998265

25X1

[redacted]

Task II

# Item 1. Submicron Measurement Error Analysis

3rd Preliminary Technical Report

Declass Review by NGA.

25X1 [redacted]

June 30, 1965

25X1 [redacted]
P. O. Box 8043
Southwest Station
Washington, D. C., 20024

Re: Contract [redacted] 25X1

Dear Sir:

Enclosed are five copies of the third preliminary report on Item 1 of Task II of the subject contract. The report is entitled "Submicron Measurement Error Analysis." The enclosed report is submitted in accordance with the requirements of the subject contract.

Very truly yours,

[redacted] 25X1

Enclosures

[redacted] 25X1

June 30, 1965

25X1 [redacted]

## Task II, Item 1, 3rd Preliminary Technical Report

## Item 1. Submicron Measurement Error Analysis

### WORK STATEMENT

Evaluate the physical and metallurgical properties of materials used in measuring engine construction to determine comparative suitability to submicron measuring. Materials to be considered are: Meehanite, steel, granite, aluminum, magnesium, and glass, and other materials that may be particularly suitable.

Evaluate physical properties and structural concepts appropriate to achievement of vibration levels and structural rigidity compatible with submicron measuring requirements. Evaluate methods of measuring the small vibration levels expected in a high performance structure.

Reports No. 1 and No. 2 dealt with the physical and metallurgical properties of materials. This report, No. 3, deals with structural rigidity and vibration control.

Submitted by:

[redacted] 25X1

[redacted] 25X1

Task II, Item 1, 3rd Preliminary Technical Report

## CONTENTS

[redacted]

25X1

Task II, Item 1, 3rd Preliminary Technical Report

## LIST OF FIGURES

# 1. SUMMARY

## 1.1 Introduction

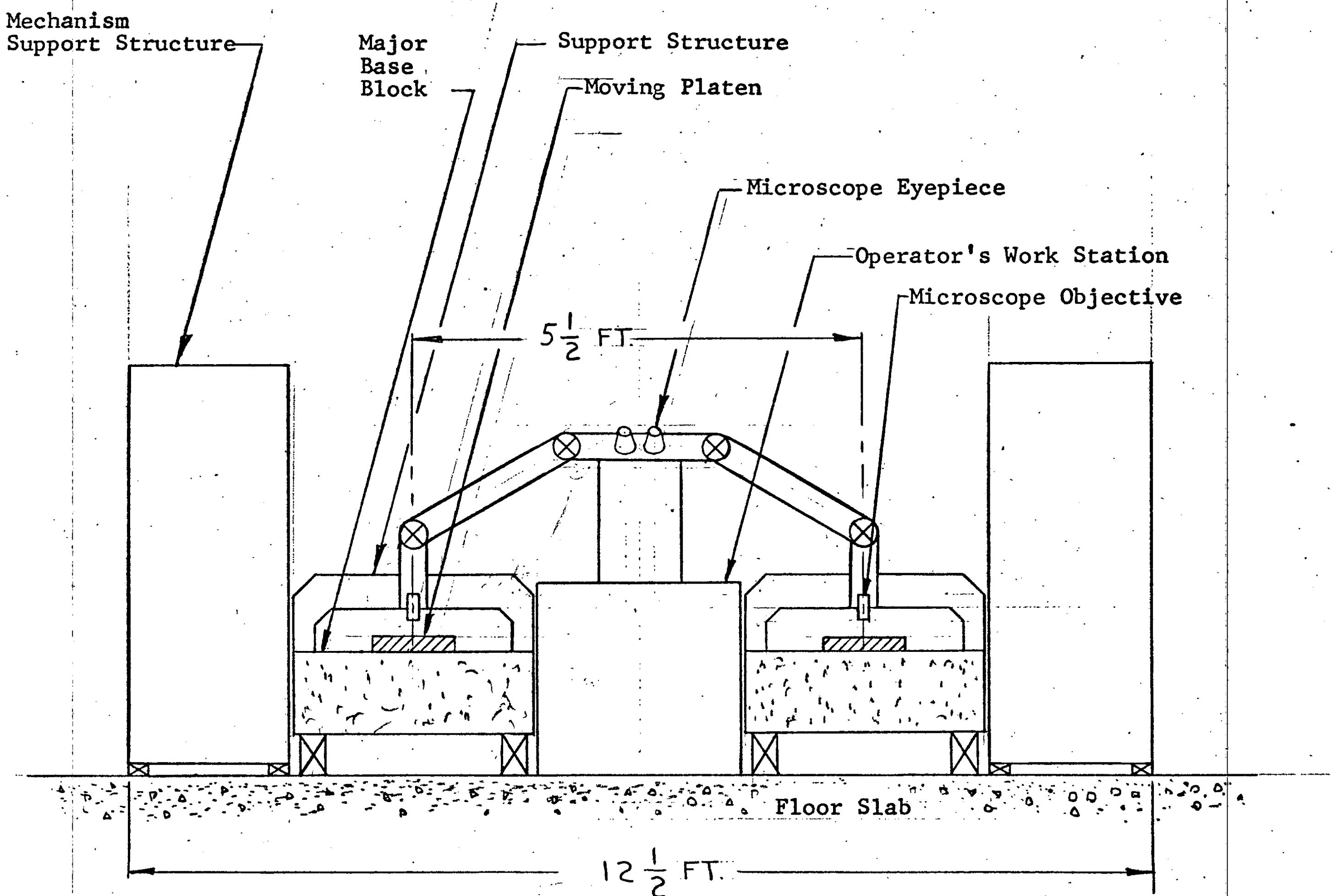
This report is intended to provide some guidelines for the preliminary design of a structure for a precision stereo comparator. The general major structural components of a stereo comparator for submicron measurement are:

a. The major base blocks
b. The moving platens
c. The microscope objective lens supports
d. The film drive and roller support structures
e. The microscope eyepiece lens support
f. The operator's work station

The film will be carried on the moving platen therefore inadvertent relative movement between the moving platen and the microscope objective must be avoided. To minimize undesired motion the structure supporting the microscope objective should be rigidly anchored to the base block. It is assumed that the microscope and the operator will remain in a fixed position and the moving platen carrying the film format will move to permit viewing different parts of the format with the microscope.

One advantageous arrangement is to place the operator's work station between the two moving platens (see Fig. 1). Such an arrangement imposes certain restrictions on the machine and its structure. For example, the two base blocks should be independently supported since a structure which would tie the two blocks together with the required stiffness appears to be impractical. The microscope eyepiece must be supported independently from the base blocks. Thus there will be relative movement between the two microscope objectives and between the objectives and the eyepiece. The microscope optics must, therefore, incorporate a pivoted link (similar to a stereo arm) which permits the above relative movements without adversely affecting the image observed by the operator. The relative movement to be accommodated will be small.

