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Grass trimmer hand-arm vibration reduction using multi-axial vibration absorber

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Abstract

In this investigation, the novel multi-axial vibration absorber is proposed to reduce the handle vibration of petrol engine grass trimmer. The proposed vibration absorber is designed using Dunkerley's equation and fabricated for testing. The experimental modal analysis of absorber is conducted to find resonance frequencies of the absorber and to validate the results obtained from equations. The experimental tests are carried on grass trimmer with absorber attached near handle location to find effectiveness of absorber in reducing hand-arm vibrations in the *x*, *y* and *z* directions. The result indicated that the total vibration value measured at the handle of grass trimmer is reduced by substantial level with the attachment of multi-axial vibration absorber.

Keywords

Multi-axial absorber; grass trimmer; modal analysis; absorber design; hand-arm vibration

Introduction

The petrol grass cutter machine is used for cutting wild grass and vegetation growth. Prolonged exposure of large magnitude of hand-arm vibration to operator causes vascular, neurological and musculoskeletal disorder, which is collectively known as hand-arm vibration syndrome (HAVS). A cross-sectional study on HAVS was carried out by Nor et al.[1] using questionnaire distributed to workers from grass and turf maintenance industry. The result shows the HAVS symptoms to moderate exposure and high exposure risk group. Mohamad et al.[2] evaluated the Hand-Arm Vibration Exposure Risk Assessment (HAVERA) variables and developed the health prediction cause–effect model of the HAVERA processes using multiple linear regressions and feedforward neural network programming. The studies showed that the vibrations are reduced at desired location on machines which can be modelled as a beam using vibration absorber.[3–9] Cha[3] used elastic mounted masses to induce single or multiple nodes along an elastic structure subjected to harmonic excitations. The focus of Cha and Rinker[4] was on enforcing nodes at desired location of damped Euler–Bernoulli beam during forced harmonic excitations using damped vibration absorbers. Cha and Buyco[5] formulated efficient approach to find the required absorber parameters, to impose nodes along an arbitrarily supported linear structure subjected to multiple harmonic excitations. Patil and Awasare[6] developed iterative procedure to find the absorber parameters required to impose node at desired locations on beam. Hao et al.[7] presented a research work on the usefulness of tunable vibration absorber to suppress vibration of electric grass trimmer. The field test results involving grass trimming operation showed that the handle vibrations are suppressed to large extents by attaching absorber at optimum location. Hao and Ripin[8] applied imposing node technique to achieve very low vibration (node) at the handle location using two cantilevered tunable vibration absorbers. Patil[9] applied imposing node method for vibration suppression of grass trimmer using single-axis vibration absorber. The grass trimmer considered as free–free beam carrying three lumped masses of cutter head, handle and engine, respectively. The total vibrations are reduced by 42% for no cutting operation and by 44% for cutting operation using imposing node method.

The frequency analysis illustrates that petrol engine grass trimmer is subjected to excitations in *x* and *z* directions due to petrol engine vibration. Nevertheless, vibration absorbers presented in prior research are effective in suppression of vibrations in single axis. Therefore, in this investigation, the novel multi-axial vibration absorber is

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designed and developed to reduce the multi-axis handle vibration of petrol engine grass trimmer. The experimental tests were performed to find efficacy of absorber in laboratory, that is, no cutting operation, and during field operation, that is, grass cutting operation.

Grass trimmer system

The specifications and additional details of the petrol grass trimmer used for experimental testing are listed in Table 1. The machine has appropriate nylon trimmer line for cutting various types of grasses and crops. It is powered by an internal combustion engine mounted at one end of hollow circular pipe and rotating cutter head at other end as shown in Figure 1. The nylon string attached to the cutter head cuts the grass. The speed of grass trimmer during grass cutting operation was in the range of 7000–7500 r/min. Therefore, for laboratory and field tests, the speed of grass trimmer is selected as 7250 r/min. The hand-transmitted vibration is measured by means of the frequency-weighted root-mean-square (rms) acceleration as per ISO 5349-1.[10] Figure 2(a) and (b) shows the axis system for the trimmer handle and pipe, respectively. As the single-axis accelerometer is available, the sequential measurement (measure in one direction at a time) of vibration is carried using lightweight mounting block which is attached to vibrating surface as depicted in Figure 2(c). The accelerometer is attached to the block for the measurement in the *x, y* and *z* directions, and all the operating conditions remain same for three-axis measurements ISO 5349-2.[11]

Table 1. Specifications of grass trimmer.

