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VIBRATION PREDICTION AND CONTROL IN MICROELECTRONICS FACILITIES

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1. INTRODUCTION

The importance of vibration in microelectronics production can be appreciated when one considers the extreme sensitivity of the "tools" and systems used in the chip fabrication process and the large amount of mechanical energy that is consumed in a modern cleanroom. The most sensitive tools used in the process are greater than one hundred times, more sensitive to vibration than a human occupying, say, a desk in an office building. At the same time the total mechanical power consumed per unit area of cleanroom floor is one hundred times, or more, that consumed by the operation of an office building. The great difference in the ratio of "power consumed" to "sensitivity" lies at the heart of the vibration problem in cleanroom design.

2. VIBRATION CRITERIA

There is a dearth of reliable information from tool manufacturers about the vibration and noise sensitivity of their products. And where information is available there is wide variation in sensitivity from tool to tool. This is not surprising considering the complexity of these tools. To provide a basis against which the vibration "quality" of process floors supporting these tools can be quantified, various criteria have been proposed. The most widely used criteria are the "generic" vibration criterion (VC) curves[1]. These curves are referenced in a recent Recommended Practice published by the Institute of Environmental Sciences. The curves are shown in Figure 1. Each curve is associated with the particular complexity (expressed in line width or detail size) of the circuitry to be fabricated in the facility.

Figure 2 shows the vibration performance, at start-up of a typical modern high-tech fabrication facility. Spectra are shown for both vertical and horizontal vibration. This facility was designed to comply with VC-D (250 microinches/sec). The measured performance complies closely with the criterion.

3. FACILITY CONFIGURATION AND SOURCES

The cross section of a typical fabrication facility is shown in Figure 3. The major features of this particular design can be summarized as follows:

(1) The "process" floor on which the tools are supported lies above a sub-fab in which the many vacuum pumps and other systems associated with the process tools are located. Increasingly, fabs are being built with two levels of sub-fab — one to carry the clean recirculation air; the other for pumps and other utilities.
(2) The Central Utility Plant (CUP) in which chillers, boilers and other major equipment are located is often, but not always, set separately from the fab building.
(3) The recirculation air fans, and sometimes the make-up air handlers, are often located on a fan-deck directly above the fab.
(4) Other major mechanical systems, including exhaust fans, house and process vacuum pumps, DI water pumps, compressors, etc., are generally set at various levels and locations within the fab building.
(5) The structural design of these facilities usually provides a Structural Isolation Break (SIB) between the process floor and the roof/fan-deck structure (also called "shell" structure). The SIB's essentially create two separate structural systems for the fab building.

The typical fabrication facility, therefore, has many discrete sources of vibration located at many positions inside and outside the building. To these sources must be added distributed sources created by the ducts and pipes that transport fluids (chilled water, ultrapure water, exhaust and recirculation air, etc.) within the building. Fluid turbulence is a significant source of broadband vibration

In addition to the mechanical sources enumerated above, there are sources associated with:

(1) The site, caused by local roads and highways, industry, railroads, construction, etc.,
(2) Vehicular movements within the site, and
(3) Walkers and material movements on or close to the process floor

In certain cases the process tools themselves can generate vibration and impact the environment of neighboring tools. Generally, however, it is the building mechanical sources that dominate the final performance, of a typical microelectronics facility.

4. VIBRATION PREDICTION METHODS

By using multiple regression analysis procedures on data collected from many operating facilities, covering a wide range of design and performance characteristics, it has been possible to develop semi-empirical prediction methods. These methods have the advantage of being straightforward and fast in their application. They are also fairly accurate although, of course, they do not take account of the details of the design. They assume that "good practice" will be followed in terms of facility layout and in the selection, placement and isolation of major mechanical systems. They also assume that care will be taken in the design and isolation of sub-fab systems (vacuum pumps, etc.) that are associated with the process tools.

The primary determinant of vibration performance, according to these semi-empirical methods, is the static stiffness of the column-supported process floor in the vertical and horizontal directions. The practice in applying these methods is to use Finite Element modeling (FEM) and other analytical techniques to determine the appropriate stiffnesses. These techniques are also used to study vibration propagation paths and the effectiveness of SIBs and other methods of attenuating these paths.

