

ACI 351.3R-18

# Report on Foundations for Dynamic Equipment

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## Report on Foundations for Dynamic Equipment

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# Report on Foundations for Dynamic Equipment

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*This report presents to industry practitioners the various design criteria and methods and procedures of analysis, design, and construction applied to foundations for dynamic equipment.*

**Keywords:** amplitude; foundation; reinforcement; vibration.

## CONTENTS

### CHAPTER 1—INTRODUCTION, p. 2

- 1.1—Background, p. 2
- 1.2—Purpose, p. 2
- 1.3—Scope, p. 2

### CHAPTER 2—NOTATION AND DEFINITIONS, p. 3

- 2.1—Notation, p. 3
- 2.2—Definitions, p. 5

### CHAPTER 3—FOUNDATION AND MACHINE TYPES, p. 5

- 3.1—General considerations, p. 5

### 3.2—Machine types, p. 5

### 3.3—Foundation types, p. 6

### CHAPTER 4—DESIGN LOADS, p. 8

- 4.1—Overview of design loads and criteria, p. 8
- 4.2—Static machine loads, p. 9
- 4.3—Dynamic machine loads, p. 11
- 4.4—Environmental loads, p. 17
- 4.5—Load conditions, p. 17
- 4.6—Load combinations, p. 18

### CHAPTER 5—IMPEDANCE OF THE SUPPORTING MEDIUM, p. 18

- 5.1—Overview and use of soil impedance, p. 18
- 5.2—Basic dynamic concepts, p. 19
- 5.3—Calculation of dynamic foundation impedances, p. 21
- 5.4—Dynamic impedance of soil-supported foundations, p. 22
- 5.5—Dynamic impedance of pile foundations, p. 28
- 5.6—Transformed impedance relative to center of gravity, p. 35
- 5.7—Added mass concept, p. 35
- 5.8—Sample impedance calculations, p. 36

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## CHAPTER 6—VIBRATION ANALYSIS AND ACCEPTANCE CRITERIA, p. 40

- 6.1—Overview, p. 40
- 6.2—Modeling for rigid foundations, p. 41
- 6.3—Modeling for flexible foundations, p. 42
- 6.4—Solution methods, p. 46
- 6.5—Frequency analysis, p. 49
- 6.6—Forced response analysis, p. 50
- 6.7—Sample calculations, p. 52

## CHAPTER 7—DESIGN AND MATERIALS, p. 56

- 7.1—Overview of design methods, p. 56
- 7.2—Concrete, p. 59
- 7.3—Reinforcement, p. 61
- 7.4—Machine anchorage, p. 62
- 7.5—Elastic support systems, p. 63
- 7.6—Grout, p. 63
- 7.7—Seismic design considerations, p. 64
- 7.8—Fatigue considerations, p. 64
- 7.9—Special considerations for compressor block post-tensioning, p. 64
- 7.10—Sample calculations, p. 64

## CHAPTER 8—CONSTRUCTION CONSIDERATIONS, p. 67

- 8.1—Subsurface preparation and improvement, p. 67

## CHAPTER 9—REPAIR AND UPGRADE, p. 67

- 9.1—Overview of need for repair, p. 67
- 9.2—Discussion of repair options, p. 68

## CHAPTER 10—REFERENCES, p. 69

- Authored documents, p. 71

## APPENDIX A—DYNAMIC SOIL PROPERTIES, p. 73

- A.1—Poisson's ratio, p. 73
- A.2—Dynamic shear modulus, p. 74
- A.3—Soil damping, p. 75
- A.4—Radiation damping, p. 76

## CHAPTER 1—INTRODUCTION

### 1.1—Background

Machinery with rotating, reciprocating, or impacting masses requires a foundation that can resist dynamic forces. Precise machine alignment should be maintained, and foundation vibrations should be controlled to ensure proper functioning of the machinery during its design service life.

Successful design of such foundations for dynamic equipment involves close collaboration and cooperation among machine manufacturers, geotechnical engineers, engineers, owners, and construction personnel. Because different manufacturers may have very different foundation acceptance criteria and their own practices with regards to foundation design requirements, strict adherence to **ACI 318** alone may not be necessarily appropriate for certain foundations that support heavy industrial equipment, such as steam turbine generators, combustion turbine generators, or

compressors. In addition, different practicing engineering firms may use design approaches based on past successful performance of foundations, even though these may not be the most economical designs. Therefore, this report summarizes current design practices to present a common approach, in principle, for various types of concrete foundations supporting dynamic equipment.

Compared to the previous edition, this document has been reorganized to make the document more systematic and user-friendly. More detailed information on the following subjects has been added on the behavior of foundations subjected to dynamic machine forces:

- a) Impedance of the supporting medium (both soil-supported and pile-supported foundations)
- b) General overview of vibration analysis (including finite-element modeling) and acceptance criteria, including finite-element analysis
- c) Determination of various soil properties required for dynamic analysis of machine foundations

Example problems have been reworked and improved with some additional details to better illustrate the implementation of the calculation procedure in a manual calculation. Latest relevant references have been added to capture the current practice.

### 1.2—Purpose

The purpose of this report is to present general guidelines and current engineering practices in the analysis and design of reinforced concrete foundations supporting dynamic equipment.

This report presents and summarizes, with reference materials, various design criteria, methods and procedures of analysis, and construction practices currently applied to dynamic equipment foundations by industry practitioners.

### 1.3—Scope

This document is limited in scope to the engineering, construction, repair, and upgrade of concrete foundations for dynamic equipment. For the purposes of this document, dynamic equipment includes the following:

- a) Rotating machinery
- b) Reciprocating machinery
- c) Impact or impulsive machinery

**ACI 351.1R** provides an overview of current design practice on grouting. Design practices for foundations supporting static equipment are discussed in **ACI 351.2R**.

There are many technical areas that are common to both dynamic equipment and static equipment foundations. Various aspects of the analysis design and construction of foundations for static equipment are addressed in **ACI 351.2R**. To simplify the presentation, this report is limited in scope to primarily address the design and material requirements that are pertinent only to dynamic equipment foundations. Engineers are advised to refer to **ACI 351.2R** for more information on the foundation design criteria (static loadings, load combinations, design strength, stiffness, and stability) and design methods for static loads. In particular, **ACI 351.2R** provides detailed coverage on the design and anchorage of equipment to concrete foundations. Note that

ACI 351.2R was published prior to a major revision to ACI 318 and some of the section numbers that it references in ACI 318 may have changed.

