Analysis of Isolation System for Impulsive Force Device with Recoil Mechanism

반동방식 충격기구의 완충시스템 해석

HyoJun Kim and BongJo Ryu

김 효 준†· 류 봉 조*

(Received September 23, 2004 : Accpeted February 21. 2005)

Key Words : Recoil Mechanism(반동방식), Impulsive Force Device(충격기구), Isolation System(완충시스템)

ABSTRACT

In this study the optimal isolation system for the prototype HIFD(high impulsive force device) is investigated. For this purpose, firstly, the dynamic behavior of a human body and a transmitted force under specific operation conditions are analyzed through a series of experimental works using the devised test setup. In order to design the optimal dynamic absorbing system, the parameter optimization process is performed using the simplified isolation system model based on the experimental results of linear impulse and transmitted force. Finally, under the parameters satisfying the constraints of the buffering displacement and the transmitted force, the performance of the designed isolation system for the prototype HIFD is evaluated by experiment.

요 약

이 논문에서는 프로토타입(prototype) 고충격력 기구에 대한 최적의 절연시스템이 연구된다. 이 목적을 위해 첫째, 고충격력 기구의 특정한 작동조건하에서 인체의 거동과 전달력이 고안된 시험 장치를 이용한 실험을 통해 분석된다. 최적 동흡진기를 설계하기 위하여, 파라미터 최적과정이 충격량과 전달력의 실험결과를 토대로 한 단순화된 절연시스템 모델을 사용하여 이루어진다. 최종적으로, 충격완화 변위와 전달력의 구속조건을 만족시키는 파라미터 하에서, 프로토타입 고충격력 기구에 대해 최적 설계된 절연시스템의 성능이 실험에 의해 평가된다.

1. Introduction

It is very important for precision equipment, devices and operators to avoid severe environmental disturbances such as vibration, shock from other machines.

Therefore, the isolation system design reducing the impulsive force has been studied and applied in many engineering fields. In order to stabilize the total system with an impulsive disturbance, a reduction of the transmitted forceis required in the level maintaining system.

As a research related to the above subject, Chehab et al.[1] Investigated the efficiency of mounting systems for foundations supporting hammer-presses and conducted a comprehensive

† 책임저자 : 정회원, 국립삼척대학교 기계공학과
E-mail : hjkim@samcheok.ac.kr
Tel : (033) 570-6322, Fax : (033) 570-6773
* 정회원, 한밭대학교 기계공학부

parametric study considering the powerful short-period impact loads. Babitsky et al.[2] Studied a vibro-impact structure in a design application of a gimballed electro-optical device. Under appearance of intense impulsive accelerations, the isolator concept is based on co-operative use of an undamped vibration isolator in combination with optimally damped bumpers installed with minimal free travel.

In a research of the dynamic absorbing system, Zhang et al.[3] derived the optimal damping factor of absorbing system in the railway vehicle, and defined the equivalent mass ratio which is able to evaluate the efficiency. Hundal et al.[4] studied the design scheme which is able to induce the optimal response using the pneumatic shock isolator.

After that, Alanoly et al.[5] performed the study of reducing acceleration and relative displacement of the body mass through semi-active actuator. The semi-active actuator can be used for absorbing the force by controlling the internal fluid flow in the actuator with the variable orifice different from fully active control system. Walsh et al.[6] studied on the absorber system with variable stiffness in order to minimize the excessive vibration which can occur in the rotatory machine using the on-off operation process.

A kind of the HIFD(High Impulsive Force Device) penetrating an object by an immediate explosive shot, the transmitted force to the mount system becomes a more important issue. Besides, in case of the HIFD with high performance, larger impact energy can be produced in the aspect of kinetics. In case of a portable HIFD supported by human body directly without special mounting structures, there is a possibility of causing serious problems to the human being[7]. Furthermore, the interaction between a human body and the HIFD is one of important performance factors in an actual operating condition.

Therefore, in order to realize the mechanism of the dynamic absorbing system reducing the transmitted force to the human body from the HIFD, correct prediction for the transmitted impulsive force to the body should be followed in the development of the device.

