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〈**Original**〉

Vibration Estimation of Synchrotron Light Source Building Using Experimental Modal Analysis

실험적 모드해석을 이용한 방사광 가속기 건물의 진동평가

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ABSTRACT

Synchrotron light source building of the accelerator has stringent vibration limits since the performance of the optical devices and electronic equipments in the laboratory is strongly influenced by the vibrations of the building. In this study, vibrations of the synchrotron light source building are estimated using experimental modal analysis and force response simulation technique. Dynamic properties of the building are identified from the modal parameters and vibration responses are predicted from the force response simulation. A double anti vibration system is designed and applied to the HVAC equipments and it has been shown that the measured vibrations of the building with the double anti vibration system satisfy the vibration criteria.

요 약

냉난방용 회전기기에 의한 방사광가속기 건물의 진동은 내부에 설치되어 있는 광학장치나 전자장비의 성능에 영향을 미치기 때문에 건설단계에서 부터 건물진동에 대한 평가가 매우 중요하다. 본 연구에서는 방사광가속기 건물의 진동을 평가하기 위해서 실험적 모우드해석법을 이용하여 냉난방기기가 위치한 바닥의 동적특성을 구하고, 하중응답 모사법을 통해 진동응답치를 계산하였다. 평가결과, 건물 바닥의 진동치는 기준치를 상회하고 있으며 이를 해결하기 위해 이중방진 시스템을 설계 제작하여 냉난방기기와 바닥 사이에 설치함으로써 바닥의 진동을 허용한도 이내로 감소시켰다.

Nomenclature

A_K : k^{th} mode shape scaling constant
[C] : Damping matrix
{F(s)} : Transformed applied forces
{F(t)} : Applied force vector in time domain
[H(ω)] : Frequency response function matrix
[K] : Stiffness matrix
[M] : Mass matrix
P_K, P_K^* : Poles of the kth mode
$[R_K]$: Residue matrix for the kth mode.
s : Complex Laplace variable.
u_K, u_K^* : Eigenvectors of the kth mode.

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$\{X(0)\}$: Initial displacement vector
$\{\dot{X}(0)\}$: Initial velocity vector
$\{X(s)\}$: Transformed displacement response
$\{X(t)\}$: Displacement vector in time domain
$\{\dot{X}(t)\}$: Velocity vector in time domain
$\{\ddot{X}(t)\}$: Acceleration vector in time domain
σ_K : Modal damping of the kth mode
ω_K : Modal frequency of the kth mode

1. 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, compressors and HVAC equipments. Specifically, HVAC equipments are installed on the utility floor under which storage rings and experimental lines are placed. Vibrations of the utility floor contribute to the movement of magnets, which cause error fields that distort the electron orbit[1,2]. Hence, it is necessary to predict the vibration to prevent the excessive vibration responses of the building structure as well as sen-