Engine	Mitsubishi TB43, Japan
Power output	1.3 hp/6500 r/min
Pipe diameter	28 mm
Gear ratio	14:19
Size ($L \times W \times H$)	1810 mm × 610 mm × 480 mm
Weight	8.3 kg

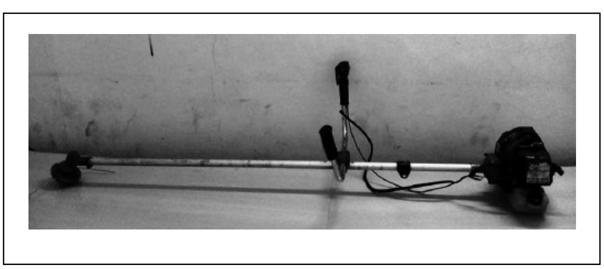

Figure 1. Grass trimmer system.

Frequency analysis of grass trimmer

The frequency analysis of grass trimmer during the grass cutting operation is carried out to find the sources of vibration. The speed of grass trimmer is kept constant around 7250 r/min for the period of measurements of accelerometer signal. The acceleration spectra of grass trimmer in the *x, y* and *z* directions near handle location are studied. Figure 3 illustrates that the grass trimmer employed in this study had highest peak in the *x*-axis of magnitude 7.8*g*, peak in the *z*-axis of magnitude 3.5*g* and in the *y*-axis 0.72*g* at 121 Hz, respectively. This frequency is correlated with the speed of the engine 7250 r/min, which is the primary source of vibration excitation. There is a second peak in *x, y* and *z* axes is of values 0.8*g*, 0.02*g* and 0.25*g*, respectively, with 90 Hz, which is due to rotating cutter head speed 5400 r/min (gear ratio 14:19). The magnitude of this source of excitation is small compared with engine excitation, hence neglected. Therefore, from frequency spectrum, it is observed that the grass trimmer is subjected to force harmonic excitation in the *x* and *z* directions due to petrol engine vibration. The vibration in the *y*-direction is very small, hence neglected.

Experimental modal analysis

The experimental modal analysis of the trimmer system with all attachments was carried out to determine the natural frequencies of the system. For modal analysis, the

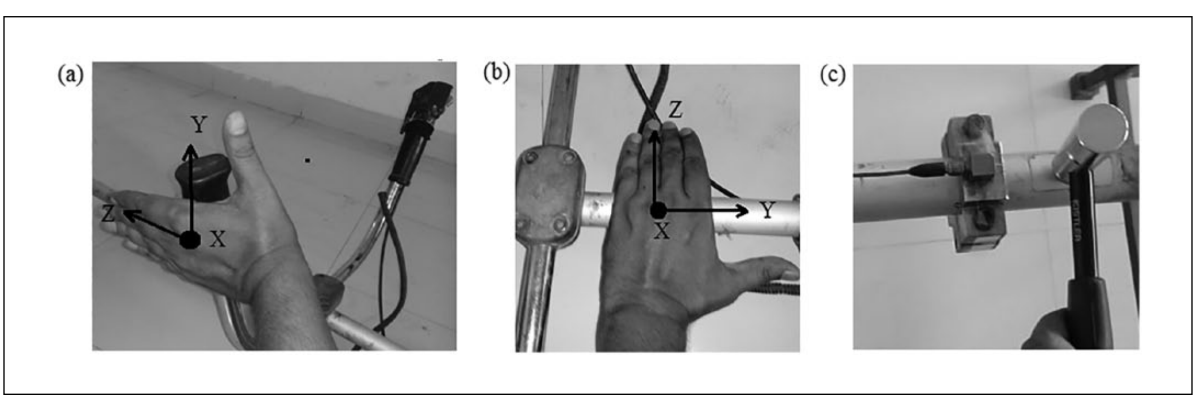

Figure 2. Axis systems on (a) trimmer handle, (b) trimmer pipe and (c) mounting block for vibration measurement.

