장비기초설계의 강성 및 진동저감에 관한 연구
Stiffness,RigidityandVibrationPreventioninPrecisionMachine FoundationDesign

박옥정* 김진호** 전한준***
Park, Ok-Jeong Kim, Jin-Ho Jeon, Han-Jun

국문요약

대형시험장비가 설치되는 실험실의 계획에서 장비의 진동은 간과할 수 없는 고려사항이다. 따라서 시험장비를 지지하는 구조물과 기초의 설계시 시험장비의 운행 중 발생하는 공진을 피할 수 있는 구조적 체계의 진동특성에 대한 적절한 평가가 요구된다. 본 논문은 동적특성을 얻기 위해 바닥구조물의 FE 모델링에 관하여 기술하였다. 또한 진동저감을 위한 시험장비기초의 설계를 위해 tuning, 진동기준, 방진시스템을 검토하였다. 시험장비의 진동을 줄이는 최선의 방법은 low tuning 이었으며 이의 구현을 위해 방진스프링과 함께 단단한 콘크리트 블록위에 시험장비를 설치하였다. 총체적인 방진시스템의 구조적 진동특성은 이동성, 힘, 속도 스펙트라를 이용해 표현되었다. 전달과 지점이동 FRF의 비를 시뮬레이션을 통해 비교함으로서 바닥 슬래브의 진동전달 정도가 관찰되었다.

1.INTRODUCTION

The machine-induced vibration could cause fatigue and cracking in beams, floors, walls, and other structural members. The load-bearing capability of a structural member may thus be compromised. Vibration isolation is a well-known and effective method for reducing vibration from machinery. Case studies have shown the installation of machinery on vibration isolators helps protect nearby precision machinery and equipment from incoming vibration, further improving the working environment.

----

* 한국철도기술연구원 책임연구원, 정회원
** 한국철도기술연구원 선임연구원, 정회원
*** 한국철도기술연구원 주임기술원, 정회원

This paper is concerned with the vibration performance of one such floor in a new test facilities building at Korea Railroad Research Institute. As this building is being developed to accommodate equipment that is highly sensitive to vibration, it is necessary to predict likely levels of vibration response at the design stage and compare those with the limits specified by the equipment manufacturers. As a result, extensive FE modeling is performed prior to construction and the configuration of the floor is developed to satisfy those vibration constraints.

This study reviews several forms of vibration limits and predicts the dynamic properties of a resiliently mounted, through the isolators are computed due to stationary dynamic loads. In addition, it presents the results of a controlled evaluation of vibrations .

2.FEMODELING

Before the modal testing is performed, an FE model of the floor is developed using the ANSYS code. Its main purpose is to serve as the basis for manual FE model updating to match the analytically calculated modal properties with their experimental counterparts. Fairly detailed modeling of the entire floor surface is performed, resulting in the 3D FE model shown in Figure 1.

In this FE model, the supporting columns above and below the slab are introduced to realistically model the boundary conditions of the floor. ANSYS SHELL63 elements, which have both bending and membrane capabilities, are chosen to model the concrete slab. Standard BEAM44 elements are used to model the steel beams and columns, making use of the offset option to model the offset of the centroid of the down stand edge beams from the centroid of the slab, in the cases where this is applicable.

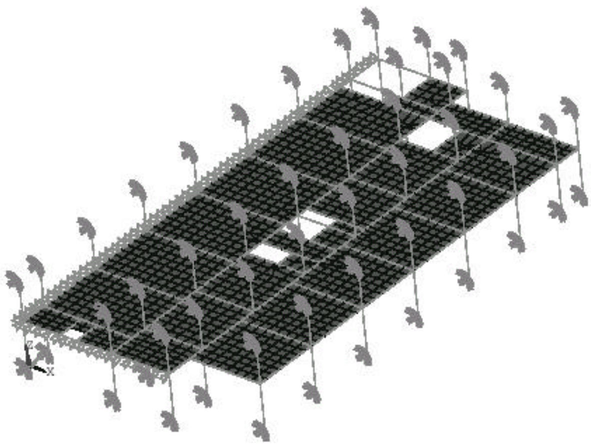

Figure 1: Pre-test FE model

The bending stiffness and material densityof the SHELL63 elements in the direction of spanning of the precast beams have to be adjusted to take account of the openings in the hollow-core precast elements. During the modeling a number of fundamental assumptions were made, namely that:

▷ the dynamic Young's modulus of concrete with a design strength of 55 MPa (grade C55) concrete is 38 GPa, its density is 2400 kg/m^3 and its Poisson's ratio is 0.2,
▷ the connection between the beams and columns is rigid, which is justifiable for the small

displacements associated with walking-induced excitation, and

▷ restraints are applied to the model by allowing no rotation and no displacement of the columns at ground and second floor (Figure 1).

A modal analysis is performed to determine first 50 modes of vibration. The natural frequencies of the first five modes are of interest as they are lower than 20 Hz. They are presented in Table 1 with their modal masses, determined using unity normalized mode shapes. The first main mode of vibration is a single curvature mode in the wider bay, occurring at 13.65 Hz. The second, third and fourth modes of vibration also occurred in the wider bay, with increasing curvature. The fifth main mode of vibration occurred in the 8 m wide bay as a single-curvature mode. Significant modes across the structure do not occur until the seventh analytical mode of vibration, at frequencies in excess of 20Hz.

Table 1. Natural frequencies and modal masses from pre-test FE model

Mode	Frequency (Hz)	Modal Mass (Tonnes)
1	13.65	53.6
2	14.76	64.3
3	15.95	59.7
4	17.82	72.1
5	20.00	58.1

3.TUNINGANDVIBRATIONCRITERIAFORSTRUCTURES

The primary goal of machinery isolation is to attenuate the vibration energy flow from a machine to adjoining areas. This is accomplished most effectively by "tuning." Tuning is the process of separating the machine operating frequency from any structural frequencies. The frequency separation should satisfy the "octave rule" as a minimum requirement. A one-octave separation means that the higher frequency is twice the value of the lower frequency.

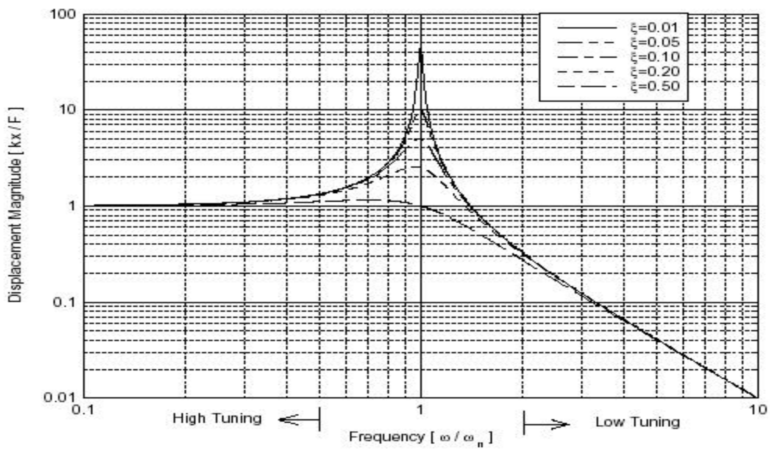

Figure 2. Steady-state response of a single-DOF system

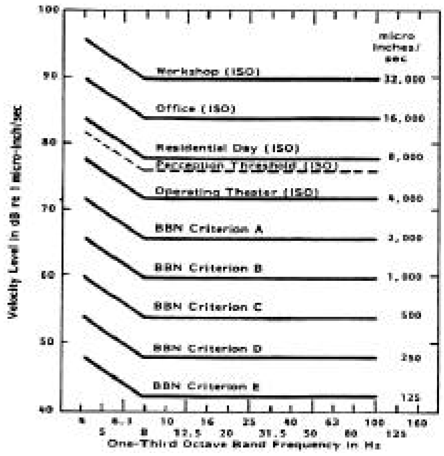

Figure 3. Floor criteria-allowable vibration

Figure 2 shows steady-state response of a single-DOF system subjected to an applied sinusoidal force. Low tuning is achieved when the operational forcing frequency is at least twice the value of the isolated natural frequency, $_{n}$. In other words, the isolated frequency is at least one octave below the operating frequency.

