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ORIGINAL ARTICLE

A CAD/CAE-integrated structural design framework for machine tools

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Abstract In this paper, a novel integrated framework for design and optimization of a machine tool structure is presented, which can greatly improve the design quality and efficiency by combining knowledge-based design and multi-stage optimization with the CAD/CAE integration technique. To realize this framework, a topology architecture model has been developed to integrate the configuration design and geometric modeling knowledge as well as the static and dynamic evaluation knowledge of machine tools with a specific topology architecture type; an analysis feature model is proposed for the integration between commercial CAD and CAE software, in which analysis features can be automatically converted to a script code for finite element analysis (FEA) through feature mapping. Based on the topology architecture model and feature-based CAD/CAE integration methodology, a two-stage design optimization process is proposed to perform the conceptual structural design of machine tools. In the first stage, the principal parameters which critically affect the performance of an entire machine are determined; then, the static and dynamic stiffness matching designs are performed to obtain the reasonable stiffness and weight of each structural part and functional component based on the stiffness model and dynamic model. In the second stage, the arrangement of ribs is determined by inferring the design knowledge; FEA is used to evaluate the performances of structural parts, and the response surface method (RSM) is applied to optimize the structural parameters to approach the stiffness and mass close to the allocated values obtained from the first stage. Re-design of a four-axis horizontal machining center with a box-in-box architecture was carried out to illustrate the design procedure in detail and to verify the feasibility and efficacy of the proposed framework. By applying the proposed framework, the total weight of the entire machine is minimized while sufficient stiffness is maintained. The results also show that the proposed framework facilitates the conceptual structural design and optimization process of machine tools.

Keywords CAD/CAE integration · Topology architecture model · Configuration design · Structure optimization · Machine tool

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Abbreviations

DZ_{span}	Span distance of the pair of Z-axis ballscrews
DX_{table}	Dimension of worktable in the X direction
PZ_{span}	Span coefficient of the pair of Z-axis ballscrews
DZ_{screw}	Length of Z-axis ballscrew
DZ_{table}	Dimension of worktable in the Z direction
DZ_{stroke}	Stroke of the machine tool in the Z direction
PZ_{screw}	Design margin of Z-axis ballscrew
DZ_{rail}	Dimension of the Z-axis guideway
PZ_{rail}	Design margin of Z-axis guideway
DZ_{mt}	Dimension of the machine tool in the Z direction

DZ_{col}	Dimension of column in the Z direction
DZ_{bed}	Dimension of bed in the Z direction
$k_{mtx}, k_{mty}, k_{mtz}$	Stiffness values of the machine tool in the X, Y, and Z directions
k_{tx}, k_{ty}, k_{tz}	Stiffness values at the tip of the cutting tool in the X, Y, and Z directions
k_{wx}, k_{wy}, k_{wz}	Stiffness values of the workpiece in the X, Y, and Z directions
$\boldsymbol{M}$	Mass matrix
$\boldsymbol{K}$	Stiffness matrix
$\boldsymbol{C}$	Damping matrix
$\boldsymbol{f}$	Excitation force vector
$\ddot{\boldsymbol{x}},\ \dot{\boldsymbol{x}},\ \boldsymbol{x}$	Acceleration vector, velocity vector, and displacement vector of structure node
f_1	First natural frequency
K_{dynamic_X}, K_{dynamic_Y}, K_{dynamic_Z}	Dynamic stiffness of the machine tool in the X, Y, and Z directions
n_p	Total number of structural part and function part
m_i	Mass of structural part or function part
M_i	Mass of structural part of a successful design case
Y	Design response
x_i, x_j	Design variable
a_0	Constant in response function
a_i	Coefficient of linear segment
a_{ii}	Coefficient of quadratic segment
a_{ij}	Interaction coefficient of linear segment
n_v	Number of design variables
ε	Statistical error in response surface model
$K_X,\ K_Y,\ K_Z$	Stiffness of structural part in the X, Y, and Z directions
m_{str}	Mass of structural part
$d_i,\ T_w$	Thickness of rib and side wall
D_r	Dimension of lightening hole
n_r	Total number of the region of column
$DX_{\mathrm{mt}},\ DY_{\mathrm{mt}},\ DZ_{\mathrm{mt}}$	Dimension of the machine tool in the X, Y, and Z directions
$DX_{\mathrm{bed}},\ DY_{\mathrm{bed}},\ DZ_{\mathrm{bed}}$	Dimension of bed in the X, Y, and Z directions
$DX_{\mathrm{col}},\ DY_{\mathrm{col}},\ DZ_{\mathrm{col}}$	Dimension of column in the X, Y, and Z directions
$DX_{\mathrm{mf}},\ DY_{\mathrm{mf}},\ DZ_{\mathrm{mf}}$	Dimension of moving frame in the X, Y, and Z directions
$DX_{\mathrm{spb}},\ DY_{\mathrm{spb}},\ DZ_{\mathrm{spb}}$	Dimension of spindle box in the X, Y, and Z directions
$DX_{\mathrm{st}},\ DY_{\mathrm{st}},\ DZ_{\mathrm{st}}$	Dimension of sliding table in the X, Y, and Z directions
$K_{\mathrm{bs}x},\ K_{\mathrm{bs}y},\ K_{bsz}$	Axial stiffness of the X-, Y- and Z-axis ballscrew
$K_{\mathrm{gxn}},\ K_{\mathrm{gxt}}$	Normal and tangential stiffness of X-axis guideway
$K_{\mathrm{gyn}},\ K_{\mathrm{gyt}}$	Normal and tangential stiffness of Y-axis guideway
$K_{\mathrm{gzn}},\ K_{\mathrm{gzt}}$	Normal and tangential stiffness of Z-axis guideway
$K_{\mathrm{sp}x},\ K_{\mathrm{sp}y},\ K_{\mathrm{spz}}$	Stiffness of spindle in the X, Y, and Z directions
$K_{\mathrm{wt}x},\ K_{\mathrm{wt}y},\ K_{\mathrm{wtz}}$	Stiffness of worktable in the X, Y, and Z directions
$A_X,\ A_Y,\ A_Z$	Amplitude of dynamic response curve in the X, Y, and Z directions
$C_X,\ C_Y,\ C_Z$	Maximum dynamic compliance in X, Y, and Z directions
m_{mt}	Mass of the entire machine tool

1 Introduction

Machine tool structure is one of the critical factors to maintain machining speed, precision, and productivity. Mechanical structure not only provides the support and accommodation for all the machine components but also contributes to static and dynamic performances [1, 2]. Therefore, the design of a high-stiffness and lightweight machine tool structure has been one of the major objectives in machine tool design, especially for high-precision machine tools [3].

However, in the current structure design process of machine tools, design activities rely heavily on the experience of engineers. For instance, some principal parameters which critically affect the performance of the machine are manually determined according to the experience of the design engineer. The parametric CAD model and finite element (FE) model of the machine tool structure are then established utilizing commercial CAD and CAE software through an interactive manner. Based on the structural analysis results, it is determined by experience again whether or not to improve the performance by changing the structural parameters. The processes will be repeated until satisfaction. It is quite clear that current design practice is inefficient. Moreover, for the experience-based or cut-and-try approaches, it is difficult to find an optimized entire machine structure of which the total weight is minimized while sufficient stiffness is maintained. Note that the machine tool is comprised of a number of structural parts and functional units which form a unified whole. So actually, even though each structural part is optimized individually, it may be of little use in improving the performance of the entire machine [4]. The reasons leading to the current status are the lack of an effective method which could conduct the static/dynamic stiffness matching design in the early design stage to guide the design optimization of structural parts, as well as a CAD/

CAE-integrated framework to support the top-down structure design procedure of the machine tool.

The structure design of a machine tool requires complicated and diversified design knowledge. Systematic management and reuse of this design knowledge can improve design efficiency and quality. Many knowledge integration [5–7] and reusing [8–12] methods have been developed for structure design in recent years. For instance, Lee et al. [13] defined four categories of design knowledge: equations, if-then rules, rules for multi-criteria decision-making, and formalized data. Then, according to them, they developed a hybrid inference architecture, which involves huge and complex knowledge of machine tools and makes the inference process effective. By systematically summarizing the expert experience, Park and Sohn [14] expressed the configuration knowledge of a machine tool in the forms of a set of formulae, e.g., stroke-decision formulae, size-decision formulae, and position-decision formulae. Liu et al. [15] proposed a knowledge-centric process management framework, in which the relevant knowledge for machine tool development is classified into six types: process knowledge, configuration knowledge, components knowledge, development standard knowledge, instance knowledge, and software application knowledge. The core issues of the configuration design of a machine tool are the selection of an appropriate type of machine tool based on the user requirements and the determination of principal design parameters [14, 16]. By systemically organizing the skeleton elements which involve datum planes, curves, sketches, parameters, equations, and rules, it is possible for the skeleton model to precisely specify the spatial arrangements of components, interfaces between components, geometrical constraints of components, dimensions of components, and kinematics of components [17]. By this manner, the configuration design knowledge and the fundamental CAD structure of the machine tool with a specific topology architecture type can be embedded in the skeleton model and can be reused in the early design stage.

High machining accuracy requires high static stiffness of the machine tool structure. It is supposed to have guiding significance for structural design of the machine tool through establishing an effective stiffness model and realizing stiffness analysis in the entire workspace. For this purpose, scholars put forward many stiffness modeling and design methods [18, 19]. For instance, Yan et al. [20–22] proposed a semi-analytic method based on a multi-axis system closed-chain stiffness model, which can be utilized to describe the integrated stiffness performance of the entire machining system. Huang et al. [23] analyzed the stiffness of machine tools by using the finite element method (FEM), from which the impacts of various components on the stiffness of entire machine were determined. Note that, in those methods, stiffness modeling and analysis are performed after the structure design. That is, they are done based on the existing CAD model of the machine tool structure. Hence, they can only be used to analyze and optimize the existing machine tool structure, but not to guide the design of it. Given that, once the design requirements of entire machine cannot be reached, structural modifications must be undertaken repeatedly to achieve the desired goals. In order to solve the above-mentioned problems, Shi et al. [24] proposed a stiffness modeling method and a stiffness matching design method of the entire machine in the early design stage. By using the proposed method, the stiffness of each component is allotted precisely, which avoids excessive or insufficient stiffness values of a component. The allotted stiffness can be used to guide the design and optimization of structural parts and the selection of functional units.

Dynamic performance is another important factor to be considered for structure design of a machine tool. Modal analysis determines the fundamental vibration mode shapes and its corresponding frequencies. Moreover, natural frequency is an evaluation index of structural dynamic stiffness [25]. In conventional precision machine tool design, the first natural frequency is expected to be higher than the machine operation frequency [26]. The harmonic response analysis is able to verify whether or not the designs will successfully overcome resonance and the harmful effects of forced vibrations [27]. Generally, there are two kinds of dynamic design and analysis methods: FEM [28] and lumped parameter method (LPM) [29]. Compared with LPM, FEM provides higher accuracy, but a relatively completed CAD model is needed to build the FE model. Therefore, it is not suitable for conceptual design due to the fact that the completed geometric model of the machine tool may not yet be available in this stage. In contrast, LPM needs fewer structural parameters which can be obtained from the conceptual model to accomplish dynamic analysis, though it has less accuracy than FEM. Therefore, it can be applied to analyze and estimate the dynamic performance of machine tool structure in the early design stage. Based on LPM, parametric dynamitic models of the machine tool with a specific topology architecture type can be established, which can be used to optimize machine configuration through the analysis of the vibration characteristic and energy distribution [14].

For the structure design of a machine tool, an integrated CAD and CAE system which could complete every design cycle effectively is desirable. Most of the current commercial CAD and CAE software packages are still stand-alone systems which are not suitable for product development in the collaborative environment. Some of the major issues involved in CAD and CAE integration are summarized in ref. [30]. Efforts have been made to solve these problems. Approaches can roughly be classified into four categories, i.e., the CAE-centric integration [31, 32], the CAD-centric integration [33–35], the feature-based integration [36–40], and the unified CAD and CAE model-based integration [30, 41]. In the CAD-centric integration and the CAE-centric integration, parametric scripting languages provided by the commercial CAE systems are usually employed to integrate the information between CAD and CAE environments, which can automate the FE modeling and analysis. The merit of these approaches lies in the fact that it is easy to accomplish the automation of the structural optimization loop. However, they have a poor bidirectional association between the CAD model and the CAE model. In contrast, for the feature-based integration and a unified CAD and CAE model-based integration, it is not only easy to accomplish the bidirectional association between the design model and the analysis model as well as the automation of the design process but also convenient to integrate the design and analysis knowledge. Note that there are various complicated loads, boundary conditions, and interfaces at connection joints which are involved in the FEA of the machine tool structure. This means that the structure analysis of machine tools needs complicated analysis modeling knowledge. Especially for the interface, as a weak link between components, it greatly affects the overall stiffness and mechanical characteristics of the assembled structure [42]. To make the model realistic and avoid the complexity in mesh generation, appropriate simplification of interfaces and geometries of connection joints is essential, through which it could increase the efficiency of preparing the analysis model, without affecting the accuracy of the results [28]. Therefore, an analysis feature model which is specially designed for a machine tool is used to integrate the CAD and CAE system in this paper. It extracts and maintains all the semantic parameters required for building the FE model and manages the process of engineering analysis with the assistance of design and analysis modeling knowledge.