## CHAPTER 2—NOTATION AND DEFINITIONS

### 2.1—Notation

|                      |                                                                                                |                  |                                                                                                                                          |
|----------------------|------------------------------------------------------------------------------------------------|------------------|------------------------------------------------------------------------------------------------------------------------------------------|
| $A$                  | = steady-state vibration amplitude, in. (mm)                                                   | $F_K$            | = force in vibration isolator spring, lbf (N)                                                                                            |
| $A_{head}$           |                                                                                                | $F_o$            | = dynamic force amplitude (zero-to-peak), lbf (N)                                                                                        |
| $A_{crank}$          | = head and crank areas, in. <sup>2</sup> (mm <sup>2</sup> )                                    | $F_{pl}$         | = lateral/longitudinal pseudo-dynamic design force, lbf (N)                                                                              |
| $A_p$                | = cross-sectional area of the pile, in. <sup>2</sup> (mm <sup>2</sup> )                        | $F_{pv}$         | = vertical pseudo-dynamic design force, lbf (N)                                                                                          |
| $a, b$               | = plan dimension of a rectangular foundation, ft (m)                                           | $F_r$            | = maximum horizontal dynamic force, lbf (N)                                                                                              |
| $a_o$                | = dimensionless frequency                                                                      | $F_{red}$        | = force reduction factor to account for the fraction of individual cylinder load carried by the compressor frame (frame rigidity factor) |
| $B_c$                | = cylinder bore diameter, in. (mm)                                                             | $F_{rod}$        | = force acting on piston rod, lbf (N)                                                                                                    |
| $B_i$                | = mass ratio for the $i$ -th direction                                                         | $F_s$            | = dynamic inertia force of slide, lbf (N)                                                                                                |
| $B_{mf}$             | = machine footprint width, ft (m)                                                              | $F_{THROW}$      | = horizontal force to be resisted by each throw's anchor bolts, lbf (N)                                                                  |
| $B_M$                | = width of mat foundation, ft (m)                                                              | $F(t)$           | = generic representation of time-varying load (force or moment) horizontal                                                               |
| $B_r$                | = ram weight, tons (kN)                                                                        | $F_{unbalance}$  | = maximum value applied using parameters for a horizontal compressor cylinder, lbf (N)                                                   |
| $b_1, b_2$           | = constants 0.425 and 0.687, respectively                                                      | $f'_c$           | = specified concrete compressive strength, psi (MPa)                                                                                     |
| $C$                  | = damping coefficient or total damping at center of resistance                                 | $f_{i1}, f_{i2}$ | = dimensionless pile stiffness and damping functions for the $i$ -th direction                                                           |
| $[C]$                | = damping matrix                                                                               | $f_o$            | = operating speed, rpm                                                                                                                   |
| $C_{CR}$             | = critical damping coefficient                                                                 | $G, G^*$         | = dynamic shear modulus of the soil, psi (MPa)                                                                                           |
| $C_{i1}, C_{i2}$     | = dimensionless stiffness and damping parameters, subscription $i = u, v, \psi, \eta$          | $G_p J$          | = torsional stiffness of the pile, lbf-ft <sup>2</sup> (N-m <sup>2</sup> )                                                               |
| $c$                  | = viscous damping constant, lbf-s/ft (N-s/m)                                                   | $G_s$            | = dynamic shear modulus of the embedment (side) material, psi (MPa)                                                                      |
| $c_i$                | = damping constant for the $i$ -th direction                                                   | $H$              | = depth of soil layer, ft (m)                                                                                                            |
| $c_i(\text{adj})$    | = adjusted damping constant for the $i$ -th direction                                          | $I_g$            | = gross area moment of inertia, in. <sup>2</sup> (mm <sup>2</sup> )                                                                      |
| $c_{ij}$             | = equivalent viscous damping of pile $j$ in the $i$ -th direction                              | $I_p$            | = moment of inertia of the pile cross section in. <sup>4</sup> (mm <sup>4</sup> )                                                        |
| CG                   | = center of gravity                                                                            | $i$              | = $\sqrt{-1}$                                                                                                                            |
| CF                   | = center of force                                                                              | $i$              | = directional indicator or modal indicator, as a subscript                                                                               |
| $c_{gi}$             | = pile group damping in the $i$ -th direction                                                  | $K$              | = stiffness or total stiffness at center of resistance, lbf/ft (N/m) or lbf-ft/rad (N-m/rad)                                             |
| $D$                  | = damping ratio                                                                                | $[K]$            | = stiffness matrix                                                                                                                       |
| $D_i$                | = damping ratio for the $i$ -th direction                                                      | $K'$             | = total stiffness at center of gravity, lbf/ft (N/m) or lbf-ft/rad (N-m/rad)                                                             |
| $D_{rod}$            | = rod diameter, in. (mm)                                                                       | $K_{ij}^*$       | = impedance in the $i$ -th direction due to a displacement in the $j$ -th direction                                                      |
| $d$                  | = pile diameter, in. (mm)                                                                      | $K_N$            | = actual negative stiffness, lbf/ft (N/m) or lbf-ft/rad (N-m/rad)                                                                        |
| $d_s$                | = displacement of the slide, in. (mm)                                                          | $K_P$            | = arbitrary chosen positive stiffness value (typically set equal to the static stiffness), lbf/ft (N/m) or lbf-ft/rad (N-m/rad)          |
| $d_{mf}$             | = distance from machine shaft centerline to top of foundation, ft (m)                          | $K_{eff}$        | = effective bearing stiffness, lbf/in. (N/mm)                                                                                            |
| $E$                  | = static Young's modulus of concrete, psi (MPa)                                                | $K_s$            | = static soil stiffness, lbf/in <sup>3</sup> (N/m <sup>3</sup> )                                                                         |
| $E_d$                | = dynamic Young's modulus of concrete, psi (MPa)                                               | $K_c^G$          | = pile group coupling impedance                                                                                                          |
| $E_p$                | = Young's modulus of the pile, psi (MPa)                                                       | $K_h^G$          | = pile group horizontal impedance                                                                                                        |
| $e_m$                | = mass eccentricity, in. (mm)                                                                  | $K_v^G$          | = pile group vertical impedance                                                                                                          |
| $F$                  | = peak value of harmonic dynamic load (force or moment)                                        | $K_\psi^G$       | = pile group rocking impedance                                                                                                           |
| $F_1$                | = correction factor                                                                            | $k$              | = individual pile stiffness at center of resistance, lbf/ft (N/m) or lbf-ft/rad (N-m/rad)                                                |
| $F_{block}$          | = force acting outward on the block from which concrete stresses should be calculated, lbf (N) | $k_{ei}^*$       | = impedance in the $i$ -th direction due to embedment                                                                                    |
| $(F_{bolt})_{CHG}$   | = force to be restrained by friction at the crosshead guide tie-down bolts, lbf (N)            | $k_{gi}$         | = pile group stiffness in the $i$ -th direction, lbf/ft (N/m) or lbf-ft/rad (N-m/rad)                                                    |
| $(F_{bolt})_{frame}$ | = force to be restrained by friction at the frame tie-down bolts, lbf (N)                      | $k_i$            | = static stiffness for the $i$ -th direction, lbf/ft (N/m) or lbf-ft/rad (N-m/rad)                                                       |
| $F_D$                | = damper force, lbf (N)                                                                        |                  |                                                                                                                                          |
| $F_{GMAX}$           | = maximum horizontal gas force on a throw or cylinder, lbf (N)                                 |                  |                                                                                                                                          |
| $F_{IMAX}$           | = maximum horizontal inertia force on a throw or cylinder, lbf (N)                             |                  |                                                                                                                                          |

