

Development of Anti-vibration Type Brush Cutter for European Commercial Market

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ABSTRACT

Honda's first anti-vibration type brush cutter UMK450E XEET was developed for the European commercial hand-held market. This development established vibration reduction technology by focusing on the shape of the mount rubbers equipped to the anti-vibration frame and the driveshaft bearing placement. The mount rubber shape was newly designed to simultaneously satisfy demands for vibration reduction, swing stiffness during brush cutting work, and actual machine own-weight support. A new bearing placement design method was established that uses the driveshaft as a dynamic damper. This bearing placement design suppresses handle vibration variation due to the static unbalance amounts of the cutting head and the engine flywheel. The newly designed mount rubber and bearing placement realized a low vibration brush cutter that satisfies the user demands in the European commercial market.

1. Introduction

Honda has deployed and expanded sales of brush cutters equipped with 1-horsepower and 1.5-horsepower engines in the European consumer hand-held market. In this development, a brush cutter equipped with a 2-horsepower engine targeting the European commercial market has been newly developed.

Brush cutters are hand-held work machines that are used for grass cutting and other work by farmers and landscapers in the 2-horsepower class commercial market. Many users work for long times a day, especially in this market. According to the results of market research (Honda survey), the working time of home users in the consumer market averages about an hour a day, whereas users in the commercial market who are the target demographic of this development were found to work for up to about six hours a day. The excitation force due to the rotation of the cutting head and the engine constantly acts on the handle during the work, so low handle vibration is important for users who work for long times. In addition, users work while adjusting the engine speed with the throttle lever according to the work environment (load), so low handle vibration over a wide engine speed range is also important. To meet these demands, Honda developed the UMK450E XEET, which is the company's first brush cutter equipped with an anti-vibration frame structure. Figure 1 shows an overall image

of the developed anti-vibration type brush cutter.

This development established two main vibration reduction technologies in order to satisfy user demands in the European commercial market. First, the shape of the mount rubber, which is the main component of the anti-vibration frame structure, was newly designed. This reduced vibration over a wide engine speed range. Next, focusing on the bearing placement of the driveshaft that transmits the engine drive force to the cutting head, a new bearing placement design method was established. This reduced the variation of the handle vibration value relative to the increased excitation force due to the imbalance of the rotor. This paper introduces these vibration reduction technologies.



Fig. 1 Image view of UMK450E XEET

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2. Development Goals

2.1. EU Physical Agents (Vibration) Directive and Machinery Directive

The EU Physical Agents (Vibration) Directive 2002/44/EC⁽¹⁾, which is a key index when establishing target values for vibration reduction, is explained below. This directive determines the standard values for vibration exposure level as a minimum requirement for protecting workers from safety and health risks. The vibration exposure action value (EAV) and vibration exposure limit value (ELV) are set as the standard values for vibration exposure level. When the daily vibration exposure level exceeds the EAV, employers are obligated to take steps such as suppressing the vibration exposure time and selecting brush cutters with lower vibration. However, the daily vibration exposure level should not exceed the ELV under any circumstances.

In order to satisfy the EAV and ELV standard values, the allowable daily working time (vibration exposure time) is indicated as a guideline according to the handle vibration value. Figure 2 shows this relationship⁽²⁾. The horizontal axis of the graph is the vibration exposure time per day, and the vertical axis is the handle vibration value. The handle vibration value is expressed by the composite value of the handle vibration values at idling speed and racing speed (133% of engine speed at engine maximum power). The two curves represent the times until the vibration exposure level reaches the EAV and ELV, respectively, according to the handle vibration value. It is necessary to set a development target value for handle vibration that satisfies the working times of the target users based on the graph in Fig. 2.

In addition, the EU Machinery Directive 2006/42/EC⁽³⁾ requires development manufacturers to describe uncertainty along with the measured handle vibration value in the brush cutter operating instructions. Uncertainty here refers to the range of variation in the vibration acceleration that occurs in the measurement procedure and in manufacturing. Anti-vibration type brush cutters in particular have a small handle vibration value compared to brush cutters without an anti-vibration frame. For this reason, users more easily feel the magnitude of the vibration acceleration variation through the handle. This makes it important to reduce uncertainty as much as possible when developing anti-vibration type brush cutters.

2.2. Development Targets

Targets (1) and (2) below were set in this development.

(1) Handle vibration

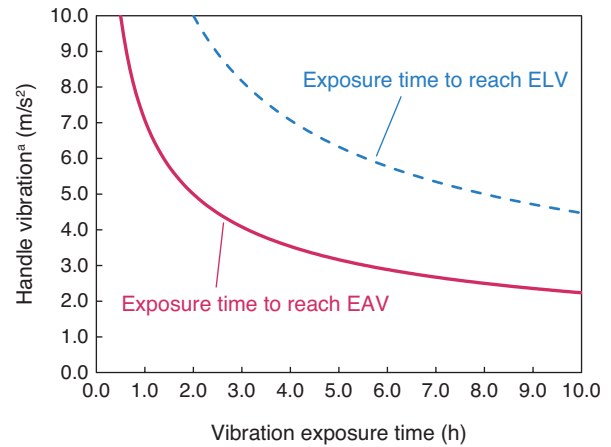
It is necessary to set a target value that can protect the target users, who according to market research work for up to about six hours a day, from safety and health risks based on the EU Physical Agents (Vibration) Directive 2002/44/EC⁽¹⁾. Therefore, a handle vibration value of 2.5 m/s² that satisfies the EAV even when working for eight hours a day was set as the development target for this development. (The handle vibration measurement method complies with

ISO22867-2011⁽⁴⁾.) An additional development target that resonance exceeding 2.5 m/s² does not occur over the entire practical engine speed range was also set.

