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Dynamic characteristic parameters identification analysis of a parallel manipulator with flexible links †

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Abstract

This paper presents dynamic parameters identification analysis of a 3-TPT parallel manipulator with flexible links by means of simulation and experiment. The 3-TPT mechanism is described in detail, and the dynamic equations of the regarded mechanism are derived analytically based on the Newton's second law. The dynamic simulation model of parallel manipulator with flexible links is built by the integration method of virtual prototyping (VP) and finite element analysis (FEA), which forms the flexible multi-body system of parallel manipulator. Frequency response curves are obtained by applied force on the simulation model, and the stiffness and damping ratio are identified, respectively, from the simulation results. The strategy has been experimentally tested on an actual 3-TPT parallel manipulator. The compared results show that the errors are very small and reasonable scope, so it is feasible for the integration method used as reference and basis for dynamic characteristic. Therefore, the integration simulation method provides a theoretical foundation and reference for dynamic optimal design of parallel manipulator. These analyses and results provide valuable insight into the design and control of the parallel manipulator with flexible links.

Keywords: Dynamic parameters identification; Flexible multi-bodies system; Parallel manipulator; Virtual prototype (VP); Finite element analysis (FEA)

1. Introduction

Parallel manipulators with high payload-to-weight ratio, high accuracy, high stiffness, low coarseness and high automation [1] are needed as a requirement of modern manufacture. To obtain the satisfied machining requirements, a parallel manipulator needs to work accurately and efficiently. However, in actual applications, due to inevitable manufacturing tolerances and assembling errors in numerous links and joints, the dynamic model parameters of parallel manipulators are always unequal to the values provided by the manufacturers [2-4]. Meanwhile, the vibration and deformation of a parallel manipulator also have negative influence on the quality of machining [5].

It is necessary to reduce, or possibly, void vibration resulting from manipulator self-oscillation when it works in the range of rated power. Good dynamic characteristic of a parallel manipulator is more important than improvement of machining property. However, the closed-loop mechanism structure of parallel manipulators is much more complex than serial ones, and the kinematics and dynamics of parallel manipulators are very complex. To achieve the potential performance of parallel manipulators, model-based dynamic controllers should be designed elaborately [6]. Therefore, vibration analysis has been an indispensable research of parallel manipulator in design and manufacturing [7-10]. The dynamic performance of a parallel manipulator is typically represented in terms of frequency response functions (FRFs) and stability lobe diagrams (SLDs) for chatter. They experimentally determine machining efficiency and quality [11].

The dynamic parameters which describe the dynamic characteristic of a manipulator are important for control algorithms, system dynamic characteristic and precision and reliability. However, the dynamic characteristic of the mechanism contains uncertainties in many parameters and many control methods are sensitive to their values. Sensitivity to parameter uncertainty occurs especially in high speed operations. Dynamic parameter identification methods have gained importance for developing a model. In general, a standard manipulator identification procedure consists of modeling, experiment design, data acquisition, signal processing, parameters estimation and model validation. If the obtained model does not pass the validation tests, one or several steps of the procedure are repeated and some of the choices are reconsidered. The identification of dynamic parameters has attracted considerable attention from numerous researchers. Atkeson et al. proposed the estimation of inertial parameters. Gautier et al. proposed a

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dynamic identification method from only the torque data. Grot Jahn et al. used the two-step approach to perform the friction and rigid body identification of robot dynamics [12, 13]. There is a method has been used extensively and found to be the best in terms of ease of experimentation and precision of the obtained values [14]. This approach is based on the analysis of the ''input/output'' behavior of the manipulator on some planned motion and on estimating the parameter values by minimizing the difference between a function of the real variables and its mathematical model. Using this method, Guegan et al. [15] identified the base dynamic parameters of an orthoglide parallel kinematic machine. Vivas et al. [16] identified the base dynamic parameters of a parallel robot which are necessary for the dynamic model, and pointed out the use of the accelerometer and rotation sensor was not very necessary. Wu et al. [17-20] added the identified and unidentified dynamic parameters to the position/force switching control system of a redundant machine tool, and the same experiments were performed. The results that the tracking errors are reduced by using the identified dynamic parameters show that the identified results are more accurate.

