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Performance improvement of the vibration reduction method by frequency-based sub-structuring and a neutralizer at vibration source device

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ABSTRACT

Recently, machine structures consist of many subsystems. Vibration of the subsystems often propagates to a whole structure. As a result, unpleasant mechanical noise and vibration occur at the whole structure. And even if countermeasure at the whole structure is the most effective solution, the manufacturer of the subsystem will be required countermeasure at the subsystem. Normally, the subsystems are far smaller than the whole structure. Therefore, compliance of a subsystem is dominated by a rigid body mode at the interested frequency. In a case like this, it can be defined that, few degrees of freedom of transmitted forces dominate the resonance of the whole structure. In this study, a design method in the case that a single degree of freedom transmitted force is dominant is organized. A countermeasure method for vibration of a whole structure using FRF based sub-structuring and a Neutralizer at a single target frequency, with a restriction that only subsystems can be structurally modified, is proposed. As a result, vibration transmission and wave propagation from a subsystem to a whole structure at a target frequency are theoretically zero. And to improve the performance of this method, the position and the size of a Neutralizer on an active subsystem is studied.

Keywords: Frequency Response Function, Modal Analysis, Wave Analysis, FRF-Based Sub-Structuring, Neutralizer

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1. INTRODUCTION

In recent years, manufacturers of a whole structure and manufacturers of a plurality of substructures (hereinafter referred to as subsystems) are often different. The whole structure is complicated, has a large number of parts, and is composed of a plurality of subsystems. Each subsystem has individual functions, and these functions are linked to each other to form the function of the whole structure.

When a subsystem supplies its function to a whole structure, mechanical behavior of the subsystem generates an exciting force, and consequently generates waves, which propagate through housings or internal structures of the whole structure. This phenomenon results in structure-borne noise, which may reach user's ears and causes noise problems. Technically considering countermeasures against this problem, it seems effective to take countermeasures at the coupling DOFs (Degrees of Freedom) of the whole structure, which affects the magnitude of the excitation point impedance, and also to take countermeasures against panel members which actually produces acoustic radiation at the whole structure. However, considering taking technical measures from the situation of a manufacturer of a subsystem, it is practically impossible. Even if it is clear that a coutermeasure for some location other than the subsystem is more effective.

We arrange the problem from a technical viewpoint. Whole structures have high mode density, and its vibration characteristics differ for each one due to its low robustness. And the subsystem of interest is often rigidly coupled to a whole structure. Under these conditions, a manufacturer of a subsystem is required to take effective measures against a whole structure only by structural modification of the subsystem.

In this research, we examine the design method which can always reduce the vibration of a whole structure at the target frequency by applying measures only for the subsystem of interest, not for a whole structure. For example, in the case that the vibration of a motor propagates to a car body, panel members of the car excited by the propagated motor vibration greatly at the natural frequency, and cause large acoustic radiation, by reduction of vibration at the narrow band around the natural frequency. Then, the design method only for narrow band frequency range is effective. In the following, the subsystem of interest, which is modifiable by a manufacturer of subsystems and has exciting force, is defined as the active part, and the other parts are difined the passive part.

In realizing vibration reduction in a narrow band, we introduce the constraint that the active part has low damping and the passive part has comparatively high damping. In the case of a car, for example, motors and pumps have lower damping than the body of a car. Therefore, the precondition for applying this method is considered to be reasonable from the viewpoint of the manufacturer of the subsystem. Then, under the condition that these two systems are rigidly coupled, we lead theoretically the design requirement for making it possible to reduce the vibration of the whole structure by taking measures only for the active part. For this derivation, we apply a design method using the FRF based sub-structuring (Seijs et al., 2016) from the viewpoint similar to the design method proposed by Kido et al. (Kido · Sueoka, 2005), (Kido et al., 2006) .