Ideally one would like to predict the vibration performance using purely analytical methods in which the contributions of all mechanical sources, whatever their nature, size and location, are included. FEM techniques offer the opportunity of such an analysis and these techniques are being used with some success now. There are many areas of difficulty, however, that must be resolved before FEM can be used as an alternative to semi-empirical methods. Some of these areas are as follows:

(1) Model Complexity. — Most modern fabrication buildings are very large, incorporating cleanroom areas in the range 75,000 to 125,000 sq ft. Computer models encompassing the whole building are large, therefore, and exercising these models can quickly strain the computing capabilities of most organizations. Even partial models that include, for instance, only the process "table" or a narrow slice of the total building (two-dimensional model) can require substantial computing time. An example of a partial finite element model having 4000 elements is shown in Figure 4. This model shows two levels of sub-fab and the shear walls required for horizontal stiffening.

(2) Soils Model.— Dynamic analysis of a half-space soil medium has generally been evaluated using three analytical methods: FEM, closed form solutions, and parametric modeling. In FEM analysis, one encounters complexities such as boundary conditions, refinement of elements to account for high frequency waves, and of course, the size of the model. Closed form solutions only exist for very simplified loads and geometries, rendering it impractical for microvibration evaluation. Parametric soil modeling holds more promise. This latter methods coupled with techniques such as the Boundary Elements Method (BEM) may allow for practical evaluation of microvibration propagation and dynamic soil-structure interaction. The soils play a critical role in determining the efficiency of vibration propagation both from sources outside the facility (the central plant, for instance) and between building components themselves (between the perimeter columns that support the fan deck and the columns that support the process floor slab, for instance). The soils also contribute to the damping of the structure since energy can pass from the structure into the ground via the foundation system. The importance of this damping mechanism is thought to be particularly important at microvibration amplitudes.

(3) Machinery Force Spectra. — Whereas it is normal practice for the manufacturers of pumps, cooling towers, fans etc. to measure the acoustic noise radiation from their products with considerable accuracy, it is virtually impossible to obtain information on the force spectra generated by these machines. Yet this information is essential if one is to predict the vibration response on the process floor using purely analytical techniques. Machines cannot be quantified simply on the basis of unbalance forces at the shaft rotational frequency. Most machines generate significant vibration, both pure tone and broadband, over the total frequency range covered by the vibration criterion curves. It is important to realize that even if these machines are vibrationally isolated they cannot be neglected as contributors to the final vibration performance. Examples of measured spectra from several machines are shown in Figure 5.

There is much work to be done, therefore, before purely analytical techniques can replace semi-empirical methods as the basis for the facility vibration design.

References

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Figure 1. Generic Vibration Criterion (VC) Curves for vibration-sensitive equipment

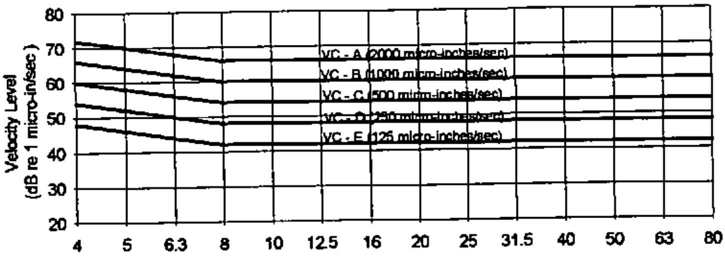


Figure 2: Velocity Levels in Representative Facility (Average plus one standard deviation levels from multiple locations)

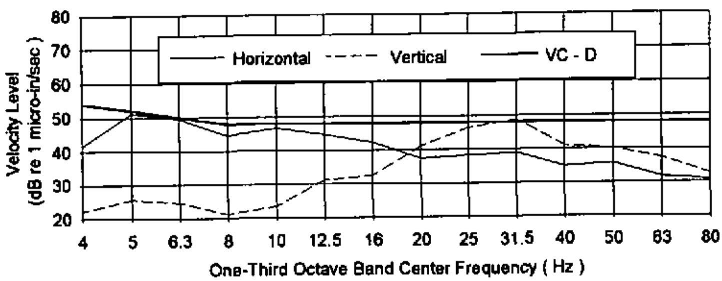


Figure 3: Typical Microelectronics Cleanroom Design Showing Vibration Control Features