Although there are many research achievements about dynamic characteristic parameters identification, most of them mainly focus on the analysis of displacement or forces from mechanism without consideration of the rigid and flexible coupling system which includes flexible link deformation and has an important effect on dynamic characteristic of parallel manipulator.

In this paper, a coupling dynamic simulation model with flexible links is built using VP and FEA method; VP technology provides tools to calculate the physical parameters such as mass, length, and inertia from 3-dimensional models. And dynamic characteristic of a kind of 3-TPT parallel manipulator is analyzed by using exciting vibration on the rigid-flexible coupling system simulation model; then the dynamic parameters such as stiffness and damping ratio can be obtained by simulation experiment on simulation model, so it is easy to obtain the parameter values. In the mechanism design phase, the dynamic performance and model-based control performance of the manipulator can be investigated based on the dynamic parameters, and the dynamic performance analysis can be used further to improve the design.

This paper is organized as follows: In Sec. 2, the geometric model of the proposed parallel mechanism is described, following the dynamic equation of mechanism in general, as well as of the exemplarily considered mechanism is also introduced in Sec. 2. In Sec. 3, simulation analysis of dynamic characteristic for parallel manipulator, the integration method of VP and FEA is proposed to form the flexible multi-body system; then simulation results of exciting vibration are analyzed based on the vibration model of parallel manipulator. In Sec. 4, experimental analysis of excitation vibration was conducted, compared with simulation results. Finally, in Sec. 5, some conclusions are given.

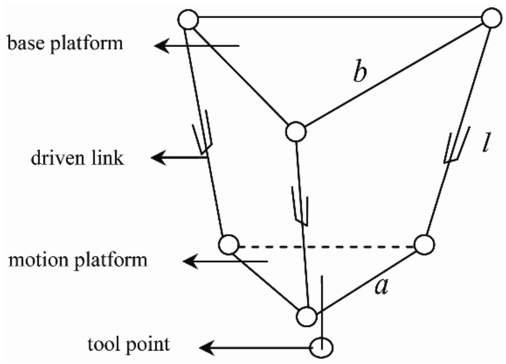


Fig. 1. Simple sketch of parallel manipulator.

2. Model of parallel manipulator

2.1 Geometric model of parallel manipulator

The 3-TPT parallel manipulator is made up of base platform, motion platform and driven links (Fig. 1). The motion platform position can be adjusted by the change length of driven links, which is connected to base platform by three parallel driven links and constraint mechanism. These driven links can control three translational degrees of freedom, which have an important influence on dynamic performance of parallel manipulator, and are regarded as flexible bodies due to their weak stiffness, and the other bodies of parallel manipulator are regarded as rigid bodies. Therefore, the flexible multi-body dynamics of parallel manipulator is formed.

2.2 Dynamic model for parallel manipulator

Modal analysis is a modern method and mean for analyzing dynamic characteristics for most engineering systems, such as mechanical systems, civil engineering structures and bridges and so on. It is a kind of coordinate transformation according its essence, and its purpose is to describe the response vector which was described in original physical coordinate in modal coordinate system, so that the complex actual structure system of a 3-TPT parallel manipulator can be simplified to be a mass-spring-damp system for analyzing and predicting dynamic response [8].

For arbitrary n freedom linear mechanical system, its vibration differential equation can be shown as follows:

$$\mathbf{M}\ddot{\mathbf{x}}(t)+\mathbf{C}\dot{\mathbf{x}}(t)+\mathbf{K}\mathbf{x}(t)=\mathbf{f}(t)\,, \tag{1}$$

or

$$\ddot{x}(t)+2\xi\omega_n\dot{x}(t)+\omega_n^2x(t)=s\omega_n f(t)\,, \tag{2}$$

where $\mathbf{M}$ is mass matrix. $\mathbf{C}$ is stiffness matrix, $\mathbf{K}$ is damping matrix, $\mathbf{f}(t)$ is external excited force, $\mathbf{x}(t)$ is displacement response vector. ω_n is natural angular frequency of system,

