Active Structural Acoustic Control of Sound Power Radiation from a Soft-Core Sandwich Panel

Kiran SAHU[1]; Jukka TUHKURI[2];

[1,2] Aalto University, Finland

ABSTRACT

In this paper, active control of harmonic sound transmitted through a soft-core sandwich panel is studied. As it has already been shown for the low frequency region, the noise transmission through a soft-core sandwich panel mainly occurs due to the flexural and the dilatational modes [Rimas Vaicaitis, NASA Technical Note, NASA TN D-8516, 1977]. Therefore, in this study, the volume velocity and weighted sum of spatial gradients methods are used to control these modes, and achieve sound attenuation in a broad frequency range. A point force actuator is used as the secondary force to control the radiation modes of the bottom faceplate. Radiated sound power from these two control theories are compared for different values of isotropic loss factor of core. Numerical studies indicate that irrespective of core loss factor weighted sum of spatial gradients method works well in a large frequency band without increasing the radiated sound power unlike the volume velocity method.

Keywords: Sound, volume velocity, WSSG I-INCE Classification of Subjects Number(s): 38.3

1. INTRODUCTION

Light sandwich panels are extensively used in many fields, because of the advantages they offer of high strength-to-weight ratios. The most important advantage of sandwich structures is that optimal designs can be obtained for different applications by choosing different materials and geometric configurations of the faceplates and the core. However, the acoustical properties of these light and stiff structures can be less desirable at low frequencies. These undesirable properties can lead to high noise levels. Therefore, new means of providing noise attenuation at low frequencies need to be established.

Sound transmission characteristics of sandwich panels have been investigated by many authors. Ford et al. [1] were the first to study the effects of dilatational modes of sandwich panels on sound transmission loss. They found that the dilatational mode of vibration depends primarily on mass of the face sheets and thickness of the core. Smolenski and Krokosky [2] included volumetric and shear terms in the strain energy calculation done by Ford et al. [1]. They noticed that the flexural modes of vibration do not change significantly with a change of the thickness or Poisson’s ratio of the core, whereas the dilatational modes of vibration respond dramatically to these properties of the core. Wave impedance analysis approach was used to calculate the sound transmission loss of sandwich panels by Dym and Lang [3,4]. They noticed that a high transmission loss can be achieved by choosing the faceplates, whose symmetric and anti-symmetric impedances have similar values. Vaicaitis [5] formulated a viscoelastic model of a sandwich panel by taking the core both as soft and hard, and analyzed the sound transmission properties of both the types of core. He found that the sandwich panels with soft viscoelastic cores exhibit noise transmission characteristics similar to those of double wall elastic panels except in the frequency range where dilatational mode occurs, and sandwich panels with hard core reduce noise significantly as compared to the elastic panels in a broad frequency range.

Conventional methods of suppressing acoustic noise using passive sound absorbers generally do not work well at low frequencies. Therefore, active methods are used to attenuate the low frequency sound. Johnson and Elliot [6] studied the effect of minimizing the radiated sound power (SPM) and cancelling the volume velocity (VVC) on the global sound power reduction. They proposed that SPM strategy, which is difficult to implement in practice, can be replaced by VVC to get large amount of

[1] kiran.sahu@aalto.fi
[2] jukka.tuhkuri@aalto.fi

reduction in sound power at low frequencies. By using VVC, Pan et al. [7] studied the control of sound transmission through a double-leaf partition.

However, VVC strategy requires a large number of sensors. For example, to accurately measure the volume velocity of a plate of size 0.278 m x 0.247 m, 16 to 25 sensors are required [8]. The increase in the number of sensors can be avoided by using a distributed piezoelectric sensor to measure volume velocity [9]; however, this sensor would need to be designed for the specific geometry. A recent control metric, termed composite velocity (also referred to as WSSG or the weighted sum of spatial gradients), has shown promise in resolving these issues. Composite velocity was developed as a weighted sum of spatial velocity gradients requiring only four sensors to measure, and was found to be relatively insensitive to sensor location on a simply-supported plate [10].

From all the above mentioned works, it is apparent that extensive studies have already been done to investigate the sound transmission through sandwich panels. However, active control of sound transmission through soft-cored sandwich panels by active means is a fairly unexplored topic. In the present investigation, an analytical study of a simply supported sandwich panel has been considered. Both VVC and WSSG strategies are used to drive a point force actuator, which is attached to the bottom faceplate of the sandwich panel. Also, the effectiveness of both the strategies has been studied for different values of isotropic loss factor of the core.