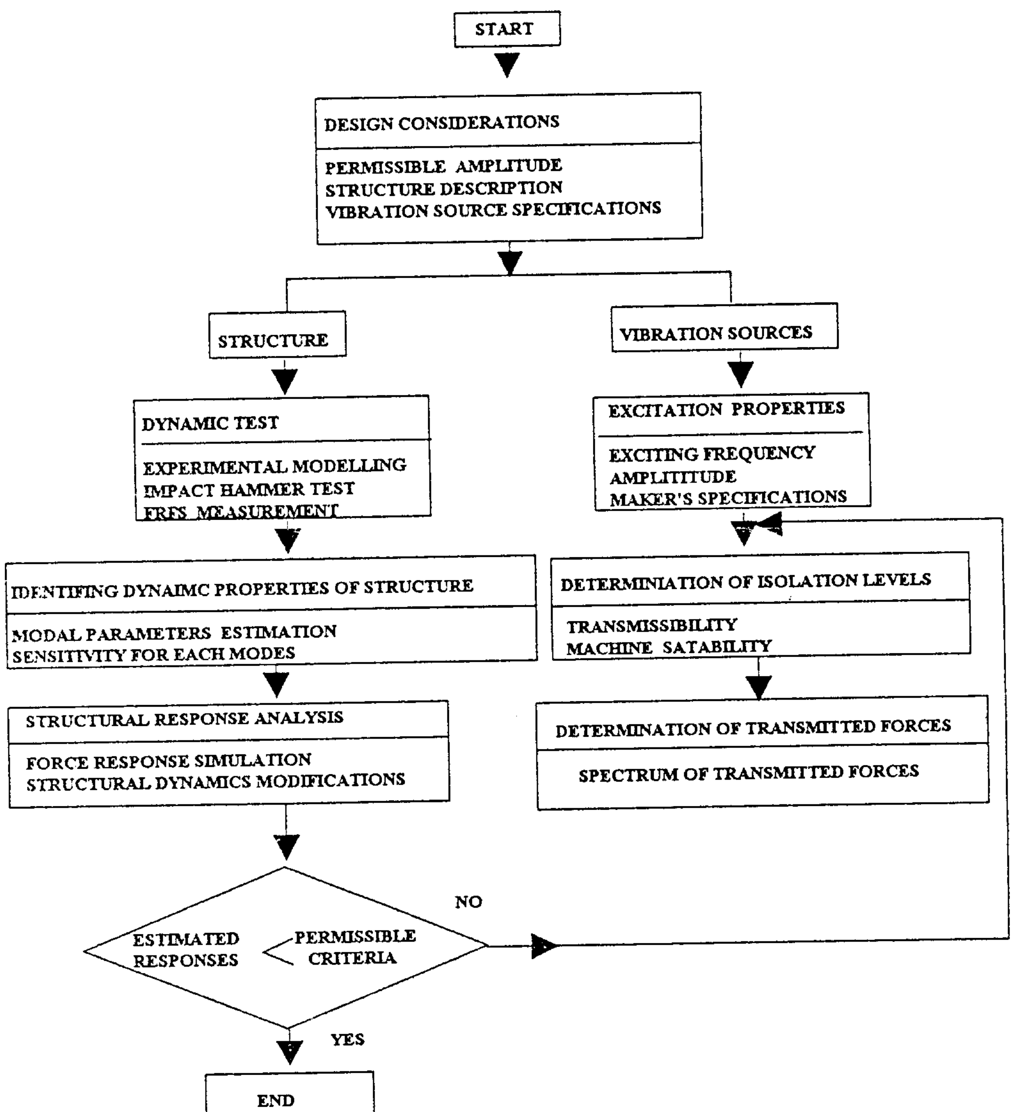


Fig. 1 Procedure for vibration estimation of the structure

sitive equipments. If the predicted values are higher than the vibration criteria, proper method must be employed to attenuate the vibration.

In this study, vibrations of the synchrotron light source building are estimated by employing a procedure, shown in Fig. 1. Experimental modal analysis involves the extraction of modal parameters by dynamically testing the actual structure. Dynamic properties of the structure can be identified from the modal parameters. Force response simulations are carried out to predict the response of the structure to input force under different operating conditions of the HVAC equipments.

2. Experimental Modal Analysis

The complete dynamic description of the structure is represented by modal parameters, such as natural frequency, damping and mode shapes. Basically, two experimental modal techniques are used for determining the modal parameters. The first is normal modal testing method. The second is frequency response function method. In this paper, parameter estimation methods using frequency response function is used to determine the modal parameters.

2.1 Equation of Motion in the Laplace Domain[1,2]

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
$\{F(t)\}$=applied force vector

When the Laplace transform is applied to linear equation of motion (1), 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 (2) can be rewritten in the system matrix form.

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

where 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 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.

2.2 Modal Parameters

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

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

The solution of equation (5) represents a complex eigenvalue problem. The complex eigenvalues are the modal frequencies and modal dampings 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. Eigenvectors$(u_K, u_K{}^*)$ correspond to these eigenvalues $(P_K, P_K{}^*)$, but these also occur in complex conjugate pairs. Hence, the eigensolutions can be described as.

