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To develop algorithm to create node at desired location on beam using absorber

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Abstract - *The purpose of present investigation is to apply Imposing node method to reduce handle vibration of grass trimmer using variable stiffness vibration absorber. The numerical simulations are conducted to find the effectiveness of proposed method in handle vibration suppression. The laboratory experimental test is carried out to validate the numerical results. The results shows that the vibrations are reduced by 75% at handle support location using proposed method..*

Key Words: Grass trimmer, handle vibration, vibration absorber, imposing node, vibration isolation

1. INTRODUCTION

The powered grass trimmers are widely used in the general upkeep of parks and wooded areas in particular for the maintenance of the roadside and also of the railway lines where the outgrowth of the bushes on the railway track must be prevented. The operator of trimmer may subjected to large magnitude of hand arm vibration and may cause complex vascular and neurological and musculoskeletal disorder, collectively named as hand-arm vibration syndrome. Various techniques are used for controlling the vibration at the handle. These includes mounting of a dynamic vibration absorber on the grass trimmer, isolate the hand from the vibrating handle with the use of anti-vibration gloves [1] and isolate the tool handle form the vibrating source by using isolators [2]. The use of vibration isolators, however results in handle with a high mass and low stiffness. The isolation performance of the anti-vibration gloves for different hand tools is influenced by the excitation spectrum or tool type [3]. Alternatively, the excessive vibrations are suppressed by attaching a tunable vibration absorber to the grass trimmer. Hao and Ripin [4] applied imposing node technique to achieve vibration reduction at handle location of the petrol engine grass trimmer using two tunable vibration absorbers. Cha [5-9] discussed the scheme to impose nodes at required locations in a harmonically excited structure using simple oscillators. Patil and Awasare [10] proposed the algorithm to find required neutralizer parameters to create nodes at desired locations on beam.

In the present research imposing node technique is applied to reduce handle vibration of grass trimmer where the engine is mounted on the grass trimmer pipe. The algorithm based on iterative method is developed and applied to find absorber frequency to impose node at handle support location of grass trimmer. The numerical and experimental tests are conducted to show the viability of proposed method.

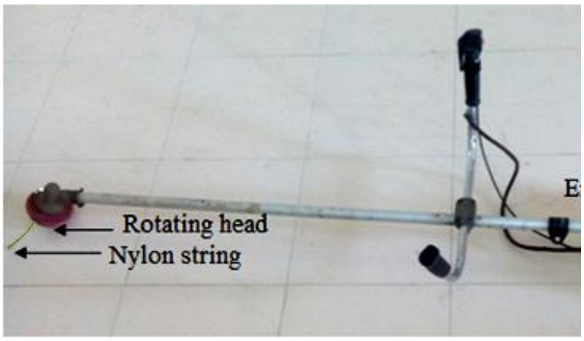

Fig.1 Grass trimmer system

1.1 Grass Trimmer System

A small petrol engine grass trimmer as shown in Figure (1) is considered in the present study. It is powered by an internal combustion engine mounted at one end of hallow circular pipe and rotating cutter head at other end. The nylon string attached to the cutter head cuts the grass. The specifications of the grass trimmer used for numerical and experimental tests are listed in Table 1 .The grass used for study is well-developed and dense for which the speed of grass trimmer was around 7000 to 7500 rpm. For laboratory and field test the speed of grass trimmer is selected as 7250 rpm. The hand-transmitted vibration is measured by means of the frequency-weighted root-mean-square (rms) acceleration as per ISO 5349-112. Figure 2(a) and 2(b) shows the axis system for the trimmer pipe and handle 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 of x, y and z direction and all the operating conditions remains same for three axis measurements ISO 5349-213.

Table-1 . Specifications of grass trimmer.

Engine	TB 43 Mitsubishi Japan
Displacement	42.7 cm³
Power output	1.3 H.P/ 6500 rpm
Fuel Tank Capacity	0.9 L
Handle	Double (W)
Pipe diameter	28mm
Gear ratio	14:19
Size (L x W x H)	1810 mm x 610 mm x 480 mm
Weight	8.3 Kg

1.2 Sub Frequency Analysis of Grass Trimmer

Sample paragraph, To find the primary source of vibration, the frequency spectrums of grass trimmer near engine support location and handle location for cutting condition are studied as shown in Figure (3). The acceleration spectrum of grass trimmer measured in x y and z axes are illustrated in Figure (4) for engine location. Only frequencies in the operating speed range 0-200 Hz are shown. Table 2 list the magnitudes of the acceleration in x y and z axes at engine and handle location of the grass trimmer.

