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Research Article

Research on Modal Parameters Identification of Parallel Manipulator with Flexible Multi-Body System

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Abstract: In this study, a new method based on simulation is proposed. And the analysis method based on flexible multi-body system of parallel manipulator is provided in the same time. Firstly, modal analysis principle of parallel manipulator was analyzed in theory and the parameters of dynamic characteristic were identified by theoretical analysis. Then vibration model of flexible multi-body for parallel manipulator was built in virtual prototype software and formed vibration system of rigid and flexible coupling for simulation analysis and from the simulation results got the value of parameters for vibration characteristic of parallel manipulator. And the dynamic characteristic parameters were identified according to the simulation results. The results showed that the simulation method and result dates are validated. So the integration simulation method is feasible, which can provide reference for dynamic optimal design.

Keywords: Dynamic characteristic, flexible multi-body system, parallel manipulator, parameters identification

INTRODUCTION

The vibration and deformation of parallel manipulator will be having important effect on quality of machining (Xu and Jiao, 1998). The parallel manipulator with high precision, low coarseness and high automation are needed as the requirement of modern manufacture. It is a main research work to reduce and void vibration for ensuring the manipulator will not generate self-oscillation when it works in the range of rated power (Sakamoto, *et al.*, 2012). Many study shows that good dynamic characteristic of parallel manipulator will be more needed and more important with gradually improvement of machining property (Jinwook *et al.*, 2000), so parameters identification analysis of dynamic characteristic is more important.

Vibration analysis have been an indispensable research content for parallel manipulator design, manufacture and use and is paid more attention as an important aspect of dynamic characteristic (Yu *et al.*, 2004). The experience and repeated measured are main means of traditional structure design while considering dynamic factor, it greatly slowdown the design speed and the design quality also hard to be optimization (Kris *et al.*, 2004). It is inevitable trend to analyze vibration characteristic of parallel manipulator and replace static design by dynamic design.

Virtual prototype simulation technique is one of main ways for structure dynamic design; it can solve some main problems for structure dynamic design (Zhu *et al.*, 2011). In this study this method is been considered for analyzing dynamic characteristic of a kind of 3-TPT parallel manipulator and the results can be used for optimizing structure, then the design performance of parallel manipulator will be improved and design cycle will be shortened.

In this study, a new method based on simulation is proposed. And the analysis method based on flexible multi-body system of parallel manipulator is provided in the same time. Firstly, modal analysis principle of parallel manipulator was analyzed in theory and the parameters of dynamic characteristic were identified by theoretical analysis. Then vibration model of flexible multi-body for parallel manipulator was built in virtual prototype software and formed vibration system of rigid and flexible coupling for simulation analysis and from the simulation results got the value of parameters for vibration characteristic of parallel manipulator. And the dynamic characteristic parameters were identified according to the simulation results. The results showed that the simulation method and result dates are validated. So the integration simulation method is feasible, which can provide reference for dynamic optimal design.

MODAL ANALYSIS PRINCIPLE FOR PARALLEL MANIPULATOR

Basic theory: Modal analysis is a modern method and mean for analyzing dynamics for most engineering

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system, such as mechanical system, civil engineering structure and bridge and so on Zhang *et al*. (2010). Modal analysis 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 structures system can be simplified to be a modal model for analyzing and predicting modal response (Li *et al*., 2011).

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

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

where, M is mass matrix. C is stiffness matrix, K is damping matrix, $f(t)$ is external excited vector, $x(t)$ is displacement response vector.
Eq. (1) is transferred by Fourier transform:

$$[-\omega^2\boldsymbol{M}+j\omega\boldsymbol{C}+\boldsymbol{K}]X(\omega)=F(\omega) \tag{2}$$

where,
ω = A variable
$X(\omega)$ = Laplace transform for $x(t)$
$F(\omega)$ = Laplace transform for $f(t)$

The response of system is :

$$X(\omega)=[-\omega^2\boldsymbol{M}+j\omega\boldsymbol{C}+\boldsymbol{K}]^{-1}F(\omega)=H(\omega)F(\omega) \tag{3}$$

where, $H(\omega)$is displacement transfer function matrix.

