Expert spindle design system

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Abstract

This paper presents an expert spindle design system strategy which is based on the efficient utilization of past design experience, the laws of machine design, dynamics and metal cutting mechanics. The configuration of the spindle is decided from the specifications of the workpiece material, desired cutting conditions, and most common tools used on the machine tool. The spindle drive mechanism, drive motor, bearing types, and spindle shaft dimensions are selected based on the target applications. The paper provides a set of fuzzy design rules, which lead to an interactive and automatic design of spindle drive configurations. The structural dynamics of the spindle are automatically optimized by distributing the bearings along the spindle shaft. The proposed strategy is to iteratively predict the Frequency Response Function (FRF) of the spindle at the tool tip using the Finite Element Method (FEM) based on the Timoshenko beam theory. The predicted FRF of the spindle is integrated to the chatter vibration stability law, which indicates whether the design would lead to chatter vibration free cutting operation at the desired speed and depth of cut for different flutes of cutters. The arrangement of bearings is optimized using the Sequential Quadratic Programming (SQP) method.

Keywords: Spindle design; Expert system; Chatter vibration; Finite element method; Optimization

1. Introduction

The spindle is the main mechanical component in machining centers. The spindle shaft rotates at different speeds and holds a cutter, which machines a material attached to the machine tool table. The static and dynamic stiffness of the spindle directly affect the machining productivity and finish quality of the workpieces. The structural properties of the spindle depend on the dimensions of the shaft, motor, tool holder, bearings, and the design configuration of the overall spindle assembly.

This research considers spindle component selection and configuration using the proposed expert system based on the digital knowledge base. The expert system with fuzzy logic is implemented as the selection system.

Eskicioglu et al. [1] developed a rule-based algorithm for the selection of spindle bearing arrangement using PRO-LOG, which is a programming language for expert systems. The bearing arrangements are determined by the cutting operation type, and the required cutting force and life of bearings. Wong and Atkinson [2] demonstrated a knowledge cell approach for diverse designs. They divided the knowledge cell into four parts; the Function, Selection, Graphics, and Logic cells.

For design optimization of spindles, Yang [3] conducted static stiffness to optimize a bearing span using two bearings, and described the methods used to solve the multi-bearing spans’ optimization method. Taylor et al. [4] developed a program which optimizes the spindle shaft diameters to minimize the static deflection with a constrained shaft mass. The Downhill Simplex Method is used to find the optimum value. Lee and Choi [5] conducted an optimization design in which they minimized the weight of the rotor-bearing system with the augmented Lagrange multiplier method. Chen et al. [6] and Nataraj and Ashrafuon [7] demonstrated the optimization results to minimize the forces transmitted by the bearings to the supports. Wang and Chang [8] simulated a spindle-bearing system with a finite element model and
compared it to the experimental results. They concluded that the optimum bearing spacing for static stiffness does not guarantee an optimum system dynamic stiffness of the spindle. Hagiu and Gafraru [9] demonstrated a system in which the bearing preload of the grinding machine is optimized. Kang et al. [10] conducted static and dynamic spindle analysis by using an off-the-shelf FE system with an added rigid disk and non-linear bearing model.

The previous research used only two support bearings, although practical spindles may use more bearings depending on the machining application. In addition, most of them optimize design parameters, such as shaft diameter, bearing span, and bearing preload, to minimize the static deflection. This paper considers more than two bearings in the spindle model and takes into account the chatter stability that is totally related to the dynamic properties of the spindle.

The overall expert spindle design system is outlined in Fig. 1. The design of the spindle with optimized bearing spacing is automated using the requirements set by the machining application, expert spindle design rules, cutting mechanics, structural dynamics and chatter stability of milling process.

2. Expert system for spindle design

The expert system for spindle design is introduced here to facilitate the design process using past experience and knowledge.