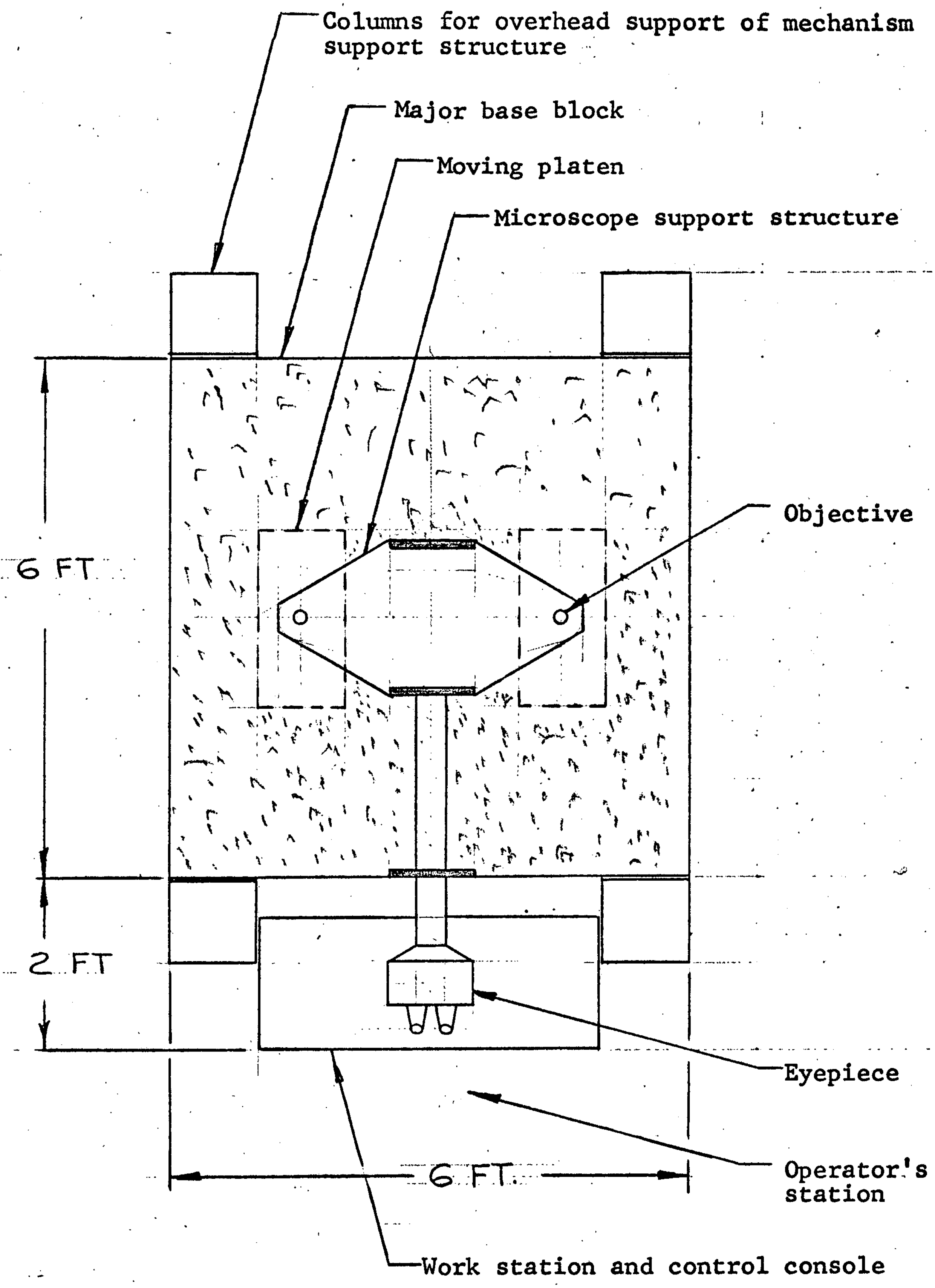
An alternate arrangement of major structural components is shown in Fig. 2. This arrangement uses one base block, approximately 6 ft x 6 ft to support both moving platens. It is a much less desirable arrangement for operating ease and efficiency, but it places the microscope optical system in one rigid structural unit. The optical relay path length is approximately 3 ft longer than the first arrangement which minimizes the optical relay path length.



A General Arrangement of Major Structural Components

(Front Elevation View)

Fig. 1



An Alternate General Arrangement of Major Structural Components

(Plan View)

Fig. 2

Other general considerations of the structure are: (a) the operator's work station must be structurally isolated from the major base block; (b) the mechanism support structure which supports the film rolls and drives, vacuum pump, blowers, etc., must be structurally isolated from the major base blocks; (c) ideally, the natural resonant frequencies of structural components should be widely separated and well damped. Items of concern are:

a. The vibration isolator supports
b. The floor slab
c. The free-free mode of the major base block
d. The free-free mode of the moving platen
e. The moving platen air bearing
f. The microscope objective support structure
g. The microscope eyepiece support structure
h. Every beam, post and bracket in the mechanism support structure
i. The operator's work station

(d) input disturbances should be minimized by dynamic balancing of motor rotors, film rollers, pumps and fans.

A detail analysis of the structural dynamics of the submicron measuring engine can be performed by an existing computer program when the preliminary design of the structure is established. The analysis will provide data for the detail structural design. Section 6 of the report indicates the preliminary design data needed for the computer analysis.

### 1.2 Summary of Report

A homogeneous granite major base block will have a fundamental dynamic frequency of about 410 cps which is satisfactory. Two blocks will weigh about 3400 lbs each. They can be readily supported by the floor with a simple support design.

Cast iron major base blocks will be much more rigid and lighter weight. The fundamental dynamic frequency will be about 765 cps and weight about 2,000 lbs each (including a 2-in. granite cap).

A fabricated steel major base block will be even more rigid. The fundamental dynamic frequency will be 1,710 cps which is excellent. The weight would also be about 2,000 lbs each (including a 2-in. granite cap). Special precautions would be required in the design to insure a well damped structure and to insure that all webs have a high fundamental mode. If the suggested two major base blocks are combined into one, the above weights will double.

Calculations of the floor slab stiffness indicate it will have a fundamental mode of 20 to 65 cps. It is estimated floor excitation will be between $10^{-2}$g and $10^{-3}$g. The floor slab vibration characteristics should be measured.

The microscope depth of field will be approximately 2 to 8 microns; therefore, the structure which supports the microscope objective lens must be rigidly attached to the major base block. If possible the microscope eyepiece should be independently supported in order to obtain the most effective operating arrangement. The microscope relay lens structure should be vibration isolated so as not to transmit vibration from the eyepiece to the major base block.

The moving platen, if made of 1-1/2-in. thick glass, will have a 120 cps fundamental mode which is too close to the estimated theoretical fundamental mode of 115 cps of the air bearing. A test of the normal transmissibility of the air bearing is needed. Low frequency pressure pulsations in the air bearing should not be allowed to exceed 1% or 2%. The lateral transmissibility of vibration across an air bearing is extremely low. Under worst conditions it should not exceed 1 to 2 millimicrons which is excellent for measuring resolutions of 1/10 micron.

Ideally there should be no mechanical connection between the moving platen and the major base block which could provide a vibration path bypassing the air bearing. Practically, special attention should be paid to minimizing electrical cables, hoses, drive and film loop connections to the moving platen and to designing the required connections for minimum transmission of vibration to the moving platen. Pneumatic and electromagnetic drives should be given special attention for this reason.

The outer structure should house all sources of vibration and shock associated with machine operation. The outer structure should be physically structurally separate from the major base block and the optical support structure. Simple conventional vibration pads are adequate for vibration isolation of the outer structure from the floor. Bolted construction and other damping techniques such as sand filling should be employed. After preliminary design, the elements of the outer structure should be computer analyzed for resonance interactions.

Rotating members of drives, pumps, and blowers should be balanced to achieve a close coincidence of the center of mass to the center of support. Recommended maximum eccentricity versus speed is given.

Fig. 2a is a composite chart of the vibration frequency data. The chart shows peak-to-peak amplitudes of vibration in microinches vs frequencies at various vibration g levels. The data may serve as design guidelines in sizing the structural components. The actual responses of the structural elements must be calculated by analyzing the structure as a dynamic model.

## 1.3 Conclusions and Recommendations

The principal conclusions of this report are:

a. A granite major base block will be satisfactory and least expensive. A superior composite steel structure with a granite or glass cap could be designed but would be more expensive.

b. The major base block or blocks should be supported on three pneumatic isolators of about 8 cps natural frequency. Automatic level recovery time should be no more than 2 sec.

c. The principal elastic mode of the floor is estimated to be in the range of 20 to 65 cps and should be measured.

d. The microscope objective lenses should be rigidly fixed to the major base blocks.

e. The microscope eyepiece should, if possible, be separately supported.

f. The operator's work station should be structurally separated from other structure and need not be vibration isolated from the floor.

