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Original scientific paper

Numerical Simulation Study on the System Dynamics of Sea Buckthorn Harvester

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Abstract: To investigate the mechanical characteristics of the vibrational air-sucking sea buckthorn harvester, a system dynamics model integrating the harvester and the sea buckthorn tree was developed in this study. The model incorporates the adsorption force of the sucking device and the mechanical interactions between branches and fruits. Through numerical simulations, the influence of excitation frequency, lifting height, vibration isolation spring stiffness, crank radius, and adsorption force on the displacement of each component in the vibration system was systematically analyzed. Unlike vibration-only or suction-only prototypes, the proposed model simultaneously incorporates vibratory excitation and vacuum suction, which yields a 15 % higher detachment rate at 25 % lower trunk acceleration, thereby mitigating tree damage. The dynamic analysis demonstrates the vibration displacement of all components peaks at an excitation frequency of 10 rad/s. Increasing the stiffness of vibration isolation springs enhances the harvester's anti-vibration performance while amplifying the relative displacement between the tree trunk, branches, and fruits. Notably, the lifting height exhibits minimal impact on the harvest rate, whereas enlarging the crank radius significantly increases vibration displacement. Additionally, the application of adsorption force promotes fruit detachment by augmenting vibrational amplitude. This study provides theoretical insights for optimizing the design of vibration-adsorption combined harvesters.

Keywords: dynamic analysis; fruit-stalk interface; numerical simulation; sea buckthorn harvester, vibration - adsorption

1 INTRODUCTION

Sea buckthorn fruits have significant economic value. In recent years, the planting scale has been increasing. However, at present, sea buckthorn harvesting is still performed manually, which is inefficient and labor-intensive, severely constraining expansion of the sea buckthorn industry. Therefore, it is very necessary to develop and promote mechanized sea buckthorn harvesting technology [1-3]. The vibrating sea buckthorn harvester is the most common type of sea buckthorn harvesting machinery and can be used to harvest different kinds of fruits. Scholars at home and abroad have conducted extensive research on the mechanized harvesting technology of sea buckthorn [4-14].

Currently, research on sea buckthorn harvesting machinery both domestically and internationally has led to the development of several methods: trunk vibration, branch vibration, air-suction, and pruning harvesting. However, the unique physical characteristics of sea buckthorn fruits pose challenges for efficient harvesting. These fruits are small (4-6 mm in diameter), with a short but relatively thick stalk (approximately 2-3 mm in length and 2 mm in diameter). They grow densely on thorny branches and adhere firmly due to the strong binding force between the stalk and the branch, measured at 0.8-1.2 N for Chinese sea buckthorn [8]. When using vibration or air-suction methods alone, the harvest rates are low, and the harvesting is incomplete. Increasing the vibration amplitude to boost the harvest rate can harm the trees. While trunk vibration can harvest the entire tree's fruits at once, its effectiveness depends on the main trunk's branches. Slender branches can dissipate much of the energy before it reaches the fruits [16, 17]. Pruning for harvesting damages the trees and resources, as pruned main branches cannot bear fruit for the next three years. This method contradicts sustainable development and sea buckthorn forestry practices.

Foreign scholars have demonstrated that for the "Indian Summer" sea buckthorn variety in western Canada, the harvest rate rises linearly with amplitude increases at frequencies of 20-25 Hz, peaking at 25 Hz and 32 mm amplitude [15]. Similarly, a Swedish-developed sea buckthorn vibrating harvester operates effectively at 25 Hz and amplitudes of 40-55 mm [17]. Compared with emerging soft robotic grippers [25-27], the proposed vibration-adsorption mechanism avoids direct grasping, thereby reducing mechanical damage.

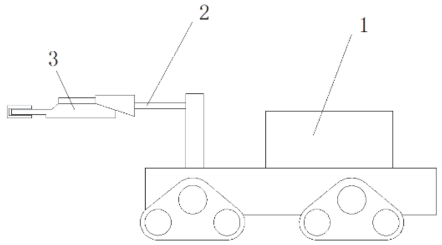
Relevant studies have shown [18] that the vibration frequencies and amplitudes of the optimal harvest of different trees are different and related to their natural frequencies. Before conducting the design of the vibrating harvesting device and the actual field experiments, it is very necessary to simulate and analyze the vibrating harvesting mechanism of the harvesting device.