The expert system combined with the fuzzy logic is used as the selection system of components for the spindle design in order to handle uncertainties in the design process as illustrated in Fig. 2. The required input data for the spindle design, such as the cutting torque and power, are computed using the laws of cutting mechanics, as described in [11]. The input data is entered into the fuzzy inference system, which is established by design experts, and is fuzzified using membership functions. The Mamdani method is used as the inference system. The fuzzified values are applied to the fuzzy rules and aggregated using the maximum method. The result of the aggregation is defuzzified using the centroid method, and a defuzzified number is obtained. The simple defuzzified number is applied to the selection rule for the spindle components. An external database, which includes material cutting coefficients, is connected to the fuzzy inference system, which users can access. The supervising engineer, who is permitted to maintain this expert system, can modify the membership function.
and database when the tendency of the fuzzy terms, such as ‘high’, ‘middle’, and ‘low’ changes as the technology evolves. In this article, transmission and lubrication types are determined using the expert system with fuzzy logic.

2.1. Selection of transmission type

Torque has to be transmitted from the motor to the spindle shaft. There are a number of transmission types [12] and the main design configurations are gear (G), belt-pulley (B),...
direct-coupling (D), and motorized (M) types as summarized in Table 1.

Fig. 3 illustrates the process used to determine the type of transmission between spindle shaft and spindle motor.

The process of transmission type selection can be described as follows:

**Step 1. Evaluation of required cutting torque and power.**

The required cutting torque is evaluated from the given cutting conditions, and compared against the spindle motor specifications. The instantaneous cutting torque $T_c$ is evaluated based on the laws of cutting mechanics as described in [11]

$$ T_c = \frac{D}{2} \sum_{j=1}^{N} F_y(\phi_j), \quad \text{with } \phi_{st} \leq \phi_j \leq \phi_{ex} \quad (1) $$

where $D$ is the diameter of the milling cutter, and $N$ is the number of teeth of the cutter, $\phi_j$ is the instantaneous angle of immersion, and $\phi_{st}$ and $\phi_{ex}$ are entry and exit angles for the cutter. The tangential cutting force $F_y$ is given by;

$$ F_y(\phi_j) = K_{ac} a h(\phi_j) + K_{tc} a \quad (2) $$

where $a$ is the axial depth of cut, $K_{tc}$ and $K_{te}$ are the cutting and the edge force coefficients, respectively. The material-dependent cutting coefficients are evaluated from cutting experiments and stored in a database. The chip thickness variation $h(\phi_j)$ is expressed as follows

$$ h(\phi_j) = c \sin \phi_j \quad (3) $$

where $c$ is the feed rate (mm/rev-tooth).

The cutting torque $T_{max c}$ required for the spindle motor is the maximum value among the instantaneous torque $T_c$ in one tooth period

$$ T_{\text{max } c} = \max(T_c) \quad (4) $$

The cutting power $P_t$ is found from,

$$ P_t = V \sum_{j=1}^{N} F_y(\phi_j), \quad \text{with } \phi_{st} \leq \phi_j \leq \phi_{ex} \quad (5) $$

where $V=\pi D n$ is the cutting speed and $n$ is the spindle speed. Typically, the worst cutting condition, the slot milling ($\phi_{st}=0, \phi_{ex}=\pi$) is considered in sizing the spindle.

**Step 2. Spindle motor specifications.**
The spindle motor specifications must be determined to identify the transmission type. The power and torque diagram is shown in Fig. 4. The relation between motor power $P_{mo}$ and motor torque $T_{mo}$ can be expressed as follows

$$ P_{mo} = \frac{2\pi}{60} n_{max} T_{mo} \quad (6) $$

where $n_{max}$ is the motor rotation speed in rpm. $n_{max}$, as shown in Fig. 4, is the maximum motor speed.

**Step 3. Classification.**

The spindle motor specifications are checked by applying the maximum cutting torque $T_{\text{max } c}$ and the required corresponding spindle speed $n$ in Fig. 4. The expert system checks if the maximum cutting torque is below the motor torque $T_{mo}$. The system also checks whether the maximum speed $n_{\text{max}}$ of the motor is greater than the spindle speed used in the target machining application. $T_{ac}$ and $n_{\text{acmax}}$ are the actual spindle torque and actual maximum spindle speed, respectively, and are expressed as follows

$$ T_{ac} = T_{\text{mo}} \frac{G_s}{G_{mo}}, \quad n_{\text{acmax}} = n_{\text{max}} \frac{G_{mo}}{G_s} \quad (7) $$

where $G_{mo}$ and $G_s$ are the gear sizes of the motor side and spindle side, respectively.