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

where

$P_K, P_K{}^*$=poles of the kth mode

σ_K=modal damping of the kth mode
ωk=modal frequency of the kth mode
u_K, $u_K{}^*$=eigenvectors of the kth mode.

2.3 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.

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

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

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

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.

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

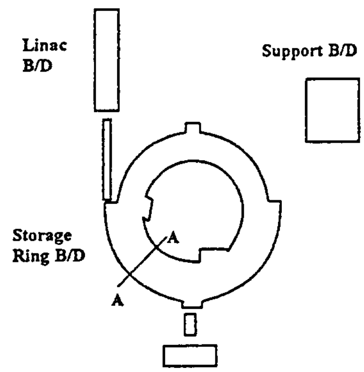
3. Modal Testing Procedure

3.1 Test Structure

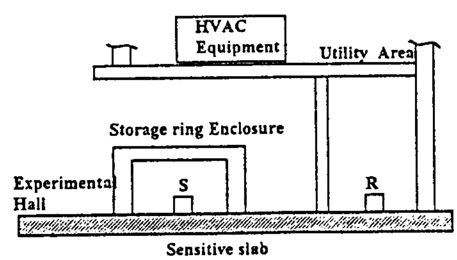
Figure 2(a) shows the plane view of the storage ring building and Fig. 2(b) is a cross sectional view. The utility area of the storage ring building consists of twelve floors which are simply supported by twelve walls. Two HVAC systems are placed on each floor. For the experimental modal testing, only one floor is considered since other floors have same size and are made of same materials. Attention is focused on the vibrations of the utility floor because small vibration of the sensitive equipments can be expected once the vibration levels of the utility floor satisfy the vibration requirements.

3.2 Measurement Locations and Condition

Impact tests have been carried out on the utility floor to measure the frequency response functions (FRF) of the structure. Figure 3 shows the instrumentation setup. A large impact hammer was devised and response signals are fed into a dual channel spectrum analyzer and also recorded on a portable tape recorder for future use. The recorded FRF signals are analyzed using STAR software[4]. The test floor is modelled with 231 nodes as shown in Fig. 4. Two HVAC systems are located at the node 25 and 165, while an accelerometer is attached to the location of the node 81.



(a) Plane view



(b) Cross section(A-A)

Fig. 2 Configurations of the test structure

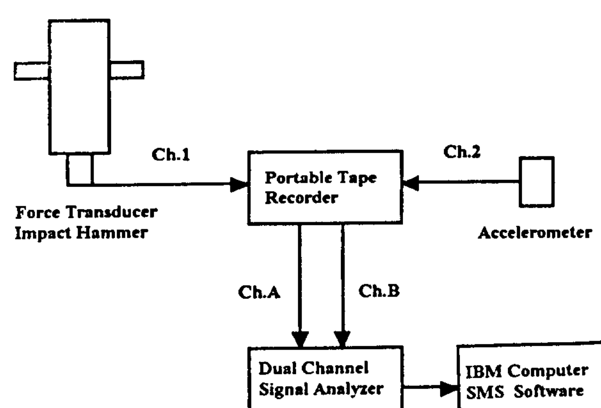


Fig. 3 Instrumentations setup

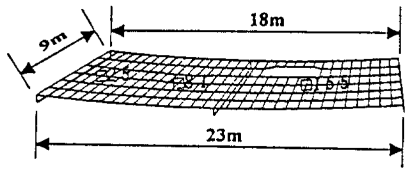


Fig. 4 Modelling of the utility floor

4. Modal Analysis and Results

Procedure for extracting the modal parameters from the measured FRFs is accomplished by the STAR package from SMS. The modal parameter estimation is done by curve fitting. Figure 5 shows an example for the measured and curve fitted FRFs. For the test structure, Table 1 shows the frequencies and dampings for the first two modes of vibration. The first two mode shapes are shown in Fig. 6.