The system response in modal principle can be written as follows:

$$X(\omega)=\sum_{r=1}^{n}\frac{\{\boldsymbol{\psi}_r\}^T\{\boldsymbol{F}\}\{\boldsymbol{\psi}_r\}}{-\omega^2 m_r+jc_r\omega+k_r} \tag{4}$$

where,
m_r = Modal mass
c_r = Modal damping
kr = Modal stiffness
$\{\psi_r\}$ = System mode vector

For the equation of relationship is more visual between response and external force, Eq. (4) can be expressed in matrix form as follows:

$$X(\omega)=\sum_{r=1}^{n}\frac{\{\boldsymbol{\psi}_r\}^T\{\boldsymbol{\psi}_r\}}{-\omega^2 m_r+jc_r\omega+k_r}\{\boldsymbol{F}\} \tag{5}$$

The transfer function form which is showed the relationship between external force on point j and response on point i can be expressed as follows:

$$H_{ij}=\frac{X_i}{F_j}=\sum_{r=1}^{n}\frac{\psi_{ir}\psi_{jr}}{-\omega^2 m_r+jc_r\omega+k_r}=\sum_{r=1}^{n}\frac{\psi_{ir}\psi_{jr}/m_r}{-\omega^2 m_r+j2\zeta_r\omega\omega_r+\omega_r^2} \tag{6}$$

And after coordinate transformation, the equation can be expressed:

$$[x]=\sum_{i=1}^{N}q_i\{\varphi_i\} \tag{7}$$

In Eq. (7), $\{\varphi_i\}$is i th order vibration mode which is determined according to $[M]\{\ddot{\boldsymbol{x}}\}+[K]\{\boldsymbol{x}\}=0$.Bring Eq. (4) into (1) the following equation can be expressed:

$$[M]\left(\sum_{i=1}^{N}\ddot{q}_i\{\varphi_i\}\right)+[C]\left(\sum_{i=1}^{N}\dot{q}_i\{\varphi_i\}\right)+[K]\left(\sum_{i=1}^{N}q_i\{\varphi_i\}\right)=\{f(t)\} \tag{8}$$

Eq. (8) left multiplicity $\{\varphi_j\}^T$ according to the orthogonality of modal shape vector about mass matrix [M] and stiffness matrix [K]. Then the following equation can be got:

$$\begin{aligned}\{\varphi_j\}^T[C]\{\varphi_i\}&=\alpha\{\varphi_j\}^T[M]\{\varphi_i\}+\beta\{\varphi_j\}^T[K]\{\varphi_i\}\\&=\begin{cases}0, i\neq j\\ \alpha m_j+\beta k_j=c_j, i=j\end{cases}\end{aligned} \tag{9}$$

where,
m_j = Modal mass
k_j = Modal stiffness
c_j = Modal damping coefficient

Then Eq. (9) can be simplified as follows:

$$m_j\ddot{q}_j+c_j\dot{q}_j+k_jq_j=\{\varphi_j\}^T\{\boldsymbol{f}(t)\} \tag{10}$$

Set $\{f(t)\} = \{F\}e^{j\omega t}$, then $q_i = Q_je^{j\omega t}$, put it into Eq. (10):

$$(-\omega^2 m_j+j\omega c_j+k_j)Q_je^{j\omega t}=\{\varphi_j\}^T\{\boldsymbol{F}\}e^{j\omega t} \tag{11}$$

$$Q_j=\frac{\{\varphi_j\}^T\{\boldsymbol{F}\}}{-\omega^2 m_j+j\omega c_j+k_j} \tag{12}$$

If the initial conditions not be considered, the displacement response of system can be got from Eq. (7):

$$\{\boldsymbol{x}\}=\begin{Bmatrix}X_1\\X_2\\\vdots\\X_N\end{Bmatrix}e^{j\omega t}=\sum_{i=1}^{N}q_i\{\varphi_i\}=\sum_{i=1}^{N}Q_i\{\varphi_i\}e^{j\omega t} \tag{13}$$

$$\begin{aligned}\{\boldsymbol{X}\}=\begin{Bmatrix}X_1\\X_2\\\vdots\\X_N\end{Bmatrix}&=\sum_{i=1}^{N}Q_i\{\varphi_i\}=\sum_{i=1}^{N}\frac{\{\varphi_j\}^T\{\boldsymbol{F}\}}{-\omega^2 m_j+j\omega c_j+k_j}\\&=\sum_{i=1}^{N}\frac{\{\varphi_j\}\{\varphi_j\}^T}{-\omega^2 m_j+j\omega c_j+k_j}\{\boldsymbol{F}\}\end{aligned} \tag{14}$$