In the design method proposed by Kido et al., it is necessary to consider a small number of the transmitted forces at coupling DOFs. Therefore, in this paper, the design method will be developed when it is assumed that the number of transmitted forces is one. As indicated by Kido et al. (Kido · Sueoka, 2005), even if the coupling DOFs consist of multiple DOFs, the dominant DOF of the transmitted forces involved in the resonance of a whole structure can be regarded as a small number of DOFs. And by applying reconstruction of DOFs(Kido et al., 2006), the dominant DOFs in the transmitted forces involved in the resonance of a whole structure can be approximated as one DOF or two DOFs. In fact, even in the actual machine vibration of a low- frequency range, dominant vibration transmission is carried out by the propagation of a torsional wave if the vibration source of the active part is a torsional type of motion, or by the propagation of a longitudinal wave if it is a linear type of motion.

In order to reduce the vibration of a whole structure based on the method of Kido et al., it is effective to reduce the dominant transmitted forces at the coupling DOFs of the active part and the passive part. Therefore, in this paper, we propose a method to set the vibration transmission from the excitation DOFs in the active part to the coupling DOFs, to zero. Since the vibration transmission is caused by the wave propagation, a Neutralizer is provided on the transmission path between the excitation DOFs and the evaluating DOFs in the active part, and it is set as a virtual fixed end for perfect reflection of the wave . As a result, the transmitted forces between the excitation DOFs and the coupling DOFs can be made zero. Khatib et al. (2005) proposes methods to form virtual fixed end for bending waves, by setting up a Neutralizer in the beam structure where wave propagation occurs in two wave modes called bending wave and bending near-field wave in the elementary theory of wave, and to control wave propagation. However, this prior research is not enough for the target of this paper in the following two points. First, it is organized from the viewpoint of control of wave propagation. And it is not organized from the viewpoint that the active part and passive part are combined, and the transmission force generated at coupling DOFs set to be 0. Secondly, although the dominant transmitted force in case that the bending wave passes through the coupling DOFs is internal force of 2 DOFs corresponding to deflection and deflection angle. As described in the preceding paragraph, in this paper, we aim to set the dominant single DOFs transmitted force generated by the longitudinal wave and the twisting wave in the elementary theory of the wave, to zero. It is necessary to define that installation of a Neutralizer is effective for realizing a virtual fixed end for longitudinal waves and twisted waves.

Since a Neutralizer is also called a dynamic vibration absorber and it is also used to suppress the vibration of the continious body, we would like to describ the difference between the usage as a general dynamic vibration absorber and the usage within the design method in this paper. In recent design methods related to vibration damping using dynamic vibration absorbers, as seen in Sugimoto et al. (Sugimoto et al., 1991) and Yamaguchi (Yamaguchi, 1990), many are new forms of dynamic vibration absorbers, and adjustment of rigidity and damping. Among them, in a previous study of a dynamic vibration absorber of the type added to a continuous body like this paper, Seto (Seto, 1984) used multiple dynamic vibration absorbers from the viewpoint of passive mode control in a continuous body, to simultaneously suppress multiple vibration modes of the whole structure. In addition, Matsuoka et al. (Matsuoka et al., 2012) proposed a method of controlling the vibration mode of a beam structure by a moment type dynamic vibration reducer. However, these viewpoints are different from the viewpoints of this paper in the following two points. Firstly, in this paper we propose a method that enables reduction of response in arbitrary narrow frequency range, and not limited to the natural frequency of the structure to be countermeasured. On the other hand, other papers specialize to reduce the vibration of the resonance peak. Secondly, in this paper, however the manufacturer of the active part can not get to know vibration characteristics of the passive part, the effect of a Neutralizer must be proved in case that a Neutralizer is installed only on subsystem of interest. Therefore, it can be pointed out that it is quite different from the design method using the information of the whole structure like Seto and Matsuoka et al.

2. FRF BASED SUB-STRUCTURING

In this paper, we focus on the analytical method proposed by Kido et al. (Kido · Sueoka, 2005) in order to analyze the vibrational coupling relationship between the active part and the passive part. Kido et al. propose a method to analyze the vibrational coupling relationship of two subsystems with the *kernel* dynamic stiffness matrix as an index based on the FRF based sub-structuring method. In this research, its *kernel* dynamic stiffness is used as a criterion for generation of total system resonance.