In response to the above problems, a vibrational air-sucking harvesting method combining vibration harvesting with sucking harvesting is proposed. In order to analyze more systematically the influence of vibration mode, parameters, and sucking force on the effect of vibration harvesting, based on the system dynamics method, this paper simplifies fruit trees into a spring-mass-damping system composed of tree trunks, branches, and fruits. A multi-degree-of-freedom vibration system composed of a walking device, a vibration device, tree trunks and fruits was established, and an air-sucking adsorption force was applied. Then, mathematical modeling of this vibration system was carried out, and numerical simulation was conducted by software, providing important theoretical references for the design and development of sea buckthorn vibration-air-sucking combined harvesting equipment. The central research question is therefore: How do vibration parameters and suction force jointly influence fruit detachment, and what is the optimal combination to maximize harvest efficiency while minimizing tree damage? The remainder of this paper is organized as follows. Section 2 presents the coupled vibration-adsorption dynamic model. Section 3 details the numerical simulation and parameter sensitivity analysis. Section 4 provides design-oriented conclusions and future work.

2 MECHANICAL CALCULATION MODEL OF SEA BUCKTHORN VIBRATION HARVESTING

2.1 Establishment of the Model of the Sea Buckthorn Vibrating Harvester

The sea buckthorn vibrational air-sucking machine transmits mechanical energy to the sea buckthorn fruits by vibrating the fruit trees. At the same time, it applies the sucking adsorption force. Under the combined effect of vibration and adsorption force, sea buckthorn fruits are harvested in batches from the sea buckthorn trees. The sea buckthorn vibrating harvester is composed of four parts, which are the walking device, the vibration isolation device, the vibration device, and the sucking device. Due to the overly complex system graphic with the addition of the sucking device, the sucking device is not given in the system structure diagram, as shown in Fig. 1. According to the principle of mechanical vibration, the entire harvester-sea buckthorn tree harvester system can be simplified into four parts: the walking device, the vibration device, the trunk, and the fruit. The system model is shown in Fig. 2. Due to the actual elements involved such as tree trunks, branches, leaves, fruits, and soil, it is very difficult to precisely describe the dynamic control equation of the system. Therefore, the connection parts between each part of the system are equivalent. A vibration isolation device is set between the vibration device and the walking device. The vibration isolation device is composed of four parallel vibration isolation springs. Its main function is to isolate the vibration generated when the harvester vibrates the tree trunk, reduce the impact of the harvester's vibration harvesting on the operator and the entire machine, and improve the comfort of the operator and the reliability of the entire machine. The excitation mode of the vibration device is excitation by the crank-slider mechanism. The following hypotheses are proposed in the model: (1) The sea buckthorn fruit tree is simplified as a cantilever beam with one end fixed, and its mechanical properties are represented by equivalent stiffness and damping. (2) When sea buckthorn branches vibrate, they can be regarded as cantilever beams with fixed ends at the connection points to the main trunk. When subjected to external forces, they undergo bending deformation, and the mechanical properties between the branches and the trunk are equivalent to the bending equivalent stiffness. (3) The fruit and the branch are connected by the fruit stalk, which can be approximately regarded as a cylinder. The mechanical properties between the fruit and the branch are equivalent to tensile stiffness. (4) Simplify the entire system into a four-degree-of-freedom mass-spring-damping vibration system. (5) The connection between the vibration device and the sea buckthorn fruit trees is regarded as rigid, and the connection between the vibration device and the walking device is made by a vibration isolation device. The mechanical characteristics are expressed by equivalent stiffness and damping. (6) The adsorption force of the harvest part during inhalation is equivalent to a concentrated force acting on the sea buckthorn fruit.



1. Walking device 2. Vibration isolation device 3 Vibration device

Figure 1 Structural diagram of sea buckthorn vibrating harvester - air-sucking type

Based on the simplified mechanical model of the system, a system dynamics model is established to analyze the harvesting performance of the harvester. Then, by analyzing the influence of the stiffness of the vibration isolation spring, the radius of the crank, the rotational speed, and the lifting height of the harvester head on the harvester effect.

Fig. 2 is a schematic diagram of the entire simplified system.