From above equation, its denominator is irrelevant to excited point j and response point i and only relevant to frequent and damping. So the transfer function and its denominator are all same in spite of any measured point in mechanical system. The numerator of Eq. (14) determined by modal mass, vibration modal value of excited point and response point, Above equation shows that the transfer function includes all parameters of dynamics, modal parameters are needed to be identified after getting transfer function, namely, measured value theoretical formula of transfer function are carried out curve fitting to get modal parameters such as nature frequency, damping ratio and mode shape.

The parameters of vibration characteristics can be identified as follows:

- **Natural Frequency:** The natural frequency can be identified from the amplitude frequency curves, because the max amplitude is happed on natural frequency.
- **Damping Ratio:** From above analysis, the transfer function of system according to Eq. (6) is showed as following:

$$H(\omega)=\frac{X(\omega)}{F(\omega)}=\sum_{i=1}^{N}\frac{\{\varphi_j\}^{\mathrm{T}}}{-\omega^2 m_j+j\omega c_j+k_j}=\sum_{i=1}^{N}\frac{\{\varphi_j\}\{\varphi_j\}^{\mathrm{T}}}{-\omega^2 m_j+j\omega c_j+k_j}$$

For getting the frequency which corresponding the maximum response, the undamped natural frequency is:

$$\omega_0=\sqrt{\frac{k}{m}}=2\pi f_0$$

The damping ratio is:

$$\xi=\frac{c}{2m\omega_0}$$

The damped natural frequency is:

$$\omega_d=\omega_0\sqrt{1-\xi^2}$$

Due to the damping of general engineering structure is small ($\xi \le 0.1$), so the damped natural frequency can be considered as followed:

$$\omega_d \approx \omega_0$$

From above we can got that: the peak value of curve of amplitude frequency response characteristics is the natural frequency of system.

For deciding the damping ration of system, it is can be got by the half power frequency. On the point of half power, the maximum value of frequency response function can be showed as follows:

$$\frac{1}{m\sqrt{(\omega_0^2-\omega^2)^2+(2\xi\omega_0\omega)^2}}=\frac{1}{\sqrt{2}}\left|H(\omega)\right|=\frac{1}{\sqrt{2}}\frac{1}{2m\xi\omega_0^2\sqrt{1-\xi^2}}$$

The two approximate solutions of above equations are:

$$\omega_a\approx\omega_0\sqrt{1-2\xi}\approx\omega_0(1-\xi)$$

Then, the maximum value of transfer function is:

$$H_{\max}=\frac{1}{4k\xi(1-\xi)},\ \omega_b\approx\omega_0\sqrt{1+2\xi}\approx\omega_0(1+\xi)$$

Then, the maximum value of transfer function is:

$$H_{\max}=\frac{1}{4k\xi(1+\xi)}$$

Then the damping ration is:

$$\xi=\frac{\omega_b-\omega_a}{2\omega_0}$$

The stiffness of system is:

$$k=\frac{1}{2A\xi}$$

So, in the vibration testing experiment, the natural frequency can be tested, from the curves of amplitude frequency characteristic, the point of half power is found out and according above computing formulas, the parameters of vibration characteristic can be got.

SIMULATION ANALYSIS OF VIBRATION CHARACTERISTIC FOR PARALLEL MANIPULATOR

It is an effective approach for structure optimization and test of dynamic characteristic to combine closely theoretical analysis, experiment and simulation, ADAMS software adopted virtual prototype technique can solves effectively above problems. The ADAMS/Vibration module is used for test forced modal shapes and analyzing frequency domain (Zhu *et al*., 2009), it can provide the function multi-input and multi-output and can describe them in vibration form, so this module is used for making analysis of vibration simulation. And the model of parallel manipulator with multi-body system for parallel manipulator is showed in Fig. 1.