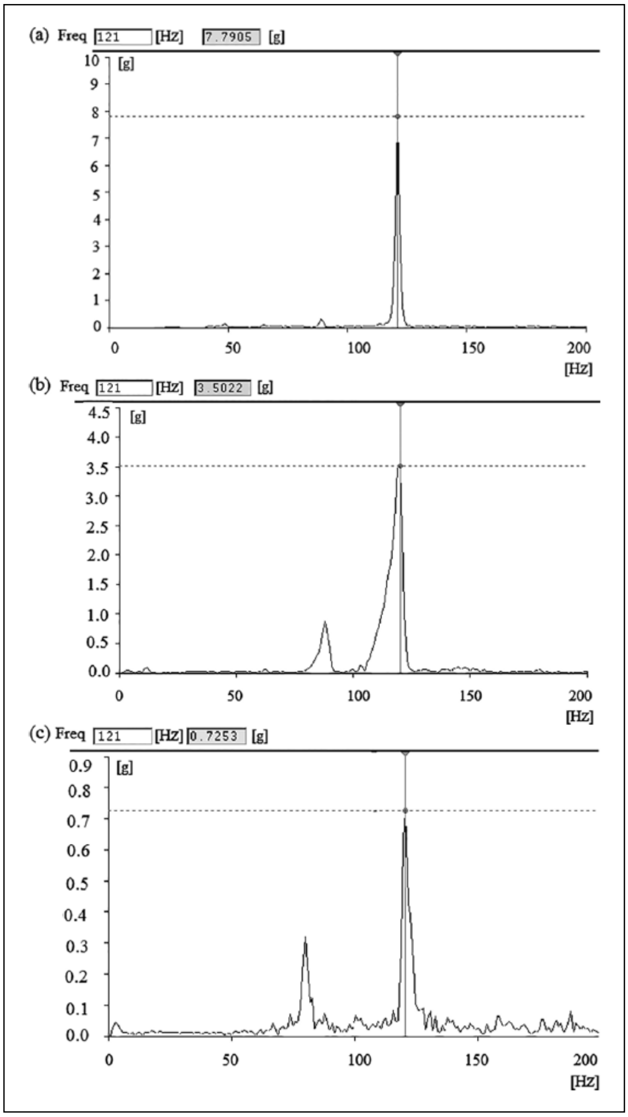

Figure 3. Frequency spectrum of grass trimmer near handle location along (a) *x*-axis, (b) *z*-axis and (c) *y*-axis.

grass trimmer suspended horizontally in a 'free–free' orientation approximated by two bungee cords. An impact hammer fitted with piezoelectric force transducer was used to excite the trimmer pipe as shown in Figure 2(c), while lightweight ICP accelerometer fixed to mounting block was used to measure the frequency response read from a fast Fourier transform (FFT) vibration analyser. Response spectrum of trimmer as given in Figure 4 shows that the first four modes get excited by impact hammer. First four natural frequencies of trimmer are 15, 51, 124 and 214 Hz, respectively. It is observed that the operating speed of the grass trimmer (7250 r/min, that is, 121 Hz) is near to the third natural frequency of the grass trimmer system (124 Hz).

Next, the grass trimmer considered as free–free beam with effective mass of cutter head, handle and engine as three lumped masses as shown in Figure 5. The grass trimmer system is subjected to harmonic excitation due to engine vibration at engine mount location in the x and z directions. The system parameters of grass trimmer are listed in Table 2. To reduce the vibrations at handle location, the multi-axial absorber (x-axis and z-axis absorber) is attached near handle support, which becomes the collocated case, that is, absorber attachment location and vibration reduction location are same. Now as explained in earlier research,[3–6] for the collocated case, the vibration absorber is required to be tuned to the operating frequency of grass trimmer (121 Hz) in order to reduce the vibrations at handle support locations.

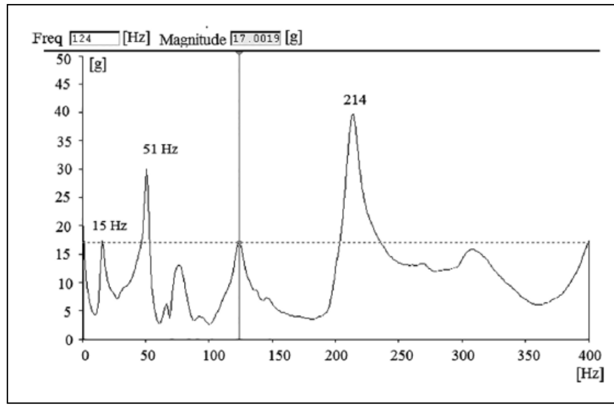

Figure 4. Frequency spectrum of grass trimmer.

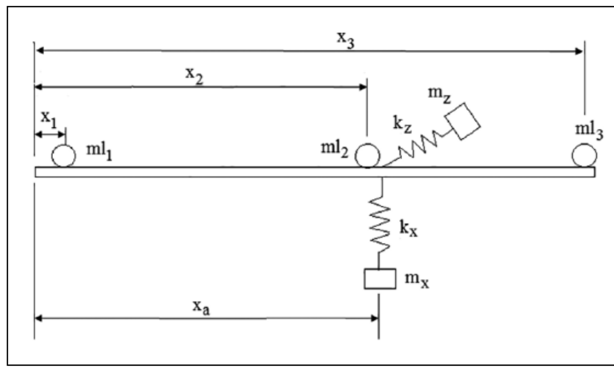

Figure 5. Grass trimmer systems with multi-axis vibration absorber.