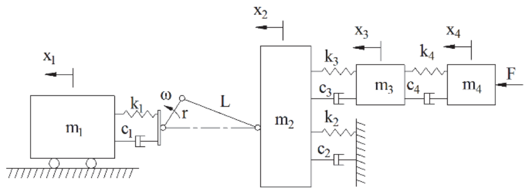


Figure 2 Vibration harvester - dynamic model of sea buckthorn tree

The meanings of each parameter in the figure are as follows: m_1 represents the quality of the harvester. The equivalent mass of the sea buckthorn tree at the empowerment point is m_2. m_3 is the equivalent mass of the branch; m_4 means the equivalent mass of sea buckthorn fruits. k_1 denotes the vibration isolation spring stiffness. The equivalent stiffness of tree trunks is k_2. The equivalent stiffness of branches is k_3. The equivalent stiffness of the fruit stalk is k_4. c_1 represents the vibration isolation spring damping. c_2 means the Equivalent damping of tree trunks. Equivalent damping of branches is c_3. Equivalent damping of fruit stalk is c_4. x_1 means the displacement of the harvester from the equilibrium position, and x_2 is the displacement of the tree trunk from the equilibrium position. x_3 is the displacement of the branch from the equilibrium position. x_4 represents the displacement of the harvester from the equilibrium position. r means the crank length. L is the length of the connecting rod. ω is the angular velocity of the crank. F is the adsorption force applied to the sea buckthorn fruit by the suction harvesting part.

Based on the above force diagram and by the theory of system dynamics, the vibration differential equation of the vibration system of the vibrating harvester is established as shown in Eq. (1) to Eq. (4).

$$m_1\ddot{x}_1 = -\left(x_1 - x_2 + \Delta x\right)k_1 - \left(\dot{x}_1 - \dot{x}_2 - \dot{x}\right)c_1 \tag{1}$$

$$\begin{aligned} m_2\ddot{x}_2 &= \left(x_1 - x_2 - \Delta x\right)k_1 + \left(\dot{x}_1 - \dot{x}_2 - \dot{x}\right)c_1 - \\ &-\left[\left(x_2 - x_3\right)k_3 + \left(\dot{x}_2 - \dot{x}_3\right)c_3\right] - \left(k_2 x_2 - c_3\dot{x}_2\right) \end{aligned} \tag{2}$$

$$\begin{aligned} m_3\ddot{x}_3 &= \left[\left(x_2 - x_3\right)k_3 + \left(\dot{x}_2 - \dot{x}_3\right)c_3\right] - \\ &-\left[\left(x_3 - x_4\right)k_4 + \left(\dot{x}_3 - \dot{x}_4\right)c_4\right] \end{aligned} \tag{3}$$

$$m_4\ddot{x}_4 = \left(x_3 - x_4\right)k_4 + \left(\dot{x}_3 - \dot{x}_4\right)c_4 + F \tag{4}$$

In the formula:

x - The absolute displacement of the crank-slider mechanism.

Δx - The relative displacement of the crank-slider mechanism.

The static cantilever formula for k_2 was corrected by a dynamic coefficient $\alpha = 1.2\text{-}1.5$, accounting for inertial effects at 10-20 rad/s.

The calculation formula of x, Δx is as follows.

$$x = L\left(1 - \frac{1}{4}\lambda^2\right) + r\left(\cos(\omega t) + \frac{1}{4}\lambda\cos(2\omega t)\right) \tag{5}$$

In the formula: λ - The ratio of the crank length to the connecting rod length.

$$\Delta x = r + L - x \tag{6}$$

By organizing Eq. (1) to Eq. (6), the following relationship can be obtained.

$$\begin{aligned} m_1\ddot{x}_1 = &-\left(x_1 - x_2 - \left(r + \frac{1}{4}L\lambda^2 - \right.\right. \\ &\left.\left. -r\cos(\omega t) - \frac{r}{4}\lambda\cos(2\omega t)\right)\right)k_1{}' - \\ &-\left(\dot{x}_1 - \dot{x}_2 - \left(-r\omega\cos(\omega t) + \frac{r}{2}\lambda\omega\cos(2\omega t)\right)\right)c_1 \end{aligned} \tag{7}$$

$$\begin{aligned} m_2\ddot{x}_2 = &\left(x_1 - x_2 - \left(r + \frac{1}{4}L\lambda^2 - r\cos(\omega t) - \right.\right. \\ &\left.\left. -\frac{r}{4}\lambda\cos(2\omega t)\right)\right)\cdot k_1 + \left(\dot{x}_1 - \dot{x}_2 - \left(-r\omega\cos(\omega t) + \right.\right. \\ &\left.\left. +\frac{r}{2}\lambda\omega\cos(2\omega t)\right)\right)c_1 - \left[\left(x_2 - x_3\right)k_3 + \right. \\ &\left. +\left(\dot{x}_2 - \dot{x}_3\right)c_3\right] - \left(k_2x_2 - c_3\dot{x}_2\right) \end{aligned} \tag{8}$$