In this study the isolation of the prototype HIFD is performed considering the dynamic characteristics of a human body structure due to a horizontal impulsive force input. Based on the simplified human structure model for the dominant human body motion, the overall isolation system including the dynamic absorbing system is modeled on the basis of experimental results and analyses between human behaviors and transmitted impulse occurring in an actual utility condition. In order to determine the design parameters of the isolation system, the parameter optimization process is performed considering the trade-off between the buffering displacement and the transmitted force. From solutions satisfying the constraints, the performance of the actual isolation system is evaluated by experiment using a devised test setup and the transmitted force to human being is predicted.

2. Description of Application System

In general, the high impulsive force device penetrates an object by an instant impact shot using explosive gas. The prototype HIFD in this study has the recoil system. The function of a recoil mechanism is to transform the extremely high interior ballistic forces acting on the recoiling parts by the high propellant pressure into tolerable loads that act on the supporting structure. The design objective of this kinds of HIFD is to attenuate very high peak loads into longer

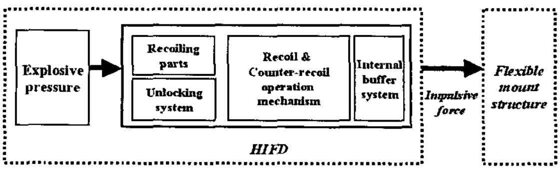

Fig. 1 Schematic diagram of application system

duration with much lower peak values through the dynamic action of the recoiling parts.

Figure 1 shows the schematic diagram of the application system including the prototype HIFD.

In this study, the supporting structure for the prototype HIFD is a human being. Unfortunately, the conventional recoiling mechanism is not enough to reduce the transmitted force to human body structure. Therefore, it is necessary to apply the additional dynamic absorbing system in the prototype HIFD system and the related study has been progressed on the basis of the dynamic model of the prototype HIFD which has the dynamic characteristics as shown in Fig. 2. In the figure, the analytical results have a good agreement with experimental ones although there are some differences caused by the unmodeled high frequency modes.

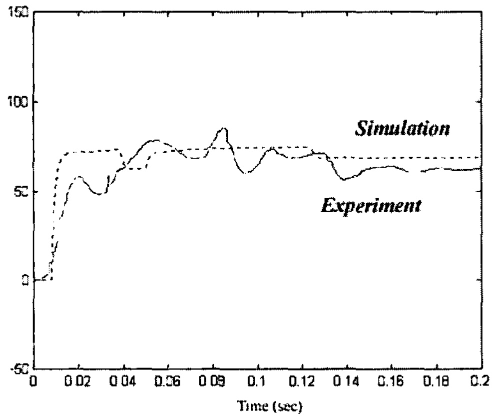

Fig. 2 Comparison of impulses generated by HIFD

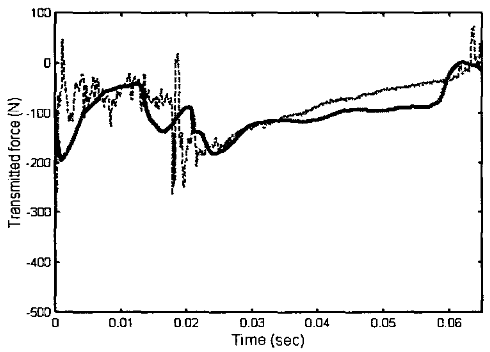

Fig. 3 Comparison of experimental result and estimation result using shear structure model(dotted: experiment, solid: prediction)

In order to analyze the dynamic characteristics of human bodies having complicated structures subjected to impact force from the HIFD, it is necessary to measure and to analyze the human behaviors as a supporting structure.

In the previous works[8~10], on the basis of experimental results on the dynamic characteristics between human bodies and HIFD, the influence of flexible mount part has been investigated using the general purpose HIFD. In the works, some experiments have been performed to measure the transmitted force from the general purpose HIFD to a human being under an actual operating condition. From the result, the transmitted impulse from HIFD can be different depending on the supporting condition such as a human mount and a rigid spring mount. In this reason, it is required to consider the mount condition for the design of the internal mechanism of HIFD.

Figure 3 shows a comparison between the test results and the corresponding analysis results using the simplified human structure model. The shear structure model[8,10,11] can be considered as an analysis model of flexible mount structure. As shown in this figure, analysis results have a good agreement with experimental ones. However, there is a slight difference in the higher modes. Higher modes effect shown in experimental results can be recovered by increasing the degrees of freedom of analysis model. However, since all modes can not be taken into account for the design of a dynamic absorbing system, it is desirable to adopt an analysis model using some principal modes[8].