If the required speed and torque for the target machining application are higher than the capacity of the motor, the designers are warned to make changes in the design. Table 2 shows the classification for the transmission selection.

**Step 4. Application of fuzzy logic.**

In the classification example shown in Step 3, the transmission possibilities are reduced from the gear, belt-pulley, direct coupling, and motorized type, to just 'gear/belt-pulley' or 'direct-coupling/motorized' mechanism. The fuzzy

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Table 2

<table>
<thead>
<tr>
<th>Classification for transmission selection</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gear</td>
</tr>
<tr>
<td>------</td>
</tr>
<tr>
<td>1)</td>
</tr>
<tr>
<td>2)</td>
</tr>
<tr>
<td>3)</td>
</tr>
<tr>
<td>(3-1)</td>
</tr>
<tr>
<td>(3-2)</td>
</tr>
<tr>
<td>4)</td>
</tr>
</tbody>
</table>
logic process for the transmission selection between the gear and belt-pulley types are shown as follows.

2.2. Selection between gear type and belt-pulley types

The following fuzzy sets are used to select the transmission between gear and belt-pulley types. These functions are set by referring to bearing catalogs [13–15] and the lead author’s 5 years of experience in a Japanese machine tool company as a spindle designer.

**Torque.** The belt-pulley type cannot transmit large torque compared to the gear. The membership function of the torque set is shown in Fig. 5. The torque can be classified as ‘Small’, ‘Middle’, and ‘Large’.

**Spindle speed.** The belt may expand due to the centrifugal force at a high speed range, which prevents the transmission of large torque. Fig. 6 shows the membership functions of the spindle speed. The belt and gear type transmissions are usually not available at a spindle speed range higher than 15,000 rpm. As a result, the membership functions of ‘Low’, and ‘Middle’ are used when the gear or belt-pulley type transmission is selected.

**Low cost versus high accuracy.** The belt-pulley transmission parts can be manufactured at a low cost because of their simplicity. In contrast, the gears are more costly due to the design and manufacturing complexity involved. On the other hand, due to the belt tension applied at the pulley point, the spindle shaft deflects and the rotation accuracy of its spindle is not as good as the gear type. The membership functions of low cost or high accuracy sets are shown in Fig. 7. The users have to select an integer weighting number between 1 and 10, which indicates the conflict between the cost and accuracy.

**Gear versus belt-pulley.** The transmission type must also be fuzzified. The membership functions of the gear or belt-pulley are shown in Fig. 8. These membership functions are used in the implication process.

The Fuzzy rules shown in Table 3 are applied to the fuzzified values of torque, speed, cost versus accuracy, and gear/belt transmission type via membership functions. These rules are defined from design principles.

The following example illustrates the expert spindle design procedure. The required cutting torque, maximum spindle speed, and the weight value for cost/accuracy are given as 180 N m, 6000 rpm, and 7, respectively. These three numbers are applied to membership functions. Note that only membership functions for low and middle speed are used for the selection of gear or belt-pulley type. All seven rules are applied to the membership functions. Fig. 9 shows the implication process for rule 4. Torque, spindle speed, and cost/accuracy are fuzzified as 0.133, 0.666, and 0.3, respectively. Since the minimum operator is used for implication, the minimum fuzzified number 0.133 is applied to the membership function for gear/belt-pulley type.

Fig. 10 shows the whole process of fuzzy logic. After implicating all seven rules, seven membership functions are obtained as shown on the right hand side of Fig. 10. These seven membership functions are aggregated and a final membership function is obtained, as shown in the lower right of Fig. 10. From the final membership function, the centroid point is computed and the value of the lateral axis is taken as a defuzzified number.
The final defuzzified value $GB$ becomes ‘4.19’, as shown in Fig. 10, and the transmission type is determined with the following selection rule:

IF $GB \geq 5$ THEN transmission is belt-pulley type
ELSE IF $GB \geq 5$ THEN transmission is gear type

Therefore, the gear type is chosen for the transmission since $GB$ is equal to 4.19.

### 3. Optimization of bearing locations

In order to apply the optimization to the spindle design, objective and design variables are established. Chatter vibration is an important issue for machine tools since it may lead to spindle, cutter and part damages.

