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Simultaneous Resonance Suppression in Response to Multiple Excitation Sources of a Brush Cutter

Yuki Koike, Shingo Tsuruoka

***Abstract*— Even with brush cutters that are equipped with rubber-mount based anti-vibration systems, handle vibration can worsen in the high engine speed zone. This may be caused by main pipe resonance which is excited by cutting head and engine vibration. In this study, we constructed a design method of dynamic vibration absorber (DVA) for suppressing the main pipe resonance simultaneously in response to both the first-order inertial force of the cutting head and engine in the high engine speed zone. We focused on the placement spans of rubber bushings that support a driveshaft inside the main pipe. By establishing two widened bushing placement spans, we utilized the bending modes of the driveshaft in these two widened span areas as multiple DVAs, called span-tuning dynamic vibration absorbers (ST-DVAs). Analysis results of the finite element method (FEM) showed that the ST-DVAs could control anti-resonance frequencies independently in each response to the first-order inertial force of the cutting head and engine. In handle vibration measurement during actual operation, ST-DVAs generated anti-resonances at the target frequencies in response to the above-mentioned excitations. As a result, at the target engine speed, ST-DVAs reduced handle vibration by 71% compared with that produced via equal-span rubber bushing placement. Hence, our study provides a design method of DVA for suppressing resonance simultaneously in response to multiple excitation sources of a brush cutter by only altering the bushing placement spans without any additional devices.**

***Index Terms*— damping, dynamic vibration absorber, grass trimmer, hand-arm vibration**

I. Introduction

BRUSH cutters are hand-held engine powered machines often used to clear land for agriculture or landscaping. Brush cutters have two main excitation sources of vibration, i.e., engine and cutting head rotational force. When designing brush cutters, reducing the handle vibration from these excitation sources is critical because long-term operators are at risk of developing hand-arm vibration syndrome (HAVS) due to excessive vibration exposure. Therefore, a number of engineering studies have been conducted aimed at reducing the vibration by using methods such as shape modification, mass distribution adjustment, and addition of typical spring-mass devices, i.e., dynamic vibration absorber (DVA) [1] – [4].

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In the actual industrial product field, brush cutters equipped with rubber-mount based anti-vibration systems are in widespread use. The anti-vibration systems can reduce the handle vibration over a wide range of frequencies.

However, even with this kind of anti-vibration type brush cutters, handle vibration can worsen in the high engine speed zone that is often used during operations. One of the main causes is that the resonance of the combined bending mode of the main pipe and driveshaft (inside the main pipe) is excited by the first-order inertial force of the cutting head and engine at specific engine speed. Here, the frequency components of these two inertial forces are different at a specific engine speed because of reduction gears (Fig. 1). Therefore, it is important to find a way to suppress the resonance simultaneously in response to these two excitation frequencies.

The aim of our study is to construct a design method that suppress the main pipe resonance simultaneously in response to both the first-order inertial force of the cutting head and engine at a specific engine speed in the high engine speed zone.

In this study, we changed the placement span of rubber bushings, which are components that support the driveshaft (Fig. 1), for suppressing the resonance. In typical brush cutters, multiple rubber bushings are placed at equal intervals. The advantage of this kind of vibration suppression method is that there is no influence to the total weight and the frame stiffness of the brush cutters. Furthermore, there is no need to add devices such as traditional DVA [3], [4].

Our previous study [5] and Okubo et al. [6] proposed methods for reducing handle vibration by altering bushing placements. Okubo et al. [6] used random iterative calculations and optimized the rubber bushing placements for reducing the handle vibration from the engine excitation force. In our previous study [5], we proposed a method to utilize a bending mode vibration of the driveshaft to act as a dynamic vibration absorber (called "ST-DVA") by altering the placement span of the rubber bushings in local region. Thus, we constructed a design method to suppress the main pipe resonance by generating an anti-resonance at a single target frequency in response to the first-order inertial force of the cutting head.

In the present paper, we propose an applied ST-DVA design method for generating anti-resonances simultaneously at target frequencies in response to multiple excitation sources, i.e., both the first-order inertial force of the cutting head and engine.