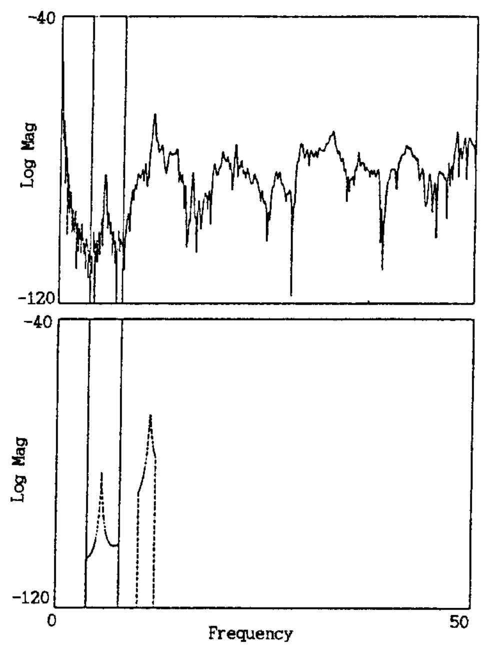


Fig. 5 Measured and curve fitted FRFs

Table 1 Modal parameters

Mode No.	Frequency [Hz]	Damping ratio [%]
1	5.6	0.65
2	11.3	1.21

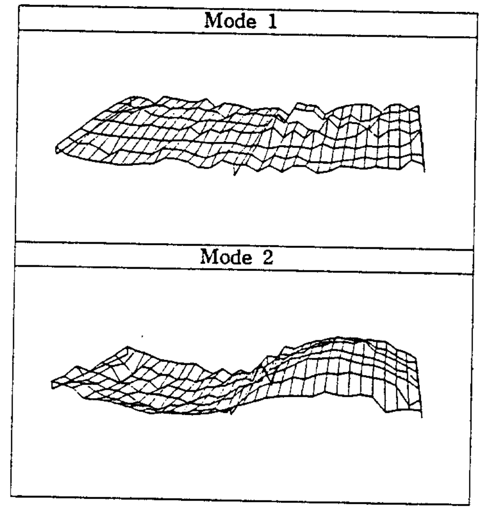


Fig. 6 The first two mode shapes

5. Force Response Simulation

Vibrations of the synchrotron light source building come from the HVAC equipments located in the utility floor. The rotational frequency of the HVAC equipments is 23 Hz. The Vibration responses of the utility floor due to the HVAC systems are simulated using force response simulation technique.

The unbalance forces generated by the HVAC equipments were measured from the field test and were below 750N at 23 Hz. Table 2 displays the operating conditions of the HVAC equipments. 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, both HVAC systems operate at the same time. The estimated vibration responses of the utility floor are tabulated in Table 3. It has been illustrated that the vibrations of the utility floor

Table 2 Operating conditions for the HVAC

No.	HVAC at the node 25	HVAC at the node 165
Case 1	ON	OFF
Case 2	OFF	ON
Case 3	ON	ON

Table 3 Estimated vibrations w/o double anti vibration system (mm/sec^2)

No.	Utility floor
Case 1	20-100
Case 2	20-150
Case 3	30-200

Table 4 Vibration criteria (RMS)

Utility floor	Sensitive slab
6 mm/sec^2	4~20 Hz ; 0.01μm
	Above 20 Hz ; 0.1μm or 1.6 mm/sec^2

exceed the vibration criteria for the building structure, shown in Table 4.