3. Design of Isolation System

3.1 Optimal Design Process

In this section, the overall isolation system including the dynamic absorbing system is modeled for the purpose of reducing the transmitted force induced by the prototype HIFD.

In the design process, it is very difficult to include the enormous mathematical model of complex human body structure. Therefore, in this study, the overall isolation system considers only dominant lower horizontal mode on the basis of experimental results.[9]

In order to determine the design parameters of the dynamic absorbing system, an optimization process[12] taking account of constraint condition is performed as follows.

The state-space equation of the overall isolation system[12] is expressed as:

$$\dot{X} = AX + BF_i \tag{1}$$

where, $$A = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -\frac{k_s}{M_s} & -\frac{c_s}{M_s} & \frac{k_s}{M_s} & \frac{c_s}{M_s} \\ 0 & 0 & 0 & 1 \\ \frac{k_s}{M_u} & \frac{c_s}{M_u} & -\frac{k_s + k_u}{M_u} & -\frac{c_s}{M_u} \end{bmatrix}$$

$$B = \begin{bmatrix} 0 & \frac{1}{M_s} & 0 & 0 \end{bmatrix}^T, \quad X = \begin{bmatrix} x_s & \dot{x}_s & x_u & \dot{x}_u \end{bmatrix}^T$$

Where M_s is the HIFD mass, M_u is the mount mass, k_s is the equivalent stiffness and c_s is the damping coefficient of the dynamic absorbing system, x_s means the HIFD displacement, x_u is the mount displacement, F_i is the impulsive force input and k_u is the mount stiffness.

In order to decide the optimal parameters of the dynamic absorbing system, the performance index J considering both the buffering displacement and the transmitted force to the mount is defined as Eq. (2) using the state variables.

$$J = \lim_{T \to \infty} \frac{1}{T} E\left[\int_0^T \left(\dot{x}_4^2 + \rho (x_1 - x_3)^2\right) dt\right] \tag{2}$$

where, ρ is the weighting factor.

Equation (2) can be rearranged in the form of Eq. (3) by using the state vector and the symmetric, positive definite matrix Q with dimensions of 4×4.

$$J = \lim_{T \to \infty} \frac{1}{T} E\left[\int_0^T X^T Q X dt\right] \tag{3}$$

where,

$$Q_{11} = \frac{k_s^2}{M_u^2} + \rho$$

$$Q_{12} = \frac{k_s c_s}{M_u^2}$$

$$Q_{13} = \frac{k_s (k_s + k_u)}{M_u^2} - \rho$$

$$Q_{14} = -\frac{k_s c_s}{M_u^2}$$

$$Q_{21} = \frac{k_s c_s}{M_u^2}$$

$$Q_{22} = \frac{c_s^2}{M_s}$$

$$Q_{23} = -\frac{c_s (k_s + k_u)}{M_u^2}$$

$$Q_{24} = -\frac{c_s^2}{M_u^2}$$

$$Q_{31} = \frac{k_s (k_s + k_u)}{M_u^2} - \rho$$

$$Q_{32} = -\frac{c_s (k_s + k_u)}{M_u^2}$$

$$Q_{33} = \frac{(k_s + k_u)^2}{M_u^2} + \rho$$

$$Q_{34} = \frac{c_s (k_s + k_u)}{M_u^2}$$

$$Q_{41} = -\frac{k_s c_s}{M_u^2}$$

$$Q_{42} = -\frac{c_s^2}{M_u^2}$$

$$Q_{43} = \frac{c_s (k_s + k_u)}{M_u^2}$$

$$Q_{44} = \frac{c_s^2}{M_u^2}$$

The performance index of Eq. (3) can be represented as follows.

$$J = Trace\left\{ Q\sum \right\} \tag{4}$$

where,

$$\sum = E\left[X^T X\right] = \begin{bmatrix} \sigma_{11} & \sigma_{12} & \sigma_{13} & \sigma_{14} \\ \sigma_{21} & \sigma_{22} & \sigma_{23} & \sigma_{24} \\ \sigma_{31} & \sigma_{32} & \sigma_{33} & \sigma_{34} \\ \sigma_{41} & \sigma_{42} & \sigma_{43} & \sigma_{44} \end{bmatrix}$$

