Proceedings of 20th International Congress on Acoustics, ICA 2010

23-27 August 2010, Sydney, Australia

Passive control of structural intensity for reducing structure-borne sound on compound plate structure

Yuta MURAKAMI (1), Nozomu NUMATA (1) and Toru YAMAZAKI (2)

(1) Graduate School of Kanagawa University, 3-27-1 Rokkakubashi, Kanagawa-ku, Yokohama 221-8686, Japan
(2) Department of Mechanical Engineering, Kanagawa University

PACS: 43.40.Dx

ABSTRACT

This paper presents a new structural design concept where the structural intensity technique is used to reduce the sound radiated from a compound plate structure. This concept is based on the modal expansion of the structural intensity on a plate. Structural intensity in modal form can be expressed by the superposition of weight coefficients and "cross-modal functions", and the weight coefficients depend on the location of the point excitation coordinates. The cross-modal function is determined by the product of two modes with spatial derivatives and is expressed in a vector field. The modal form of structural intensity gives the desired distribution of structural intensity in terms of changing the weight coefficients (excitation conditions) and/or the cross-modal functions (structural design). The cross-modal functions can be classified into two types of power flow: vortex-type and straight-type distributions. In the case of the vortex-type cross-modal function, the power propagated through the plate is zero because the integral of vortex flow is zero. Then, the modal form of the intensity suggests that enhancement of the vortex-type cross-modal function relative to the intensity leads to less power transmission through the plate. Vortex-type intensity on a plate would be useful for interrupting power transmission between plate subsystems of compound plate structures. On the other hand, the straight-type cross-modal functions would be useful for promoting the power transmission. In this study, numerical simulations are carried out to demonstrate the interruption of the power transmission as a result of generating vortex-type structural intensity on the middle plate of a three-plate structure (J-shaped structure).

1. INTRODUCTION

Structural intensity, which was introduced by Noiseux [1] and Pavic [2] in the 1970s, is defined as the instantaneous rate of energy transfer per unit area of structure. The structural intensity field indicates the magnitude and direction of vibrational energy flow at any point in a structure. Structural intensity analysis of a simply supported plate was carried out by Gavric [3] using the finite element method (FEM). The measurement [4] and calculation of the structural intensity distribution provide the power transmitted through any path and the locations of energy sources and sinks.

The structural intensity technique has been used to diagnose defects in structures [5] and to estimate statistical energy analysis (SEA) parameters [6]. Recently, Xu [7] suggested that an optimized design/modification or vibration control of a coupled stiffened plate system could be obtained by using the structural intensity technique. Liu [8] concluded that the structural intensity technique can assist in finding effective control positions for dampers and actuators, as well as in identifying vibro-acoustic interior noise sources and predominant panels that can be modified to reduce the interior noise level. However, these two papers provide no description of a method for designing a structure for which the structural intensity fields are calculated solely by FEM analysis.

We are also attempting to develop a new process for designing a structure with low noise and vibration from the point of view of the structural intensity concept. To achieve this goal, two major tasks must be accomplished: one is developing a structural design method that can provide the desired intensity distribution and the other is defining the relation between the structural intensity distribution, vibration shapes and the sound radiated from structures.

For design methods using structural intensity, we have proposed the modal expansion of structural intensity for flexural vibration on a beam [9]. In the modal expansion, the structural intensity on the beam is expressed by superposition of the product of the weight function and the product of two natural modes with a spatial derivative that is called the "cross-modal function" as well as the vibration response in modal form. Then, the structural intensity field can be controlled by changing the natural modes or excitation conditions. We have also proposed a procedure for selecting an excitation point to realize the desired intensity distribution [10]. This procedure is based on the modal expansion of intensity and is focused in particular on the weight coefficient that is mainly dependent on the location of the point excitation. By using the procedure, the location of a point excitation can be determined to enhance particular cross-modal functions.

To define the use of the structural intensity technique for reducing noise, we focus on vortex-type structural intensity. Nakagawa et al. have studied the relationship between structural intensity and acoustic intensity to reduce structure-borne noise [11]. Notably, a point excitation near the nodal line of the resonant mode generates vortex-type structural intensity. Furthermore, when the distribution of structural intensity on a plate is a vortex, the sound radiation efficiency drastically changes to become either larger or smaller than that of the non-vortex-type structural intensity pattern. We have also reported that the occurrence of vortex-type structural intensity under a point excitation near the nodal line is explained by the modal expansion of structural intensity [12]. Furthermore, we have found that the vortex-type cross-modal function transmits no power through the plate and that vortex-type structural intensity may also interrupt power transmission to neighboring plates [12].

This paper discusses a new structural design concept where the structural intensity technique is used to reduce the sound radiated from a compound plate structure. The concept is based on the power transmission of cross-modal functions with vortex-type and straight-type distributions. The vortex-type cross-modal function without transmitting power is useful for interrupting power propagation to a neighboring plate, while the straight-type cross-modal functions is useful for promoting power transmission. First, the modal form of the structural intensity on a flat plate is summarized. Then, we propose a new structural design concept based on the properties of the two types of cross-modal functions. Finally, numerical simulations are carried out to demonstrate the interruption of power transmission as a result of generating vortex-type structural intensity on the middle plate of a three-plate structure (J-shaped structure).