2. MODELING

Consider a physical system consisting of a rectangular soft-cored sandwich panel, as shown in Figure 1. A harmonic plane wave is incident on the top surface of the sandwich panel and the bottom face of the sandwich panel is subjected to a control force. It is assumed that the sandwich panel is flat and simply supported at all four edges. It is also assumed that the faceplates and the soft viscoelastic core of the panel are isotropic.

Since a very soft viscoelastic core is considered, Poisson's ratio of the material is nearly zero; hence, the core can be approximated as a viscoelastic spring. Assuming the small deflection theory of plates based on the classical plate theory, the governing equations of motion can be written as,

$$D_t \nabla^4 w_t + \left(\frac{1}{3}\rho_c h_c + \rho_t h_t\right)\ddot{w}_t + \frac{1}{6}\rho_c h_c \ddot{w}_b + c_t \dot{w}_t + k_c\left(w_t - w_b\right) = p_e(x,y,t) \tag{1}$$

$$D_b \nabla^4 w_b + \left(\frac{1}{3}\rho_c h_c + \rho_b h_b\right)\ddot{w}_b + \frac{1}{6}\rho_c h_c \ddot{w}_t + c_b \dot{w}_b + k_c\left(w_b - w_t\right) = p_c(x,y,t) \tag{2}$$

where $D_t = E_t h_t^3 \big/ \left\{12\left(1-\upsilon_t^2\right)\right\}$ and $D_b = E_b h_b^3 \big/ \left\{12\left(1-\upsilon_b^2\right)\right\}$ are the bending stiffness of the top and the bottom faceplates, respectively; $\nabla^4 = \partial^4/\partial x^4 + 2\partial^4/\partial x^2 \partial y^2 + \partial^4/\partial y^4$; w_i, ρ_i, E_i, h_i, c_i, and υ_i are the displacement in the direction normal to the panel, density, Young's modulus, thickness, damping coefficient, and Poisson's ratio respectively; subscript $i = t, b$ and c refers to the top and the bottom faceplates and the core, respectively; a superposed dot indicates a time derivative. The terms $(1/3)\rho_c h_c$ and $(1/6)\rho_c h_c$ represent the contributions of the mass of the viscoelastic core to the displacement amplitude of the top and the bottom faceplates. The viscoelastic spring constant of the core material is $k_c = E_c(1+j\beta)/h_c$, where E_c is the elastic modulus of the core in the direction normal to the plate surface, j is the imaginary number, β is the isotropic loss factor of the core, p_e and p_c are the oblique plane wave incident on the top faceplate and the secondary force applied to the bottom faceplate, respectively.

If the structural vibration is assumed to be described by the summation of M modes, and both the top and the bottom faceplates are simply supported, the forced response can be expanded in terms of normal modes as [5],

$$w_t(x,y,t) = \sum_{m=1}^{M} D_{tm}(t)\,\phi_m(x,y), \quad w_b(x,y,t) = \sum_{m=1}^{M} D_{bm}(t)\,\phi_m(x,y) \tag{3}$$

where D_{tm} and D_{bm} are the generalized coordinates of the top and bottom faceplates, respectively; t represents time, and the mode shape functions $\phi_m(x,y)$ satisfy the orthogonal property and normalized as,

$$S_p = \int_{S_p} \left[\phi_m(x,y)\right]^2 dS \tag{4}$$

where S_p is the area of the sandwich panel.

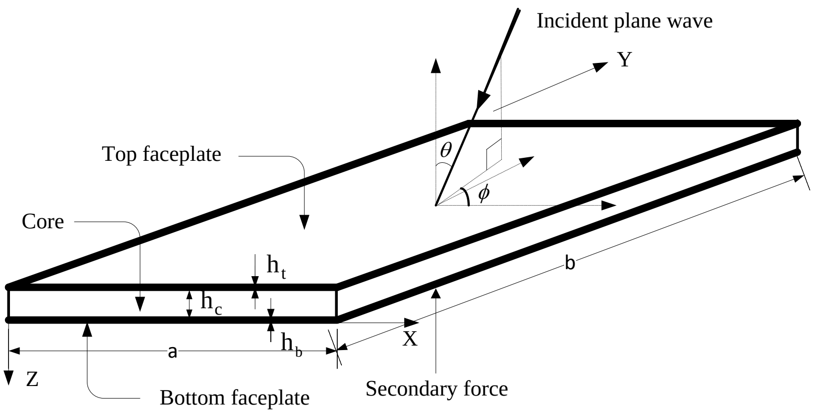

Figure 1 - Sandwich panel excited by a plane wave incident at angles θ and ϕ, and a control force.