The covariance propagation equation to the state-space representation of Eq. (1) satisfies the Equation (5).

$$A\sum + \sum A^T + B\Xi B^T = 0 \tag{5}$$

where assumed that the impulsive disturbance F_i having intensity Ξ satisfies the following Eq. (6) and (7).

$$E\left[\dot{F}_i(t)\dot{F}(t+\tau)\right] = \Xi\delta(t-\tau) \tag{6}$$

$$E\left[\dot{F}_i(t)\right] = 0 \quad (t \geq 0) \tag{7}$$

Therefore, the components of solution σ_{ij} can be obtained as follows.

$$\sigma_{11} = \frac{\xi_{11}}{\mu_{11}} + \frac{2k_s^2 k_u m_s m_u + k_s^3 m_u^2}{2c_s k_s k_u^3 m_s^2}$$

$$\sigma_{12} = 0$$

$$\sigma_{13} = \frac{\xi_{13}}{\mu_{13}} + \frac{k_s k_u m_s m_u + k_s^2 m_u^2}{2c_s k_u^3 m_s^2}$$

$$\sigma_{14} = -\frac{1}{2k_u m_s}$$

$$\sigma_{21} = \sigma_{12}$$

$$\sigma_{22} = \frac{c_s^2 k_u + k_s^2 m_s + 2k_s k_u m_s + k_u^2 m_s + k_s^2 m_u}{2c_s k_u^2 m_s^2}$$

$$\sigma_{23} = \frac{1}{2k_u m_s}$$

$$\sigma_{24} = \frac{c_s^2 k_u + k_s^2 m_s + k_s k_u m_s + k_s^2 m_u}{2c_s k_u^2 m_s^2}$$

$$\sigma_{31} = \sigma_{13}$$

$$\sigma_{32} = \sigma_{23}$$

$$\sigma_{33} = \frac{\xi_{33}}{\mu_{33}}$$

$$\sigma_{34} = 0$$

$$\sigma_{41} = \sigma_{14}$$

$$\sigma_{42} = \sigma_{24}$$

$$\sigma_{43} = \sigma_{34}$$

$$\sigma_{44} = \frac{c_s^2 k_u + k_s^2 m_s + k_s^2 m_u}{2c_s k_u^2 m_s^2}$$

where,

$$\xi_{11} = c_s^2 k_s k_u m_s + k_s^3 m_s^2 + 3k_s^2 k_u m_s^2 + k_u^3 m_s^2 + c_s^2 k_s k_u m_u + 2k_s^3 m_s m_u$$

$$\xi_{13} = c_s^2 k_u m_s + k_s^2 m_s^2 + 2k_s k_u m_s^2 + k_u^2 m_s^2 + c_s^2 k_u m_u + 2k_s^2 m_s m_u$$

$$\xi_{33} = c_s^2 k_u m_s + k_s^2 m_s^2 + k_s k_u m_s^2 + c_s^2 k_u m_u + 2k_s^2 m_s m_u + k_s^2 m_u^2$$

$$\mu_{11} = 2c_s k_s k_u^3 m_s^2$$

$$\mu_{13} = 2c_s k_u^3 m_s^2$$

$$\mu_{33} = \mu_{13}$$

From the above components of solution, σ_{ij}, the performance index J of Eq. (4) is represented in the following form.

$$J = \frac{k_s^2}{2c_s k_u m_s^2} + \frac{c_s}{2m_s^2 m_u} + \left(\frac{1}{2c_s k_s} + \frac{1}{2c_s k_u}\right) \times \rho \tag{8}$$

From a calculation minimizing the performance index of Eq. (8) about stiffness and damping, respectively, the optimal stiffness coefficient k_{op} and damping coefficient c_{op} of the dynamic absorbing system be obtained as Eq. (9) and (10).

$$k_{op} = \sqrt[3]{\frac{k_u m_s^2 \rho}{2}} \tag{9}$$

$$c_{op}=\sqrt{m_u\times\left\{\frac{k_s^2}{k_u}+\left(\frac{m_s^2}{k_s}+\frac{m_s^2}{k_u}\right)\times\rho\right\}} \tag{10}$$

3.2 Implementation of Isolation System

In the design of a newly developed HIFD, it is required both a performance improvement and a consideration of actual operating condition. The excessive transmitted force gives rise to an unwanted body motion and it is limited for the safety in a human being. In this section, on the basis of the previous results, the isolation system for the prototype HIFD has been implemented as shown in Fig. 4.