2. MODAL EXPANSION OF STRUCTURAL INTENSITY FOR FLEXURAL VIBRATION ON A FLAT PLATE

2.1 Vibration displacement in modal form

For flexural vibration of a plate excited at the coordinates (x_F, y_F) by a point force F with angular frequency ω, the spectrum of the flexural displacement ξ at the point (x, y) can be expressed as

$$\zeta = \sum_{n=1}^{N} \alpha_n \phi_n(x, y), \tag{1}$$

where the n-th weight coefficient of displacement is

$$\alpha_n = \frac{F\phi_n(x_F, y_F)}{\omega_n^2(1 + j\eta_n) - \omega^2}. \tag{2}$$

Here, N is the number of retained modes, η_n is n-th modal loss factor and j is a complex unit. ω_n is the n-th natural angular frequency and $\phi_n(x, y)$ is the mass-normalized natural mode shape.

2.2 Definition of structural intensity on a flat plate

The structural intensity for flexural vibration on a uniform flat plate can be expressed in terms of the out-of-plane displacement ζ as follows:

$$I_x = -\frac{\omega D}{2}\mathrm{Im}\left[\left(\frac{\partial^3\zeta}{\partial x^3} + \frac{\partial^3\zeta}{\partial x\partial y^2}\right)\zeta^* + \left(\frac{\partial^2\zeta}{\partial x^2} + \nu\frac{\partial^2\zeta}{\partial y^2}\right)\left(\frac{\partial\zeta}{\partial x}\right)^* - (1-\nu)\frac{\partial^2\zeta}{\partial x\partial y}\left(\frac{\partial\zeta}{\partial y}\right)^*\right], \tag{3}$$

where I_x is the intensity in the x-direction, D is the flexural rigidity and ν is Poisson's ratio of the plate. Complex conjugates are denoted by an asterisk and Im [] denotes the imaginary part of a complex number.

2.3 Modal expansion of structural intensity on a plate

The modal expansion of structural intensity can be derived by substituting Eq. (1) into (3) to obtain the following:

$$I_x(x, y) = \sum_{m=1}^{N}\sum_{n=1}^{N} \beta_{mn}\Phi^x_{mn}(x, y). \tag{4}$$

Here, β_{mn} is the weight coefficient of structural intensity and is mainly dependent on the excitation point as described in

$$\beta_{mn} = \frac{\omega D}{2}\left(\mathrm{Re}[\alpha_m]\mathrm{Im}[\alpha_n] - \mathrm{Im}[\alpha_m]\mathrm{Re}[\alpha_n]\right), \tag{5}$$

and Φ^x_{mn} is the x-direction component of the cross-modal function given by

$$\Phi^x_{mn} = \Phi^{x\ sf}_{mn} + \Phi^{x\ bm}_{mn} + \Phi^{x\ tm}_{mn}, \tag{6}$$

where $\Phi^{x}_{mn}{}^{sf}$, $\Phi^{x}_{mn}{}^{bm}$ and $\Phi^{x}_{mn}{}^{tm}$ correspond to the shear force, bending moment and torsional moment components, respectively. These are calculated using the following equations as the product of two mode shapes with spatial derivatives:

$$\Phi^{x\ sf}_{mn} = \phi_m\left(\frac{\partial^3\phi_n}{\partial x^3} + \frac{\partial^3\phi_n}{\partial x\partial y^2}\right) - \phi_n\left(\frac{\partial^3\phi_m}{\partial x^3} + \frac{\partial^3\phi_m}{\partial x\partial y^2}\right), \tag{7a}$$

$$\Phi^{x\ bm}_{mn} = -\frac{\partial\phi_m}{\partial x}\left(\frac{\partial^2\phi_n}{\partial x^2} + \nu\frac{\partial^2\phi_n}{\partial y^2}\right) + \frac{\partial\phi_n}{\partial x}\left(\frac{\partial^2\phi_m}{\partial x^2} + \nu\frac{\partial^2\phi_m}{\partial y^2}\right), \tag{7b}$$

$$\Phi^{x\ tm}_{mn} = (1-\nu)\frac{\partial\phi_m}{\partial y}\frac{\partial^2\phi_n}{\partial x\partial y} - (1-\nu)\frac{\partial\phi_n}{\partial y}\frac{\partial^2\phi_m}{\partial x\partial y}. \tag{7c}$$

Note that the cross-modal functions have no orthogonality.

When the damping of the plate is small and the excitation force acts at the resonance frequency, the structural intensity in modal form, as given in Eq. (4), can be approximated as

$${}^rI_x \approx \sum_{n=1}^{N} \beta_{rn}\Phi^x_{rn}, \tag{8}$$

where

$${}^r\beta_n = \beta_{rn} = -\frac{\omega_r D}{2}\mathrm{Im}[\alpha_n]\mathrm{Re}[\alpha_n]. \tag{9}$$

The superscript r denotes the variables at the resonance frequency. The intensity and cross-modal function in the y-direction can be obtained by exchanging x and y.