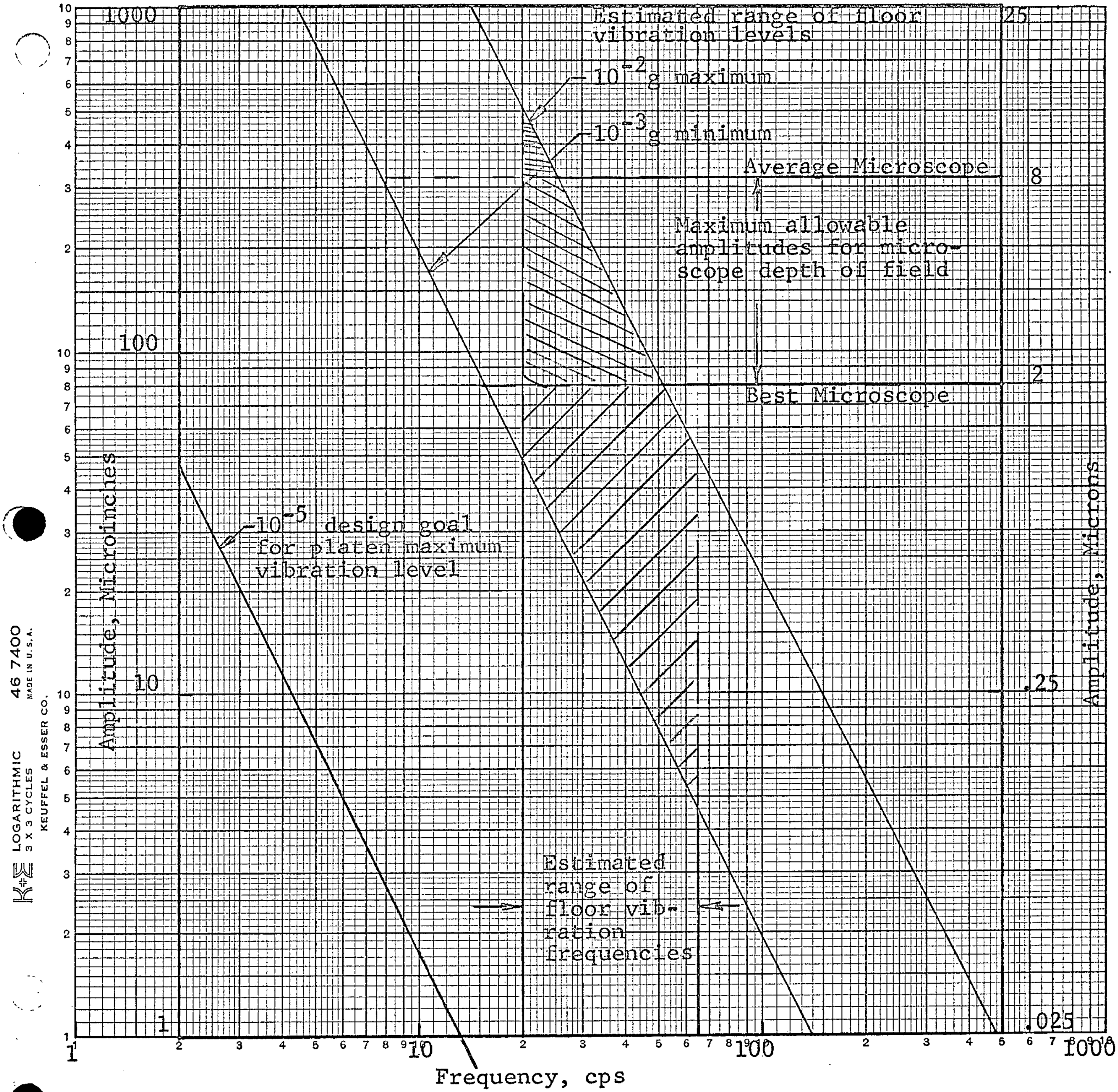
g. An outer structure should house all machine sources of vibration and shock and should be structurally separate from the major base block. Conventional vibration pads can be used.

h. Air bearings will provide excellent isolation of horizontal vibration. Transmissibility should be well below the 1/10 micron measuring resolution desired.

i. Transmissibility of vertical vibration by the air bearings may have an undesirable characteristic and should be measured.

At this stage recommendations are principally concerned with filling gaps in our information:

a. Measure floor resonance and damping.



Composite of Vibration Frequency Data

Fig. 2a

b. Measure natural damping of granite.

c. Measure transmissibility of an air bearing normal to the air cushion.

We suggest that the customer have the above measurements made during the preliminary design phase so that the data will be available for the detail analysis of the structure.

## 2. THE MAJOR BASE BLOCK

### 2.1 Composite or Homogeneous Construction

The basic structural dynamic requirements of the major base block are as follows:

a. The fundamental bending frequency of the free-free structure should be well above 300 cps.

b. The weight of the block must be such that the loads transmitted through the supports to the floor will not exceed the allowable design loads of the floor.

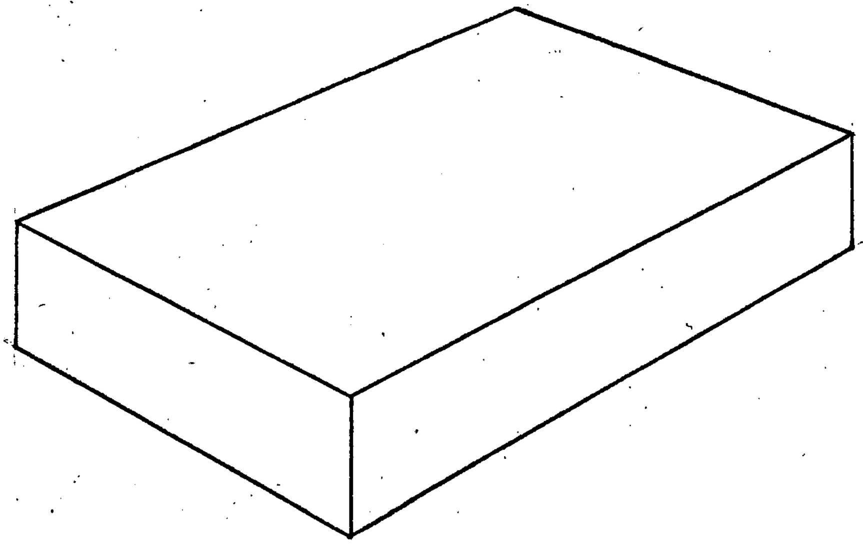
c. The material and construction of the block should provide high damping to minimize vibration loads.

Three methods of the major block construction have been considered. Each of the methods is described below.

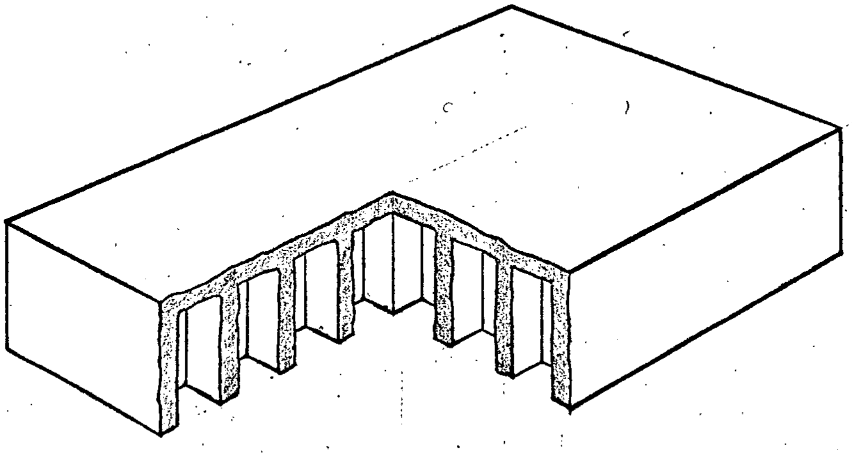
The first construction method is simply a solid granite block (see Fig. 3a) supported by a number of isolators along the fundamental mode node line of the block. This is an homogeneous construction. The surface of the granite can be smoothed and polished. The surface is relatively free of corrosion and easy to keep clean. The material damping of granite is high; however, the flexural rigidity per distributed weight of the homogeneous granite is low compared to cast iron or steel structures. The cost of the homogeneous granite block is lower than other construction methods.

The second method is also a homogeneous construction. The block will be cast to form a waffle-like cast iron structure (see Fig. 3b). With this type of construction the distributed weight, M (lb/in.), along the length dimension is lower than granite. Assuming an 18-in. depth block, the EI/M will be somewhat higher than granite which results in higher bending frequency. Cast iron also has good damping characteristics. One of the disadvantages is that the surface would require protection from rust and corrosion. Care must be exercised in sizing the webs such that the natural frequencies of the webs are well above the fundamental block bending frequency.