2.1 FRF based sub-structuring method used for the analysis of interest

The relationship between the active part and the passive part will be described with reference to Fig. 1. The excitation DOFs and the evaluation DOFs in the whole structure are indicated by node 1 and 4 on the left of Fig. 1 respectively. The free body diagram obtained by dividing the whole structure into two subsystems is shown on the right side of Fig. 1. The excitation DOFs is on the active part side, and the evaluation DOFs is on the passive part side. The coupling DOFs between the active part side and the passive part side is node 2, 3 respectively. The basic formula of the FRF based sub-structuring method in this subdivision is as follows:

$$\mathbf{X}_4^{\mathrm{AB}}(\omega) = \mathbf{H}_{43}^{\mathrm{B}}(\omega)\left(\mathbf{H}_{22}^{\mathrm{A}}(\omega) + \mathbf{H}_{33}^{\mathrm{B}}(\omega)\right)^{-1}\mathbf{H}_{21}^{\mathrm{A}}(\omega)\mathbf{F}_1(\omega)\,, \tag{1}$$

where $\mathbf{X}_4^{\mathrm{AB}}(\omega)$ is the response displacement of the evaluation DOF 4 in the passive part B, when the force $\mathbf{F}_1$ is added to the excitation DOF 1 of the whole structure in the frequency domain. $\mathbf{H}$ is a compliance matrix, the number on the left side of the subscript on the lower right is the response side, and the number on the right side shows the number of degrees of freedom on the excitation side. Also, the subscripts A and B on the right shoulder indicate that they are related to the active part A and the passive part B, and in the case of AB, they are quantities related to the whole structure. Also, $\mathbf{g}_2^{\mathrm{A}}$ and $\mathbf{g}_3^{\mathrm{B}}$ are internal forces in the coupling DOFs, which corresponds to the amount called transmitted force in the field of vibration analysis

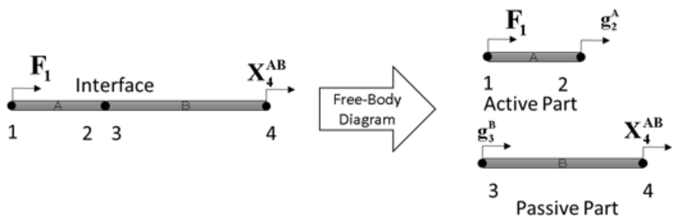


Fig. 1 Free body diagram of an active part and a passive part.

2.2 Reduction of analysis DOFs focusing on dominant transmitted power

The relationship between the transfer function of the active part and the passive part is shown in Fig. 2. In this paper, the active part is very small compared with the passive part, and the active part is a rigid structure compared with the passive part. Therefore, in the low frequency band where vibration and noise of the whole structure become problematic, the compliance of the active part is dominated by the rigid body mode and there is no elastic vibration mode. In such a case, as shown by Kido et al. (Kido · Sueoka, 2005), even if the joint part has multiple coupling DOFs, since the joint part behaves like a rigid body, DOFs of the dominant transmitted forces, which participates in the resonance of the whole structure, becomes small. As mentioned in the introduction, in this paper, in particular, a design method will be developed in the case that the transmitted force of single DOFs is dominant.

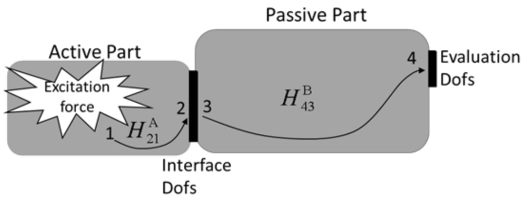


Fig. 2 Frequency response function from excitation force which is added to node1, to evaluation dofs at node 4.

2.3 Conditions for occurrence of resonances of a whole structure

Following the analysis method proposed by Kido et al. (Kido · Sueoka, 2005), the occurrence condition of the resonance is described as the relation between the transfer function of the active part and the passive part. As stipulated in the previous section, if it is assumed that single DOF is dominant in the transmitted force, the number of excitation DOF and evaluation DOF are both one. Equation 1 can be represented by a scalar, and is expressed as,

$$X_4^{AB} = H_{43}^{B}\left(H_{22}^{A} + H_{33}^{B}\right)^{-1} H_{21}^{A} F_1 . \tag{2}$$

From this equation, the resonance condition of the whole structure becomes as follows,

$$H_{22}^{A} + H_{33}^{B} = 0 . \tag{3}$$

Even when both the number of degrees of exciting DOFs and the number of degrees of evaluation DOFs are other than 1, this formula is always established when the transmitted force of single DOFS is dominant. At frequencies where the condition of Eq. 3 holds, the inverse number term of Eq. 2 becomes infinite, and the whole structure resonates.