2.4 Normalization of cross-modal functions

When we investigate the correlation between structural intensity and the cross-modal functions, it is convenient to normalize the cross-modal functions as follows:

$$\Phi^{x,reg}_{rn} = \frac{\Phi^x_{rn}}{|\Phi_{rn}|_{\max}}, \tag{10}$$

$$|\Phi_{rn}| = \sqrt{\left(\Phi^x_{rn}\right)^2 + \left(\Phi^y_{rn}\right)^2}, \tag{11}$$

$${}^r\beta^{reg}_n = {}^r\beta_n \times |\Phi_{rn}|_{\max}. \tag{12}$$

Here, $|\Phi_{rn}|_{\max}$ is the maximum norm of the cross-modal function. In this paper, we refer to the cross-modal functions and weight coefficients described in the Eqs. (10) and (12), respectively.

2.5 Method for selecting the position to apply the point force

Here, we propose a method for selecting the position at which to apply the excitation force such that the desired structural intensity related to the cross-modal functions is realized. Modifying the position where the force is applied changes only the weight coefficients ${}^{r}\beta_n^{reg}$. Accordingly, to excite a particular cross-modal function, we can select an excitation position to increase the weight coefficient corresponding to the cross-modal function relative to the other coefficients.

The selection method has three steps: step 1, calculate cross-modal functions; step 2, select a cross-modal function to be primarily excited; step 3, determine the position at which to apply a force to increase the weight coefficient corresponding to the cross-modal function selected in the step 2. From Eqs. (2), (9), (12), it follows that when the excitation force acts at the resonance frequency, that is, the excitation frequency $\omega = \omega_r$, the weight coefficients are expressed as follows:

$$
{}^{r}\beta_n^{reg} = -\frac{F^2 D_p}{2\eta_r \omega_r} \times \frac{\left(\omega_n^2 - \omega_r^2\right)}{\left\{\left(\omega_n^2 - \omega_r^2\right)^2 + \eta_n \omega_n^4\right\}} \times \left|{}^{r}\Phi_n\right|_{\max} \times \phi_r\left(x_F, y_F\right)\phi_n\left(x_F, y_F\right). \tag{13}
$$

The first term on the right side is affected by only material properties and the magnitude of the excitation force. The second term is dependent on the relation between the excitation and natural frequencies. The third term is the magnitude of the cross-modal function, which is dependent on natural mode shapes. The last term is described by the natural mode shapes at the excitation position. The second and third terms decrease with increasing separation between the excitation frequency ω_r and the natural frequency ω_n.

Therefore, we can consider the following two important points to select the excitation location. One is to consider only the product of φ_r and φ_n for increasing or decreasing the weight coefficients ${}^{r}\beta_n^{reg}$. The other is to consider the difference between the excitation frequency and the natural frequency corresponding the cross-modal function; this difference should be small.

3. CONCEPT FOR NOISE REDUCTION BY INTERRUPTION AND PROMOTION OF POWER TRANSMISSION BASED ON STRUCTURAL INTENSITY TECHNIQUE

In this section, we introduce a new structural design concept where the structural intensity technique is used to reduce the sound radiated from a compound plate structure. The concept is based on the modal expansion of structural intensity on a flat plate. First, we discuss the corresponding transmitted power of cross-modal functions to introduce the concept. Then, we describe the concept in detail.

3.1 Transmitted power of cross-modal functions

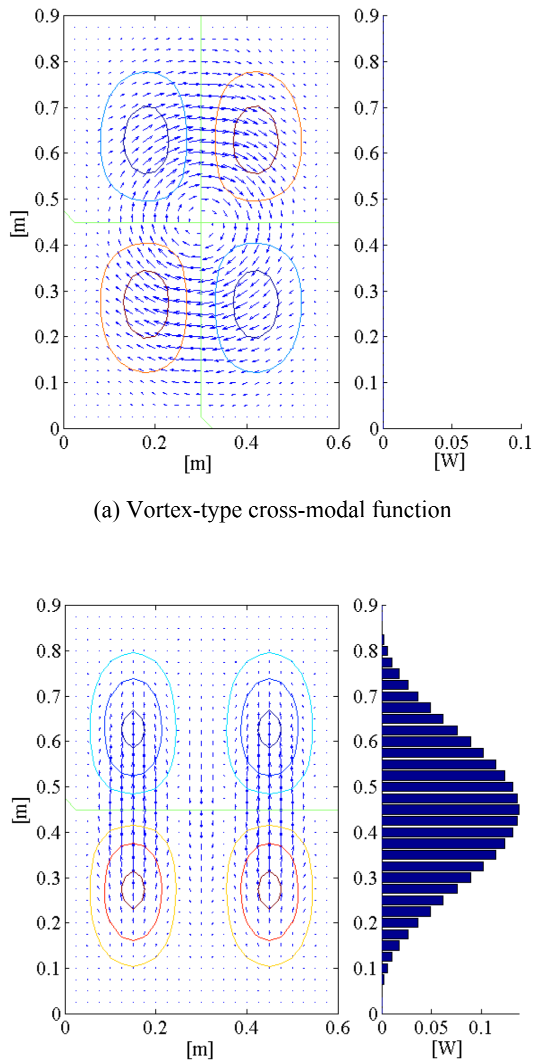
The cross-modal functions can be classified into two types of power flow: vortex-type and straight-type distributions. As shown in Figure 1(a), the vortex-type cross-modal function is real vortex flow. The center of the vortex is at the intersection point of the mode product, which is shown as a contour plot. The integration of intensity across the plate which corresponds to the transmitted power is zero. On the other hand, the straight-type cross-modal function, as shown in Figure 1(b) is flow crossing the plate and power transmission is promoted.

