한국소음진동공학회 '93추계학술대회논문집 pp. 157 ~ 161.

실험적 모우드 해석을 이용한 방사광 가속기 건물의 진동제어

(VIBRATION CONTROL OF SYNCHROTRON LIGHT SOURCE BUILDING USING EXPERIMENTAL MODAL ANALYSIS)

박상규* °이홍기* 권형오*
(Sang Kyu Park, Hong Ki Lee and Hyeong Oh Kweon)

ABSTRACT

Optical devices and electronic equipments used in the laboratory of the synchrotron light source building of the accelerator have stringent vibration limits.

In order to control the vibration of the building structure and HVAC systems which are main vibration sources are evaluated using experimental modal analysis.

Double anti-vibration system is used for the HVAC system and results show that the double anti-vibration system reduces the vibrations of the building to acceptable levels.

INTRODUCTION

Synchrotron light source, which is a very sensitive equipment, requires environment with extremely limited vibrations and should be located in areas as far from external vibration sources as possible. However, in many cases, the synchrotron light source building is subjected to vibrations from rotating machinery such as pumps, fans and compressors. Specifically, HVAC systems are installed near slab on which the storage rings and experimental lines are placed. These contribute to the movement of magnets, which cause error fields that distort the electron orbit.[(1)(2)] To attenuate the vibration, emphasis should be given to the stability of the concrete slab structure and the control of forces generated by the HVAC systems.

For this purpose, impact tests have been performed on the structure of the building to determine thenstructuren properties such as natural frequency, damping and mode shapes. Unbalance forces generated by HVAC systems were also measured from the field tests. With these informations known, force response simulation has been carried out under different operating conditions of the HVAC systems to estimate the vibration response levels at the storage ring area. The procedure is summarized in Figure 1.

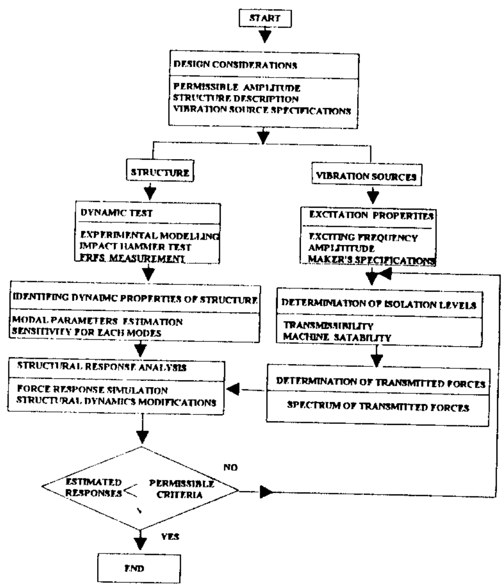


Figure 1. Procedure for vibration control of the structure

EXPERIMENTAL MODAL ANALYSIS

The complete dynamic description of structures is represented by modal parameters, which are natural frequency, damping and mode shapes. Basically, two experimental modal techniques are used for determining the modal parameters. The first is normal modal testing. The second is frequency response function method. In this paper, frequency reponse function(FRF) method is employed to determine the modal parameters.

* 유니슨산업(주) 부설 유니슨기술연구소, 정회원

Equation of Motion in the Laplace Domain[3][4]

For a multiple degree of freedom system, the elastic motion can be written by the following linear differential equations.

$$[M]\{\ddot{X}(t)\}+[C]\{\dot{X}(t)\}+[K]\{X(t)\}=\{F(t)\} \quad (1)$$

where,

[M]=the mass matrix
[C]=the damping matrix
[K]=the stiffness matrix
$\{X(t)\}$= displacement vector
$\{\dot{X}(t)\}$=velocity vector
$\{\ddot{X}(t)\}$=acceleration vector
$\{F(t)\}$=appied force vector

When the Lalpace transform is applied to the equation, the transformed equation becomes.

