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# Investigation of moving fixture on deformation suppression during milling process of thin-walled structures

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ABSTRACT

Dynamic deformation of the thin-walled structures during milling process will influence the machining accuracy and surface quality of the final component. Present paper is concerned with deformation suppression by proposing a new method, which is realized by supporting a fixture element at the projection area of the tool-workpiece contact zone on back surface of the workpiece. During milling, the fixture element will move with the milling cutter at the same velocity. A dynamic model of the new cutter-workpiece-fixture system is constructed to analyze the workpiece deformation. A machining case is implemented to demonstrate the effectiveness and feasibility of the proposed method and model.


## Introduction

Thin-walled components, such as jet engine compressor and turbine blades, are usually machined from blocks [1]. But the low rigidity of these parts will lead to undesirable deformation during processing, which has a negative effect on machining accuracy and the surface quality of the final workpiece unless they are suppressed [2]. Many approaches have been developed to control the milling deformation of flexible components, such as sacrificial structures method [1], damping method [3,4], electromagnetic induction [5], online compensation [6], milling path planning [7–11] et al.. Among these methods, the fixture design plays an important role. It is a promising method which has drawn much attention.

To design a proper fixture for deformation suppression, the precondition is to analyze and model the machining deformation of the structures with fixture elements accurately. Wardak et al. [12], Ratchev et al. [13], Siebenaler and Melkote [14] predicted the workpiece deformation in a fixture system using FEM. FEM is a widely used method and it can predict the deformation accurately but time- consuming. Therefore, many analytical prediction methods were developed. Wang et al. [15] proposed an analytical procedure to investigate the influence of fixture layout on the dynamic deformation of the multi-span thin-walled structures. Mesherki et al. [16,17] proposed an analytical formulation for the dynamics of the multi-pocket thin-walled structures considering the fixture constraints, and the varying dynamics of the structure caused by the material removal were also discussed by the same author [18].

On the basis of accurate deformation analysis, the next step is to design a proper fixture. In this stage, many factors including layout, numbers, clamping force and sequence should be taken into account. Many algorithms and methods were proposed to optimize the layout, the clamping sequence and the clamping force [19–25], but most of them were about the rigid workiece machining cases. This paper will focus on the investigation about the flexible workpieces. Aoyama and Kakinuma [26] designed a fixture system which could support the thin and compliant workpiece securely and decreased the milling deformation, but they ignored the influence of fixture layout on the deformation response of the system. Wan and Zhang [27] treated the frame-structure components with fixture constraints as Mindlin plate (that take into account the in-plate shear strain during analyze its deformation or vibration) with simultaneous elastic edges and internal supports. According to the model, a nonlinear programming problem related to the frequency sensitivity was constructed to optimize the fixture layout to guarantee the maximum natural frequency of the fixture-workpiece system. Liu et al. [28] optimized the numbers and locations of the fixtures during end milling of thin-walled components. Chen et al.

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**Fig. 1.** Multi-pocket thin-walled structures in engineering.

[29] constructed a multi-objective model to optimize the fixture layout and clamping force, and the model was solved by genetic algorithm. Zeng et al. [30] optimized the locations, numbers and applied force of fixture elements simultaneously based on a dynamical model of fixture-workpiece-cutter system in which the milling force was treated as disturbance excitation while the fixture was treated as control input. Qin et al. [31] analyzed and optimized the fixture clamping sequence of low-stiffness workpiece. Thermal deformation during machining is also important. Ying and Tetsutaro [32] optimized the fixture design considering the cutting heat in face milling process. Vasundara and Padmanaba [25] reviewed the recent development of machining fixture layout design, analysis and optimization method.