(2) Uncertainty of handle vibration value

Users can freely replace the cutting head according to the work application. This means that cutting head replacement may unexpectedly increase the excitation force and excite resonance of the pipes and other parts. Such resonance is a factor increasing the uncertainty of the handle vibration value. A key to reducing uncertainty is to reduce the variation of the handle vibration value relative to the variation in the static unbalance amount of the equipped cutting head. In addition, it is also desirable to reduce the variation of the handle vibration value relative to the variation in the static unbalance amount of the engine flywheel, which is the main rotating component. In consideration of the respective variation in the static unbalance amounts of the cutting head and the engine flywheel, the target for this development was set to keeping the variation (uncertainty) of the handle vibration value to less than 2.0 m/s². (The handle vibration measurement method complies with ISO22867-2011⁽⁴⁾.)

In addition to the above-mentioned development targets (1) and (2), this development also took into account the swing stiffness described below from the viewpoint of enhancing merchantability. To reduce handle vibration, it is necessary to design small spring constants for the mount rubber equipped to the anti-vibration frame. However, when the spring constants of the mount rubber are small,



$$a: \text{Handle vibration (m/s}^2\text{)} = \left[\frac{1}{2} (Id^2 + Ra^2) \right]^{1/2}$$

Id : vibration value at idling speed (m/s²)

Ra : vibration value at racing speed (m/s²)

Note: Id and Ra are three-axis composite values of each filtered acceleration response (r.m.s. value). Racing speed is engine speed of 133% of speed at maximum engine power.

Fig. 2 Relationship between handle vibration and allowable vibration exposure time

the swing stiffness during brush cutting work decreases. Decreased swing stiffness here refers to an increase in the angular displacement delay of the working part (cutting head) relative to the angular displacement of the handles during the swing operation of brush cutting work. Figure 3 shows an image of angular displacement delay. The mount rubber was designed bearing in mind that such angular displacement delay of the working part may lead to reduced merchantability, such as reduced work efficiency and an increased feeling of fatigue due to an inability to operate as intended.

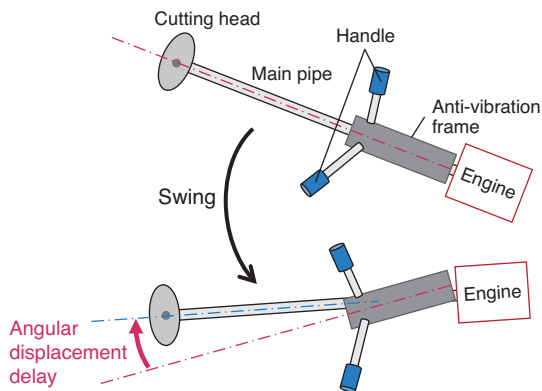


Fig. 3 Angular displacement delay of cutting head from handle swing motion

3. Vibration Reduction Technology

This development established a new mount rubber design and a new bearing placement design method to achieve the development targets described in Section 2.2.

3.1. Mount Rubber

3.1.1. Anti-vibration frame structure

Figure 4 shows an outline of the anti-vibration frame structure from the engine to the handle attachment part in the developed anti-vibration type brush cutter⁽⁵⁾. The anti-vibration frame has a structure that holds a resin box frame with two mount rubbers each in the front and rear for a total of four mount rubbers. The mount holder and clutch housing fastened to the main pipe support each mount rubber, and the handle pipe is fastened to the resin box frame via the handle holder. This type of anti-vibration frame structure enables reduction of the vibration transmitted to the handle from the cutting head and engine, which are sources of vibration.

Next, the features of the mount rubber shape and mount rubber support structure are described. Figure 5 shows an example of a mount rubber shape in the examination process, and Fig. 6 shows the mount rubber support structure. The mount rubber shape has an outer wall part

and a center hole-shaped part connected by multiple ribs. As shown in Fig. 6(a), the resin box frame has support parts on the front and rear ends into which the outer wall parts of the mount rubbers are inserted. Meanwhile, the mount holder and clutch housing have structures with pin-shaped protrusions in two places each, as shown in Fig. 6(b). The structure supports the mount rubbers from the inner wall side by inserting these pin-shaped protrusions into the hole-shaped parts in the center of the mount rubbers. Figure 6(c) shows an image of the condition with each part assembled. As a result, this structure supports the mount rubbers by the outer wall part and the center hole-shaped part, and reduces the vibration transmission rate with the rib shapes in-between.