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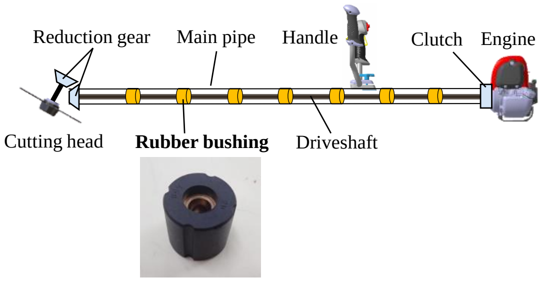

Fig. 1. Basic brush cutter structure. Adaptation drawing of our previous paper [5] with permission of American Society of Mechanical Engineers ASME; permission conveyed through Copyright Clearance Center, Inc.

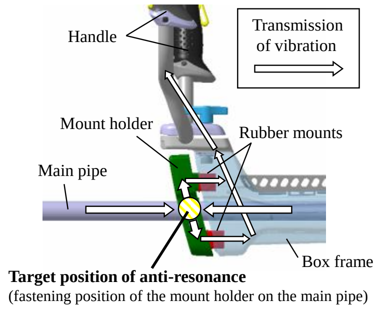

Fig. 3. Vibration transmission path and target anti-resonance position. Adaptation drawing of our previous paper [5] with permission of American Society of Mechanical Engineers ASME; permission conveyed through Copyright Clearance Center, Inc.

II. Main Concept and Analysis Method

A. *Structure of the anti-vibration type brush cutter*

Here, we explain the structure of the anti-vibration type brush cutter targeted in this study (Fig. 2). The primary components are the main pipe, engine, cutting head/gear case, handle, and anti-vibration system. In the anti-vibration system, a handle pipe is mounted on a box frame via a handle holder. The box frame is supported by four rubber mounts. The four rubber mounts are held in place by a mount holder and a clutch housing fastened to the main pipe. Therefore, the handle is supported in floating condition with respect to the main pipe. Thus, the vibration transmitted to the handle from the cutting head and engine can be reduced over a wide range of frequencies.

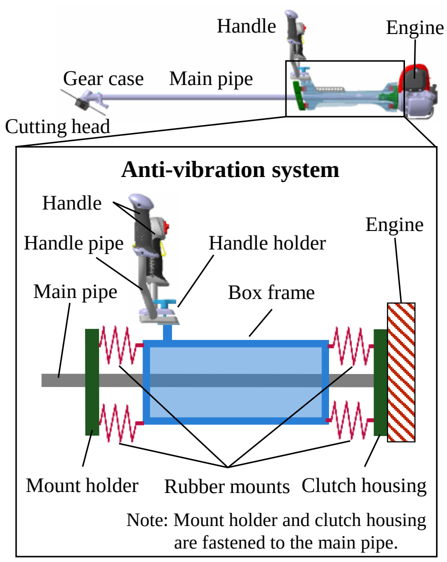

Fig. 2. Structure of anti-vibration type brush cutter. Adaptation drawing of our previous paper [5] with permission of American Society of Mechanical Engineers ASME; permission conveyed through Copyright Clearance Center, Inc.

Fig. 3 shows the transmission path of the main pipe resonance vibration and the target position for generating anti-resonance. The main pipe resonance vibration is transmitted to the handle via the mount holder. Therefore, to reduce the vibration transmission to the handle, generating anti-resonance at the mount holder fastening position on the main pipe is effective. In this study, we targeted to generate anti-resonance at this position in each response to the cutting head and engine excitation.

B. *Concept of span-tuning dynamic vibration absorbers (ST-DVAs)*

Firstly, we explain about the ST-DVA that is for a single excitation frequency (Fig. 4). Full details of the ST-DVA are described in our previous study [5]. As shown in the figure, a widened bushing placement span should be established (labeled as “Wide span area”). Here, the target position of anti-resonance should be included in this wide span area. The first bending mode of the driveshaft supported at both ends by two rubber bushings with this wide span area is regarded as the ST-DVA. By changing the placement span of the rubber bushings #5 and #6, the ST-DVA can generate an anti-resonance at a single target frequency in response to the single excitation source (e.g., first-order inertial force of the cutting head).