Grass trimmer employed in this study had highest peak in the x-axes of magnitude 7.8 g at 121 Hz. This frequency is correlated with the speed of the engine 7250 rpm which is the primary source of vibration excitation. The vibration magnitude at 121 Hz in the y axis was 0.72 g. and peak in z-axis of magnitude 0.18g. There is a second peak in x y and z axis with 90 Hz which is due to rotating cutter head speed 5400 rpm (gear ratio 14:19). The magnitude of this source of excitation is small compared to engine excitation, hence neglected. Meanwhile near handle location the magnitude of vibration at 121 Hz in the x- axis was 4.7g, y axis was 0.77 g, and in z- axis was 0.52g. These results provide a guideline for the vibration suppression of grass trimmer used in the present study.

2.MATHMATICAL MODEL AND ALGORITHM DEVELOPMENT

The grass trimmer modelled 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 force of forcing amplitude F with frequency ω_e, $f(t) = Fe^{j\omega_e t}$, acting at $x_f = 0.98L$ due to engine vibration. The absorber is modelled as single degree of freedom spring-mass-damper system having mass m_1, variable stiffness k_1 and damping coefficient c_1 of the absorber. The system parameters and material properties used in the numerical and experimental tests are listed in Table 3. The weight of each end mass of the absorber is selected as 200 gm, so as to limit the percentage increase in the total weight of the trimmer system. Note that the structural damping of the absorbers used is considered as equivalent viscous damping as discussed in Dayou and Brennan[14].

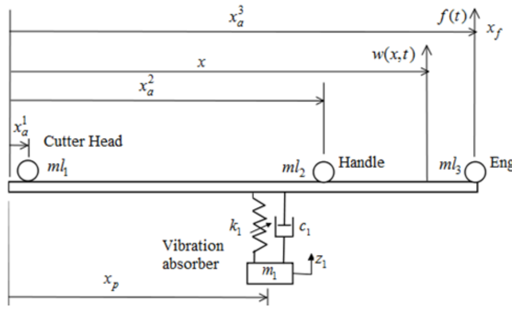

Fig.2 Model of grass trimmer with structurally damped vibration absorber.

Using the assumed-modes method the deflection of the beam(grass trimmer pipe) at any point x along the length is given by

$$w(x,t) = \sum_{i=1}^{N} \phi_i(x)\eta_i(t), \quad (1)$$

where N is the number of modes used in the assumed-modes expansion, $\phi_i(x)$ are the eigenfunctions of the undamped pipe and $\eta_i(t)$ are corresponding generalized co-ordinates.

Applying Lagrange's equations and assuming simple harmonic motion with same response frequency as the excitation frequency, the following equations of motion are obtained

$$\left(-\omega_e^2\begin{bmatrix}[M] & 0\\ 0^T & [m]\end{bmatrix} + j\omega_e\begin{bmatrix}[C] & [R_c]\\ [R_c]^T & [c]\end{bmatrix} + \begin{bmatrix}[K] & [R_k]\\ [R_k]^T & [k]\end{bmatrix}\right)\begin{bmatrix}\underline{\eta}\\ \underline{z}\end{bmatrix} = \begin{bmatrix}F\underline{\phi}(x_f)\\ 0\end{bmatrix} \quad (2)$$

where $\underline{\eta} = [\eta_1 \quad \eta_2 \quad \ldots \quad \eta_N]^T$, $\underline{z} = [z_1]^T$, $m = [m_1]$, $c = [c_1]$, $k = [k_1]$

The $N \times N$ $[M]$, $[C]$ and $[K]$ matrices of equation (2) are

$$[M] = [M^d] + \sum_{i=1}^{3} ml_i \underline{\phi}(x_a^i)\underline{\phi}^T(x_a^i),$$

$$[C] = [C^d] + c_1\underline{\phi}(x_p)\underline{\phi}^T(x_p),$$

$$[K] = [K^d] + k_1\underline{\phi}(x_p)\underline{\phi}^T(x_p) \quad (3)$$

where $[M^d],[C^d]$ and $[K^d]$ are diagonal matrices whose i-th elements are M_j, C_j and K_j are the generalized masses, damping and stiffnesses of beam.
Vector of the eigenfunctions of the beam

$$\underline{\phi}(x)=[\phi_1(x) \quad \phi_2(x)\ldots\phi_N(x)]^T$$

$$\underline{\phi}(x_a^i)=[\phi_1(x_a^i) \quad \phi_2(x_a^i)\ldots\phi_N(x_a^i)]^T \; \phi(x_p)=[\phi(x_p)]$$

$$\underline{\phi}(x_f)=[\phi_1(x_f) \quad \phi_2(x_f)\ldots\phi_N(x_f)]^T. \tag{4}$$

The matrices in equation (2) $[R_c]$ and $[R_k]$ are given by

$$[R_c]=[-c_1\phi(x_p)]$$
$$[R_k]=[-k_1\phi(x_p)] \tag{5}$$

The damping matrix of the beam in non dimensional form is given by

$$[C^d]=2\zeta_{bj}\omega_{bj}[I] \qquad j=1,\ldots N \tag{6}$$

In above equation the experimental approach can be used[15], to measure the damping in a beam by using experimental modal analysis which determines the modal damping ratios ζ_{bj} and natural frequencies ω_{bj}.