$k_i^*$  = frequency-dependent impedance in the  $i$ -th direction  
 $k_i(\text{adj})$  = adjusted static stiffness for the  $i$ -th direction, lbf/ft (N/m) or lbf-ft/rad (N-m/rad)  
 $k_i^*(\text{adj})$  = adjusted frequency-dependent impedance in the  $i$ -th direction  
 $k_{ij}$  = stiffness of pile  $j$  in the  $i$ -th direction, lbf/ft (N/m) or lbf-ft/rad (N-m/rad)  
 $k_{ij}^{st}$  = static stiffness of an individual pile  $j$  in the  $i$ -th direction, lbf/ft (N/m) or lbf-ft/rad (N-m/rad)  
 $k_s$  = soil modulus of subgrade reaction, lbf/in<sup>3</sup> (N/m<sup>3</sup>)  
 $k_{st}$  = static stiffness constant  
 $k_u^*$  = horizontal impedance of supporting medium  
 $k_v^*$  = vertical impedance of supporting medium  
 $k_\psi^*$  = rocking impedance of supporting medium  
 $k_\eta^*$  = torsional impedance of supporting medium  
 $k(\omega)$  = frequency ( $\omega$ )-dependent dynamic impedance  
 $L$  = length of connecting rod, in. (mm)  
 $L_M$  = greater plan dimension of the mat foundation, ft (m)  
 $L_{mf}$  = machine footprint length, ft (m)  
 $L_P$  = lateral distance from center of resistance to individual piles, ft (m)  
 $l$  = depth of embedment, ft (m)  
 $l_p$  = pile length, ft (m)  
 $M$  = mass, lbm (kg)  
 $[M]$  = mass matrix  
 $M_h$  = hammer mass, including any auxiliary foundation, lbm (kg)  
 $M_o$  = overturning moment on foundation, lbf-ft (N-m)  
 $MR$  = mass ratio of concrete foundation to machine  
 $M_r$  = ram mass, including dies and ancillary parts, lbm (kg)  
 $M_{res}$  = foundation overturning resistance, lbf-ft (N-m)  
 $M_\Delta$  = added mass, lbm (kg)  
 $m$  = mass of the machine-foundation system; lbm (kg)  
 $m_d$  = slide mass including the effects of any balance mechanism, lbm (kg)  
 $m_r$  = rotating mass, lbm (kg)  
 $m_{rec}$  = reciprocating mass in a reciprocating machine, lbm (kg)  
 $m_{rot}$  = rotating mass in a reciprocating machine, lbm (kg)  
 $m_s$  = added mass (inertial), lbm (kg)  
 $N$  = number of piles  
 $(N_{bolt})_{CHG}$  = number of bolts holding down one cross-head guide  
 $(N_{bolt})_{frame}$  = number of bolts holding down the frame, per cylinder  
 $NT$  = normal torque, lbf-ft (N-m)  
 $P_{ALL}$  = allowable bearing pressure, ksf (kPa)  
 $P_{heads}$   
 $P_{crank}$  = instantaneous head and crank pressures, psi (MPa)  
 $P_{max}$  = maximum bearing pressure, ksf (kPa)  
 $P_s$  = power being transmitted by the shaft at the connection, horsepower (kilowatts)  
 $R$  = circular foundation radius, equivalent translation radius of rectangular foundation, ft (m)  
 $R_i$  = equivalent radius of rectangular foundation, ft (m)

$R_{\psi a}, R_{\psi b}$  = equivalent rocking radius of foundation about a- and b-axis, respectively, ft (m)  
 $R_\eta$  = equivalent torsional radius of foundation, ft (m)  
 $r$  = length of crank, in. (mm)  
 $r_i$  = radius of the crank mechanism of the  $i$ -th cylinder, in. (mm)  
 $r_o$  = pile radius or equivalent radius, in. (mm)  
 $S$  = press stroke, in. (mm)  
 $S_{all}$  = allowable foundation settlement, in. (mm)  
 $S_f$  = service factor, used to account for increasing unbalance during the design service life of the machine  
 $S_{i1}, S_{i2}$  = dimensionless stiffness and damping parameters for side layer, subscription  $i = u, v, \psi, \eta$   
 $S_{max}$  = maximum foundation settlement, in. (mm)  
 $SV_R$  = seismic shear force due to the rigid foundation and other rigid components, lbf (N)  
 $SV_s$  = seismic shear force due to the superstructure, machine and other flexible components, lbf (N)  
 $SV_{seismic}$  = total seismic shear force machine-foundation system, lbf (N)  
 $s$  = pile center-to-center spacing, ft (m)  
 $[T]$  = transfer matrix  
 $T_M$  = mat foundation thickness, ft (m)  
 $T_{min}$  = minimum required anchor bolt tension, lbf (N)  
 $t$  = time, s  
 $u$  = displacement amplitude, in. (mm)  
 $u_0$  = peak displacement amplitude, in. (mm)  
 $V_c$  = compressive velocity of a pile, ft/s (m/s)  
 $V_F$  = transmissibility factor  
 $V_{La}$  = Lysmer's analog wave velocity, ft/second (m/s)  
 $V_{max}$  = maximum allowable bearing vibration, in. (mm)  
 $V_{peak}$  = peak velocity, in./s (mm/s)  
 $V_{RMS}$  = root mean square velocity, in./s (mm/s)  
 $V_s$  = shear wave velocity of the soil, ft/s (m/s)  
 $v_h$  = post-impact hammer velocity, in./s (mm/s)  
 $v_o$  = reference velocity = 18.4 ft/s (5.6 m/s) from a free fall of 5.25 ft (1.6 m)  
 $v_r$  = ram impact velocity, ft/s (m/s)  
 $W_a$  = equipment weight at anchorage location, lbf (N)  
 $W_f$  = weight of the foundation, tons (kN)  
 $W_m$  = machine weight, tons (kN)  
 $W_r$  = rotating weight, lbf (N)  
 $y$  = generic representation of displacement (translational or rotational), in. (mm) or rad  
 $y'(\dot{y})$  = generic representation of velocity (translational or rotational), in./s (mm/s) or rad/s  
 $y''(\ddot{y})$  = generic representation of acceleration (translational or rotational), in./s<sup>2</sup> (mm/s<sup>2</sup>) or rad/s<sup>2</sup>  
 $y_c$  = crank pin displacement in local y-axis, or distance from the center of gravity to the base support, in. (mm)  
 $y_e$  = distance from the center of gravity to the level of embedment resistance, ft (m)  
 $z_c$  = crank pin displacement in local z-axis, in. (mm)  
 $z_p$  = piston displacement, in. (mm)  
 $\alpha$  = angle between battered piles and vertical piles, rad  
 $\alpha_h$  = ram rebound velocity to impact velocity ratio

- $\alpha_i$  = pile dynamic interaction factor in the  $i$ -th direction, subscription  $i = z$  (axial), HH (horizontal), MM (in phase rocking), MH (sway rocking)  
 $\alpha_j$  = coefficients,  $j = 1$   
 $\beta_i$  = rectangular footing coefficient for the  $i$ -th direction  
 $\beta_j, \gamma_j$  = coefficients,  $j = 1$  to 4  
 $\beta_m$  = material damping ratio  
 $\gamma_c$  = concrete density, lbf/ft<sup>3</sup> (kN/m<sup>3</sup>)  
 $\phi$  = phase angle  
 $\{\phi\}$  = mode shape vector  
 $\phi_i$  = mode shape factor  
 $\eta$  = tuning ratio  
 $\theta$  = phase angle, or angle between the direction of load action and the plane in which piles lie, rad  
 $\rho$  = soil mass density, lbf/ft<sup>3</sup> (kg/m<sup>3</sup>)  
 $\rho_p$  = pile mass density, lbf/ft<sup>3</sup> (kg/m<sup>3</sup>)  
 $\Delta$  = peak amplitude (translational or rotational), in. or rad  
 $\mu$  = coefficient of friction  
 $\nu$  = Poisson's ratio of the soil  
 $\omega$  = circular frequency of motion, rad/s  
 $\omega_d$  = damped circular natural frequency, rad/s  
 $\omega_i$  = undamped circular natural frequency for the  $i$ -th mode, rad/s  
 $\omega_n$  = undamped circular natural frequency, rad/s  
 $\omega_o$  = circular operating frequency of a machine or other driving force, rad/s  
 $\omega_{su}, \omega_{sv}$  = circular natural frequencies of a soil layer in horizontal ( $u$ ) and vertical ( $v$ ) directions, rad/s  
 $u, v,$   
 $\psi, \eta$  = subscriptions used for notating horizontal, vertical, rocking, and torsional direction, respectively

## 2.2—Definitions

Please refer to the latest version of ACI Concrete Terminology for a comprehensive list of definitions. Definitions provided herein complement that resource.

**root cause analysis**—collective term that describes a wide range of approaches, tools, and techniques used to uncover causes of problems.