$\omega_n = \sqrt{K/M}$, ξ is damping ration of system, $\xi = c/\sqrt{KM}$, s is static sensitivity, $s = 1/K$, for a specific system, s is a constant; so let $s=1$, then Eq. (1) can be expressed as follows:

$$\ddot{x}(t) + 2\xi\omega_n\dot{x}(t) + \omega_n^2 x(t) = \omega_n f(t) . \tag{3}$$

When initial condition is zero, Eq. (3) is transferred by Laplace Transform, and the system transfer function $H(\omega)$ can be written to the following equation:

$$H(S) = \frac{X(S)}{F(S)} = \frac{\omega_n^2}{S^2 + 2\xi\omega_n S + \omega_n^2} . \tag{4}$$

Then we can get the frequency response function as

$$H(\omega) = \frac{1}{1-(\frac{\omega}{\omega_n})^2 + j2\xi(\frac{\omega}{\omega_n})} . \tag{5}$$

Therefore, the amplitude frequency expression and phase frequency expression of whole mechanism can be obtained as follows:

$$A(\omega) = \frac{1}{\sqrt{[1-(\frac{\omega}{\omega_n})^2]^2 + 4\xi^2(\frac{\omega}{\omega_n})^2}} , \tag{6}$$

$$\varphi(\omega) = -\arctan\frac{2\xi(\frac{\omega}{\omega_n})}{1-(\frac{\omega}{\omega_n})^2} . \tag{7}$$

To get the frequency corresponding to the maximum response we should analyze amplitude frequency and phase frequency characteristic curves of single degree of freedom system shown in Fig. 2.

According to the above analysis, the parameters of vibration characteristics can be identified as follows:

(1) Natural frequency

The natural frequency can be identified from the amplitude frequency curves, because the max amplitude is happed on natural frequency [10].

(2) Damping ratio

From Fig. 2 and Eq. (2) we can get the undamped natural frequency:

$$\omega_n = \sqrt{\frac{K}{M}} . \tag{8}$$

The damping ratio is: $\xi = \frac{c}{2M\omega_n}$.

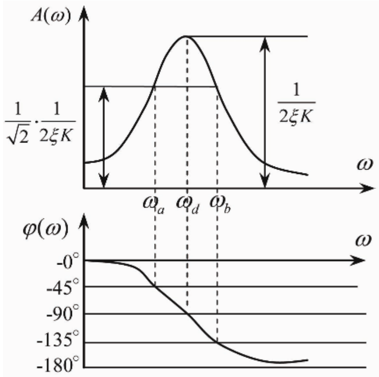


Fig. 2. The amplitude frequency and phase frequency characteristic curves.

The damped natural frequency is: $\omega_d = \omega_n\sqrt{1-\xi^2}$.

Since the damping of a general engineering structure is small $(\xi \le 0.1)$, the damped natural frequency can be considered as follows:

$$\omega_d \approx \omega_0 . \tag{9}$$

From the above analysis we can obtain that the peak value of the curve of amplitude frequency response characteristic is the natural frequency of the system.

For deciding the damping ratio of the system, the half power frequency can be used [12]. On the point of half power, the maximum value of frequency response function can be shown as follows:

$$\frac{1}{M\sqrt{(\omega_n^2-\omega^2)^2 + (2\xi\omega_n\omega)^2}} = \frac{1}{\sqrt{2}}A(\omega) = \frac{1}{\sqrt{2}}\frac{1}{2M\xi\omega_n^2\sqrt{1-\xi^2}} .$$

The two approximate solutions of above equations are:

$$\omega_a \approx \omega_n\sqrt{1-2\xi} \approx \omega_n(1-\xi) . \tag{10}$$

Then, the maximum value of transfer function is:

$$A(\omega)_{\max} = \frac{1}{4K\xi(1-\xi)} , \omega_b \approx \omega_n\sqrt{1+2\xi} \approx \omega_n(1+\xi) .$$

The maximum value of transfer function is:

$$A(\omega)_{\max} = \frac{1}{4K\xi(1+\xi)} . \tag{11}$$

The damping ratio is:

$$\xi = \frac{\omega_b - \omega_a}{2\omega_n} . \tag{12}$$

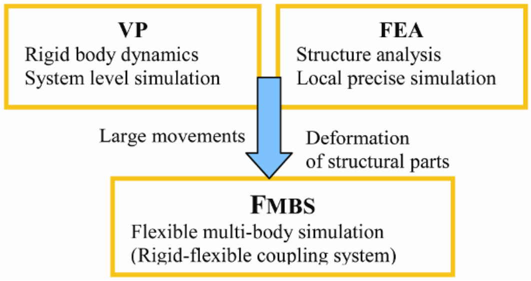


Fig. 3. Integration of FEA and VP in one system.