Design of vibration absorber

In order to reduce vibration in the x and z directions, multi-axial absorber is proposed as shown in Figure 6. It consists of mutually perpendicular threaded rod which comprises cantilevered masses mounted on beam fixed at the centre. The fixed centre is attached directly to the vibrating structure. The resonance frequency of device is adjusted by moving the masses towards or away from the base support along the cantilever beam, which alters

Table 2. The system parameters of grass trimmer.

Length of the trimmer rod	L	1.5 m
Mass of rod	m	0.9 kg
Weight of cutter head	ml_1	0.895 kg
Weight of handle	ml_2	0.833 kg
Weight of engine	ml_3	5.21 kg
Cutter head location	x_1	27 mm
Handle location	x_2	1020 mm
Engine location	x_3	1480 mm
Absorber attachment location	x_a	1000 mm

Table 3. Dimensions and material properties of absorber used in design.

Density of rod and end mass	ρ_m	7800 kg/m^3
Modulus of elasticity	E	2.1×10^{11} N/m^2
Total length of each rods	L	0.100 m
Position of absorber mass from support	l	0.09–0.05 m
Diameter of each rods	d_b	0.006 m
Mass of each end mass	m_a	0.1 kg

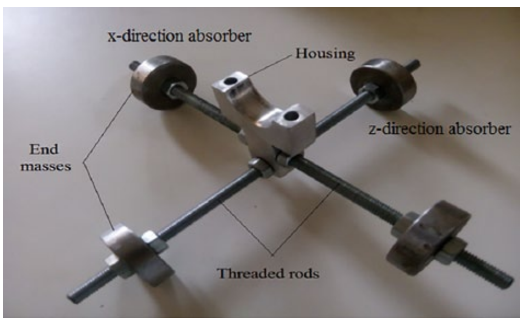

Figure 6. Tunable dual mass multi-axial absorber.

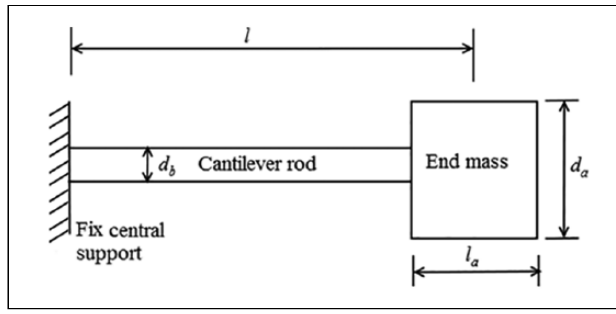

Figure 7. Dimensions of the absorber system used for discrete system analysis.

the effective stiffness in the system and alters its resonance frequency. The x-direction and z-direction device each is capable of suppressing vibration of grass trimmer in respective direction. The dimensions and material properties of the absorbers are listed in Table 3. The weight of each end mass of the absorber is selected as 200 g, so as to limit the percentage increase in the total weight of the trimmer system. Generally, when calculating the natural frequencies of complex dynamic linear systems, the finite element analysis method is applied. However, it is good practice to first apply a method (here Dunkerley's equation[12] is used) to approximately calculate the natural frequency of that system to get a feel for the value of the natural frequency. For the sake of brevity, only Dunkerley's equation results are reported in this section.

The absorber is modelled as dual cantilevered mass, which is represented as two discrete systems. The rod itself is assumed to be one vibration system, and the absorber mass at the end of the rod is another system. Figure 7 shows the half part of the cantilevered masses which are symmetric at the fixed centre. If the damping of the system is neglected, the natural frequency of the absorber can be determined using Dunkerley's equation

$$\frac{1}{\omega^2} = \frac{1}{\omega_a{}^2} + \frac{1}{\omega_b{}^2} \tag{1}$$

The natural frequency of the cantilever rod ω_b of mass m_b

$$\omega_b = \sqrt{\frac{12.7EI}{m_b l^3}} \tag{2}$$

The natural frequency of cantilever rod of negligible mass with concentrated mass m_a attached at the end ω_a is given by

$$\omega_a = \sqrt{\frac{3EI}{m_a l^3}} \tag{3}$$

Substituting equations (2) and (3) into equation (1) gives

$$\omega_1 = \sqrt{\frac{38.1EI}{\left(3m_b l^3 + 12.7m_b l^3\right)}} \tag{4}$$

From above equation, the fundamental frequencies of absorber when the absorber mass is at the position l are calculated. Figure (8) shows the plot of natural frequencies of absorber with change in mass position along the rod length. The designed resonance frequencies of the absorbers are kept less than the required resonance frequencies, because at the time of experimentations, the required resonance frequencies are obtained by moving end mass