## CHAPTER 3—FOUNDATION AND MACHINE TYPES

### 3.1—General considerations

The type, configuration, and installation of a foundation or support structure for dynamic machinery may depend on the following factors:

- Site conditions such as soil characteristics, topography, seismicity, climate, and other effects
- Machine base configuration such as frame size, cylinder supports, pulsation bottles, drive mechanisms, and exhaust ducts
- Process requirements such as elevation requirements with respect to connected process equipment and support requirements for piping
- Anticipated loads such as the equipment static weight, along with loads developed during construction, startup, operation, shutdown, and maintenance

e) Allowable amplitudes of vibration associated with each dynamic load case

f) Construction requirements such as limitations or constraints imposed by construction equipment, procedures, techniques, or the sequence of construction

g) Operational requirements such as accessibility, settlement limitations, temperature effects, and drainage

h) Maintenance requirements such as temporary access, laydown space, in-plant crane capabilities, and machine removal considerations

i) Regulatory factors, owner requirements, or building code provisions such as tied pile caps in seismic zones

j) Economic factors such as capital cost, useful or design service life, and replacement or repair cost

k) Environmental requirements such as secondary containment or special concrete coating requirements

l) Recognition that certain machines, particularly large reciprocating compressors, rely on the foundation to add strength and stiffness that is not inherent in the structure of the machine

### 3.2—Machine types

**3.2.1 Rotating machinery**—This category includes gas turbines, steam turbines, and other expanders; turbo-pumps and compressors; fans; motors; and centrifuges. These machines are characterized by the motion of rotating components.

Unbalanced forces in rotating machines are created when the mass centroid of the rotating component does not coincide with the center of rotation (Fig. 3.2.1). This dynamic force is a function of the mass of the rotating component, speed of rotation, and the magnitude of the eccentricity of offset. The offset or eccentricity should be minor under manufactured conditions when the machine is well balanced, clean, and without wear or erosion. Changes in alignment, operation near resonance, turbine blade loss, and other malfunctions or undesirable conditions can greatly increase the force applied to its bearings by the rotor.

**3.2.2 Reciprocating machinery**—For reciprocating machinery, such as compressors or diesel engines, a piston moving in a cylinder interacts with a gas through the kinematics of a slider crank mechanism driven by, or driving, a rotating crankshaft. Individual inertia forces from each cylinder are inherently unbalanced with dominant frequencies at one and two times the rotational frequency (Fig. 3.2.2).

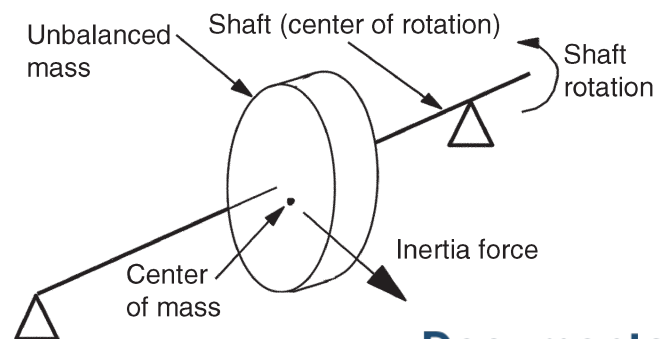


Fig. 3.2.1—Rotating machinery diagram

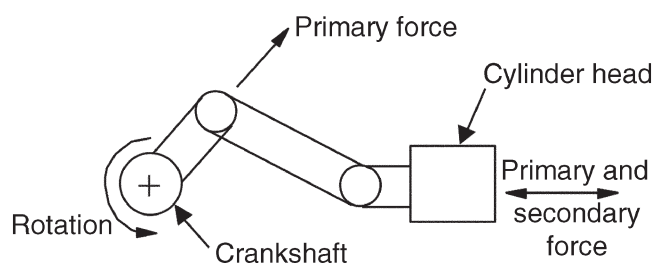


Fig. 3.2.2—Reciprocating machine diagram.

The unbalanced forces and moments generated by reciprocating machines with more than one piston are dependent on the crank arrangement. The optimum crank arrangement that minimizes loading is generally not possible because the mechanical design will be optimized to satisfy the operating requirements. This leads to piston/cylinder assemblies and crank arrangements that do not completely counter-oppose; therefore, unbalanced loads occur, which should be resisted by the foundation.

Individual cylinder fluid forces act outward on the cylinder head and inward on the crankshaft (Fig. 3.2.2). For a rigid cylinder and frame, these forces are internally balanced in the machine, but deformations of large machines can cause a significant portion of the forces to be transmitted to the mounts and into the foundation. Particularly on large reciprocating compressors with horizontal cylinders, it is inappropriate and unconservative to assume the compressor frame and cylinder are sufficiently stiff to internally balance all forces. Such an assumption has led to many inadequate mounts for reciprocating machines.

**3.2.3 Impulsive machinery**—Equipment, such as forging hammers and some metal-forming presses, operate with regulated impacts or shocks between different parts of the equipment. This shock loading is often transmitted to the foundation system of the equipment and can propagate into the surroundings and is a factor in the design of the foundation.

Closed die forging hammers typically operate by dropping a weight (ram) onto hot metal, forcing it into a predefined shape. While the intent is to use this impact energy to form and shape the material, there is significant energy transmission, particularly late in the forming process. During these final blows, the material being forged is cooling and less shaping takes place. Thus, pre-impact kinetic energy of the ram converts to post-impact kinetic energy of the entire forging hammer. As the entire hammer moves downward, it becomes a simple dynamic mass oscillating on its supporting medium. This system should be well damped so that the oscillations decay sufficiently before the next blow. Timing of the blows commonly range from 40 to 100 blows per minute. The ram weights vary from a few hundred pounds to 35,000 pounds (16 tons). Impact velocities in the range of 25 ft/s (7.6 m/s) are common. Open die hammers operate in a similar fashion but are often of two-piece construction with a separate hammer frame and anvil.

Forging presses perform a similar manufacturing function as forging hammers but are commonly mechanically or hydraulically driven. These presses form the material at

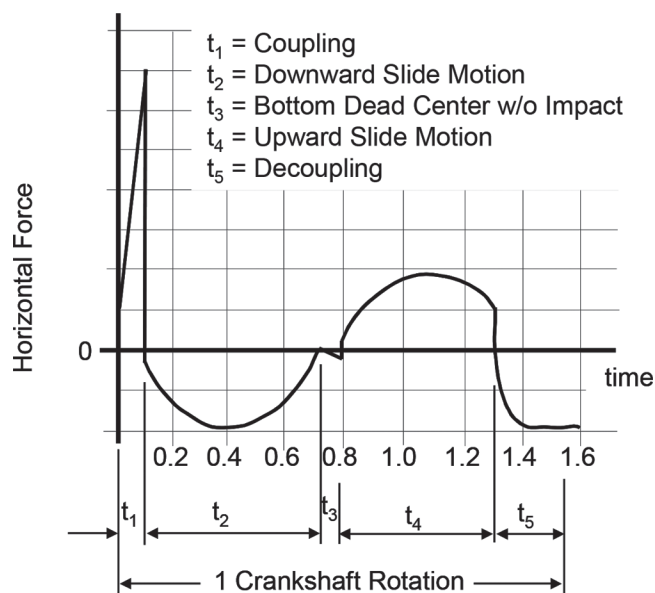


Fig. 3.2.3—Example of a forcing function for a forging press.

low velocities but with greater forces. The mechanical drive system generates horizontal dynamic forces that the engineer should consider in the design of the support system. Rocking stability of this construction is important. Figure 3.2.3 shows a typical example of a horizontal forcing function through one full stroke of a forging press.

Mechanical metal forming presses operate by squeezing and shearing metal between two dies. Because this equipment can vary greatly in size, weight, speed, and operation, forces and design criteria used for the foundation design can vary greatly. Speeds can vary from 30 to 1800 strokes per minute. Dynamic forces from the press develop from two sources: the mechanical imbalance of the moving parts in the equipment and the response of the press frame as the material is sheared (snap-through forces). Imbalances in the mechanics of the equipment can occur both horizontally and vertically. Generally, high-speed equipment is well balanced. Low-speed equipment is often not balanced because the inertia forces at low speeds are small. The dynamic forces generated by all of these presses can be significant as they are transmitted into the foundation and propagate into the subgrade.