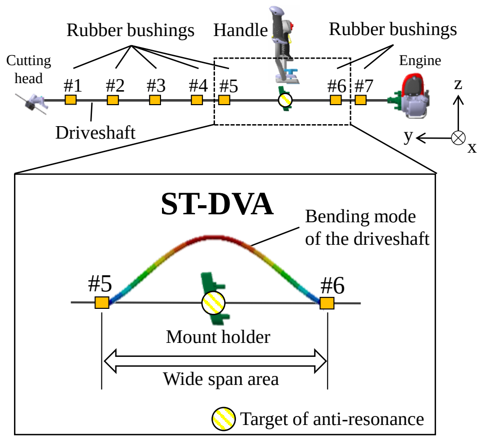

Fig. 4. ST-DVA outline. Adaptation drawing of our previous paper [5] with permission of American Society of Mechanical Engineers ASME; permission conveyed through Copyright Clearance Center, Inc.

Next, we explain about the span-tuning dynamic vibration absorbers (ST-DVAs), which is an applied design method of ST-DVA, for suppressing resonance simultaneously in response to both the first-order inertial force of the cutting head and engine at a target engine speed.

Fig. 5 shows the ST-DVAs outline. As shown in the figure, two ST-DVA were established. The ST-DVA-#1 corresponds to the wide span area between the bushings #5 and #6, and the ST-DVA-#2 corresponds to the wide span area between the bushings #2 and #3. The ST-DVA-#1 was installed so that the target position of anti-resonance was within the area between the bushings #5 and #6.

The ST-DVA-#1 was used for the excitation source that has greater influence on the handle vibration selected from

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either the first-order inertial force of the cutting head or engine. The ST-DVA-#2 was used for the other excitation source. Such an arrangement setting was based on the assumption that the vibration reduction effect was greater when the DVA was installed near the target position for generating anti-resonance. In this study, we used the ST-DVA-#1 for the cutting head excitation.

The natural frequency of the ST-DVA-#1 was tuned by the placement span of the bushings #5 and #6. The natural frequency of the ST-DVA-#2 was tuned by the placement span of the bushings #2 and #3. As shown in Fig. 6, these natural frequencies were calculated via finite element method (FEM) analysis model where the positions of the outer cylindrical surface of each rubber bushing #1 to #7 and both ends of the driveshaft were constrained. The driveshaft was connected to the inner surface of each rubber bushing under the condition that relative displacement was free only in the y-axis direction. Table I shows the specifications of the components.

As shown in Fig. 5, one of the remaining rubber bushings #4 was placed between the rubber bushings #3 and #5. By constraining the driveshaft displacement between the rubber bushings #3 and #5 in this way, each natural frequency of the ST-DVA-#1 and ST-DVA-#2 can be tuned independently to each other. The remaining two rubber bushings #1 and #7 were placed manually at any position near the cutting head and the engine respectively.

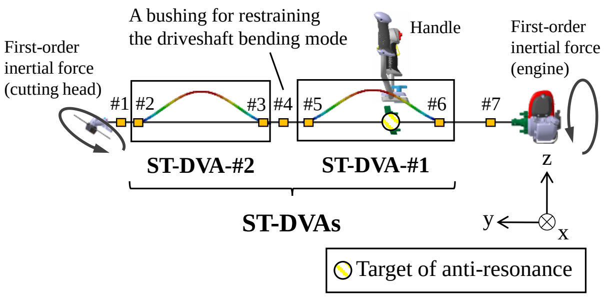

Fig. 5. Outline of ST-DVAs for multiple excitation sources.

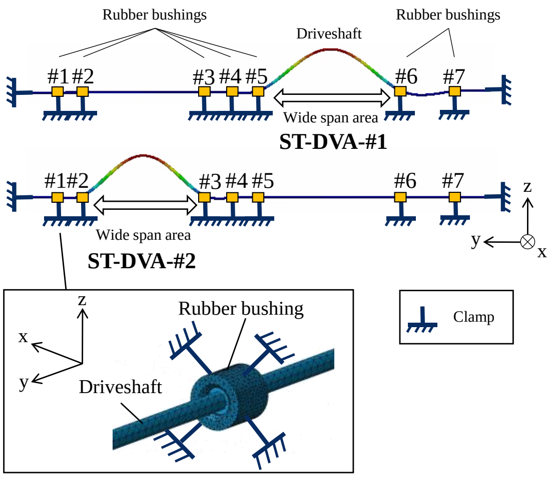

Fig. 6. Natural frequency analysis of driveshaft and rubber bushings.

