STRUCTURAL OPTIMIZATION OF A HANDHELD TILLER HANDRAIL BY VIBRATION MODAL ANALYSIS
/
基于振动模态分析的微耕机扶手架结构优化

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ABSTRACT

Exposure to severe vibration while tilling presents a serious health and safety risk to a handheld tiller operator. Handrail of the tiller transfers vibration from tiller to operator with an essential extent. Taking handrail of Shineray SR1Z-80 tiller as a case study, its vibration characteristics were investigated by means of Computational Modal Analysis and Experimental Modal Analysis, resulting in good agreement on natural frequency and mode shape within interested frequency range of the vibration, and the average relative error of natural frequency between them is less than 4%. Handle, rod and cross bar are main sections of the handrail that affect the vibration with decreasing level of influence regarding displacements. Structural optimization of the handrail was obtained by orthogonal factorial experiment design technique and statistical analysis of variance. The optimal dimensions of the handrail are handle length 170 mm, rod length 470 mm, and cross bar length 390 mm.

摘要

耕作时的强烈振动对微耕机操作者造成严重的健康和安全危险。其中大部分的振动由扶手架传递给微耕机的操作者。本研究以鑫源 SR1Z-80 微耕机的扶手架为例，通过计算模态分析和试验模态分析研究了其扶手架的振动特性。在关注的频率段内，两种模态分析方法得到的扶手架固有频率和模态振型一致性很好，其中固有频率的相对误差小于 4%。扶手、扶手杆和横杆是影响扶手架振动性能的主要部分，他们对振动的影响程度（位移）依次降低。通过正交试验设计和方差分析，得到了扶手架的优化结构，其对应的优化尺寸参数为：扶手长度 170mm，扶手杆长度 470mm，横杆长度 390mm。

INTRODUCTION

Handheld tillers are typically propelled forward by the diesel or gasoline engine rotating rotavators and the power of the engine is normally less than 7.5 kW. The handheld tillers are widely used in hilly lands, greenhouses and orchards, due to their advantages of small volume, light weight, simple structure and easy transfer in the farm land. But for a handheld tiller, the connections between subsections (e.g. engine and transmission, transmission and frame, transmission and handrail, etc.) are rigid, thus operators are wholly exposed to severe vibration transmitted directly to their hands. Resulting from the engine, tilling rotavators and their coupling, the vibration is often pretty complex, contains many frequencies, occurs in several directions and changes over time, and it may cause injuries to sensory nerves, the vascular, muscles, bones and joints, influence the operators' performance capability, or present a serious health and safety risk to the operators *(Su et al., 1989)*. Under usual working conditions, in 10% of the exposed operators, vascular disorders of the hand, namely "vibration white finger", can appear after three years of continuous use of handheld tillers *(Ragni et al., 1999)*.

Many scholars studied the vibration characteristics of a handheld tiller handrail for better comfort level while operating. Goglia *et al.* (2004) studied the vibration of a small agricultural tractor transmitted to the driver's hands and body during work.

The transfer mechanism and the effects of vibration transmitted were analysed. *Li et al*. (2016) measured the acceleration at the handles of a handrail and engine cover along different directions under four working conditions, namely no-load idle speed, no-load half-throttle, no-load full throttle, and field operation full throttle. Yang and Meng (2005) employed the virtual prototype technology to study the vibration of the

GN31 cultivator and found that the vibration at the handles were reduced effectively by adding an object (1.5-3.0 kg) to the handrail-rod near the handrail cross-bar, with 79%-88% maximum acceleration reduction. Ying *et al.* (1994) designed a rubber vibration absorber on the handles of GN-5 walking tractor by increasing the damping and decreasing the vibration transmission. The vibration absorber reduced the acceleration by 41.1% and prolonged the tolerable time of daily exposure to vibrations by 126.4%. Yang (2005) developed a vibration isolation handle for the small-sized tiller on the basis of vibration mechanism, with 50%-80% reduction of vibration transmitted to the handle.

Currently, most studies of the vibration while tilling are focused on observation, evaluation and influence of vibration on the operators, etc., and the study on the handrails ergonomics is insufficient to provide the required information so as to decrease their vibration sensitivity and to improve tilling operation comfort. Taking the handrail of an SR1Z-80 handheld tiller as a case study, on the basis of vibration characteristics analysis and the corresponding statistics analysis of the handrail, structural optimization of the handrail was conducted, so as to provide available design theory and reference to develop new high quality tillers with features of high efficiency, low labour intensity, and good operation comfort.

