

Analysis and Design of Elevated Foundations for Reciprocating Machines

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Synopsis: Most reciprocating machines are critical to the operation of the plant and, therefore, the supporting structure must be carefully designed to avoid potential undesirable behavior such as attainment of a resonance condition. The designer is then confronted with selecting the best possible techniques in order to accomplish a trouble-free condition. Available engineering analysis tools include theory of vibrations, half-space theory, soil-structural analysis computer programs and rational modeling techniques. This paper summarizes and reviews the steps that must be considered during design of the supporting structure for an elevated reciprocating machine and provides practical guidelines which serve to obtain a realistic and useful design. Four different models are presented and discussed and an example problem is used to illustrate the main features and results of each model. It is concluded that the combination of the best modern scientific tools and modeling technique coupled with practical guidelines yields a reliable structural configuration.

Keywords: foundation; machine bases; structural design; vibration.

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INTRODUCTION

Reciprocating machines are relatively heavy machines that generate vibrating forces of substantial magnitude at low operating frequencies. Reciprocating machines are being manufactured in increasingly larger sizes and oftentimes, require that they be placed at higher than grade elevations to accommodate exhaust piping connections below their bases. Therefore, even though the preferred supporting foundation for reciprocal-type machines has been a flat, relatively thick slab or block at grade, it is now common to design elevated foundations to support reciprocating compressors. Relatively wide and thick blocks bearing directly on the soil or supported by piles and having piers rising from the slab to support the various machine components are often used for elevated reciprocating compressors. This type of structural arrangement is common in larger sized reciprocating machines, specially those used in high pressure piping (for example, low density polyethylene) process applications.

The operating frequencies of reciprocating machines may lie very close to the natural frequencies of the supporting structure in several of the vibrating modes, thus creating possible resonance conditions in the machine-foundation system. The absolute magnitude of vibration amplitude at the resonance condition becomes a controlling criteria because of the closeness of operating and natural frequencies. In order to

achieve acceptable vibration amplitudes it becomes necessary to make the structure very rigid so that the fundamental frequency of the structural system increases to a value above the operating machine frequency. The difference between the fundamental frequency and the operating frequency should be substantial say, a difference of at least 20 but preferably 50 percent. In addition, the ratio of the deflections in the structure caused by the dynamic force to the equivalent static load deflections (magnification factor) as well as the maximum velocity, should be within acceptable limits. This design approach requires that modeling of the foundation structure be as close to the real structure as is consistent with the available tools of analysis so that differences between the model and prototype structure behavior be minimized.

Contrary to other structural systems, dynamic-sensitive structures are difficult to retune and any modification or replacement will result in substantial financial losses not only due to repair costs but also to production losses. Therefore, to guard against uncertainties and to assure proper functioning of the structure, it is imperative that suitable analytical procedures be developed and used which not only provide more accurate dynamic response values but also formally use the latest available analysis techniques. Some of these techniques are lumped parameter model analysis (1-3), soil structure interaction (2-4), elastic half-space theory (3-6) and the latest computer programs available to the profession (7,8). This paper illustrates integration of current knowledge in these areas for the design of elevated structures supporting reciprocating compressors.

Modeling of the real structure, machine and the supporting soil is of critical importance in obtaining results that will approach the actual performance of the combined soil-structure interactive medium (9, 10). A comprehensive analytical approach using four different models of the type of foundation under study are presented and discussed in this paper. Two of the models are developed according to conventional classical procedures while the other two models are analyzed by using current concepts developed in the fields of structural dynamics and soil dynamics. The relative advantages and disadvantages of these models are discussed.

A complete design for a structure supporting a reciprocating machine will include the following steps:

- a) Obtaining all information including machine characteristics, center of gravity, unbalanced forces and moments, excitation frequencies, geometry and special clearance and height requirements.

- b) Geotechnical conditions at the site and evaluation of soil design parameters.
- c) Preliminary design of the supporting structure.
- d) Static and Dynamic analysis.
- e) Checking the proposed design for acceptability. If not acceptable, the design is modified and the analysis repeated.

These steps are considered in detail in reference 10. An example problem illustrating the use of suggested analysis techniques for the design of an elevated reciprocating machine is presented. The paper also includes a discussion of alternate modeling techniques.

FOUNDATION LAYOUT

The design engineer either by himself or with the help of the geotechnical consultant should establish the layout for the foundation structure. There are two common types of foundations used: concrete block footing placed directly on the soil or rock, and concrete footing supported by piles or piers. The preference of one system over the other should be decided by taking into consideration: relative economy, settlement, bearing capacity of the soil, vibration isolation, and the level of the underground water table. Pile or pier-supported footings are the exception and are used only where poor soil conditions are found.

Preliminary sizing and geometrical member arrangement constitutes the initial and often, the most tedious design phase for elevated foundations. Although this preliminary phase is partly based on the experience of the designer, suggested guidelines can be useful in arriving at a satisfactory final design. It should be emphasized that the general guidelines for trial sizing are only useful in the initial phase and are no substitute for a thorough dynamic analysis and check as described later. These general guidelines include the following (10):

1. The designer should carefully analyze equipment size and clearance requirements to assure that sufficient space is allocated to equipment, anchor bolts, piping and clearance for installation, maintenance and operation. That is, physical space limits and requirements should be clearly identified and considered.

2. The bottom of the foundation mat should be placed no higher than the minimum founding depth recommended by the soil consultant. This generally includes considering the location of adequate bearing strata, water table, depth of frost penetration, paving elevation and special local soil conditions. However, in very poor soils, the geotechnical consultant may recommend the use of piles.

3. The top of the footing should be kept a minimum of 0.3 m (1 ft.) above the finished floor or pavement to prevent damage from surface water runoff.

4. The minimum thickness of the foundation slab is seldom less than one-fifth the least slab dimensions or one-tenth the largest slab dimension.

5. Increased damping in the rocking mode is recommended, thus, the width perpendicular to the axis of rocking should be 1.5 times the vertical distance from the bottom of the slab to the machine center of gravity. The additional damping is recommended in view of the degree of uncertainty in some of the design parameters.

6. For large reciprocating machines, the authors recommend that the embedded depth in soil be increased so that 50 to 80 percent of the slab is soil-embedded. This will result in increased damping for all modes of vibration. Increased damping will minimize the amplitude of vibrations even close to the resonance condition and is considered an additional preventative measure to avoid undesirable displacements.

7. The total mass of the structure including the mat should preferably be no less than 5 times the mass of the machine for reciprocating-type machines. For centrifugal machines, the ratio is usually no less than 3.

8. The maximum static bearing pressure for soil-supported foundations should not exceed one-half of the allowable static soil pressure. For pile-supported foundations, the heaviest loaded pile should not carry over one-half of its allowable load.

9. The center of resistance of the soil should be within 5 percent of the side measured parallel to the eccentricity of all superimposed loads for soil-supported foundations. For pile-supported foundations, the centroid of the piles should also be within 5 percent of the side measured parallel to the eccentricity of the superimposed loads.

10. The columns or piers should be checked for individual member resonance with the machine acting frequency. The lowest natural frequency of the columns or piers is approximately given by

$$f_n = \frac{44800 (f'_c)^{1/4}}{\sqrt{pL}}$$

Where f'_c is the concrete strength in psi (1 psi = 6895 Pa), p is the actual column or pier axial stress in psi and is usually in the 40-300 psi range (276-2070 kPa), L is the column or pier height in inches (1 in. = 25.4 mm) and f_n is in cycles per minute. The formula above is obtained by substituting the axial deformation in a column $\delta = pL/E$ into the formula giving the natural frequency of a single degree of freedom system with no damping:

$$f_n = (g/\delta)^{1/2}/(2\pi) \text{ with } E = 57000 (f'_c)^{1/2} \text{ psi and } g = 386.4 \text{ in/sec/sec.}$$

MODELING OF IDEALIZED SYSTEM

A sketch for a proposed foundation structure is shown in Figure 1 and may be represented by four models. Models A and B are conventional engineering practice models, while Models C and D are models which give a more realistic representation of the prototype system behavior.

Model A

This model consists of a single-lumped mass supported by linear and rotational springs. The total mass is equivalent to the sum of the masses of machines, piers and the foundation. The entire system is assumed to be a perfectly rigid body (block-type foundation) resting on soil and, therefore, the elastic-half-space theory is applicable. An idealized model for the system is shown in three parts, Figure 2 (a), (b) and (c), each representing independent modes. These are the only possible modes of vibration because the excitation forces are acting in the given direction.

Model B

This model only considers the superstructure and assumes that the mat slab is massive and, therefore, the piers in Figure 1(c) will function as cantilevers having full fixity at their base, see Figure 3. The superstructure is modeled as a multi-lumped system with each mass having three linear degrees of freedom. Excitation forces are applied at the joints. The influence of the soil is not considered, except for proportioning of the slab based on an allowable bearing capacity. The natural frequencies and response values for individual members of the superstructure are obtained in this model.

Model C

This model is an improvement over Model A, see Figure 4. The model includes a two lumped-mass system with six possible degrees of freedom for each mass. The mass corresponding to the upper half of the superstructure plus machines is lumped at joint 1 while the mass corresponding to the lower half of the superstructure and the mat slab is lumped at joint 2. The element connecting the two masses represents the equivalent stiffness of the superstructure. The soil spring constants are attached to each of the joints and represent the equivalent six soil stiffnesses supporting the mass at joint 2. This model assumes that the superstructure and the foundation are rigid and their masses and stiffnesses can be represented by equivalent lumped parameters (lumped masses and springs). This model provides overall information with regards to the lowest natural frequencies and also yields the response values at the machine attachment point and at the foundation levels which are the usual points of interest. However, the model does not provide complete dynamic results at various points of machinery support and foundation level.

Model D

This model is an improvement in relation to the previous models, not only with respect to degree of reliability of results but also with regards to availability of sufficient information at all points of interest. The approach is based on lumping the masses at the nodes. The foundation slab is modeled using finite element procedures and is supported by soil-springs at the node points, see Figure 5. There are other methods which account for the stiffness of the foundation slab and provide equally good results, such as "Equivalent Frame Method" as described in the American Concrete Institute Building Code, Requirements for Reinforced Concrete (ACI-318). The interaction of the soil's stiffness with the foundation structure is

obtained through the use of the elastic half-space theory. The calculation of stiffness for the structure is quite complex and is generally done through the use of computer programs. This model yields comprehensive results not only about the dynamic behavior of the structure but also of the foundation (9). In addition, through various mode shapes, it is possible to determine the distress situation that may occur at various points of interest. This model is highly recommended for large reciprocating machines supported on elevated piers and is used in the example problem presented in this paper.

STATIC AND DYNAMIC ANALYSIS

After a trial configuration is selected, the structure is modeled as discussed above, an actual static and dynamic analysis is performed and the predicted behavior is compared to certain admissible performance guidelines. These may include: design for the equivalent (conservative loads), natural frequencies of the structure for possible resonance, maximum amplitude of deformation due to the application of the dynamic loads and checking the structure and foundation for the effect of the dynamic loads.