In this paper, a CAD/CAE-integrated framework for structural design of a machine tool is proposed. A topology architecture model has been developed to support the top-down structure design and static/dynamic stiffness matching design of machine tools. An analysis feature model and a feature mapping method are employed to integrate the CAD and CAE system and realize the automation of the FE modeling and analysis. Miscellaneous and complicated engineering knowledge is embedded into the software package alone with the design process, predefined model, and design knowledge base. A two-stage design optimization process for the structure design of a machine tool is proposed. In the first stage, the principal parameters which critically affect the performance of the entire machine, including the static stiffness and the weight of structural parts, are determined. In the second stage, knowledge-based reasoning and the RSM-based optimization method are employed; thus, optimized structural parts which can guarantee the static and dynamic performance of the entire machine are obtained.

The rest of this paper is organized as follows. The integrated framework for the structural design of machine tools is proposed in Section 2. Section 3 systematically introduces the topology architecture model. The proposed feature-based CAD/CAE integration approach is presented in Section 4. The two-stage design optimization process of the machine tool structure is described in Section 5. In Section 6, re-design of a horizontal machining center is chosen as a case study to illustrate the feasibility and efficacy of the proposed framework. Finally, a brief summary and some directions for future work are given in Section 7.

2 Integrated framework for structure design of machine tool

Figure 1 illustrates the architecture of the proposed integrated framework which consists of a structure design module, a structure analysis module, and a structure optimization module. Each module is introduced as follows.

The structure design module includes four submodules. Configuration design module employs the skeleton model and constraint-solving as well as rule-based reasoning method, thus produces a conceptual design model of machine tool structure. The stiffness and dynamic design module gets the principal parameters from the skeleton model. Based on the stiffness model and the dynamic model, the static and dynamic stiffness matching design of machine tool can be performed to obtain the reasonable stiffness and mass values of the structural parts and the functional units. Then the functional unit selection module uses the obtained results to choose the functional units and standard parts. The structural parts design module employs design templates and knowledge interference method to produce the parametric CAD model of structural parts.

The structure analysis module supports the structure design module and the structure optimization module, and its functions are to validate and evaluate the static and dynamic performance of the designed structure. By exploiting the analysis feature model and feature mapping method, the FE modeling and analysis of the entire machine and structural parts can be automated.

The structure optimization module employs the RSM-based optimization method to find the best parameters of structural parts to approach the stiffness and mass values obtained in the stiffness and dynamic design module. As an optimizing task manager, the structure optimization module is responsible for exchanging data between the structure design module and the structure analysis module, updating the CAD model, rebuilding the FE model as well as controlling the optimizing process and carrying out the algorithm.

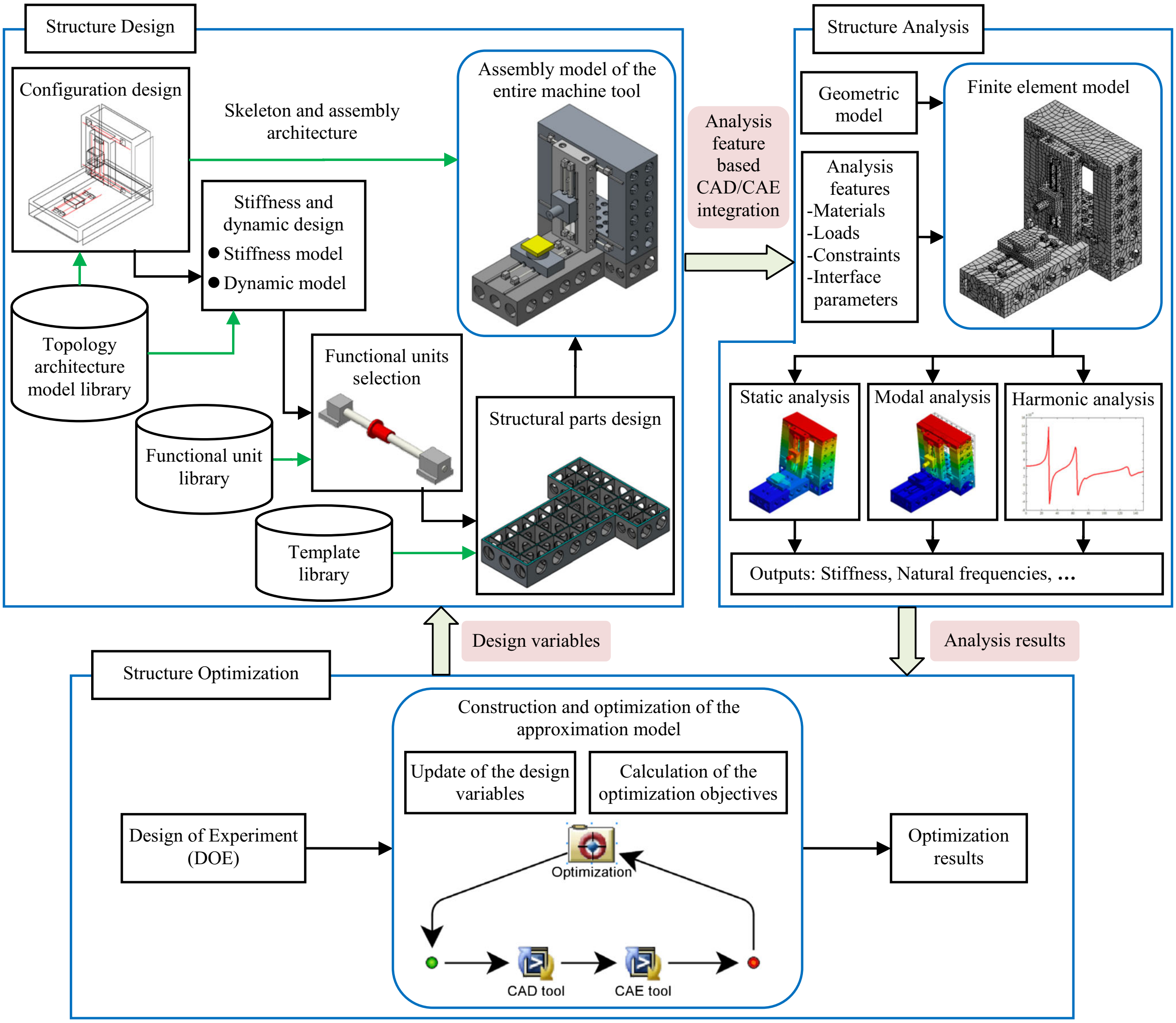

Fig. 1 Architecture of integrated structure design framework for machine tool

3 Topology architecture model

The top-down design method is essentially the breaking down of a system to elucidate its compositional sub-systems [43], which mainly includes the conceptual design stage and the detailed design stage [44]. In the conceptual design stage [45], decisions are made for the definition of functional requirements, selection and location of parts, and definition of the relations between parts. Most of the important factors including the overall characteristics and the principal parameters of a product are determined in the conceptual design stage. This means a conceptual model should be established firstly and the knowledge that drives design and modeling should be embedded into it with the help of CAD tools.

In order to support the structure design of the entire machine tool in the early design stage, a topology architecture model is proposed to integrate the geometric modeling knowledge as well as static and dynamic evaluation knowledge. As shown in Fig. 2, the topology architecture model of a machine tool consists of a skeleton model, a stiffness model, and a dynamic model. The skeleton model contains all the critical design parameters as well as the fundamental CAD structure of the machine tool with a specific topology architecture type. The stiffness model and the dynamic model, which are established according to the topology architecture of the machine tool, are used to evaluate the static and dynamic performance in the conceptual design stage. Once the basic layout dimensions of the skeleton model are determined, the static and dynamic stiffness matching design is performed to obtain the reasonable stiffness and mass values of each structural part and functional unit, which can be used to guide the design of structural parts and the selection of functional units.

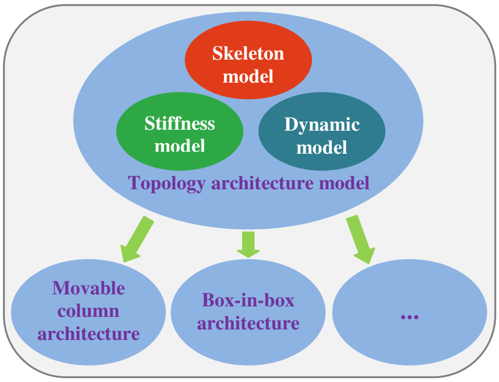

Fig. 2 Topology architecture model

3.1 Skeleton model

By systemically organizing the skeleton elements, especially the geometric entities including datum planes, datum axes, and sketches, a skeleton model can be used to express the architecture of the entire machine and the topologic connecting relationships between components, e.g., structural parts and functional units. Meanwhile, by adding non-geometric entities such as parameters, equations, and rules, it can precisely specify the space layout, the principal sizes and geometric constraints of components, and the interfaces between components [46]. Therefore, a completely defined skeleton model can effectively integrate configuration design knowledge of the machine tool with the specific topology architecture type.

Figure 3 demonstrates the skeleton model of a horizontal machining center with a box-in-box architecture. Figure 3b shows the skeleton model of the *Z*-axis ballscrew subassembly. In the skeleton model, plane *P_Bed_Up* represents the up assembly position of the bed. Sketch *S_Table* and plane *P_Table* represent the basic shape and the assembly position of the worktable respectively. Sketch *S_Shaft* and planes *P_Nut*, *P_Bearing_Front*, *P_Bearing_Rear*, *P_Seat_Front*, and *P_Seat_Rear* represent the assembly of the ballscrew; besides, planes *P_Nut* and *P_Table* specify the assembly relationship between the nut and the worktable; plane *P_Nut* also specifies the spatial position of the nut; furthermore, planes *P_Seat_Front* and *P_Seat_Rear* specify the spatial position of the front and the rear bearing seats and the assembly relationship with the bed. By changing the position of *P_Nut*, it is capable of performing the motion simulation of the worktable and the interference checking of the machine tool.

Parameter DZ_{span} in Fig. 3b is a function of the dimension of the worktable:

$$DZ_{span} = DX_{table} \cdot PZ_{span} \tag{1}$$

where the value of parameter PZ_{span} is 0.6–0.9 obtained by interviewing experienced engineers.

Similarly, the length of the *Z*-axis ballscrew can be obtained by the equation pre-built in the skeleton model:

$$DZ_{screw} = DZ_{table} + DZ_{stroke} + 2PZ_{screw} \tag{2}$$

where parameter PZ_{screw} is 200–500 mm. Parameters DZ_{table} and DZ_{stroke} are obtained by inference based on machining requirements of the workpiece, while PZ_{screw} is obtained by interviewing experienced engineers.

Other principal parameters in the *Z* direction of the machining center are given as Eqs. (3)–(5).

$$DZ_{rail} = DZ_{table} + DZ_{stroke} + 2PZ_{rail} \tag{3}$$

$$DZ_{mt} = DZ_{rail} + DZ_{col} + PZ_{mt} \tag{4}$$

$$DZ_{bed} = DZ_{mt} \tag{5}$$

where parameters PZ_{rail} and PZ_{mt} are both 500–1000 mm. Parameter DZ_{col} is obtained from the dimension of the column in the *X* direction by the built-in formulae in the skeleton model. Parameters PZ_{rail} and PZ_{mt} are obtained by interviewing experienced engineers.