Note that we can roughly estimate the approximate flow distribution of a cross-modal function from the mode product $\varphi_m\varphi_n$, which is shown in Figure 1 as a contour plot. This estimation is possible because the mode product is the energy potential of the cross-modal function. The estimation guidelines are as follows:

(1) The directionality of the cross-modal function at a point corresponds to the direction from the antinode to the other antinode of the mode product on the plate.

(2) Vortex flow of the cross-modal function is created at the center at the intersection points of the node lines.

These guidelines allow calculation of the spatial derivatives of natural modes as shown in Eq. (9) to be avoided when estimating the cross-modal functions.



(a) Vortex-type cross-modal function

(b) Straight-type cross-modal function

Figure 1. Examples of vortex-type and straight-type cross-modal functions $\mathbf{\Phi}_{mn} = (\Phi^x_{mn}, \Phi^y_{mn})$ (vectors), the mode product $\varphi_m\varphi_n$ (contours) and the corresponding power transmission along the y-axis (bars)

3.2 Concept for reducing structure-borne sound by interruption and promotion of transmitted power

According to the modal expression of the intensity, the enhancement of the vortex-type cross-modal function relative to the total intensity would result in less power transmission through the plate, that is, interruption of power. In contrast, the enhancement of the straight-type cross-modal function relative to the intensity would increase power transmission, that is, promotion of power.

Figure 2 shows an overview of the proposed structural design concept based on “the interruption and promotion of transmitted power” for reducing structure-borne sound in mechanical structures.

Let us consider a structure consisting of plate-like subsystems, as shown in Figure 2. A driving unit such as an engine or motor is assumed to be set on one subsystem. Under such conditions, the power input to the excited subsystem is transmitted to neighboring subsystems, the transmitted power then propagates to neighbors, and further propagation continues in this manner. At last, the sound is radiated from a subsystem that is exposed to a space that is required to be quiet. The power transmission through the structure can be evaluated in terms of structural intensity.

If the power transmitted to the radiating subsystem can be reduced, the radiated sound would in turn be decreased. Hence, we consider the vortex-type intensity on a subsystem or subsystems between the source and the radiating subsystem because the vortex-type cross-modal function transmits no energy through the subsystem. This is the interruption of transmitted power by the vortex-type intensity. On the other hand, the residual power input must be transmitted to subsystems unrelated to the radiating subsystem. Then, the straight-type intensity would be useful for promoting power transmission to the unrelated subsystems.

To realize the vortex-type or straight-type intensity and cross-modal functions, we carry out two procedures. One is to change the excitation location to control the weight coefficients ${}^{r}\beta_n^{reg}$ of the intensity, which is especially related to the weigh function α_n of vibration displacement. The other is to change the cross-modal function in order to modify the natural modes, that is, to modify the structure. Currently, we are studying these two procedures. In this paper, the procedure of changing the excitation location is used to generate vortex-type and straight-type intensity.

4. VERIFICATION OF CONCEPT BY NUMERICAL SIMULATIONS

In this section, we verify the concept described in Section 3.2 by numerical simulations.

4.1 Analysis conditions

A J-shaped structure, as shown in Figure 3, is considered. This compound plate structure consists of three flat rectangular plates made of steel. The dimensions of the plates are 0.35 m in width, 1.5 mm in thickness and 0.5 m, 0.4 m or 0.25 m in length. All boundaries excluding the coupling lines are simply supported.

A point excitation force is applied at a point on plate 1. Plate 3 is taken as the radiating subsystem, for which vibration should be small to minimize noise radiation. The straight-type intensity on plate 2 is examined to evaluate the promotion of power transmission. The vortex-type intensity on plate 2 is examined to evaluate the interruption of power transmission.

The numerical simulations are carried out using Femap for Nastran to calculate the natural mode shapes and frequencies. Matlab is used to calculate the structural intensity and cross-modal functions, and FFT/Actranas is used to calculate the radiated sound at a virtual observation plane parallel to plate 3 at a distance of 0.01 m.

The plates are made of steel with the following properties: Young's modulus, E=206 GPa; Poisson's ratio, ν=0.32; and mass density, ρ=7834 kg/m^3. The magnitude of the harmonic excitation point force is F=1 N and the two excitation angular frequencies of $\omega=\omega_6$=2*π*191 rad/s and $\omega=\omega_{12}$=2*π*335 rad/s are considered. The calculation of the vibration displacement and intensities in Matlab is based on Eqs. (1) and (8) for a number of retained modes of N=25 and modal damping of 0.01.