MATERIAL AND METHODS

The SR1Z-80 handheld tiller was manufactured by Chongqing Shineray Agricultural Machinery Co., Ltd. The vibration characteristics of its handrail were studied by means of Computational Modal Analysis and Experimental Modal Analysis. The model parameters of these two modal analyses such as natural frequency and mode type were compared and validated. The optimal structure parameters were obtained by orthogonal factorial experiment design technique and statistical analysis of variance. The outline of the study was shown in fig. 1.

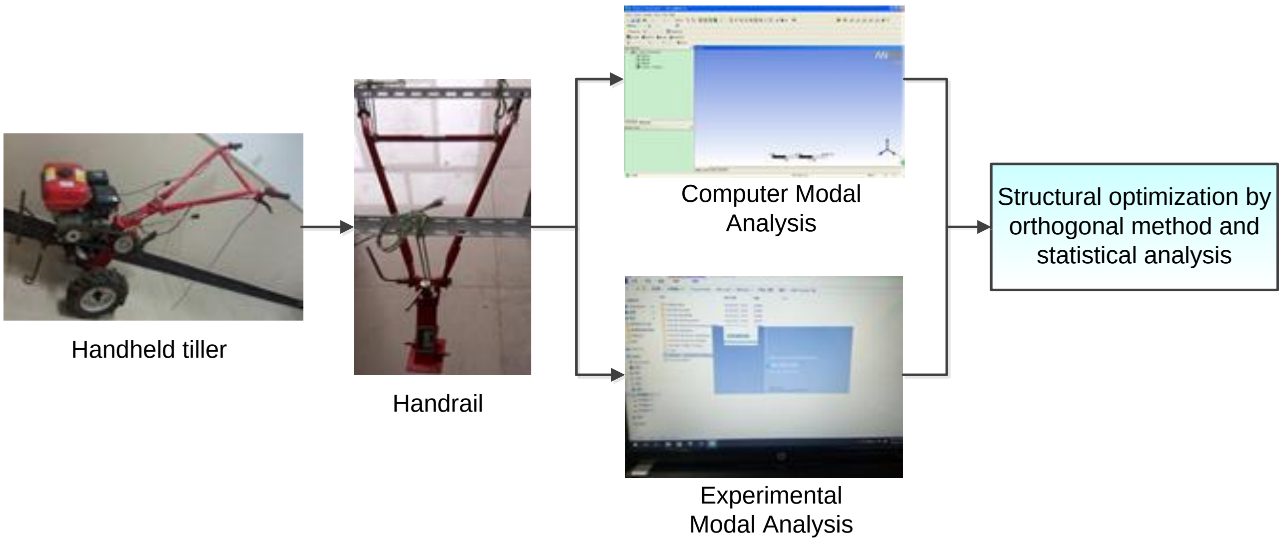


Fig. 1 - Outline of the study

Computational Modal Analysis

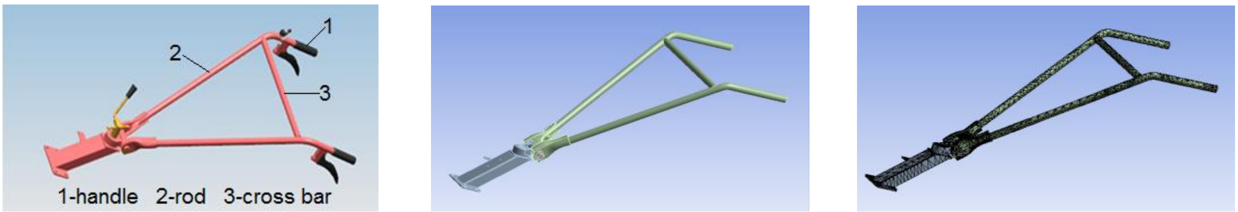
FEM model of the handrail

According to practical structure and parameters of handrail of the SR1Z-80 handheld tiller, the 3-D parametric solid model of the handrail was built in UG software package, as shown in fig. 2(a).

Import the solid model into ANSYS Workbench. By considering the effect of small sub-sections, such as chamfers, threads, etc. which were subjected to small forces during operation, on the meshing and solving of the handrail FEM model, some simplifications were made for the solid model. Then, the simplified 3-D model was obtained, as shown in fig. 2(b).