The availability of electronic digital computers having great calculating speed and analytical power has resulted in substantial advancement in designing complex machine foundations. Structural software packages predict the natural frequencies, the deformations and the forces in the structure (7, 8). In many cases, the structure or soil parameters are not exactly known. For example, the shear modulus of the soil, G , may vary by $\pm 25\%$ or more at points below the foundation even for a careful well-designed field investigation (10). The effect of this variation in G may be considered by changing one input parameter and re-running the program. Thus, the soil-structural behavior may be predicted for possible ranges of G . This is a most convenient feature of computer use. A flow chart summarizing the steps that occur during a computer analysis is given in Figure 6 and is typical of all structural software packages commercially available. The flow chart depicts specific stages using the ICES-STRU DL software package (7, 8).

ACCEPTABILITY AND FINAL DESIGN

The computer results must be checked against certain acceptable performance criteria. These include:

1-Static Conditions

- a) The maximum soil pressure or pile load should not exceed 50% of the allowable capacity.
- b) Settlement must be uniform, i.e., the center of gravity of all static loads should be within 5% of any linear dimension (and preferably coincide) with the footing or pile group centroid.
- c) The combined center of gravity of the static plus dynamic loads should be within 0.5% of any linear dimension from the footing or pile group centroid. The axis of rocking should coincide with a principal axis of the footing. The total settlement should be less than the permissible deflection for attached piping (very small for high pressure piping).

2-Dynamic Conditions

- a) Resonance - The acting machine frequencies should differ by at least $\pm 20\%$ and preferably $\pm 50\%$ for reciprocating machines from the 10 lowest natural frequencies of the structure.
- b) Maximum Vibration Amplitude - The maximum amplitude must fall in an acceptable range for the acting machine frequencies, for example in zones A or B of Figure 3-3 of Reference 10 or Figure 10-2 of Reference 4. A number of other acceptable vibration amplitude charts are available (10-13).
- c) Velocity - The maximum resulting velocity is also checked. The velocity may be calculated from

$$v = 2\pi f \text{ (cps)} \times \text{(displ. amplitude)}$$

Acceptable velocities are given in Table 10-4 of Reference 4 and Table 3-2 of Reference 10. Maximum horizontal peak velocities in excess of 4 mm/sec. (0.16 in./sec.) are generally, to be avoided.

- d) Acceleration - The maximum resulting acceleration as calculated from

$$a = 4\pi^2 f^2 (\text{cps}) \times (\text{displ. amplitude})$$

is checked, for example, using Figure 3-3 of Reference 10 or Figure 10-2 of Reference 4. However, if the maximum vibration amplitude and velocity (b) and (c) above are acceptable, an acceleration check is not required.

- e) The ratio of the deflections caused by the dynamic forces to the deflections caused by the equivalent static loads (the magnification factor) should be less than 1.5.

EXAMPLE PROBLEM

The elevated structure shown in Figure 1 is analyzed using the four previously described Models A-D. The structure and soil parameters and the machine unbalanced dynamic forces, see Figures 7 and 8 required for the analysis of these models are:

Soil:	G (shear modulus)	=	165.5 MPa
	ν (Poisson's ratio)	=	0.40
	γ (density)	=	179.6 kg/m ³
Structure:	E (mod. of elast.)	=	21525.4 MPa
	ν (Poisson's ratio)	=	0.17
	γ (density)	=	245.0 kg/m ³
Machine:	Unbalanced dynamic forces (see Figures 7 and 8):		
	F _y (vertical) Primary:	6619 sin wt N	
	F _x (horizontal) Primary:	39746 cos wt N	
	F _x (horizontal) Secondary:	102970 cos 2wt N	
	M _y (vertical) Primary:	37412 sin wt N-m	
	M _x (horizontal) Primary:	444938 cos wt N-m	
	M _x (horizontal) Secondary:	80395 cos 2wt N-m	

This soil data is obtained from a site investigation normally provided by the geotechnical consultant while the machine unbalanced forces are given by the machine manufacturer. The value of ω (excitation frequency) is 20.94 rad./sec. These dynamic forces are applied in the appropriate directions at a point of Model A as shown in Figure 2. The dynamic forces are applied at the shaft axis above joint 2 in Models B and D. In Model C, all of these forces are applied at the location of mass at joint 1. The other parameters for the individual models are given below. These parameters include the effect of depth of foundation embedment in soil calculated by formulas given in Reference 3.

Model A (see Figure 2)

Vertical (Y-direction):	m	$= 107253 \text{ kg}$
Excitation	k_y	$= 8.4671 \times 10^9 \text{ N/m}$
	D_y	$= 1.03$

Lateral (x-direction), Rocking (Ψ -direction), and Pitching

(ϕ -direction) Excitations:	m	$= 107253 \text{ kg}$
	I_Ψ	$= 2711525 \text{ kg-m}^2$
	I_ϕ	$= 1266095 \text{ kg-m}^2$
	k_L	$= 6.653 \times 10^9 \text{ N/m}$
	k_Ψ	$= 5.0573 \times 10^{11} \text{ N-m/rad.}$
	k_ϕ	$= 2.9744 \times 10^{11} \text{ N-m/rad.}$
	D_L	$= 0.68$
	D_Ψ	$= 0.46$
	D_ϕ	$= 0.30$
Torsional Excitation:	I_θ	$= 2635161 \text{ kg-m}^2$
	k_θ	$= 4.4934 \times 10^{11} \text{ N-m/rad.}$
	D_θ	$= 0.26$

Parameters relating to mass (m , I) are calculated at the centroid of the foundation. Spring constants terms (k), and damping ratios (D) were evaluated from the elastic-half-space theory (2-4) (developed for a single degree of freedom equivalent lumped parameter system) by using the soils information.

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Model B (see Figure 3)

Spring constants and lumped masses are calculated by the computer internally using the computer program ICES-STRUDL (7, 8) and a damping ratio of 0.02 (normally assumed for concrete construction) is used for all modes of oscillation.

Model C (see Figure 4)

For Joint 1, (mass of half height of piers and machine)

$$\begin{aligned}m_1 &= 29207 \text{ kg} \\I_{\psi 1}, I_{\phi 1}, I_{\theta 1} &= \text{nominal values}\end{aligned}$$

For Joint 2, (half height of piers and entire mat slab)

$$\begin{aligned}m_2 &= 78046 \text{ kg} \\I_{\psi 2} &= 1622273 \text{ kg-m}^2 \\I_{\phi 2} &= 724468 \text{ kg-m}^2 \\I_{\theta 2} &= 2346741 \text{ kg-m}^2\end{aligned}$$

Spring constants of member 1,

$$\begin{aligned}k_{x1} &= 1.0930 \times 10^{10} \text{ N/m} \\k_{y1} &= 3.3951 \times 10^{10} \text{ N/m} \\k_{z1} &= 1.5761 \times 10^{10} \text{ N/m} \\k_{\psi 1} &= 1.2870 \times 10^{11} \text{ N-m/rad.} \\k_{\phi 1} &= 1.7463 \times 10^{11} \text{ N-m/rad.} \\k_{\theta 1} &= 1.1554 \times 10^{11} \text{ N-m/rad.}\end{aligned}$$

Spring constants for member 1 are calculated by applying unit forces and moments in Model B. Spring constant values at joint 2 for the soil are calculated using the elastic-half-space theory assuming an equivalent lumped parameter system. In the model, these springs have been represented by three linear springs at newly created joints 3, 4, 5 and 6 so that the model can then be analyzed using the computer program. The value of

the springs and distances between the joints are calculated using the following equations:

$$\begin{aligned}k_m &= 2k_e e^2 \\k_1 &= 2k_e \\e &= \sqrt{k_m/k_e}\end{aligned}$$

Where k_m and k_e are the soils rotational and linear spring stiffnesses, respectively, and k_e is the stiffness of equivalent linear springs acting at a distance e from the centroid of pressure. The values of equivalent springs used in the model are:

At Joints 3, 4:	$k_{x3} = k_{x4}$	$= 2.7211 \times 10^{10} \text{ N/m}$
	$k_{y3} = k_{y4}$	$= 2.5400 \times 10^{10} \text{ N/m}$
	$k_{z3} = k_{z4}$	$= 6.5832 \times 10^9 \text{ N/m}$
At Joints 5, 6:	$k_{x5} = k_{x6}$	$= 1.2160 \times 10^9 \text{ N/m}$
	$k_{y5} = k_{y6}$	$= 2.5400 \times 10^{10} \text{ N/m}$
	$k_{z5} = k_{z6}$	$= 3.2792 \times 10^{10} \text{ N/m}$

and the values of e_x and e_z are 10.93 and 8.04 meters, respectively. Members 2, 3, 4 and 5 have been assumed to have negligible mass with infinite stiffnesses since their sole purpose is to convert the rotational and linear stiffnesses into concentrated linear stiffnesses. Damping ratios of 0.10 are used throughout as an equivalent average value for all modes of oscillation.

Model D (see Figure 5)

The stiffness matrix and the masses are lumped at the node points and are calculated internally by the computer program (7, 8) while the soil spring constants at the supports are calculated using the elastic-half-space theory (2-4). The total value of linear spring constants is distributed to the individual node points in contact with the soil in proportion to their contact area. For non-rigid mats, alternate distributions have been suggested (10). The moments of these spring constants are then calculated at the centers of pressure for comparison and adjustment with the torsional spring constants calculated

for the equivalent lumped system. An average damping ratio of 0.10 is used for all modes of oscillation. The mass of the equipment is added at joints 1, 2, 3, 5, 7 and 11.