There are significant number of parameters in a typical spindle design process, such as the dimensions of the spindle shaft, housing, and collars. However, the most effective design parameters need to be selected to optimize the spindle design in practice. There are numerous constraints on the geometric design of spindle parts, and design dimensions which are usually coupled with each other. For example, if the diameter of the spindle shaft changes, the bore diameter of the housing also has to be changed, where more parameters need to be taken into account which may lead to a convergence problem in optimization algorithm. Since the objective function is highly non-linear, the Sequential Quadratic Programming (SQP) method is used in

### Table 3
Fuzzy rules for transmission selection (gear/belt-pulley)

<table>
<thead>
<tr>
<th>Torque</th>
<th>Spindle speed</th>
<th>Cost /Accuracy</th>
<th>Gear/Belt</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 IF</td>
<td>Small AND Middle AND Low</td>
<td>High accuracy Low cost</td>
<td>Gear</td>
</tr>
<tr>
<td></td>
<td>Large</td>
<td>Low</td>
<td>Belt</td>
</tr>
<tr>
<td>2 IF</td>
<td>Small AND Middle AND Low</td>
<td>High accuracy Low cost</td>
<td>Gear</td>
</tr>
<tr>
<td></td>
<td>Large</td>
<td>Low</td>
<td>Belt</td>
</tr>
<tr>
<td>3 IF</td>
<td>Middle AND Middle AND Low</td>
<td>High accuracy Low cost</td>
<td>Belt</td>
</tr>
<tr>
<td></td>
<td>Large</td>
<td>Low</td>
<td>Gear</td>
</tr>
<tr>
<td>4 IF</td>
<td>Small AND Middle AND Low</td>
<td>Low cost</td>
<td>Gear</td>
</tr>
<tr>
<td></td>
<td>Large</td>
<td>Low</td>
<td>Belt</td>
</tr>
<tr>
<td>5 IF</td>
<td>Small AND Middle AND Middle</td>
<td>High accuracy Low cost</td>
<td>Belt</td>
</tr>
<tr>
<td></td>
<td>Large</td>
<td>Low</td>
<td>Gear</td>
</tr>
<tr>
<td>6 IF</td>
<td>Middle AND Middle AND Middle</td>
<td>High accuracy Low cost</td>
<td>Belt</td>
</tr>
<tr>
<td></td>
<td>Large</td>
<td>Low</td>
<td>Gear</td>
</tr>
<tr>
<td>7 IF</td>
<td>Middle AND Middle AND Middle</td>
<td>High accuracy Low cost</td>
<td>Belt</td>
</tr>
<tr>
<td></td>
<td>Large</td>
<td>Low</td>
<td>Belt</td>
</tr>
</tbody>
</table>

Fig. 9. Implication process for rule 4.
the optimization of the spindle design. The iterative
optimization operations can be expressed as the following
equation

\[ x_{k+1} = x_k + \alpha^* d \quad (8) \]

where \( k \) is the iteration number, \( x_{k+1} \) is the new design
variable vector, \( x_k \) is the current design variable vector, \( d \) is a
vector search direction, and \( \alpha^* \) is the scalar quantity that
defines the distance moving in direction \( d \). The optimization
algorithm used in this paper is shown in Fig. 11.

Objective function. The cutting conditions, the depth of
cut and spindle speed, must be under the stability lobes in
order to avoid chatter vibrations in metal cutting [16].
The location of stability pockets, the lobes, is dependent on
the natural frequencies of the spindle system, and the
allowable depth of cut depends on the dynamic stiffness of
the modes. This paper proposes automated tuning of the
spindle modes in such a way that chatter vibration free
pockets of stability is created at the desired spindle speed
and depth of cut. The tuning of the spindle dynamics is
achieved by optimizing the distribution of bearings along
the spindle shaft.

When the critical depth of cut of the stability lobes at the
cutting spindle speed \( (a_{Clim}) \) (Fig. 12) is maximized,
the cutting is the most stable. Therefore, the objective function \( f_{ob} \) is simply defined as follows

\[
\text{Minimize} : \quad f_{ob} = \sum_{i=1}^{N} W_i(a_{\text{Climb}}), \quad i = 1, 2, \ldots, N_f
\]

where \( W_i \) and \( a_{\text{Climb}} \) are the weight and critical depth of cut for the \( i \)th cutter, respectively, and \( N_f \) is the total number of cutters with different flutes.