**3.2.4 Other machine types**—Other machinery generating dynamic loads include rock crushers and metal shredders. While part of the dynamic load from these types of equipment tend to be based on rotating imbalances, there are also random characteristics to the dynamic forces that vary with the particular operation and design.

### 3.3—Foundation types

**3.3.1 Block-type foundation**—Dynamic machines are preferably located close to grade to minimize the elevation difference between the machine dynamic forces and the center of gravity of the machine-foundation system (Fig. 3.3.1). The low location also reduces the moments due to horizontal forces. The ability to use such a foundation

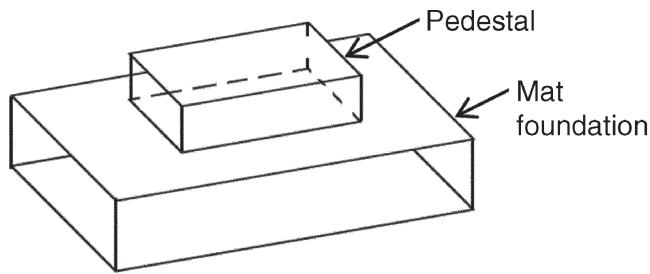


Fig. 3.3.1—Block-type foundation.

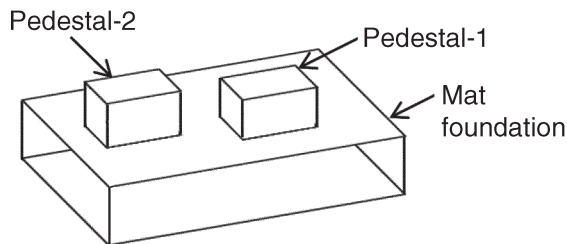


Fig. 3.3.2—Combined block-type foundation.

tion primarily depends on the quality of surface soils. Block foundations are nearly always designed as rigid structures. The dynamic response of a rigid block foundation depends only on the dynamic load, mass, dimensions, and soil characteristics.

**3.3.2 Combined block-type foundation**—Combined blocks are used to support closely spaced machines (Fig. 3.3.2). When designing combined block-type foundations, it is important to consider the combination of forces from two or more machines and possible lack of stiffness of a larger mat foundation.

**3.3.3 Tabletop-type foundation**—Elevated support is needed when access is required to the underside of the machinery for ducts, piping, maintenance platforms, or the machine has to be elevated for process reasons (Fig. 3.3.3). Tabletop structures are considered to be flexible, hence their response to dynamic loads can be quite complex and depend both on the motion of its discrete structural members (columns, beams, and footing) and the subgrade upon which it is supported.

**3.3.4 Tabletop with isolators**—Isolators (springs and dampers) located at the top of supporting columns are sometimes used to minimize the response to dynamic loading (Fig. 3.3.4). The effectiveness of isolators (or the degree of isolation, as defined in [Chapter 6](#)) depends on the damping, machine speed, and natural frequency of the foundation. Descriptions for the technical details of this type of support are provided in [6.3.3](#).

**3.3.5 Spring-mounted equipment on block foundation**—Occasionally, machinery are mounted on springs to minimize forces from connecting piping (Fig. 3.3.5). The springs are then supported on a block-type foundation. This arrangement has a dynamic effect similar to that for tabletops with vibration isolators. Other types of equipment are spring mounted to limit the transmission of dynamic forces. Technical details are described in [6.3.3](#).

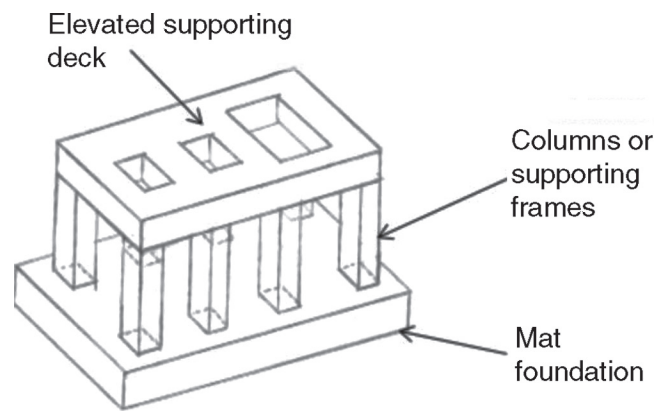


Fig. 3.3.3—Tabletop-type foundation.

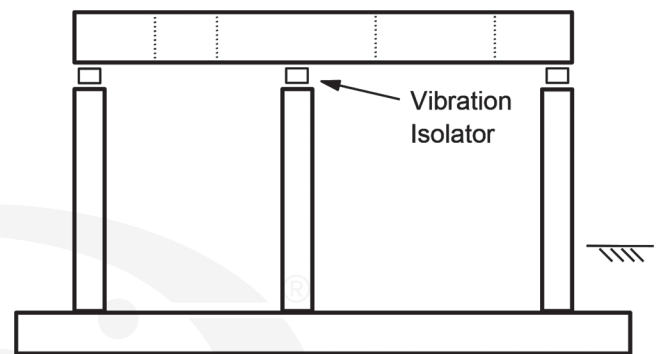


Fig. 3.3.4—Tabletop with isolators.

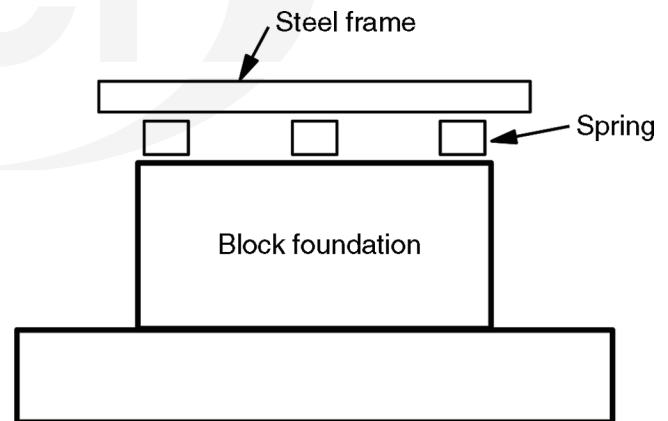


Fig. 3.3.5—Spring-mounted equipment on block foundation.

**3.3.6 Inertia block**—Dynamic equipment on a structure may be relatively small in comparison to the overall size of the structure. In this situation, dynamic machines are usually designed with a supporting inertia block to shift structure/machine natural frequencies away from machine operating speeds and reduce amplitudes by increasing the inertia (Fig. 3.3.6).

**3.3.7 Deep foundations**—Any of the previously mentioned foundation types may be supported directly on soil or on deep foundations such as piles or caissons (Fig. 3.3.7). Deep foundations are generally used where soft ground conditions result in low allowable bearing pressures and excessive

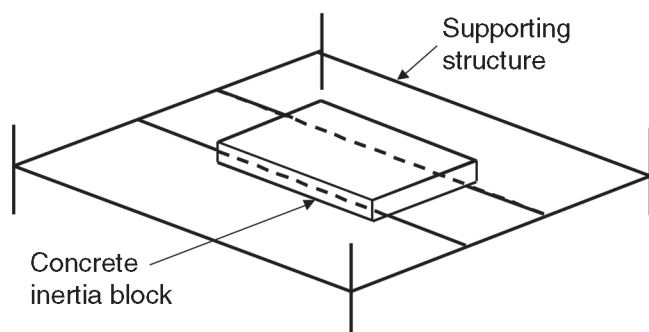


Fig. 3.3.6—Inertia block.