3.2 Stiffness model

In order to achieve the top-down design of the stiffness, a stiffness model and a stiffness matching design method [24] are employed in our study. To establish the stiffness model of the machine tool, the stiffness coefficient [24] is firstly introduced to characterize the stiffness of structural parts and functional units. The deformation model of the entire machine can be established based on a multi-body system theory. Based on

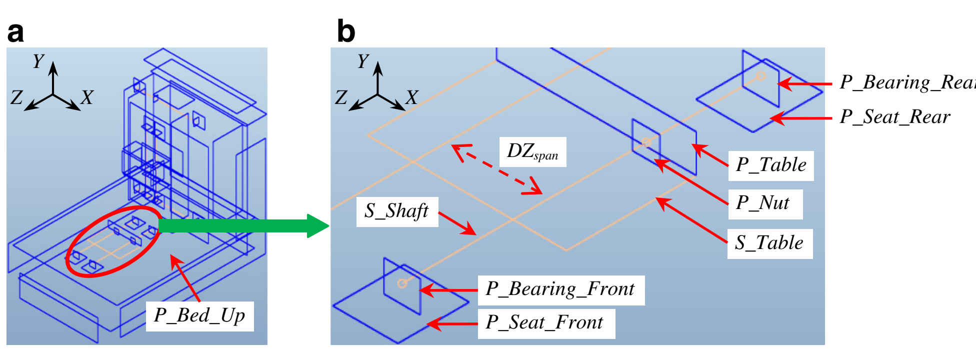

Fig. 3 Skeleton model of the horizontal machining center with a box-in-box architecture. **a** Skeleton model of the entire machine tool, **b** skeleton model of the *Z*-axis ballscrew subassembly

the simultaneous equations of the static equilibrium equations, the deformation compatibility equations, and the physical equations, the equations of the stiffness coefficients for the deformations of components are then established. The three-direction (3D) stiffness model is finally obtained by substituting the equations into the deformation model that reflects the stiffness characteristics of the machine tool. The 3D stiffness models of the machine tool with a box-in-box architecture, which is divided into the cutting tool loop and the workpiece loop, are expressed as follows [24].

$$\begin{bmatrix} \frac{1}{k_{tx}} \\ \frac{1}{k_{ty}} \\ \frac{1}{k_{tz}} \end{bmatrix} = \begin{bmatrix} \frac{1}{k_{sx}} + \frac{1}{k_{hx}} + \frac{1}{k'_{fzx}} + \frac{1}{k'_{\mathrm{ygnx}}} + \frac{1}{k'_{fxx}} + \frac{1}{k'_{\mathrm{ygtx}}} + \frac{1}{k'_{czx}} + \frac{1}{k'_{\mathrm{xgnx}}} + \frac{1}{k_{bsx}} \\ \frac{1}{k_{sy}} + \frac{1}{k_{hy}} + \frac{1}{k'_{fzy}} + \frac{1}{k'_{\mathrm{ygny}}} + \frac{1}{k_{bsy}} + \frac{1}{k'_{cyy}} + \frac{1}{k'_{\mathrm{xgty}}} + \frac{1}{k'_{czy}} + \frac{1}{k'_{\mathrm{xgny}}} \\ \frac{1}{k_{sz}} + \frac{1}{k_{hz}} + \frac{1}{k'_{fzz}} + \frac{1}{k'_{\mathrm{ygnz}}} + \frac{1}{k'_{czz}} + \frac{1}{k'_{\mathrm{xgnz}}} \end{bmatrix} \tag{6}$$

$$\begin{bmatrix} \frac{1}{k_{wx}} \\ \frac{1}{k_{wy}} \\ \frac{1}{k_{wz}} \end{bmatrix} = \begin{bmatrix} \frac{1}{k_{px}} + \frac{1}{k_{rx}} + \frac{1}{k'_{bxx1}} + \frac{1}{k'_{\mathrm{zgtx1}}} + \frac{1}{k'_{bxy1}} + \frac{1}{k'_{\mathrm{zgty1}}} + \frac{1}{k'_{bxx2}} + \frac{1}{k'_{\mathrm{zgtx2}}} \\ \frac{1}{k_{py}} + \frac{1}{k_{ry}} + \frac{1}{k'_{byy1}} + \frac{1}{k'_{\mathrm{zgny1}}} + \frac{1}{k'_{byy2}} + \frac{1}{k'_{\mathrm{zgny2}}} \\ \frac{1}{k_{pz}} + \frac{1}{k_{rz}} + \frac{1}{k_{bsz}} + \frac{1}{k'_{byz}} + \frac{1}{k'_{\mathrm{zgnz}}} \end{bmatrix} \tag{7}$$

$$\begin{bmatrix} \frac{1}{k_{mtx}} \\ \frac{1}{k_{mty}} \\ \frac{1}{k_{mtz}} \end{bmatrix} = \begin{bmatrix} \frac{1}{k_{tx}} + \frac{1}{k_{wx}} \\ \frac{1}{k_{ty}} + \frac{1}{k_{wy}} \\ \frac{1}{k_{tz}} + \frac{1}{k_{wz}} \end{bmatrix} \tag{8}$$

where Eqs. (6) and (7) are the stiffness models of the cutting tool loop and the workpiece loop, respectively, and Eq. (8) is the stiffness model of the entire machine tool obtained by combining Eqs. (6) and (7). For a detailed description of parameters used in Eqs. (6) and (7), please refer to [24]. Based on the stiffness model, the stiffness matching design can be performed to obtain the reasonable stiffness values of structural parts and functional units.

3.3 Dynamic model

Calculation of the structural dynamic behavior is necessary to ensure stable operation and proper relative displacement between the tool and the workpiece [47]. Due to the fact that details of structural parts are not yet available at the concept design stage, the lumped parameter model is employed to evaluate and optimize the dynamic behavior of the machine tool, e.g., vibration characteristic and energy distribution. The 3D parametric dynamic model [48] of a horizontal machining center with a box-in-box architecture is shown in Fig. 4. The undamped free vibration differential equation is shown in Eq. (9). The governing equation describing the motion of a multi-degree-of-freedom system with viscous damping is given by Eq. (10).

$$\boldsymbol{M}\ddot{\boldsymbol{x}} + \boldsymbol{K}\boldsymbol{x} = 0 \tag{9}$$

$$\boldsymbol{M}\ddot{\boldsymbol{x}} + \boldsymbol{C}\dot{\boldsymbol{x}} + \boldsymbol{K}\boldsymbol{x} = \boldsymbol{f} \tag{10}$$

Structural parameters of the dynamic model, as shown in Fig. 4, can be obtained from the skeleton model introduced in Section 3.1; stiffness and damping parameters of functional units are determined as follows: parameters of the guideway and the ballscrew which are selected according to the results of the stiffness match method described in Section 3.2, can be obtained from the handbook of manufacturers; the connecting stiffness and damping parameters of bolts for fixing interfaces are determined according to literature [49, 50].

Based on the parametric dynamic model, structure dynamic optimization can be performed. Design variables include the mass of the structural parts and the functional units. The objective of the optimization problem is to minimize the total mass and maximize the first natural frequency and the dynamic stiffness in X, Y and Z directions of the entire machine.

3.4 Object-oriented modeling of machine tools

Object-oriented programming (OOP) may be regarded as a collection of interacting objects, as opposed to the conventionally procedure-oriented programming [51]. In the OOP, each object is capable of receiving messages, processing data, and

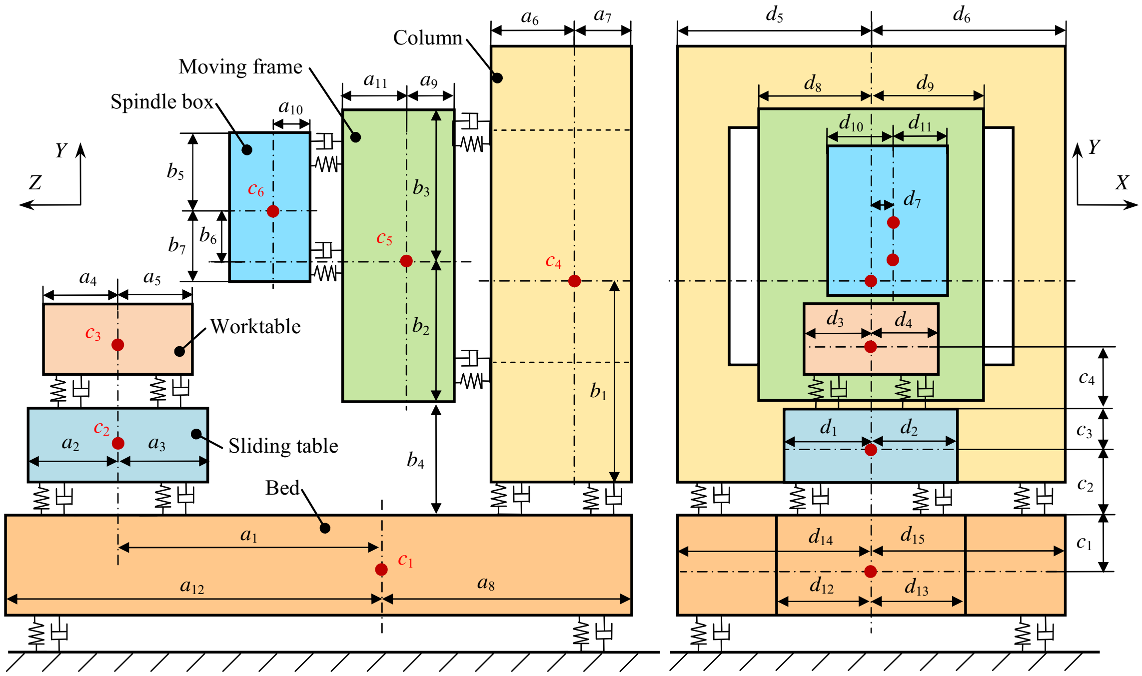

Fig. 4 Parametric dynamic model of the horizontal machining center with a box-in-box architecture [48]

sending messages to other objects for managing complexity—abstraction, inheritance, association, and communication [52]. These virtues make the object-oriented approach significantly facilitate the design and implementation of complex software systems. Figure 5 depicts the Unified Modeling Language (UML) diagram of the machining center. With a topology architecture model as its core, it provides the user with a set of classes which can be used to store and transmit design information and manage design knowledge and product models as well as design process.

Class *TopologyModel* represents the superclass for the topology architecture model of the machine tool. It stores and manages the design models, analyzes and maintains the completeness of the parameters, and governs the conceptual design process. It is composed of three classes: *Skeleton*, *StiffnessModel*, and *DynamicModel*. Class *Skeleton* stores and manages the geometric parameters and constraints of components, space layout of components, and the joining methods between components. These parameters are defined as a set of attributes of class *Skeleton*. The attributes can be initialized by traversing the geometric and non-geometric entities of the skeleton model as well as the fundamental CAD structure of the machine tool which is embedded in the skeleton model. By adjusting the corresponding parameters, inferring the design knowledge which is embedded in the skeleton model, and updating the related skeleton elements, a conceptual model of the designed machine tool can be established. Class *StiffnessModel* stores and manages the parameters which are necessary for the stiffness model. For a stiffness model of the machine tool, MATLAB, code for static stiffness prediction and stiffness matching design, is compiled as a standalone executable firstly. Then, through calling the external executable files by class *StiffnessModel*, static stiffness matching design can be performed. The same is true for class *DynamicModel*.

Class *MachineToolAssembly* represents the assembly model of the machining center. It stores and manages the design models, analyzes and maintains the completeness of the parameters, and governs the structural design process. Class *MachineToolAssembly* is composed of an array of *Component* objects. It gets information from *Skeleton* and generates the parametric assembly model of the machine tool by traversing assembly relationship and the fundamental CAD structure of machine tool which is embedded in the skeleton model.

Class *Component* is an abstract class representing the component of the machine tool, inherited by subclass *StructuralPart*, *FunctionPart*, and *ConnectingJoint*. It contains the related skeleton elements, geometric parameters, and the parametric CAD model of the component. Class *Component* gets geometric parameters from class *MachineToolAssembly*, generates and updates the parametric CAD model according to the attribute of *DrivingParameters*. Besides, it gets stiffness and mass parameters from the associated classes *StiffnessModel* and *DynamicModel*, respectively, as design constraints.

Class *StructuralPart* stores and manages design parameters of the structural part, such as outer dimensions, thicknesses of walls and ribs, shape and dimensions of lightening holes, as well as the arrangement of ribs.