To investigate the effects of the interruption and promotion of power transmission, we evaluate the power input by a point excitation, the power transmitted between plates and the radiated sound power. The power input P_{in} is calculated from

$$P_{in} = \frac{1}{2}\mathrm{Re}\left[F\left\{j\omega\varsigma\left(x_F, y_F\right)\right\}^*\right], \quad (14)$$

where * indicates a complex conjugate. The transmitted power P_{tra} from plate i to plate j is calculated from the structural intensity as

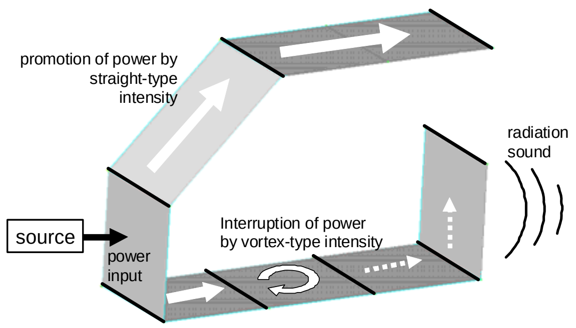


Figure 2. Concept of interruption and promotion of power transmission between subsystems by using structural intensity in modal form

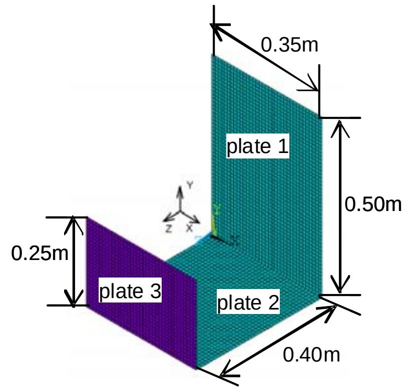


Figure 3. J-shaped test structure with simply supported boundaries

$$P_{tra} = \int I_y(x,y)\,dx \;, \tag{15}$$

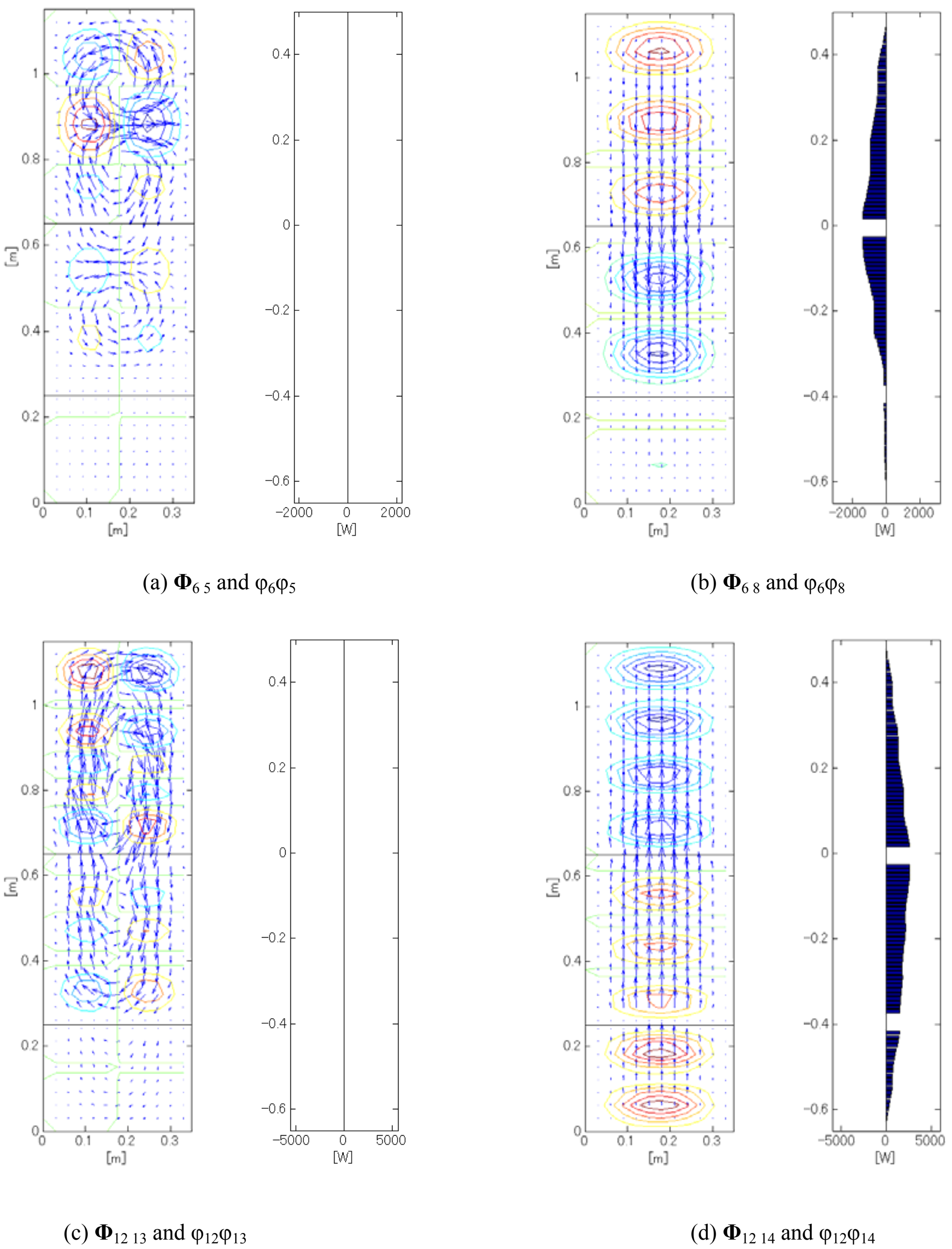
where $I_y(x, y)$ is the intensity in the vertical direction, parallel to the coupling line at the point (x, y) on each plate; and y is the coordinate of the point along the coupling line. Positive intensity is defined as the intensity of power transmitted from plate 3 to plate 2 and from plate 2 to plate 1. Finally, the sound power P_{rad} is evaluated using the following equation:

$$P_{rad} = \int AI(x,y)\,ds \;, \tag{16}$$

where $AI(x, y)$ is the normal acoustic intensity at point (x, y) on the observation plane and ds is the area for $AI(x, y)$. In this paper, to evaluate the concept described above, we consider only radiating plate 3 to calculate sound power.