Substituting Eq. (3) into Eqs. (1) and, (2) and omitting the co-ordinate axis and the summation sign in the interest of brevity and doing some algebraic manipulations, one will obtain the displacement of the top and the bottom faceplates as,

$$w_t(x,y,\omega) = \frac{1}{S_p}\sum_{m=1}^{M}\int_0^a\int_0^b \left[H_m^{t1} p_e(x,y,\omega) + H_m^{t2} p_c(x,y,\omega)\right]\left\{\phi_m(x,y)\right\}^2 dx\,dy \tag{5}$$

$$w_b(x,y,\omega) = \frac{1}{S_p}\sum_{m=1}^{M}\int_0^a\int_0^b \left[H_m^{b1} p_e(x,y,\omega) + H_m^{b2} p_c(x,y,\omega)\right]\left\{\phi_m(x,y)\right\}^2 dx\,dy \tag{6}$$

Here, ω is the frequency of incident pressure. The expressions for H_m^{t1}, H_m^{t2}, H_m^{b1} and H_m^{b2} are given as,

$$H_m^{t1} = \frac{\gamma_m}{a_t\left(\lambda_m\gamma_m - \sigma_t\sigma_b\right)},\ H_m^{t2} = \frac{\sigma_t}{a_b\left(\lambda_m\gamma_m - \sigma_t\sigma_b\right)},\ H_m^{b1} = \frac{\sigma_b}{a_t\left(\lambda_m\gamma_m - \sigma_t\sigma_b\right)},\ H_m^{b2} = \frac{\lambda_m}{a_b\left(\lambda_m\gamma_m - \sigma_t\sigma_b\right)},$$

$$\sigma_t = \frac{1}{a_t}\left(\frac{1}{6}\rho_c h_c\omega^2 + k_c\right),\ \sigma_b = \frac{a_t}{a_b}\sigma_t,\ \lambda_m = -\omega^2 + j\frac{c_t}{a_t}\omega + \frac{1}{a_t}\left[D_t\left\{\left(\frac{m_1\pi}{a}\right)^2 + \left(\frac{m_2\pi}{b}\right)^2\right\}^2 + k_c\right],$$

$$\gamma_m = -\omega^2 + j\frac{c_b}{a_b}\omega + \frac{1}{a_b}\left[D_b\left\{\left(\frac{m_1\pi}{a}\right)^2 + \left(\frac{m_2\pi}{b}\right)^2\right\}^2 + k_c\right],\ a_t = \frac{1}{3}\rho_c h_c + \rho_t h_t,\ a_b = \frac{1}{3}\rho_c h_c + \rho_b h_b.$$

The eigenfrequencies of the coupled system can be calculated by setting $c_t = c_b = \beta = 0$ and using

$$\lambda_m\gamma_m - \sigma_t\sigma_b = 0 \tag{7}$$

Eq. (7) will give two characteristic values for each modal indices (m_1, m_2). Theses roots represent the in-phase flexural and the out-of phase dilatational vibration resonance frequencies of the sandwich panel.

2.1 Overview of Control Theory

Since the primary sound is incident on the top faceplate and the control force is implemented on the bottom faceplate, so only the expression for composite velocity or weighted sum of spatial gradients (WSSG) for bottom faceplate is calculated, which is given by [10],

$$\text{WSSG} = \alpha\left(w_b^2\right) + \beta\left(\partial w_b/\partial x\right)^2 + \gamma\left(\partial w_b/\partial y\right)^2 + \delta\left(\partial^2 w_b/\partial x\partial y\right)^2 \tag{8}$$

When the spatial derivatives $\partial/\partial x,\ \partial/\partial y,\ \partial^2/\partial x\partial y$ of Eq. (6) are taken, it can easily be seen that each of these terms will be scaled by $m_1\pi/a,\ m_2\pi/b,\ m_1m_2\pi^2/ab,$ respectively. This gives an obvious method to determine weights for each of these terms. By choosing $\alpha = 1,\ \ \beta = a/m_1\pi,\ \ \gamma = b/m_2\pi$ and

$\delta = ab/m_1 m_2 \pi^2$ the magnitudes of each of the spatial gradient terms are scaled to be the same as the magnitude of the $\partial w_b/\partial t$ term. Using these weights yields an extremely uniform field over the surface of the bottom faceplate at a specific mode.