$$\begin{aligned} m_3\ddot{x}_3 = &\left(x_2 - x_3\right)k_3 + \left(\dot{x}_2 - \dot{x}_3\right)c_3 - \\ &-\left[\left(x_3 - x_4\right)k_4 + \left(\dot{x}_3 - \dot{x}_4\right)c_4\right] \end{aligned} \tag{9}$$

$$m_4\ddot{x}_4 = \left(x_3 - x_4\right)k_4 + \left(\dot{x}_3 - \dot{x}_4\right)c_4 + F \tag{10}$$

The vibration curve of the harvester can be obtained by solving the dynamic Eq. (9) to Eq. (10) through numerical methods.

3 SIMULATION ANALYSIS OF DYNAMIC CHARACTERISTICS OF HARVESTERS

3.1 Calculation of Simulation Parameters

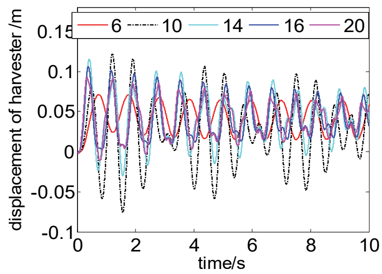
Considering that the size of sea buckthorn trees affects the harvest effect, in this paper, 10 sea buckthorn trees with diameters ranging from 35 to 45 mm and all in the fruiting stage within the same sea buckthorn planting base were selected for measurement. In this study, the main dynamic parameters of the sea buckthorn fruit tree were calculated. If the sea buckthorn tree is equivalent to a cantilever beam, then the stiffness of the trunk is $3EI/L^3$, $I = \dfrac{\pi d^4}{64}$. Where: E - the elastic modulus of the trunk. I - the moment of inertia of the tree trunk. L - the height from the ground to the holding point. d - the diameter of the tree trunk. The mass of the sea buckthorn fruit harvester (excluding the vibration device) is $m_1 = 1500$ kg. During harvest, the crank length *dd* is $r = 40$ mm, the working frequency is $\omega = 20$ rad/s, and the ratio of the crank to the connecting rod length is $\lambda = 0.25$. The vibration isolation device can be composed of 4 vibration isolation springs. The elastic coefficient of a single vibration isolation spring is $k_1 = 7 \times 10^4$ N/m, and the damping coefficient is $c_1 = 400$ N·s/m. When the height from the ground is 400 mm, the equivalent mass of the tree trunk in $m_2 = 235$ kg. The equivalent stiffness is $k_2 = 312.5 \times 10^3$ N·m^{-1}, $c_2 = 5000$ N·s/m, and the obtained dynamic parameters are shown in Tab. 1 [19-24].

Table 1 Mechanical parameters of the harvester

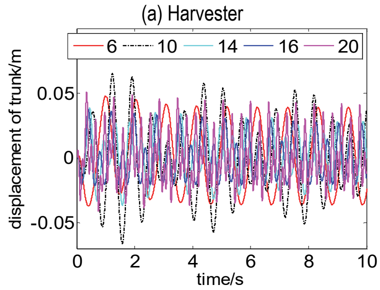
Parameters	Numerical value / kg	Parameters	Value / N·m^{-1}	Parameters / N·s/m	Value / N·m^{-1}
m_1	1500	k_1	2.8×10^5	c_1	1600
m_2	235	k_2	3.1×10^5	c_2	5000
m_3	5.5	k_3	433	c_3	0.04
m_4	0.0006	k_4	125	c_4	0

3.2 Analysis of the Vibration Characteristics of the Picking Machine by Crank Speed

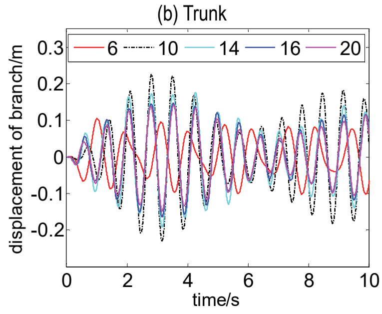
In this paper, two vibration isolation springs are selected, with a height of 400 mm from the ground. The crank length $r = 40$ mm, and the working frequencies are respectively set as 6 rad/s, 10 rad/s, 14 rad/s, 16 rad/s, and 20 rad/s. The ratio of the crank to the connecting rod length $\lambda = 0.25$. Without considering the adsorption force for the time being, the vibration displacement curves of the sea buckthorn harvester, the sea buckthorn tree body, branches and fruits were simulated and analyzed using the dynamic model of the sea buckthorn harvester, as shown in Fig. 3.