3.1.2. Mount rubber design requirements

(1) Spring constant of vibration displacement region

The region where the hole-shaped part in the center of the mount rubber is displaced by an input load due to vibration in the condition with the user holding the handles was defined as the vibration displacement region. Based on

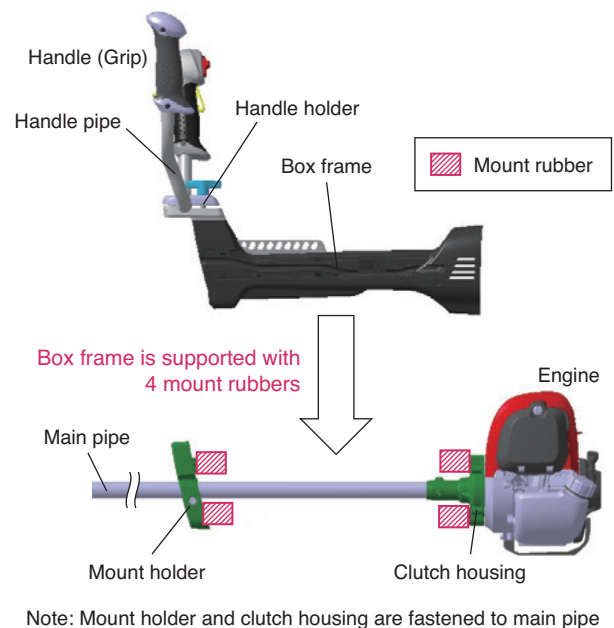


Fig. 4 Anti-vibration frame structure

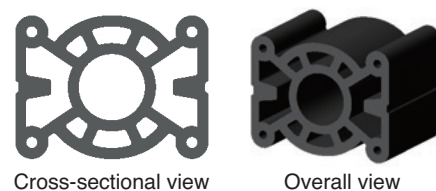


Fig. 5 Mount rubber

preliminary testing results, the expected displacement in this region was set to less than around 3 mm. In such a vibration displacement region, reduction of the spring constant is demanded. To achieve the target handle vibration value, this development examined the shape on the premise of a mount rubber shape with many gaps, as shown in Fig. 5. Here, from the viewpoint of “(3) Actual machine own-weight support” described hereinafter in this chapter, there is a concern of large deformation and buckling of the mount rubber if the spring constant in the actual machine own-weight direction is too small. Therefore, in the vibration displacement region, the basic guideline was to reduce the spring constant of the swing direction displacement.

(2) Spring constant of swing displacement region

The region where the hole-shaped part in the center of the mount rubber is displaced by the input load of swing operation as shown in Fig. 3 was defined as the swing displacement region. Based on preliminary testing results, the expected displacement in this region when the mount rubber shape has many gaps, such as shown in Fig. 5, was set to around 3 mm or more. In such a swing displacement region, it is necessary to increase the spring constant to secure sufficient swing stiffness. To achieve the contrary

spring constant demands of both the vibration displacement and swing displacement regions, this development designed the mount rubber shape so that the spring constant in the swing displacement direction is nonlinear according to the displacement.

(3) Actual machine own-weight support (low vibration, durability)

The developed anti-vibration type brush cutter has a structure that supports the actual machine own-weight of approximately 10 kg with mount rubbers in four locations. The mount rubbers use a shape with many gaps to reduce vibration, so there is a concern that if the mount rubbers deform greatly or buckle due to the actual machine own-weight, the increased spring constant may lead to increased vibration and reduced durability. Therefore, from the viewpoint of durability and low vibration, consideration was given to avoid the occurrence of large deformation and buckling due to the actual machine own-weight.

3.1.3. Examination of mount rubber specifications

The specification examination processes for satisfying each design requirement of the mount rubber are described below.

(1) Spring constant of vibration displacement region

In examining the spring constant in the vibration displacement region, multiple levels of mount rubbers with different spring constants were prototyped in the initial development stage. Based on the correlation between the spring constant and the measured handle vibration value, the spring constant (spring constant in the swing direction: 5.2 N/mm) that achieved the target handle vibration value was set as the design target value.

Next, Computer Aided Engineering (CAE) analysis was used to conduct a mount rubber shape parameter study. Specifically, enforced displacement analysis was performed on various mount rubber shapes with different thickness distributions and numbers and arrangement of ribs, as shown in Fig. 7, and the spring constant of each