4.2 Calculation results and discussion

First, we present examples of the cross-modal functions of the J-shaped structure in Figure 4 in the expansion view. The cross-modal functions are calculated from the natural modes with spatial derivatives by using Eq. (6). The cross-modal functions $\mathbf{\Phi}_{6\,5}$ and $\mathbf{\Phi}_{12\,13}$ are vortex-type, and $\mathbf{\Phi}_{6\,8}$ and $\mathbf{\Phi}_{6\,14}$ are the straight-type. Their positive and negative powers transmitted through the plate are also shown in Figure 4. The powers of the vortex-type cross-modal functions are zero. In the case of the straight cross-modal function shown in Figure 4(b), negative power is observed, which signifies that the power through the plates goes from plate 1 to plate 3. Hence, we can see that the transmitted power of the cross-modal function is the same as that for the flat plate shown in Figure 1. Our concept as described in Section 3.2 is therefore useful



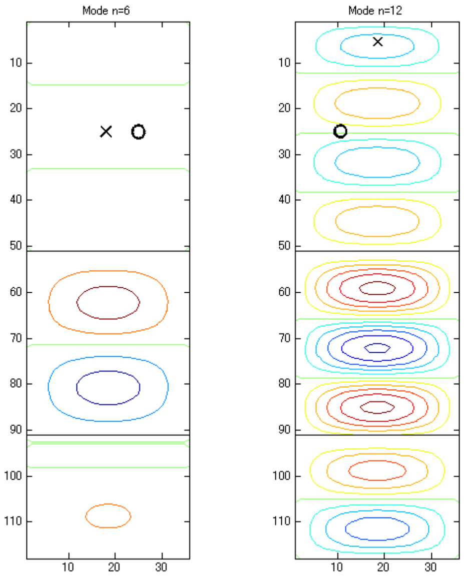
(a) $\mathbf{\Phi}_{6\,5}$ and $\varphi_6\varphi_5$

(b) $\mathbf{\Phi}_{6\,8}$ and $\varphi_6\varphi_8$

(c) $\mathbf{\Phi}_{12\,13}$ and $\varphi_{12}\varphi_{13}$

(d) $\mathbf{\Phi}_{12\,14}$ and $\varphi_{12}\varphi_{14}$

Figure 4. Examples of cross-modal functions (vectors), mode products (contours) and corresponding transmitted powers of the cross-modal functions (bars)



(a) φ_6 (b) φ_{12}

Figure 5. Two resonance modes considered and each point excitation location: ○ for the vortex-type intensity and × for the straight-type intensity

in the case of a compound plate structure.

Next, we present two verification results for excitation frequencies of 191 Hz at the 6th resonance and 355 Hz at the 12th resonance. These resonance modes are shown in Figure 5. For each resonance excitation frequency, we consider two excitation points × and ○ shown in Figure 5; the position × is for creating straight-type intensity on plate 2 to evaluate power promotion, and the position ○ is for creating vortex-type intensity to evaluate power interruption. These excitation points are selected by our method presented in Section 2.5. We calculate the intensity, the power input, the transmitted power and the radiated sound power.