TABLE I
DRIVESHAFT AND RUBBER BUSHING SPECIFICATIONS

	Young's modulus (MPa)	Mass (kg)	Diameter (mm)	Length (mm)
Rubber bushing	20	0.02	24	23
Driveshaft	200,000	0.64	8	1546

C. Finite element method model for frequency response analysis

The FEM model for frequency response analysis in this study is practically same as described in our previous study [5]. More details, which include properties of each element are described there.

Fig. 7 shows an outline of the FEM model for frequency response analysis. The FEM model that consists of 3D shapes and 1D elements was used. The cutting head and the engine were modeled as lumped and inertia mass.

Fig. 8 shows the input setting of sine waves for simulating the first-order inertial force of the (a) cutting head and (b) engine. The magnitude of each first-order inertial force was input as a function that increases with the square of the frequency. For two orthogonal axial directions, sine waves with phases shifted by 90 ° were input in the case of Fig. 8(a) and (b).

Using this FEM model, we evaluated the frequency responses on the main pipe at the mount holder fastening position that was the target position for generating anti-resonance described in Fig. 3. We calculated the frequency responses for each first-order inertial force of the cutting head and engine individually. The rubber bushing placements applied to the frequency response analysis in this study are shown in Table II.

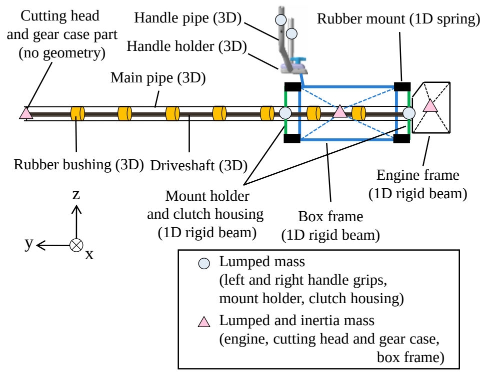

Fig. 7. Outline of FEM model for frequency response analysis. Adaptation drawing of our previous paper [5] with permission of American Society of Mechanical Engineers ASME; permission conveyed through Copyright Clearance Center, Inc.

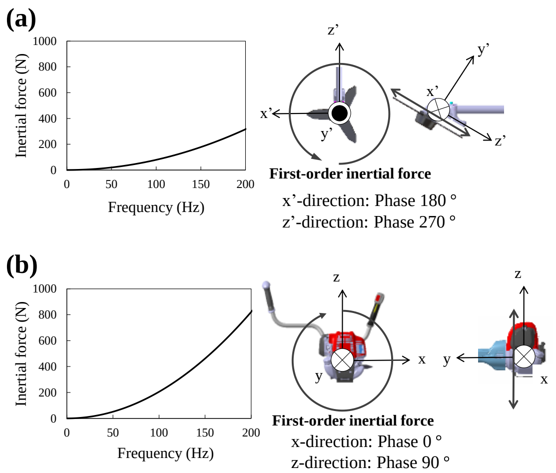

Fig. 8. Excitation force magnitude and direction/phase input setting of sine waves for simulating first-order inertial force: (a) cutting head excitation and (b) engine excitation. (a) is adaptation drawing of our previous paper [5] with permission of American Society of Mechanical Engineers ASME; permission conveyed through Copyright Clearance Center, Inc.

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TABLE II
RUBBER BUSHING PLACEMENTS CORRESPONDING TO EACH NATURAL FREQUENCY OF ST-DVAS AND EQUAL-SPAN PLACEMENT

Natural frequencies of ST-DVAs		Rubber bushing placements[a] (mm)						
ST-DVA-#1	ST-DVA-#2	#1	#2	#3	#4	#5	#6	#7
120 Hz	110 Hz	1433	1405	889	848	758	281	98
120 Hz	115 Hz	1433	1399	895	848	758	281	98
120 Hz	140 Hz	1433	1373	921	848	758	281	98
120 Hz	155 Hz	1433	1359	935	848	758	281	98
Equal-span placement		1313	1125	938	750	563	375	188

[a]Note: Origin point is the end of the main pipe on engine side. Rubber bushing placements are the distances from the origin point in the longitudinal direction of the main pipe.