As triangle meshes have good performance for the vibration analysis, triangle meshing scheme was employed to mesh handrail of the tiller by automatic meshing method, with mesh size 3 mm. There are 51628 nodes and 27032 elements for the meshed handrail.

Handrail parameters were defined, as shown in table 1. Behaviour of handrail contact surface was defined as bonded. Then, the handrail FEM model was obtained, as shown in fig. 2(c).



(a) 3-D model **(b) Simplified 3-D model** **(c) FEM model**

Fig. 2 - Finite element model of the handrail

Table 1

Parameters of the handrail

Material	Density/kg.m^{-3}	Poisson's ratio	Elastics modulus/Pa	Contact
Steel Q235	7.83e3	0.35	2.07e11	Bonded

Modal Analysis

The modal analysis was performed in Ansys Workbench. No constraints or loads were applied to FEM model for Computational Modal Analysis, namely a free modal analysis approach was employed to determine the vibration characteristics (natural frequencies and mode shapes) of the handrail. Frequency range from 2 Hz to 200 Hz was focused on for the analysis, with weighting factor W_h greater than 0.1 within the frequency range (the frequency-weighting characteristic for hand-transmitted vibration W_h reflects the assumed importance of different frequencies in causing injury to the hand).

In panel Details of "Analysis Settings", specify Range Minimum and Range Maximum as 2 Hz and 200 Hz, respectively. Keep default option of "Solver Type" as "Program Controlled". Then the modal analysis is available for carrying out. The qualified natural frequencies of different modes, non-rigid body modes, and the corresponding mode shapes were obtained. According to the modal analysis, there are 5 non-rigid body modes within the frequency range focused on.

Experimental Modal Analysis

Experiment setup

As the handrail was simple in structure and light in weight, it was hung by an elastic polyamide rope with 8 mm diameter for the experiment setup as shown in Fig. 3. Main equipment and instruments components were as follows: modal force hammer 086C03 of sensitivity 2.335 mV/N and sensor 356A16 of sensitivity 98.2mV/g, made by PCB Piezotronics, Inc., USA; signal analyser LMS SCADAS Mobile of 24 channels, made by SIEMENS, Germany.