6. Double Anti Vibration System

To reduce the vibrations of the utility floor, a double anti vibration system, shown in Fig. 7 is designed. It consists of a HVAC equipment, a seismic mass, and eight springs. The equations of motion of the double anti vibration system can be given as

$$M_1\ddot{Z}_1+K_1Z_1+K_2(Z_1-Z_2)=0$$
$$M_2\ddot{Z}_2+K_1(Z_1-Z_2)=F_0\sin\omega t \quad (11)$$

where M_1 and M_2 are the masses of the machine and the additional mass, K_1 and K_2 are the spring constants, and Z_1 and Z_2 are the displacements of the masses M_1 and M_2, respectively. F_0 is the driving force and ω is the forcing frequency. The general equations of motion of the two masses can be written as

$$Z_1(t)=\frac{F_0\omega_2^2/M_1}{f(\omega)}\sin\omega t \quad (12)$$

$$Z_2(t)=\frac{(\omega_1^2/M_2+\omega_2^2/M_1-\omega^2/M_2)F_0}{f(\omega)}\sin\omega t \quad (13)$$

where

$$\omega_1^2=\frac{K_1}{M_1},\quad \omega_2^2=\frac{K_2}{M_2} \quad (14)$$

$$f(\omega)=\omega^4-\left[\omega_2^2\left(1+\frac{M_2}{M_1}\right)+\omega_1^2\right]\omega^2+\omega_1^2\omega_2^2 \quad (15)$$

Transmissibility, which is the ratio of the transmitted force to the driving force, is given as

$$\text{Transmissibility}=\frac{K_1Z_1}{F_0}=\frac{\omega_1^2\omega_2^2}{f(\omega)} \quad (16)$$

The double anti vibration system for the HVAC equipments is desinged for the transmissibility to be 0.01. Table 5 and 6 show the specification of the double anti vibration system and the estimated vibrations obtained from the force response simulation, respectively. As shown, estimated vibration in Table 6 are 10% of those in Table 3. This is due to the fact that the transmissibility is 0.1.

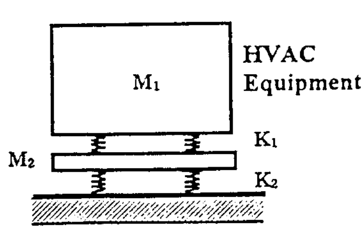


Fig. 7 Double anti vibration system

Table 5 Specification of double anti vibration system

Item	Description
M_1	4170 kg
M_2	3250 kg
K_1	3270 kg f/cm
K_2	750 kg f/cm

Table 6 Estimated vibrations with double anti vibration system (mm/sec^2)

No.	Utility floor
Case 1	0.2-1
Case 2	0.2-1.5
Case 3	0.3-2

7. Vibration Measurement at Site

After building construction and machine installation were completed, vibration responses of the synchrotron light source building were measured with all machines operated. The machines include exhaust fans and twenty four HVAC equipments with double anti vibration systems. The exhaust fans were installed on the ceiling of the building. Figure 8 shows the measurement locations in the synchrotron light source building. No. 1 and 2 indicate the locations on the top of a HVAC system and utility floor, respectively. Figures 9 through 14 illustrate the frequency spectrum of the vibrations and Table 7 summarizes the results. In the table, U and S mean the utility floor and storage ring, respectively. Blanks indicate no peak responses at the corresponding frequencies. Vibrations of the utility floors and storage ring are in the ranges of vibration criteria. Comparison of the responses on No. 1 with those on No. 2 indicates that the double anti vibration system reduces the vibrations of the utility floor to about 10% of the HVAC vibrations. This means that the double anti vibration system is very effective to reduce the vibrations of the HVAC equipment. Note that there are marked changes in the response of utility floors and the storage ring areas at the frequency of 19 Hz, as shown in Figs. 11, 13 and 14. This