3. FORMATION OF A VIRTUAL FIXED END ON AN ACTIVE PART

In this chapter, we show how to design wave reflectors required for the design method proposed in the next chapter. This reflector is used to make the wave generated at the vibration source in the active part completely reflected by the transmission path without reaching the coupling DOFs. In order to realize complete reflection, it is desirable to provide a virtual fixed end. Therefore, we focused on a Neutralizer (Khatib et al., 2005) as a way to realize passive complete reflection without using active control. Khatib et al. indicated a method of designing a reflector for bending waves, but the bending wave is a vibration transmission with two DOFs, which is out of the object of this paper. In this paper, we propose a method of designing a Neutralizer for longitudinal waves, because it is considered that the transmission with longitudinal wave and twisting wave is dominant in vibration transmission of single DOFs in this paper. Since the longitudinal wave and the twisting wave in the elementary theory are represented by a homogeneous differential equation, the design method for the longitudinal wave can be applied to the twisted wave almost as it is.

We derive the reflection and transmission characteristics of longitudinal wave when a Neutralizer is regarded as discontinuity for wave propagation. Figure 3 shows the configuration of a Neutralizer for longitudinal waves. A Neutralizer is single DOFs vibration system, consists of a point mass and a spring, and its one end is rigidly connected to the active part. The reflection and transmission characteristics can be obtained by solving the displacement sequence and the force balance in the section of a Neutralizer (x=0) by using the displacement and the relationship between force and wave amplitude. Along with the movement of a Neutralizer, the force that the spring gives to the active part can be expressed by Eq. 4 (Yasuda, 2012) as with a general dynamic vibration absorber,

$$F_N = \frac{\kappa m\omega^2}{\kappa - m\omega^2} U e^{j\omega t} . \tag{4}$$

Also, the displacement and the internal force at the negative face at the position x are denoted by $U_\mathrm{L}(x)$,$F_\mathrm{L}(x)$, the values at the positive face are denoted by $U_\mathrm{R}(x)$, $F_\mathrm{R}(x)$ and these are expressed using wave amplitude (Khatib et al., 2005) and the following equations,

$$U_\mathrm{L}(x) = a_\mathrm{L} e^{-jkx} + b_\mathrm{L} e^{jkx} , \tag{5}$$

$$U_\mathrm{R}(x) = a_\mathrm{R} e^{-jkx} + b_\mathrm{R} e^{jkx} , \tag{6}$$

$$F_\mathrm{L}(x) = ES(-jka_\mathrm{L} e^{-jkx} + jkb_\mathrm{L} e^{jkx}) , \tag{7}$$

$$F_\mathrm{R}(x) = ES(-jka_\mathrm{R} e^{-jkx} + jkb_\mathrm{R} e^{jkx}) , \tag{8}$$

where the wave number of the medium at the position x is k, Young's modulus is E, and cross-sectional area is S. Also, amplitudes of the forward and backward waves on the negative side of position x are referred to as a_L and b_L, and the amplitudes on the positive side are denoted as a_R and b_R. At this time, continuity of displacement and the balance of force at the Neutralizer attachment section (position x = 0) can be expressed as follows:

$$U_\mathrm{L}(0) = U_\mathrm{R}(0) , \tag{9}$$

$$F_\mathrm{R}(0) + F_\mathrm{N} - F_\mathrm{L}(0) = 0 . \tag{10}$$

By substituting Eq. 4 - 8 into these Eq. 9 and 10, we arrange the relation of the wave amplitude to the form of the scattering matrix of the equation below,

$$\begin{Bmatrix} b_\mathrm{L} \\ a_\mathrm{R} \end{Bmatrix} = \begin{bmatrix} r_\mathrm{LR} & t_\mathrm{RL} \\ t_\mathrm{LR} & r_\mathrm{RL} \end{bmatrix} \begin{Bmatrix} a_\mathrm{L} \\ b_\mathrm{R} \end{Bmatrix} . \tag{11}$$