As it is described above, only the volume velocity of bottom faceplate is needed, which is calculated using elemental radiator method given by Johnson and Elliot [6].

3. RESULTS & DISCUSSIONS

Here, the analytical results of radiated sound power from the bottom faceplate of the sandwich panel and its active control using volume velocity and weighted sum of spatial gradients (WSSG) methods are presented. A simply supported soft-core sandwich panel is considered for the numerical study whose top and bottom faceplates are assumed to be of aluminum alloy and the core is of lightweight low modulus viscoelastic material. The material properties of the faceplates and the core are given in Table 1 and the eigenfrequencies associated with the structural mode of the sandwich panel are calculated using Eq. (7) and shown in Table 2. The dimensions of the sandwich panel are same as taken by Vaicaitis [5], where the sandwich panel is $0.25\ \text{m} \times 0.508\ \text{m}$ and the thicknesses of each faceplate and the core are 0.51 mm and 6.35 mm, respectively. A plane wave of pressure amplitude 1 Pa is incident on the top faceplate of the sandwich panel at $\theta = 45^0$ and $\phi = 45^0$. A control force in the form of a point force actuator is acted at the middle of the bottom faceplate and a sensor is located at $(0.15\text{m},\ 0.29\text{m})$ to measure the WSSG. The magnitude and the phase of the control force for each frequency have been simulated to minimize the WSSG at the sensor location. For the analytical simulation, volume velocity has been found by discretizing the plate into 55 elements and estimating the velocity across each individual element as the velocity at the center of the element. This number of elements was calculated as sufficient to give an accurate measure of volume velocity based on the methods described by Sors and Elliott [8]. A total of 55 structural modes are used for the simulations in the frequency range of 0 to 550 Hz and no significant difference has been noticed in simulations for higher number of modes.

Table 1 – Physical parameters of faceplates and core

Aluminum faceplates	$E_t = E_b = 72.4$ GPa, $\upsilon_t = \upsilon_b = 0.3$, $c_t = c_b = 27.13\ \text{N-sec/m}^3$, $\rho_t = \rho_b = 2770\ \text{kg/m}^3$
Viscoelastic Core	$E_c = 34500$ Pa, $\upsilon_c = 0$, $\rho_c = 277\ \text{kg/m}^3$

Table 2 – Eigenfrequencies of the sandwich panel

Mode number	Flexural eigenfrequency (Hz)	Dilatational eigenfrequency (Hz)
(1,1)	19.3	402.3
(1,2)	30.6	403.2
(1,3)	49.5	405.7
(2,1)	66	408.9
(1,4)	75.9	411.2

The radiated sound power from the bottom faceplate at different values of isotropic loss factor of the viscoelastic core has been calculated and shown in Figure 2. Bullets and the small circles show the flexural and dilatational modes of the sandwich panel, respectively. It can be observed from these figures that the noise transmission characteristic of an elastic panel $(\beta = 0)$ and viscoelastic panel $(\beta > 0)$ are similar for frequencies upto 250 Hz, however, after this, viscoelastic panel with soft core significantly attenuates sound power as compared to the elastic panel, which very well agrees with the work done by Vaicaitis [5]. It can be observed that at 402 Hz, where the first dilatational mode occurs, a panel with viscoelastic soft core can attenuates around 40 dB of sound. These results show that the noise transmission into the cavity mainly occurs by flexural modes till 250 Hz and by dilatational modes in the frequency region from 400 to 500 Hz. Therefore, in the following section, volume velocity and WSSG methods are used to control both the flexural and the dilatational modes to get control over the frequency range considered in this study that is from 0 to 550 Hz.