ANALYSIS AND DISCUSSION OF RESULTS

Analysis of Model A was performed manually using the following formulas which may be derived from the equation of motion of a single-degree-of freedom system (1, 4):

$$f_n \text{ (natural frequency, rpm)} = (60/2\pi)\sqrt{k/m}$$

$$A \text{ (max. displ. amplitude)} = \frac{F_0/k}{\sqrt{(1-r^2)^2 + (2Dr)^2}}$$

Where m , k and D are the mass, spring constant and damping ratio parameters of the model, F_0 is the amplitude of the excitation force, and r is the ratio of the excitation frequency to the natural frequency. The solution for Models B, C and D was obtained using the ICES-STRUDL computer program (7, 8) in view of the complexity of the problem, the amount of calculation work, and the desired accuracy of the results. However, the solution for Model C can be obtained manually by decoupling some of the modes and eliminating the rotational degree of freedom at joint 1 (which does not appear to affect the displacement response). Four independent models could be considered as follows: (1) vertical direction with two degrees of freedom at joints 1 and 2 in the y direction; (2) torsional θ -direction with a single degree of freedom at joints 1 and 2 in the z direction and a single degree of freedom in the Ψ -direction at joint 2; (4) longitudinal and pitching directions with two degrees of freedom at joint 2 in the Φ direction. If these four alternate models are developed, then the representation of joints 3, 4, 5 and 6 (Figure 4) becomes unnecessary, the joints are eliminated and the actual soil springs are applied at joint 2. The equations of motion for these four alternate models can be written using the dynamic equilibrium equations, (1, 10). The solution of (1) and (2) above for the natural frequencies and response values can be obtained through formulas available in theory of vibrations text books (1). Solution for (3) and (4) can be found using some of the procedures developed for multiple degrees of freedom systems. The natural frequencies

and mode shapes are calculated by an iterative procedure (Stodola-Vianello method) and the response values are found using model superposition in reference 10. Response values must be calculated separately for each group of excitation forces having a given acting frequency. The resulting response for each group of dynamic forces (all forces in each group having the same frequency) must be combined by using the principle of superposition to obtain the final result. It should be noted that the total maximum response is obtained by adding the maximum absolute values of the response due to each group of equal acting frequency forces, irregardless of the time at which these maxima occur. This is a conservative approach. Calculation work to obtain the total response requires the use of a scientific electronic hand-held calculator with an accuracy of at least ten significant digits.

Computer programs available to the profession are quite versatile and not only provide the natural frequencies (eigenvalues) and the mode shapes (eigenvectors) or response values, but also give the forces and moments experienced by the structure due to the application of dynamic forces. The first twenty modes were included in the computer program in determining the response values although ten could have been used with little difference in the results. Note that only the first ten modes are considered for resonance condition checking. The time increment used was 0.0125 sec. (14) which is one-twelfth of the secondary smallest period (0.15 sec). Response values were calculated for a total time span of 1.5 sec. which includes five cycles at the primary operating frequency (200 rpm) and 10 cycles at the secondary operating frequency (400 rpm). The significant results in this example are the natural frequencies, mode shapes and the displacement response at the machinery support level and at the foundation level. The results for natural frequencies and vibration response are presented in Tables I and II, respectively.

Discussion of Table I

1. Fundamental Frequency - Model D gives the lowest value (658 rpm) for the fundamental frequency followed by Model C (693 rpm), Model A (755 rpm) and Model B (2707 rpm). Model D yields the lowest natural frequency because the correct deflected configuration of the structure is used and this, in turn, means that the system possessing the least amount of energy has been considered. Since fewer masses (thus fewer degrees of freedom) have been used in Model C, additional modeling errors show up in the results. For Model A, only one mass is used with a single degree of freedom and this results in larger errors and thus gives higher frequency results. However, in Model B which used a larger number of masses and used the same computer program as for Model D, the results come out much higher than in the other

models. This phenomenon can be attributed to the lack of interaction of the foundation mass with the superstructure, that is, since the superstructure was fixed at its mat end, the structural stiffness is much higher than when the mat and soil flexibilities are considered.

2. Longitudinal Mode (z-direction) - The first fundamental frequency occurs in the z-direction in all models except Model B. In Model B, it shows up as the third mode (3,339 rpm) indicating that the superstructure is considerably stiffer in the z-direction. However, it should be noted that this mode is coupled with the pitching (Φ -direction) mode of the system which does include the foundation. The foundation's soil resistance is lower in pitching than in rocking (x-direction) for this rectangular foundation, therefore, the order of the modes is affected as noted in Model D. This behavior does not occur in Model B, because of the lack of interaction of the foundation (and soil mass) with the superstructure. This mode is significant because the natural frequency is closer to the excitation frequencies. However, the dynamic response is not influenced by this factor, because no excitation force is acting in this direction.

3. Lateral Mode (x-direction) - This is the second mode in Model C and D while it is the first mode in Model B. Models C and D show natural frequencies of 701 rpm and 717 rpm, respectively, while Model B shows 2707 rpm. The natural frequency of Model C is comparable with Model D, thus indicating that the assumptions made in Model C are reasonably correct. This mode is also significant due to the fact that there are dynamic forces acting in this direction. In Model A, there is no distinction between lateral and longitudinal modes, (i.e., the natural frequency is 755 rpm in both cases), due to the assumptions made in the elastic-half-space theory.

4. Vertical Mode (y-direction) - This is the third mode and is obtained in Models C and D at frequency values of 841 and 880 rpm, respectively, and is also the second mode in Model A with frequency of 857 rpm while in Model B it is the 8th mode with frequency of 5592 rpm. This mode is significant because the dynamic force acts in this direction. Therefore, this mode makes a significant contribution to the total response in Models A, C, and D, while in Model B the contribution of this mode is negligible because the natural frequency (5592 rpm) is several times larger than the excitation frequencies (200 and 400 rpm). It should be noted that in Model A and C, the frequencies are very close to the frequency obtained in Model D, thus indicating that a correct representation has been made for these models.

5. Pitching Mode (Φ -direction) - This mode is a rotation about the x-axis, and is significant because the dynamic moment acts in this direction. This is the fourth mode and is attained in Models A, C and D at frequencies value of 1,317, 1,016 and 981 rpm, respectively. The higher values in Models A and C can be attributed to probable errors made in calculating the equivalent lumped parameters. This type of mode is coupled with the vertical mode (y-direction) for Model C and occurs as the sixth and seventh modes. In Model D, it is coupled with the lateral mode (z-direction) and also shows up in the fifth mode (1,038 rpm). In Model B, the pitching mode occurs in the fourth mode (4,165 rpm), again in the seventh mode (5,371 rpm) and finally in the tenth mode (6,525 rpm). This mode occurs uncoupled in Model A due to its mode independent nature.

6. Rocking Mode (Ψ -direction) - This mode is a rotation with a Ψ -angle about the z-axis. It is a higher order mode and is found to be unimportant for Models B, C and D. In Model A, this mode has a natural frequency of 1,478 rpm and is coupled with the excitation in the longitudinal direction.

7. Torsional Mode (Θ -direction) - This mode is a rotation with a Θ -angle about the y-axis and is significant because a dynamic moment is acting in this direction. Because of the multi-degree lumped mass characteristic used in Models B and D, this mode is difficult to define. For Model A, it shows up as the third mode (1,259 rpm) and Model C it is the fifth mode (1,262 rpm). The value of natural frequencies is the same in both models, because this mode exhibits independent behavior, as discussed above in the analysis techniques description. The same also occurs for the vertical mode, where Models A and C exhibit natural frequencies of 857 and 841 rpm, respectively, in the y-direction.

8. Resonance Condition - One of the basic requirements in foundations for reciprocating compressors is to attain a natural frequency which is higher than the excitation frequency by at least 50%. This was achieved in all the models. The soil under the foundation affords considerable damping but is not effective at this frequency ratio, that is, damping becomes significant at frequency ratios approaching 1. However, there may be some inherent errors in the results due to uncertainties as to the correct value of the soil parameters, which may result in a reduced fundamental frequency. Under those conditions, the influence of damping is significant and helps to compensate for possible calculation errors.

Discussion of Table II

The sums of maximum displacement response in the three linear directions (x, y, and z) are listed in Table II (a). The response is found at the lumped masses for both excitation frequencies in Models A and C while for Models B and D, the response occurs either at the base support or at the machine level. These values are then compared with allowable values established by industry (Figures 3-3 or 10-2 of Reference 10 and 4, respectively). In some engineering offices a comparison is made with regards to the resulting velocity to determine the classification of machine operation (Table 3-2 of Reference 10 or Table 10-4 of Reference 4). Two sets of acting frequencies, i.e., primary and secondary are involved in this example problem, therefore, the resulting value is obtained by the root mean square (RMS) method. These velocity RMS values are listed in Table II (b).

1. Displacement and Velocity at Base Support (Foundation Level) - By following the criteria given above for general limits of vibration amplitude at a particular frequency, the response values for all models are below the acceptable limits for machine and machine foundations and fall in a zone "easily noticeable to persons" for both of the excitation frequencies which, according to industry standards, is acceptable. The displacement response is maximum in the lateral direction because the dynamic forces are acting in that direction. Model C shows the largest response value 0.00212 in. or 0.054 mm) which is slightly higher than in Model D (0.00189 in. or 0.048 mm). However, Model A shows a value of 0.00082 (0.021 mm) because the value of D_H (calculated damping ratio in the elastic-half-space theory)¹¹ used was 0.68 while for Model C and D an equivalent average value of 0.10 was used. Model B does not provide any displacement amplitude value, since the model bases were considered fixed. Comparison of velocity amplitude with the values on "General Machinery-Vibration Severity Data" as discussed above, shows that Models C and D are classified as "good" and Model A is classified as "very good," which is also acceptable according to industry standards.

2. Displacement and Velocity at Machinery Level - The magnitude of vibration amplitude which occurs at this level is generally a cause of concern to engineers in foundation design. A number of industry standards may be followed for design acceptance. According to one limitation, zone B of Figure 10-2 of Reference 4 or Figure 3-3 of Reference 10, the horizontal amplitude of vibration should not exceed 0.003 in. (0.076 mm) for 200 rpm and 0.0025 in. (0.064 mm) for 400 rpm. This requirement is met by all models except Model C. If the general machinery-vibration-severity data is followed, then the

allowable range for "fair" operation is 0.08 in/sec to 0.16 in/sec (2.0 mm/sec to 4.0 mm/sec). This limit is met by all the models. Some industry standards consider 0.16 in/sec (4.0 mm/sec) as the upper desirable value of maximum velocity for reciprocating compressors.

Since the proposed configuration meets all static and dynamic design criteria, the structure is acceptable.

CONCLUSIONS

Four models representing an elevated pedestal foundation were dynamically investigated. The investigation identified certain relative advantages for each model representation as follows:

Model A - This model representation may only be used for estimating the lowest natural frequency (fundamental) of the machine-structure system and can be used without the aid of computer programs. All other results are erratic and unsatisfactory. Thus, under certain conditions, this model may be useful and inexpensive in terms of calculation time and computer cost.

Model B - This model provides information concerning the natural frequencies of individual members. However, resulting fundamental frequency and response values are incorrect because the effect of the mat and soil is neglected. Thus, the model is judged unsatisfactory for the calculation of the natural frequencies and amplitude of vibration.

Model C - This model provides overall information with regards to the lowest natural frequency of all basic modes and also gives the displacement response at some points of interest. The model gives somewhat higher than true response values. Also, dynamic results are not available at many machinery support points and hence, this model is not recommended for a large size reciprocating compressor. Model C, however, consumes only 5% of the computer time required for the solution of Model D and may, therefore, be considered if computer time is limited.

Model D - This model yields comprehensive results not only about the dynamic behavior of the structure but also of the foundation. In addition, through various mode shapes, it is possible to detect possible distress situations that may occur at various points of interest. This model is highly recommended for large-size compressor machines supported by an elevated foundation such as the example problem considered here.

