness and 1st resonance frequency*

Figure 10 show that by increasing the number of bearing will increase spindle stiffness and its corresponding natural frequency, but it is not linear.

### 4.4. Spindle length effect

In this case, length modification is only conducted on front part of the shaft (spindle head), while dimensions from front bearing to the rear are kept unchanged. The modification results in natural frequency shift, as shown on Tab. 3, but the thermal characteristic is relatively unchanged.

*Tab. 3. Spindle length effects on spindle first natural frequency*

| Length of spindle head (mm) | Stiffness (N/µm) | Natural frequency (rpm) | |
|---|---|---|---|
| | | 1 | 2 |
| 6 | 219 | 3180 | 8640 |
| 12 | 197 | 3420 | 8460 |
| 24 | 151 | 3660 | 8160 |
| 36 | 118 | 4080 | 7560 |
| 48 | 90 | 4500 | 6480 |

## 5. Conclusion

Mathematical models for identifying static stiffness, dynamic stiffness and temperature rise in machine tool spindle have been developed. The models have conformed well to experimental measurement. Based on the models, variation of bearing stiffness, bearing span, number of bearing, spindle diameter and spindle length will change spindle static and dynamic characteristics.

Increasing bearing preload will increase spindle stiffness and first harmonic frequency. There is one optimum bearing distance, which results in maximum static stiffness. Increasing number of bearing will increase static stiffness but it is not linear. More bearing will also shift natural frequency to the right but also generate new resonance frequencies.

The method discussed in this paper can be used by spindle designer, during design stage, for rough estimation of the optimum spindle geometry, bearing configuration and preload value, which produces spindle with high static and dynamic stiffness.

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