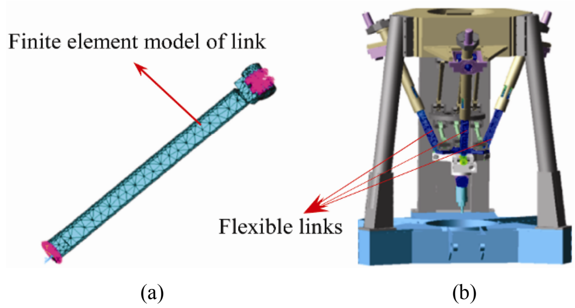


Fig. 4. The model of parallel manipulator with flexible links.

The stiffness of the system is:

$$K = \frac{1}{2A\xi} . \quad (13)$$

According to the above analysis, in the vibration testing experiment, the natural frequency can be tested from the curves of amplitude frequency characteristic and the point of half power also can be found, so according to the above computing formulas, the parameters of vibration characteristic can be obtained.

3. Simulation analysis of dynamic characteristic for parallel manipulator

3.1 The simulation model by using VP and FEA

It is an effective approach for structure optimization and test of dynamic characteristic to closely combine theoretical analysis, simulation and experiment. A virtual prototype of a parallel manipulator is a computer simulation model of the physical product that can be presented, analyzed, tested and evaluated. Its core parts are rigid multi-body system kinematics, dynamics modeling and corresponding numerical algorithm, and it can realize large movements and be analyzed on system-level. In rigid multi-body system, parts have mass and inertia properties but no deformation. To transform existing geometry models, VP and FEA provide standard formats to be imported from 3D-CAD-Models. It makes it possible to integrate flexible bodies in rigid multi-body system. Small linear deformations could be included in the system and the advantages of both FEA and VP have to be integrated [21] during the simulation of a parallel manipulator. The flexible multi-body system is used to simulate the dynamic behavior of parallel manipulator. To get dynamic characteristics of a parallel manipulator exactly, the correct simulation is the integration of large movements under consideration of small deformations in the structural components [22]. Virtual prototype and the finite element method are integrated in one system as shown in Fig. 3.

3.2 The dynamic simulation model with rigid-flexible coupling system

As is known, a parallel mechanism is a polycyclic closed chain mechanism of complex multi-body system, so its dynamic modeling is complex. At present, the establishment of dynamics equations of parallel manipulators is always put forward with considering the whole mechanism as a multi-rigid-body system, which is not considering the deformation of slender bodies. Moreover, the elastic deformation of slender bodies has a significant effect on the output response of the mechanism system; if it is dealt with simplified as a rigid body, the actual results of parameters identification will have bigger errors. Therefore, using rigid bodies and flexible bodies coupled dynamics simulation is closer to the real situation.

Finite element models of flexible links are divided into thousands of elements with proper element type, and its attributes and corresponding parameters are set [23]. For connecting rigid bodies with flexible bodies, the exterior nodes of links of parallel manipulator are set by finite element method, which are built on the center of relative gyration of flexible body to other parts in a mechanical system [24], and dealt with by setting rigid region in Fig. 4(a). When the MNF files are generated they can be transformed into VP for simulation. After MNF files are introduced into VP, the motion pairs are set up between exterior nodes and rigid bodies to connect flexible bodies and rigid bodies. When the flexible bodies are joined in other rigid bodies, we apply drive forces and control procedure on the coupling system and carry out the simulation. The model of rigid-flexible coupling system of parallel manipulator, and the color change stands up the value of stress change in motion process in Fig. 4(b).