Structures and foundations supporting reciprocating compressors may be properly modeled and analyzed when a medium to large size computer is available having dynamic analysis and finite element capabilities. State-of-the-art concepts such as half-space theory, depth of embedment effects, and damping of all types, may be included in this recommended model representation. However, for preliminary design or for situations where the computer is not available or is too costly to operate, Model A may provide sufficient information and may be used.

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Table I--Natural frequencies and modes (see Figures 2-5 for joint locations)

Mode No.	MODEL A		MODEL B		MODEL C		MODEL D	
	Freq. (rpm)	Mode Type	Freq. (rpm)	Mode Type	Freq. (rpm)	Mode Type	Freq. (rpm)	Mode Type
1	755	Latl. or Long.	2707	Latl. @ Jt. 11,12	693	Long. @ Jt. 1	658	Long. @ Jt. 11
2	857	Vertical	3210	Latl. @ Jt. 8,9	701	Latl. @ Jt. 1	717	Latl. @ Jt. 11
3	1259	Torsional	3339	Long. @ Jt. 11	841	Vert. @ Jt. 1	880	Vert. @ Jt. 9
4	1317	Pitching	4165	Pitching	1016	Pitching	981	Pitching
5	1478	Rocking	4292	Multi-Coupled	1262	Torsional	1038	Pitching
6			5065	Multi-Coupled	1304	Pitching	1070	Multi-Coupled
7			5371	Pitching	2357	Pitching	1354	Multi-Coupled
8			5592	Vert. @ Jt. 1	3025	Rocking	1584	Multi-Coupled
9			5302	Multi-Coupled	3890	Vert. @ Jt. 1	1876	Multi-Coupled
10			6525	Pitching	35464	Rotation of Jt. 1	1922	Multi-Coupled

NOTE: CORRESPONDENCE OF AXES DIRECTIONS:

Lateral → x-axis

Longitudinal → z-axis

Pitching → Φ , rotation @ x

Vertical → y-axis

Rocking → Ψ , rotation @ zTorsional → Θ , rotation @ y

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Table II (a)--Displacement response at points of interest

DIRECTION OF OSCILLATION	LOCATION	MAXIMUM DISPLACEMENT AMPLITUDE, INCHES $\times 10^{-3}$ (mm)			
		MODEL A	MODEL B	MODEL C	MODEL D
Lateral (X)	Base Supp	0.82 (0.021)	*	2.12 (0.054)	1.89 (0.048)
	Machine Level	1.31 (0.033)	0.12 (0.003)	4.30 (0.109)	2.49 (0.063)
Vertical (Y)	Base Supp	0.03 (0.001)	*	0.04 (0.001)	1.45 (0.037)
	Machine level	0.72 (0.018)	0.04 (0.001)	0.05 (0.001)	1.12 (0.028)
Longitudinal (Z)	Base supp	not calcul.	*	0.04 (0.001)	1.06 (0.027)
	Machine level	not calcul.	0.36 (0.009)	0.10 (0.003)	1.62 (0.041)

* PIER BASES CONSIDERED FIXED

Table II (b)--Velocity response at points of interest
(calculated by the RMS Method)

DIRECTION OF OSCILLATION	LOCATION	MAXIMUM VELOCITY AMPLITUDE, INCHES/SEC. (mm/sec.)			
		MODEL A	MODEL B	MODEL C	MODEL D
Lateral (X)	Base supp	0.0298 (0.757)	*	0.0671 (1.740)	0.0652 (1.656)
	Machine Level	0.0474 (1.204)	0.0046 (0.117)	0.1556 (3.952)	0.0966 (2.454)
Vertical (Y)	Base supp	0.007 (0.018)	*	0.0008 (0.020)	0.0537 (1.364)
	Machine level	0.0247 (0.627)	0.0011 (0.028)	0.0010 (0.025)	0.0295 (0.749)
Longitudinal (Z)	Base supp	not calcul.	*	0.0008 (0.020)	0.0307 (0.780)
	Machine level	not calcul.	0.0090 (0.229)	0.0042 (0.107)	0.0441 (1.120)

* PIER BASES CONSIDERED FIXED

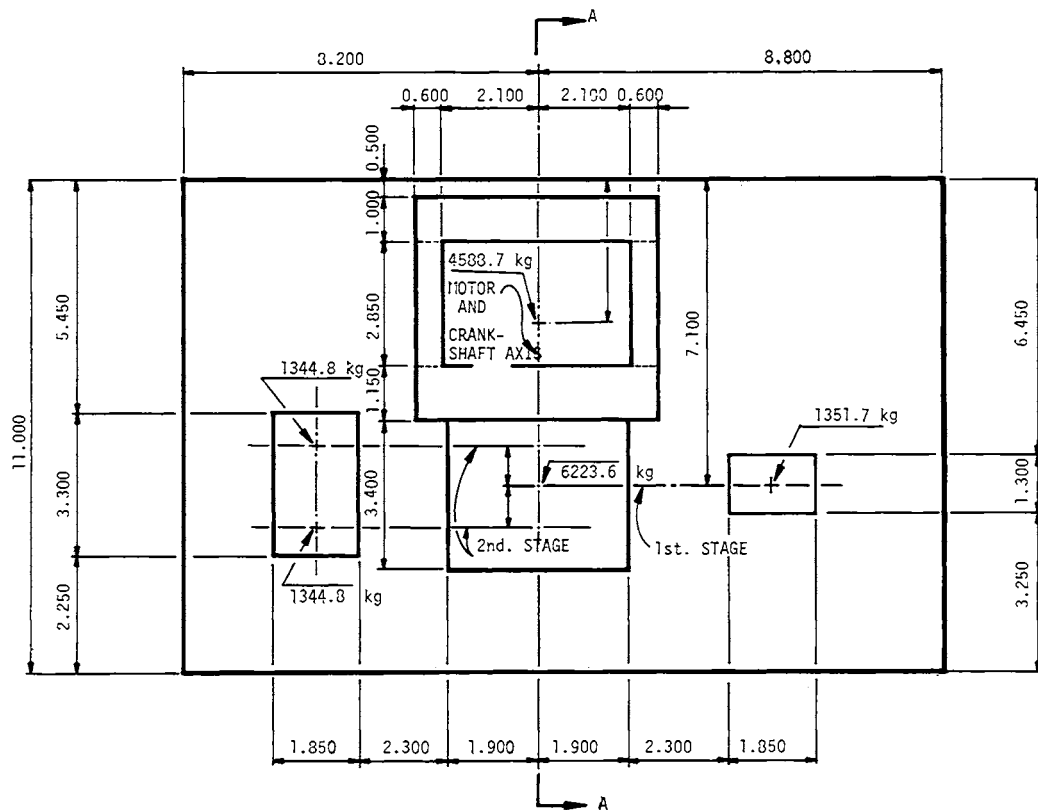


Fig. 1 (a)--Foundation layout

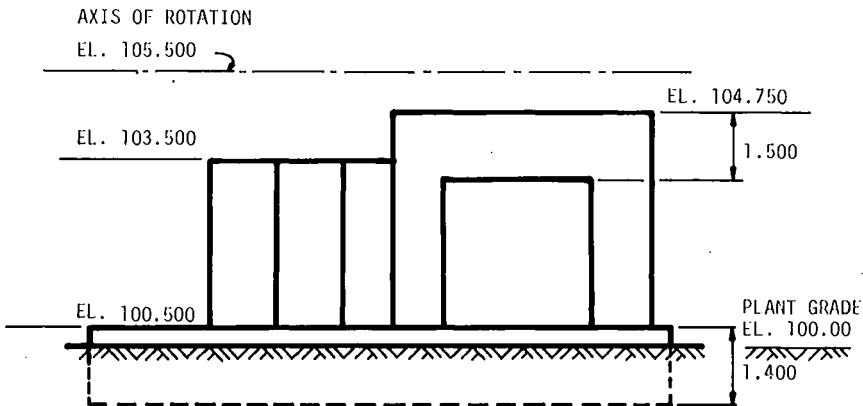


Fig. 1 (b)--Section A-A

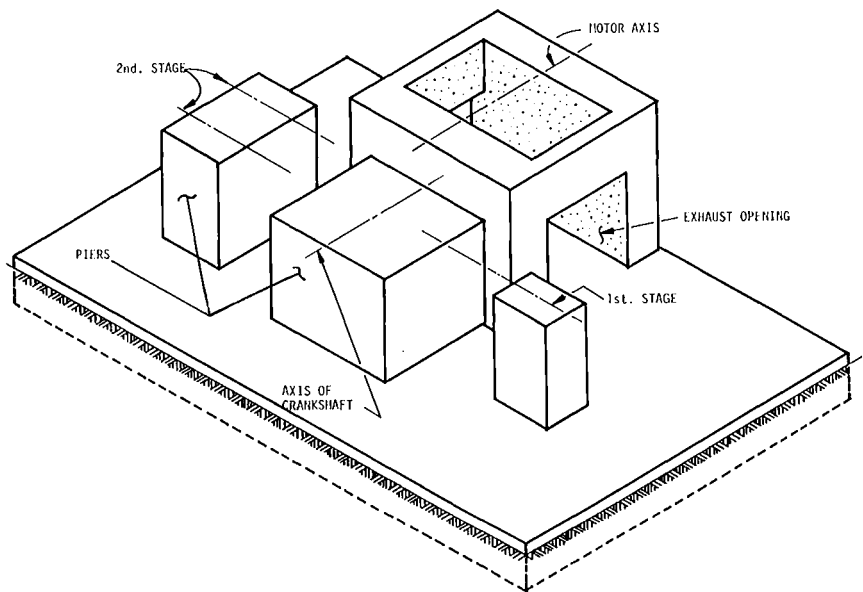
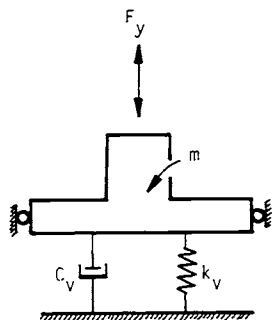
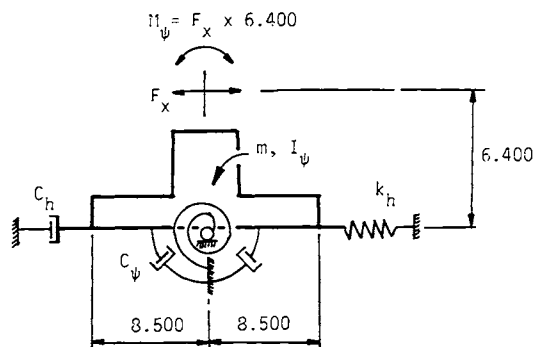