Using second equation of Equation (2), the $\overline{z}_1$ are found to be

$$\overline{z}_1=\frac{\omega_1^2+j\omega_e c_1/m_1}{\omega_1^2-\omega_e^2+j\omega_e c_1/m_1}\phi_i\overline{\eta} \tag{7}$$

In Equation (7) resonance frequency ω_1 and damping coefficient c_1 of the absorber are given by

$$\omega_1=\sqrt{k_1/m_1} \quad c_1=2\zeta_1\omega_1, \tag{8}$$

where ζ_1 is the damping ratio of the absorber
Equation (7) is substituted into the first equation of equation (2) and then solving for $\overline{\eta}$ to obtain

$$\overline{\eta}=\left\{-\omega_e^2[M]+j\omega_e^2[C^d]+[K^d]+\sigma_1\phi(x_1)\phi^T(x_1)\right\}^{-1}F\phi(x_f) \tag{9}$$

Where,

$$\sigma_1=\frac{(m_1\omega_1^2+jc_1\omega_e)\omega_e^2}{\omega_e^2-\omega_1^2-j\,c_1/m_1\,\omega_e} \tag{10}$$

Substituting equation (9) in to equation (2) to induce node at desired locations, x_q along the beam requires that

$$W(x_q)=\underline{\phi}^T(x_q)\left\{-\omega_e^2[M]+j\omega_e[C^d]+[K^d]+\frac{(m_1\omega_1^2+jc_1\omega_e)\omega_e^2}{\omega_e^2-\omega_1^2-j\,c_1/m_1\,\omega_e}\phi(x_p)\underline{\phi}^T(x_p)\right\}^{-1}$$

$$F\underline{\phi}(x_f)=0 \tag{11}$$

Equation (11) is used to find the resonance frequency of the absorber ω_1 for absorber mass m_1 and damping coefficient c_1 at which the value of the displacement of beam $W(x_q)$ become zero to impose nodes at x_q.

The algorithm based on iterative procedure developed in reference Patil and Awasare (2016) is modified to accommodate the damping in the absorber and the procedure is as given below.
The algorithm procedure to find the frequency of absorber ω to impose a node

1) Set the initial frequency of the absorber $\omega<\omega_e$.
2) Compute $W(x_q)$ using equation (11).
3) Increase the frequency of absorber ω in steps up to $\omega>\omega_e$ and find $|W(x_q)|$ for each increment.
4) Select the frequency of absorber ω_1 at which $|W(x_q)|$ is minimum.

It should be noted that if the method does not converge to zero value of $|W(x_q)|$, or does not give the feasible value of ω (the value of ω return by iteration is very high), this implies that the node cannot be enforced at the selected location for the given value of mass m. In such cases the new value of mass m is selected, to get a feasible value of ω and to $|W(x_q)|$, converge to zero. In addition, absorber attachment location x_p can also be changed if no feasible solution is found.
It is important to mention that for damped beam and absorber, the amplitudes i.e. $W(x_q)$ and $\overline{z}_1$, are complex quantities. Hence absolute values are used for comparison, for plots and for list in the tables.

3. NUMERICAL SIMULATIONS

The code is developed in MATLAB using algorithm developed in the previous section to perform the numerical simulations for imposing nodes at the handle support location on trimmer pipe using vibrations absorber. The eigenfunctions $\phi_i(x)$, generalized masses M_i and generalized stiffnesses K_i for grass trimmer which is modelled as free-free beam are given by

$$\phi_i(x)=\frac{1}{\sqrt{\rho L}}\left(\sinh\beta_i x+\sin\beta_i x+\frac{\sin\beta_i L-\sinh\beta_i L}{\cosh\beta_i L-\cos\beta_i L}(\cos\beta_i x+\cosh\beta_i x)\right) \tag{12}$$

$$M_i = 1 \quad \text{and} \quad K_i = (\beta_i L)^4 \, EI/(\rho L^4) \tag{13}$$

where $\beta_i L$ satisfies the following transcendental equation

$$\cos \beta_i L \cosh \beta_i L = 1 \tag{14}$$

In the following simulation results the masses, frequencies and vibration amplitudes are non-dimensionalized by dividing by ρL, $\sqrt{EI/(\rho L^4)}$ and $F/(EI/L^3)$ respectively. The plots are generated as shown in Figure 6 (a), (b) and (c), for absorber resonance frequency, vibration amplitude of the beam at the node location and absorber mass amplitude for different absorber attachment locations for given absorber mass. The plot shows that the trimmer pipe displacement at node location is minimum for absorber attachment location $x_p = 0.655L$. Choosing this absorber attachment location, resonance frequency required to impose node obtained from the plot is $41.74\sqrt{EI/(\rho L^4)}$ i.e. 121.56 Hz. The absolute steady state response of the trimmer pipe with and without absorber is shown in Figure (7). Responses shows that vibrations are suppressed to a great extent at node locations $x_q = 0.68L$ therefore isolating trimmer handle form vibration.