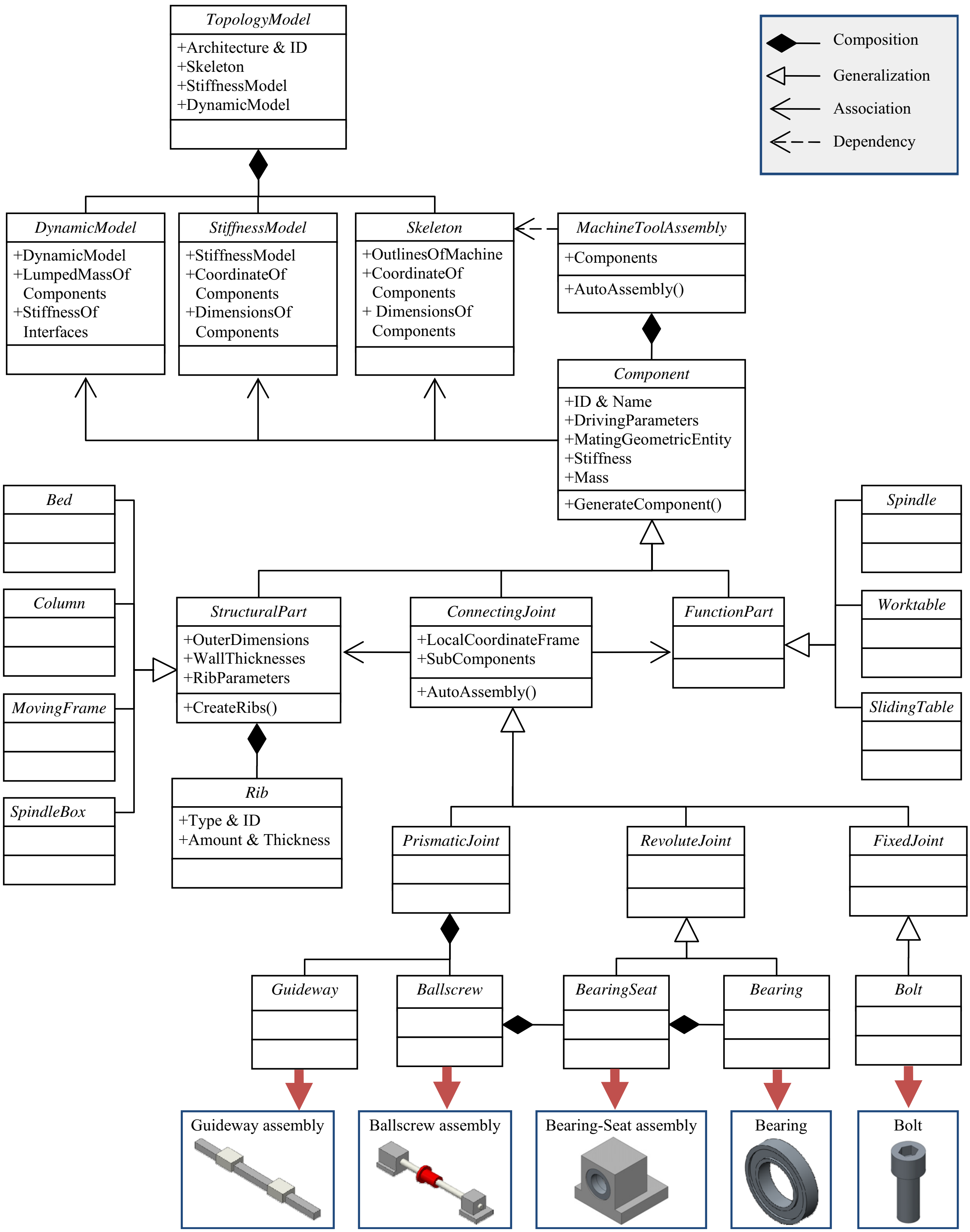

Fig. 5 UML diagram of machining center

Class *ConnectingJoint* contains geometric parameters, connecting stiffness and damping parameters, modeling knowledge, and corresponding assembly models of connecting joints. It is inherited by subclass *PrismaticJoint*, *RevoluteJoint* and *FixedJoint*, which represent the typical connecting joints used in the machine tool.

To facilitate integration of different topology architecture models into the framework, an independent library is established. For each topology architecture model included in the library, a parametric skeleton model, two executable files for static and dynamic stiffness matching design, and their own parameter configuration files are stored. The parameter configuration files, which store the names and classifications of model parameters in text format, are used to maintain the parameter mapping relation between the class and the topology architecture model. Therefore, based on the established object-oriented model and the topology architecture model library, any well-defined topology architecture model of the machining center, can be integrated in the proposed framework.

4 CAD/CAE/iSIGHT integration

A handful of software packages that are capable of serving as multidisciplinary design optimization (MDO) frameworks exist in industry and academia [53]. iSIGHT is one of the leading commercial software suites that represent the state of the art. In our study, parametric modeling of the machine tool was carried out by Pro/ENGINEER; meshing and numerical analysis were carried out by ANSYS. The design optimization platform based on iSIGHT which integrated the above software is shown in Fig. 1.

4.1 CAD/CAE integration

4.1.1 Analysis feature modeling

The interface primarily determines whether the simulation results approach the real characteristics of the system [28, 54, 55]. Generally, there are three types of interfaces in the machine tool structure: rolling interface, sliding interface, and fixed interface. For the FE modeling of the machine tool system, proper simplification of the interface can improve modeling efficiency without losing the accuracy of performance estimation. For example, the FE modeling of a guideway (rolling guide) assembly is shown in Fig. 6. As shown in Fig. 6a, it contains two types of interfaces, rolling interface and bolted fixed interface. To make the model more realistic, the slider and the rail are modeled as solid elements and connected by spring elements at the rolling interface [55]; the bed and the sliding table are modeled as solid elements and connected by spring elements with the rail and the slider, respectively, at the corresponding bolted fixed interfaces. By this manner, the entire machine tool can be modeled as several sets of solid elements which represent the solid bodies of all components, and are connected by several sets of spring elements at the interfaces.

Regarding the structure design of the machine tool, it is expected that the integration process between CAD and CAE systems should be seamless to make the optimization of structural parts and the performance evaluation of the entire machine at different configurations automatically. To meet these requirements, a feature-based CAD/CAE integration approach [56, 57] is developed by exploiting the application programming interface (API) of commercial CAD and CAE systems. In our study, the analysis feature is defined as an informative unit representing a region of interest in a FE model. And it can be described by an aggregation of properties of the model. The relevant properties are referred to as feature attributes, including parametric values and geometric entities they are associated to.

In the proposed feature-based CAD/CAE integration approach, two sets of analysis features, namely solid-body feature and interface feature, are defined. The object-oriented model of the analysis feature for the machine tool is demonstrated in Fig. 7. Class *SolidBodyFeature* represents the solid model of a single part of the machine tool. It contains a number of sub-features such as *MaterialFeature*, *ElementFeature*, *LoadFeature*, and *ConstraintFeature*. Class *InterfaceFeature* represents the interface feature between two solid bodies,

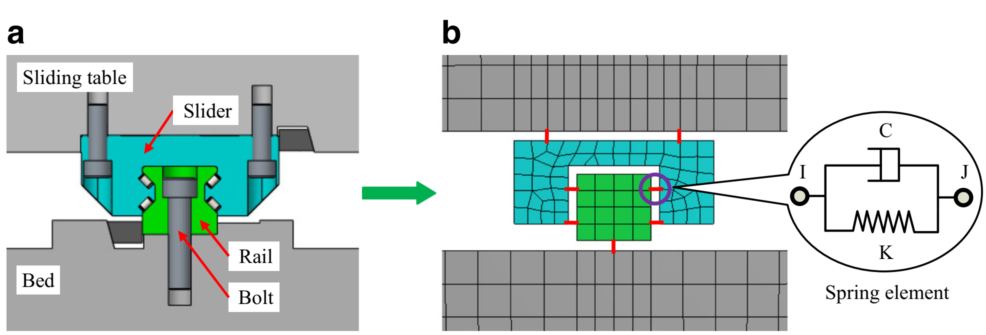

Fig. 6 Finite element modeling of the guideway. **a** CAD model of the guideway assembly. **b** Equivalent FE model of the guideway with spring elements

which is simplified into a series of spring elements in the CAE environment. When the spring elements are used to compose an interface feature, configuration parameters such as the amount, layout, and stiffness and damping coefficients are calculated according to both the type of the interface and the parameters of the corresponding functional unit.

In the CAD environment, with the aid of the application programming interface (API) of the CAD system, the information needed for structural analysis, such as materials, loads, and constraints, is stored in text format and attached to the associated geometric entities in the form of "attribute." Here, the associated geometric entities are points, lines, faces and bodies, etc. For instance, the attribute of the material is attached to the solid body of the structural part once the material is assigned or selected by the designer. Similarly, the attribute of the interface feature is attached to the faces of the joint surface. In the opposite way, the information of these added attributes and its associated geometric entities can be extracted and organized into an analysis feature model of the machine tool.

In the CAE environment, by using the built-in import/export functions supported by most of the commercial CAD software, the geometric data of the designed model can be transferred into CAE environment via an intermediate data format, such as STEP (Standard for Exchange of Product data), IGES (Initial Graphics Exchange Standard), parasolid, STL, etc. But the non-geometric information for analysis, i.e., material properties, loads, and constraints, cannot be directly transferred between the CAD and CAE environments [58]. Note that most of the commercial CAE systems provide scripting language that support programming and I/O command, such as APDL (ANSYS Parametric Design Language) for ANSYS, PCL