From the literatures mentioned above, it can be concluded that the fixture elements support the workpiece at some fixed points in their investigation, which is aimed to minimize the deformation of the whole component. However, the surface quality is mainly related to the contact part between the tool and the workpiece. Therefore, there is no need to concern the deformation suppression of the whole structures. Considering this fact, present investigation proposes a new method which uses just one fixture supporting element at the back surface of the projection area of the tool-workpiece contact zone. During milling, the element will move with the cutter at the same velocity. This paper will mathematically model the proposed novel method. Thus, the rest of this paper is organized as follows: The method is descripted and then the dynamic deformation model of cutter-workpiece-fixture system is presented in Part 2. Case study is implemented to validate the method in Part 3. The whole paper is concluded in Part 4.

## Mathematical modeling of the novel method

### *Method description*

There exist many different shapes of thin-walled structures in engineering practice, such as thin-walled multi-framed and multi-pocket structures as shown in Fig. 1. For simplicity, a simple structure shown in Fig. 2 is used to illustrate the proposed method in this paper. It is also widely studied by other investigators [8,33,34]. Schematic diagram of the new method is shown in Fig. 3 (*a*). A fixture element is supported at the back surface of component and it will move with the cutting tool during milling. Fig. 3 (*b*) shows the schematic diagram of Zeng's method for comparison. Main differences between the two methods are: (i) the fixture element is mobile for the new method while it is fixed in Zeng's method; (ii) only one element is needed in present paper while many elements are needed in Zeng's method.

### *Deformation model*

To analyze the deformation, the component is simplified to a thin-walled plate subjected to the milling force. The fixture element is simplified to an oscillator which supports the plate at its back surface. Besides, it is assumed that the separation between the workpiece and the moving fixture won't happen during milling [35]. The simplified model is shown in Fig. 4. During machining, both the milling force and the oscillator will move at a velocity of the feeding speed along the feeding direction. According to Altintas [36], the milling force has three components. But only the milling force which is perpendicular to the plate surface will lead to its transverse deformation. Thus, the forces which are parallel to the surface plate are ignored. Suppose the feeding speed is $v(t)$, the milling path is denoted by $(\xi(t),\eta(t))$, the parameters of the oscillator are $m$, $c$ and $k$ respectively, the deformation of the plate is $w$. According to the thin-walled shell theory, $w$ satisfies the following formulation.

$$\begin{cases} D\left(\nabla^2\nabla^2 w+\chi\nabla^2\nabla^2\left(\dfrac{\partial w}{\partial t}\right)\right)+\rho h\dfrac{\partial^2 w}{\partial t^2} \\ =\left(P_0-F_z(\phi)\right)\delta\left(x-\xi(t)\right)\delta\left(y-\eta(t)\right) \\ P_0=c\dfrac{dz(t)}{dt}+k\left(z(t)-z_0\right) \end{cases} \quad (2.1)$$

where $D$ denotes flexural stiffness and its expression is $D=Eh^3/12(1-\upsilon^2)$ in which $\upsilon$ is Passion ration and $E$ is Young's modulus, $\nabla^2\nabla^2$ is bi-harmonic operator, $\chi$ is viscous damping coefficient, $\rho$ is the density of the plate, $h$ is the thickness of the plate, $F_z(\varphi)$ is milling force which is normal to the plate surface, $P_0$ is the counterforce of the oscillator, $\delta$ is Dirac function, $z(t)$ denotes the distance from the mid-plane of the structure to the mass center of the oscillator.

z($t$) satisfies the following equation.

$$m\frac{d^2}{dt^2}\left(z(t)+w\left(\xi(t),\eta(t),t\right)\right)+c\frac{d}{dt}z(t)+k\left(z(t)-z_0\right)=0 \quad (2.2)$$

where $w(\xi(t),\eta(t),t)$ is the deformation of the milling force applied point.

The solution of Eq. (2.1) can be expanded in terms of plate mode-shapes as [37]

$$w(x,y,t)=\sum_{i=1}^{+\infty}\sum_{j=1}^{+\infty}T_{ij}(t)W_{ij}(x,y) \quad (2.3)$$

where $W_{ij}(t)$ denotes the $(i,j)$ order mode shape, $T_{ij}(t)$ denotes corresponding mode coefficient. Substituting Eq. (2.3) into Eq. (2.1)