The excitation vibration model will be built based on a rigid-flexible coupling system. There are three basic building blocks in a vibration model: input channels, output channels, and actuators. Actuators are vibrate or drive the system in the frequency domain. The actuators ‘act’ on the input channels you specify. Input channels define location and direction of vibratory input. They accept actuators as sources of vibratory input and are used to plot. Output channels measure vibratory response. The system response and report results can be measured in the frequency and the model of excitation vibration of parallel manipulator as shown in Fig. 5.

3.3 Parameter identification results of simulation

There are two kinds of milling type of a parallel manipula-

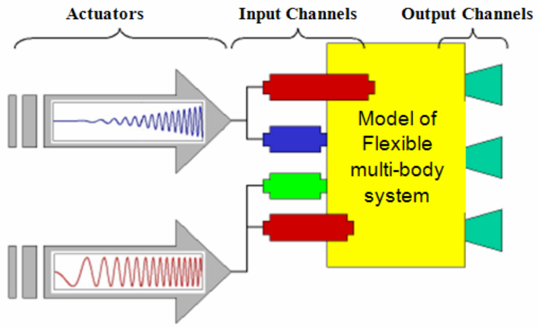


Fig. 5. Model of manipulator excitation.

tor: parallel manipulator without load, which the links speed and driven force are being measured in certain speed and acceleration condition; and feed with uniform speed and bear milling force. Milling force can be calculated by the corresponding equation, the main milling force F_t can be calculated by the following equation [25]:

$$F_t = C_F a_e^{1.1} f_z^{0.8} d_0^{-1.1} a_p^{0.95} z \ , \tag{14}$$

where C_F is coefficient of milling force; a_e is milling width(mm); a_p is milling depth(mm); f_z is the feed of every gear tooth(mm/z); d_0 is the outer diameter of milling tool(mm); z is tooth number of milling cutter.

The initial condition of cutting parameters is given by: $C_F = 808$, $a_e = 12$ mm, $a_p = 2$ mm, $f_z = 0.1$ mm/z, $d_0 = 16$ mm, $z = 6$.

Milling force can be calculated according to Eq. (14) and initial condition: $F_t \approx 1060N$, the magnitude of $f(t)$ in the Eq. (1). And the force F_t is applied on tool point as external excited force according to above calculation when simulation is carried out. The initial phase is 0, and the vibration analysis is carried out with rapid scanning. The displacement response can be obtained in X, Y, Z direction and space; the results are shown in Figs. 6 and 7 (the frequency range is from 0.01 to 800 Hz).

The frequency response function of tool point is shown in Fig. 6. The maximum displacement responses in X direction occurred on 129 Hz; the maximum displacement responses in Y direction occurred on 129 Hz, and displacement response was 0.035 mm; the maximum displacement responses in Z direction occurred on 129 Hz, and displacement response was 0.015 mm; the maximum displacement responses in space occurred on 129 Hz, and displacement response was 0.052 mm.

The maximum value of transfer function can be obtained according to Eq. (11), and the damping ratio also can be calculated according to Eq. (12). The stiffness of the system can be obtained according to Eq. (13).

From Fig. 7, we can get the numbers as follows:

$$\omega_0 = 129Hz, \mathrm{H}_0 = 0.045mm$$
$$\omega_a = 126Hz, \omega_b = 132Hz \ .$$

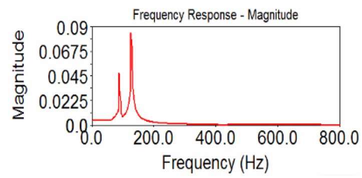


Fig. 6. Frequency response of tool point.

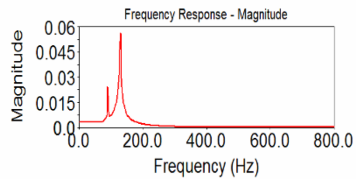


Fig. 7. Frequency response of motion platform.

According to Eqs. (11)-(13), the stiffness and damping can be obtained as follows:

$$k = 4.78E5(N/m); \xi = 0.0232 .$$

The frequency response function of the motion platform is shown in Fig. 7. The natural frequency and amplitude can be obtained as follows:

$$\omega_0 = 129Hz, \mathrm{H}_0 = 0.032mm$$
$$\omega_a = 126Hz, \omega_b = 132Hz \ .$$

According to Eqs. (11)-(13), the stiffness and damping can be obtained as follows:

$$k = 1.12E6(N/m); \xi = 0.0279 .$$

From the above simulation results, the natural frequency of the 3-TPT parallel manipulator is about 129 Hz, so this frequency should be avoided in the parallel manipulator working process.