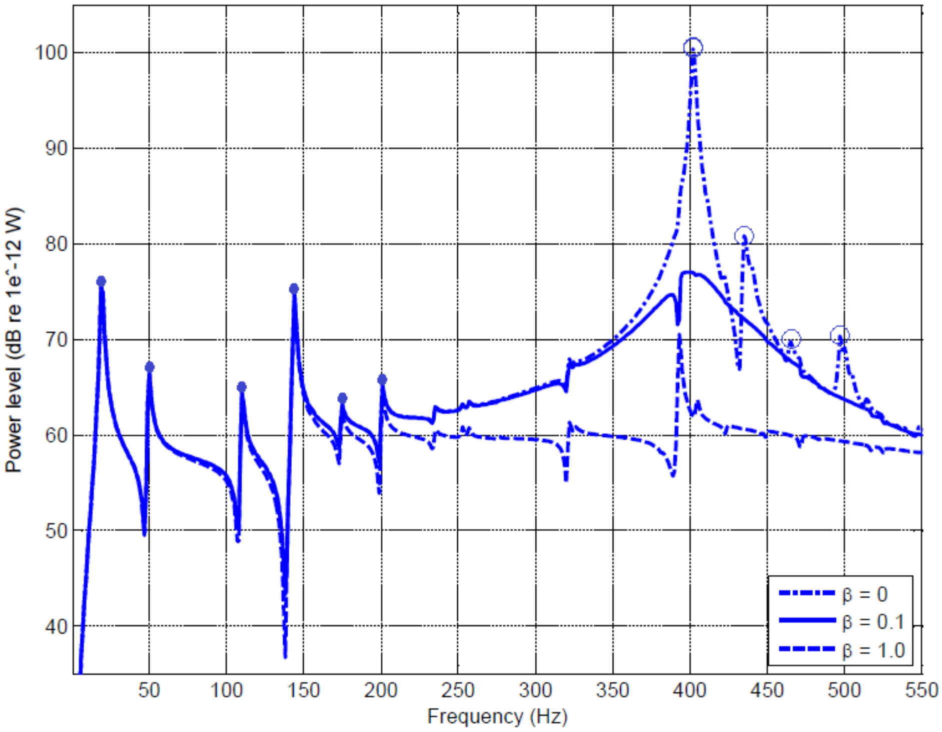

Figure 2 – Radiated sound power from the bottom faceplate at different values of core loss factor