In order to calculate \( a_{\text{Climb}} \), the Frequency Response Function (FRF) at the tool tip is required, which is evaluated by Timoshenko Beam based Finite Element model of the spindle.

**Design variables.** The design variables for the chatter free spindle design are the bearing spans. The number of the design variables depends on the bearing arrangement and the transmission type. Fig. 13 shows the design variables of the motorized spindle with five bearings (four front and one rear bearing). The \( x(1) \) to \( x(6) \) are the design variables which define the bearing locations. The \( x \) is determined automatically with the subtraction of the other known length from the total spindle length.

**Initial conditions.** In order to prevent optimization to converge to one local minimum as opposed to a global minimum, three different initial conditions are used and the final design is chosen from the three emerging optimal solutions.

**Calculation of FRF at the tool tip.** In order to compute the FRF at the tool tip, which is required for the chatter stability analysis, the tool holder and cutter are taken into account in the specifically developed FE model where the damping ratio of each mode is set to 0.03, which is typical in the mechanical system. The system uses an in house developed FE algorithm based on standard Timoshenko Beam model.

By using the FE method the following forced vibration equation can be obtained for the spindle system:

\[
[M] \ddot{x} + [C]x + [K]x = F
\]

In modal coordinates, it can be expressed as,

\[
[\ddot{\mathbf{y}}] + [\mathbf{C}]\mathbf{y} + [\mathbf{K}]\mathbf{y} = \mathbf{F}
\]

where \( \mathbf{M} = P^T M P \), \( \mathbf{C} = P^T C P \) and \( \mathbf{K} = P^T K P \) are the modal mass matrix, damping matrix, and stiffness matrix, respectively. These are all \( m \times m \) diagonal matrices; \( \mathbf{F} = P^T F \) is modal force; \( \mathbf{x} = P \mathbf{y} \), where \( P \) is mode shape which is an \( n \times m \) matrix.

Eq. (10) is decoupled as an SDOF system

\[
m_i \ddot{y}_i + c_i \dot{y}_i + k_i y_i = \ddot{f}_i, \quad i = 1, \ldots, m
\]

where \( m_i, c_i, k_i \) are the diagonal elements of matrices \( \mathbf{M}, \mathbf{C}, \mathbf{K} \), respectively.

The transfer function for Eq. (12) is

\[
G_i(s) = \frac{y_i(s)}{f_i(s)} = \frac{1/m_i}{s^2 + 2\zeta_i \omega_{ni}s + \omega_{ni}^2}, \quad i = 1, \ldots, m
\]

where \( \zeta_i = c_i/(2m_i \omega_{ni}) \) is the damping ratio for the \( i \)th mode; \( \omega_{ni} = \sqrt{k_i/m_i} \) is the \( i \)th natural frequency.

From Eq. (13)

\[
y_i(s) = G_i(s)\ddot{f}_i(s), \quad i = 1, \ldots, m
\]

In matrix form, Eq. (14) becomes

\[
\mathbf{Y}(s) = \mathbf{G}(s)\mathbf{F}(s)
\]

Due to \( \mathbf{X}(s) = \mathbf{PY}(s) = \mathbf{PG}(s)\mathbf{F}(s) = \mathbf{PG}(s)\mathbf{P}^T \mathbf{F}(s) \), the transfer function for the spindle system is obtained as follows:

\[
\mathbf{H}(s) = \frac{\mathbf{X}(s)}{\mathbf{F}(s)} = \mathbf{PG}(s)\mathbf{P}^T
\]

The element in the transfer function matrix \( \mathbf{H}(s) \) is

\[
h_{jk} = \frac{x_j(s)}{F_k(s)} = \sum_{i=1}^{m} P_{ji} P_{ki} G_i(s) = \sum_{i=1}^{m} \frac{u_j u_k}{s^2 + 2\zeta_i \omega_{ni}s + \omega_{ni}^2}
\]

where \( u_j \) is the element of the mass normalized mode shape \( u \).