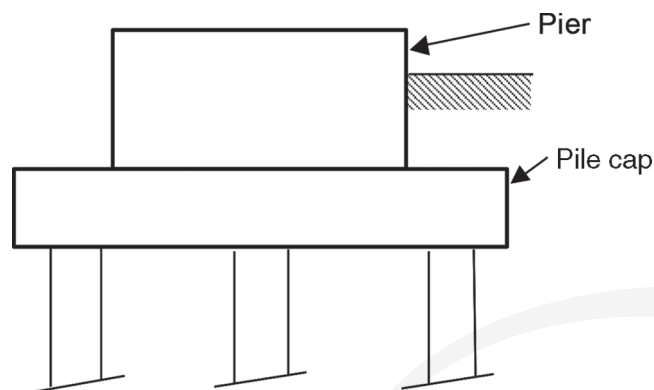


Fig. 3.3.7—Deep foundation.

settlement for a mat-type foundation. Piles use end bearing, frictional side adhesion, or a combination of both, to transfer axial loads into the underlying soil. Transverse loads are resisted by soil pressure bearing against the side of the pile cap and piles or by battered piles. Various types of piles are used, including drilled piers, caissons, auger cast piles, and driven piles.

## CHAPTER 4—DESIGN LOADS

### 4.1—Overview of design loads and criteria

The design of concrete foundations that support dynamic machinery involves several tasks:

- a) Defining the anticipated loads
- b) Properly transferring these loads to the foundation and then to the supporting medium (supporting soil, pile, or structure)
- c) Establishing the performance criteria, and providing for these through proper proportioning and detailing of structural members.

Behind this straightforward series of tasks lies the need for careful attention to the interfaces between machine, mounting system, concrete foundation, and the interaction with subgrade media.

The loads on machine foundations may be divided into one of three categories: equipment loads; environmental loads; and installation/maintenance loads.

The equipment loads on machine foundations may be both static and dynamic. Static loads are principally a function of the weights of the machine and all its auxiliary equip-

ment. Dynamic loads, which occur during the operation of the machine, result from forces generated by unbalance, inertia of moving parts, or both, and by the flow of fluids and gases for some machines. The magnitude of dynamic loads varies as a function of time; and primarily depends upon the machine's operating speed and the type, size, weight, and arrangement (position) of moving parts within the machine casing.

The basic goal in the design of a machine foundation is to limit the amplitude of motion for the machine and foundation. The goal is to neither endanger the satisfactory operation of the machine nor disturb people working in the immediate vicinity (Gazetas 1983). Allowable amplitudes depend on the speed, location, and criticality or function of the machine. Other limiting dynamic criteria affecting the design may include avoiding resonance and excessive transmissibility to the supporting soil or structure. Thus, a key ingredient to a successful design is the careful engineering analysis of the soil-foundation response to dynamic loads from the machine operation.

The foundation's response to dynamic loads can be significantly influenced by the supporting medium (soil or subgrade/bedrock) on which it is constructed. Consequently, critical soil parameters, such as the dynamic soil shear modulus, are preferably determined from a field investigation and laboratory tests rather than relying on generalized correlations based on broad soil classifications. Due to the inherent variability of soil, the dynamic response of machine foundations is often evaluated using a range of values for the critical soil properties (as discussed in [Chapter 5](#) and [Appendix A](#)).

Furthermore, a machinery support structure or foundation is designed with adequate structural strength to resist the worst possible combination of loads occurring over its design service life. This often includes limiting soil-bearing pressures to well within allowable limits to ensure a more predictable dynamic response and prevent excessive settlements and soil failures.

Foundations supporting reciprocating or rotating compressors, turbines, generators and motors, presses, and other machinery should withstand all the forces that may be imposed on them during their design service life. Machine foundations are unique because they may be subjected to significant dynamic loads during operation in addition to gravity, wind, and seismic loads. The magnitude and characteristics of the operating loads depend on the type, size, speed, and layout of the machine.

Generally, the weight of the machine, center of gravity, surface area(s), and operating speeds are readily available from the manufacturer of the machine. Establishing appropriate values for dynamic loads is best accomplished through careful communication and clear understanding between the machine manufacturer, owner, and engineer responsible for the foundation design. The communication should include but not be limited to the purpose, planned use for the loading information, and the definition of the information provided. It is in the best interest of all parties (machine manufacturer, engineer responsible for the foundation design, machine owner, and operator) to ensure effective definition and

communication of data and its appropriate use. Machines always experience some level of unbalance, vibration, and force transmitted through the bearings. Under some off-design conditions, such as wear, the forces may increase significantly. The machine manufacturer and engineer responsible for the foundation design should work together so that their combined knowledge achieves an integrated machine-foundation system that robustly serves the needs of its owner and operator and withstands design loads.

Usually the machine loads, including normal operation loads, emergency loads, catastrophic loads, and accidental loads, are provided by the machine manufacturer. Seismic loads and their distribution pattern from machine to the foundation at the machine support points should be provided by the manufacturer based on the machine configuration and mounting system. The total seismic loads for the foundation analysis should include the seismic loads induced from the machine mass and the seismic loads induced from foundation concrete mass. However, the engineer responsible for the foundation design should have a good understanding of each load and how the seismic load is transferred to the foundation. The machine seismic loads provided by the machine vendor may need to be adjusted based on the seismic design code requirements. Seismic loads from the mat foundation are, in general, out of phase from the seismic loads due to the machine and superstructure, unless the machine or superstructure are rigid. Therefore, the seismic shear due to the foundation and other rigid members ( $SV_R$ ) and the seismic shear due to the superstructure + machine and other flexible ( $SV_S$ ) members should be calculated by similar combinations such as square root of sum of the squares (SRSS) approach as follows

$$SV_{seismic} = (SV_R^2 + SV_S^2)^{0.5} \quad (4.1)$$

Close cooperation and coordination with the machine vendor is essential for a successful foundation design.

Sections 4.2 to 4.4 provide some of the commonly used loading descriptions, evaluations, or both, for determining machine-induced forces and other design loads for foundations supporting machinery. They include definitions and other information on static and dynamic loads. These loads should be provided by the machine manufacturer, and alternative assumptions should be made when such data are unavailable or are under-predicted.

## 4.2—Static machine loads

**4.2.1 Dead loads**—A major function of the foundation is to support gravity (dead) loads due to the weight of the machine, auxiliary equipment, piping, valves, and dead weight of the foundation structure. The weights of the machine components are normally supplied by the machine manufacturer. The distribution of the weight of the machine on the foundation depends on the location of support points and on the rigidity or flexibility of the machine frame. Typically, there are multiple support points and, thus, the distribution is statically indeterminate. In many cases, the machine manufacturer provides a loading diagram showing the vertical loads at each support point. Sometimes the center of gravity loca-

tion of the machine components provided by the machine manufacturer can be used to distribute machine dead loads to the supporting points.

Some machine manufacturers provide the weights of turbine and generator at certain support points. These forces are due to axial movements of the turbine-generator rotor only at the bearing casing connection points of the combined radial/axial bearing (thrust bearing). These weights should be used in the dynamic analysis, seismic analysis, or both, of the foundation.

**4.2.2 Live loads**—Live loads are produced by personnel, tools, and maintenance equipment and materials. The live loads used in design should be the maximum loads expected during the design service life of the machine. For most designs, live loads are uniformly distributed over the floor areas of platforms of elevated support structures or to the access areas around at-grade foundations. Typical live loads vary from 60 lbf/ft<sup>2</sup> (3 kPa) for personnel to as much as 250 lbf/ft<sup>2</sup> (12 kPa) for maintenance equipment and materials (ASCE/SEI 7). For steam turbine generator machines, it is common to also consider a value of 400 to 500 lbf/ft<sup>2</sup> (19 to 24 kPa) for machine parts (such as rotors and casings) in a laydown area.

**4.2.3 Normal operating loads**—Static operating loads include the weight of the machine contents during normal operation. Static operating loads include forces such as the drive torque developed by some machines at the connection between the drive mechanism and driven machinery. Static operating loads can also include forces caused by thermal growth of the machinery equipment and connecting piping. Time-varying (dynamic) loads generated by machines during operation are covered in 4.3.