4. Experimental analysis of excitation vibration

The modern parallel manipulator is developed towards high speed, high precision and high productivity, which requires that the parallel manipulator structure must have good dynamic characteristics. Test modal analysis has an important significance for researching the dynamics of structure and optimization design [26]. The manipulator is a complex assembly, including many parts; its internal includes a joint surface between parts, so it is hard to solve dynamic characteristics accurately only relying on theoretical analysis [27]. The experiment modal analysis is a kind of research method of dynamic characteristics in frequency domain for parallel manipulator. Its features combine theoretical analysis and experiment tests and carrying out curve fitting between external excited force and response transfer function. It

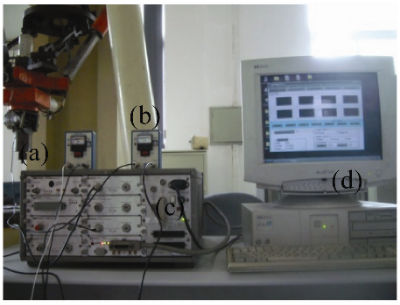


Fig. 8. The apparatus and equipments of experiment.

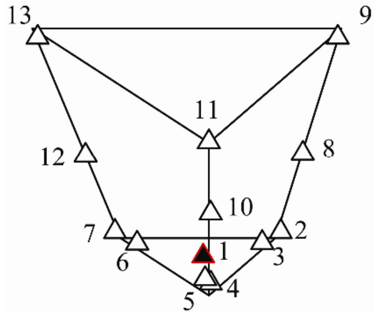


Fig. 9. The disposal position of test points.

utilizes modal parameter identification for getting the frequency and mode shape of a parallel manipulator.

4.1 Experimental setup and measuring

There are some experimental equipments and parallel manipulator as shown in Fig. 8: (a) accelerometer (CL-YD-141); (b) charge amplifier (YE5852A); (c) dynamic signal analyzer (HP3566A/3567A); (d) computer; (e) excitation force hammer(PZT 291M); (f) force sensors (CA-YD-141); (g) power amplifier (YE5873); (h) signal conditioning and amplifiers (YE5852A).

Generally, there are two measuring methods for a modal test. First, the constantly moving three-direction acceleration sensors are measuring the output response signal by using a single point fixed pulse hammer excitation; second, the various points are measured by the constantly moving hammer under the fixed three-direction acceleration sensors. The second method is used in the modal test.

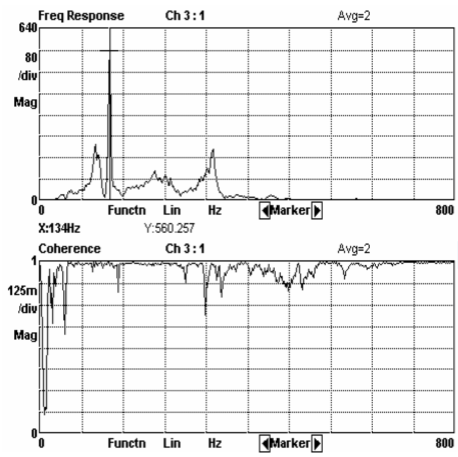
To fully reflect the dynamic characteristics of a parallel manipulator, based on the characteristics of the machine structure, the selected 13 measuring points and the excitation point are arranged on the parallel manipulator as shown in Fig. 9. Point 1 is the response point of the X, Y and Z direction for picking up a signal. The selected measurement points in the model are parallel manipulator structure feature nodes; these points can determine the location and the order of the venue and percussion. They are arranged by the following principles: measuring points in principle correspond with the theoretical model node to facilitate the theoretical calculations and comparison of the measured structure. According to test experience, if the conditions permit, the measurement points can be increased appropriately at the key points. The measurement sensors arranged on parallel manipulator are shown in Fig. 10.

