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Motion interference analysis and optimal control of an electronic controlled bamboo-dance mechanism

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Abstract. An electric bamboo-dance mechanism was designed and developed to realize mechanism of automation and mechanization. For coherent and fluent motion, ANSYS finite element analysis was applied on movement interference. Static structural method was used for analyzing dynamic deflection and deformation of the slender rod, while modal analysis was applied on frequency analysis to avoid second deformation caused by resonance. Therefore, the deformation in vertical and horizontal direction was explored and reasonable optimization was taken to avoid interference.

1 Introduction

Bamboo Dance is a popular entertainment in Guangxi, Hainan and other southern regions of China as a national cultural heritage. The purpose of this project is to design and develop an electrically controlled bamboo-dance mechanism (BDM) (see Figure 1) to realize motion automation and mechanization. It mainly consists of swing rods connected by longitudinal rods on both sides. They are subjected to horizontal movement. A fine control of the rods' movement will benefit the BDM on its stability and practicability. The lateral movement of the swing rods is controlled by the scissor mechanism movement, carried by the longitudinal rods. Hence it is necessary to analysis the scissor mechanism and longitudinal rods' movement to optimize the design of BDM.

Figure 1: Prototype of electronic controlled bamboo dance mechanism

2 UG model and finite element model (FEM) setup

2.1 UG Model setup for horizontal movement mechanism

Horizontal movement of the actual structure is shown in Figure 2, the motion can be decomposed into a bamboo pole dance institutions horizontally and vertically, and here we only study the interference problem of longitudinal rods, i.e., only study the structure related to horizontal movement.

Figure 2: The UG diagram of horizontal movement of the structure

2.2 FEM setup

The rod Model was setup with unit BEAM189. It is suitable for structure analyze ranged from slender to medium chunky beams. The unit is based on Timoshenko beam theory with consideration of shear deformation effects [1]. The longitudinal rods and scissor mechanism are steel C1045, which its basic parameters are shown in Table 1 [2]:

Table 1: The FEM physical values of steel C1045

Name	Value
Elasticity modulus	EX = 2.09×10^{11} Pa
Poisson's ratio	PRXY = 0.269
Density	DENS = 7890 kg/ m^3

3 Static analysis

The rod is a slender rod, which has a greater theoretical static interference. So a middle support was adapted to reduce deflection and for proper motion. In addition, a dancer's common movement of 'pole stepped' was introduced into the model in order to simulate dynamic immunity and deformation of the rod [3, 4].

3.1 Finite element analysis (FEA) of longitudinal rod

Table 2: The ideal force conditions of longitudinal rod

Axial coordinate	Force
（0 , 0）	null
（0.08, 0）	support point
（0.16, 0）	31N
（0.46, 0）	1.4N
（0.76, 0）	1.4N
（1.06, 0）	1.4N
（1.44, 0）	1.4N
（1.74, 0）	31N
（1.82, 0）	support point
（1.9 , 0）	null

Based on the ideal working condition (condition I) shown in Table 2, FEA was carried out and the results are shown in Figure 3. The maximum deformation was 0.3mm rod located in the middle of the rod and gradually decreased to the ends of the rod. The snap ring of scissor mechanism has an outer diameter of 15.2mm, while the figure for longitudinal bar is 15.1mm, with a gap of 0.1mm between them. According to the node deformation nephogram, the 14th node of the forefront snap ring presented deformation of 0.157mm. Obviously, it had surpassed the maximum gap, hence will lead to over-bend of the longitudinal rod and will cause stuck on the ends.

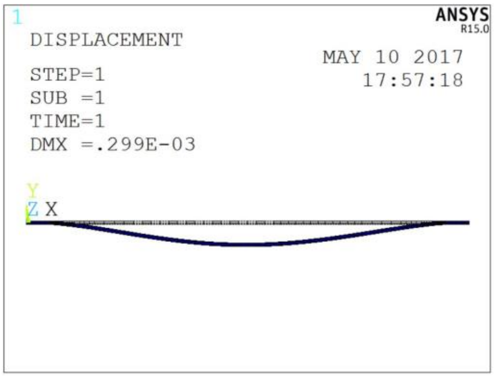

Figure 3: The deformation of longitudinal bar in the ideal conditions

Simulation of ‘tread pole’ (condition II) were also performed by ANSYS for analyzing vertical deformation when dancer step on the bar by mistake. Assumed dancer’s maximum weight F as 100Kg, force F1 = F / 4 = 250N, bearing of the rod is 250N. The forces condition is shown in Table 3.