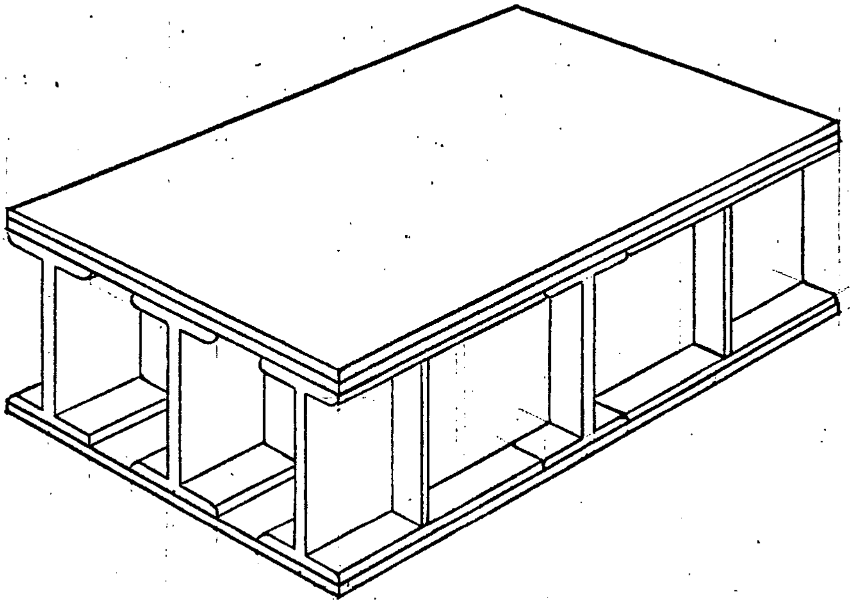
The third construction is a composite type made of a bolted steel framed structure 18 in. high with a granite or glass top (see Fig. 3c). The steel-framed structure could be of sandwich type. The compartments can be filled with sand to provide damping. A bolted instead of welded structure



(a) Homogeneous granite block



(b) Cast iron block with waffle pattern bottom



(c) Composite steel block with granite or glass top plate

Magnet Base Block Construction Fig. 3

is preferred because of higher structural damping. Of the methods considered, the composite type of construction generally will yield highest EI/M. The total weight of the block is also the lowest.

## 2.2 Weight and Rigidity

As stated in 2.1 the weight requirement of the block is that the support load should not exceed the allowable floor load. Since the support is of point load nature and the floor is of concrete slab construction, the punching shear load will be the criterion in calculating the internal stresses of the floor slab. By providing a steel base plate under the support the loading may be considered a distributed load.

Preliminary estimates of weights for the three types of block construction have been made. The granite block would weigh approximately 3400 lbs. The weight of the cast iron or the steel-framed blocks is approximately 2,000 lbs (including a 2-in. thick granite or glass surface plate).

Assuming the moving plate and the associated equipment would weigh around 200 lbs, the gross weight to floor for the granite block (worst case) will not exceed 3,600 lbs which will be distributed among three supports. If one of the supports receives 1,800 lbs and the support base is 10 in. in diameter, the shear load along the perimeter would not exceed 60 lbs/in. Assuming the support rests on a 10-in. concrete slab, the concrete shear stress would be 6 lbs/in.$^2$ which is well within the allowable.

## 2.3 Principal Elastic Mode of Vibration

The fundamental free-free bending frequenicies of the three block structures have been calculated and listed below:

| Block Type | M(lbs/ft) | EI(in.$^2$/lb) | f(cps) |
|---|---|---|---|
| Granite | 570 | $2 \times 10^{10}$ | 410 |
| Cast Iron | 330 | $4 \times 10^{10}$ | 765 |
| Steel-framed | 330 | $20 \times 10^{10}$ | 1710 |

The calculations were based on 6 ft-span free-free beams.

## 2.4 Vibration Isolation of the Blocks

To minimize floor excitations to the major base block, it is desirable to support the block on isolators.

The mass-spring resonance frequency of the isolators should be well below the fundamental mode of the floor slab. It is undesirable that the isolators be too soft since the moving platen would then cause tilt of the base block. If the moving platen is 10% of the weight of the base block, the tilt could be as much as 10% of the static deflection of the isolators. To eliminate permanent tilt of the major base block, a self-leveling pneumatic type isolator such as the Barry Controls SERVA-LEVL should be used. The very soft standard 2 cps SERVA-LEVL has a level recovery time of 15 to 20 sec which is too long. To prevent interaction of the level recovery system and the vibration isolation action the level recovery time constant must be long compared to the vibration isolator time constant. Probably an 8 cps mount with a 1 to 2 sec level recovery time would be a good compromise.

## 2.5 Interaction with the Floor Slab

In general, if the floor structure is of reinforced concrete construction, the fundamental bending frequency will be above 15 cps. A preliminary estimate of the bending frequency of the floor where the submicron measuring system will be located is somewhere between 20 and 65 cps.

The criteria set forth in 2.1 and 2.4 were generated from the knowledge of the floor structure.

The relative displacement transmissibilities considering a system indicated in Fig. 6a can be expressed by the following equation:

$$\frac{X_3}{X_1} = \sqrt{\frac{\left[\left(2\zeta_1 \frac{\omega_F}{\omega_A}\right)^2 + 1\right]\left[\left(2\zeta_2 \frac{\omega_F}{\omega_B}\right)^2 + 1\right]}{\left\{\left[1-\left(\frac{\omega_F}{\omega_A}\right)^2\right]^2 + \left(2\zeta_1 \frac{\omega_F}{\omega_A}\right)^2\right\}\left\{\left[1-\left(\frac{\omega_A}{\omega_B}\right)^2\right]^2 + \left(2\zeta_2 \frac{\omega_F}{\omega_B}\right)^2\right\}}}$$

where

$\omega_A$ = Floor excitation resonant frequency

$\omega_B$ = Isolation system frequency

$\omega_F$ = Major base block resonant frequency

$\zeta_1$ = Damping ratio of the isolation system

$\zeta_2$ = Damping ratio of the base block

Sections 6 and 7 will discuss further analytical and experimental work to determine the over-all structural dynamic performance of the submicron measuring system. The floor excitations and interaction with the measuring system are of major consideration.

## 3. THE MICROSCOPE OBJECTIVE SUPPORT

### 3.1 Microscope Depth of Field

The allowable relative movement of the film platen and the microscope objective support is governed by the microscope depth of field.

$$DF = \frac{\lambda \sqrt{n^2 - (NA)^2}}{(NA)^2}$$

where

$\lambda$ = Wave length of light = $0.5 \times 10^{-6}$ meters

n = Index of refraction of air = 1

NA = Numerical aperture which varies from 0.25 to 0.5

The DF range varies from 80 to 320 microinches (2 to 8 microns).

### 3.2 Some Structure Criteria

The microscope objective support structure criteria are:

a. The resonant frequency should be at least twice that of the major base block.

b. The relative peak-to-peak movement should be less than 80 microinches (2 microns) under any excitations.

c. The optical linkage components connecting the eyepieces should not transmit appreciable vibration load to the microscope objective structure.

## 4. THE MOVING PLATEN

### 4.1 General Size and Construction Considerations

The required observable format size for each platen is 10 in. x 20 in. The glass plate can therefore be assumed to be approximately 1 ft x 2 ft. To obtain adequate flatness (approaching an optical flat) it will need to be at least 1 in. thick and may need to be 2 in. thick. The weight of the plate only will, therefore, be from 25 to 50 lbs. Additional thickness will be required for vacuum slots. In addition, structure will be required for supporting the glass plate, for air bearings, for the ways, for the X-axis measuring engine, for the intermediate ways and for the Y-axis measuring engine. The total weight will probably not exceed 200 lbs for each moving platen of the stereo pair.

### 4.2 Air Bearing Normal Transmissibility and Pulsation

The air bearings which will support the moving platen on the base blocks will help to isolate the platen from vibrations in the base. The air bearing will act as a damped spring and with the supported mass will have a frequency response characteristic dependent on the spring-mass constants. Amplification at the resonant frequency will depend upon the damping. As a first approximation of transmissibility we assumed the air bearing would follow the same principles as an air bearing isolator in which the natural resonant frequency is determined by the air column height. The air column height was assumed to be the thickness of the air cushion in the air bearing. The resonant frequency is therefore estimated to be:

$$f_n = \frac{1}{2\pi} \sqrt{\frac{g\,\gamma}{y}}$$

where

$f_n$ = Resonant frequency, cps

$g$ = Gravity constant = 386 in./sec$^2$

$\gamma$ = Ratio of specific heats of air = 1.4

$y$ = Thickness of air cushion = 0.002 in. (assumed)

thus

$$f_n = \frac{1}{2\pi} \sqrt{\frac{386\ (1.4)}{0.002}} = 82.5 \text{ cps}$$

The curve of Fig. 4 illustrates the estimated theoretical transmissibility of the air bearing. Note that below 25 cps a 0.002-in. thick air bearing will transmit vibration at 1:1. Between 25 cps and 150 cps an 0.002 -in. thick air bearing will amplify vibration as much as 1.4:1. Above 150 cps it attenuates vibration and at 300 cps the transmissibility is 0.3. Transmissibility is defined as the output amplitude divided by the input amplitude.