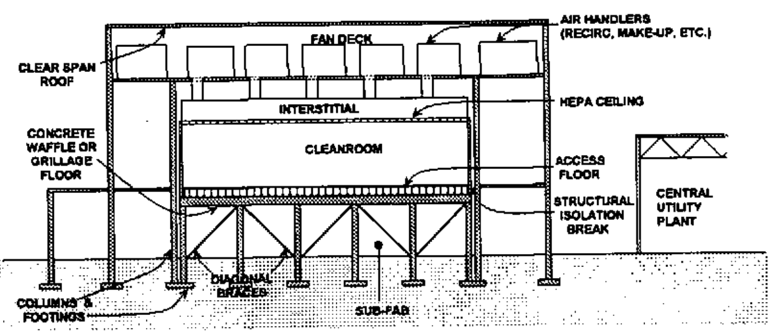


Figure 4: Finite Element Analysis Structural Model

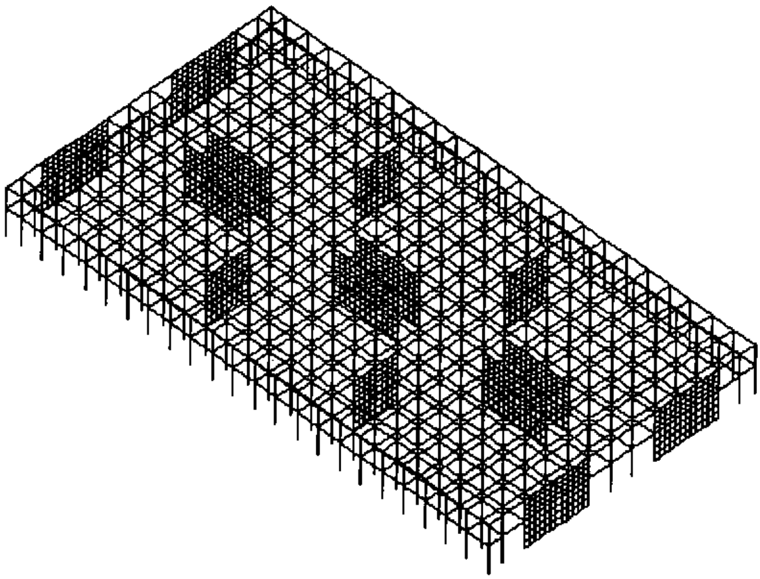
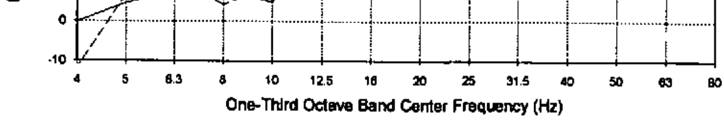


Figure 5: Average Force Spectra for Various Machines



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ISSUES OF MINIMISATION OF NOISE IN MACHINES TREATED AS MULTI-SOURCE SYSTEMS

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1. INTRODUCTION

Designing of machines with a minimum noise levels both at the operator's cab and in the external field is currently one of the more important technical concerns. Growing threats that noise poses to environment leads to increase of requirements connected with norms, and thus requires that these issues are taken into account already at the machine's construction stage. The basic difficulty lies in the fact that by using theoretical means we can approximately (generally not very precisely) predict the acoustical results of a ready product. Acoustic field inside the machine's structure is not a free field and as a matter of fact it cannot defined in theoretical manner. For a constructor the only points of reference are the prototype's acoustical properties and detailed analyses of similar structures, enabling estimation, with some approximation, of the final effect in the new machine. The author of this study participated in numerous research projects probing both into the acoustic properties of machines (hydraulic excavators) and into sound-insulating materials and structures, which resulted in creation of rich "data banks" [5]. On the basis of identification measurements he also worked out numerous algorithms aiding design of sound-insulating casings and covers for machines [2,3,4]. This work presents a proposal of an algorithm aiding the decision of selection of a sound-insulating structure in case when the noise at the operator's cab results from activity of a number of sources.

2. GENERAL CHARACTERISTIC OF THE CALCULATION MODEL

Machine can be treated as a system of sources of noise and mechanical vibration of which part can be to a certain degree correlated with each other. In practice, the level of noise at the operator's cab is influenced by one or by a number of dominant sources which usually are power units or elements of the working unit. Analysis of the machine's vibroacoustic properties should be conducted on the basis of uniform recording of vibration and noise which would enable evaluation of transformation of vibration energy into acoustic energy (and vice versa), and thus identification of various propagation paths [4]. Let us assume [1] that the real, measured signal in the frequency domain at the systems output point (at the operator's cab) can be defined by the following relationship:

$$X=\sum P_i H_i+\Phi+\Phi^* \tag{1}$$

where: X - Fourier transform of real existing signal $x(t)$,

$P_i=\mathscr{F}p$ - Fourier transform of excitation,

H_i - transmittance,

Φ - error of the model,

Φ^* - error of linear approximation.