On the design stage, it is can pre-evaluate dynamic performance of parallel manipulator to utilize virtual prototype to simulate, so that parallel manipulator product can be analyzed for finding structure weakness, revision and optimization, then put forward improvement suggestions for perfecting the prototype of parallel manipulator. This kind of virtual prototype

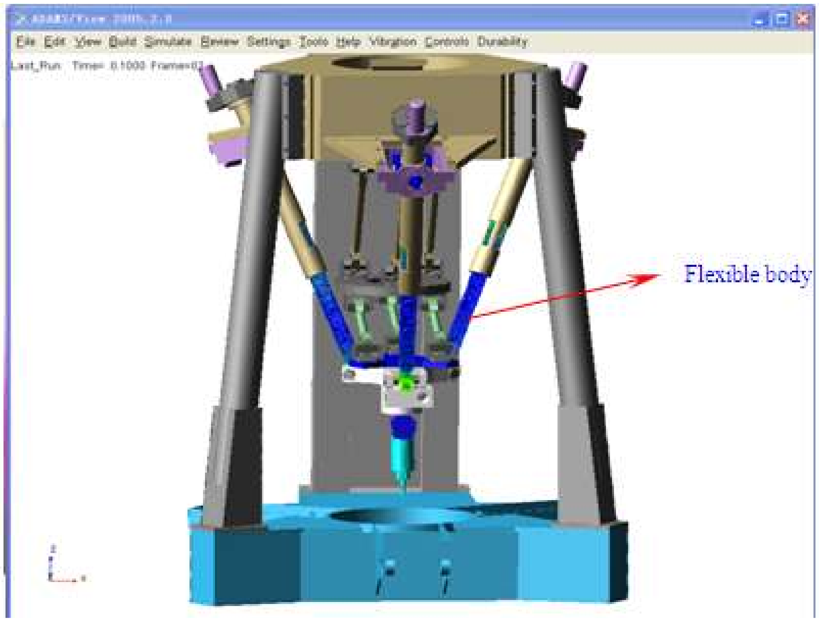

Fig. 1: The model of parallel manipulator with multi-body system

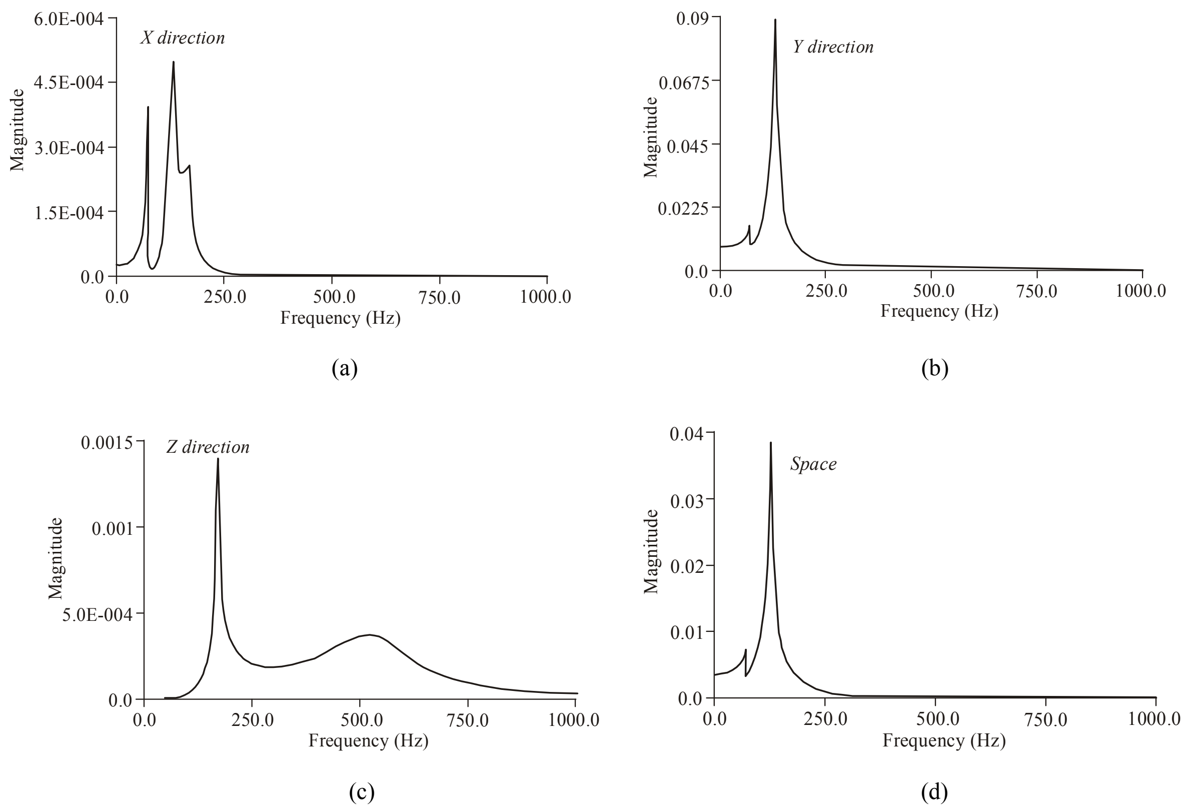

Fig. 2: Frequency response of motion platform

simulation technique proceeds from multi-body dynamics analysis and utilizes computer simulation and operation to get the results, its model needs to be built and improved according to the measurement results for assuring it conform to reality.

There are two milling types when simulating work situation of parallel manipulator, one is parallel manipulator is working with no-load, under this condition, the links speed and driven force are be measured when tool is moving with certain speed and