For an 0.001-in. thick air bearing $f_n$ = 115 cps. The curve will be similar but shifted to the right as shown.

The actual response of an air bearing to vibration inputs may differ appreciably from the estimated theoretical transmissibility shown here. A more realistic theoretical analysis immediately becomes exceedingly complex and no dynamic analysis of an air bearing has been found in the literature. A measurement of the frequency response of a typical air bearing is recommended since it appears that the air bearing will amplify vibration in the frequency range of the floor inputs and at the fundamental mode of the glass platen.

Pressure pulsations in the air bearing, if they are of low frequency, will change the air cushion thickness. If the air cushion is 0.001-in. thick then a 10% pressure variation will cause the platen to move 2-1/2 microns which is unacceptable for the depth of the field of the best microscopes. A 1% pulsation would cause only 1/4 micron movement of the platen which is acceptable.

## 4.3 Air Bearing Lateral Transmissibility and Pulsation

For horizontal vibration the air bearings will be a highly effective isolator. Motion will be transmitted across the bearing by shear in the air cushion. The shear force is given by:

$$F = A \mu \frac{dV}{dy}$$

where

$F$ = Force on supported member, lbs

$A$ = Area of supporting air cushion, $ft^2$

$\mu$ = Viscosity of air = $.04 \times 10^{-5}$ lb sec/$ft^2$

$\frac{dV}{dy}$ = Velocity gradient of the air across the thickness y of the air cushion in ft/sec/in.

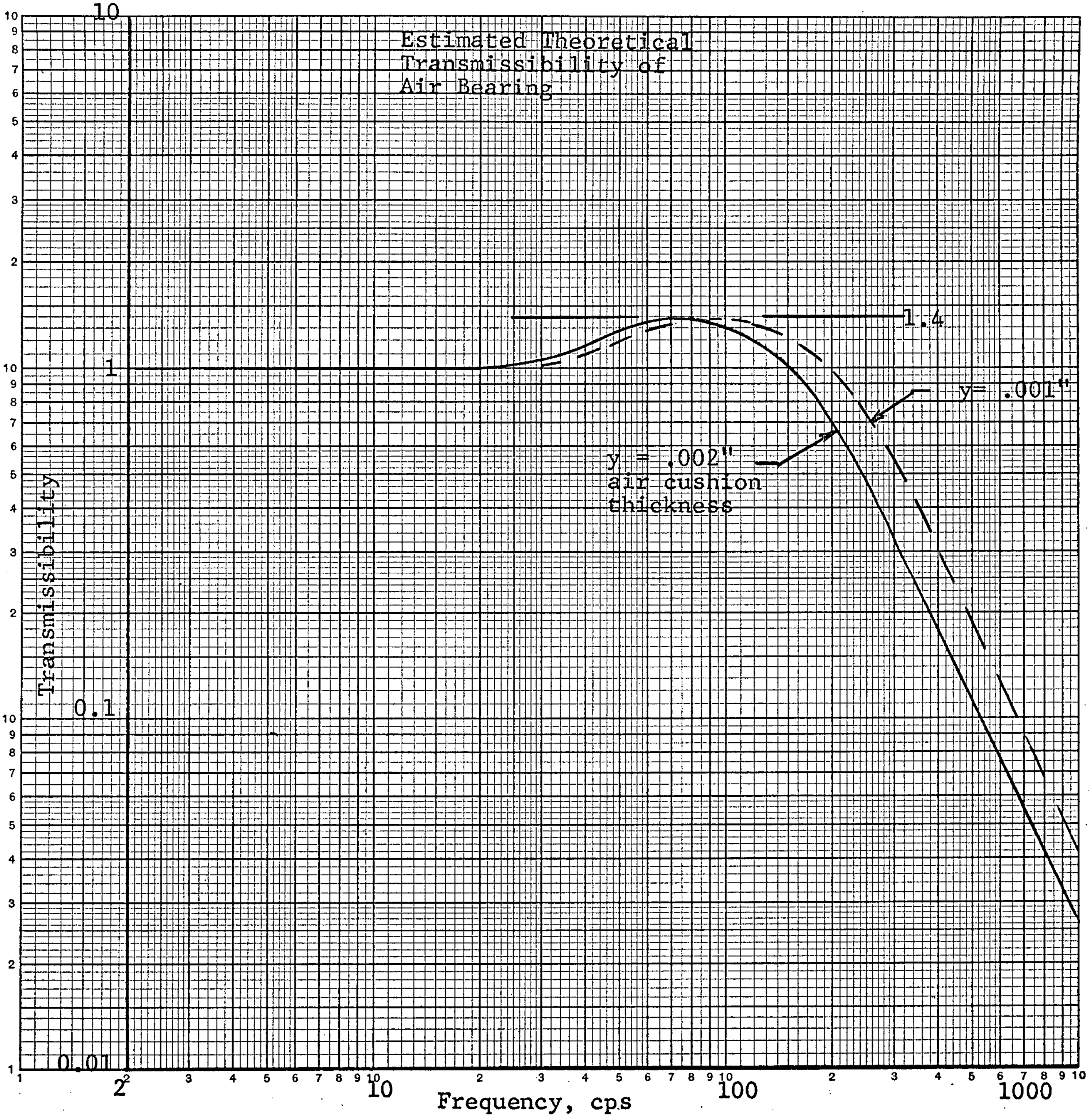

Estimated Theoretical
Transmissibility of
Air Bearing
10
1
0.1
0.01
Transmissibility
1.4
y= .001"
y = .002"
air cushion
thickness
2
10
100
1000
Frequency, cps

Fig. 4

By assuming some fairly extreme worst case conditions, we can demonstrate that lateral transmissibility across the air cushion is negligible.

Assume a base vibration of 300 microinch amplitude and about 65 cps (i.e., 400 rad/sec), then peak velocity of the base will be 0.12 in./sec. The force transmitted to the supported member will be:

$$F = \frac{40}{144} \times 0.4 \times 10^{-5} \times \frac{.12}{.001} = 1.33 \times 10^{-3} \text{ lbs}$$

for $A = 40 \text{ in.}^2 = 40/144 \text{ ft}^2$

$\mu = 0.4 \times 10^{-5}$ lb sec/ft$^2$

$dV = 0.12$ in./sec

$dy = 0.001$ in. air cushion thickness

For a 200-lb supported weight this is less than $10^{-5}$g which corresponds to approximately 1-1/2 millimicrons amplitude. Such amplitude is negligible compared to the 1/10 micron resolution desired.

## 5. THE OUTER STRUCTURE

### 5.1 Some Structure Criteria

The outer structure will support the film drive system and other operating equipment. Although the outer structure will be isolated from the floor and the major block, any excitation of the outer structure to the floor will be transmitted to the major base block. The criteria of the outer structure design are listed below:

a. The resonant frequencies of all the components and their mountings to the outer structure should be well separated from the resonant frequencies of the floor slab, block isolation system, and the first elastic mode of the major base block.

b. Bolted structure is preferred to welded structure for higher structural damping.

### 5.2 Vibration Isolation of the Structure

The purpose of the isolation of outer structure from the floor is to minimize floor excitation. With a proper conventional design of mounting pad the goal can be achieved. It is not necessary to incorporate a pneumatic isolation system as required for the major block. Care must be exercised in designing the isolation mount such that the isolation frequency is not near the floor resonant frequency.