$$s^2[M]\{X(s)\} - s[M]\{X(0)\} - [M]\{\dot{X}(0)\} + s[C]\{X(s)\} - [C]\{X(0)\} + [K]\{X(s)\} = \{F(s)\} \quad (2)$$

where,

$\{X(s)\}$ = the transformed displacement vector
$\{F(s)\}$ = the transformed applied forces
$\{X(0)\}$ = initial displacement
$\{\dot{X}(0)\}$ = initial velocity
s = complex Laplace variable($\sigma + j\omega$)

When initial conditions are zero, the equation of motion(2) can be rewritten in the system matrix form.

$$[B(s)]\{X(s)\} = \{F(s)\} \quad (3)$$

where the system matrix [B(s)] is $s^2[M]+s[C]+[K]$ and contains all the information concerning the physical properties of structure.

The alternative of the system matrix, transfer function matrix, is defined as the inverse of the system matrix.

$$\{X(s)\} = [H(s)]\{F(s)\} \quad (4)$$

where [H(s)] is the transfer function matrix.

Modal Parameters

When no external forces are present, the equation(3) is reduced to the homogeneous equation. That is, modal parameters are the solutions of the following equation.

$$[B(s)]\{X(s)\} = \{0\} \quad (5)$$

The solution of equation(5) represents a complex eigenvalue problem. The complex eigenvalues are the modal frequencies and modal damping which cause the determinent of [B(s)] to be zero for nontrival solution.

$$\text{Determinant } |B(s)| = 0 \quad (6)$$

If there are N DOFs in equation(5), there are 2N eigenvalues, but these always occur in complex conjugate pairs. There are eigenvectors(u_K, u_K^*) corresponding to each of these eigenvalues(P_K, P_K^*), but these also occur in complex conjugate pairs. Hence, the eigensolution can be described as;

$$\begin{aligned} &P_K = \sigma_K + j\omega_K, \\ &P_K^* = \sigma_K - j\omega \\ &\qquad\qquad\qquad \text{for } k= 1 \text{ to } N \quad (7) \\ &[B(P_K)]\{u_K\} = \{0\} \\ &[B(P_K^*)]\{u_K^*\} = \{0\} \end{aligned}$$

where,

P_K, P_K^* = poles of the kth mode
σ_K = modal damping of the kth mode
ω_K = modal frequency of the kth mode
u_K, u_K^* = eigenvectors of the kth mode

Frequency Response Function

From the identity relationship of system matrix and transfer function matrix, transfer function matrix [H(s)] can be rewritten in partial fraction form shown in the following equation.

$$[H(s)] = \sum_{k=1}^{n} \left[\frac{[R_K]}{s - P_K} + \frac{[R_K^*]}{s - P_K^*} \right] \quad (8)$$

where $[R_K]$ = the residue matrix for kth mode. The transfer function matrix can also be written in terms of poles and mode shapes.

$$[H(s)] = \sum_{k=1}^{n} \left[\frac{A_K\{u_K\}\{u_K\}^T}{s - P_K} + \frac{A_K^*\{u_K^*\}\{u_K^*\}^T}{s - P_K^*} \right] \quad (9)$$

where A_K = the mode shape scaling constant for mode k.

If the transfer function matrix is limited to values of s having zero real part, this special case results in the frequency response function matrix.

$$[H(\omega)] = \sum_{k=1}^{n} \left[\frac{A_K\{u_K\}\{u_K\}^T}{j\omega - P_K} + \frac{A_K^*\{u_K^*\}\{u_K^*\}^T}{j\omega - P_K^*} \right] \quad (10)$$

MODAL TESTING PROCEDURE

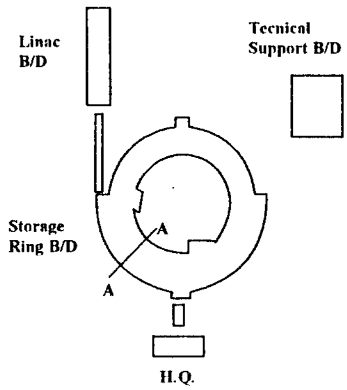
Test Structure

Twenty four HVAC systems will be placed in the utility area of the storage ring building with steel-reinforced concrete. Only a part of the utility area will be considered in this study to simplify the problem. Figure 2(a) shows the plane view of the storage ring building and figure 2(b) is a cross section view in which a HVAC with a double anti-vibration system is shown on the utility area.