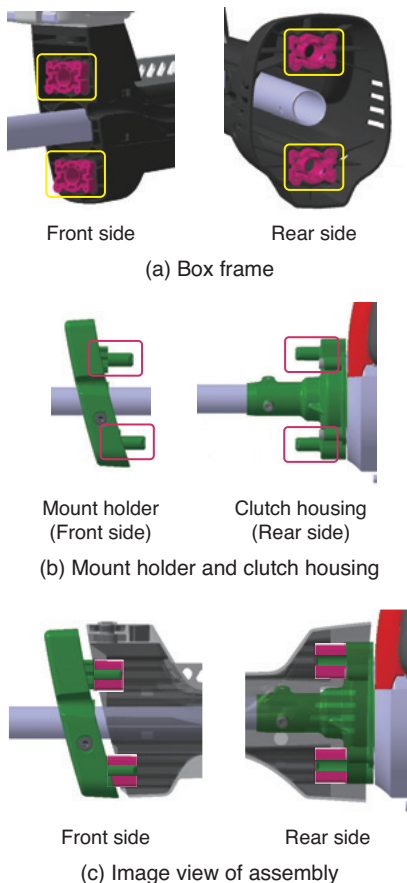


Fig. 6 Support structure of mount rubbers

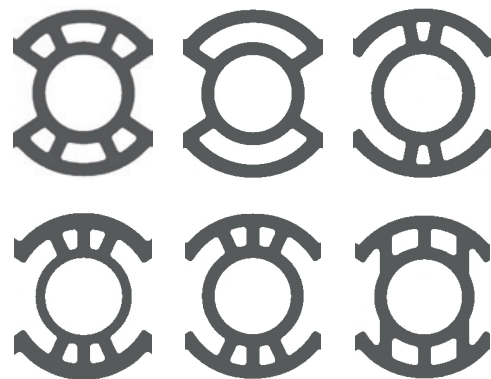


Fig. 7 Examples of mount rubber shape in parameter study

shape was calculated. Figure 8 shows an image of enforced displacement analysis. The two parts of a simple box frame shape simulating the mount rubber insertion part and the mount rubber were modeled. This model was used to calculate the spring constant by outputting the reaction force value when the hole-shaped part in the center of the mount rubber is forcibly displaced. Based on the tendencies understood through the parameter study, a shape was found that satisfies the target spring constant (Fig. 9).

(2) Spring constant of swing displacement region

Comparative verification was performed with a competing anti-vibration type brush cutter, and the spring constant target values were set to secure sufficient swing stiffness. To achieve the spring constant target values of both the vibration displacement and swing displacement regions, spring constant characteristics that are nonlinear relative to swing direction displacement, as shown in Fig. 10, were set as the design target. Note that the slope of

the graph corresponds to the spring constant.

A structure with protrusion shapes on both ends relative to swing direction displacement was adopted to satisfy the nonlinear spring constant characteristics shown in Fig. 10. CAE enforced displacement analysis was performed to adjust the protrusion shape. Figure 11 shows the protrusion shapes of the initial and final specifications, and Fig. 12 shows the CAE analysis results of the spring constants corresponding to each shape. The protrusion shape of the final specification achieved high spring characteristics with a spring constant (graph gradient) that is the target gradient or more in the swing displacement region, while also achieving low spring characteristics with a graph gradient that is the target gradient or less in the vibration displacement region.

(3) Actual machine own-weight support (low vibration, durability)