**4.2.3.1 Operating torque loads**—Machines such as compressors, turbines, and generators require some form of drive mechanism, either integral with the machine or separate from it. When the drive mechanism is nonintegral, such as a separate electric motor, reciprocating engine, and gas or steam turbine, it produces a net external drive torque on the driven machine. For turbine-generators, fluid or magnetic coupling between the rotor and stator of the generator can produce an overturning torque about the longitudinal axis (shaft) equal in magnitude and opposite in direction on the driver and driven machine. For example, the normal operational torque load from turbines is oriented in the opposite direction to that of shaft rotation, whereas for the generator, the normal operational torque load is oriented in the direction of shaft rotation (Fig. 4.2.3.1a). The normal torque (sometimes called drive torque) is generally applied to the foundation as a static force couple in the vertical direction acting about the centerline of the shaft of the machine based on the highest expected output of the machine. The magnitude of the normal torque is often calculated as

$$NT = \frac{(5250)(P_s)}{f_o} \quad (\text{lbf-ft})$$

$$NT = \frac{(9550)(P_s)}{f_o} \quad (\text{N-m}) \quad (4.2.3.1a)$$



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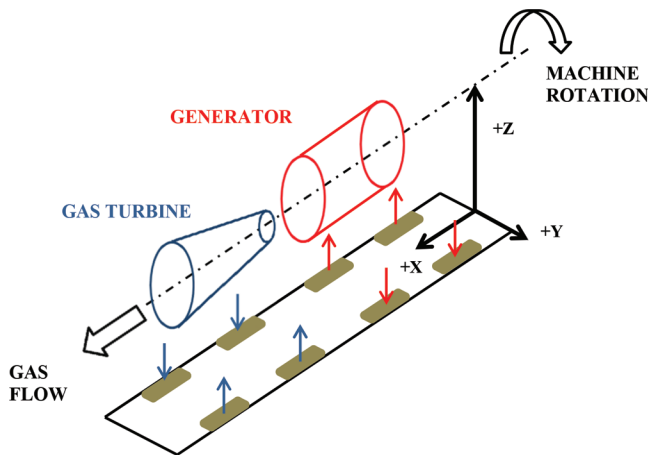


Fig. 4.2.3.1a—Equivalent forces for torque loads.

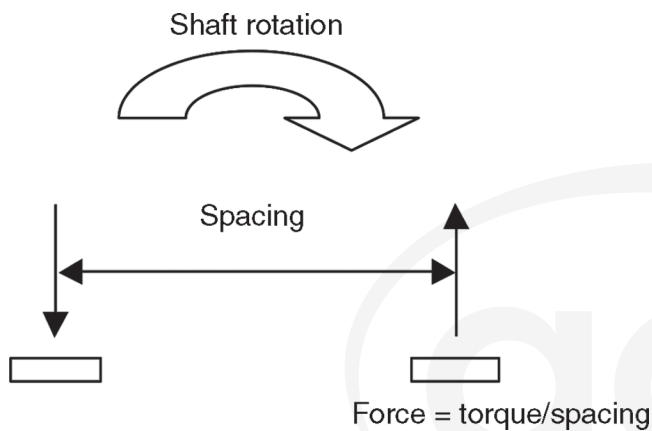


Fig. 4.2.3.1b—Equivalent forces for torque loads—torque resisted by longitudinal equipment soleplates.

The torque load is generally resolved into a vertical force couple by dividing it by the center-to-center distance between longitudinal soleplates or anchor points (Fig. 4.2.3.1b). When the machine is supported by transverse soleplates only, the torque is applied along the width of the soleplate assuming a straight line variation of force (Fig. 4.2.3.1c).

The torque on a generator stator is applied in the same direction as the rotation of the rotor and can be high due to startup or an electrical short circuit.

**4.2.3.2 Condenser vacuum loads**—For steam turbine-generator machines, usually a condenser is connected to the low-pressure (LP) turbine and may share one common foundation with the steam turbine-generator. The condenser generates vacuum loads on the steam turbine (LP turbine) when an expansion joint is provided between the condenser duct and the LP turbine exhaust neck. Condenser vacuum loads are normally based on zero absolute pressure. The vacuum loads can act vertically on the tabletop of the foundation if the condenser is located directly below the LP turbine, and it can also act laterally to the foundation if the condenser is not located directly below the LP turbine. The magnitude of the vacuum load can be estimated as the standard atmospheric pressure (14.7 psi [101.4 kPa]) times the

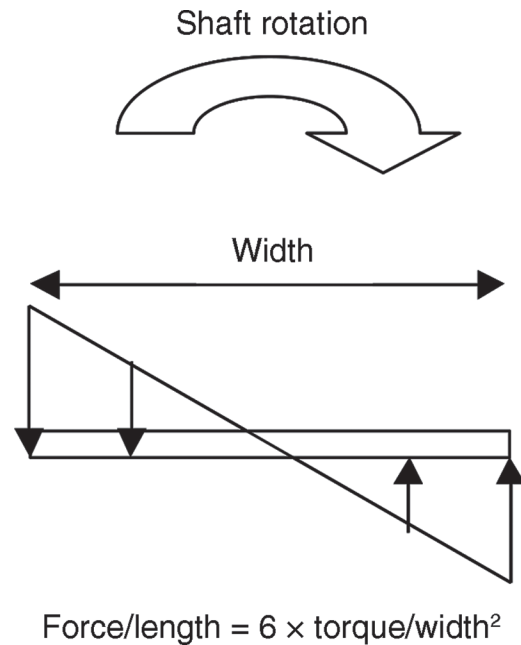


Fig. 4.2.3.1c—Equivalent forces for torque loads—torque resisted by transverse equipment soleplates.

neck duct opening area of the LP turbine. The vacuum loads are normally supplied by the machine manufacturer.

**4.2.3.3 Axial thrust loads**—Axial loads result from gas-flow momentum changes through gas turbines, or from the pressure differences between the gas inlet and exhaust. These loads generally are treated as a static load whose magnitude varies depending on the normal operation level exhaust of back pressure or shut-down level of exhaust back pressure. The axial thrust loads are normally supplied by the machine manufacturer.

**4.2.3.4 Piping loads**—Piping forces can be calculated based on the pipe stress analysis, taking into account the pipe weight (including insulation weight), fluid weight, fluid dynamics, thermal expansion/contractions, valve operations, and environmental load conditions (seismic, wind, and ice). These forces are transferred to the foundation by pipe supports, attachments, or anchors attached to the foundation along the piping system.

The piping load results from steam pipes and circulating water pipes attached to the turbine sections. The resultant piping loads acting on various points of support and guidance for the turbine casing reach the foundation as forces. During a valve trip, the turbine foundation can be subjected to significant piping valve trip loads. The piping loads usually are determined by the mechanical pipe engineers.

**4.2.3.5 Thermal loads**—Changing temperatures of machines and their foundations cause expansions, contractions, and distortions, causing the various parts to try to slide on the support surfaces. Thermal loads or self-straining loads may be divided into one of three loads types:

- Thermal friction loads
- Thermal expansion/contraction loads
- Temperature change and temperature gradient change on the foundation member cross section

Thermal friction load occurs at the machine supports (at sole plates) from static friction resistance to the thermal movement of the machine components. The magnitude of the resulting frictional forces depends on the magnitude of the temperature change, the location of the supports, and on the condition of the support surfaces. The thermal forces do not impose a net force on the foundation to be resisted by supporting medium (soil or piles) because the forces on any surface are balanced by equal and opposite forces on other support surfaces. Thermal forces, however, may govern the design of the grout system, pedestals, and hold downs and may cause thermal stresses in the supporting concrete structures, including girders, slabs, and columns/pedestals.