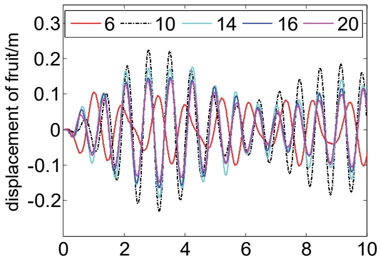
(a) Harvester



(b) Trunk



(c) Branch



(d) Fruit

Figure 3 Displacement curves of each part at different crank speeds

A comprehensive analysis of Fig. 3 can lead to the following conclusions.

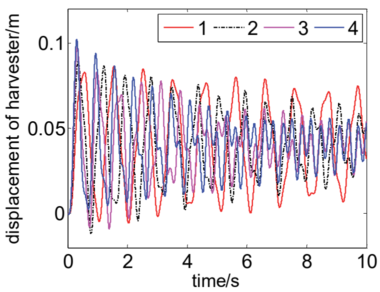
(1) The vibration displacements of each part of the system obtained through the simulation and calculation of the harvesting maneuverability model fluctuate within a certain range over time. The position movement of the harvester varies within ±0.15 m, the displacement of the tree trunk varies within ±0.06 m, and the branches and fruits vary within ±0.22.

(2) A comparative study of the excitation frequency and the amplitude of the vibration curve reveal that for the harvester and the tree body, the vibration displacement of the branches and fruits is the greatest when the excitation frequency is 10 rad/s, which has certain reference value for the design of the harvester.

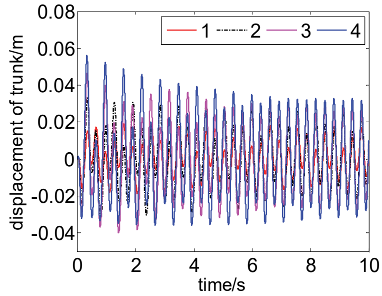
(3) The study of the period of the vibration curve reveals that the vibration curve shows a certain periodicity. The periods of vibration are different in each part.

3.3 The Influence of Vibration Isolation Springs on the Dynamic Characteristics of Dynamic Harvesters

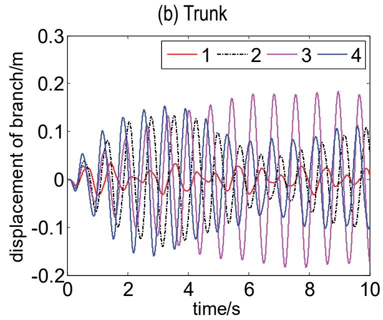
The lifting height is the height from the ground where the chuck of the vibration device clamps the tree trunk. The lifting height was selected as 400 mm, the crank length as $r = 40$ mm, the working frequency as $\omega = 20$ rad/s, and the ratio of the crank to the connecting rod length as $\lambda = 0.25$. The vibration curves of the sea buckthorn harvester, the sea buckthorn tree body, branches, and fruits were simulated and analyzed using the dynamic model of the sea buckthorn harvester, as shown in Fig. 3. Take the springs of the vibration isolation device as one vibration isolation spring, two vibration isolation springs, three vibration isolation springs, and four vibration isolation springs respectively. The elastic coefficient of a single vibration isolation spring is $k_1 = 7 \times 10^4$ N/m, and the damping coefficient is $c_1 = 400$ N·s/m. This paper analyzes the dynamic characteristics of the system. Fig. 4 respectively presents the vibration displacement curves of each part of the harvester varying with the stiffness of the vibration isolation springs.



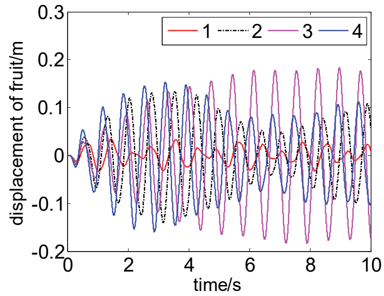
(a) Harvester



(b) Trunk



(c) Branch



(d) Fruit

Figure 4 Displacement curves of each part under different stiffness of vibration isolation springs

The following conclusions can be drawn from Fig. 4:

(1) For the vibration displacement curve of a harvester, the greater the spring stiffness, the greater the maximum value of the vibration displacement at the beginning. However, the maximum value of the steady-state amplitude decreases with the growth of stiffness (the number of springs). This indicates that if the number of vibration isolation springs is increased (that is, the stiffness is increased), the vibration isolation effect can be improved.