First, the shapes that realize low vibration in the state

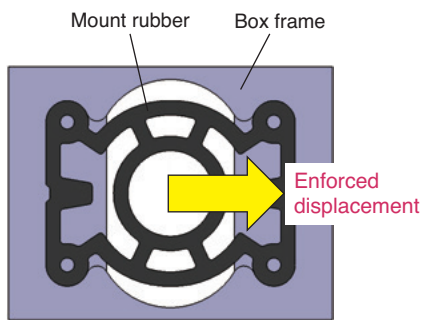


Fig. 8 Image view of enforced displacement CAE analysis

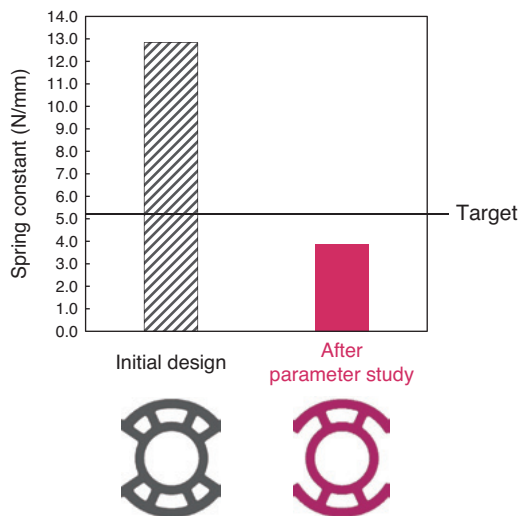


Fig. 9 CAE results of each mount-rubber spring constant

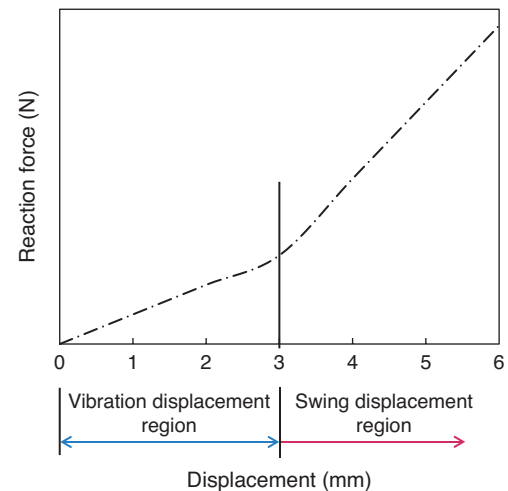


Fig. 10 Target nonlinear spring constant characteristic

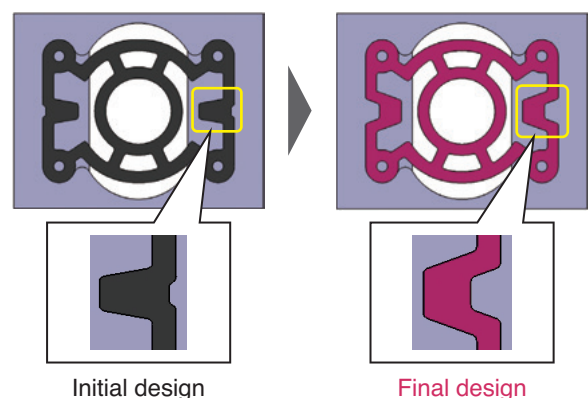


Fig. 11 Comparison of protrusion shapes

with the actual machine own-weight supported were examined. This development focused on the clearance at the bottom of the mount rubber insertion part, as shown in Fig. 13. In the state with the actual machine own-weight supported, the clearance setting affects the deformation state of the mount rubber, and the spring constants and handle vibration value change. Therefore, the clearance setting of the insertion part also becomes a key design element in addition to the mount rubber shape. This development conducted a parameter study of the clearance by CAE enforced displacement analysis and actual measurement of handle vibration, taking into account the displacement due to the actual machine own-weight. This CAE enforced displacement analysis modeled the two parts of a simple box frame shape simulating the mount rubber insertion part and the mount rubber as in Fig. 8. In this model, the reaction force value during enforced displacement of the hole-shaped part in the center of the mount rubber was output in two analysis steps as follows. In the first analysis step, enforced displacement was produced in the direction that the actual machine own-weight acts. The enforced displacement here referred to the mount rubber deformation state when mounted on an actual

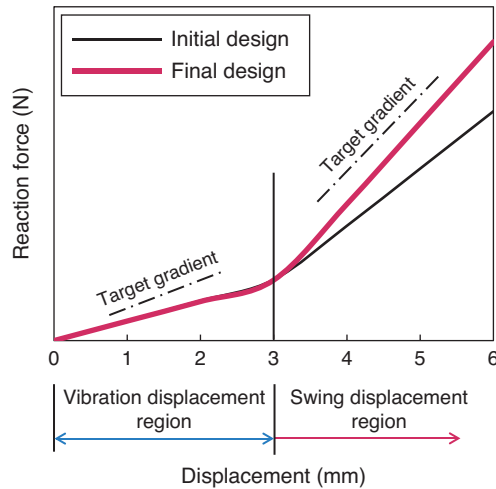


Fig. 12 Nonlinear spring constant characteristics

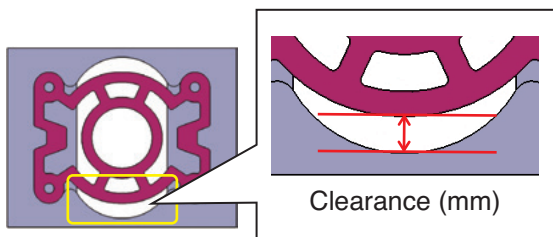
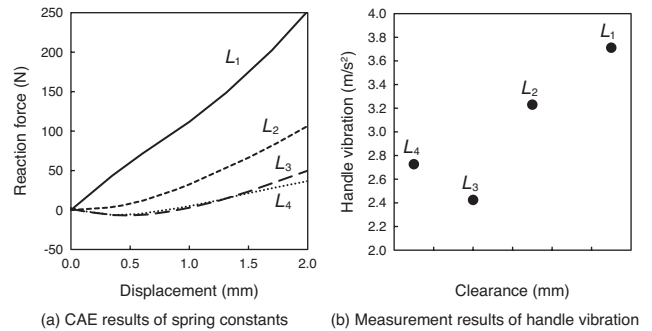


Fig. 13 Clearance at bottom of mount rubber

brush cutter, and was set to produce the same deformation state. In the next analysis step, enforced displacement was produced in the swing displacement direction as in Fig. 8. Figure 14(a) shows the results of enforced displacement analysis of each clearance. The displacement on the horizontal axis indicates the enforced displacement in the swing direction, and the gradient of the graph corresponds to the spring constant. In addition, Fig. 14(b) shows the actual measurement results of the handle vibration value for each clearance. Based on the results in Figs. 14(a) and (b), the clearance was set to the clearance (L_3 in the figure) at which both the spring constant of the vibration displacement region and the handle vibration value are the lowest.

Next, the reasonableness of the shape was confirmed from the viewpoint of durability. First, the mount rubber deformation state was visually confirmed in the condition with the actual machine own-weight supported. The results confirmed that at the set clearance, there was no site with large local elongation or self-contact between inner walls. In addition, durability tests on a test bench and during actual work under the condition of continuous engine operation, and durability tests in various assumed work modes were also conducted. These tests confirmed that there was no damage to the mount rubber.