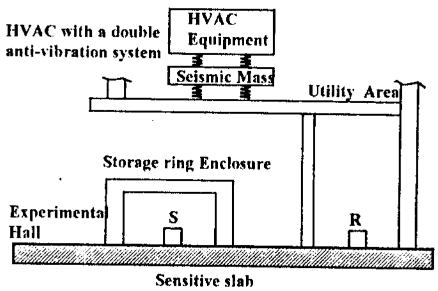
Measurement Locations and Conditions

Impact tests have been carried out on the utility area to measure the FRFs of the structure. Figure 3 shows the instrumentation setup. A large impact hammer was devised and response signals were fed into a dual channel spectrum analyzer and also recorded on a portable tape recorder for future use. The recorded FRF signals were analyzed using STAR software[4]. The test area is modelled with 231 nodes as shown in figure 4. Two HVAC systems are located at the node 25 and

165, while an accelerometer is attatched to the location of the node 81.



(a) Plane view



(b) Cross section(A-A)

Figure 2. Configuration of the test structure

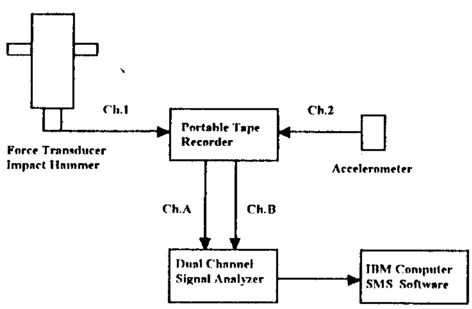


Figure 3. Instrumentation setup

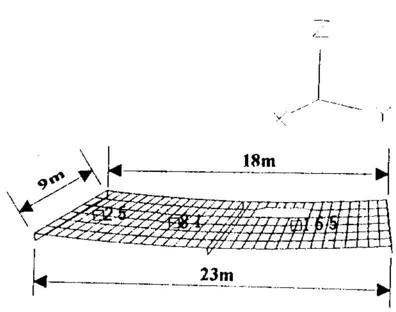
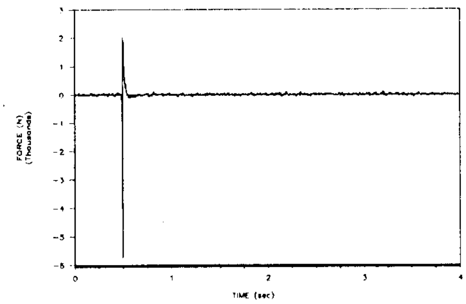


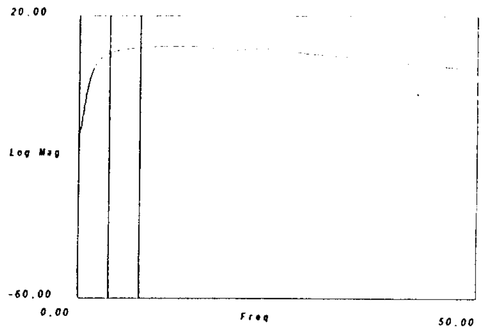
Figure 4. Modelling of utility area

Excitation

Time signals and the averaged spectrum resulting from impact show that the excitation energy level is adequate within the frequency range of interest(figure 5).