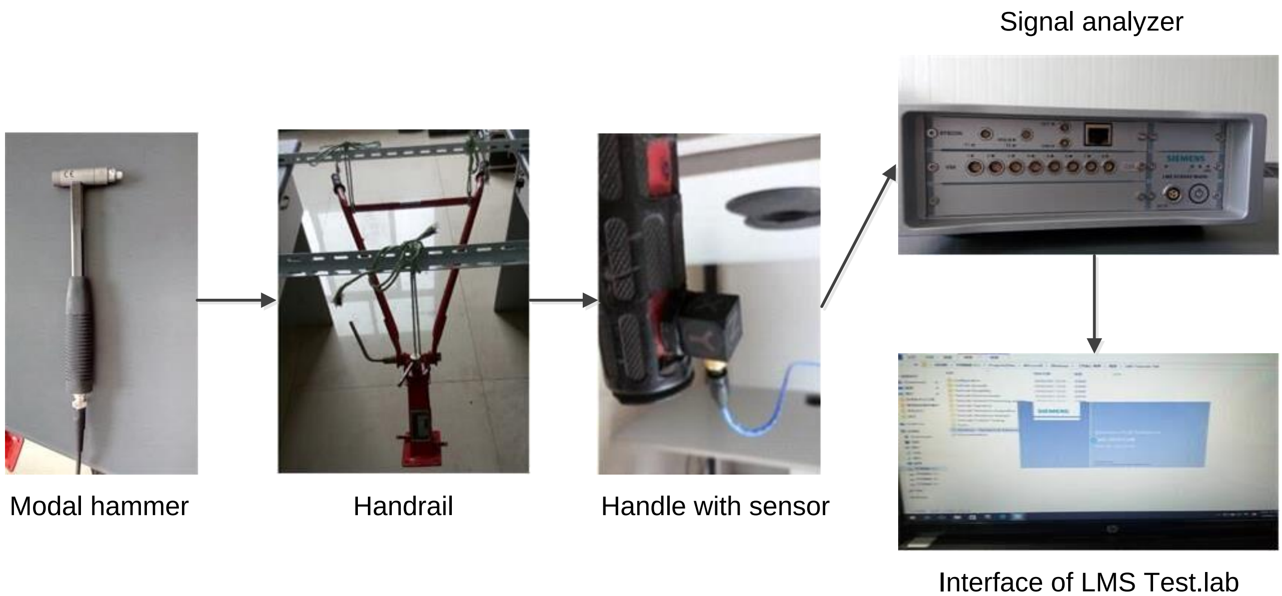


Fig. 3 - Experiment setup

Experiment procedure

According to exciting position sequences, information of 47 exciting points was defined in the testing system of the handrail investigated, by LMS Test.lab Impact Testing. The 3-D model of the handrail was built by combination of lines and surfaces determined by these exciting points in panel Geometry. Frequency 1024 Hz for the modal test was specified in panel Impact Scope. Driving points and average times for the exciting, trigger level, bandwidth, and window function were specified in panel Impact Setup through exciting trials, including auto-increment, roving hammer and auto reject with overload. The sensor locates at the 44th exciting point, and driving point starts from the first point and moves to the next sequentially, 5 times per driving point with 3-5 s intervals for the tapping excitation. Then, the frequency response function of each driving point can be obtained automatically in the LMS impact testing system.

Icons of Band, Stabilization, and Shapes are available for selection in panel PolyMax. Steady state diagrams were obtained after specification of frequency range by clicking the icon of Band, natural frequency characteristics were identified after enabling Automatic Modal Parameter Selection, Multivariant MIF or Complex MIF by clicking the icon of Stabilization, and the mode shapes of different order were obtained by clicking the icon of Shapes.

According to Modal Assurance Criterion (MAC) presented by West in 1986 *(Xiong, 2014)*, the values of MAC matrices were obtained by Auto-MAC method in panel Modal Validation. MAC is a dimensionless parameter, within a range of [0, 1]. The higher the MAC value, the greater the relevance.

Structural optimization

Experiment design

Orthogonal factorial experiment design technique based on Taguchi method was used to arrange experiments. Main control factors affecting vibration characteristics of the handrail were defined as follow: factor A: handle length; factor B: cross bar length; factor C: rod length. Levels of each control factor were shown in table 2. Experiments were designed in accordance with orthogonal array $L_9(3^4)$, a 3-level 4-factor array with 9 runs, and their arrangements were shown in table 3.

Table 2

Levels of control factor

Level	Handle length A /mm	Cross bar length B /mm	Rod length C /mm
1	170	390	470
2	200	420	520
3	230	450	570

Table 3

Experiment arrangements and results of natural frequency

No.	A	B	C	Blank column	Natural frequency /Hz
1	1	1	1	1	69.65
2	1	2	2	2	60.82
3	1	3	3	3	59.73
4	2	1	2	3	56.32
5	2	2	3	1	58.94
6	2	3	1	2	57.62
7	3	1	3	2	62.03
8	3	2	1	3	61.26
9	3	3	2	1	55.79

Optimization

Statistical analyses of range and variance were performed to obtain the impacts and their significance of each factor on the handrail vibration characteristics. Optimal engineering average was employed to obtain the optimal level combination of the control factors, and as a result, the structural optimization for the handrail was available.

RESULTS

Handrail vibration characteristics

The vibration characteristics (natural frequencies and mode shapes) of the handrail were obtained by means of Computational Modal Analysis and Experimental Modal Analysis.

Natural frequency

The natural frequencies of the non-rigid mode orders within the frequency range aforementioned were listed in table 4. As can be seen in the table, the natural frequencies by Computational Modal Analysis are in good agreement with those by Experimental Modal Analysis, with average relative error less than 4%. For the Experimental Modal Analysis, MAC values of each mode order are higher than 0.9, which shows that results by Experimental Modal Analysis are reliable. Therefore, Computational Modal Analysis can be used to analyse and optimize structure parameters of the handrail of a tiller from vibration characteristics.