Through the above examination process, the final mount rubber specification was found (Fig. 15).



Note: (a) L_1, L_2, L_3, L_4 are clearance values ($L_1 > L_2 > L_3 > L_4$)
(b) Clearance amount is 1.0 mm on scale

Fig. 14 Spring constants and handle vibration depending on clearance values

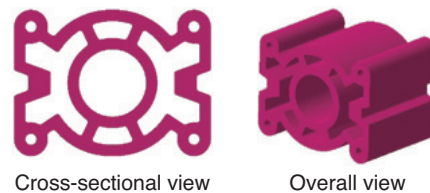


Fig. 15 Mount rubber of UMK450E XEET

3.2. Driveshaft Bearing Placement Technology

3.2.1. Basic drivetrain component structure

Figure 16 shows the basic drivetrain component structure. The rotational force of the engine is transmitted to the cutting head through the centrifugal clutch, driveshaft, and reduction gear. The driveshaft is supported by multiple rubber bearings positioned inside the main pipe. In addition to rubber bearings, resin liner materials are also sometimes used as bearing parts for typical brush cutters. The developed anti-vibration type brush cutter UMK450E XEET uses a support structure with multiple rubber bearings (hereinafter, bearings).

3.2.2. Dynamic damper concept

A new bearing placement design method was established to suppress resonance of the main pipe relative to the increased excitation force due to the unbalance amounts of the cutting head and engine flywheel. Although the number of bearing placements differs for each brush cutter, the bearings are generally placed at equal (or nearly equal) spans, as shown in Fig. 16⁽⁶⁾. By contrast, this development set a bearing placement with unequal spans and used the driveshaft as a dynamic damper^{(5), (7)} (Fig. 17). As shown in the figure, an area (the wide span area in the figure) with an extremely wide placement span between two adjacent bearings is provided. The first bending mode vibration of the driveshaft portion supported at both ends in this area is used as a dynamic damper. The natural frequency of the first bending mode vibration of the driveshaft can be set to any frequency by adjusting the placement span of the two bearings at both ends of the wide span area (the distance between #5 and #6 in the figure). This enables the generation of anti-resonance in the main pipe (mount holder attachment position) for any frequency. The transmission rate of resonance vibration to the handle is suppressed by generating anti-resonance at the mount holder attachment position.

As an applied design technique, the remaining bearing placement spans (#1 to #4 in the figure) can also be changed to add dynamic damper areas. This enables simultaneous suppression of resonance for multiple excitation frequencies.

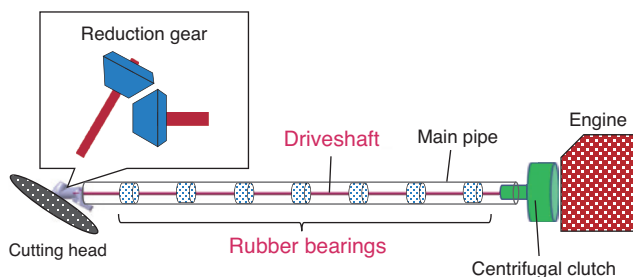


Fig. 16 Drivetrain component structure

This design method only changes the bearing placement, and does not call for installation of additional devices or changes to part shapes (e.g., changing the main pipe diameter, etc.). This enables implementation of resonance measures without impacting the total weight or swing stiffness, which are important from the viewpoint of merchantability.

3.2.3. Bearing placement of UMK450E XEET

This development set the bearing placement shown in Fig. 18. Dynamic damper areas were provided in two locations to simultaneously suppress resonance relative to the excitation force of both the cutting head and the engine, which have different frequency components due to the reduction gear. The target frequencies for resonance suppression were the respective frequencies of the first-order components of cutting head and engine rotation corresponding to racing speed. The natural frequencies of the two dynamic dampers were set by the respective bearing placement spans (between #2 and #3 and between #5 and #6 in the figure) according to these two target frequencies.

3.2.4. Confirmation of effects

The handle vibration for the bearing placement of the dynamic damper concept (Fig. 18) and the equal-span bearing placement (Fig. 16) were compared by measurement during actual operation. The handle vibration measurement method complied with ISO22867-2011⁽⁴⁾. The vibration acceleration of the left handle was measured when the engine speed was varied from idling to wide open throttle. Figure 19⁽⁷⁾ shows the results. Figure 19(a) and (b) show the respective handle vibration tracking data for the first-order components of cutting head and engine vibration. Note that the rotational speed ratio of the reduction gear between the cutting head and the engine is