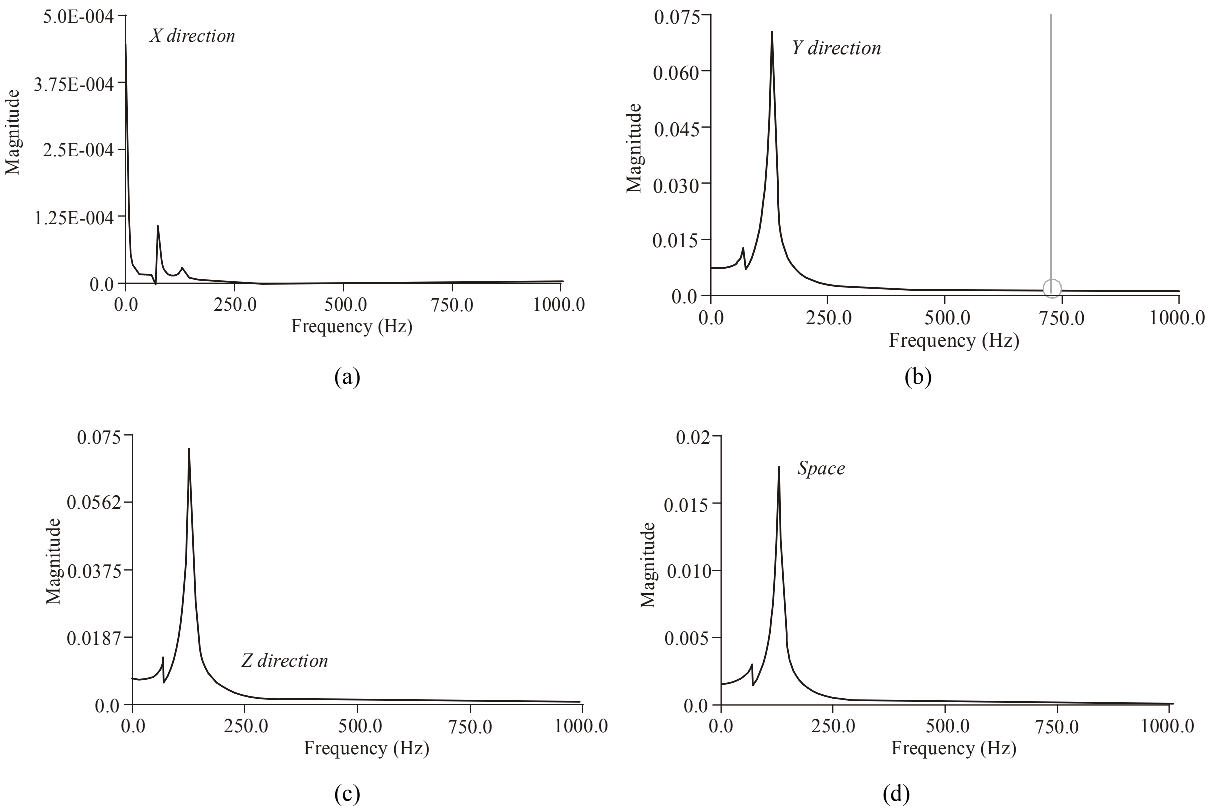

Fig. 3: Frequency response of translation platform

acceleration; the other is tool feed with uniform speed and bear milling force. For milling tool with different materials process different workpiece, when working parameters are determined, the milling force can be calculated by corresponding equation (Zhang *et al*., 2009), take milling tool with W18Cr4V and workpiece with 45# for example, the main milling force F_t can be calculated by the following equation:

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

where,

C_F = Coefficient of milling force, it is 808
α_e = Milling width (mm)
α_p = Milling depth (mm)
f_z = The feed of every gear tooth (mm/z)
d_0 = The outer diameter of milling tool (mm)
z = Tooth number of milling cutter

According to the structure parameters of parallel manipulator, the end milling tool and workpiece with steel 45# are chosen, then corresponding processing parameters are:

$C_F = 808$, $\alpha_e = 12$ mm, $\alpha_p = 2$ mm, $f_z = 0.1$ mm/z,d_0 = 16 mm, z = 6,So $F_t \approx 1060$N

The force 1000 N is applied on tool tip as external excited force according to above calculation when simulation is carried out. The initial phase is 0, the vibration analysis is carried on with rapid scanning style. ADAMS software can output the displacement response in X, Y, Z direction and space, the results are showed in Fig. 2 to 3 and the frequency range is from 1 to 1000 Hz.

In Fig. 2 it shows the frequency response of motion platform, the maximum displacement responses in X direction occurred on 128Hz, displacement response is 0.00048 mm; the maximum displacement responses in Y direction occurred on 130 Hz, displacement response is 0.09 mm; the maximum displacement responses in Z direction occurred on 130Hz, displacement response is 0.015 mm; the maximum displacement responses in space occurred on 130Hz, displacement response is 0.035 mm:

$$\omega_{01} = 125.2Hz; \ \ \mathrm{H}_{01} = 0.037mm$$
$$\omega_{a1} = 122.8Hz, \omega_{b1} = 127.2Hz$$

Stiffness and damping are as following:

$$k_{01} = 7.69E5(N/m); \xi_{x1} = 0.0176;$$