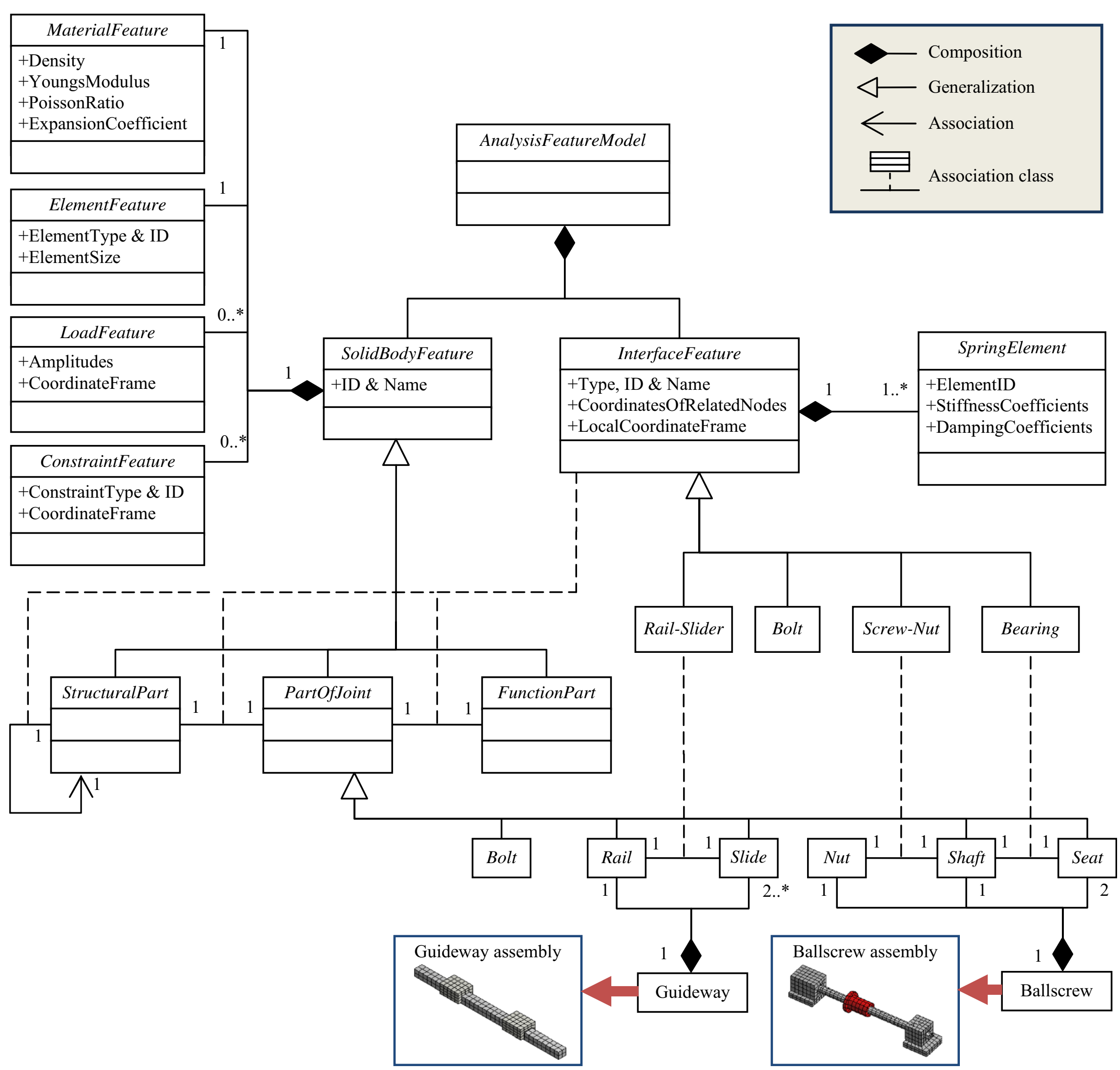

Fig. 7 Object-oriented model of analysis feature for machine tool

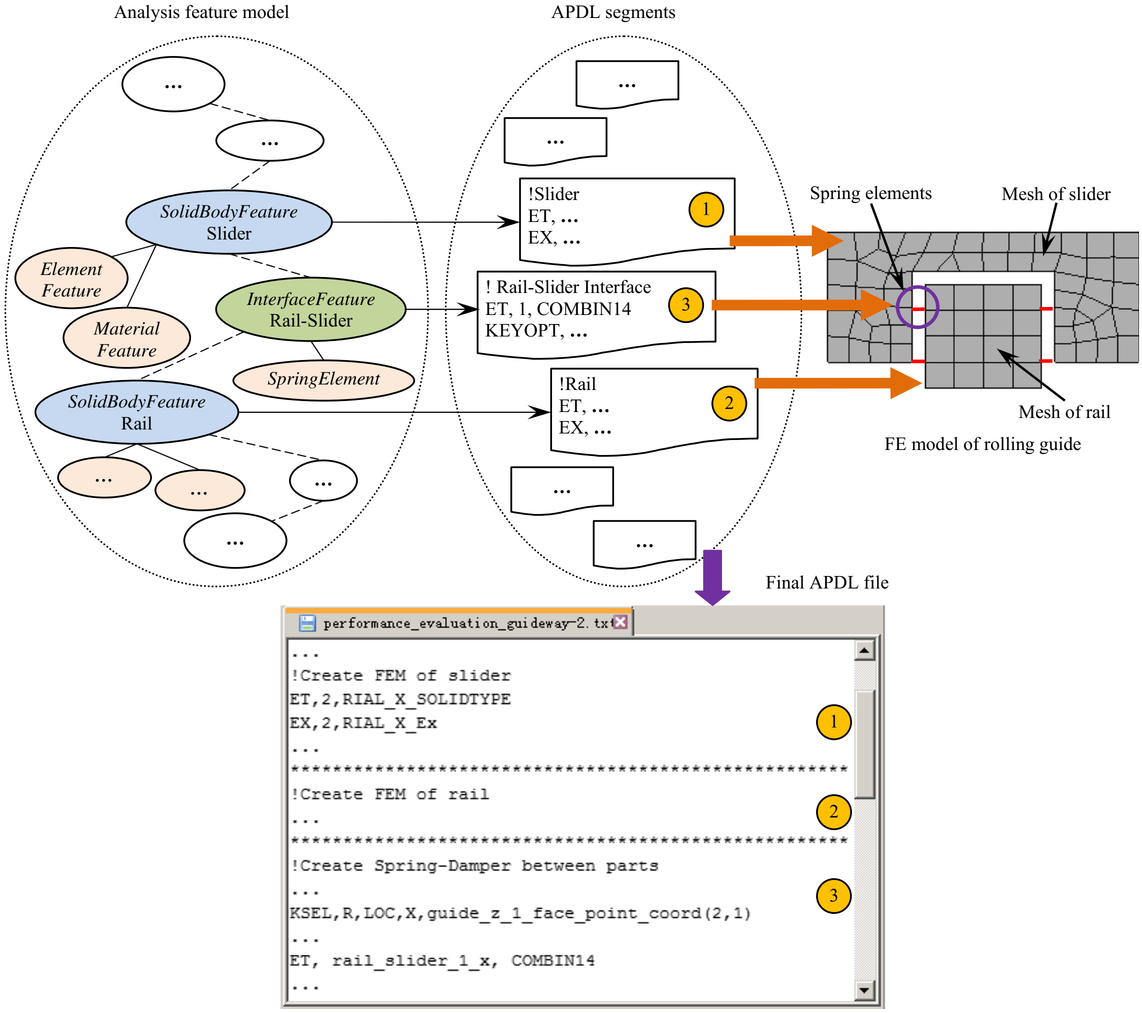

Fig. 8 Mapping process from analysis features of rolling guide to APDL script codes

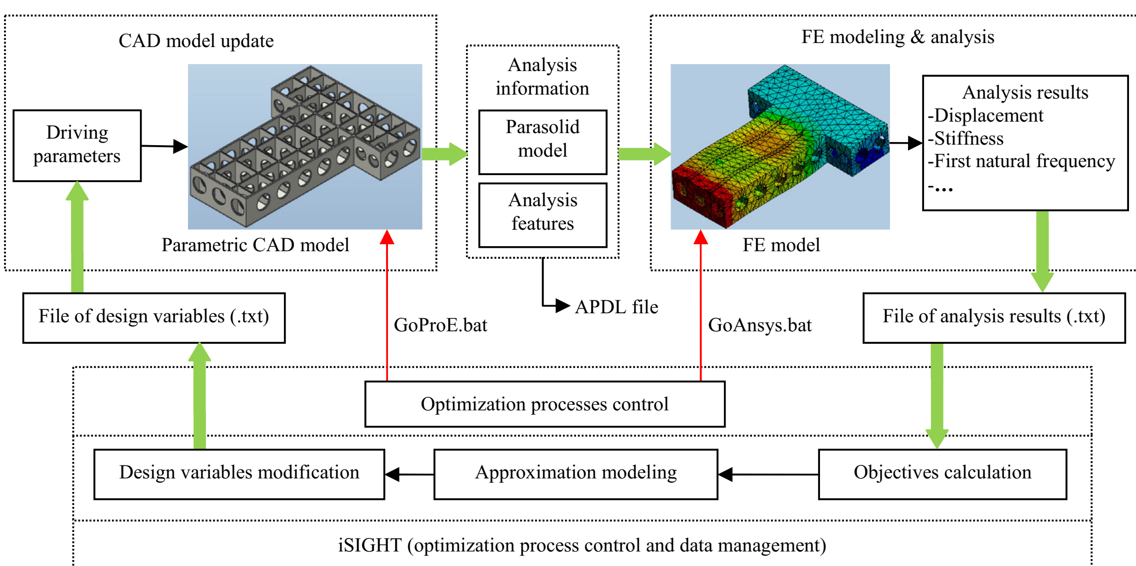

Fig. 9 Integrated optimization framework for machine tool structure

(Patran Command Language) for MSC/Patran, and Bacon (a script language that can be used to mimic GUI operations) for SAMCEF. By the programming, FE modeling, analysis job creation, and result viewing can be performed automatically. Thus, by converting the analysis feature model into scripting language, it is effective to integrate non-geometric information in the CAE system and finally convenient to automate the FE modeling and simulation of the machine tool.

4.1.2 Analysis feature mapping

Figure 8 shows the mapping process from the analysis feature model to the APDL scripts. Each solid-body feature or interface feature of the analysis feature model corresponds to a single APDL segment. The corresponding APDL segment contains specific key words and parameters. The APDL segment of a solid-body feature contains key words about meshing, material properties assigning, constraints and loads

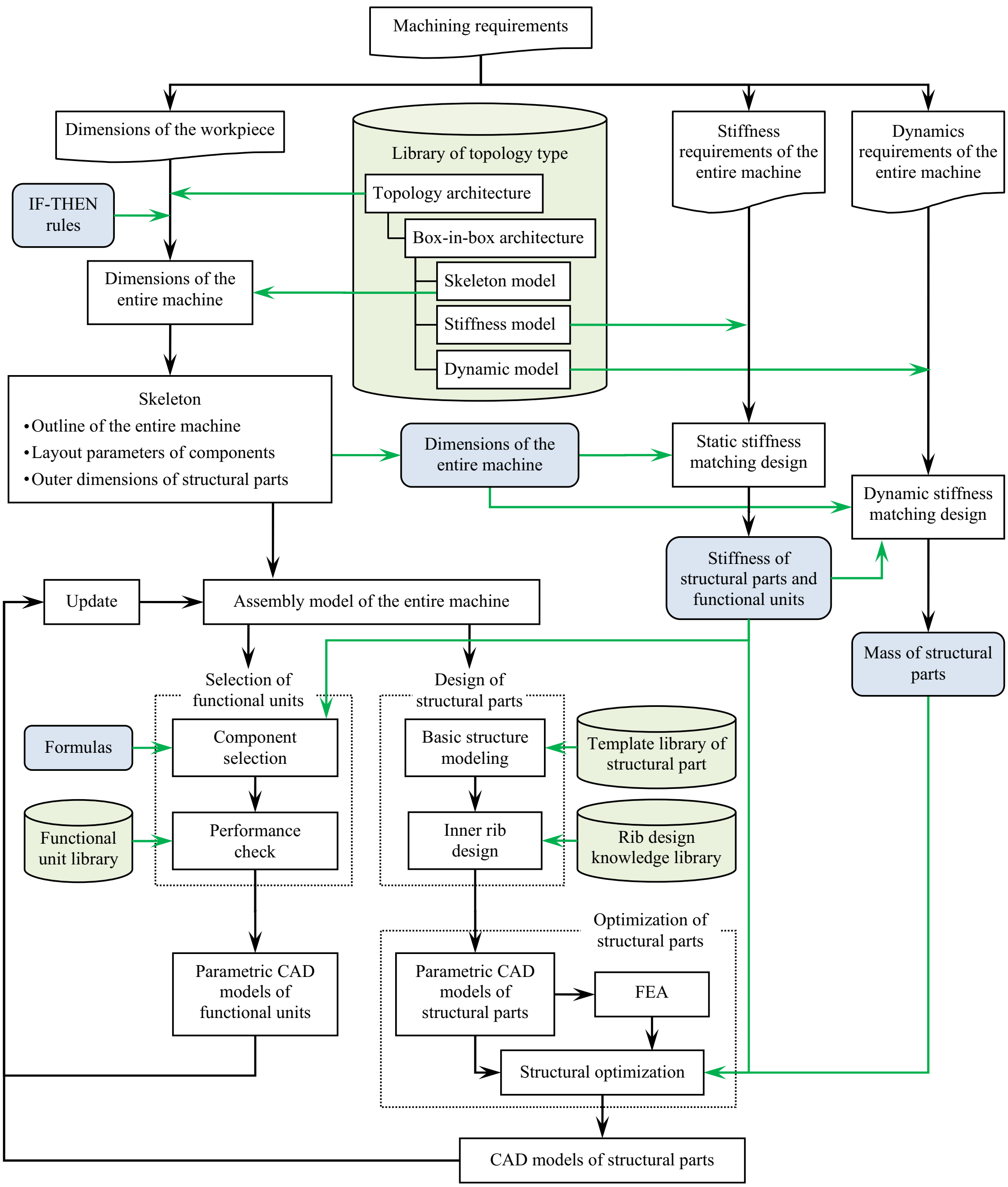

Fig. 10 Structural design procedure of machine tool

adding, parameters of which can be obtained from the attributes of the corresponding analysis feature, while the APDL segment of an interface feature contains key words about creation of spring elements. The amount and layout information of spring elements can be obtained from the interface feature. The stiffness and damping values can be directly obtained from the properties of spring element feature. Once all the APDL segments are created, they are combined into a single APDL scripting code to perform the FE modeling and analysis.

4.2 CAD/CAE/iSIGHT integration

Figure 9 shows the integrated optimization framework for the machine tool. As shown in Fig. 9, iSIGHT serves as an integrator that integrates both the CAD and CAE components by means of a batch file and a text file. By running the batch file of "GoProE.bat", it invokes Pro/ENGINEER to read the file of design variables, update the geometric model, and export it as Parasolid format for the FE modeling. By running the batch file of "GoAnsys.bat", it starts ANSYS to run the APDL file to perform the FE modeling and analysis, and output the analysis results file. Here, three files are exploited for the integration of CAD, CAE, and iSIGHT. The file of design variables is used for transferring the design parameters between CAD component and iSIGHT. The file of analysis results is used for transferring the simulation result data between CAE component and iSIGHT. The APDL file is used for the integration of geometric and non-geometric information between CAD and CAE software. Meanwhile, iSIGHT also serves as a controller that governs the model-evaluate-remodel loop, modifies design variables, extracts analysis results, and manages data exchange between software packages.

5 Structure design of the machine tool based on the topology architecture model

To realize the top-down design of the machine tool structure, a two-stage design optimization process is proposed in this study. The detailed procedure is shown in Fig. 10.

5.1 Machine type selection and configuration design

By comprehensively considering the customer requirements and the required machine properties, such as stiffness, cutting capability, occupying space, and manufacturing cost, an appropriate topology architecture type of the machine tool is selected firstly. The joining methods among component parts are also determined in this step.