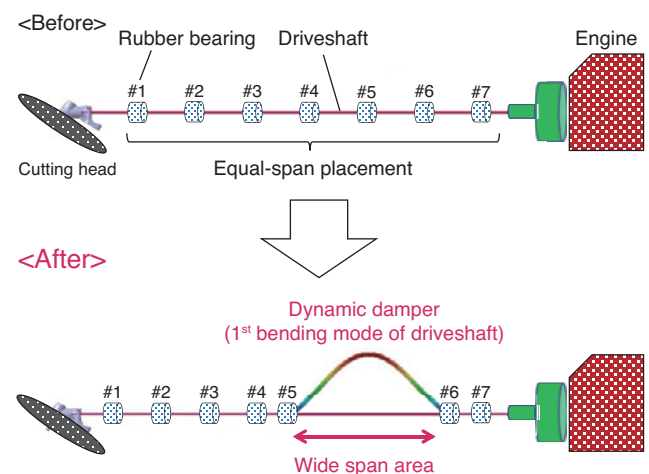


Fig. 17 Dynamic damper outline

0.77:1. As shown in the figure, the vibration levels at the target frequencies of the first-order components of both cutting head and engine vibration were suppressed by 72% and 58%, respectively, compared with the equal-span bearing placement.

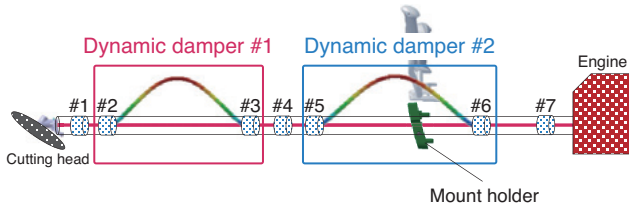
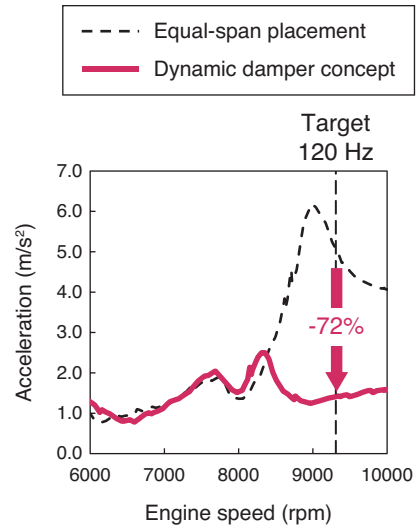
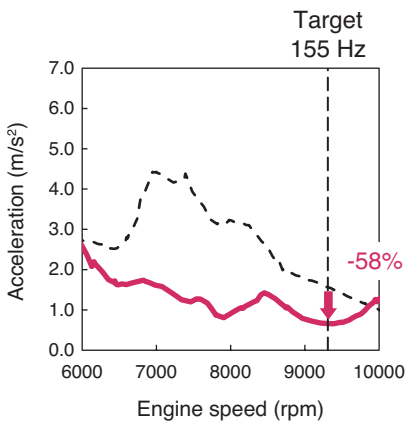


Fig. 18 Bearing placement of UMK450E XEET



(a) Vibration of first-order component of cutting head



(b) Vibration of first-order component of engine

Fig. 19 Measurement results of handle vibration

4. Confirmation of Degree of Achievement

The degree of achievement of the development targets (1) and (2) described in Section 2.2. is shown below.

4.1. Handle Vibration

Figure 20 shows the measurement results of handle vibration⁽⁸⁾. The handle vibration value on the vertical axis is the composite value of the handle vibration values at idling speed and racing speed as described in Section 2.1. A steel blade and a nylon blade, which are typical cutting heads, were used for measurement. Steel and nylon cutting heads with average unbalance amounts were mounted and the handle vibration was measured. Figure 20 confirmed that the target handle vibration value of less than 2.5 m/s^2 was achieved for both the left and right handles.

In addition, Fig. 21 shows the measurement results of handle vibration when the engine speed is varied. The vibration level was confirmed to be less than 2.5 m/s^2 over the entire engine speed range, and there were no resonance peaks that exceeded 2.5 m/s^2 .