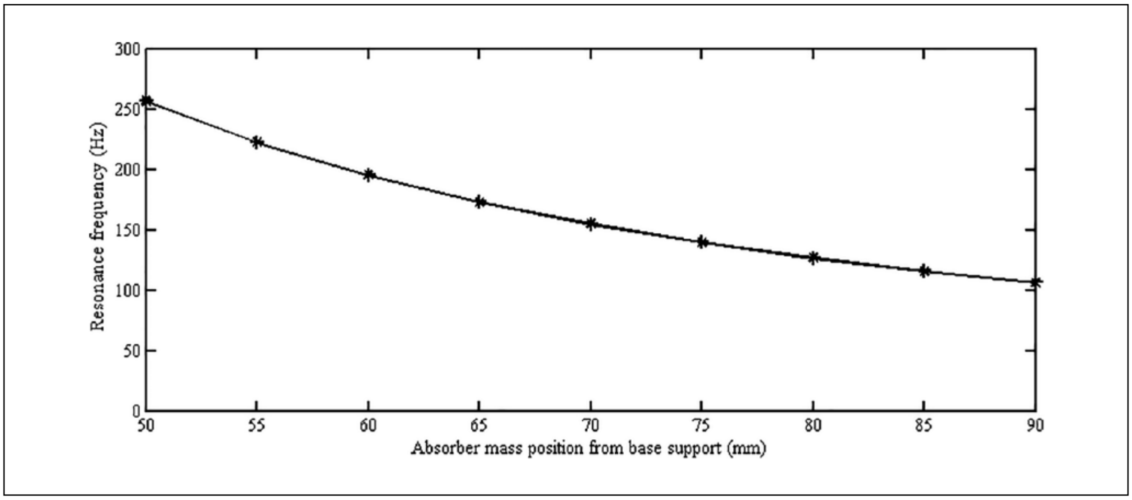

Figure 8. Variation of frequencies of absorber with change in mass position along the rod length obtained by Dunkerley's equation.

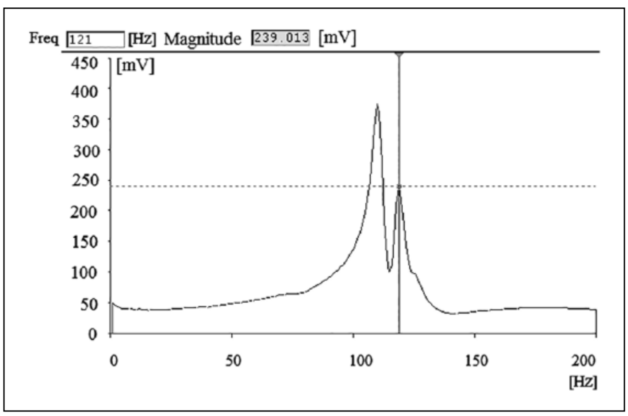

Figure 9. Frequency spectrum of absorber.

towards the base support. Figure 6 illustrates the absorber developed using parameters listed in Table 3.

The experimental modal analysis of the absorber was carried out to determine the resonance frequencies of absorber for different mass positions on the rods. Response spectrum of absorber is depicted in Figure 9. It is observed that the first two modes get excited by impact hammer. This is the experimental validation of the discussion from Hill and Snyder[13] that the first two modes of absorber are characterised by vertical displacement of masses, with the masses in phase for first mode and out-of-phase for second mode and frequencies are in close proximity for these modes. Figure 10 describes the relationship between the mass position and the resonance frequencies of the absorber. It is observed that the resonance frequency of the absorber increases linearly as mass moves towards the base support. Comparison of Figures 9 and 10 shows that the frequencies obtained from Dunkerley's equation are slightly higher than experimental results. In Dunkerley's equation for finding the stiffness of the rod, it is considered as of uniform diameter; nevertheless, actual rod has threads which reduce its stiffness. In addition, for Dunkerley's equation, the centre housing is considered as rigid one; however, the developed absorber housing is of aluminium having particular stiffness. For these reasons, the frequencies calculated by Dunkerley's equation are higher than of experimental modal analysis frequencies.