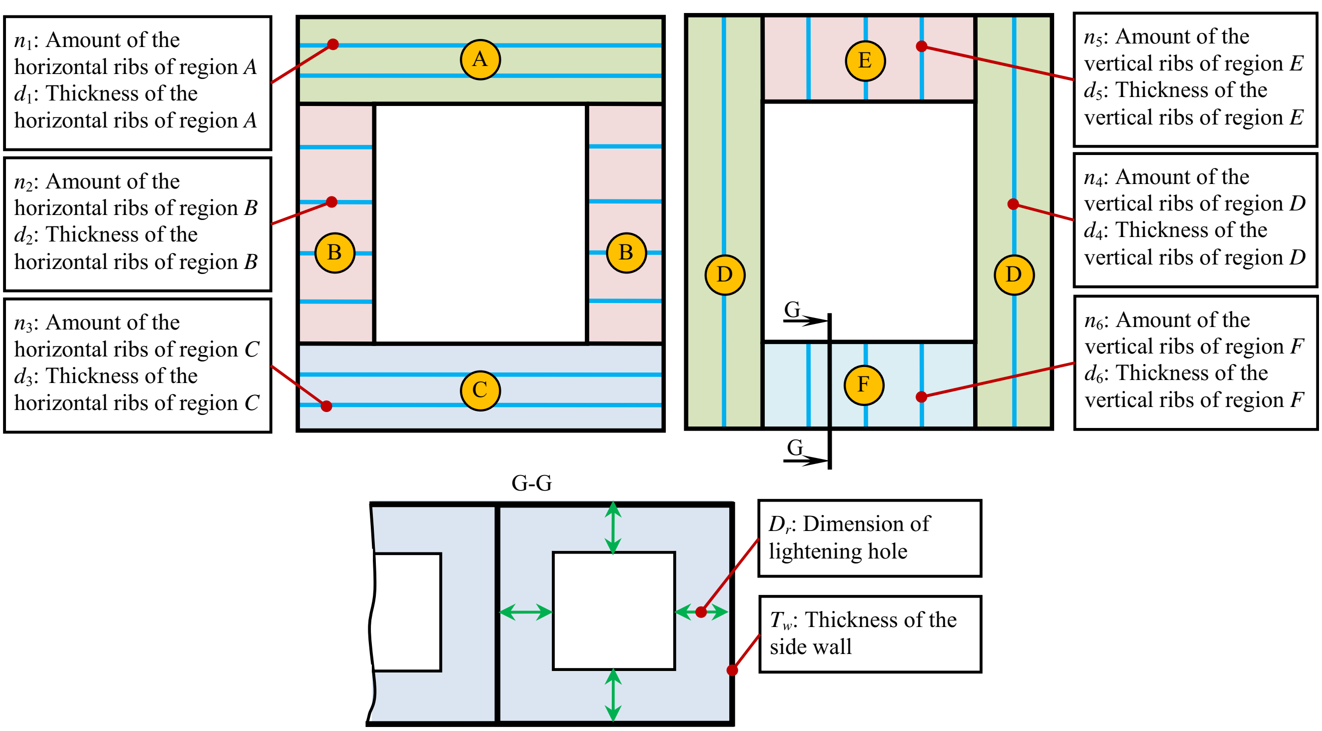

Fig. 11 Design template of the column

Table 1 Design requirements of the machining center

Design parameters	Values
Maximum dimensions of the workpiece (*X*/*Y*/*Z*)	1000/1000/1000 mm
Table-board dimensions of the worktable (*X*/*Z*)	1000/1000 mm
Strokes (*X*/*Y*/*Z*)	1200/1100/1100 mm
Minimum static stiffness (*X*/*Y*/*Z*)	50/50/50 N/μm
Natural frequency range	10–65 Hz

Then, the principal design parameters and locations of important components, which critically affect the performance of the machine, are determined by inferring the design knowledge which is embedded in the skeleton model. These parameters include the size of the worktable, the stroke of the feed mechanism, the size and position of the each structural part, etc.

5.2 Stiffness matching design

Based on the stiffness model in Section 3.2, the stiffness matching design can be performed. The optimization procedure [24] of applying the linear programming method to obtain the allocated stiffness of the structural parts and functional units is as follows.

(1) The target stiffness of the entire machine is determined according to the machining requirements. Once the necessary geometric parameters of stiffness model are

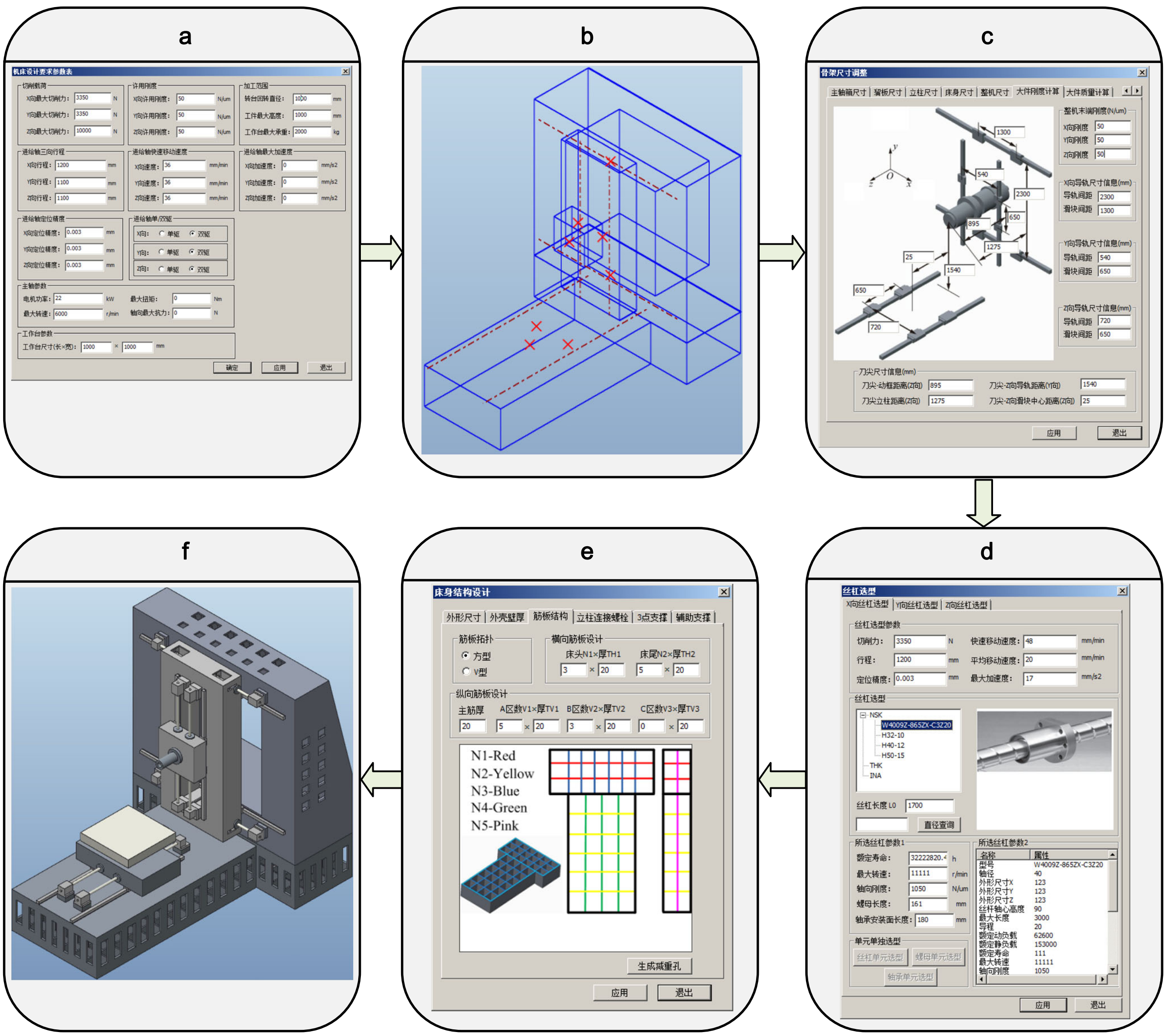

Fig. 12 Screenshots displaying the structural design processes. **a** Design specifications. **b** Machine type selection and configuration design. **c** Static and dynamic stiffness matching design. **d** Selection of functional units. **e** Design of structural parts. **f** Assembly model of the entire machine

extracted from the skeleton model, the stiffness model is then converted into linear equations, which are used as linear constraints in the linear programming.

(2) The ranges of stiffness coefficients of the structural parts and functional units which are necessary for the linear programming method are obtained from a database. The database was built based on simulations of similar structural part and the manufacturers' data.

(3) The objective of the optimization problem is to maximize the stiffness coefficients of corresponding structural part which have the greatest impact on the stiffness of the entire machine.

(4) Based on the linear programming method, obtain the reasonable stiffness of the structural parts and functional units.

5.3 Dynamic design

Material properties and weight of structural parts, as well as joining methods among component parts, have great influence on the dynamic performance of the machine tool. Because the joining methods among component parts and materials of structural parts have been determined at the machine type selection step, the weight of the structural parts and functional units are the main factor affecting the dynamic performance of the machine tool. The specific procedures of applying the RSM to optimize the weight of structural parts and functional units are listed below.

(1) Define the factor levels of each variable and use the Box-Behnken design (BBD) to construct the experiment design [59].

(2) Define the objective function as shown in Eq. (11).

$$\begin{cases} \max \ f_1 \\ \max \ \ K_{Dynamic\_X},\ K_{Dynamic\_Y},\ K_{Dynamic\_Z} \\ \min \ \ \sum_{i-1}^{n_p} m_i \\ s.t. \ \ \ m_i \in [0.8M_i, 1.2M_i],\ \ i = \{1,\ 2,\ \cdots,\ n_p\} \end{cases} \tag{11}$$

where M_i is the mass of the structural part of a successful design case which has similar specifications with the designed machine tool. If there is no successful design case for reference, the value of M_i can be set to the mass of the initial design of the structural part described in Section 5.5.

(3) Perform the numerical simulation and evaluate the results according to the predefined objective function by the dynamic model as mentioned in Section 3.3. The quadratic polynomial function as shown in Eq. (12) is used to approximate the relationships among the objective function and the variables.

$$Y = a_0 + \sum_{i=1}^{n_v} a_i x_i + \sum_{i=1}^{n_v} a_{ii} {x_i}^2 + \sum_{i=1}^{n_v - 1} \sum_{j=2}^{n_v} a_{ij} x_i x_j + \varepsilon \tag{12}$$

(4) Based on the quadratic relationship, obtain the reasonable mass of each structural part.

Table 2 Some principal dimensions of the machining center

Symbol	Value (mm)	Symbol	Value (mm)
DX_{mt}	3340	DX_{mf}	1300
DY_{mt}	3630	DY_{mf}	2900
DZ_{mt}	4500	DZ_{mf}	380
DX_{bed}	3340	DX_{spb}	680
DY_{bed}	630	DY_{spb}	650
DZ_{bed}	4500	DZ_{spb}	300
DX_{col}	3340	DX_{st}	1388
DY_{col}	3000	DY_{st}	437
DZ_{col}	1200	DZ_{st}	800

5.4 Selection of functional units

Functional units include spindle, worktable, turntable, and standard parts, such as ballscrew, guideway, etc. According to the principal parameters extracted from the skeleton model and the reasonable stiffness obtained by the stiffness matching design, an appropriate functional unit is selected from the functional unit library. Performance check is also carried out to evaluate whether it meets other design requirements. Then, the corresponding parametric CAD model is generated and loaded to update the assembly model of the entire machine in accordance with the geometrical constraints and

Table 3 Allocated stiffness of structural parts

Component	*X* direction (N/μm)	*Y* direction (N/μm)	*Z* direction (N/μm)
Bed	832	1053	–
Column	–	816	453
Moving frame	451	–	310
Spindle box	2261	2310	3037