Since each propagation path can be defined theoretically with the use of a linear model which gives a very bad approximation, the basis of our analysis will be the identified model, which requires definition, in empirical manner, of the functions Φ and Φ^* or of their sum. The proposed approach assumes that the linear effects will be so weak, that a sufficiently good approximation will be obtained by accounting for the non-linear effects in an additive way (or after simple transformations [1] with the use of the multiplication manner).

3. CALCULATION ALGORITHM

Let us analyze the problem more precisely on the example of a hydraulic excavator powered by two not insulated power units equipped with identical combustion engines. The level of remaining sources is lower than $\Delta L > 10$ dB.
The aim of the algorithm is to find a proper characteristics for an appropriate structure placed on the noise propagation path to decrease the level of noise by a given value.
Description of each propagation path in the frequency domain results directly from equation (1), after showing the acoustical transmittance as a product of transmittances and subjecting both sides of the equation to the operator of change of scale to dB (by finding the logarithm) [1].
Both paths of noise propagation have been defined with assumption of the following denotations:

L_s - noise level of the source,

L_w - growth of noise level resulting from the existing engine casing,

D_o - acoustic insulating power of the existing engine casing,

L_p - decrease of noise level along the engine - operator's cab path calculated as for a free field,

$D_o^{\prime}$ - acoustic insulating power of the new (additional) insulating structure,

Δ - error of path noise propagation description.

The above quantities are calculated with the use of known acoustic formulas. Thus the decrease of noise level for each propagation path is defined by the following relationship:

$$\Delta L_i = L_{si} + D_{oi} + D'_{oi} + L_{wi} + L_{pi} + \Delta_i \tag{2}$$

Summing up of the noise levels in the sources should be done in a "logarithmic" manner, that is by adding appropriate powers (intensities) of noise. Thus, the summary level at the operator's cab will defined by the following relationship:

$$\Delta L = 10\log\left(10^{0.1\Delta L_1} + 10^{0.1\Delta L_2}\right) + D_k + \Delta^* \tag{3}$$

where: D_k - denotes cab's insulating power,

Δ^* - total error of nonlinearity (according to Φ^* function).

Relationships (2) and (3) are vector equations with a number of components equal to the number of bands in the conducted analysis. Practically the analysis is conducted in 1/3-octave bands, since these offer sufficient precision for selection of appropriate acoustical characteristics for designed structures.

The proposed model already enables certain estimation of results of application of various structural solutions (with the use of a rich data bank), but the precision of such proceedings is small.

3. IDENTIFICATION PROCEDURE

There are various techniques of identification of the assumed model. In the simplest case it is possible to assume the same structure of the vector of corrections for both paths which gives us possibility to skip one or two identification measurement, that is the one connected with definition of the difference between the level of the selected source and the level at the operator's cab. The unknown Δ functions are introduced into equations (2) by means of algebraic addition to the right-hand sides of these equations. The summary level in the cabin is thus calculated:

$$L_c^{(A)} = L_c^{(A)}(\Delta_1, \Delta^*) \tag{4}$$

and the identification equations are constructed

$$\Delta^{*(A)} = L_c^{(A)}(\Delta_1, \Delta) - L_{ni}^{(A)} \tag{5}$$

where: L_{ni} - denotes real (measured) level at the operator's cab, A - denoted work parameters.

The vector of corrections, determined from equation (5), is entered into corrected equations (2). The advantage of an algorithm obtained in such a way is that when there is no possibility to conduct identification (which happens when a new machine is designed), then it is possible to enter appropriate values equal to 0 and make initial estimations.

4. CALCULATION PROGRAM

The program that has been worked out in the MATLAB environment on the basis of the procedure that was delineated above, enables estimation of the noise level at operator's cab in the function of acoustical parameters of engine and cabin casings, and on the basis of new additional sound-insulating structures (the acoustical characteristics of which are found in the data bank) placed on the propagation paths. The input parameters are: structure dimensions and noise levels of the sources. This gives the designer broad opportunity for action. While designing a new machine we can select appropriate parameters of designed casings/shields by modifying the already existing structure, we can study advisability of changes of acoustic insulating power for existing elements, or we can analyze influence of new structures. In response to proposal of use of casings or linings with known parameters, we obtain both the spectrum structure and the summary level of noise, with correction A, at the operator's cab.

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