From the weighting factor ρ satisfying the constraint conditions of the trade-off between the transmitted force and the buffering displacement, the design parameters k_{op} and c_{op} of the isolation system for the prototype HIFD are decided from Eq. (9) and (10). The design parameter k_{op} and c_{op} were decided as 7040(N/m) and 2139 (Nsec/m), respectively.

On the basis of the parameters obtained from the optimal design process, a series of experiments for evaluating the practical performance of the isolation system was conducted for the prototype HIFD with the dynamic absorbing system in an operating condition as shown in Fig. 4.

The general dynamic absorbing system is capable of converting the kinetic energy at the instant of impact into heat using a flow resistance through narrow oil escape holes(orifices) between a piston chamber and an inner cylinder in a different way of the dynamic vibration absorber.[13,14]

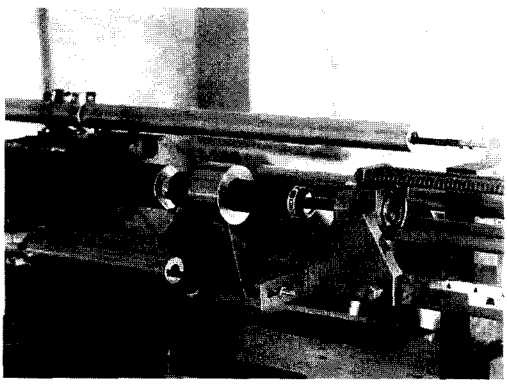

Fig. 4 Configuration of experimental setup

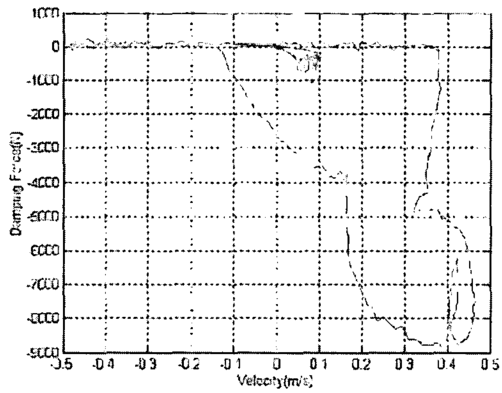

Fig. 5 Experimental force-velocity characteristic curves of dynamic absorber

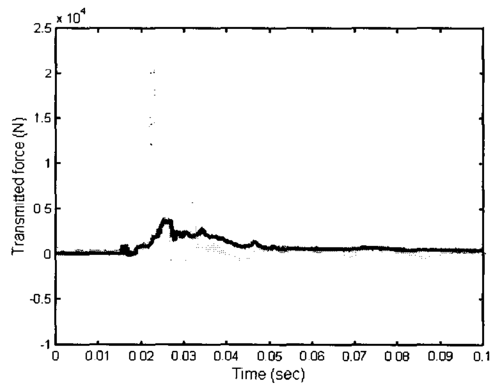

Fig. 6 Comparison of transmitted forces to mount(dotted : W/O, solid : W/ dynamic absorbing system)

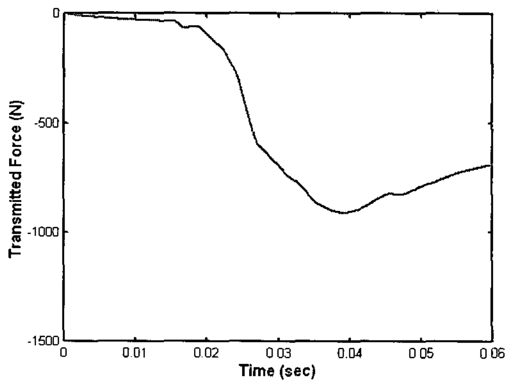

Fig. 7 Prediction of transmitted force to human body

The system includes the commercial dynamic absorber that is close to previous study because of difficulty to make the actualdynamic absorber within the launching time for prototype isolation system. As mentioned, it should have a different behavior between compression and extension operation. Considering the efficient condition of the prototype HIFD operation, the dynamic absorber is approximately undamped in extension for a quick return. Figure 5 shows the dynamic characteristics of the shock absorber equipped with the isolation system.