4.2.1 Verification at the 6th resonance

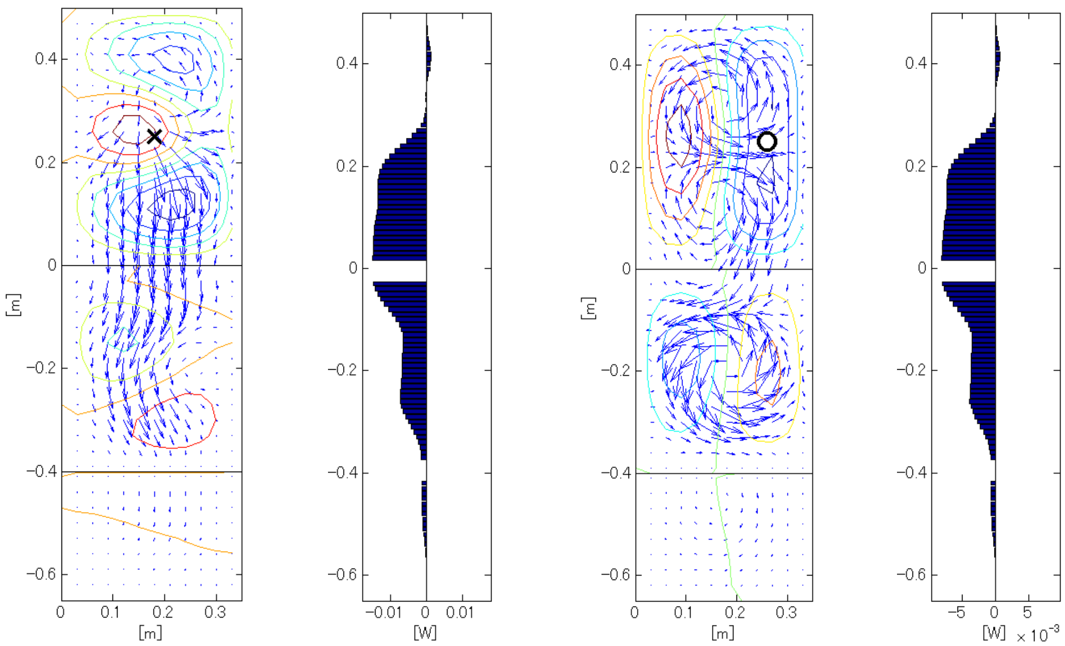
Figure 6 shows the results of the calculated total structural intensity (vectors), vibration displacement (contours) and transmitted power (bars) through the width of the plates. Figure 6(a) shows the results for promoting power transmission by generating straight-type intensity, and Figure 6(b) shows the results for interrupting power transmission by generating vortex-type structural intensity. Table 1 shows the comparison of the powers and vibration energies of the plates between the straight-type and vortex-type intensity. The power input, the transmitted power between plates and the radiation power are estimated from Eqs. (14), (15) and (16),respectively. The vibration energy E_i of plate i is estimated from

$$E_i = m_i V^2 , \tag{17}$$

Table 1. Comparison of powers and energies for the 6th resonance

	Straight-type intensity	Vortex-type intensity	Ratio[a]
P_{in}	1.00000	1.00000	1.00000
$P_{tra,12}$	0.78936	0.85827	0.91971
$P_{tra,23}$	0.07269	0.08096	0.89790
P_{rad}	0.00147	0.00164	0.89432
E_1	0.01816	0.01204	1.50839
E_2	0.05595	0.06062	0.92302
E_3	0.00553	0.00616	0.89868

[a]Ratio of the value in the case of vortex-type intensity to that in the case of straight-type intensity.



(a) Straight-type intensity by excitation at (0.18, 0.25) (b) Vortex-type intensity by excitation at (0.26, 0.25)

Figure 6. Results of calculated total intensities (vectors), transmitted power (bars) and vibrational displacement (contours) for the 6th resonance

where m_i is the total mass of plate i and V_i^2 is the spatial average velocity on plate i.

The values in Table 1 are under the power input is 1.0. “Ratio” in Table 1 refers to the ratio of the values in the case of enhanced vortex-type intensity to that in the case of straight-type intensity. The ratio of the radiated sound power is 0.89432, which shows the sound power radiated from plate 3 with the vortex-type structural intensity on plate 2 is much smaller than that with the straight-type intensity. This result suggests that the power transmitted to plate 3 that radiates sound can be interrupted by the vortex-type intensity on plate 2. The interrupted power is stored in the vibrational energy of plate 1. This is expressed as the ratio of E_1 in Table 1. The value of the ratio is 1.50839, which is lager than 1.0. As a result, we find that the vortex-type intensity on plate 2 interrupts the power transmission from the source to the radiating plate but the energy on the source plate is increased. On the other hand, the results indicate that straight-type intensity on plate 2 promoted power transmission.

Table 2. Comparison of powers and energies for the 12th resonance

	Straight-type intensity	Vortex-type intensity	Ratio[a]
P_{in}	1.00000	1.00000	1.00000
$P_{tra,12}$	0.19507	0.00022 W	0.25241
$P_{tra,23}$	0.00225	0.00015 W	0.01045
P_{rad}	0.00006	0.00014 W	0.26044
E_1	0.03227	0.011041	3.09857
E_2	0.00915	0.02664	0.34358
E_3	0.00163	0.00788	0.20632

[a] Ratio of the value in the case of vortex-type intensity to that in the case of straight-type intensity.

4.2.2 Verification at the 12th resonance

Here, we present addition results that verify the design concept proposed in this paper. These results are for the excitation frequency of 355 Hz at the 12th resonance.

Figure 7 and Table 2 show the results, which are analogous to the results in Figure 6 and Table 1 for the 6th resonance. The two excitation points shown in Figure 5(b) are considered for generating straight-type and vortex-type intensity on the J-shaped structure.

A comparison of the values in Tables 1 and 2 indicates that power transmission is interrupted to a greater extent at the 12th resonance than at the 6th resonance. This is because the ratios of the sound power and the vibration energy in Table 2 differ more greatly than those in Table 1.

From the two verification results described above, we can conclude that the concept proposed in Section 3.2 is useful for reducing structure-borne sound by interruption and promotion of transmitted power.