III. ANALYSIS RESULTS AND DISCUSSION

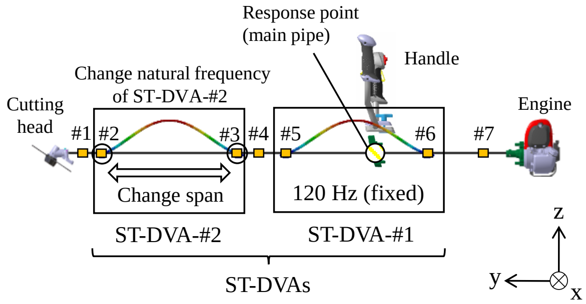
A. *Frequency response analysis results*

Fig. 9 shows the frequency response of acceleration on the main pipe at the mount holder fastening position. The natural frequency of the ST-DVA-#1 was fixed at 120 Hz and that of the ST-DVA-#2 was changed by the placement span of the bushings #2 and #3. Fig. 9(a) and (b) are the acceleration response to the first-order inertial force of the cutting head and engine respectively. The x-axis and z-axis directions were taken as the response directions to the first-order inertial force of the cutting head and engine respectively. The critical damping ratio was set at 0% to make it easier to identify anti-resonance. The response results of equal-span placement are also shown for reference.

As shown in Fig. 9(a), anti-resonance frequency was observed at 120 Hz in each case of where the ST-DVAs were established. Here, the anti-resonance frequency coincided with the natural frequency of the ST-DVA-#1. On the other hand, as shown in Fig. 9(b), anti-resonance frequency changed depending on the natural frequency of the ST-DVA-#2. However, there was a discrepancy between the anti-resonance frequency and the natural frequency of the ST-DVA-#2.

What is important here is that by fixing the natural frequency of the ST-DVA-#1 to 120 Hz, the anti-resonance frequency in response to the cutting head excitation was kept at 120 Hz regardless of the change in the natural frequency of the ST-DVA-#2. This indicates that we can control the anti-resonance frequencies independently for each excitation force of the cutting head and the engine. In other words, by changing the natural frequency of the ST-DVA-#2 with that of the ST-DVA-#1 fixed at specific frequency, we can generate anti-resonances at each target frequency for both the first-order inertial force of the cutting head and engine.

When actually designing the ST-DVAs, we should tune the natural frequency of the ST-DVA-#2 while checking the anti-resonance frequency via the frequency response analysis as shown in Fig. 9(b). This is because of a frequency discrepancy between the natural frequency of the ST-DVA-#2 and the anti-resonance frequency generated by the ST-DVA-#2. This kind of frequency discrepancy occur when the ST-DVA is installed at the area that does not include target position of anti-resonance like ST-DVA-#2.