As depicted in Figure 4, the maximum deformation of 13mm lies in the central rod and gradually decreases to the both ends. While the deformation of the forefront snap ring was over 3mm and had led to vertical movement interference.

Table 3: The analog stampede force conditions of longitudinal rod

Axial coordinate	Force
（0 ， 0）	null
（0.08， 0）	support point
（0.16， 0）	31N
（0.46， 0）	1.4N
（0.76， 0）	251.4N
（1.06， 0）	1.4N
（1.44， 0）	1.4N
（1.74， 0）	31N
（1.82， 0）	support point
（1.9 ， 0）	null

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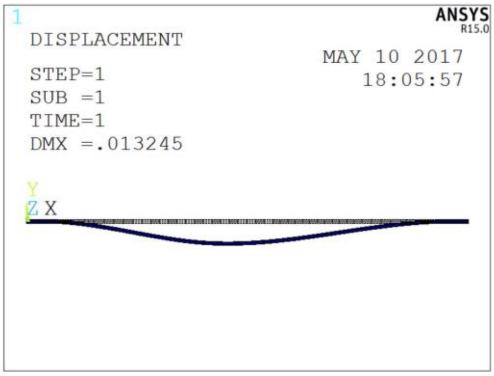


Figure 4: The deformation of longitudinal bar in analog stampede conditions

3.2 Optimization

'Tread pole' comes with extreme deflection and deformation. To avoid it, a support can be induced onto a vertical direction, so to form an intermediate between the longitudinal support rods, hence reducing deflection. A finite element constraint was applied as an intermediate support for simulation. It was set between 0.86 to 1.06m as the position of the support with applied vertical restraint, the results are shown in Figure 5.

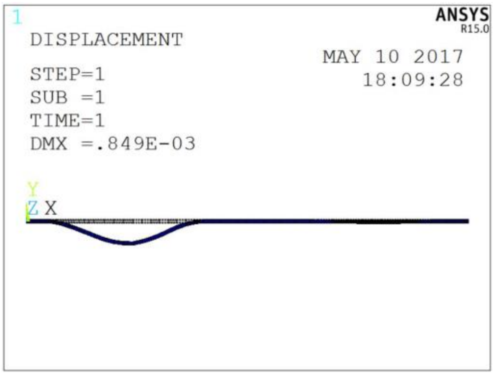


Figure 5: Deformation of longitudinal rod with intermediate support under 'tread pole' conditions

Only a half-wave was displayed, which represented deformation of 'tread pole'. Because the maximum load of 250N was applied on point (0.76, 0), the front section appeared obvious bending. The rear section bore forces as normal condition and presented different deformation with intermediate support. Nodes 39 deformed only 0.015mm, which was much less than the static deformation of 0.3mm and the rod gap of 0.1mm. The maximum deformation of the rod was only 0.85mm, which is the same order of magnitude as that of static load. This value was obviously less than the 13mm deformation of condition II, showing that the middle support can significantly decrease deformation and deflection. Also, the results indicated that under the first condition with middle support, the rod could avoid vertical movement interference and realize smooth movement.