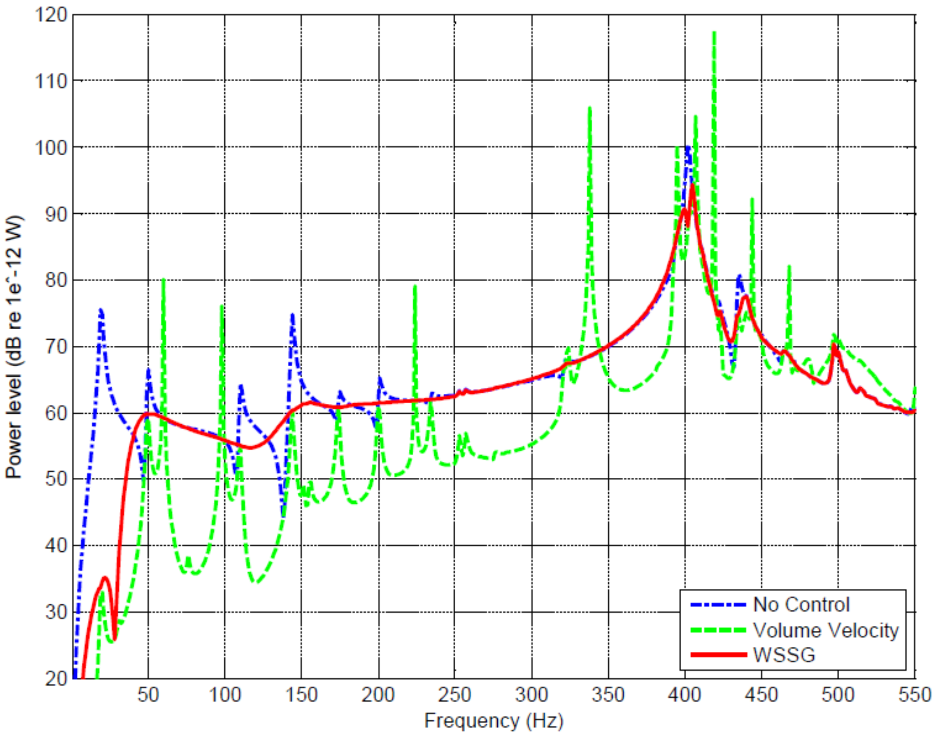

Figure 3 – Radiated sound power from the bottom faceplate at $\beta = 0$

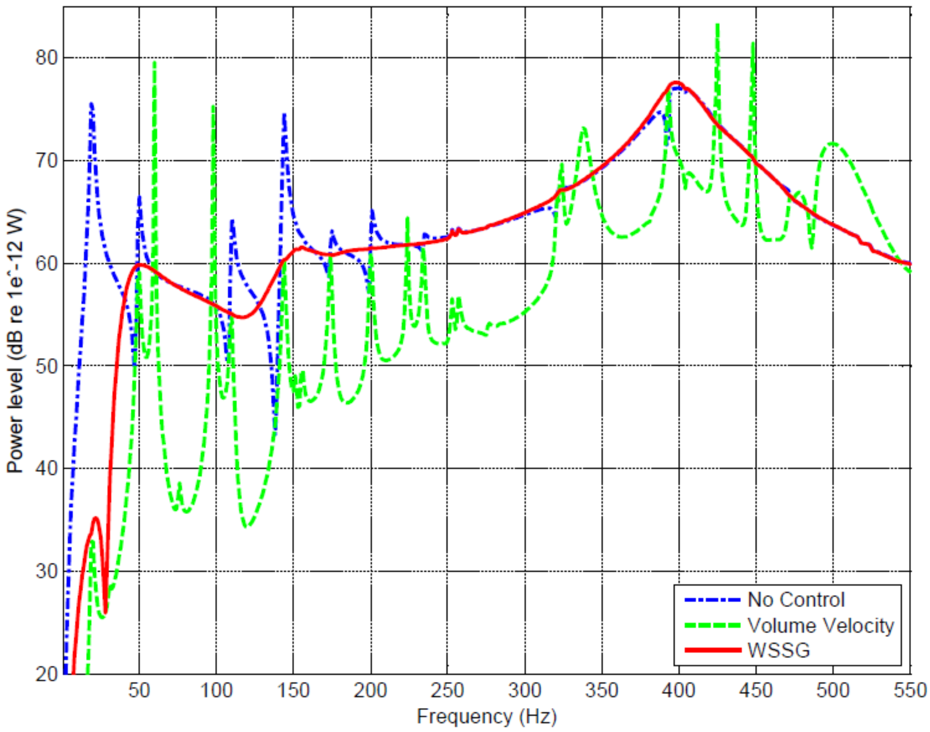

Figure 4 – Radiated sound power from the bottom faceplate at $\beta = 0.1$

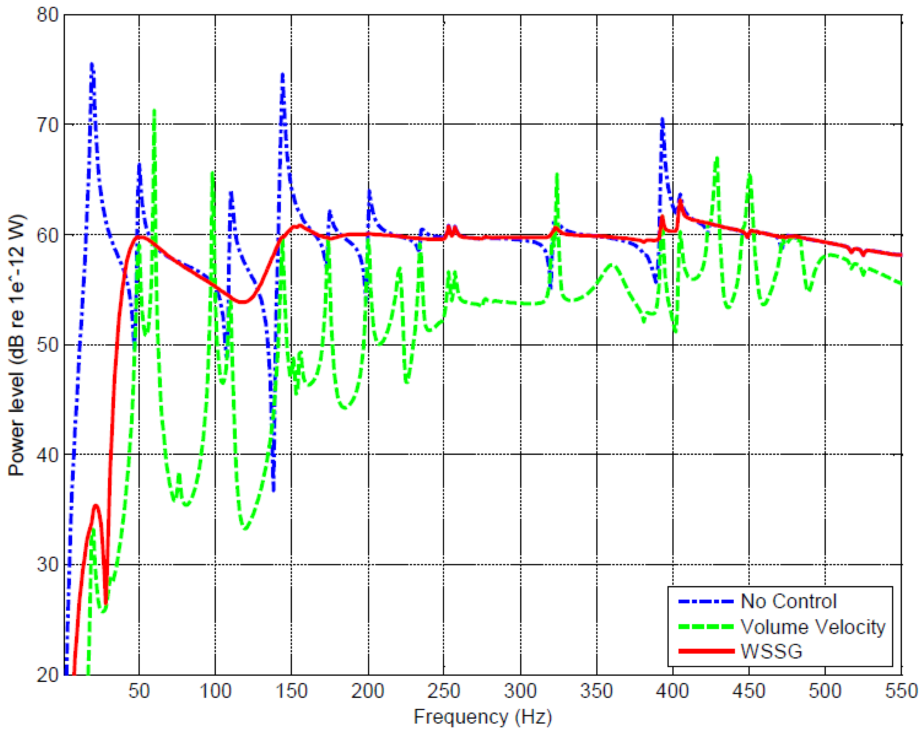

Figure 5 – Radiated sound power from the bottom faceplate at $\beta = 1.0$

Both the uncontrolled and controlled radiated sound power using the two objective functions are plotted for $\beta = 0, 0.1, 1.0$ and shown in Figures 3, 4 and 5, respectively. The point force actuator convincingly control the flexural modes, which are dominating in the frequency region from 0 to 250 Hz, and dilatational modes, which occur in the frequency range from 400 to 500 Hz and hence, attenuate the sound inside the cavity in a broad frequency range, that is, from 0 to 550 Hz. Also, it can be observed that volume velocity method does considerably better, but considering the number of sensors involved, it is less practical in implementation. Also, an important note is that the maximum increase in radiated power is less for WSSG method as compared to the volume velocity method. This is an important consideration for cases where the structural excitation is narrowband in nature. For those cases, it is possible that implementation of active control could increase the radiation for some frequencies of excitation, and it is desired to minimize those possible undesired amplifications. Thus, having a small maximum increase in radiated power is a desirable feature of an effective active control scheme.

4. CONCLUSIONS

This paper reports a study of active control of the radiated sound power by controlling the flexural and dilatational modes of a soft-core sandwich panel using the volume velocity and weighted sum of spatial gradient (WSSG) methods. WSSG method shows better results while only measuring the displacement at four locations on the plate, whereas the volume velocity method requires 55 accelerometers. In particular, WSSG achieved improved control when compared to volume velocity at natural frequencies and modes