Here, the reflection coefficient of the wave incident on the negative surface at the position x is defined as the transmission coefficient, and the reflection coefficient of the wave incident on the positive surface is defined as the transmission coefficient. As a result, the reflection/transmission coefficient of the longitudinal wave on the Neutralizer attachment section is derived as follows:

$$r_{\mathrm{LR}} = r_{\mathrm{RL}} = \frac{\kappa\Omega^2}{2jkES(1-\Omega^2)-\kappa\Omega^2}, \tag{12}$$

$$t_{\mathrm{LR}} = t_{\mathrm{RL}} = \frac{2jkES(1-\Omega^2)}{2jkES(1-\Omega^2)-\kappa\Omega^2}, \tag{13}$$

where the ratio of the natural angular frequency $\omega_N = \sqrt{\kappa/m}$ when Neutralizer alone is fixed to the ground and the angular frequency of the harmonic excitation force is set as $\Omega = \omega/\omega_N$.

In the case where the transmission coefficient of Eq. 13 is zero, the longitudinal wave is totally reflected. In this condition, $\omega = \omega_{\mathrm{N}}$ is completed, and it is understood that this is the case where the natural frequency of a Neutralizer in the situation that a Neutralizer alone is fixed to the ground, and exciting frequency of harmonic excitation are the same. In this case, from the Eq. 12, the reflection coefficient is $r_{\mathrm{LR}} = r_{\mathrm{RL}} = -1$. This means the virtual fixed end realizes, that is, equal to the fixed end for the propagation wave.

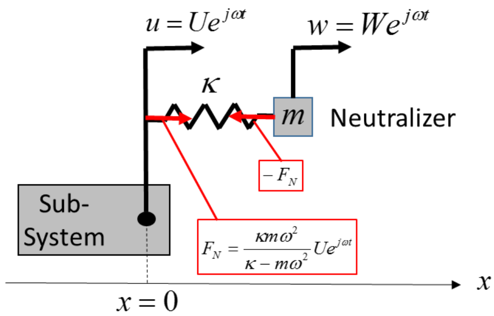


Fig. 3 Neutralizer is considered as a discontinuity of wave propagation. Reflection and transmission coefficients can be calculated by solving the equillibrium at the position of a neutralizer, using the relationship between displacement, force, and wave amplitude.

4. POPOSED DESIGN METHOD

The following design method was developed on the premise that the passive part has relatively large damping. The passive part has a larger structure than the active part in order to perform some mechanical function and generally has large damping, so it can be considered applicable to many cases. In addition, as described in section 2.2, assuming that the dominant transmitted force in the coupling DOFs is single, the Neutralizer design method derived in the previous chapter is invoked

4.1 Controllable parameters for vibration reduction

Parameters to be controlled in the proposed vibration reduction method will be described. In the Eq. 2, H_{33}^{B} and H_{43}^{B} are compliance of the passive part, so it is impossible to make a change in the position of the manufacturer of the active part. Only the transfer function H_{22}^{A} and H_{21}^{A} related to the active part can apply the change.

4.2 Control of magnitude of transfer function using Neutralizer

By installing a Neutralizer, as a manufacturer of the active part, we show a concrete method to control transfer function. First, consider the position suitable for installation of a Neutralizer. In actual machines, the exact location of the vibration source of the active part can often not be specified. However, as shown in Fig. 2, it is common that it lies somewhere on the active part side to the left of the coupling DOFs. Therefore, a Neutralizer will be installed near the coupling DOFs.

Consider the magnitude of the transfer function H_{21}^{A} and H_{22}^{A}, when a Neutralizer is installed in the vicinity of the coupling DOFs 2 within the possible range between the exciting DOFs 1 and the coupling DOFs 2. First, if a device that perfectly reflects wave of specific frequency is installed between the exciting DOFs 1 in the active part, and the coupling DOFs 2, the wave generated by excitation cannot reach to the coupling DOFs 2. Therefore, the magnitude of the transfer function becomes zero at this frequency,

$$\left|H_{21}^{\mathrm{A}}(\omega_t)\right| = 0\,. \tag{14}$$

Also, the magnitude of H_{22}^{A} at this frequency ω_t becomes extremely small,

$$\left|H_{22}^{\mathrm{A}}(\omega_t)\right| \approx 0\,. \tag{15}$$

As shown in Fig. 4, as a result of the formation of the virtual fixed end of the wave by a Neutralizer in the vicinity of the coupling DOFs, after the wave generated by exciting the coupling DOFs 2 propagates to a Neutralizer, the incident wave and the reflected wave, which is antiphase, cancel each other. At this time, Eq. 15 holds.