Experimental testing of trimmer with absorber

The experimental test is carried out to determine the effectiveness of the absorber in vibration reduction at handle support location during non-cutting operation and cutting operation. Figure 11(a) shows the experimental setup for vibration measurement of grass trimmer with and without absorber. The grass trimmer suspended horizontally in a 'free–free' orientation approximated by two bungee cords as shown. The vibration amplitudes are measured at 30 points on the trimmer surface by the accelerometer and recorded by vibration analyser to plot experimental steady-state response of trimmer pipe rotating at 7250 r/min, that is, 121 Hz.

To find the effectiveness of the absorber in the x and z directions, the absorber is mounted on the grass trimmer pipe near handle location at 1000 mm as illustrated in Figure 11(b). First, x-direction absorber is tuned by moving masses in or out such that the vibration in the x-direction at handle mount location of pipe is reduced to the minimum level. Next by same method, z-direction absorber is tuned to reduce vibration the in z-direction. The vibration amplitudes in x, y and z directions were measured on the trimmer surface by the accelerometer and recorded by vibration analyser to plot experimental steady-state response of trimmer pipe with absorber.

It is evident form experimental steady-state response for trimmer pipe with multi-axial absorber shown in Figure 12 that the amplitude of vibration is suppressed

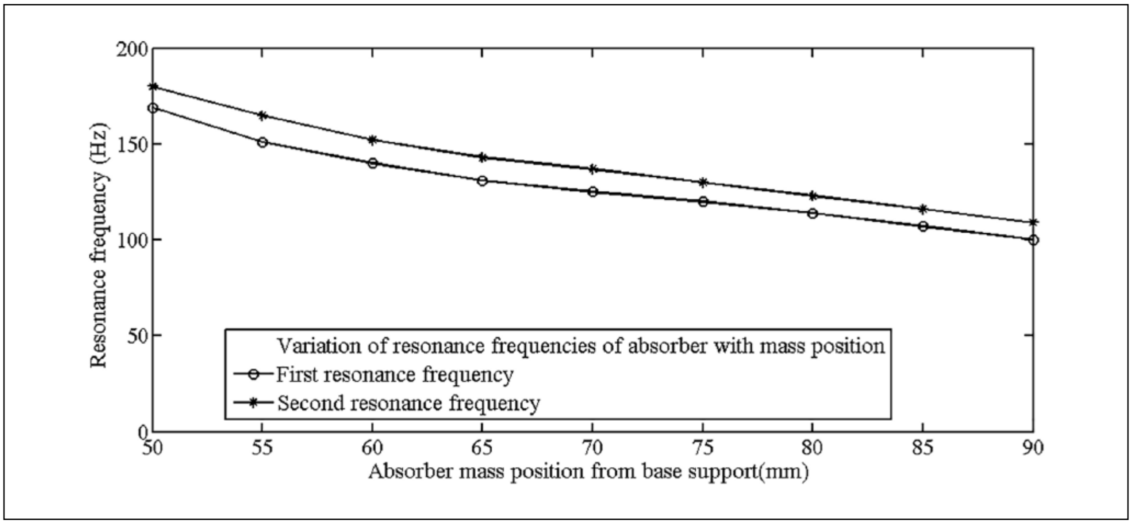

Figure 10. Variation of frequencies of absorber with change in mass position along the rod length obtained by experimental modal analysis.

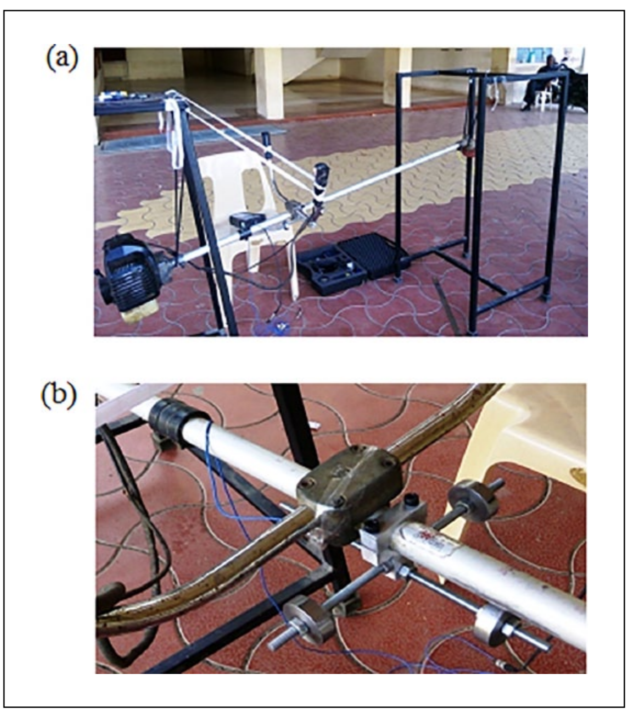

Figure 11. (a) Experimental testing of grass trimmer during non-cutting condition and (b) absorber attachment near handle location.