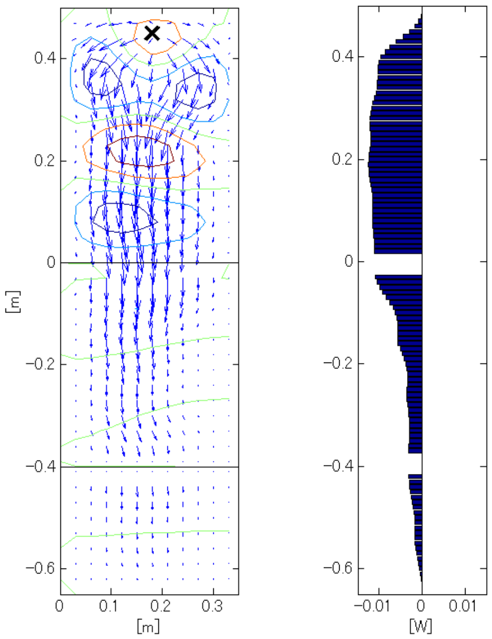
5. CONCLUSIONS

We have investigated a new structural design procedure based on the structural intensity technique. The main conclusions of this study are as follows:

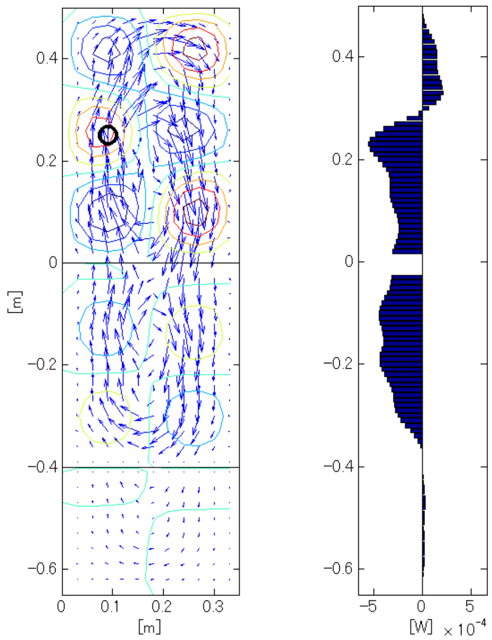
1) A new concept for reducing structure-borne sound, that is, “interruption and promotion of transmitted power by using structural intensity technique” was proposed.

2) The proposed concept is based on the modal expansion of structural intensity and on the cross-modal function that does not contribute to the transmission of power through subsystems.

3) Numerical simulations using a J-shaped structure were carried out to verify the effectiveness of the concept.



(a) Straight-type intensity by excitation at (0.18, 0.45)



(b) Vortex-type intensity by excitation at (0.09, 0.25)

Figure 7. Results of the total intensities (vectors), transmitted power (bars) and vibration displacement (contours) for the 12th resonance

ACKNOWLEDGEMENTS

This work was supported by a Grant-in-Aid for Scientific Research (C) No. 21560251.

REFERENCES

1 D.U.Noiseux, “Measurement of power flow in uniform beams and plates”, *Journal of Acoustical Society of America*, **47,** 238-247, (1970).
2 G.Pavic, “Measurement of structure borne wave intensity – Part I: formulation of the methods”, *Journal of Sound and Vibration*, 4**9**(2), 221-230, (1976).
3 L.Gavric, G.Pavic, “A finite element method for computation of structural intensity by the normal mode approach”, *Journal of Sound and Vibration*, **164**(1), 29-43, (1993).
4 T.Yamazaki, M.Kamata, S.Ohno, “Practical approach of structural intensity measurement applied for box-like structure”, *Proceedings of Fifth International Congress on Sound and Vibration*, Vol.5, 2543-2550, (1997).
5 H.P.Lee, S.P.Lim, M.S.Khun, “Diversion of energy flow near crack tips of a vibrating plate using the structural intensity technique”, *Journal of Sound and Vibration*, **296**, 602-622, (2006).
6 T.Yamazaki, M.Kamata, “Structural Vibration Analysis Using Structural Intensity Measurement”, *Transactions of the Japan Society of Mechanical Engineers*, Series C, Vol.**65**, No.633, 1772-1777, (1999). (in Japanese)
7 X.D.Xu, H.P.Lee, C.Lu, “Power flow paths in stiffened plates”, *Journal Sound and Vibration*, **282**, 1264-1272, (2005).
8 Z.S.Liu, H.P.Lee, C.Lu, “Passive and active interior noise control of box structures using the structural intensity method”, *Applied Acoustics*, **67**, 112-134, (2006).
9 T.Yamazaki, “Modal Expansion of Structural Intensity for Flexural Vibration on Beams”, *Transactions of the Japan Society of Mechanical Engineers*, Series C, Vol.**73**, No.731, 1948-1954, (2007) (in Japanese)
10 Numata.N, Yamazaki.T, “Determination of A Force Applying Vortex-Typed Structural Intensity”, *Proceedings of the 17th International Congress on Sound and Vibration*, Submitted (2010)
11 Nakagawa N., Higashi A. and Sekiguchi Y., “Study on Sound Generating Mechanism Using Vibrational Energy and Sound Energy (2nd Report, Relationship between Structural Intensity Vortex Flow and Sound Field)”, *Transactions of the Japan Society of Mechanical Engineers*, Series.C, Vol.**62**, No.593, 97-103, (1996) (in Japanese).
12 T.Yamazaki, Y.Murakami, N.Numata, “Sound and Vibration on Plate in Vortex Structural Intensity”, *Proceedings of International Workshop on Environment & Engineering 2009* (IWEE 2009), 1B4 (2009).