(2) For the vibration displacement curve of the tree trunk, as the spring stiffness increases, the maximum value of the vibration displacement of the tree trunk increases, from 0.019 m with one spring to 0.056 m with four springs, an increase of 194.2%.

(3) For the vibration displacement curves of branches and fruits, as the number of springs increases from 1 to 4, the maximum value of the vibration displacement of branches and fruits increases from 0.034 m to 0.18 m and then decreases to 0.15 m. The overall trend is increasing.

Based on the above analysis, it can be known that appropriately increasing the stiffness of the vibration isolation spring can enhance the anti-vibration performance of the harvester, and at the same time increase the vibration displacement between the tree body, branches and fruits. This provides a certain reference for the design of sea buckthorn harvesters.

3.4 The Influence of Lifting Height on the Dynamic Characteristics of Dynamic Pickers

This paper selects the crank length as r = 40 mm, the working frequency as 20 rad/s, the ratio of the crank to the connecting rod length as 0.25, two vibration isolation springs, the elastic coefficient of each vibration isolation spring as 7 × 10^4 N/m, the damping coefficient as 400 N·s/m, and the adsorption force is not considered for the time being.

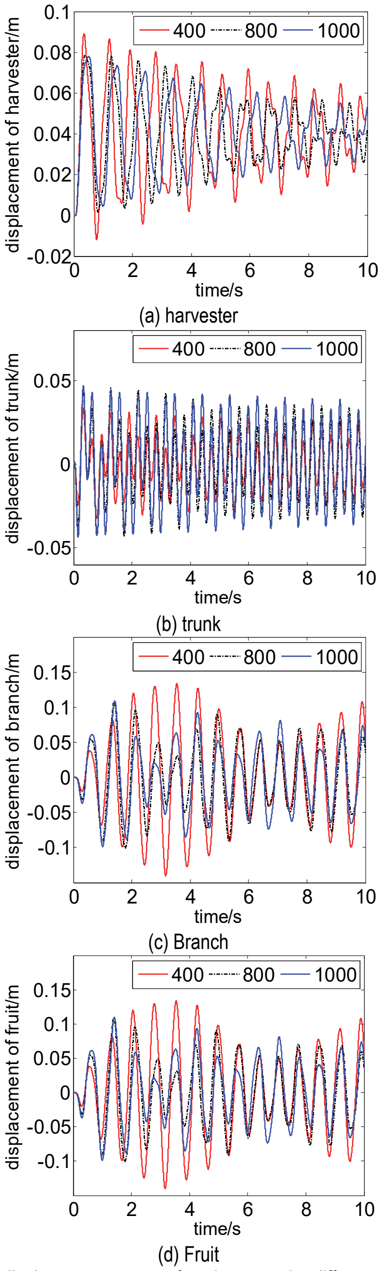


Figure 5 The displacement curves of each part under different lifting heights

The influence of the lifting height on the dynamic characteristics of the harvester was analyzed respectively with the lifting heights of 400 mm, 800 mm and 1000 mm. When the lifting height is 400 mm, the equivalent mass of the sea buckthorn tree is 235 kg, the equivalent stiffness is 312.5 × 10^3 N·m^{-1}, and it is 5.0 × 10^3 N·s·m^{-1}. When the height from the ground is 800 mm, the equivalent mass of the sea buckthorn tree is m_2 = 190 kg. The equivalent stiffness is k_2 = 166.6 × 10^3 N·m^{-1}, c_2 = 3.2 N·s·m^{-1}. When the height from the ground is 1000 mm, the equivalent mass of the sea buckthorn tree is m_2 = 168 kg, the equivalent stiffness is k_2 = 128.0 × 10^3 N·m^{-1}, c_2 = 2.4 N·s·m^{-1}. The displacement curves of each part change with the applied height as shown in Fig. 5 respectively.