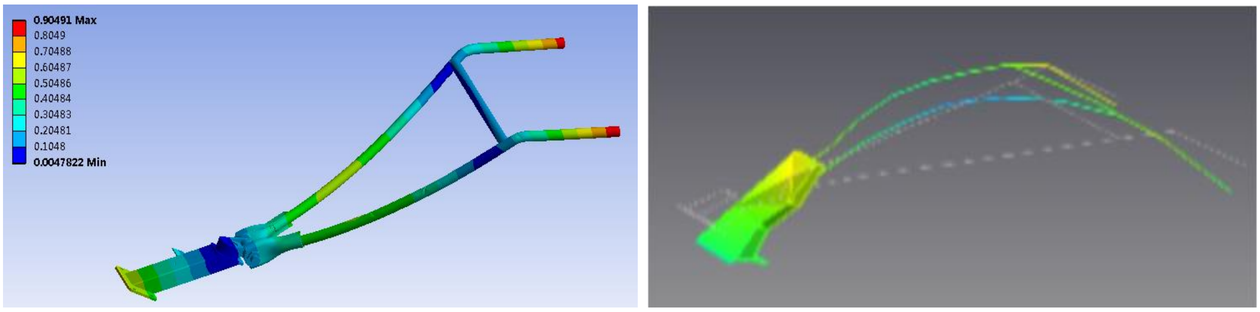
Table 4

Comparison of natural frequency

Mode order	Natural frequency by Computational Modal Analysis /Hz	Natural frequency by Experimental Modal Analysis /Hz	Relative error /%
1	61.84	64.39	4.1
2	89.61	84.04	6.2
3	112.90	116.29	3.0
4	137.33	143.46	4.5
5	177.05	174.78	1.3

Mode shape

The mode shapes of the non-rigid mode orders within the frequency range can be extracted from results of Computational Modal Analysis and Experimental Modal Analysis, and animations of the vibration are also available. According to literature *(Zhou, et al., 1999)*, human hands are generally sensitive to vibration of frequency f_s about 50 Hz *(Wen, et al., 2002)*. Considering frequency range of (0.75-1.3) f_s, namely 37.5-65 Hz, human hands are relatively sensitive to the vibration within the range with respect to handrails of the small scale agricultural machinery. Because the natural frequency of the first non-rigid mode order of the handrail falls in the frequency range 37.5-65 Hz, only mode shapes of the first non-rigid mode order were extracted by Computational Modal Analysis and Experimental Modal Analysis, and their contours were shown in fig.4.



(a) Computational Modal Analysis **(b) Experimental Modal Analysis**

Fig. 4 - Mode shape of the first non-rigid mode order

From the vibration animations of the first non-rigid mode order, the handrail largely vibrates up and down. The maximum vibration is at handrail handle, with a displacement of 0.90491 mm, and the minimum vibration locates at connection part of the handle and cross bar, and at direction shift part of the handrail, with a displacement 0.0047822 mm, as shown in fig. 4 (a). The mode shape of the first non-rigid mode order from Computational Modal Analysis shows good agreement with that from Experimental Modal Analysis, as compared fig.4 (b) and fig.4 (a). Furthermore, the mode shapes of the other non-rigid mode orders within the frequency range also have agreement mutually.

Consequently, Computational Modal Analysis can well reveal vibration characteristics of the handrail, with small errors compared with Experimental Modal Analysis, and then it can be employed to analyse and optimize the structure of a handheld tiller handrail.

Structural optimization

As the natural frequencies of the first non-rigid mode order of the handrail fall in the frequency range 37.5-65 Hz, generally sensitive to human hands, the vibration characteristics of the first non-rigid mode order are necessary for the structural optimization, and it is better to shift the natural frequency of a handrail away from this frequency range for the comfortable reason to operators of the handheld tillers. The natural frequency results of the handrail first non-rigid mode order by Computational Modal Analysis were obtained as shown in table 3.

Statistical analyses of range and variance were performed to obtain the impacts and their significance of each control factor on the handrail natural frequency. Range analysis results were shown in table 5, and variance analysis results were shown in table 6. The values in cells of each level of the control factors in table 5 represent mean natural frequencies of the corresponding levels and factors. The delta values of each factor represent the biggest change of mean natural frequencies of the factor, namely the impact level of each factor. The numbers in the rank row indicate the impact significance of the control factors.