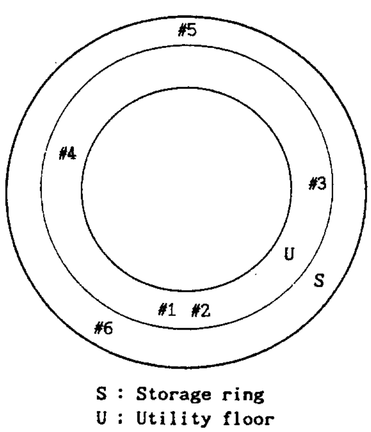


Fig. 8 Locations of vibration measurement

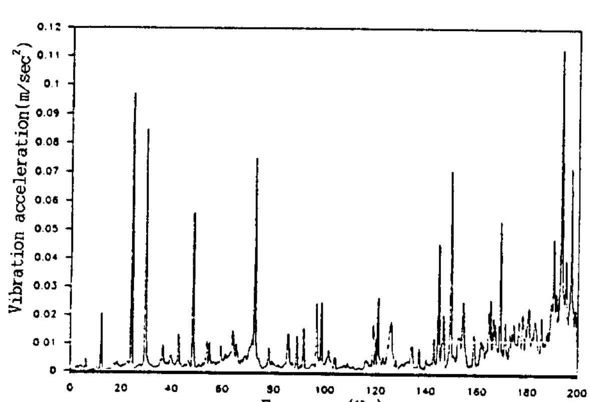


Fig. 9 Measured vibrations of HVAC with double anti vibration system at the No. 1

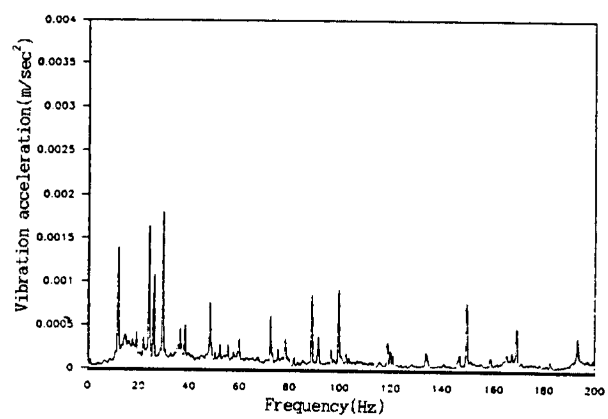


Fig. 10 Measured vibrations of utility floor with double anti vibration system at the No. 2

Table 7 Summary of measures vibrations with double anti vibration system

Location		Amp	Peak frequency(Hz)					
			12	19	23	27	30	50
U	1	a	21.0		100.0		85.0	55.0
	2	a	1.4	0.4	1.6		1.8	0.8
	3	a	0.7	3.7	2.4		1.0	0.7
	4	a	1.2	0.3	2.3		0.4	1.0
S	5	a		0.1	0.03		0.06	0.15
		d		0.007	0.001			
	6	a		0.1	0.01	0.01	0.04	0.03
		d		0.007				

S : Storage ring
U : Utility floor
a : Acceleration (mm/sec²)
d : Displacement (μm)

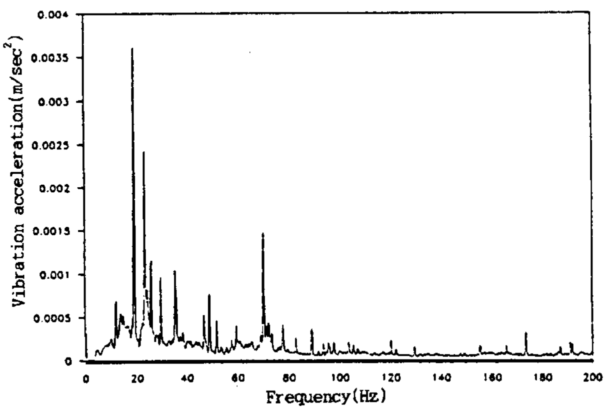


Fig. 11 Measured vibrations of utility floor with double anti vibration system at the No. 3