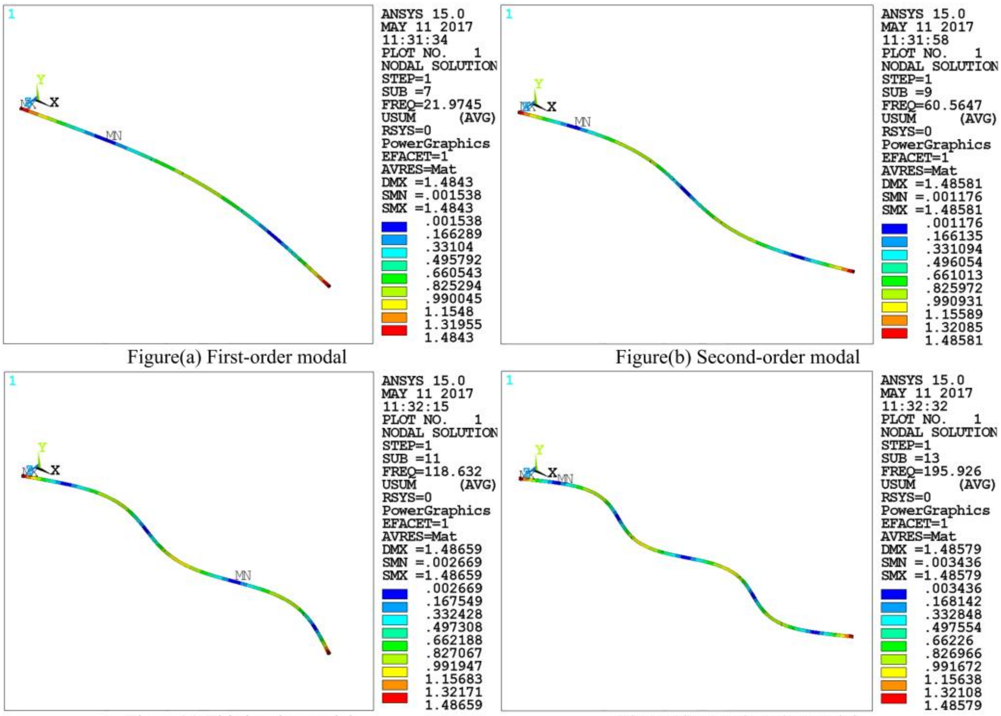
4 Modal analysis

4.1 Modal analysis of longitudinal bar

Modal analysis of a mechanism has two types: freedom modal analysis and constrained modal analysis [5]. Here we adopt the first type, at the same time using Lanczos (Block Lanczos) method with sparse matrix solver, which has the merit of fast convergence [6, 7]. The analysis only performed on first to fifteen modal, with effective modal from the seventh to the fifteenth order. As the mechanism is not only structural symmetric but also force symmetric, the frequencies present the same law [8]. Therefore, we only need to analyze one frequency of the two same values. The modal frequencies are shown in Table 4.

Table 4: The modal frequencies

Order	Natural frequency (Hz)	Vibration mode
1	21.974	1^{st} order bending
2	60.565	2^{nd} order bending
3	118.632	3^{rd} order bending
4	195.926	4^{th} order bending



Figure(a) First-order modal   Figure(b) Second-order modal

Figure(c) Third-order modal   Figure(d) Fourth-order modal

Figure 6: Modal analysis of longitudinal bar

It can be seen that the mode of vibration is substantial sway, which will cause interference between snap ring and longitudinal rod (see Figure 6). The best way to avoid this is to avoid resonance.

4.2 Motor's rated output speed to avoid resonance

Motor parameters of BDM are as follows: a PMDC permanent magnet DC gear motor [2] was selected with Shaft torque 8.9N·m and revolution speed of 3000r/min.

$$f = np/60 \tag{1}$$

Where, n is motor speed, r/min; f is power frequency, Hz; p is pole pairs of motor rotating magnetic field.

As the motor speed is controllable, each output with specific speed corresponds to a specific frequency. Based on previous modal analysis result on longitudinal bar, the rotation frequency must be away from resonance frequency. This can be realized by directly adjusting motor's input voltage, or using SCM to control motor input voltage automatically. For example, when the output speed of controller is 3000r/min, f is 50Hz, the vibration frequency is close to that of the longitudinal rod of 60.565Hz, and may lead to resonance. Therefore, when controlling the output speed of DC motor, it must be aware to the vibration frequency of the bars under 3000r/min to avoid resonance.

Table 5 is derived based on the above equation. The results of shows that in order to avoid the first order and the second order resonance, the following vibration stability criterion must be complied: $0.85f > f_p$ or $1.15f < f_p$. Where f is natural frequency; f_p is oscillation frequency.

It shows that the appropriate motor revolution should be between 2000r/min and 2500r/min, thereby horizontal deformation and deflections could be avoided.

Table 5: The table of resonance frequency

Order	Natural frequency (Hz)	Motor revolution	Motor frequency (Hz)	Vibration mode
1	21.974	1200r/min	20Hz	1st order sideway bending
2	60.565	3600r/min	60Hz	2nd order sideway bending
3	118.632	7200r/min	120Hz	3rd order sideway bending
4	195.926	11400r/min	190Hz	4th order sideway bending

5 Conclusions

Static structural analysis and modal analysis was applied on the analysis of horizontal movement of BDM. The conclusions are:

(1) Finite element model of the rod were setup. Analyze of vertical deflections and static deformation showed that: in order to prevent interference between the rod and snap ring of scissor mechanism, intermediate support shaft seat was applied and reduced vertical deformation.

(2) Based on four orders modal analysis of lever, it was found that the vibration is happened horizontally, which caused serious movement interference between the rod and snap ring. In order to prevent resonance, by Adjusting the motor speed, the frequency changed accordingly, thereby the modal frequencies of lever and horizontal vibration are avoided. It was deduced that the vibration was minimized when the rod revolution speed was 2000-2500r/min.

The above analysis had practically solved the problems of horizontal movement interference and optimized control, and had provided a primary solution for reliability and harshness of BDM.

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