(a) Time signal of impact



(b) Spectrum

Figure 5. Time signal and Averaged spectrum resulting from impact hammer

MODAL ANALYSIS AND RESULTS

Procedure for extracting the modal parameters from the measured FRFs was accomplished by the STAR package from SMS[4]. The modal parameter estimation was done by curve fitting. Figure 6 shows an example of the measured and curve-fitted FRFs. For the test structure, Table 1 shows the frequency and damping for the first two modes of vibration. The first two mode shapes are shown in figure 7.

Table 1. Frequency and damping for the first two modes

Mode No.	Frequency (Hz)	Damping ratio(%)	Remark
1	5.6	0.65	Bending about X-axis
2	11.3	1.21	Bending about Y-axis

FORCE RESPONSE SIMULATION(FRS)

The major source of vibrations in the SLS building is HVAC equipments located in the utility area. Unbalance force generated by the HVAC equipment was measured from the field

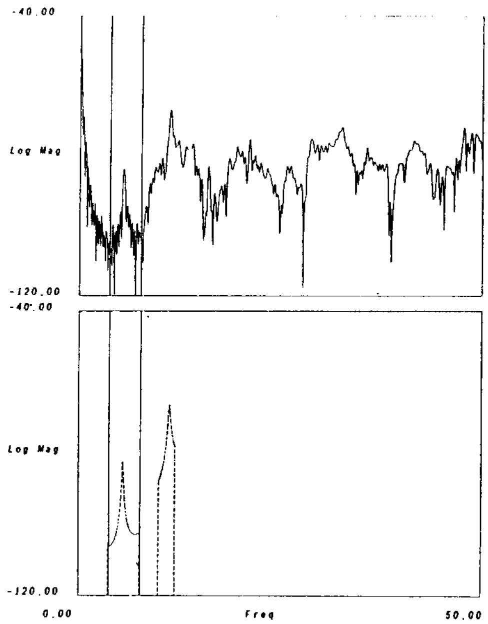


Figure 6. Plot of the measured FRF and curve-fitted FRF

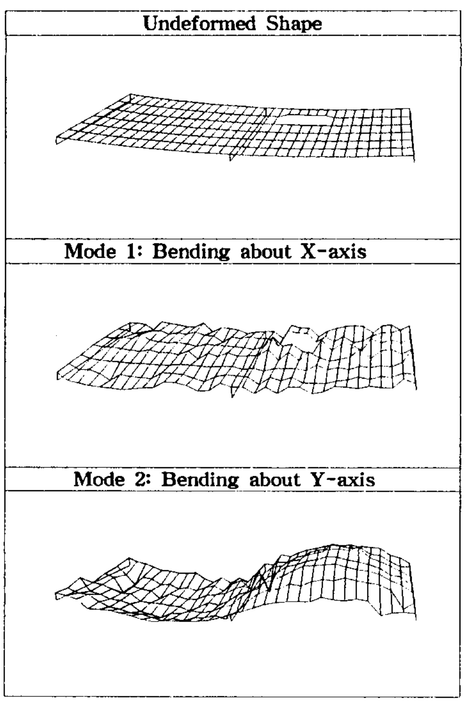


Figure 7. The first two mode shapes

tests and was about 750N (23 Hz) over a frequency range from 1 to 50Hz[5].

Vibration responses due to the two HVAC systems under different operating conditions are simulated at the utility area and the storage ring floor using FRS technique of STAR software. The different operating conditions are explained in table 2. Case 1 indicates the condition under which the HVAC system at the node 25 operates, while that at the node 165 does not. Case 2 is the opposite condition. For the case 3 and 4, both HVAC systems operates at the same time, but the phase is different. In fact, it is very difficult to see if both systems are in phase or out of phase. Case 3 and 4 simulate the worst and the best conditions, respectively.