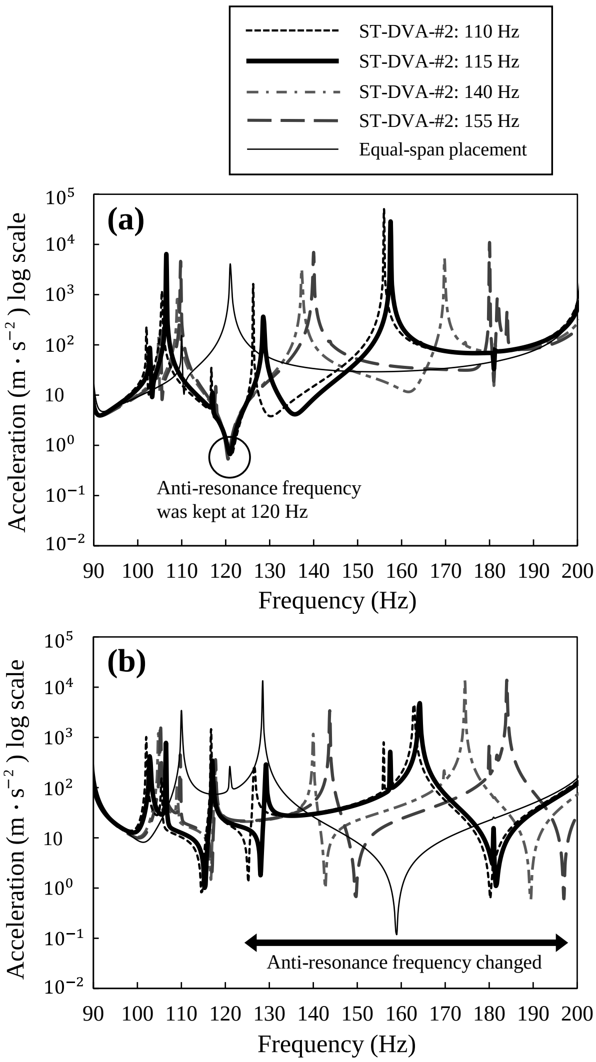


Fig. 9. Frequency response of acceleration: (a) response to cutting head excitation (x-axis direction) and (b) response to engine excitation (z-axis direction).

B. *Comparison of vibration motion*

Fig. 10 and Fig. 11 show the comparison of vibration motion between the case of (a) equal-span placement and the case of (b) where the ST-DVAs (ST-DVA-#1: 120 Hz, ST-DVA-#2: 115 Hz) were established. The critical damping ratio was set at 5% that approximately corresponds to that of the actual machine. The display magnification of the displacement in the figure was set at 1500 times to make it easier to compare the vibration motion. Fig. 10 is the response at 120 Hz to the first-order inertial force of the cutting head, and Fig. 11 is the response at 128 Hz to the first-order inertial force of the engine. Here, as shown in Fig. 9(a) and (b), these frequencies of 120 Hz and 128 Hz correspond to the resonance frequencies in the case of

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equal-span placement and also correspond to the anti-resonance frequencies in the case of where the ST-DVAs (ST-DVA-#1: 120 Hz, ST-DVA-#2: 115 Hz) were established.

As shown in Fig. 10(a) and Fig. 11(a), the combined third bending modes of the main pipe and the driveshaft were observed in response to both the cutting head and engine excitations in the case of equal-span placement. On the other hand, by establishing the ST-DVAs (Fig. 10(b) and Fig. 11(b)), the main pipe displacements at the target position of anti-resonance were suppressed in response to both the excitations. This indicates that the ST-DVAs can suppress the main pipe resonances simultaneously at each target frequency in response to both the first-order inertial force of the cutting head and engine.

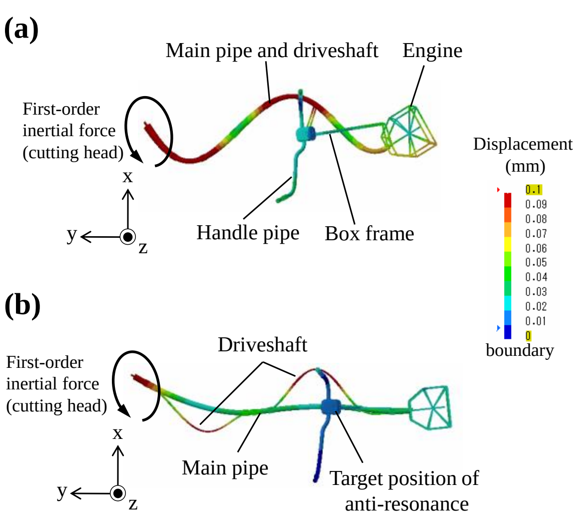

Fig. 10. Comparison of vibration motion at 120 Hz under cutting head excitation: (a) equal-span placement and (b) ST-DVAs (ST-DVA-#1: 120 Hz, ST-DVA-#2: 115 Hz).