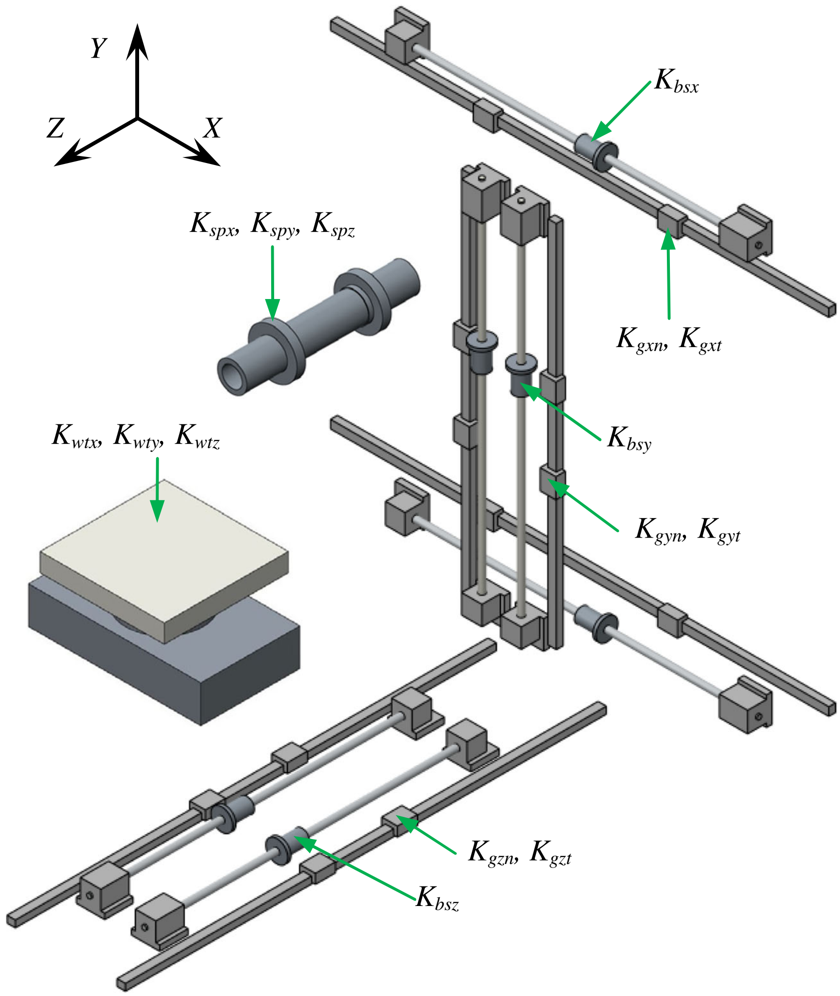

Fig. 13 Stiffness of functional units

spatial position of components which are specified in the skeleton model.

5.5 Design optimization of structural parts

Structural parts are basically shell structures for lower mass with ribs inside for reinforcement. By interviewing experienced engineers, rib design knowledge of typical structural part is acquired. Then, the parametric templates of the typical structural part are established in which the acquired empirical knowledge, design expertise, design standard, and code are embedded through the built-in formulae and if-then rules. Figure 11 shows the design template of the column of a machining center with a box-in-box architecture. Considering the castability of structural parts, some design knowledge of ribs is summarized as follows.

- The side wall thickness of the structural part should be 30–50 mm.
- The minimum distance between the rib and wall is 100 mm.
- The minimum distance between ribs is 200 mm.
- Ribs are classified into two categories: main rib and assisting rib.
  - There should be at least one main rib to support the guideway, while the assisting ribs are arranged upon the space left.
  - The main rib thickness should be 80–100% of the wall thickness.

Table 4 Allocated stiffness of functional units

Symbol	Value (N/μm)	Symbol	Value (N/μm)
K_{bsx}	303	K_{gzt}	1694
K_{bsy}	306	K_{spx}	357
K_{bsz}	306	K_{spy}	357
K_{gxn}	2380	K_{spz}	1410
K_{gxt}	1298	K_{wtx}	2500
K_{gyn}	2380	K_{wty}	7200
K_{gyt}	1694	K_{wtz}	2500
K_{gzn}	2380		

Table 5 Some solutions of the Pareto optimal set obtained after performed dynamic design

No.	Bed (t)	Column (t)	Moving frame (t)	Spindle box (t)	f_1 (Hz)	A_X (μm)	A_Y (μm)	A_Z (μm)	$A_X - A_Y$ (μm)
1	11.24	10.29	1.59	0.67	29.73	10.98	3.68	33.91	7.29
2	10.37	10.17	1.59	0.59	30.42	10.37	3.27	35.24	7.1
3	10.78	9.45	1.6	0.63	30.01	10.82	3.46	34.64	7.37
4	10.71	9.23	1.62	0.57	30.5	10.48	3.15	36.39	7.33
5	11.09	10.22	1.59	0.64	30	10.71	3.48	34.41	7.23
6	13.9	11.25	1.61	0.48	31.461	9.303	2.963	39.478	6.340
7	10.36	10.31	1.59	0.53	31.1	9.79	3.04	37.19	6.75
8	11.11	10.13	1.62	0.55	30.668	10.206	3.075	37.048	7.132
9	14.17	11.73	1.59	0.47	31.680	8.973	2.982	39.495	5.991
10	12.97	11.12	1.6	0.47	31.692	9.091	2.978	39.833	6.113

  - The assisting rib thickness should be 60–100% of the wall thickness.

- Lightening holes are generally circular or rectangular in shape with an average area of 50–60% of the rib.

Factors that affect the stiffness and weight of structural parts and then affect the static and dynamic performance of the entire machine include the outline dimensions of structural parts, the basic arrangement of ribs, the thickness of ribs/walls and dimensions of lightening holes [60]. In our study, the outline dimensions of structural parts are determined by inferring the design knowledge which is embedded in the skeleton model in the configuration design stage. The amounts and the distribution of ribs are determined by inferring the empirical knowledge which is embedded in the design template. Therefore, the thickness of ribs/walls and dimensions of lightening holes are defined as the optimization variables. The objective of the optimization problem is to maximize static stiffness in the X, Y, and Z directions and the first natural frequency as well as to minimize the volume and displacement of the structural part. The integrated optimization framework as mentioned in Section 4.2 is used to perform the design optimization of the structural part. The specific procedures of applying the RSM to optimize the structural part are listed below.

(1) Define the factor levels of each variable and use the Latin hypercube design (LHD) to construct the experiment design [61].

(2) Define the objective function as shown in Eq. (13).

$$\begin{cases} \max \ f_1 \\ \max \ K_X, \ K_Y, \ K_Z \\ \min \quad m_{\mathrm{str}} \\ s.t. \qquad 20 \le d_i \le 30, \quad i = \{ \ 1, \ 2, \ \cdots, n_r \} \\ \qquad\qquad 50 \le D_r \le 100 \\ \qquad\qquad 30 \le T_w \le 50, \end{cases} \tag{13}$$

where n_r is the total number of the region in which the rib distributed as shown in Fig. 11.

(3) Perform the FE analysis and evaluate the results according to the predefined objective function. In this step, the geometrical model is generated according to the design points that were constructed by DOE. The FE model is generated and FE analysis is performed by using the CAD/CAE integration approach mentioned in Section 4.1. The quadratic polynomial function as shown in Eq. (12) is used to approximate the relationships among the objective function and the variables.

(4) Based on the quadratic relationship, NSGA II is applied to obtain the Pareto optimal set of the structural part.

Table 6 Design variables of the column

Design variables	d_1 (mm)	d_2 (mm)	d_3 (mm)	d_4 (mm)	d_5 (mm)	d_6 (mm)	D_r (mm)	T_w (mm)
Lower bound	20	20	20	20	20	20	50	30
Upper bound	30	30	30	30	30	30	100	50

Table 7 Mechanical properties of material used in FEA

Material	Density (kg/m^3)	Elastic modulus (GPa)	Poisson's ratio
Cast iron	7300	145	0.27

(5) Finally, the solution, which is most close to the allocated stiffness and mass of the structural part, is selected from the Pareto set as the optimization result. The obtained parameters are then used to update the parametric CAD model of the structural part.

6 Case study

The proposed framework is illustrated in this section with the re-design of a four-axis precision horizontal machining center with a box-in-box architecture. The machining center is conceived for finish machining of box-type parts, such as gearbox body, engine cylinder block, etc. The representative milling operations include face milling and boring. For the face milling operation, tools involved have diameter range from 60 to 160 mm with tooth number from 6 to 15. The recommended rotating speed ranges for cast iron and aluminum alloy are 400–600 and 1000–2000 r/min, respectively. Hence, the excitation frequency range is about 65–500 Hz. For the boring operation, the recommended rotating speed range is 70–100 r/min; thus, the excitation frequency range is about 1.2–1.7 Hz. Therefore, in order to avoid the excitation frequencies [62], the natural frequency range of the machine structure should be 10–65 Hz. In general, values between 10 and 25 N/μm are considered to be adequate for general-purpose milling machines [63]. To make the machine well suited for high-precision machining, the structural stiffness is targeted to be on the order of 50 N/μm. Some design requirements of the machining center are summarized in Table 1.

6.1 Configuration design and structural part design

Figure 12 shows the key steps and the main interfaces of the structural design of the machine tool. The main design steps are as follows:

(1) Input the design requirements through interface as shown in Fig. 12a.
(2) Select the box-in-box type of machine tool from the library of the topology architecture model. The skeleton model of the selected type is shown in Fig. 12b. The principal dimensions of the entire machine and structural parts are determined according to the formulas and rules embedded in the predefined skeleton model. Some principal dimensions are summarized in Table 2.
(3) Perform static stiffness matching design and dynamic design. The interface of static stiffness matching design and dynamic design is shown in Fig. 12c. Reasonable stiffness values of structural parts obtained after stiffness matching design are listed in Table 3, while the stiffness of functional units is depicted in Fig. 13 and the allocated stiffness values are listed in Table 4. Table 5 shows some solutions which are selected from the Pareto optimal set obtained after dynamic design is performed. In Table 5, solution 7 is chosen as the optimal solution due to its higher first nature frequency, lower amplitude of dynamic response in the three directions, lower difference between amplitudes of dynamic responses in the X and Y directions, and lower weight of the entire machine.
(4) Select functional units according to the allocated stiffness and design requirements through interface as shown in Fig. 12d.
(5) Design structural parts by interactively inputting or modifying the thickness and the layout parameters of ribs as shown in Fig. 12e.
(6) Assemble all the components into the assembly model of the entire machine, as shown in Fig. 12f.

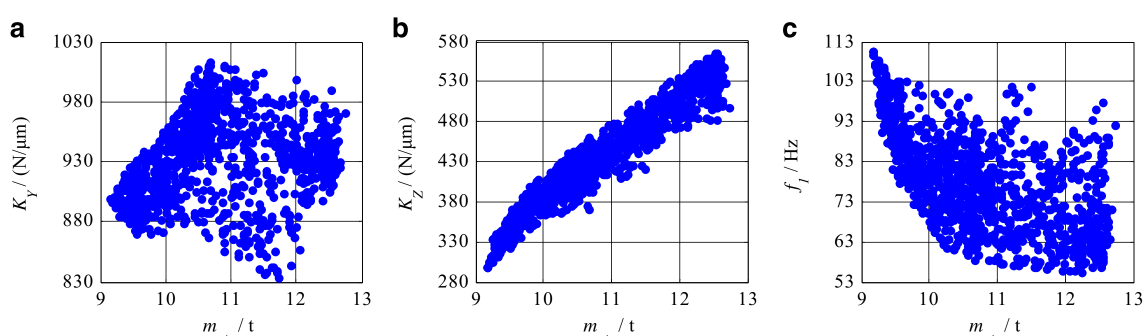

Fig. 14 Pareto plot of multi-objective optimization of the column. **a** Pareto plot of K_Y vs m_{str}. **b** Pareto plot of K_Z vs m_{str}. **c** Pareto plot of f_1 vs m_{str}

Table 8 Comparison of the optimum solution and original design of the column

Variables and responses	d_1 (mm)	d_2 (mm)	d_3 (mm)	d_4 (mm)	d_5 (mm)	d_6 (mm)	D_r (mm)	T_w (mm)	K_Y (N/μm)	K_Z (N/μm)	f_1 (Hz)	m_{str} (t)
Original design	20	20	20	20	20	20	75	35	757.3	375.6	59.7	10.85
Optimized design	28.4	20.4	23	27.2	28	28.2	89.1	32.8	964.1	407.5	74.4	10.31

6.2 Structural parts optimization

The column is taken as an example to illustrate the structure optimization procedures of structural parts. Because the outer dimensions and the rib amount have been determined in the structural design stage, the dimension of the lightening hole and the thickness of the rib and side wall are chosen as the design variables. Note that the stiffness in the X direction of the entire machine is decided by the ballscrew in the X direction for it has the minimal stiffness among the components in this direction [24]. Therefore, the stiffness of the column in the X direction is not taken as the optimization target because it is much greater than that of the ballscrew. The objective of the optimization problem is to maximize the stiffness in the Y, Z directions and the first natural frequency, while minimizing the mass of the column. The design variables are depicted in Fig. 11, and their upper and lower bounds are listed in Table 6. The material properties of cast iron are shown in Table 7.