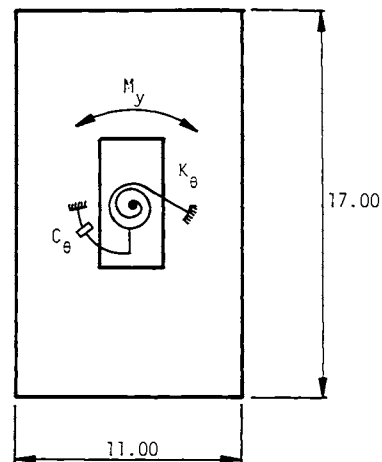
Fig. 1 (c)--Isometric view of elevated foundation



(a) VERTICAL EXCITATION



(b) HORIZONTAL AND ROCKING EXCITATION



(c) TORSIONAL EXCITATION

Fig. 2--Model A, three equivalent lumped parameter system

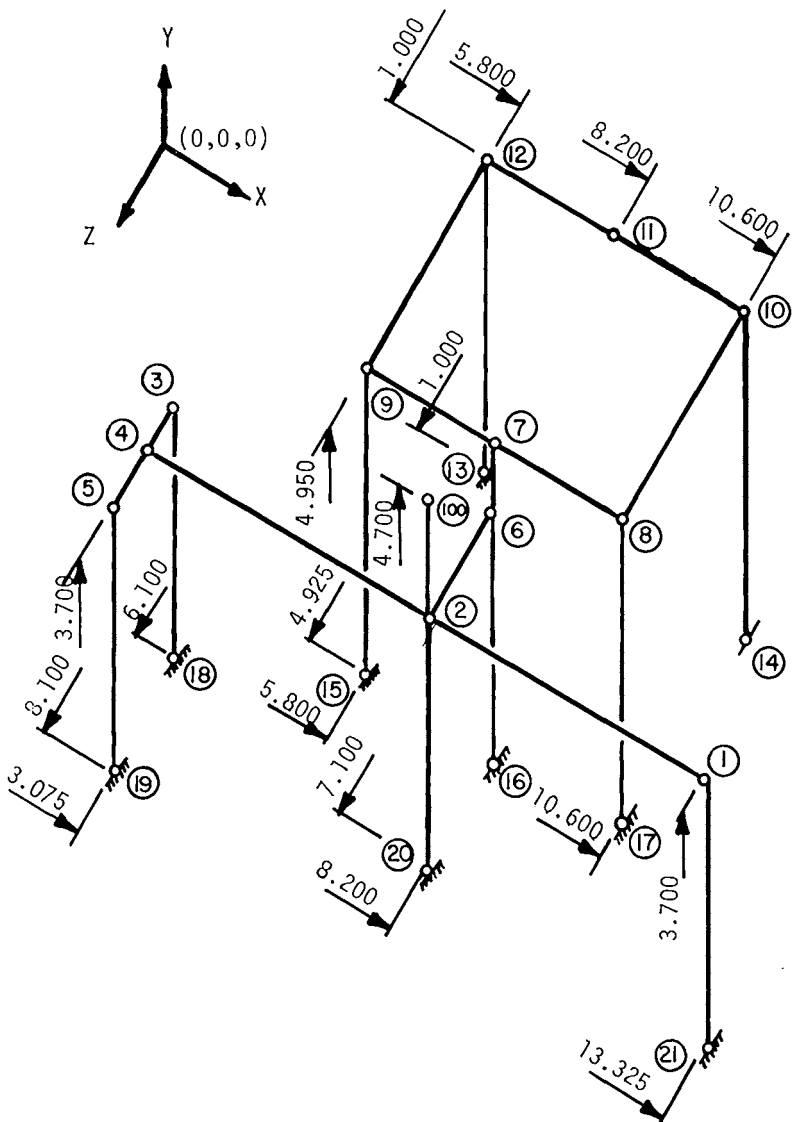


Fig. 3--Model B, multi-lumped mass of superstructure

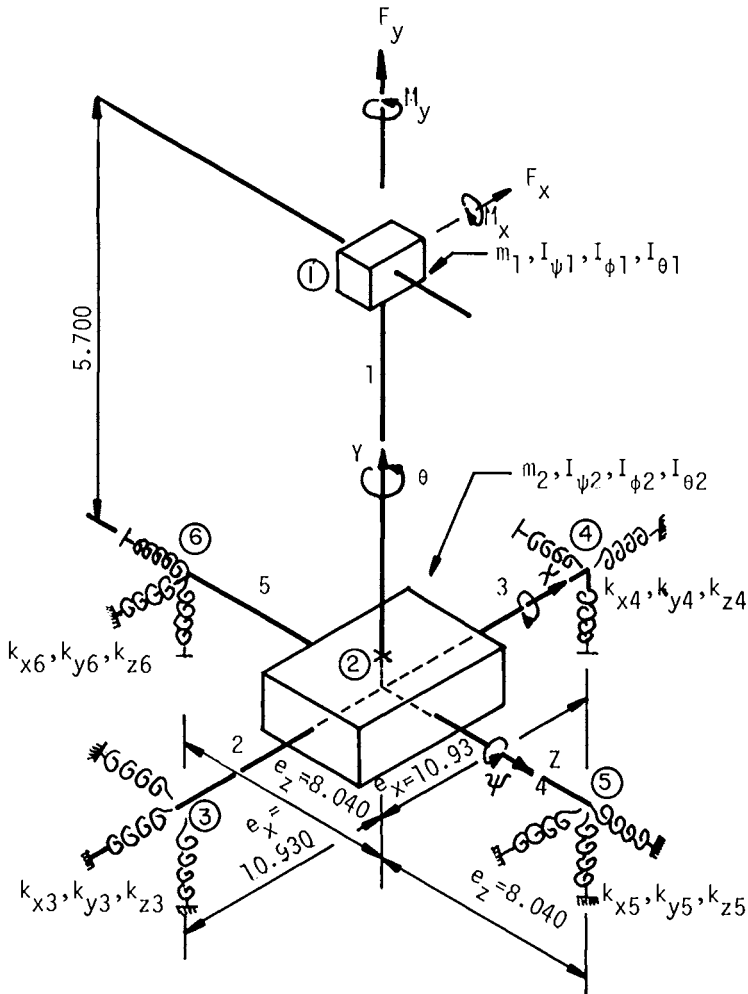


Fig. 4--Model C, two lumped mass with coupled soil-structure interaction (dimensions in mm)