**Calculation of critical depth of cut for chatter stability** \((a_{\text{Climb}})\), the milling chatter stability theory developed by Budak–Altintas [16] is used to evaluate critical depth

\[
a_{\text{lim}} = -\frac{2\pi A_R}{NK_c} \left(1 + \left(\frac{A_1}{A_R}\right)^2\right)
\]

where \( K_c \) is the cutting coefficient, and \( N \) is the number of teeth. \( A_R \) and \( A_1 \) are real and imaginary values of an eigenvalue \( \Lambda \), which is obtained using the FRF at the tool tip. The details of the chatter stability theory can be found in [11,16].
4. Application of the expert spindle design system

The proposed system is demonstrated against a commercially existing machine tool (Mori Seiki SH-403) as shown in Fig. 14 for comparison. The main spindle specifications of SH-403 are shown in [17]. The spindle has a motorized transmission with oil–air type lubrication with four bearings at the front and one at the rear. The maximum spindle speed is 20,000 rpm and the power and torque properties of the spindle motor are set from the data shown in [17]. It is assumed that the user wishes to use the machine predominantly in cutting aircraft parts made from Al7075-T6 with a four-fluted end mill with a desired depth of cut of 3 mm and 15,000 rpm spindle speed. The most common cutting conditions are listed in Table 4.

From the power and torque diagram of the spindle motor of SH-403, the motor torque \( T_{\text{max}} \) at the cutting speed 15,000 rpm is found to be as 8.8 N m. On the other hand, the cutting torque \( T_{\text{mo}} \) required for the cutting conditions shown in Table 4 can be calculated as 4.796 N m. The maximum motor rotation speed \( n_{\text{max}} \) is 20,000 rpm and the desired speed \( n \) is 15,000 rpm. Therefore, \( T_{\text{max}} \leq T_{\text{mo}} \) and \( n \leq n_{\text{acmax}}(= n_{\text{max}}) \), so either the direct coupling type or the motorized type are the possible transmission type.

In order to select the transmission type from either the direct coupling type or the motorized type, ‘Spindle Speed’, ‘High Dynamic Stiffness vs. Low Balancing Vibration’, ‘Low Thermal Effect vs. Small Noise’, and ‘Low Replacement Operation Cost vs. Low Replacement Parts Cost’ need to be input. The spindle speed is automatically set from the maximum motor speed. In this case, the numbers are assumed from the concepts of the machine written in the SH-403 catalog [17]. The fuzzy weight numbers are set

Table 4
Most common cutting conditions for SH-403

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cutter diameter</td>
<td>19.05 mm</td>
</tr>
<tr>
<td>Number of flutes</td>
<td>4</td>
</tr>
<tr>
<td>Material to be cut</td>
<td>Al7075-T6</td>
</tr>
<tr>
<td>Cutting spindle speed</td>
<td>9000 rpm</td>
</tr>
<tr>
<td>Depth of cut</td>
<td>3 mm</td>
</tr>
<tr>
<td>Width of cut</td>
<td>19.05 mm (slotting)</td>
</tr>
<tr>
<td>Feed rate</td>
<td>0.1 mm flute</td>
</tr>
</tbody>
</table>

Fig. 14. Spindle configuration of SH-403.

Table 5
Selection Results for Mori-Seiki SH-403

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Spindle name</th>
<th>Actual design</th>
<th>Expert system</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case 1</td>
<td>Mori Seiki SH-403 [17]</td>
<td>T Motorized</td>
<td>Motorized</td>
</tr>
<tr>
<td></td>
<td></td>
<td>L Oil air</td>
<td>Oil air</td>
</tr>
<tr>
<td></td>
<td></td>
<td>L Oil air</td>
<td>Oil air</td>
</tr>
<tr>
<td>Case 3</td>
<td>Weiss Spindle MAL UBC Version</td>
<td>T Belt-pulley</td>
<td>Belt-pulley</td>
</tr>
<tr>
<td></td>
<td></td>
<td>L Grease</td>
<td>Gear</td>
</tr>
<tr>
<td></td>
<td></td>
<td>L Oil Air</td>
<td>Oil Air</td>
</tr>
<tr>
<td></td>
<td></td>
<td>L Grease</td>
<td>Grease</td>
</tr>
</tbody>
</table>

T, transmission; L, lubrication; D. coupling, direct coupling
as ‘3’, ‘8’, and ‘4’, respectively. Similarly, the fuzzy weight of the lubrication system is set.