To evaluate the static stiffness, the concentrated force of 1 N in three directions is applied at the center of the upper-left/or right slider when the moving frame is located in the middle of the column in the X direction, according to the definition of the stiffness coefficient [24]. The boundary condition is that the column is fixed at the bottom plane.

Figure 14 shows the Pareto plot obtained from solving the multi-objective optimization problem of the column. It can be seen that as mass increases, the first natural frequency decreases while stiffness in the Y and Z directions increases. According to the principle of solution selection mentioned in Section 5.5, optimum design variables in multi-objective optimization of the column are determined, as shown in Table 8. The dimensions and responses of the original design of column are also listed in the table. The optimization mechanism of other structural components is similar. Table 9 lists the optimal results.

6.3 Design result verification

After structure optimization of all the structural parts have been carried out, the assembly model of the machine tool is updated, and the finite element model is generated according to the modeling method depicted in Section 4.1. Then, static, modal, and harmonic response analyses are carried out to verify the design results. In the static analysis, cutting forces of 1 N in the X, Y, and Z directions are applied to the positions of the cutting tool and the worktable, respectively, to calculate the relative static deflection between them. In the harmonic response analysis, the machine structure is excited by a series of sine excitation forces with the average value of 1 N imposed at the cutting tool. A frequency range from 0 to 500 Hz and the mode superposition method are chosen so as to give an adequate response curve.

Figures 15, 16, and 17 show the static and dynamic performance of the original and re-designed structure of the machining center, respectively. Principal characteristic parameters of the original design and the re-design of the machine tool are summarized in Table 10. Compared with the original design, static stiffness in the X, Y, and Z directions of the re-designed structure tends to

Table 9 Comparison of the optimum solution and original design of other structural components

Responses	Bed				Moving frame				Spindle box				
	K_X (N/μm)	K_Y (N/μm)	f_1 (Hz)	m_{str} (t)	K_X (N/μm)	K_Z (N/μm)	f_1 (Hz)	m_{str} (t)	K_X (N/μm)	K_Y (N/μm)	K_Z (N/μm)	f_1 (Hz)	m_{str} (t)
Original design	597.7	1117.3	62	12.49	582.1	394.1	179.7	2.01	1212.3	1189.5	2587.9	558.3	0.49
Optimized design	851.2	1190.4	79.2	10.35	515.5	362	223.5	1.59	2629.5	2388.7	3263.7	574.5	0.53

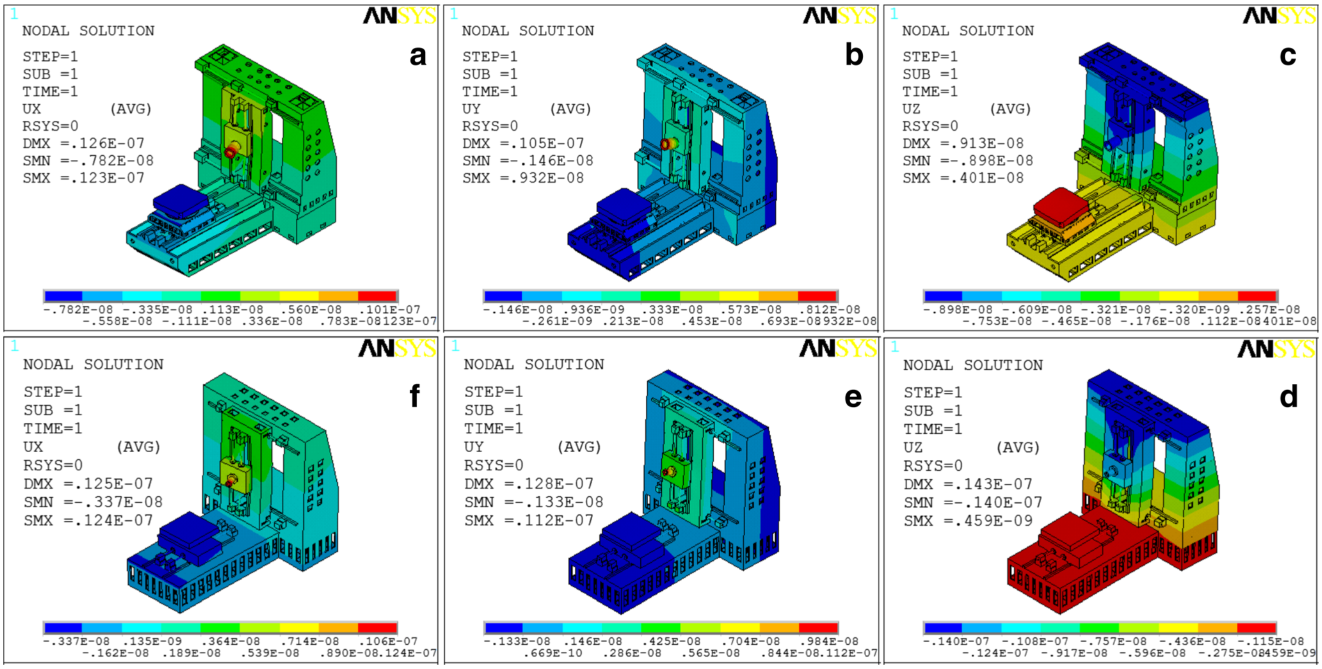

Fig. 15 Static stiffness analysis results of the entire machine. **a**–**c** Deformation in the *X*, *Y*, and *Z* directions of the original design result. **d**–**f** Deformation in the *X*, *Y*, and *Z* directions of the re-design result

accordance and the dynamic performances increase remarkably. Specifically, the first natural frequency is improved by 5.6%, and the maximum compliances in the *X*, *Y*, and *Z* directions are reduced by 23.3, 20, and 9.9%, respectively, while the overall weight of the machine tool is reduced by 8.1%.

7 Conclusions

In order to improve the design efficiency and quality, a novel integrated framework and systematic design methodology for top-down structure design of machine tools is presented in this paper. The following conclusions can be summarized as follows:

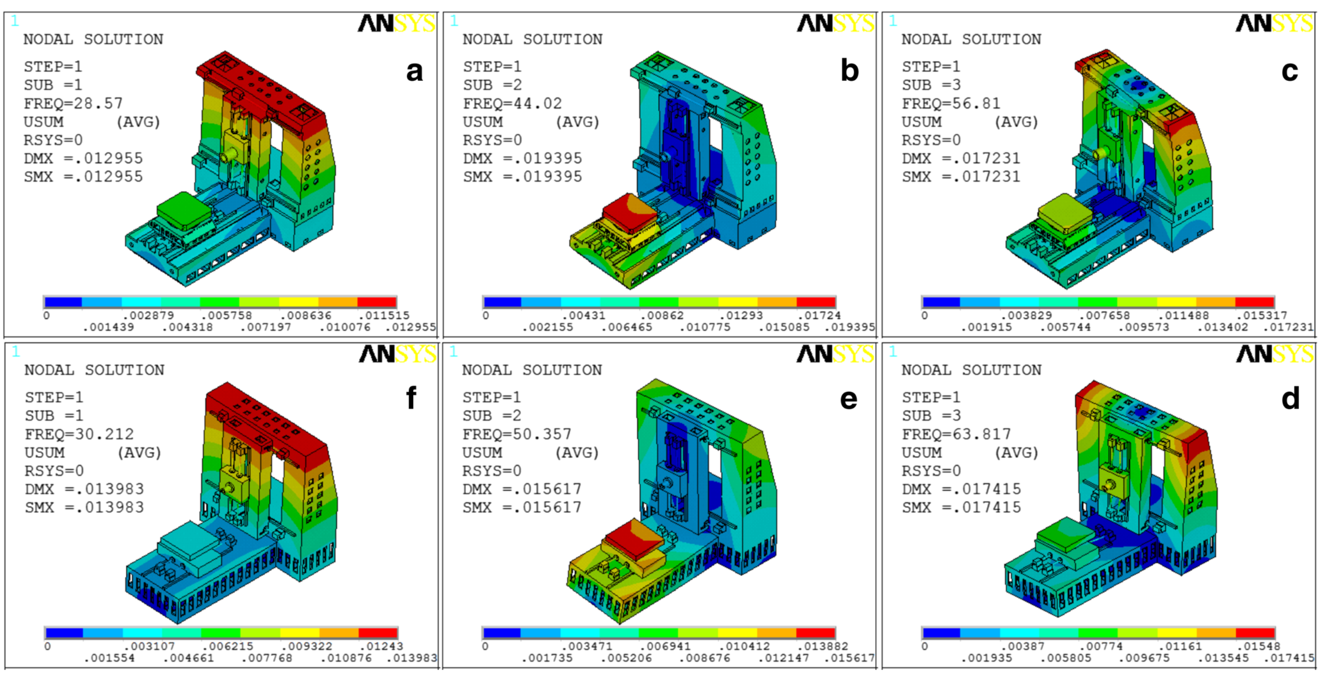

Fig. 16 Modal analysis results of the entire machine. **a**–**c** 1st–3rd order mode shapes of the original design result. **d**–**f** 1st–3rd order mode shapes of the re-design result

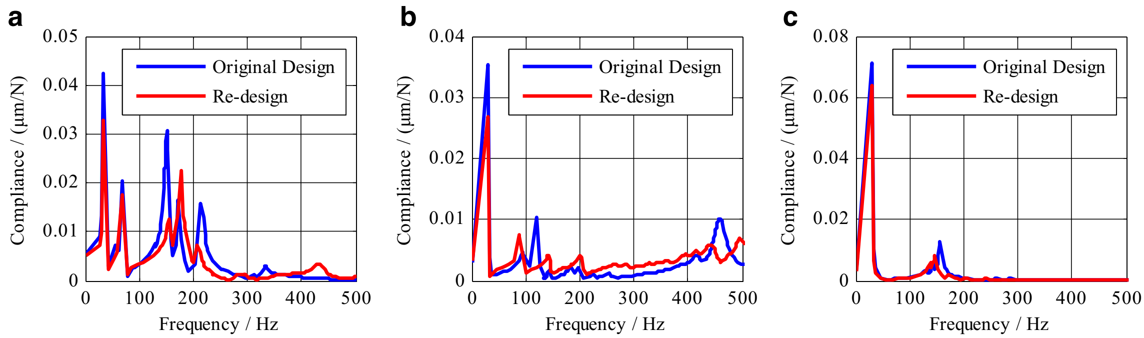

Fig. 17 Harmonic response of the entire machine. **a**–**c** Harmonic responses in the *X*, *Y*, and *Z* directions

(1) A topology architecture model has been developed to integrate the configuration design, geometric modeling, and static and dynamic performance evaluation knowledge of a machine tool with a specific architecture type. Through the pre-built relations/empirical formulas and if-then rules, the empirical knowledge, design expertise, design standard, and code are embedded in the skeleton model. Based on the stiffness model and dynamic model, static and dynamic stiffness matching design can be performed, through which the reasonable allocated stiffness and mass of structural parts and functional units can be obtained in the early design stage.
(2) An analysis feature model is proposed for the integration between commercial CAD and CAE software. By exploiting the analysis feature model and the feature mapping technique, the integration process between CAD and CAE systems could be seamless to automate the structure optimization loop and the performance evaluation of the entire machine at different configurations.
(3) A two-stage design optimization process is proposed to perform the structural design of machine tools. In the configuration stage, an appropriate topology architecture type of machine is selected firstly. Then, the principal parameters which critically affect the performance of the entire machine, including the static stiffness and the weight of structural parts and functional units, are determined. Simultaneously, the performance requirements of the entire machine are transformed into design constraints of the structural parts and functional units. In the design and optimization stage of structural parts, the arrangement of ribs is determined by inferring the empirical knowledge, design standard, and code which are embedded in the design template firstly. Then, DOE and RSM are applied to perform parameter optimization of structural parts subject to the constraints obtained from the first stage. Since the constraints are transformed from the requirements of the entire machine, the design optimization in this stage can guarantee the performance of the entire machine.

The feasibility of the proposed integrated framework and design methodology has been verified by re-design of a horizontal machining center with a box-in-box architecture. Further work will be carried out to enrich the frequently used topology architecture models of machine tools and illustrate the effectiveness of the proposed approach in the structure design and optimization of parallel kinematic machines (PKMs).

Table 10 Comparison of the original design and re-design results of the entire machine

	K_X (N/μm)	K_Y (N/μm)	K_Z (N/μm)	f_1 (Hz)	C_X (μm/N)	C_Y (μm/N)	C_Z (μm/N)	m_{mt} (t)
Requirements	50	50	50	10–65	–	–	–	–
Original design	53.3	92.8	84.7	28.6	0.043	0.035	0.071	29.5
Re-design	67.8	83.6	76.4	30.2	0.033	0.028	0.064	27.1

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