In order to analyze the test results precisely, the experimental setup was constructed by the specific measuring system under rigidly mounted condition.

Figure 6 shows the experimental results of the transmitted forces to the rigid mount for the prototype HIFD with and without the dynamic absorbing system, respectively. As shown in the results of Fig. 6, the maximum force of prototype HIFD with the dynamic absorbing system expressed about 83 % reduction rate from 22850(N) to 3940(N). Figure 7 depicts the prediction result of the transmitted force to a human body using the proved human structure model in section 2.

These results imply that the undesirable human behavior induced by the impulsive disturbance can be diminished. Therefore, the enhancement of operating performance with more stable operating condition can be achieved.

4. Conclusion

In this paper, the study on the isolation of the prototype HIFD is performed considering the dynamic motion of human body structure due to shock force input. Based on the experimental and analytical results between the human behaviors and the transmitted impulse occurring in an actual operating condition, the overall isolation system including the additional dynamic absorbing system is modeled. In order to determine the design parameters of the isolation system, the parameter optimization process is performed. By using the solutions satisfying the constraints, the performance of the implemented isolation system is evaluated by experiment using a devised test setup and the designed isolation system for the prototype HIFD results in the 83 % reduction rate of the maximum transmitted force compared to the original system.

Reference

(1) Babitsky, V. I. and Veprik, A. M., 1998, Universal Bumpered Vibration Isolator for Severe Environment, Journal of Sound and Vibration, Vol. 218 No. 2, pp. 269~292.

(2) Chehab, A. G. and Naggar, M. H., 2003, Design of Efficient Base Isolation for Hammers and Presses, Soil Dynamics and Earthquake Engineering, 23, pp. 127~141.

(3) Zhang, W., Matsuhisa, H., Honda, Y. and Sato, S., 1989, Vibration Reduction of a Railway Wheel by Cantilever-type Dynamic Absorbers, JSME International Journal, Vol. 32, No. 3, pp. 400~405.

(4) Hundal, M. S. and Fitzmorris, D. J., 1985, Response of a Symmetric Self-damped Pneumatic Shock Isolator to an Acceleration Pulse, Shock and Vibration Bulletin, Vol. 55, No. 1, pp. 139~154.

(5) Alanoly, J. and Sankar, S., 1988, Semi-active Force Generators for Shock Isolation, Journal of Sound and Vibration, Vol. 126, No. 1, pp. 145~156.

(6) Walsh, P. L. and Lamancusa, J. S., 1992, A Variable Stiffness Vibration Absorber for Minimization of Transient Vibrations, Journal of Sound and Vibration, Vol. 158, No. 2, pp. 195~211.

(7) Harris, C. M., 1997, Shock and Vibration Handbook, McGraw-Hill.

(8) Ryu, B. J., Kim, H. J., Choe, E. J., Lee, S. B., Kim, I. W. and Yang, H. S., 2003, Transmitted Force Estimation of Prototype HIF Device

Considering Human Behavior, International Congress and Exposition on Noise Control Engineering, pp. 4035~4042.

(9) Kim, H. J. and Choe, E. J., 2003, Analysis of Optimal Dynamic Absorbing System Considering Human Behavior Induced by Transmitted Force, International Journal of the Korean Society of Precision Engineering, Vol. 4, No. 6, pp. 38~43.

(10) Lee, S. B., Choe, E. J., Ryu, B. J. and Kim, H. J., 2004, Isolation of GOHIF Device Considering Human Body Structure, International Congress and Exposition on Noise Control Engineering, Paper No. 347.

(11) Fertis, D. G., 1995, Mechanical and Structural Vibrations, Wiely Inter. Science.

(12) 김효준 등, 2002, "고충격 발생기구의 완충 시스템 해석", 한국소음진동공학회논문집, 제 12 권, 제 5 호, pp. 389~396.

(13) Ram, Y. M. and Elhay, S., 1996, The Theory of a Multi-degree-of Freedom Dynamic Absorber, Journal of Sound and Vibration, Vol. 195, No. 4, pp. 607~615.

(14) Ma, S. and Semercigil, S. E., 1997, A Modified Passive Tuned Absorber for Secondary Structures, Journal of Sound and Vibration, Vol. 208, No. 3, pp. 349~366.