at handle location, that is, at 1020 mm from 108 to 22 μm the in x-direction, from 78 to 36 μm in the z-direction, and the in y-direction from 38 to 22 μm. The percentage reduction in the amplitude at handle location is 78%, 53% and 38% in the x, z and y directions, respectively. For the experimental steady-state response of trimmer pipe with single-axis absorber, the vibration at handle location is reduced from 108 to 25 μm the in x-direction, from 78 to 75 μm the in z-direction and in the y-direction from 38 to 35 μm. Therefore, the percentage reduction in the amplitude with single-direction absorber is 76%, 3.5% and 7.8% in the x, z and y directions, respectively. Therefore, multi-axis absorber is more effective in handle vibration reduction compared with single-axis absorber. It is to be noted that the weight of the single-axis absorber is 480 g and the percentage increase in the weight of grass trimmer system due to absorber is 0.06%. The weight of the multi-axis absorber is 700 g, which increases the weight of grass trimmer system to 0.088%. The amplitude at handle location can be further reduced by increasing the absorber mass, but it increases the total weight of the system.

Furthermore, it is apparent form Figure 12 that the vibration amplitudes of trimmer pipe with the multi-axis absorber are reduced for the span from 0 to 1020 cm, that is, from the cutter head to handle, in the x and z directions. It is worth commenting that, in the theoretical model, there was no coupling between bending and axial motion; however, an experimental result shows reduction in axial vibration. The purpose of current investigation is vibration reduction in the x and z directions, which is large compared with the y-direction (axial) vibrations. Hence, the model presented here is limited to bending and axial motion.

Handle vibration measurements

The measurement of acceleration of handle vibration near hand grip position is carried out with and without single- and multi-axis absorber during grass cutting operation as shown in Figure 13. The measured values of acceleration are listed in Table 4. The reduction of vibration was calculated by subtracting the vibration acceleration of the grass

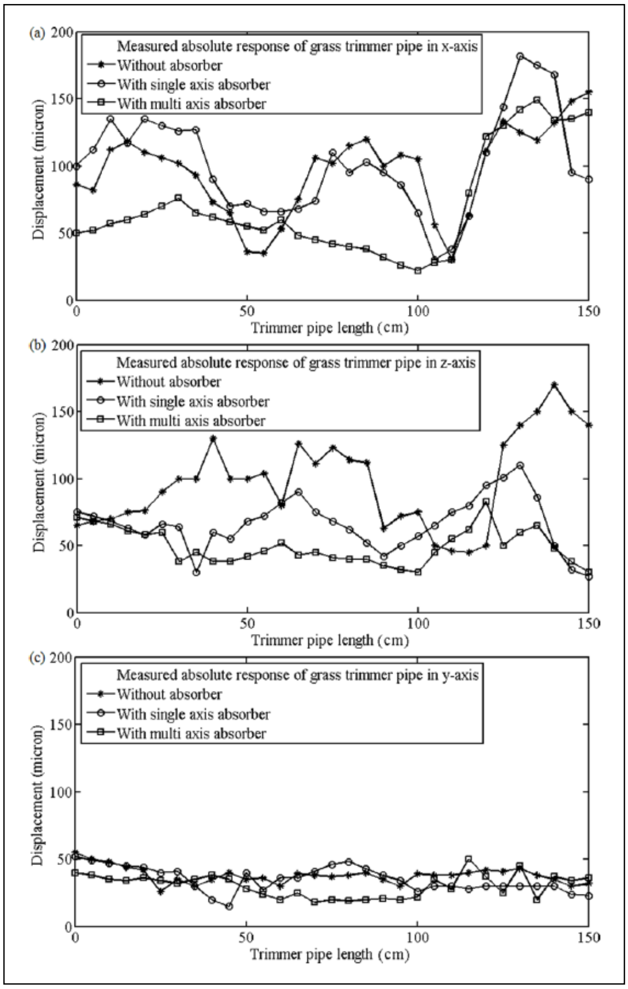

Figure 12. Experimental measured steady-state deformed shape of trimmer pipe along (a) *x*-axis, (b) *z*-axis and (c) *y*-axis.