Figure 3 shows the family of vibration criteria, which is used in this study. These criteria are specified in terms of velocity amplitude, as it has been shown that vibration sensitivity is, in general, a function of vibration amplitude within a specific frequency range rather than being a function of stiffness, frequency, or floor type alone.

4.ISOLATIONSYSTEM

Isolation systems for reducing machinery vibration typically use either a resilient material or steel coil springs as isolation material. Resilient material type isolation is available in simple sheet forms that offer precision leveling and hydraulic level-assist capabilities. Steel coil spring isolators are available with either frictional or viscous fluid type damping systems. Resilient material and steel coil spring type isolators are the two most common types used today. Each type has different features and advantages.

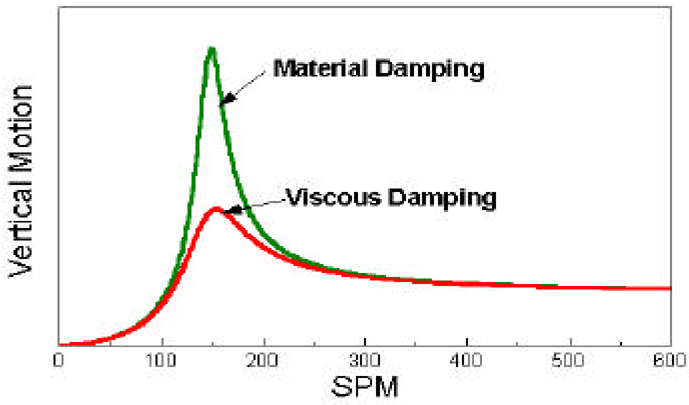

Figure 4. Relative motion comparison between material and viscous damping

Isolators or machinery mounts are used to install a wide variety of industrial machinery. They offer excellent vibration isolation and press stability and are usually installed directly to the press foot without any modifications. They offer advantages over Layered Elastomeric Systems in stability, leveling, alignment, and ease of installation. They also have a swiveling feature that keeps the machine foot properly supported and the isolator's elastomer evenly loaded when the bottom of the machine foot is not parallel with the supporting surface. This distributes load to the foundation and foot evenly. Static plus dynamic stresses are usually less than 4.8 MPa. Typical isolation performance for press applications is 50 ~ 95 percent versus anchoring to concrete.
Spring isolators offer excellent shock and vibration isolation due to their low stiffness and dynamic natural frequency (4 ~ 7 Hz). Isolation performance is around the 80 percent range for hammers and up to 98 percent for presses. Spring isolators have either viscous or frictional damping systems to provide a fast decay of motion between press cycles or hammer blows. Since viscous dampers are almost purely velocity dependent and frictional dampers provide steady resistance, the viscous damper transmits less force to the foundation (Figure 4).

5.VIBRATIONTRANSMISSION

Resilient material isolators, which are mainly considered in this study, are very effective in isolating the impact force that occurs between the machineand the foundation due to the stretching and contraction of the tie rods. These forces occur predominately in the 90-120 Hz range. Since these isolators typically have natural frequencies in the 12- 20 Hz range, a high level of isolation is possible. For example, if the disrupting frequency (f_d) of a forging press is 90 Hz and the isolator's natural frequency (f_n) is 12 Hz, the resulting ratio is 7.5. Transmitted vibration is 0.02 (2 percent) or 98 percent vibration isolation (Figure 5).

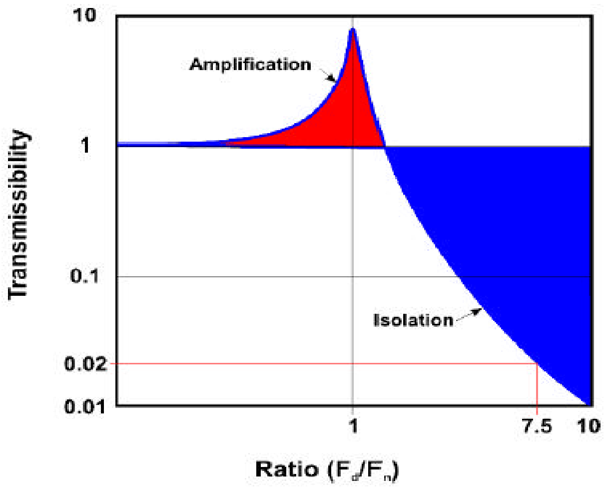

Figure 5. Transmissibility of the test machine vibration

Ewins (2000) defines transmissibility of vibration as the ratio between the response levels at two points, say j and k, due to a single excitation, say at DOF i. Following these definitions, the transmissibility can be written as:

$$ {}_iT_{jk}(w) = \frac{H_{ji}(w)}{H_{ki}(w)} \qquad (1) $$

where $H_{ji}(\omega)$ and $H_{ki}(\omega)$ are FRFs corresponding to the responses at points j and k, respectively, due to an excitation at DOF i.

Based on the above definitions, transmissibility can be regarded as a quantity that indicates the ratio of vibration levels between two points on the structure. More specifically, the transmissibility in this paper is defined as the ratio between the point and transfer mobility FRFs corresponding to two points, where the first point is located on one floor while the other point is located on the other floor exactly above the first point at the same position.

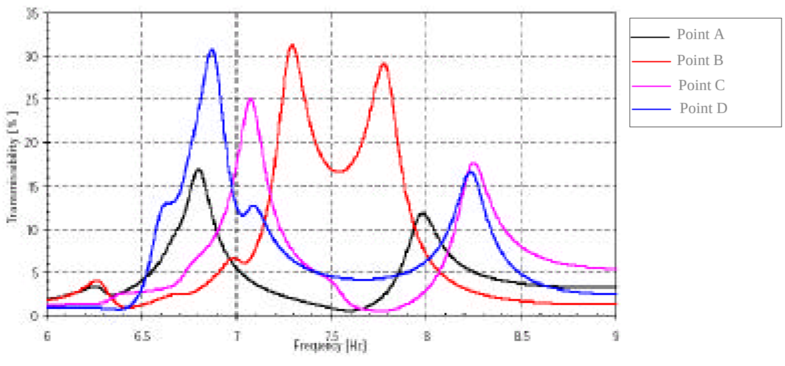

Figure 6. Transmissibility of vibrations for the isolation system

The frequency range from 6 to 9 Hz is considered with the frequency resolution of 0.0025 Hz. The ratios between transfer and point mobility FRFs for those two points, as a measure of transmissibility, are calculated, and the results are presented graphically in Figure 6. It is interesting to note that the highest transmissibility occurred at 6.87 Hz, which is not exactly at one of the resonance frequencies of either floor.

6.CONCLUSIONS

This study has given a qualitative explanation of machinery isolation. The best low tuning technique tends to be the method of mounting the machine on a stabilizing mass with isolation springs. This allows stiffer springs to reduce dynamic deflection. The minimum dominant frequency in the laboratory bay is above 18 Hz. A varying level of vibration transmission of the floor having closely spaced natural frequencies had been identified. Maximum transmissibility of 30% occurred at 6.87 Hz, which is close to but not exactly at one of the resonance frequencies of the floor

The results of relevance for the analysis and design of vibration isolating systems was obtained. All mobility and isolator dynamic stiffness or impedance data can be presented, as well as applied dissipation loss factors for the components and the combined dynamic load spectra used.

REFERENCES

1. H. Amick (1999), "A frequency-dependent soil propagation model", Conference on Optomechanical Engineering and Vibration Control, Denver, *SPIE Proc. Vol. 3786*

2. C. Harris (1996) "*Shock and Vibration Handbook*", 4th ed., McGraw-Hill, New York

3. C- H. Dowding, *Construction Vibrations*, (Prentice-Hall, 1996)

4. M. Sanderson, and A. Fredö (1995) "Direct measurement of moment mobility, Part I: Theoretical study". J Sound Vibr, 179, pp 669-684

5. M. Sanderson (1995) "Direct Measurement of Moment Mobility, Part II: An Experimental Study". J Sound Vibr, 179, pp 685-696