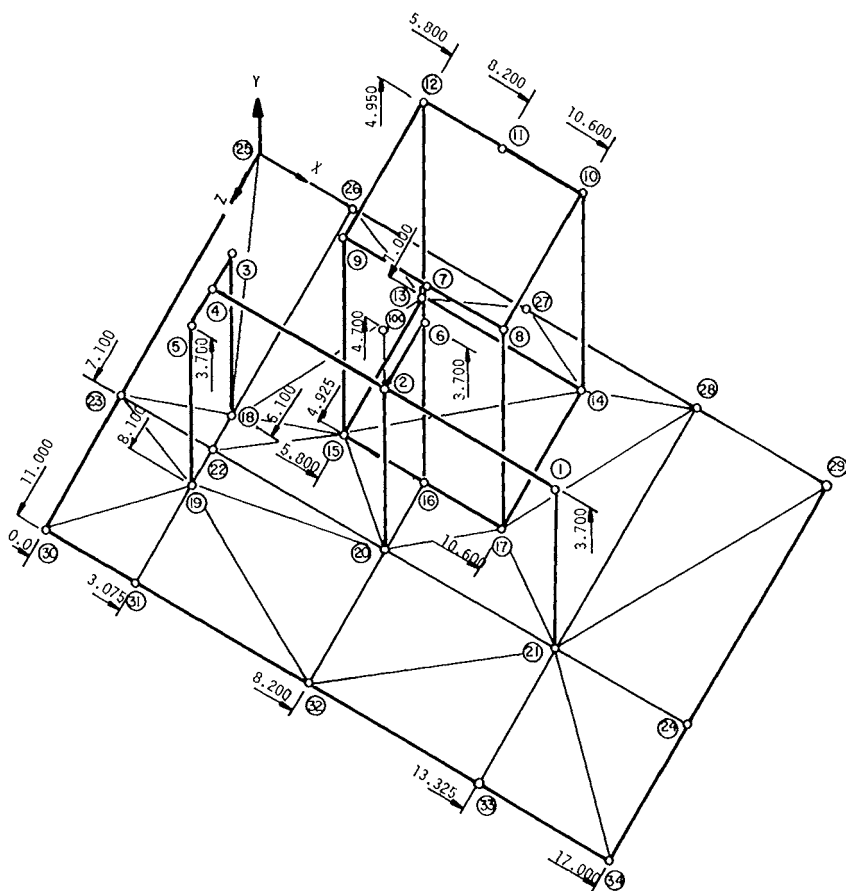


Fig. 5--Model D, multi-lumped mass with soil-structure interaction

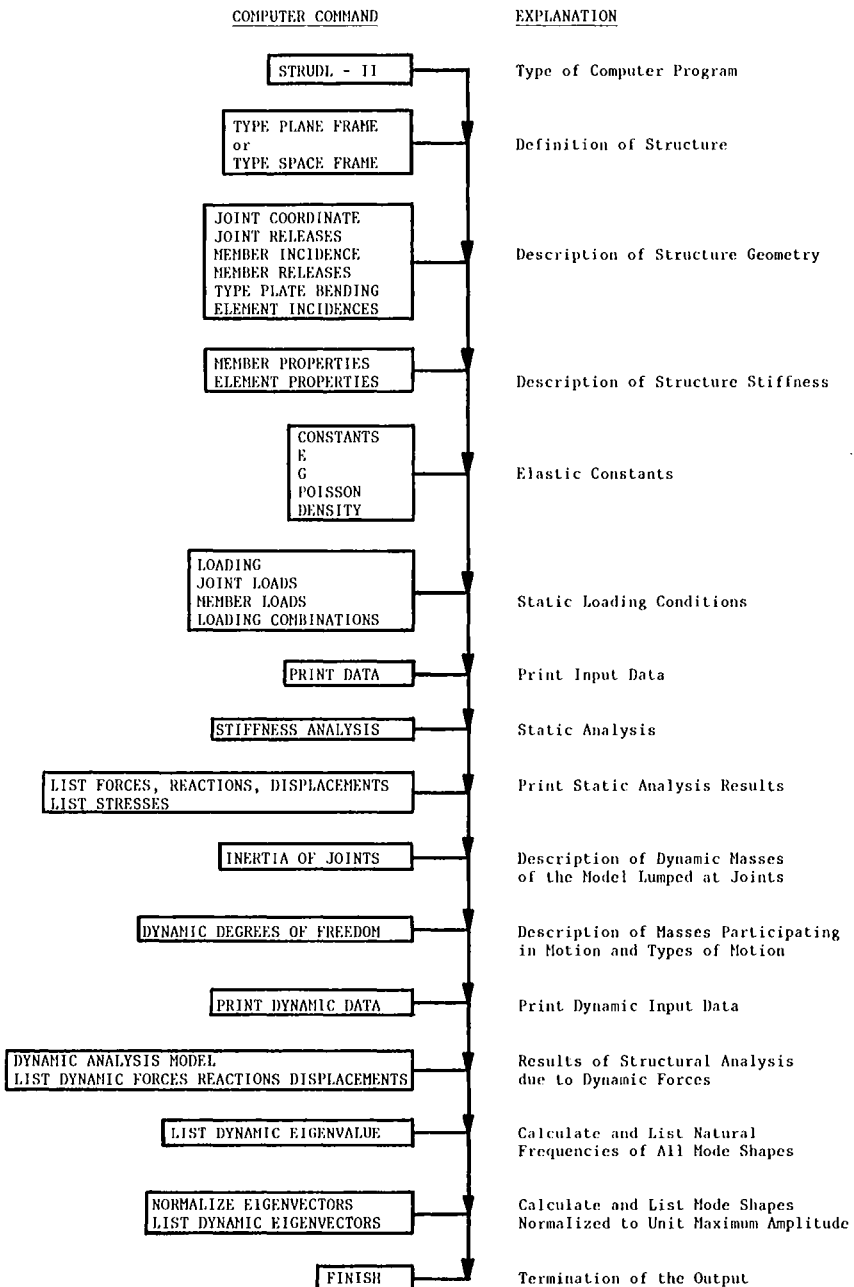


Fig. 6--Computer program flow chart

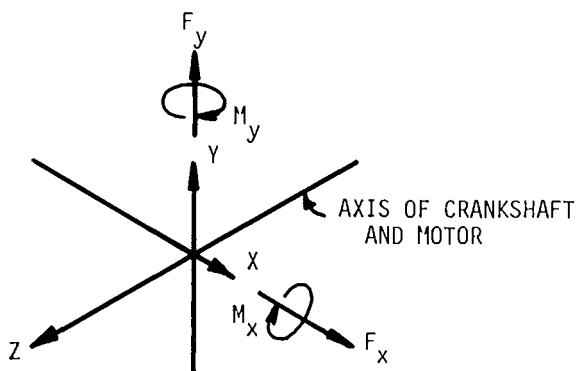


Fig. 7--Directions of unbalanced inertia force, reciprocating compressor

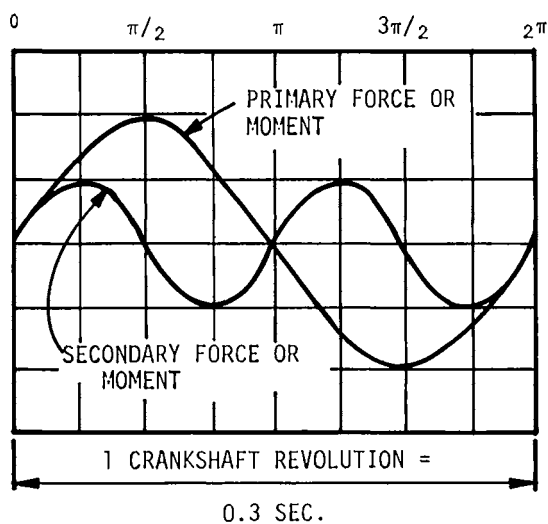


Fig. 8--Frequency of primary and secondary forces and moments

Controlling Automobile Shredder Vibration Through Pneumatic Isolation

By Donald E. Baxa and Robert Ebisch

Synopsis: This paper is a case study of pneumatic isolation systems in large automobile recycling machines. Machines of increasing size and power, plus a public that is increasingly insistent on its right to an environment free of noise and vibration, equal potential problems for recycling plants. In response to this situation, a hammermill/shredder operation making successful use of pneumatic isolation has been designed and constructed. The first, and still the largest, operation of its kind, the operation has demonstrated a valuable method of containing machine vibrations that might otherwise ignite hostility in the surrounding community and quite possibly bring on expensive legal action. The particular system described was installed in a 4,000-horsepower hammermill at an Indiana recycling firm. Besides effecting a more than 90 percent reduction in ground vibration, the system allows for easier and more efficient machine leveling and includes systems to warn of and localize pressure loss, to minimize routine mechanical shock, and to contain extraordinary shocks resulting from exploding gas tanks and loss of hammers within the machine. The system has now been in use for two years with no reported problems.

Keywords: concrete bases; hammer milling; machine bases; noise reduction; shock mechanics; shredders; vibration isolators.

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Robert Ebisch was employed with Engineering & Applied Science for several years as a part-time technical writer and editor. He moved to Washington, D.C. to work for the magazine, Chemistry (changed to SciQuest in April, 1979). He presently works for Building Design and Construction in Chicago.

Introduction

Vibration can be a costly and unpopular problem. Most people do not realize that rotating, reciprocating and impacting equipment can send enough vibration energy through the ground to annoy or damage nearby communities.

The shaking of an automobile shredder in one midwestern recycling plant recently brought on a \$500,000 lawsuit from nearby residents. The industry finally won its case by making minor changes, but not before an injunction forced corrective work and costly downtime. The vibrations, though reduced, continued to be a nuisance, and the recycler's community relations may never completely recover. This is only one example of the many ground vibration problems that exist throughout the country in mining, forging, foundry, auto shredding and other operations.

In past years, as equipment has grown in size and power, vibration problems have become increasingly more severe. At the same time, the public is becoming less tolerant of industrial vibration. State and local governments have written and are enforcing ordinances that limit the amount of ground vibration an industry can generate. Meeting head-on, these trends have caused a lot of trouble for industrial operations. The example discussed in the remainder of this paper shows one viable means of protecting the community from vibration when the exciting force cannot be designed out.

Case History and Description

In this instance, an Indiana-based scrap processing firm had already sunk pilings to support their 4,000 horsepower hammermill type shredder when they received news of the previously discussed \$.5 million lawsuit against one of their competitors for vibration nuisance. The spectre of similar problems in their own operation compelled them to seek advice.

Representatives of the Indiana company contacted the University of Wisconsin-Extension and the author regarding the possibility for design of a system to control the vibration they expected in their operation. Joint discussions of the problem led to the development of three alternative approaches:

1. Use a Slab-on-Grade Foundation similar to the one that had created the problems for their competitor.
2. Mount the slab on pilings to help limit the excursion of the unit.
3. Use properly designed vibration isolation techniques to limit the amount of vibration transmitted to the community.

Ordinarily these machines are mounted on either piling or slab-on-grade foundations. Slab-on-grade installations can and have had settling and community vibration problems. Piling foundations work well for most purposes, but they could still transmit vibration into the community and put greater stress on the machine as well.

Company representatives and UW-Extension engineering specialists decided that an inertia-block isolation system would work best. However, to get a 99 percent isolation efficiency for the 600 rpm (10 Hz) shredder would have required a steel spring roughly two feet in length with a static deflection of 10 inches. Since this system would be difficult to install and maintain and might prove laterally unstable, pneumatic isolation was the only alternative.

With pneumatic isolation there is no static deflection requirement. In order to have the isolator support the load, the design height (Reference 1) can be adjusted by varying the air pressure. By keeping the design position within the specified range, lateral stability is no problem. In general, all of the difficulties of steel springs are overcome.

As can be seen in Figure 1, a natural frequency of 1 Hz would be required for 99 percent isolation. This frequency was achieved by using a double convolution airmount set at 9.5 inches. Having a total volume of about 4600 cubic inches, the airmount operated at 160 PSI (Figures 2 and 3), with an external reservoir to produce the required volume. Connected to the airmount by an intervening orifice, the reservoir also provided additional damping to accommodate both the impact and out-of-balance forces produced by the shredder operation (Reference 1). For additional information on design features see References 2 and 3.

It is desirable to have a center of gravity at the top of the airmounts in order to decouple the system and provide stabil-

ity; therefore, an inertia block weighing about 800,000 pounds was designed. With the additional weight of the hammermill and the electric motor that drives it, the total weight that was isolated was about 1,100,000 pounds.

This weight was supported by sixteen airmounts. Even with this amount of mass it was felt that snubbers would still be required. Two of the design criteria were: 1) design against the loss of up to two hammers, each weighing 420 pounds, operating at a radius of four feet and a speed of 600 rpm, and 2) design against the possibility of an exploding gas tank inside the shredder. A vertical snubber can be seen in Figure 5, and horizontal snubbers are the two pipe arrangements in the middle of Figure 6.

A model of the system can be seen in Figures 7 and 8. The system consists of a piling sub-foundation on which rests a framework of two wide-flange beams. The airmounts, in turn, are secured to these, and they lift the single wide-flange framework that rests on top. From this upper beam, hanging rods run to the inertia concrete base. Although not shown in Figures 7 or 8, the motor mounts onto the portion that protrudes above the upper framework, and the shredder mounts on the lower flat section. See Figures 9 and 10.

Innovative features of the design include:

1) Easy and efficient leveling. The isolators are grouped and manifolded into banks of five, five, and six. Each of the three isolator banks are separately pressurized and separately controlled. Settling can be a problem in any large horsepower operation and here settling was expected in at least one or two pilings. A differential pressurization system makes it possible to accommodate such settling by simply adjusting valves. The only means of leveling for a system supported by pilings or slab-on-grade is to remove the machine and shim underneath it. The leveling controls are seen in Figure 11.

2) Each isolator has a check valve that allows air in but not out, so that pressure loss in one isolator does not affect the others, even though they are fed through the same manifold.

3) A pressure-loss warning system. A series of pressure switches trigger an alarm when pressure in any isolator, manifold or main goes below a present level.

4) As already described, additional damping to minimize shock was incorporated by orifice damping between the cylindrical air reservoir and each isolator.

5) Power and water come into the slab through hoses and cables that lie on a floating tray. Because of the vertical and horizontal mobility required for the slab, connections have to be able to move with respect to the ground (Figure 6).



The machine's mobility on its bed of air does not amount to much more than $\pm 1/4$ inch, but this provides significant stress relief to the system. This stress relief, although not well documented, has been shown to give equipment additional life and to reduce maintenance (Reference 4).

Even though the environment is potentially hazardous to the airmount, no difficulty has been reported in the five years that this system has been operating. To date it has experienced several exploding gas tanks and the loss of one and two hammers at a time, with no observed damage to the isolation system. The only maintenance required is periodic cleaning of the pit inside of which sits the inertia block. No settling of pilings has been observed.

Although the system was designed to have an isolation of 99 percent, something in excess of 90 percent was achieved. This was enough that vibration levels in the community were beneath the threshold of perception. Isolation beyond that threshold was not deemed necessary and was therefore sacrificed in order to gain two design advantages. One of these advantages was the orifice between the airmount and reservoir; it provides benefits that have already been discussed, but its inclusion in the system meant less isolation efficiency. Second, the feeding chute was excluded from isolation, which to some degree short-circuited the inertia-block vibrations into the ground but saved considerable cost. Plans were made to correct the short circuiting if it proved too great, but this was not the case.