higher than the fourth mode. This is due to the fact that WSSG could control the second, third, and fourth radiation modes whereas volume velocity could not. Also, the maximum increase in radiated sound power is very high in volume velocity method as compared to WSSG method. However, both the volume velocity and the WSSG methods able to control the flexural and dilatational modes, and therefore control the radiated sound power in a large frequency band.

ACKNOWLEDGEMENTS

The authors acknowledge the financial support received from the National Graduate Program in Engineering Mechanics (Finland).

REFERENCES

1. Ford RD, Lord P, Walker AW. Sound transmission through sandwich constructions. J Sound Vib. 1967;5(1):9-21.
2. Smolenski CP, Krokosky EM. Dilatational-mode sound transmission in sandwich panels. J Acoust Soc Am. 1973;54(6):1449-1457.
3. Dym CL, Lang MA. Transmission of sound through sandwich panels. J Acoust Soc Am. 1974;56(5):1523-1532.
4. Dym CL, Ventres CS, Lang MA. Transmission of sound through sandwich panels: A consideration. J Acoust Soc Am. 1976;59(2):364-367.
5. Vaicaitis R. Noise transmission by viscoelastic sandwich panels. NASA Technical Note, NASA TN D-8516;1977.
6. Johnson ME, Elliot SJ. Active control sound radiation using volume velocity cancellation. J Acoust Soc Am. 1995;98(4):2174-2186.
7. Pan X, Sutton TJ, Elliot SJ. Active control of sound transmission through a double-leaf partition by volume velocity cancellation. J Acoust Soc Am. 1998;104(5):2828-2835.
8. Sors TC, Elliott SJ. Volume velocity estimation with accelerometer arrays for active structural acoustic control. J Sound Vib. 2002;258(5):867-883.
9. Gardonio P, Lee YS, Elliott SJ. Analysis and measurement of a matched volume velocity sensor and uniform force actuator for active structural acoustic control. J Acoust Soc Am. 2001;110(6):3025-3031.
10. Fisher JM, Blotter JD, Sommerfeldt SD, Gee KL. Development of a pseudo-uniform structural quantity for use in active structural acoustic control of simply supported plates: An analytical comparison. J Acoust Soc Am. 2012;131(5):3833-3840.