Figure 13. Measurement of acceleration of handle vibration near hand grip position during grass cutting operation.

trimmer when operating without the absorbers from the condition with the absorbers installed. The handle vibrations are measured by means of the frequency-weighted rms acceleration, expressed in m/s^2 as per ISO 5349-1.[10] Table 5 illustrates the frequency-weighted rms acceleration (x_h, y_h and z_h axes) and vibration total value measured at handle of grass trimmer during no cutting (laboratory testing) and cutting operations (field testing). The reduction of frequency-weighted rms accelerations during no cutting operation for multi-axis absorber were 3.62 m/s^2 in x_h-axis, 3.83 m/s^2 in y_h-axis, 1.72 m/s^2 in z_h-axis and total vibration by 5.33 m/s^2. The percentage reduction for multi-axis absorber in the x_h-axis was 82.05%, 62.07% in the y_h-axis, 50.34% in the z_h-axis and 63.98% for the total vibration value. Similarly, for cutting operation, reduction of frequency-weighted rms accelerations was 2.52 m/s^2 in x_h-axis, 4.81 m/s^2 in y_h-axis, 3.13 m/s^2 in z_h-axis and total vibration by 6.19 m/s^2; percentage reduction in the x_h-axis was 60.24%, 74% in the y_h-axis, 62.92% in the z_h-axis and 67.35% for the total vibration value. However, for single-axis absorber, the percentage reduction of frequency-weighted rms accelerations in the x_h-axis was 73.07%, 25.44% in the y_h-axis, 46.35% in the z_h-axis and 38.77% for the total vibration value for non-cutting operation. For cutting operation, the percentage reduction for single-axis absorber in x_h, y_h and z_h axes are 30.8%, 58.07% and 63.96%, respectively, with total vibration of 43.96%.

Conclusion

In this research, multi-axial vibration absorber was proposed to suppress vibration of petrol engine grass trimmer handle. The experimental modal analysis result of absorber reveals that the absorber has resonance frequency from 100 to 180 Hz in both directions, which is useful in tuning the absorber to desired resonance frequency. The absorber is attached near handle location, which is the collocated case; therefore, it is tuned to the operating frequency of grass trimmer so as to reduce the vibrations near attachment locations. The result shows that the single-axis vibration absorber reduces the total vibrations by 39% for no cutting operation and by 44% for cutting operation. Nevertheless, multi-axial vibration absorber developed in this research reduces the total vibrations by 63% for no cutting operation and by 68% for cutting operation.

Table 4. Measured rms acceleration of trimmer handle.

Condition	Absorber	*x*-direction (m/s^2)	*y*-direction (m/s^2)	*z*-direction (m/s^2)	Total vibration (m/s^2)
No cutting	Without absorber	34	47.5	26.4	64.10
	With single-direction absorber	9.2	35.4	14.2	39.24
	Reduction	24.8	12.1	12.2	24.86
	With two-directional absorber	6.1	18	13.1	23.08
	Reduction	27.9	29.5	13.3	41.02
Cutting	Without absorber	32.2	50	38.3	70.73
	With single-direction absorber	13.5	34.6	13.8	39.62
	Reduction	18.7	15.4	24.5	31.11
	With two-directional absorber	12.8	13	14.2	23.11
	Reduction	19.4	37	24.1	47.62

Table 5. Frequency-weighted rms acceleration of trimmer handle.

Condition	Absorber	*x*-direction (m/s^2)	*y*-direction (m/s^2)	*z*-direction (m/s^2)	Total vibration (m/s^2)
No cutting	Without absorber	4.42	6.17	3.43	8.33
	With single-direction absorber	1.19	4.6	1.84	5.1
	Reduction	3.23	1.57	1.59	3.23
	% Reduction	73.07	25.44	46.35	38.77
	With two-directional absorber	0.793	2.34	1.703	3.00
	Reduction	3.627	3.83	1.727	5.33
	% Reduction	82.05	62.07	50.34	63.98
Cutting	Without absorber	6.5	4.186	4.979	9.19
	With single-direction absorber	4.49	1.75	1.79	5.15
	Reduction	2.00	2.43	3.18	4.04
	% Reduction	30.8	58.07	63.96	43.96
	With two-directional absorber	1.664	1.69	1.846	3.00
	Reduction	2.522	4.81	3.133	6.19
	% Reduction	60.24	74	62.92	67.35

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