The following conclusions can be drawn from Fig. 5:

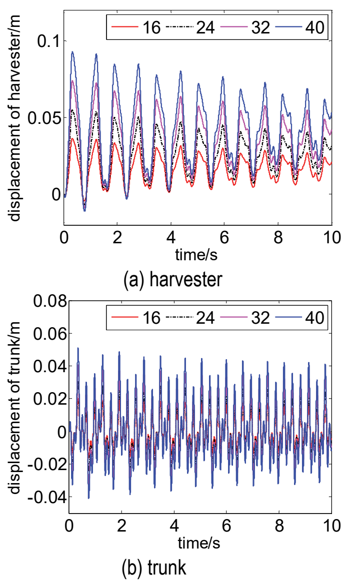
(1) For the harvester, it can be seen from the history of its vibration displacement changing over time that when the height of the support point increases, both the amplitude and the period of the displacement vibration curve change. When the height of the support point increases from 400 mm to 1000 mm, the maximum amplitude of the vibration displacement decreases from 0.089 m to 0.078 m, a reduction of 12.4%.

(2) For the tree body, when the height of the support point increases, the amplitude of the displacement vibration curve changes, but the period is almost unaffected. When the height of the support point increases from 400 mm to 1000 mm, the maximum amplitude of the vibration displacement increases from 0.033 m to 0.047 m, an increase of 42.4%.

(3) For branches and fruits, when the height of the support increases, the amplitude of the displacement vibration curve changes, but the period is almost unaffected. When the height of the support increases, the maximum amplitude of the vibration displacement first decreases and then increases, but the overall trend is still decreasing. When the enhanced height increased from 400 mm to 1000 mm, the maximum amplitude of the vibration displacement of branches and fruits decreased from 0.134 m to 0.110 m, a reduction of 21.8%.

3.5 The Influence of Crank Radius on the Dynamic Characteristics of Dynamic Pickers

This paper selects the lifting height as 400 mm, two vibration isolation springs, the working frequency ω = 20 rad/s, and the ratio of the crank to the connecting rod length λ = 0.25. Ignoring the adsorption force for the time being, this paper analyzes the influence of the crank radius on the dynamic characteristics of the harvester with the crank lengths of 16 mm, 24 mm, 32 mm, and 40 mm respectively. The vibration curves of the harvester, sea buckthorn tree body, branches and fruits are shown in Fig. 6.



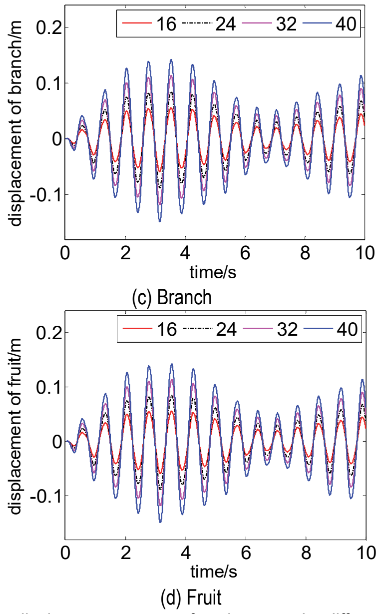


Figure 6 The displacement curves of each part under different crank radii

The following conclusions can be drawn from Fig. 6:

(1) For each part, when the crank radius gradually increases, it only affects the amplitude of the vibration curve of each part and has no effect on the period.

(2) For each part, the amplitudes of the displacement vibration curves of each part of the vibration system increase with the increase of the crank radius. When the crank length increased from 16 mm to 40 mm, the maximum amplitude of the displacement vibration curve of the harvester over time increased from 0.037 m to 0.093 m, an increase of 154.8%. The maximum amplitude of the displacement vibration curve of the tree body over time increased from 0.020 m to 0.051 m, an increase of 155.0%. The maximum amplitude values of the displacement vibration curves of branches and fruits over time both increased from 0.056 m to 0.142 m, an increase of 154.8%.

Based on the above analysis, it can be known that when the crank radius of the harvester exciter increases from 16 mm to 40 mm, an increase of 150%, the maximum displacement of each part of the vibration system increases, and the increase amplitude is relatively large, with the maximum increase exceeding 150%. This indicates that using a larger crank radius can increase the vibration displacement. Relevant studies have shown that when the displacement of vibration is greater, it is more beneficial for improving the harvest rate. Of course, an increase in the crank radius will inevitably lead to a larger size of the exciter of the harvester, which is also a factor to be considered and provides a certain theoretical basis for the design of the harvester.