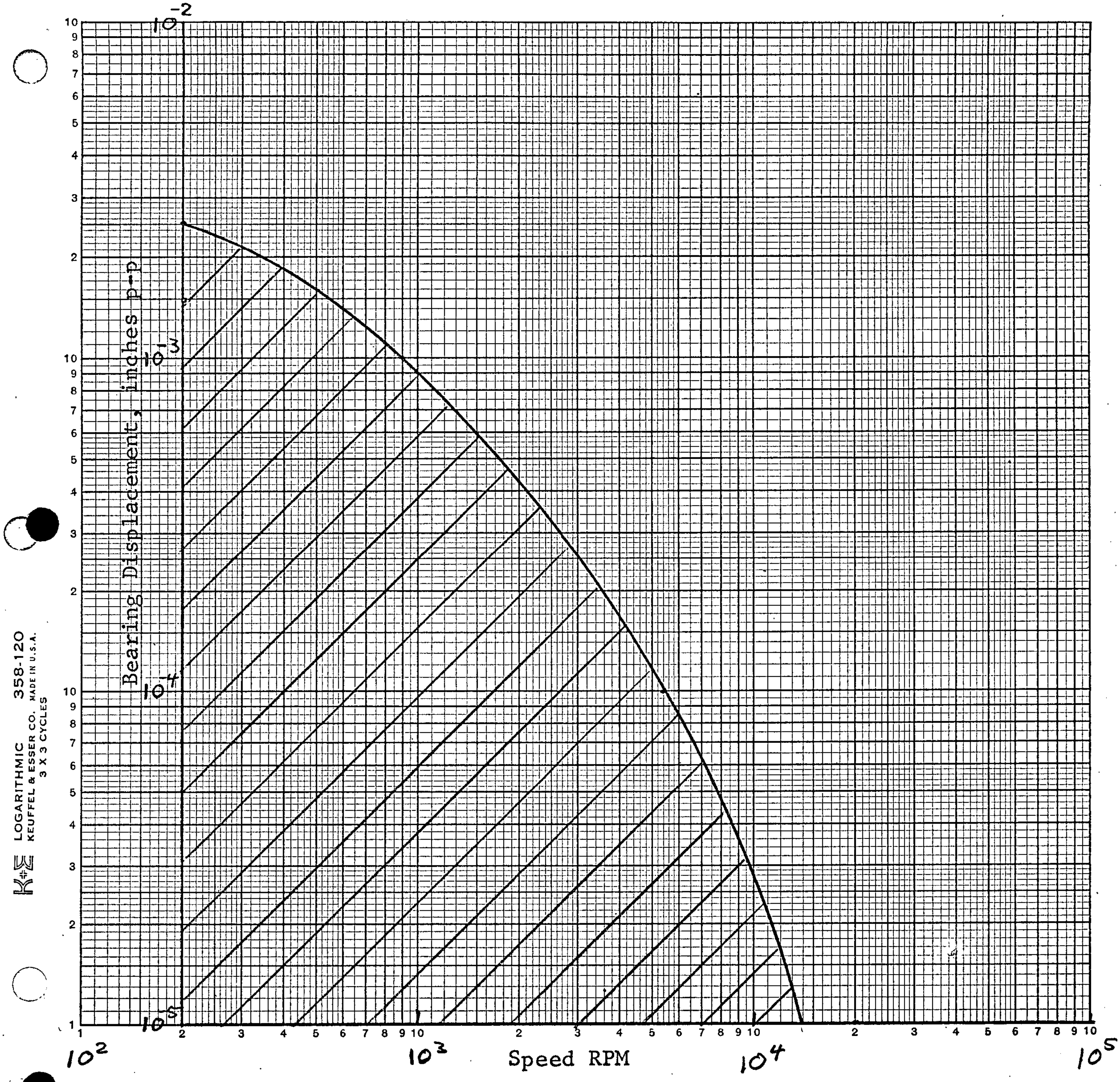
### 5.3 Some Criteria for the Drives, Pumps and Blowers

To minimize the sources of vibration in the outer structure, good balancing of the rotating components of the drives, pumps, and blowers are essential.

The criteria to achieve good balancing are as follows:

a. The permissible unbalance tolerances for these items in terms of millimeter-grams of permissible residual unbalance per kilogram of rotor weight should be in the order of 0.5. This is equivalent to center of gravity displacement of $20 \times 10^{-6}$ in.

b. The bearing displacement versus speed of the rotating equipment should be in the shaded area of Fig.5.



Machinery Vibration Limit for Very Smooth Running

Fig. 5

## 6. DETAIL ANALYSIS OF THE STRUCTURE FOLLOWING PRELIMINARY DESIGN

### 6.1 Analytical Approach

One of the main factors that affect the performance of the submicron measuring system is the understanding of the structural dynamic behavior of the system.

The dynamic response of the major base block will be of chief concern. Block vibration loads will be transmitted to the moving platen and the measuring system as well as the microscope supports. When the preliminary machine design is established, a detail study of the block structure under floor excitations can be made. Dynamic behavior of other structural and mechanical components can also be analyzed to obtain the over-all performance limits of the submicron measuring system.

An existing IBM 7094 computer routine is available for detail analysis of the structural dynamic responses of the major base block. The capabilities of the routine will be described in 6.2.

The performance of the block isolation system should also be well understood. The pneumatic system is a non-linear system. The stability and response characteristics of the servo-control loop should be simulated by an analog or digital computer.

### 6.2 The Computer Program for Structural Dynamic Analysis

An IBM 7094 computer program named LESAR (Linear Elastic Structural Analysis Routine) has been developed recently by Ying-Nien Yu. A brief description of the routine is given below.

LESAR performs static and dynamic analyses of any linear structure which can be idealized as one of the following frameworks:

Type 1 - Three dimensional pin-jointed framework (examples of structures which can be so idealized--spaceframes, membranes, box beams).

Type 2 - Three dimensional rigid-jointed framework (e.g., spaceframes, shells, box beams, curved beams).

Type 3 - Two dimensional rigid-jointed framework, loaded in-plane (e.g., bars, curved planar beams, rings, planar frames).

Type 4 - Two dimensional rigid-jointed framework loaded normal-to-plane, also called a two dimensional grid (e.g., plates, planar frames, arches, beams).

A framework is defined as a system of uniform weightless bars connected together at joints to form a stable structure. At the joints, inertias are lumped and loads are applied. The bending, torsional, and axial stiffnesses of the bars simulate the elastic properties of the corresponding structure. The lumped inertias and loads represent the actual distributed inertias and applied forces.

The framework and its environment can be described by the following quantities:

a. Coordinates of joints

b. Geometry and elastic properties of bars connecting the joints

c. Lumped inertias at joints

d. Restraints at joints

e. Loads applied at joints

Given these quantities as input LESAR will perform computations to provide the following output:

1. Stiffness matrix for the structure

2. Up to 12 Eigenvalues and Eigenvectors (modes and frequencies)

3. Response to loading conditions:

   a. Deflections and internal loads (moments, shears, torques, and axial forces) within the structure for static loads.

   b. Time history of deflections and internal loads for transient excitations.

   c. Deflections and internal loads as a function of frequency for harmonic excitation.

4. Summary of the maximum design loads occurring at joints.

The program is versatile as far as the structural types and excitations are concerned. The only limitation is in the total number of degrees of freedom which is 102. For the base block analysis, Type 4 structure will be used. The maximum number of degrees of freedom is not expected to exceed 80.

The use of this routine will provide structural dynamic analysis of the submicron measuring engine structure to a sophisticated degree. Computer time will normally not exceed 5 minutes per set of input conditions.

In the present study, LESAR was used to analyze the free vibration of the floor slab.

In future structural dynamic analysis of the major base block, the block-support system will be organized as a mechanical system as given in Fig. 6a.

Considering vertical motion only, the following data will be generated:

1. Three rigid body frequency modes (1 vertical, 2 rotational with respect to block surface).
2. Up to 9 elastic modes and frequencies.
3. Static and dynamic deflections and stresses to given loading conditions.

Similar models can be established for motion sideways to yield other rigid body modes, but this is not believed necessary. The stiffness of the block in the plane direction is high, and it is not necessary to carry out the elastic motion computation of the block. Knowing the isolation stiffness horizontally, the rigid body modes can be hand computed without resorting to computer routine.

### 6.3 Servo-Control Loop of the Isolation System

The isolation system stiffness requirement will depend on the dynamic behavior of the major base block. The level recovery time requirement of the system will be derived from the following considerations:

1. The moving platen will be free of oscillations or unstable motions throughout the slew rate range of the measuring engine.
2. Should a two-base-blocks system be chosen, the relative motions of the blocks should be small and the recovery time should be such that the motion disturbances of the eyepieces will be at a practical minimum.