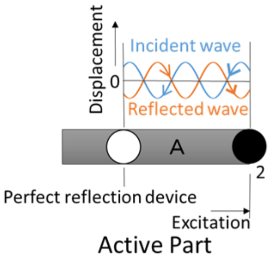


Fig. 4 Condition of incident wave and reflected wave in case that node 2 of the active part is excited in longitudinal direction, and a perfect reflection device for the longitudinal wave is set very closely to node 2. As the wave reflects at the device and its phase turns over at the device, the displacement at the node 2 is very small.

4.3 Design procedure when applying the proposed method to actual structures

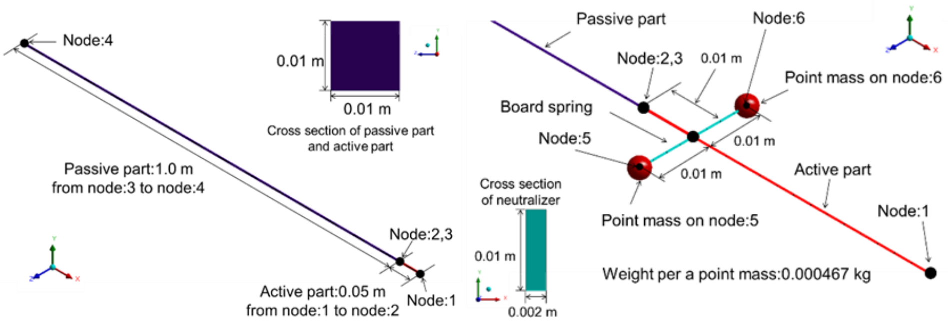
The design procedure for installing a Neutralizer on the actual structure is indicated in the itemization below.

1. In the noise and vibration problem, confirm that the exciting DOFs is in the active part, and noise and vibration of passive part, occurred by propagated vibration from the active part, is recognized as a problem.
2. In the noise and vibration problem, determine the frequency that is most desired to be reduced.
3. Confirm that the dominant vibration transmission DOF is 1, in the frequency range decided in 2. Transfer path analysis (Seijs et al., 2016) is applicable.
4. Confirm that the active part has a lower damping than the passive part.
5. Design a Neutralizer that has a natural frequency of the specified frequency at 2 and vibrates in the direction of specified transmission freedom at 3.
6. Attach the Neutralizer to the part of the active part as close as possible to the coupling DOFs of the active part and the passive part.

5. NUMERICAL VERIFICATION

Numerically verify the validity of the proposed design method. The object is the longitudinal vibration of the thin rod. Fig. 5 shows the system used to verify the design method. Cross section shape of the thin rod is 10x10 mm, and length of the active part length is 50 mm and passive part length is 1 m. Also, in the active part, a Neutralizer is installed.

The vibration of the system in Fig. 5 was obtained by finite element analysis. In preparing a finite element (hereinafter referred to as FE) model, material of the active part is aluminium, and material of the passive part is mild steel. For damping of the passive part, we give a proportional viscous damping of $\alpha = 3.0 \times 10^{-8}$ and $\beta = 3.0 \times 10^{-6}$. The excitation degree of freedom was taken as the positive direction of *x* at the position marked as Node 1 in Fig. 5. In addition, the evaluation DOF is set to the positive direction of *x* at the position denoted as Node 4 in Fig. 5. In addition, a Neutralizer is installed at the point moved from Node 2 of the active part to Node 1 by 0.01 m. Two leaf springs deforming in the *x* direction are installed symmetrically across the thin rod, and concentrated masses are connected to the tip of each, and this is regarded as Neutralizer. Both leaf springs are phosphor bronze plates with a thickness of 2 mm and a width of 10 mm. The frequency at which the response is desired to be reduced is set to 2450 Hz and the concentrated mass at the tip of the two leaf springs is set at 0.467 g so that the natural frequency of the Neutralizer alone is 2450 Hz. The mass of Neutralizer is 2.23 g each, and it has no damping. The boundary condition of this model is free to the surroundings.