The estimated vibration responses without and with a double anti-vibration system for the HVAC are tabulated in the tables 3 and 4, respectively. In the tables, R and S indicate the two locations on the sensitive slab as shown in figure 2. Simulation results indicate that the double anti-vibration system reduces the vibration level to 1% of response for the system w/o isolator. This means that the double anti-vibration system is well designed and satisfies the environmental vibration criteria provided by customers(Table 5).

Table 2. Operating conditions for FRS

No.	HVAC at 25	HVAC at 165	Remark
case 1	on	off	
case 2	off	on	
case 3	on	on	in phase
case 4	on	on	out of phase

Table 3. Estimated vibrations w/o double anti-vibration system

(unit: mm/sec^2)

No.	Utility area	Sensitive slab	
		R	S
case 1	20-100	0.018	0.003
case 2	20-150	0.027	0.003
case 3	30-200	0.036	0.004
case 4	30-200	0.036	0.004

VIBRATION MEASUREMENT AT SITE

To check the effectiveness of the system and the safety of the building, vibration responses were measured when the HVAC equipments with a double anti-vibration system operated on the utility area. Results, shown in figure 8, indicates that the vibration level(at

Table 4. Estimated vibrations with double anti-vibration system

(unit: mm/sec^2)

No.	Utility area	Sensitive slab	
		R	S
case 1	0.2-1	0.18×10^{-3}	0.03×10^{-3}
case 2	0.2-1.5	0.27×10^{-3}	0.03×10^{-3}
case 3	0.3-2	0.36×10^{-3}	0.04×10^{-3}
case 4	0.3-2	0.36×10^{-3}	0.04×10^{-3}

Table 5. Vibration criterion values

Utility area	Sensitive slab	
	R	S
6mm/sec^2	0.01μm (4-20 Hz) 1.6 mm/sec^2 (above 20 Hz)	

node 25) is about 2.8 mm/sec^2 at 23 Hz. Table 6 shows the measured vibrations of the test structure with a double anti-vibration system.

Comparing these with the estimated results shown in the table 4, the measured vibration level is higher. However, the measured vibration is within acceptable ranges since the double anti-vibration is conservatively designed. It is thought that the main cause of the error results from differences of the structure properties or HVAC unbalance forces.

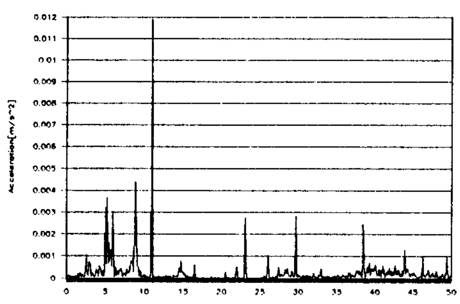


Figure 8. Measured vibration at DOF 25

Table 6. Measured vibrations with double anti-vibration system

(unit: mm/sec^2)

Utility area	Sensitive slab	
	R	S
1-3	below 0.02	

CONCLUSIONS

The procedure shows the application of experimental modal analysis and force response simulation technique to the actual engineering problem. The analysis produces a mathematical model of the test structure and force response simulation can be carried out to design an effective anti-vibration system. Results from this procedure can be utilized to control unacceptable vibration responses.

REFERENCES

(1) J. B. Godel, Stability and Vibration Control in Synchrotron Light Source Buildings, Brooklaven National Laboratory, Associated Uninersties, Inc., NSLS, Upton, NY 11973.

(2) R.O. Hettel, Beam and Steering at the Stanford Synchrotron Radiation Laboratory, IEEE Transactions on Nuclear Science, Vol. NS-30, No.4, August 1983.

(3) D. J. Ewins, Modal Testing: Theory and Practice, Reserch Studies Press Ltd., Letchworth, Hertfordshire, England, 984.

(4) SMS STAR Theory and Applications, Issue A:25, January 1990.

(5) H. K. Lee, Experimental Behavior Analysis of Double Anti-Vibration System, KSNVE, Vol.2, No.4, December 1992.