4.2 Experimental results of excitation vibration

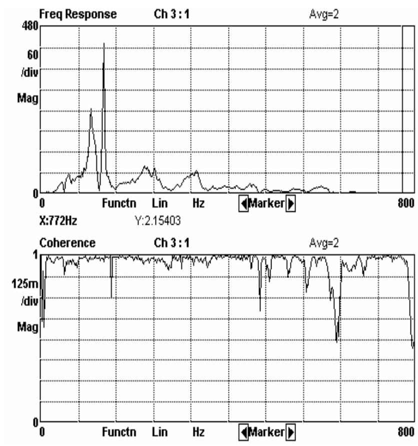
Suitable results should be chosen by judgment due to the randomness of the experiment result. Many tests should be done at every testing point and the results recorded if the value of the coherence function is close to 1. The results recorded are shown in Fig. 11.

The displacement responses of tool point and motion plat-

Fig. 10. The disposal position of sensors on parallel manipulator.



(a) Tool point



(b) Motion platform

Fig. 11. Experimental results of frequency response function.

Table 1. Comparisons of the results of simulation and experiment.

		Natural frequency (First modal order)	Natural frequency (Second modal order)
Simulation	Tool point	96	129
	Motion platform	96	129
Experiment	Tool point	105	134
	Motion platform	105	133
Error	Tool point	8.5%	3.71 %
	Motion platform	9.5%	3.75%

form are shown in the frequency range from Hz 0.01 to 800 Hz in Fig. 11. Frequency responses of tool point and motion are shown, respectively, in Figs. 11(a) and (b). The maximum displacement responses of tool point occurred on 134 Hz and displacement response was 0.0428 mm. The maximum displacement responses of motion platform occurred on 133 Hz, and the displacement response was 0.052 mm. From Fig. 11, we can get the numbers as follows:

$$\omega_0 = 134Hz, \mathrm{H}_0 = 0.052mm$$
$$\omega_a = 136Hz, \omega_b = 132.5Hz .$$

According to Eqs. (11)-(13), the stiffness and damping are got obtained as follows:

$$k = 7.39E5(N/m); \xi = 0.0130 .$$

The frequency response function of motion platform is shown in Fig. 11(b). The natural frequency and amplitude can be obtained based on the above analysis.

$$\omega_0 = 133Hz, \mathrm{H}_0 = 0.0428mm$$
$$\omega_a = 131Hz, \omega_b = 135Hz .$$

According to Eqs. (11)-(13), the stiffness and damping are got as follows:

$$k = 7.79E5(N/m); \xi = 0.015 .$$

The natural frequency is close to simulation results, so it can be noticed that the simulation results are smaller than experiment.

The stiffness of the parallel manipulator needs to be increased to improve machining precision from the identification results. The natural frequency between simulation and experimental results is shown in Table 1. The test points cannot affect the whole dynamic characteristic of parallel manipulator in theory analysis and simulation process. But it has some difference in the experimental results, due to the different configuration and endured force. It shows that the error of comparison result simulation with experiment is very small and reasonable scope, so it is feasible that the integration method is used as reference and basis for dynamic characteristic.

5. Conclusions

(1) An integration method of VP and FEA in one system is proposed for dynamic characteristic analysis of parallel manipulator based on the flexible multi-body dynamics system. The compared results show that the simulation results are validated by using the excitation vibration method in experiment process, which provides an efficient way to study theoretical foundation and reference for dynamic optimal design of other mechanical products.

(2) The stiffness and damping ratio were identified, respectively, from the simulation and experiment results, which can be provide reference and basis for structure design and dynamic characteristic analysis of parallel manipulator.

(3) The dynamics modeling of rigid-flexible coupling multibody system is based on the assumption that the system's components are perfect. However, friction, material nonuniformities and manufacturing and assembly errors in actual systems have very important effects on simulation results.

(4) The precision of the model in CAD system determines the accuracy of the estimated parameters. Due to the manufacturing error of the parts, the CAD model is not identical with the real parts and the accuracy of the estimated parameters values is affected.

Acknowledgment

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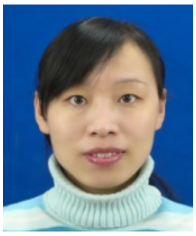

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