3.1 Vibration reduction of grass trimmer handle due to variations in speed

The operating speed of grass trimmer varies during its operation. In order to evaluate the effectiveness of absorber with variations in operating speed, simulation is conducted to find the response of trimmer due to change in excitation frequency. Figure (8) shows the absolute steady state response of the trimmer pipe with and without absorber with the variation in speed from 7200 rpm to 7320 rpm. The dotted line in figure represents the response of trimmer pipe without absorber. It is observed that the node position gets shifted from handle location with change in speed till the displacement at handle location remains small compared to the displacement without absorber.

The petrol engine grass trimmer is operated at the speed which suited the task. The speed of the trimmer for medium dense grass is around 6000 rpm whereas for well-developed and dense grass, the cutting speed is around 7500 rpm. The total operating range for the trimmer is around 1500 rpm. In the present study, test is conducted for cutting dense grass at speed around 7250 rpm and simulation results show that absorber is effective for the speed range of ± 60 rpm. In order to apply imposing node method for medium dense grass the absorber resonance frequency need to be change with the change in speed of the grass trimmer and this is the limitation of the present absorber.

4. EXPERIMENTAL TESTING OF TRIMMER WITH AND WITHOUT ABSORBER

In first step a tuned vibration absorber was designed and constructed as illustrated in Figure 9(a). The dimensions and material properties of the absorber are listed in Table 4. The experimental modal analysis of the absorber was carried out to determine the resonance frequencies of absorber for different mass positions on the rods. Figures 9(b) describe the relationship between the mass position and the resonance frequencies of the absorber. The weight of the absorber is 480gm and the percentage increase in the weight of grass trimmer system due to absorber is 0.06%. Next, Figure (10) 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 in Figure (10). The vibration amplitudes are measured at thirty points on the trimmer surface by the accelerometer and recorded by vibration analyzer to plot experimental steady state response of trimmer pipe rotating at 7250 rpm (i.e.

$\omega_e \approx 41.5\sqrt{EI/(\rho L^4)}$ or $f_e = 121$ Hz).

To find the effectiveness of the absorber the x direction, the absorber is mounted on the grass trimmer pipe at location 980mm i.e. $x_p = 0.655L$ and it is tuned by moving masses in or out such that the vibration at handle mount location of pipe is reduced to the minimum level. The vibration amplitudes were measured on the trimmer surface by the accelerometer and recorded by vibration analyzer to plot experimental steady state response of trimmer pipe with absorber. Response in Figure 11 (a) shows that vibrations are suppressed to a great extent at node locations therefore isolating trimmer handle form vibration. Comparing the steady state deformed shapes of trimmer pipe obtained from numerical experiments and experimental test i.e. Figure (7) with Figure 11(b), there is good agreement between numerical and experimental results.

It is evident form experimental steady state response of trimmer pipe shown in Figure 11(a), that the amplitude of vibration is suppressed at handle location i.e. at 1020 mm from 101 micron to 25 micron. The percentage reduction in the amplitude at handle location is 75%. The amplitude at handle location can be further reduced by increasing the absorber mass but it increases the total weight of the system. Note that the displacement at node location cannot reduce to zero due to presence of the damping in the absorber and trimmer pipe.

5. CONCLUSIONS

The purpose of this research is the investigation of imposing node method in vibration reduction of continuous elastic structure i.e. beam subjected to harmonic excitation using vibration absorbers. The following conclusions are accomplished from the investigation carried in this research work

1. The variable stiffness tunable vibration absorbers are effectively used to reduce the vibration by imposing nodes at desired locations on the beam..
2. Novel algorithms, to find the absorber parameters to impose nodes at chosen locations on beam subjected to harmonic excitations, are presented. The generalized program is developed which is used to determine the absorber parameters and it is easily modified to accommodate beam with different boundary conditions.
3. The algorithm is simple, easy to code and the same algorithm is utilized for damped and undamped absorbers.
4. The variable stiffness vibration absorber is used to suppress the handle vibration of petrol engine grass trimmer by imposing node at handle support location
5. The absorber frequency required to impose node at desired location on trimmer pipe is determined by the novel procedure developed in the present investigation. The plots are generated to assist the selection of the absorber attachment location on trimmer pipe.
6. The numerical simulation shows that the vibrations of trimmer pipe are suppressed by the absorber.

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