This system's effectiveness in the community was demonstrated when opponents of the installation protested to the city council that their neighborhood would literally shake if the machine were allowed to go into operation. A company officer then announced to the gathering that the machine had already been in operation for two weeks. No one had noticed the difference.

Since installation of this particular hammermill shredder, the author has designed a second system for a device called a rotary motor mangler. It is a three-thousand-horsepower recycling unit with a bottom discharge, and one of the design requirements was that it be completely above grade. A model of this system can be seen in Figure 12. The house on top contains the drive motor. The motor mangler sits over the hole in the inertia base, which can be seen separately in Figure 13. This system is supported by 18 airmounts with a total isolated weight of about 1,250,000 pounds. An interesting feature of the system is that the reservoir and orifice are integral parts of the airmount. This system is also indirectly mounted on pilings, which are capped by the sub-foundation that supports the airmounts.

The principal advantage of this isolation system over that designed for the hammermill is a reduction in cost, this due to the fact that the base and the inertia block were entirely reinforced concrete rather than reinforced concrete and beamwork.

Construction on this system has not been completed, but operation is anticipated for sometime in 1980.

Conclusion

This paper has examined two case studies of pneumatic vibration isolation for large horsepower rotary impact installations. One system has been in operation for over two years without a single problem, the other is 75 percent complete. The pneumatic approach to vibration control has been proven successful. It is economical, reliable, does not interfere with operation, and offers better than 90 percent isolation.

By removing machinery from contact with the ground and by eliminating the channels through which vibration would otherwise travel, pneumatic isolation protects the processor against injunctions, lawsuits and work stoppages. If pneumatic isolation is installed only after the trouble begins, as an afterthought or remedy, it can cost three to four times more than if it is installed at the same time as the machinery.

A pneumatic isolation system should be expected to cost about five percent of the total installation cost for machines like those we have described in this article, and the percentage expense decreases in proportion to the weight of the machine. A 4,000-horsepower hammermill installation, running about \$2 million, will cost approximately \$50,000 more with pneumatic isolation than with a simple slab-on-piling foundation. Whether the added expenditure is worthwhile is a decision that will vary with the individual operator. But if the operation is within about 1,000 feet of a residential zone, pneumatic isolation should definitely be considered.

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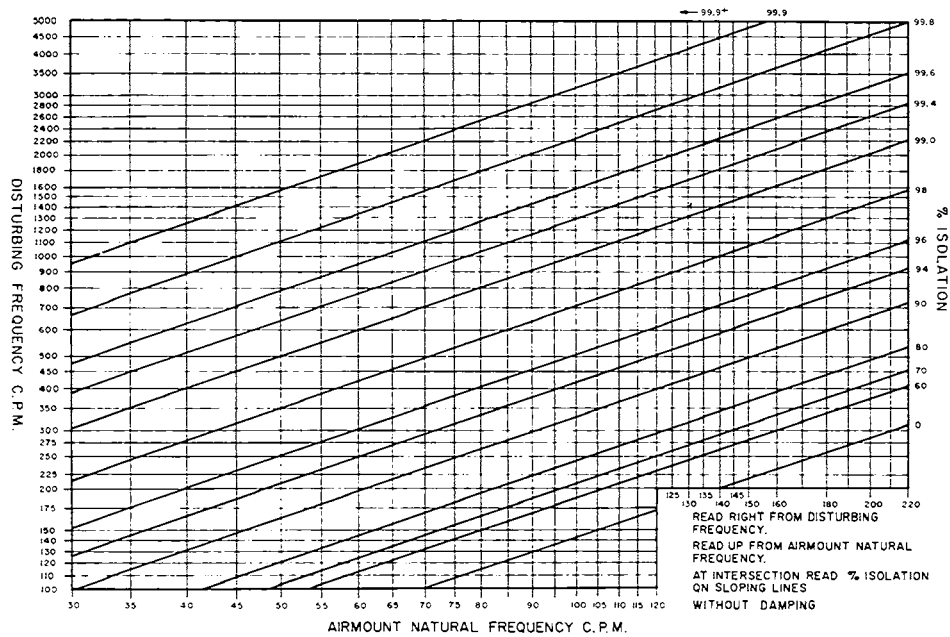
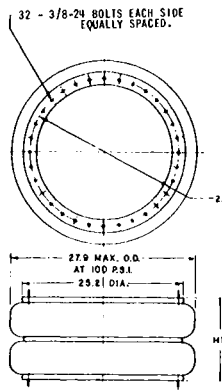


Fig. 1--Airmount isolation efficiency

EXAMPLE OF DYNAMIC CHARACTERISTICS	
DESIGN HEIGHT.....	9 IN.
VOLUME.....	.4378 IN. ³
PRESSURE.....	.60 P.S.I.
EFFECTIVE AREA.....	.4525 IN. ²
LOAD.....	.27150 LBS.
RATE.....	.6710 LBS./IN.
FREQUENCY.....	.93 C.P.M.



REFERENCE DATA	
ASSEMBLY.....	NAD-11650
ASSEMBLY WEIGHT.....	31.4 LBS.

211 SERIES STATIC DATA

RECOMMENDED DESIGN POSITION
STATIC PRESSURE 0 TO 100 PSI

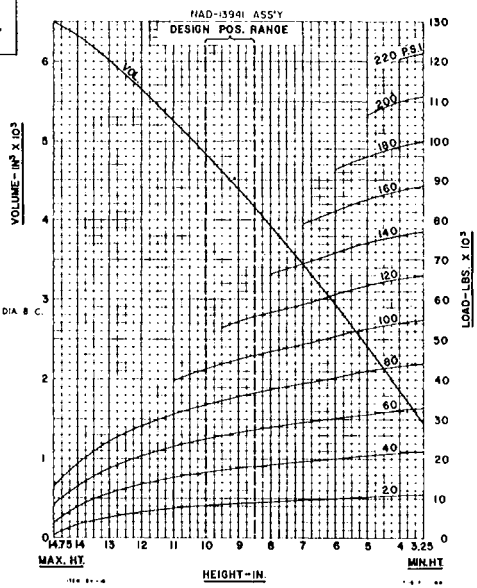


Fig. 2--Technical data for airmount Model 211. Specifications for Model 215 (four-ply) are identical to Model 211 (two-ply) except that they allow higher pressure utilization. (Courtesy of Firestone Industrial Products Company.)