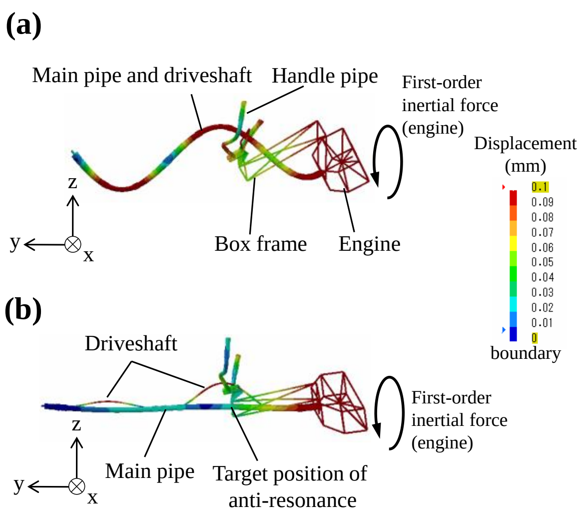

Fig. 11. Comparison of vibration motion at 128 Hz under engine excitation: (a) equal-span placement and (b) ST-DVAs (ST-DVA-#1: 120 Hz, ST-DVA-#2: 115 Hz).

IV. Actual Measurement Result of Handle Vibration

For verifying the effect of the ST-DVAs, handle vibration was measured in actual operation condition. The measurement target was the anti-vibration type brush cutter that was practically same specification with the FEM model as described in Sec. II. The operator held the handle during the measurement in accordance with ISO22867: 2011. The engine speed was swept from idle to full throttle. Taking the response point as the left handle, the acceleration response was measured in the xyz-directions using a triple-axis acceleration pickup. The acceleration response in each direction was filtered according to EN ISO5349-1. The three-axis composite values of each filtered acceleration response in the xyz-directions were evaluated as the handle vibration values of the first-order components of the cutting head and engine.

We set the target engine speed for vibration reduction at a racing speed as described in ISO22867: 2011. Here, the racing speed corresponds to 133% of the maximum output engine speed. The maximum output engine speed of the target brush cutter was 7000 rpm, therefore, the target engine speed for vibration reduction was set at 9310 rpm. The rotation speed ratio of the cutting head to the engine is 0.77 to 1.0, therefore, target frequencies in response to the first-order components of the cutting head and engine were 120 Hz and 155 Hz respectively at the engine speed of 9310 rpm. For suppressing the handle vibration at these target frequencies, the natural frequencies of the ST-DVAs were set as shown in Table III. Here, the natural frequencies of the ST-DVA-#1 and ST-DVA-#2 were fine-tuned considering that the assumed critical damping ratio of the target brush cutter was 5%.

Fig. 12 shows the measurement results for handle vibration where the rubber bushing placements were taken as shown in Table III. Fig. 12(a) and (b) are the tracking data for the first-order components of the cutting head and engine respectively. Fig. 12(c) is the square root value of sum of square of the first-order components of the cutting head and engine. From Fig. 12(a) and (b), acceleration levels of the cutting head component at 9310 rpm (120 Hz) and of the engine component at 9310 rpm (155 Hz) were reduced by 72% and 58% respectively compared with those produced via the equal-span placement. As shown in Fig. 12(b), the acceleration level of the engine component around 7200 rpm (120 Hz) was also reduced. This indicates that the ST-DVA-#1 with the natural frequency of 120 Hz suppressed the acceleration response level of the engine component of 120 Hz in addition to that of the cutting head component of 120 Hz.

As a result, as shown in Fig. 12(c), the square root value of sum of square of the first-order components of the cutting head and engine at the target engine speed of 9310 rpm was reduced by 71% compared with that produced via the equal-span placement.

These results indicate that the ST-DVAs generated anti-resonances simultaneously at the target frequencies in response to both the first-order components of the cutting head and engine.

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TABLE III
RUBBER BUSHING PLACEMENTS FOR MEASUREMENT

Natural frequencies of ST-DVAs		Rubber bushing placements[a] (mm)						
ST-DVA-#1	ST-DVA-#2	#1	#2	#3	#4	#5	#6	#7
120 Hz	163 Hz	1433	1353	941	848	758	281	98
Equal-span placement		1313	1125	938	750	563	375	188

[a]Note: Origin point is the end of the main pipe on engine side. Rubber bushing placements are the distances from the origin point in the longitudinal direction of the main pipe.

