Vibration Analysis and Vibration Damping Device Design of Power Tiller

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Abstract. Power tiller is one of the most representative agricultural machinery in hilly mountain areas of China. While the excessive vibration of the power tiller caused great physical and psychological harm to its operators. The object of this study was to present the generation and transmission mechanisms of vibration on the power tiller, and provide several measurements for the vibration isolation and damping. A dynamics model of the single cylinder engine was established firstly. Based on the analysis of the vibration transmissibility, the vibration-absorption mechanism and the critical path of the vibration control are proposed. Finally, an isolation device was designed and developed to reduce the handle’s vibration of the power tiller.

1. Introduction

Power tiller is a multi-purpose hand-held agricultural machine for rotary tilling and other farm operation, which is particularly suitable for small fields. So the power tiller is one of the most representative agricultural machinery in hilly mountain areas and it plays an irreplaceable role in advancing the process of agricultural mechanization in these areas. With the widely use of power tillers, a comfortable working environment for the operator becomes an important consideration. One of the major concerns has been the exposure to a severe hand–arm vibration while operating them because of the cantilever type of the power tiller’s handle grip. Hand-arm vibration causes discomfort to the operator and results in early fatigue. A period of days and months exposure to this vibration would cause physical, physiological and musculoskeletal disorders of the operators. The occupational health disorders of the prolonged exposure to hand–arm vibration on the operators are referred to as vibration-induced white finger (VWF) or hand–arm vibration syndrome (HAVS) [1, 2]. A great deal of previous research have been undertaken to investigate the vibration mechanism and transmission characteristics of the hand–arm system [3-5]. And several other efforts have been made to develop vibration dampers and isolators [6, 7]. But these anti-vibration devices are not widely used because of the high cost, less effective or negative impact on the operation. This study was undertaken with the specific objectives of presenting the dominating source of vibration and the transmission mechanism, and providing several vibration isolators and dampers with the characteristics of simple, practical and low cost.
2. Theoretical consideration
The vibration sources of power tiller come from the engine and the working process of tilling, weeding and harvesting. Engine vibration is the critical factor causing the hand-arm vibration, which is mainly caused by speed fluctuation of the single cylinder. This vibration is transmitted from chassis to the handle, and causes the hand-transmitted vibration of the operator. Therefore the engine’s vibration characteristics and transmission mechanism on the power tiller was studied firstly.

2.1. Dynamics model of single cylinder
The power mechanism of the engine is called the crankshaft and connecting rod mechanism. With the reciprocal mass of this combinational equipment, the dynamic unbalance of the crank is the fundamental reason for producing vibration. The structure of single cylinder is illustrated in Fig. 1.

The piston stroke is obtained using equation (1).

\[ S = l + r - l \cos \beta - r \cos \alpha \]  
(1)

Where: S is the piston stroke; l and r are the length of rod and crank, respectively; \( \alpha \) and \( \beta \) are the angle of the crank and rod, respectively.

With a symbol \( \lambda = r/l \), S could be expressed by equation(2).

\[ S = r(1 - \cos \alpha) + l(1 - \sqrt{1 - \lambda^2 \sin^2 \alpha}) \]  
(2)

The piston stroke S could be reduced to equation (3) with the expansion of square root and omitting the 4th and upper orders.

\[ S = r(1 - \cos \alpha + \frac{\lambda}{2} \sin^2 \alpha) = r(1 + \frac{\lambda}{4} - \cos \alpha - \frac{\lambda^2}{4} \cos 2\alpha) \]  
(3)

The piston’s velocity and acceleration can be obtained by taking derivatives:

\[ \dot{S}_k = r\omega (\sin \omega t + \frac{\lambda}{2} \sin 2\omega t) \]  
(4)

\[ \ddot{S}_k = r\omega^2 (\cos \omega t + \frac{\lambda}{2} \cos 2\omega t) \]  
(5)
Where: $\omega$ is the angular velocity of the crankshaft.

With Newton’s second law of motion, the inertia force of the piston $F_z$ is obtained.

$$F_z = -m_z \ddot{S}_z = -m_z r \omega^2 (\cos \omega t + \lambda_p \cos 2\omega t)$$

(6)

2.2. Design principle of vibration isolation device

As shown in Fig. 2, the vibration system with the isolation device is simplified to a mass-spring-damper model.

As previously mentioned, the exciting force of the engine presents the form of sine function. So the real-time exciting force could be expressed as $F = F_0 \sin \omega t$. And the maximum dynamic pressure $F_{max}$ is $F_0$ without the vibration isolation device. The mass, stiffness and damping coefficient of the isolation system are denoted by $m$, $k$ and $c$, respectively. And the equations of forced oscillation could be expressed as follows:

$$x = B \sin(\omega t - \varphi)$$

(7)

$$B = \frac{F}{k} \frac{1}{\sqrt{(1 - \lambda^2) + (2\xi \lambda)^2}}$$

(8)

Where: $B$ is the vibration amplitude; $F$ is the dynamic pressure; $\xi$ is the damping ratio and $\lambda$ is the frequency ratio.

So with the reducing dynamic force by isolation system, the dynamic pressure imposed on the chassis contains two parts:

$$F_k = kx = kB \sin(\omega t - \varphi)$$

(9)

$$F_c = c\dot{x} = c\omega B \cos(\omega t - \varphi)$$

(10)

And the total dynamic pressure applied to the chassis is:

$$F_D = F_k + F_c = kB \sin(\omega t - \varphi) + c\omega B \cos(\omega t - \varphi)$$

(11)

From equation (11), $F_k$ and $F_c$ are harmonic excitations with the same frequency and 90 degrees phase differences. From equation (7) to (11), it’s known the maximum pressure to the chassis is:

$$F_T = \sqrt{(kB)^2 + (cB\omega)^2} = kB \sqrt{1 + (2\xi \lambda)^2} = F \frac{\sqrt{1 + (2\xi \lambda)^2}}{\sqrt{(1 - \lambda^2) + (2\xi \lambda)^2}}$$

(12)

The transmissibility of the vibration excitation with the isolation system is:

$$\eta = \frac{F_T}{F} = \frac{\sqrt{1 + (2\xi \lambda)^2}}{\sqrt{(1 - \lambda^2) + (2\xi \lambda)^2}}$$

(13)
With the application of the appropriate isolation device, the vibration of the chassis from motor excitation is reduced effectively.

3. Design of the vibration isolation device
As shown in Fig.3, a set of isolation device was designed using spring and rubber based on the previous mass-spring-damper model.

![Figure 3. Vibration-absorption Mechanism of the damper.](image)

The isolation device contains four sets of spring and rubber parts, which were linked by two support plates. The upper and lower support plates were constrained by the bolt and nut to prevent separation. This device was installed between the chassis and the handle of the power tiller (Fig.4).

![Figure 4. Install location of the Isolation device on power tiller.](image)

4. Verification of the isolation device
As shown in Fig.5, A 3-D multi-body dynamics model of the power tiller was constructed to simulate its vibration with and without the isolation device. The motor’s excitation was simplified to rotating uniformly at 2500 rounds per minutes which is sinusoidal wave as derived previously. The acceleration at the end of the handle was extracted with and without the effect of isolation device (Fig.6). Through the using of isolation device, the amplitude of the acceleration on the handle was reduced by 40%.

![Figure 5. 3-D multi-body dynamics model of the power tiller](image)

![Figure 6. Accelerations at the end of the handle with and without the isolation device](image)

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