(a)Whole structure (b)Enlarged view

Fig. 5 Beam elements for FE validation with a neutralizer. Node 2 at the end of an active part and node 3 at the end of a passive part are rigidly connected. Two neutralizers, which consist of point masses and board springs, are set at a node, which locates at 0.01m in the direction of node 1 from node 2.

We discuss the calculation result of this analysis target. Fig. 6 shows the response displacement X_4^{AB} in *x* direction in node 4 when node 1 is excited in *x* direction, in which the characteristics in the case that a Neutralizer is not equipped in a red solid line, and the case that a Neutralizer is equipped in a blue solid line. By attaching a Neutralizer, it can be seen that the response displacement X_4^{AB} is greatly reduced at the response reduction target frequency of 2450 Hz.

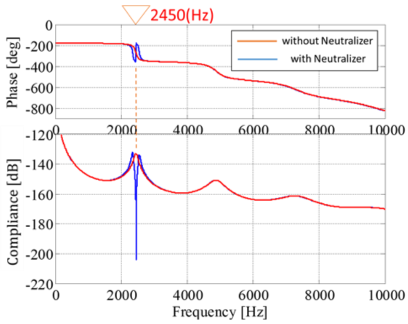
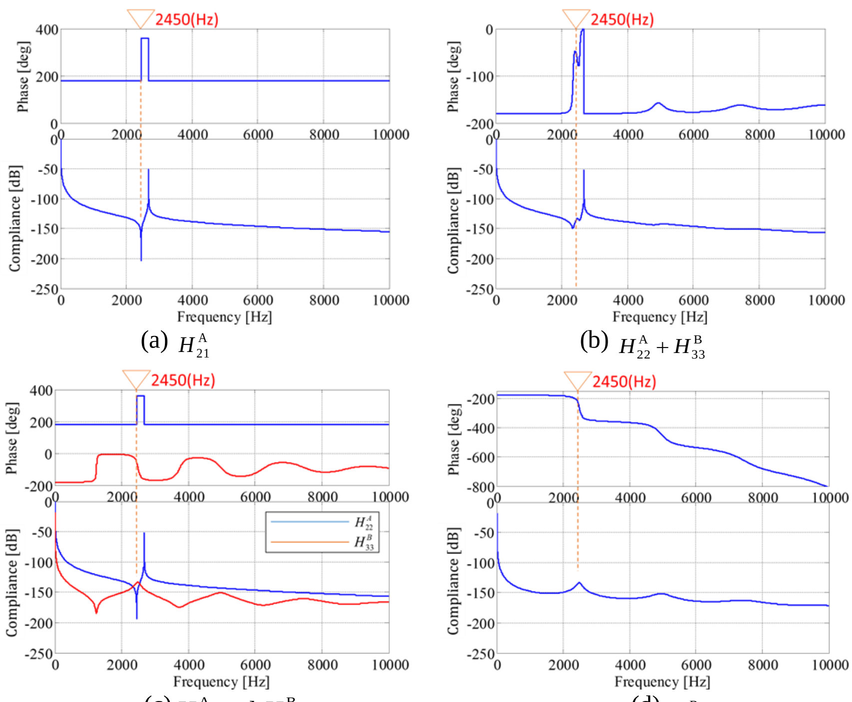


Fig. 6 Displacement for the case with and without a neutralizer. By attaching a neutralizer, compliance is largely reduced at 2450 Hz. Theoretically, displacement is 0 at the frequency, but for convenience of frequency resolution, it is nearly 0.