Table 5

Range analysis

Level	Factor A	Factor B	Factor C	Blank column
1	63.40	62.67	62.84	61.46
2	57.63	60.34	57.64	60.16
3	59.69	57.71	60.23	59.10
Delta	17.32	14.86	15.60	7.07
Rank	1	3	2	

Table 6

Variance analysis

Source of variance	Degree of freedom	Sum of squares	Mean sum of squares	*F*-ratio	Critical *F*-ratio
Factor A	2	51.34	25.67	6.14	$F_{0.25}(2,2)=3.0$
Factor B	2	36.85	18.42	4.41	$F_{0.10}(2,2)=9.0$
Factor C	2	40.56	20.28	4.85	
Blank Column	2	8.36	4.18	Error	
Total	8	137.11			

Range analysis and variance analysis shows that: factor of handle length has the highest significant impact level on handrail natural frequency and it is followed by factors of rod length and cross bar length, sequentially. The *F*-ratio of each control factor was compared to a critical value corresponding to a certain pre-selected probability, resulting in probabilities of 86.0%, 81.5% and 82.9% that control factors are in fact due to chance because of handle length, cross bar length, and rod length, respectively.

According to optimal engineering average strategy *(Wang, 2004)*, $A_1B_1C_1$ is the optimal level combination of the control factors that benefits to the handrail vibration characteristics and it shifts the natural frequency away from the frequency range 37.5-65 Hz.

Vibration characteristics of the optimized handrail

The parameters of the optimal level combination of $A_1B_1C_1$ are as follows: handle length 170 mm, cross bar length 390 mm, and rod length 470 mm. According to the national standard of the People's

Republic of China (GB10395.10-2006), the parameters of the handrail optimal level combination qualify for the technical means for ensuring safety of walk-behind powered rotary tillers.

By Computational Modal Analysis, the mode shape of the first non-rigid mode order of the optimized handrail was obtained, as shown in fig.5. The natural frequency of the first non-rigid mode order is 69.65 Hz, and the maximum displacement and minimum displacement are 0.82063 mm and 0.0017485 mm, respectively, which shows good improvement in vibration characteristics compared to the original handrail.

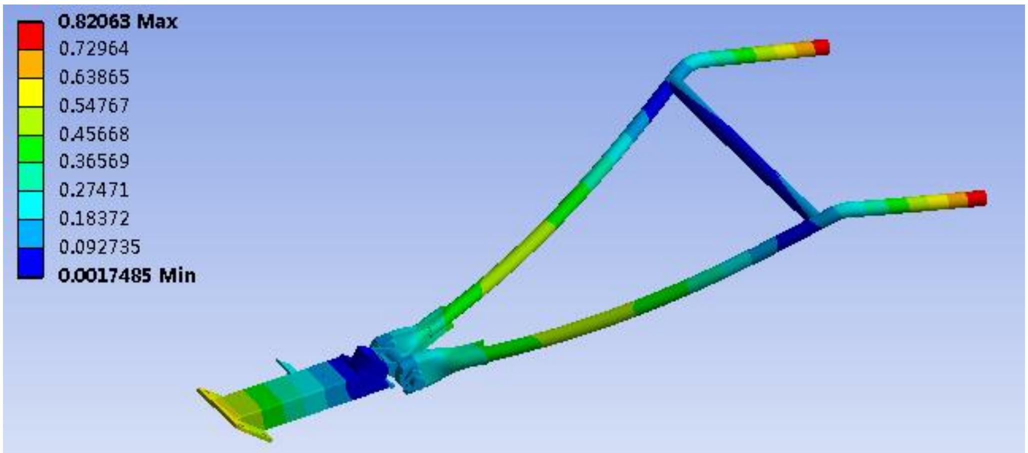


Fig. 5 - Mode shape of the first non-rigid mode order of optimized handrail

CONCLUSIONS

Hand-transmitted vibration presents a serious health and safety risk to operators of handheld tillers. Taking handrail of SR1Z-80 tiller as a case study, the vibration characteristics were investigated by means of vibration modal analysis. Structural optimization of the handrail was obtained by orthogonal factorial experiment design technique and statistical analysis. The main conclusions are as follows:

(1) Modal analysis can well reveal vibration characteristics (natural frequency and mode shape) of the handrail. The natural frequencies by Computational Modal Analysis are in good agreement with those by Experimental Modal Analysis, with average relative error less than 4%.

(2) Handle length has the highest significant level of impact on natural frequency of the handrail, and it is followed by rod length and cross bar length, sequentially. Their probabilities in fact due to chance are 86.0%, 82.9%, and 81.5%, respectively.

(3) The handrail optimal dimensions are handle length 170 mm, rod length 470 mm, and cross bar length 390 mm. The optimized handrail shows good improvement in vibration characteristics compared to the original handrail: natural frequency of the first non-rigid mode order 69.65 Hz, maximum displacement 0.82063 mm, and minimum displacement 0.0017485 mm.

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