In Fig. 3 it shows the frequency response of translation platform, the maximum displacement responses in X direction occurred on 10Hz, but the value of displacement response is very small and can be neglected; the maximum displacement responses in Y direction occurred on 130 Hz, displacement response is 0.07 mm; the maximum displacement responses in Z direction occurred on 130Hz, displacement response is 0.072 mm; the maximum displacement responses in space occurred on 130Hz, displacement response is 0.0175 mm:

$$\omega_{01} = 125.4Hz; \quad H_{01} = 0.0174mm$$
$$\omega_{a1} = 123.2Hz, \omega_{b1} = 127.6Hz$$

Stiffness and damping are as following:

$$k = 1.64E6(N/m); \xi = 0.0175;$$

From above simulation results, the following conclusion can be go: the resonance frequency of 3-TPT parallel manipulator is 130Hz in the range from 1 Hz to 1000Hz, so this frequency should be avoid when parallel manipulator is working.

CONCLUSION

In this study, we want to seek for parameters identification analysis method for parallel manipulator. From this analysis process and simulation, we endeavor to seek some common modeling methods of identification for multi-body system of parallel manipulator. And in the course of this exploration, we are able to see that simulation method with virtual prototype technology is inevitable trend.

The simulation is carried out and get the resonance frequency is on 130Hz and displacement response is 0.035 mm; It shows that the simulation results are high precision according experiment results; it can be provide reference and basis for parallel manipulator structure design and dynamic characteristic analysis.

The simulation technique of parameters identification for parallel manipulator has some advantages, such as lower cost and shorter period compared with experiment, so it has reference value for related measuring. But the dynamics modeling of rigid-flexible coupling system is based on the assumption that the system's components are perfect. However, friction, material non-uniformities and manufacturing and assembly errors in actual systems have very important effects on simulation results.

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