Factors of evaluation point response X_4^{AB} reduction are discussed. Figure 7 shows $H_{21}^{A}, H_{22}^{A}+H_{33}^{B}$, H_{22}^{A} & H_{33}^{B}(overwriting) in a state in which a Neutralizer is attached. H_{21}^{A} Shows 0 at the target frequency of 2450 Hz of a Neutralizer as shown in Fig. 7(a). This is because a Neutralizer which is installed between node 1 and node 2 blocks wave propagation. Figure 7(b) shows *kernel* compliance, which is obviously zero at the target frequency of a Neutralizer 2450 Hz, and Eq. 3 which is the resonance condition of the whole structure is not satisfied. This is because, as seen in Fig. 7(c), the condition of Eq. 16 is satisfied. Then, as shown in Fig. 7(d), due to damping of the passive part, it does not become infinite at the same frequency and does not resonate. From these facts and the Eq. 2, it is understood that the Eq. 18 $X_4^{AB}(\omega_t)=0$ is realized due to the Eq. 14 $\left|H_{21}^{A}(\omega_t)\right|=0$.



(a) H_{21}^{A} (b) $H_{22}^{A}+H_{33}^{B}$ (c) H_{22}^{A} and H_{33}^{B} (d) H_{43}^{B}

Fig.7 Compliance in case that a neutralizer is attached on an active part. (a) equals nearly 0 by convenience of frequency resolution. (b) kernel compliance is not 0 at 2450 Hz. (c) is smaller than , and accordingly it is proved that Eq.(17) is satisfied. (d) is not diffused. Therefore, response is directly proportional to , and consequently vibration reduction is achieved.

6. PERFORMANCE IMPROVEMENT STUDY

For performance improvement study of the Neutralizer, a parameter study was carried out. According to Eq. 4, if both m (mode equivalent mass) and κ(mode equivalent stiffness) increases, reaction force by the Neutralizer F_N will increase. For that purpose, we increased m and κ, with keeping equivalent $\omega_n = \sqrt{\kappa/m}$.

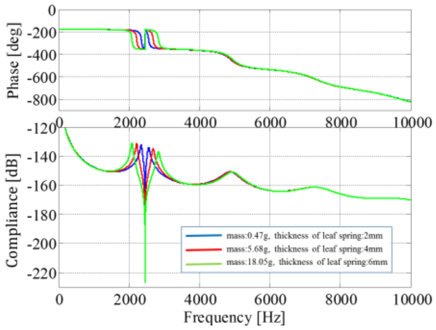


Fig.8 Displacement with Neutralizers, for the case that combination of three masses and rigidity adjusted so that the primary natural frequency is the same. An effective frequency zone tends to increase with mass weight and stiffness of the Neutralizer.

According to Fig. 8, effective frequency zone tends to increase with mass weight and stiffness of the Neutralizer. This tendency can be understood from Eq. 4, that mode equivalent stiffness and mode equivalent mass increase with constant ratio, reaction force by the Neutralizer increases. But currently, actual vibration reduction amount is not predictable. This will be clear in future study.

7. CONCLUSION

The purpose of countermeasures against the problem that the exciting force in the active part propagates to the passive part with large damping which is rigidly coupled, the structure borne noise is generated in the passive part, and it is regarded as an unpleasant noise. In almost all cases, it is impossible for manufacturers of active parts to take measures other than the active parts. Therefore, while not being able to know the vibration characteristics of the passive part, the idea and the procedure of the design to reduce the compliance of the whole structure are shown.

(1) In the vibration transmission from the active part to the passive part, when the transmitted force of one DOF is dominant, the active part has almost no damping, and in the condition that the passive part has relatively large damping, it is shown that the response at any DOFs of the passive part can be made zero, by setting the single DOFs compliance from excitation DOFs to coupling DOFs with passive part, to zero.

(2) In order to make the compliance from exciting DOF in the active part to the coupling DOFs with passive part zero, we focused on preventing the propagation of wave. To realize this, we showed that a Neutralizer should be provided in the vicinity of the coupling DOFs on the active part side, and realize a virtual fixed end. As a design guideline for the actual structure, install a Neutralizer on the active part, as close as possible from the coupling DOFs with the passive part, and match the excitation direction in the exciting DOFs with the DOFs of vibration of the Neutralizer.

(3) For designing Neutralizer to prevent propagation of longitudinal waves, calculation formula of reflection and transmission coefficient was derived. In actual products, the vibration transmission from the active part to the passive part is often dominated by the direction of the twisting wave or the longitudinal wave

direction. The method of calculating the reflection transmission coefficient for the derived longitudinal wave can be applied to the twisted wave almost as it is.

8. REFERENCES

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