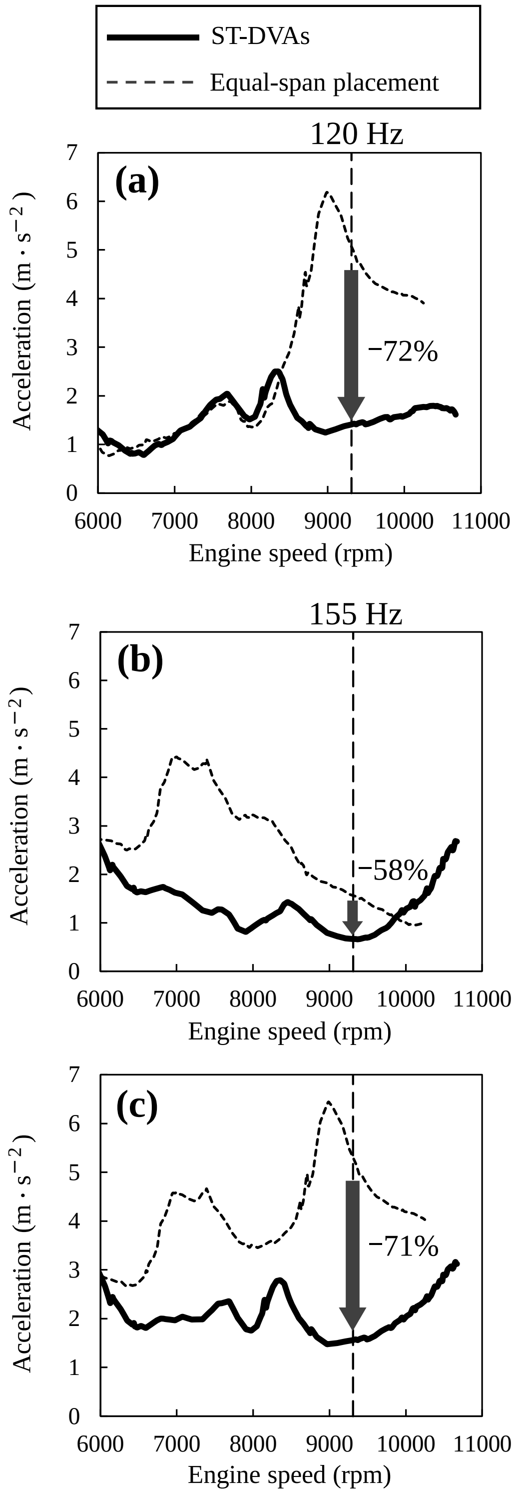

Fig. 12. Measurement results of handle vibration: (a) first-order component of cutting head, (b) first-order component of engine, and (c) square root value of sum of square of first-order components of cutting head and engine.

V. CONCLUSION

In this study, we have described a design method for suppressing the main pipe resonance simultaneously in response to multiple main excitation sources of the anti-vibration type brush cutter. The findings obtained are summarized below.

(I) By establishing two widened span areas of the rubber bushings, first bending modes of the driveshaft in these areas acted as multiple DVAs, called ST-DVAs. The natural frequencies of the ST-DVAs could be tuned by the placement spans of the rubber bushings.

(II) In FEM analysis, by tuning the natural frequencies of the ST-DVAs, the ST-DVAs controlled the anti-resonance frequencies independently for each first-order inertial force of the cutting head and the engine.

(III) Actual measurement results of the handle vibration indicated that the ST-DVAs generated anti-resonances simultaneously at the target frequencies in response to both the first-order components of the cutting head and engine excitation. Hence, at the target engine speed, the handle vibration of the square root value of sum of square of the first-order components of the cutting head and engine could be reduced by 71% compared with that produced via the equal-span rubber bushing placement.

The importance of our work is that the ST-DVAs can be constructed by only altering the placement spans of the rubber bushings, which are primary brush cutter components. Therefore, compared with the conventional multiple DVAs that has been practically used, the ST-DVAs has advantages in that it does not increase total weight and does not require any additional devices. Our findings can be applied to many brush cutters that have multiple bushings inside the main pipe.

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