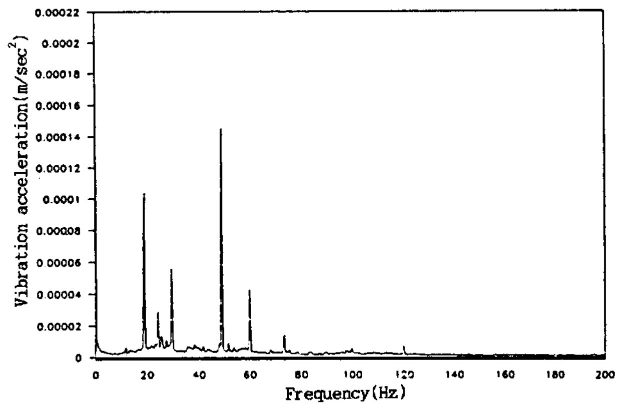


Fig. 13 Measured vibrations of storage ring with double anti vibration system at the No. 5

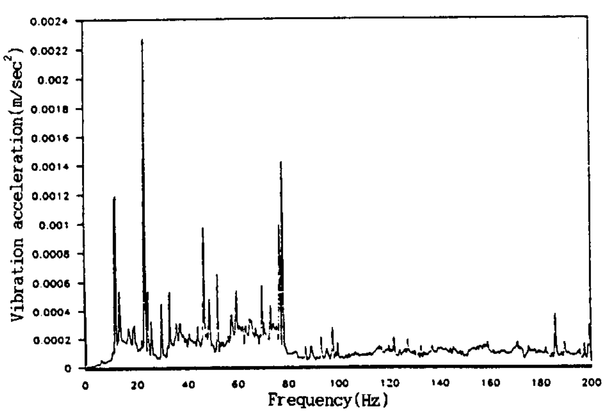


Fig. 12 Measured vibrations of utility floor with double anti vibration system at the No. 4

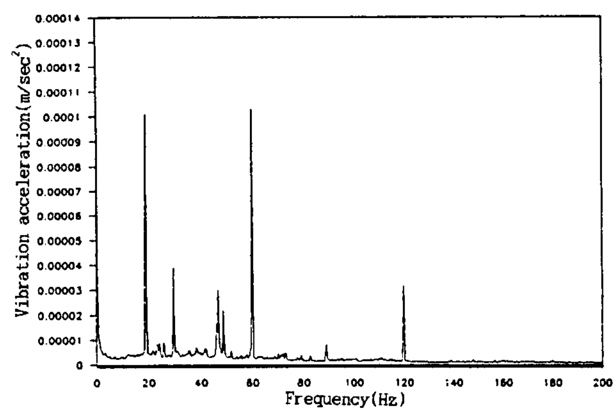


Fig. 14 Measured vibrations of storage ring with double anti vibration system at the No. 6

corresponds to the rotational frequency of the exhaust fans. This would suggest that the malfunction of the exhaust fans can cause sudden increase of the vibration levels of the utility floors and the storage ring areas.

8. Conclusions

Vibrations of the synchrotron light source building are estimated using experimental modal analysis and force response simulation technique. Estimation results show that the vibration levels of the utility floor of the building are higher than the vibration criteria. A double anti vibration system is designed and applied to the HVAC equipments to reduce the vibration amplitudes of the utility floor. As a result, following conclusions are obtained.

(1) It is shown that the estimation procedure can be effectively used to evaluate the vibrations of the structure.

(2) Double anti vibration system can be employed to reduce the vibration levels of the utility floor.

(3) Mesured vibrations of the building with double anti vibration system satisfiy the vibration criteria of the synchrotron light source building.

(4) Care should be taken when the exhaust fans are operated because they can be a possible source of the vibrations of the structure.

References

(1) J.B. Godel, 1973, Stability and Vibration Control

haven National Laboratory, Associated Univer-
in Synchrotron Light Source Building, Brook-
sities, Inc. NSLS, Upton, NY 11973.

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

(3) D.J. Ewins, 1984, Modal Testing : Theory and Practice, Research Studies Press Ltd., Letchworth, Hartfordshire, England.

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

(5) H.K. Lee, S.E. Rhee, 1992, "Experimental Behaviour Analysis of Double Anti Vibration System," Journal of KSNVE, Vol. 2, No. 4, pp. 281 ~292.