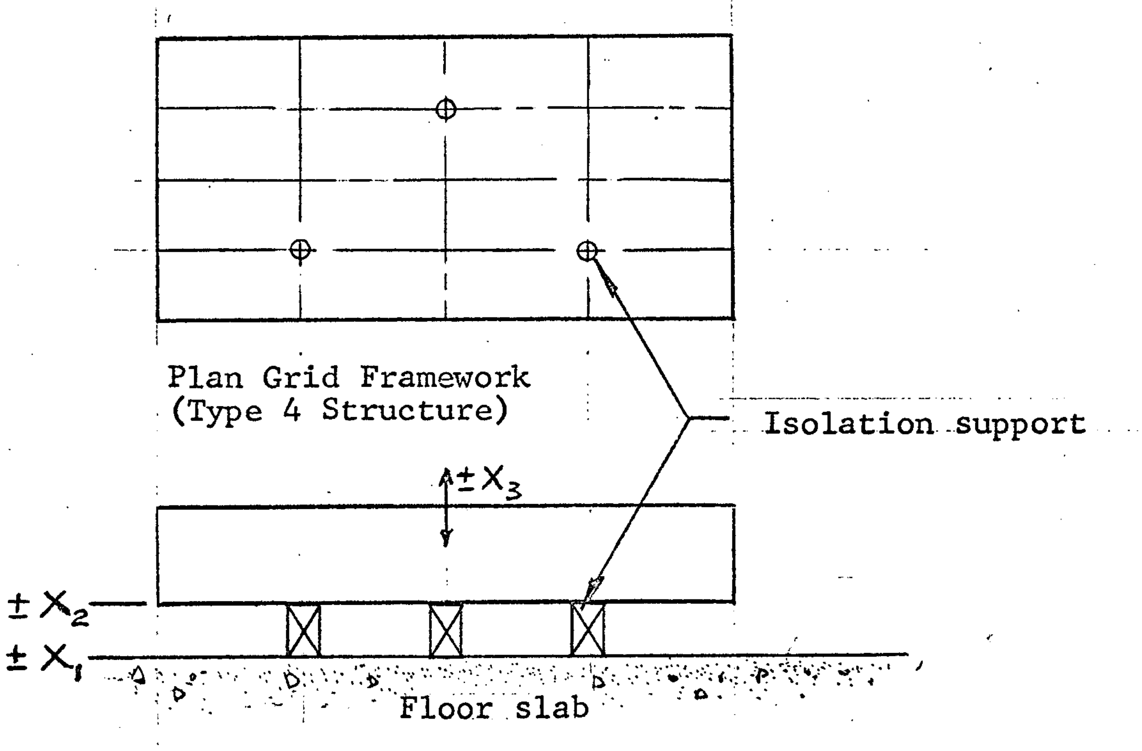
As indicated in 2.4, an 8 cps mount with a 1 to 2 sec. level recovery time may be desirable for the pneumatic isolation system. The position-sensing valve design of SERVA-LEVL would probably not permit the fast response requirements.

In general, a fast response pneumatic system would require an electro-pneumatic controller which actuates a servo valve through error signals received from displacement and/or velocity sensors.

Fig. 6b shows a typical servo-control loop mechanism of the pneumatic isolation system.

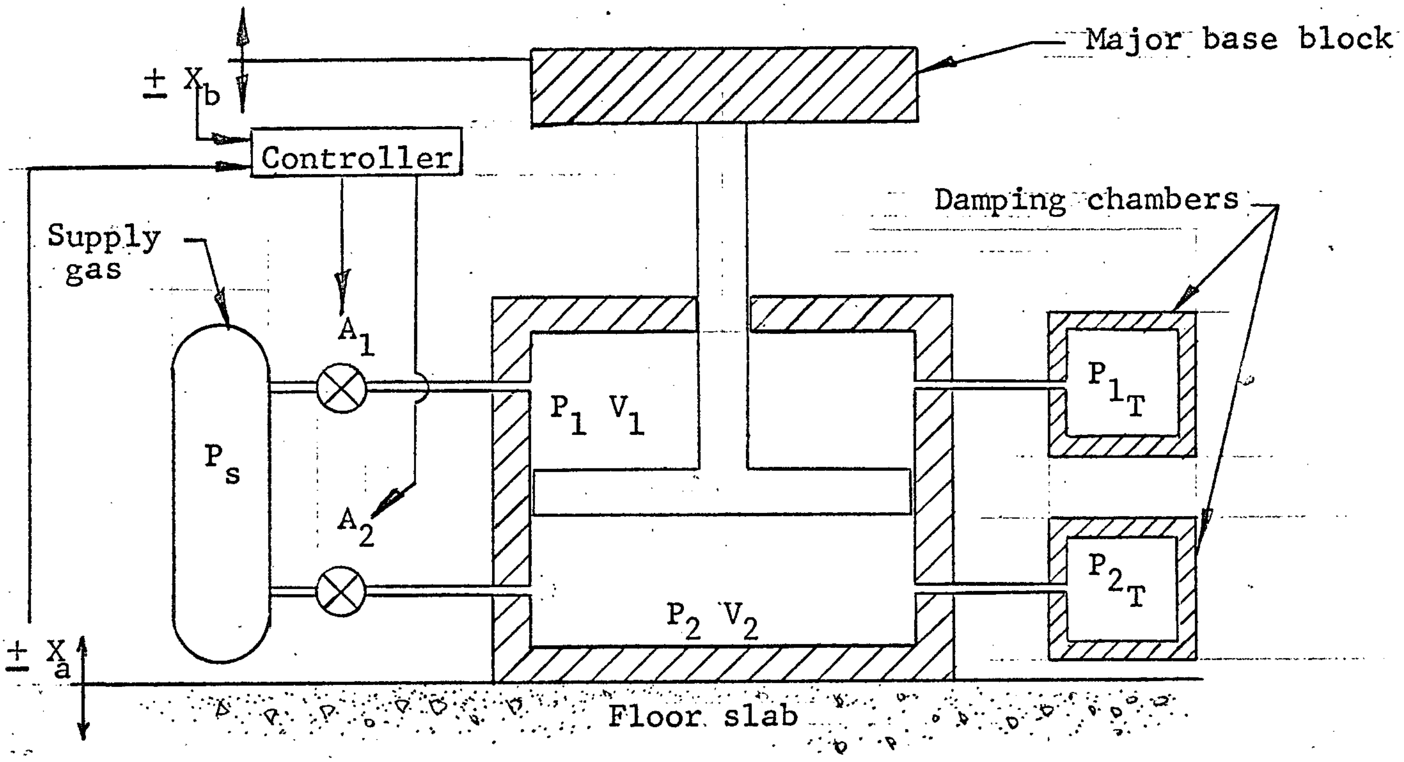
Since the pneumatic transfer functions are non-linear, the servo-control system will involve a set of non-linear differential equations relating the parameters $P_1$, $P_2$, $V_1$, $V_2$, $P_{1_T}$, $P_{2_T}$, $P_s$, $A_1$, $A_2$, $X_a$, $X_b$, and the controller transfer function. The solutions of the equations can be programmed for either a digital or analog computer.

The degree of sophistication in the servo-control loop is dependent on the requirements derived after preliminary structural dynamic design of the complete submicron measuring system.



Major Base Block Mechanical System

Fig. 6a



Pneumatic Isolator Servo-Control System

Fig. 6b

## 7. METHODS OF MEASURING STRUCTURE PERFORMANCE

### 7.1 Floor Dynamic Environmental Data

For detail structural dynamic analysis of the design of the submicron measuring system, the dynamic environmental data of the floor where the system will be located will be needed.

The data of interest are the following:

a. Fundamental resonant frequency
b. Damping characteristics
c. Acceleration environment

These data will be used to determine the isolation requirements of the outer structure and the major base blocks.

The resonant frequency and the damping characteristics of the floor slab can be measured by simply applying a hammer blow (using a lead hammer) and recording of acceleration-time history. In general, the fundamental resonant frequency and the damping ratio can be obtained simultaneously from the acceleration-time history record by measuring the period and calculating the decay rate. However, should the noise level of the floor be higher than the hammer blow or if more than one frequency is excited, the data will be difficult to reduce.