3.6 The Influence of Adsorption Capacity on Harvesting Effect

Four vibration isolation springs were selected, with an added height of 400 mm, a crank length r = 40 mm, and working frequency ω is 20 rad/s, respectively. The ratio of the crank to the connecting rod length was $\lambda = 0.25$, and the applied adsorption force was 2 N [9]. The vibration displacement curves of the harvester, the sea buckthorn tree, the branches and the fruits were simulated and analyzed using the dynamic model of the sea buckthorn harvester. As shown in Fig. 7.

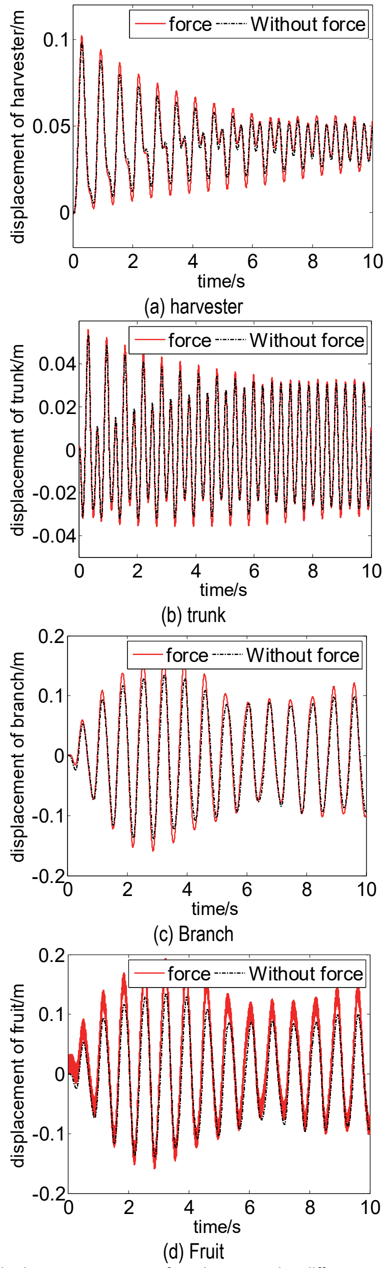


Figure 7 The displacement curves of each part under different adsorption forces

The following conclusions can be drawn from Fig. 7:

(1) For each part, when adsorption force is added, the amplitudes of the displacement vibration curves of each part all increase, but the increase amplitudes are different. The displacement increase amplitude on the fruit is the largest.

(2) After adding the adsorption force, the vibration displacement increases, which will be conducive to the shedding of sea buckthorn fruits. This indicates that the harvesting method of vibration combined with adsorption is feasible and provides a certain reference for the development of harvesters.

4 CONCLUSION

Based on the theory of system dynamics and considering the influence of branches and fruits, a dynamic analysis and calculation model for sea buckthorn harvesters and tree trunks, branches and fruits was established. Taking specific examples, the vibration conditions of the harvester, tree trunks, branches and fruits were analyzed and calculated, and the variation relationships among the excitation frequency, lifting height, stiffness of the vibration isolation spring, crank radius, adsorption force and vibration displacement were studied. Design guidelines derived from this study are as follows: (1) Employ a crank radius of 35-40 mm to achieve sufficient displacement without excessive mechanism size; (2) Set excitation frequency at 10 rad/s to exploit resonance between fruit and branch; (3) Combine four vibration-isolation springs ($k_1 \approx 2.8 \times 10^5$ N/m) with 2 N suction force to maximize relative fruit displacement while keeping trunk acceleration below 50 m/s².

The vibration displacement of each part of the vibration system is maximum when the excitation frequency is 10 rad/s, which provides a certain reference for the design of the harvester. Appropriately increasing the stiffness of the vibration isolation springs can not only enhance the anti-vibration performance of the harvester, but also increase the vibration displacement between the tree body, branches, and fruits. The growth in the lifting height has little effect on the harvest rate of sea buckthorn. A larger crank radius can increase the vibration displacement. The greater the vibration displacement, the more beneficial it is for improving the harvest rate. Of course, an increase in the crank radius will lead to a larger size of the harvester exciter. After adding the adsorption force, the vibration displacement of sea buckthorn fruits increases, which is conducive to the shedding of sea buckthorn fruits. This provides certain reference value for the design of the harvester.

Future work will validate the model in a field trial with 30 trees of varying diameters, and extend the model to include nonlinear fruit-stalk interface properties. This will integrate a soft suction cup array to enable selective harvesting of clustered fruits.

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