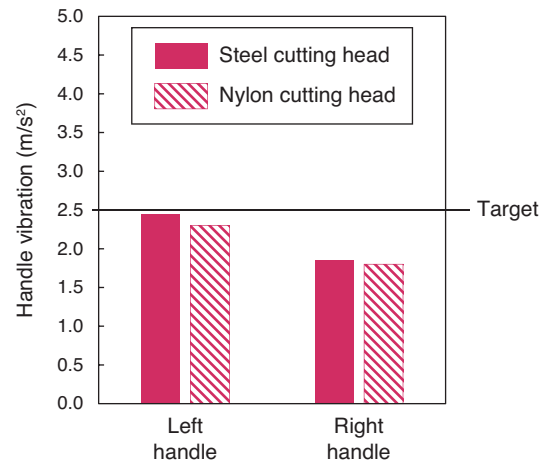


Fig. 20 Measurement results of handle vibration

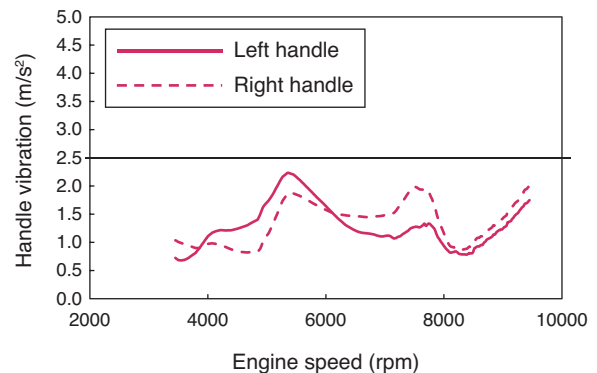


Fig. 21 Measurement result of handle vibration

4.2. Uncertainty of Handle Vibration Value

Figure 22 shows the measurement results of handle vibration when the respective static unbalance amounts of the cutting head and engine flywheel are changed in the bearing placement of the dynamic damper concept (Fig. 18). For reference, Fig. 22(a) also shows the measurement results for the equal-span bearing placement (Fig. 16). The handle vibration value on the vertical axis is the composite value of the handle vibration values at idling speed and racing speed. Figure 22(a) and (b) confirmed that the bearing placement of the dynamic damper concept suppresses the increase in the handle vibration value relative to increases in the respective static unbalance amounts of the cutting head and the engine flywheel. In addition, Fig. 22(a) confirmed that the change in bearing placement contributes to reducing the variation of the handle vibration value relative to the increase in the static unbalance amount.

Figure 23 shows the measurement results of the handle vibration value uncertainty⁽⁸⁾. The handle vibration is the

composite value of the handle vibration values at idling speed and racing speed. The uncertainty conforms to the contents described in Sections 2.1. and 2.2. This figure confirmed that the target uncertainty of less than 2.0 m/s^2 is achieved for both the left and right handles when equipped with either a steel or nylon cutting head.

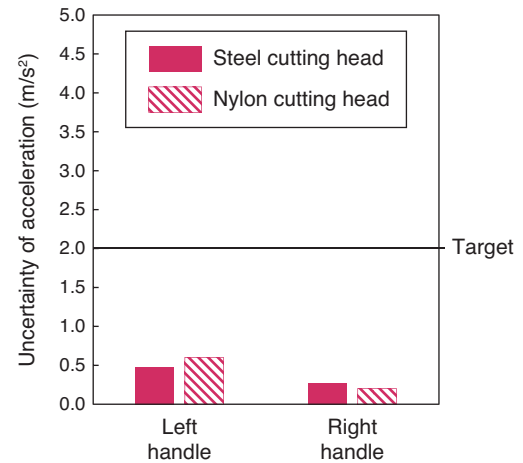


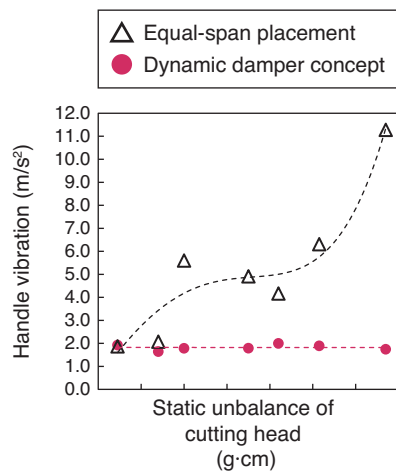
Fig. 23 Measuring uncertainty of handle vibration

5. Conclusion

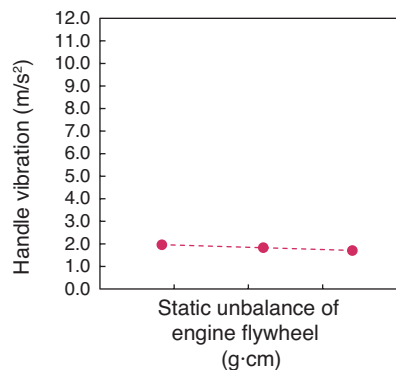
Vibration reduction technology focusing on the mount rubber shape and the driveshaft bearing placement was established to satisfy user demands in the European commercial market. The key points are described below.

- (1) A mount rubber shape with nonlinear spring constant characteristics relative to swing direction displacement was designed in order to both secure swing stiffness and reduce handle vibration. Effective use was made of a CAE parameter study in the design process to optimize the mount rubber rib shape and insertion part clearance. This achieved a handle vibration of 2.5 m/s^2 or less while supporting the actual machine own-weight.
- (2) A bearing placement design method was established that can suppress resonance at any frequency by using the driveshaft as a dynamic damper. This reduced the first-order component of cutting head vibration by 72% and the first-order component of engine vibration by 58% at the respective target frequencies compared to the equal-span bearing placement. It also suppressed the increase in handle vibration relative to increases in the respective static unbalance amounts of the cutting head and the engine flywheel, and achieved the target value for uncertainty of the handle vibration value of less than 2.0 m/s^2 .

These vibration reduction technologies realized a low vibration brush cutter that can satisfy the demands of target users in the European commercial market.



(a) Left handle vibration to static unbalance of nylon cutting head



(b) Left handle vibration to static unbalance of engine flywheel

Note: (a) Static unbalance amount is 2.0 g·cm on scale
(b) Static unbalance amount is 0.3 g·cm on scale

Fig. 22 Measurement results of handle vibration to static unbalance amount

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