An alternate measuring method is to attach a small vibration exciter (such as an electrodynamic shaker with sweep frequency range of 5 cps to 100 cps) to the floor supplying the excitations. The accelerometer response will again be recorded. At resonant frequency, a survey on adjacent points should be made so that the damping ratio can be determined.

To measure the acceleration environmental condition of the floor a portable acceleration recorder will be employed. The sensitivities of the recorder should be in the order of $10^{-3}$g and $10^{-5}$in. Measurements should be made in all three directions.

### 7.2 Granite Damping Characteristics

Having obtained the floor environmental data, the major block and isolation system design can proceed. One of the factors in choosing the major block construction technique is the damping characteristics of the three types of construction considered.

Damping data are available for cast iron and steel-framed structures. Research will be needed to obtain the material damping data of the granite. In the event that data are either not existant or not complete, a resonant survey of a specimen granite block should be performed on an electrodynamic shaker.

The material damping energy is generally a function of internal stresses; thus in performing the resonant survey, the input acceleration level of the shaker should be comparable to the level of floor acceleration environment.

### 7.3 Tests of Critical Items

It may be necessary to perform vibration level tests on critical items of the submicron measuring system for the following reasons:

a. Items which are either moving parts (rotors, fans, etc.) or parts transmitting motions (gear, linkage, etc.) will generate vibration sources. These parts must be properly balanced or tested to be sure that the frequencies are well separated from other major items of the system.

b. Items which directly affect the measurements of the film coordinates (measuring engine, moving platen, etc.) should also be tested to insure structural integrity.

Specific determination of which items are critical can be made during preliminary design of the submicron measuring instrument.

# APPENDIX

## Free Vibration Analysis of 20'x20'
## Floor Slab by IBM 7094 Computer Program

The floor structure is made of a non-uniform two dimensional plate. Hand calculations of the modes and frequencies will be laborious and inaccurate. A computer analysis of the free vibration of the 20'x20 concrete floor slab has been carried out.

In this Appendix section we shall be concerned with the structural dynamic model of the floor slab. In a subsequent report, the input/output of the computer run and the results of the computation will be discussed.

Figure 7a shows the grid framework of the floor slab. Figure 7b shows a cross section of the floor slab.

A number of assumptions of the floor slab model are given below:

(1) The effective depth of the members adjacent to the columns is 32".

(2) The effective depth of the remaining members is 8".

(3) The concrete strength is 2000 psi which results in $E_c$= 2,000,000 psi.

(4) The distributed weight of the slab is 125 psf.

(5) Members 3-26,11-27, 15-28, and 23-29 in Figure 7a are dummy members simulating the stiffness of the adjacent floor bays. Joints 1, 5, 21, 26, 27, 28, and 29 are restrained in all directions.

The input data will then be organized based on the model. The input and output data of the LESAR run will be tabulated in the future report.

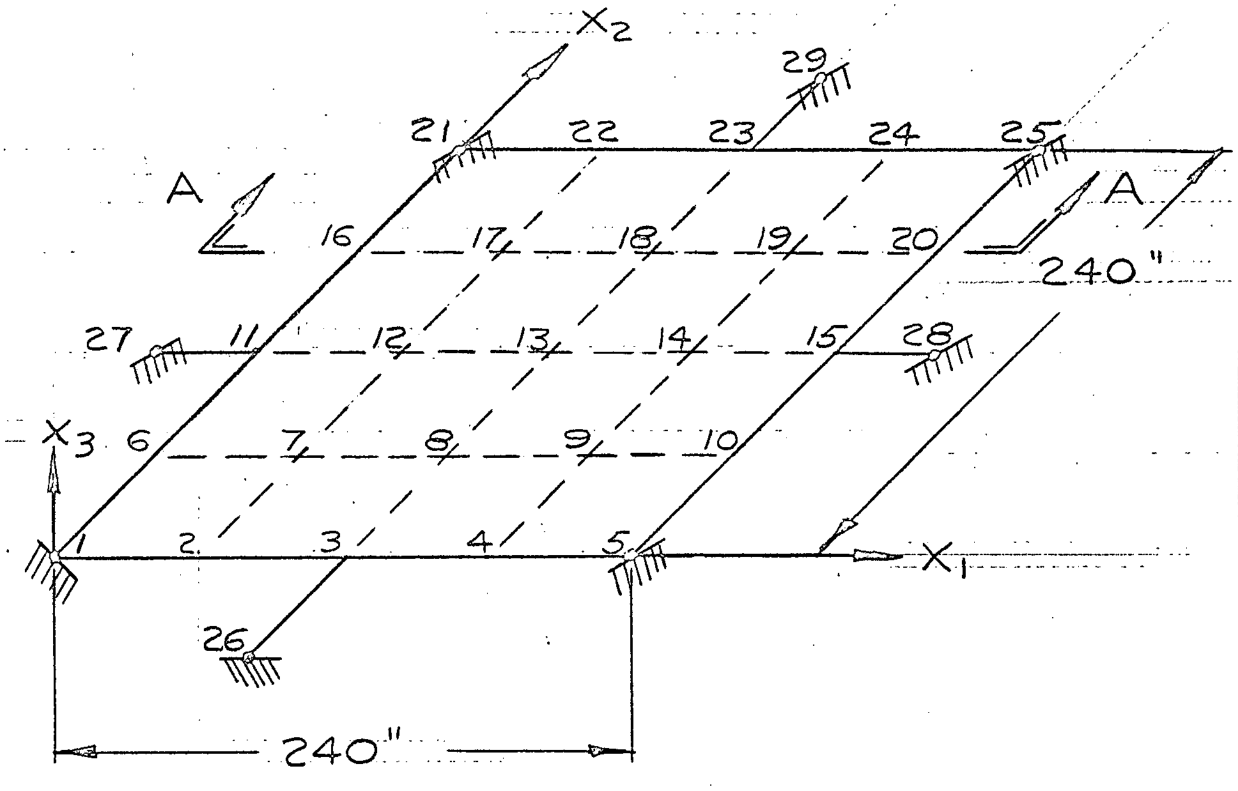


Figure 7a    Floor Slab Grid Framework

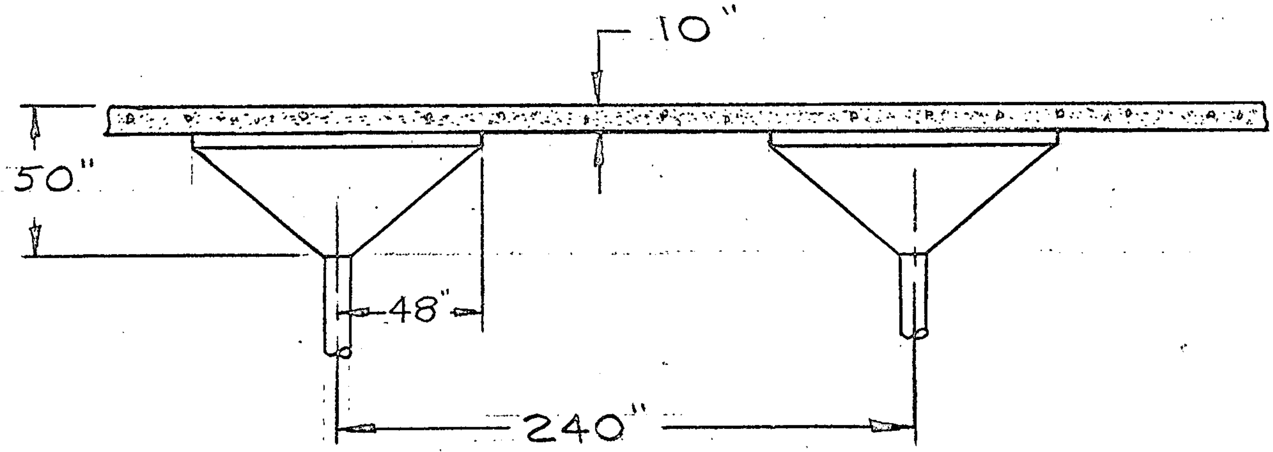


Figure 7b    Section A-A of Floor Slab

998265

[redacted]

25X1

# Task II

## Item 1. Submicron Measurement Error Analysis

## 3rd Preliminary Technical Report