Table 5 shows the results of the spindle components’ selection. The Expert Spindle Design System selected the proper transmission and lubrication type.

Table 6 shows the list of agreements between the actual design and the design that uses this expert system. The results attained via the expert system match those of the actual design in all five cases. Therefore, the proper spindle components can be selected with the proposed expert
system, that is, the rule base and the membership functions of this system are defined properly.

4.1. Optimization for bearing locations

The bearing locations are optimized with the same cutting conditions shown in Table 4. The design constants and bearing parameters are set suitably as shown in Fig. 15, which are not the dimensions of the Mori Seiki SH-403.

The bearing spans’ constraints are set by considering the spaces of lubrication devices as shown in Fig. 16.

Three types of initial bearing spans are set without violating the constraints. The following three spans are used as the initial spans, as shown in Fig. 17.

The SQP is used to optimize the bearing spans. The gradients objective function $f_{ob}$ and Hessian matrix cannot be calculated analytically. They are therefore obtained using a numerical differentiation and Quasi-Newton methods, respectively. The minimum change in the design variables for finite difference gradients is set to 0.01 mm so that the first partial derivative $\nabla f_{ob}$ is derived by subtracting 0.01 mm from the bearing spans. The BFGS approximation technique is used to update the Hessian matrix. The termination tolerance of the objective function values is set to 0.0001. In the case that the optimized value difference between the previous and current values is less than 0.0001, the calculation stops computing and the final value is shown as the optimized value.

Fig. 18 shows the chatter stability lobes for a four flute cutter computed from the three initial designs shown in Fig. 17 and the final optimized design. The desired spindle speed is 9000 rpm, and the depth of cut is 3 mm. The cutting is not stable for all three initial designs, but it becomes stable after optimization. The optimized spindle configuration is shown in Fig. 19.

Multiple cutters with different flutes can also be optimized for the same spindle. The same spindle and cutting conditions described above are used, but three cutters with two, three, and five flutes are used on the spindle. Only the first initial design is considered here. Fig. 20a and b show the stability lobes for the original and optimized designs, respectively. In the original design, the cutting is not stable for the five-fluted cutter, and it is close to the unstable regions for two and three fluted cutters. For the optimized design, there are very big
5. Conclusions

This paper presents an Expert Spindle Design system for machine tool engineers. It proposes an alternative method to the present design practice, which is based on the past experience of individual designers, while attempting to eliminate costly trials by using the laws of machine design, solid mechanics, and metal cutting dynamics in an integrated fashion.

The design configurations and membership functions are stored in a knowledge base using sets of design rules based on the past experience and laws of cutting mechanics. Fuzzy logic is used as an inference engine in the proposed expert system. The fuzzy logic can deal with design uncertainties such as high, medium and low speeds or large/small torque required from the spindles where the exact numerical values are difficult to set rigidly by the designers. The membership functions can be updated by the designers as the rules change due to technological advances in industry. The expert system leads to automatic generation of spindle configuration which includes drive shaft, motor type and size, transmission mechanism between the motor and shaft, and tool holder style.

While the configuration and sub-components of the spindle are based on the torque, power, and speed requirements from the machine tool, the exact locations of the bearings must be determined based on the chatter vibration stability of the spindle. The paper proposes a bearing spacing optimization strategy for the spindles configured by the expert system or designed by the engineers. The designer provides initial estimates of the bearing locations including constraints. The configured spindle is analyzed by a proposed Finite Element Analysis (FEA) algorithm based on Timoshenko Beam elements. The Frequency Response Function (FRF) of the spindle at the tool tip is obtained by the modal analysis module of the FEA algorithm. The bearing locations are optimized iteratively until the designed spindle satisfies chatter vibration free cutting constraints. Sequential Quadratic Programming (SQP) is used as an optimization method in identifying optimal bearing locations. The proposed expert spindle design strategy is demonstrated in designing several industrial size spindles used in industry.