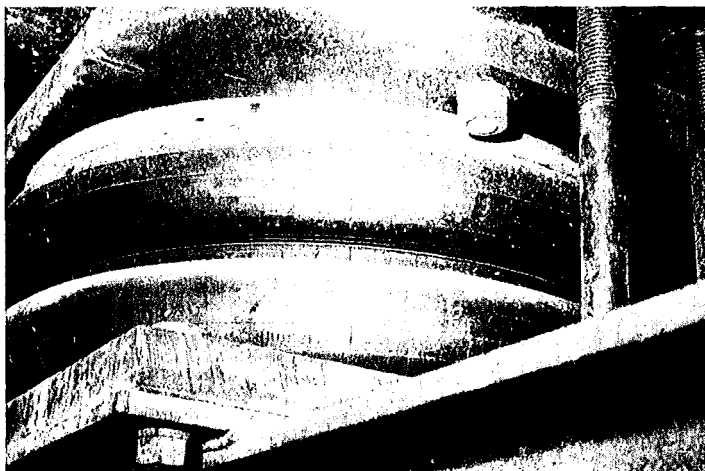


Fig. 3--Model 215 airmount in place

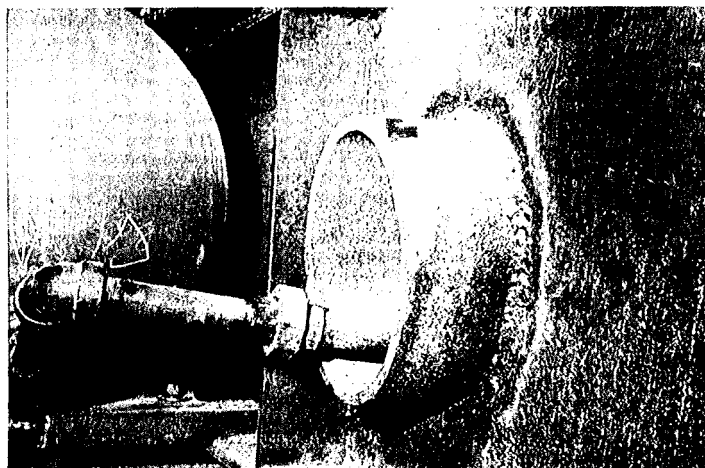


Fig. 4--Surge tank and connection to airmount

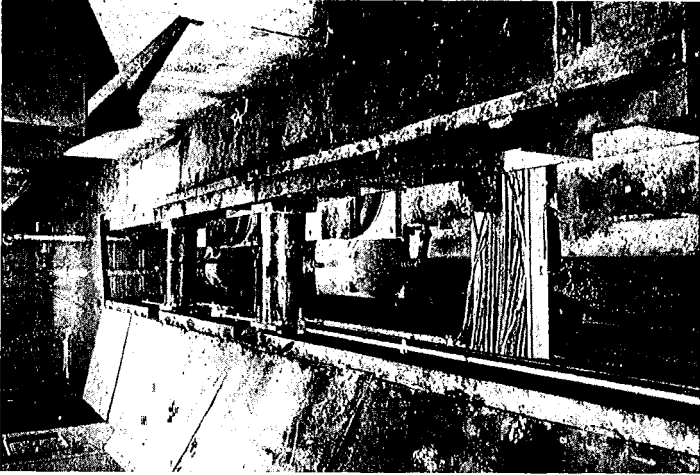


Fig. 6--Horizontal snubbing units

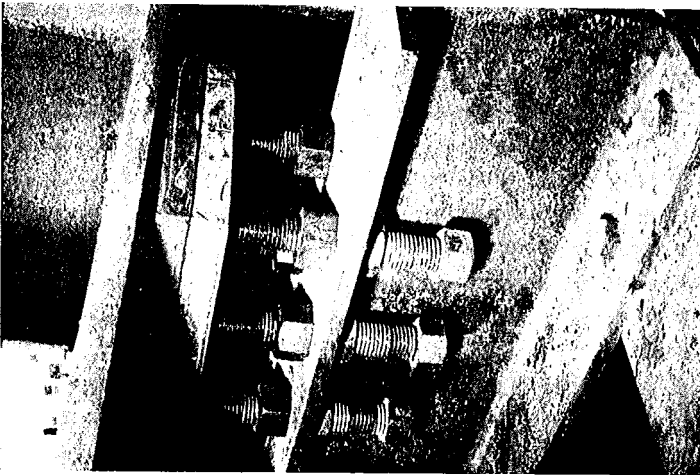


Fig. 5--Vertical snubbing unit

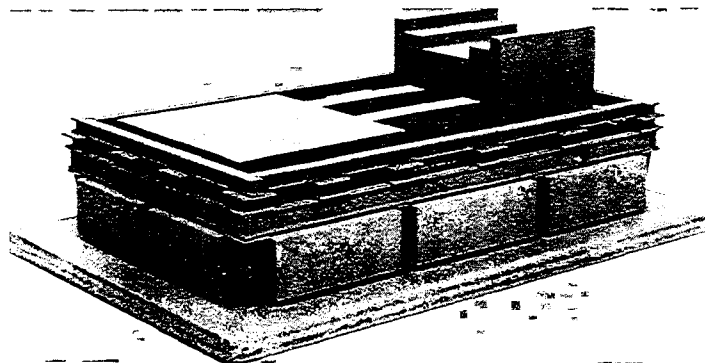


Fig. 7--Oblique view of auto shredder system model



Fig. 8--Side view of auto shredder system model

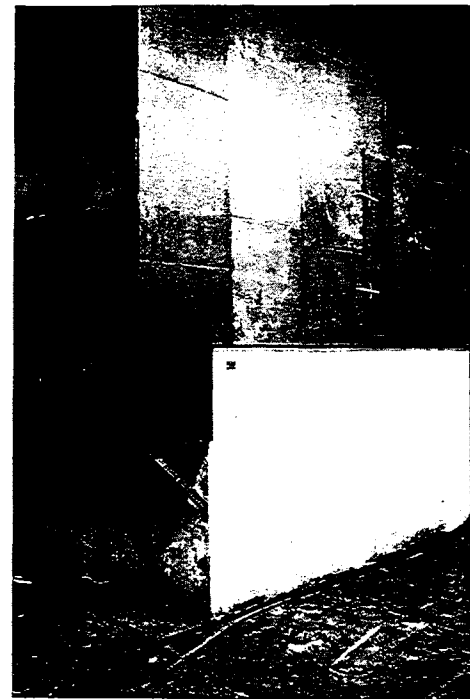


Fig. 9--Motor pedestal and motor enclosure

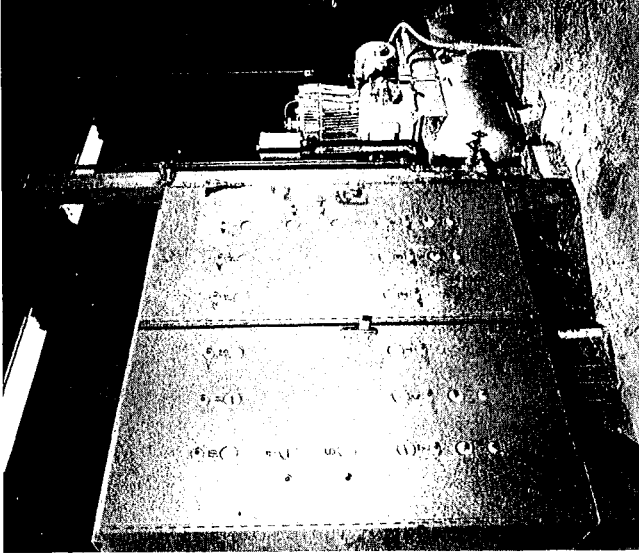


Fig. 11--Control panel for air supply and monitoring



Fig. 10--Auto shredding unit in place on foundation

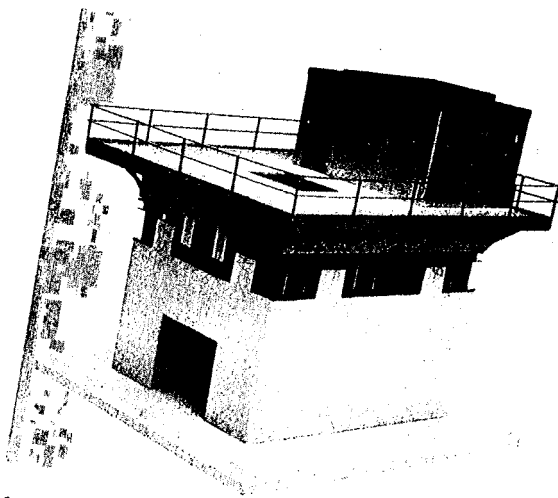


Fig. 12--Model of motor mangler isolation system

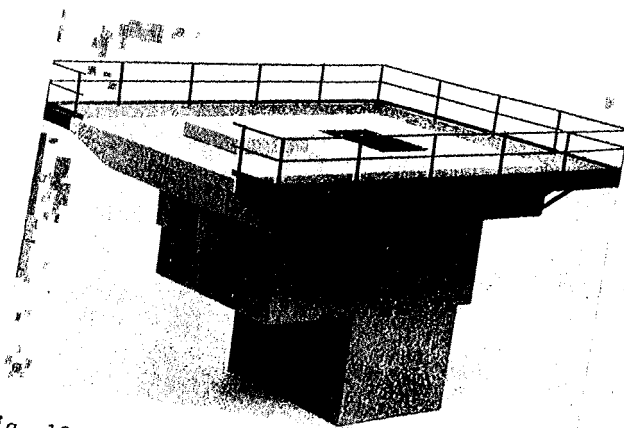


Fig. 13--Model of motor mangler inertia block

The Technology of Baseplate Grouting

**By Gary R. Hall, George W. Read, Jr.,
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Synopsis: The properties of portland cement based nonshrink grouts are discussed. A review of the basic engineering concepts involved is presented. A brief discussion on the chemistry involved in grout technology is presented. Performance criteria are discussed in some detail, along with methods of testing for the performance criteria. A brief description of the various grouting systems available to the market place is presented.

Keywords: expansion; fluidizing; grout; grouting; machine bases; portland cements; shrinkage; strength; stresses.

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Until very recently the subject of grouting of baseplates has been limited to a discussion on the "state-of-the-art." Now the "art" of grouting has approached the state wherein we can begin to speak meaningfully of the "technology of grout." The dictionary defines grout as being any material or mixture of materials which will harden or set, used to fill an existing space or void in a construction. This simplistic definition does not quite encompass those necessary and vital areas of concern in grouting technology.

Grouting technology includes, but is not necessarily limited to, a knowledge of:

1. The basic theory of a relatively inelastic system undergoing cyclical and acyclical load disturbances;
2. The products of hydration involved in the Portland cement/water system and the equilibria established under varying conditions;
3. Load imbalances generated by machine operation;
4. The effects of thermal and chemical environment upon the grout and embedded bolts, studs, etc.; and
5. Load stress transfer mechanisms.

Based upon these five areas, this paper will outline factors to be considered for baseplate design, grout consistency selection, design of grout configuration and interaction between baseplates, anchor bolts, shim packs, shearkeys and the grout monolith itself.

LOAD DISTURBANCE AND RESPONSE

An examination of the mechanisms involved in the deformation of the amorphous concrete indicates that concrete is a viscoelastic material. Being amorphous in nature, concrete is heterogeneous on the macroscopic level. However, concrete and grout are

considered to be isotropic from a consideration of their responses to applied loads. That is, in response to applied loads, a Portland cement based grout will have essentially the same properties in all directions.

Stress, or the intensity of the applied force at any point in the structure, is measured as the applied force of load acting upon a unit area in a plane. Depending upon the mode of application, stresses may be dynamic or static. Grouts experience both static and dynamic stresses. For example, compressive stresses are quite often static and essentially uniaxial. On the other hand, an angle-engine, reciprocating pump or compressor is prone to residual imbalance. In operation these compressors may deliver a wide array of dynamic loads to a grout pad, including impact forces, torsional-vibration loads, and alternating and reversed forces. These loads may apply uniaxial, biaxial or triaxial stresses to the grout. Grouts are often subjected to shear stresses as well, including torsional shear stress.

Essentially a grout slab or pad is a structural configuration with two dimensions being large when compared to a third. Therefore, it may behave as either a plate or a panel. If the grout is acted upon by forces in a direction tangent to the two large surfaces, it is considered a panel and the loads are taken as being in-plane. Out-of-plane loads acting perpendicular to the large surfaces render the body a plate.

Let us consider a grout pad as a plate or panel and let us assume that grout obeys Hooke's Law, which states that stress is proportional to strain and is independent of time, and that we are generally dealing with a very small strains. We can, therefore, use the generally accepted concepts of structural mechanics to predict the responses to load disturbances.

Perhaps the four most important quantities of response are Young's modulus, compressive modulus, modulus of rigidity and Poisson's ratio.

I shall merely show the definition formula for these properties:

$$E = \text{Young's Modulus} = \text{tensile mod. of elast.} = \frac{S_t}{L_t}$$

$$G = \text{Shear or Rigidity Modulus} = \frac{\tau}{\gamma}$$

$$H = \text{Poisson's Ratio} = \frac{\text{Lateral Strain}}{\text{Longitudinal Strain}}$$

$$G \text{ may be calculated from } E \text{ by: } G = \frac{E}{2 + 2\mu}$$

For incompressible soft materials this reduces to: $E = 3G$

We may also approximate the compressive modulus by:

$$E_c = 33 (S_c)^{3/2}$$

Where S_t = tensile stress

L_t = tensile strain

S_c = compressive stress

τ = shear stress

γ = shear strain

Compressive strength is generally in the range of 3 to 5 times the shear strength for concrete and grout.

CHEMISTRY OF PORTLAND CEMENT

A knowledge of elementary Portland Cement hydration products and the equilibria established under varying conditions is essential to grout technology. The engineer must be cognizant of the various phase assemblages possible and know which compounds to expect in his application, as the properties of the grout are directly influenced by the phase assemblage. For example, the primary cement forming compound in Type III Portland cement is C_3S which reacts with water to form $C_3S \cdot H_2O$ gel (tobermorite).

The primary cementing action is due to the tobermorite gel. The cementing action is, for the most part, a function of the tremendous surface area of the gel. A typical surface area might be 200 to 400 m^2/gm . Tobermorite has a variable composition as verified by both an x-ray diffraction and the scanning electron microscope, which show that the tobermorite is in a colloidal gel form. The scanning electron microscope reveals that tobermorite consists of sheets and fibers. As these morphological forms contract and intertwine a dense, cohesive structure results. It is the properties of this gel phase that controls much of the engineering properties of hardened concrete. The relative ease with which capillary water is lost upon drying allows considerable shrinkage to occur. The shrinkage is somewhat reversible upon rewetting but not completely. Such shrinkage can cause uneven bearing surfaces, generate tensile stresses and result ultimately in cracking of the grout, if not controlled.

Fired and vitrified ceramic materials owe their strength to the formation of glass phases within their matrix, whereas, cements and concretes derive their strength from the formation of hydrated phases or gels. These are usually colloidal in character.

With hydraulic cements an agglomeration process occurs with a resulting bonding reaction. A chemical reaction is involved and chemical bonds are formed. In order to function in a bonding