It is not possible to exactly calculate the exact thermal loading because it depends on numerous factors, including distance between anchor points, magnitude of temperature change, the material properties, the condition of the sliding surface, and the magnitude of the vertical load on each soleplate; it is, therefore, approximated. The static friction resistance has to be overcome before thermal movements occurs and many engineers use this value for the design loads. The magnitude of the frictional load may be calculated as follows

$$\text{friction force} = (\text{friction coefficient}) \times (\text{normal load acting through soleplate}) \quad (4.2.3.5)$$

The friction coefficient generally varies from 0.2 to 0.5. Loads acting through the soleplate include machine dead load, normal torque load, anchor bolt load, and piping loads.

Normally, the expected thermal deflection at various bearings is estimated by the manufacturer based on past field measurements on existing units. The machine erector then compensates for the thermal deflection during installation.

Mandke and Smalley (1989, 1992) and Smalley (1985) illustrate the effects of thermal loads and deflections in the concrete foundation of a large reciprocating compressor and their influence on the machine.

Heat transfer to the foundation can be by convection across an air gap and by conduction through points of physical contact. The resultant temperature gradients induce deformations, strains, and stresses.

When evaluating thermal stress, the calculations are strongly influenced by the stiffness and restraint against deformation for the structural member in question. Therefore, it is important to consider the self-relieving nature of thermal stress due to deformation and concrete cracking to prevent being overly conservative in the analysis. As the thermal forces are applied to the foundation member by the machine, the foundation member changes length and thereby provides reduced resistance to the machine forces. This phenomenon can have the effect of reducing the thermal forces from the machine.

Accurate determinations of concrete surface temperatures and thermal gradients are also important. Under steady-state normal operating conditions, temperature distributions across structural sections are usually linear. The air gap between the machine casing and foundation provides a

significant means for dissipating heat, and its effect should be included when establishing surface temperatures.

**4.2.4 Special loads for foundations**—The following special static load conditions are recommended in some proprietary standards for large equipment on foundations to ensure adequate strength and deflection control, especially if the appropriate information is not provided by the equipment manufacturer:

a) Vertical force equal to 50 percent of the total weight of each machine

b) Horizontal force (in the transverse direction) equal to 25 percent of the total weight of each machine

c) Horizontal force (in the longitudinal direction) equal to 10 percent of the total weight of each machine

These forces are additive to normal gravity loads and are considered to act at the centerline of the machine shaft. Loads a, b, and c are not considered to act concurrently with one another.

**4.2.5 Construction and maintenance loads**—Construction (erection and installation), hydro test, and maintenance loads are temporary loads required for installing or dismantling machine components during erection or maintenance. Erection loads are usually furnished in the manufacturer's foundation load drawing and should be used in conjunction with other specified dead, live, and environmental loads. Maintenance loads occur any time the equipment is being drained, cleaned, repaired, and realigned or when the components are being removed or replaced. Loads may result from maintenance equipment, davits, and hoists. Guidance on construction loads is provided in ASCE/SEI 37. Environmental loads can be reduced per ASCE/SEI 37 when combined with maintenance or construction loads.

A hydro-test load is the weight of water or other fluid that is placed temporarily into a vessel to confirm the integrity of the equipment.

**4.2.6 Machine catastrophic loads**—Machine catastrophic loads are caused by machine malfunction such as generator short circuit, turbine loss-of-blade/loss-of-bucket or a bowed rotor in a steam turbine. Catastrophic loads are generally applied at the bearings of the machine shaft vertically or horizontally and are not considered to act simultaneously with seismic loads, as there is a very low probability of these loads occurring at the same time. Often in the design for an electrical-short-circuit-induced torque load, the emergency drive torque is treated as an equivalent static load and the magnitude is determined by applying a magnification factor to the normal torque. Although these loads are time-dependent, they are normally given as equivalent static loads to simplify the foundation design efforts. These loads are normally provided by the machine manufacturer. Consultation with the generator manufacturer is necessary to establish the appropriate magnification factor.

### 4.3—Dynamic machine loads

**4.3.1 Rotary machine loads due to unbalanced masses**—These rotating loads are generated by the rotating motion of one or more impellers or rotors. Unbalanced rotating machines are created when the mass centroid of the rotating

part does not coincide with the axis of rotation. In theory, it is possible to precisely balance the rotating elements of rotating machinery. In practice, this is never achieved; slight mass eccentricities always remain. During operation, the eccentric rotating mass produces centrifugal forces that are proportional to the square of machine speed. Centrifugal forces generally increase during the design service life of the machine due to conditions such as machine wear, rotor play, and dirt accumulation.

A rotating machine transmits dynamic force to the foundation predominantly through its bearings (with small, generally unimportant exceptions such as seals and the air gap in a motor). The forces acting at the bearings are a function of the level and axial distribution of the unbalance, the geometry of the rotor and its bearings, the speed of rotation, and the dynamic characteristics of the rotor-bearing system. At or near a critical speed, the force from rotating unbalance can be substantially amplified, sometimes by a factor of five or more due to resonance.

The engineer should request the machine manufacturer to provide the following information:

a) Design levels of unbalance and basis—This information defines the unbalance level that is the basis for calculating the subsequent transmitted forces.

b) Dynamic forces transmitted to the bearing pedestals under the following conditions:

1. Unbalance levels over operating speed range
2. At highest vibration when running at critical speeds
3. At a vibration level where the machine is just short of tripping due to high vibration
4. Maximum level of upset condition the machine is designed to survive (for example, loss of one or more blades)

Items 1 and 2 document the predicted dynamic forces resulting from levels of unbalance assumed in design for normal operation. Using these forces, it is possible to predict the normal dynamic vibration of the machine and its foundation.

Item 3 identifies a maximum level of transmitted force under which the machine could operate continuously without tripping; the foundation should have the strength to tolerate such a dynamic force on a continuous basis.

Item 4 identifies the higher level of dynamic forces that could occur under occasional upset conditions over a short period of time. If the machine is designed to tolerate this level of dynamic force for a short period of time, then the foundation should also be designed to tolerate it for a similar period.

If an independent dynamic analysis of the rotor-bearing system is performed by the manufacturer, the end user, or a third party, the results likely provide some or all the above dynamic forces transmitted to the foundation.

By assuming that the dynamic force transmitted to the bearings equals the rotating unbalanced force generated by the rotor, information on unbalance can be used to estimate or confirm the transmitted force.

**4.3.1.1 Machine unbalance load provided by manufacturer**—When the mass unbalance (eccentricity) is known or stated by the manufacturer, the resulting dynamic force amplitude is

$$F_o = m_r e_m \omega_o^2 S_f \quad (\text{lbf}) \quad (4.3.1.1)$$

$$F_o = m_r e_m \omega_o^2 S_f / 1000 \quad (\text{N})$$

where  $F_o$  should be distributed to the bearings considering the rotor/shaft span between bearings as a simple beam and the center of mass of the rotor location within the span.

**4.3.1.2 Machine unbalance load (meeting industry criteria)**—Many rotating machines are balanced to an initial balance quality either in accordance with the manufacturer's procedures or as specified by the purchaser. **ISO 1940-1** and **ASA/ANSI S2.19** define balance quality in terms of a constant  $e_m \omega_o$ . For example, the normal balance quality  $Q$  for parts of process-plant machinery is 0.25 in./s (6.3 mm/s). Typical balance quality grade examples are shown in Table 4.3.1.2. To meet these criteria, a rotor intended for faster speeds should be better balanced than one operating at a slower speed. Using this approach, Eq. (4.3.1.1) can be rewritten as

$$F_o = m_r Q \omega_o S_f \quad (\text{lbf}) \quad (4.3.1.2a)$$

$$F_o = m_r Q \omega_o S_f / 1000 \quad (\text{N})$$

where  $Q$  is in inch-pound and SI systems is ft/s and mm/s, respectively.

**API 617** and **API 684** work with maximum residual unbalance  $U_{max}$  criteria for petroleum processing applications. The mass eccentricity is determined by dividing  $U_{max}$  by the rotor weight. For axial and centrifugal compressors with maximum continuous operating speeds greater than 25,000 rpm, API 617 establishes a maximum allowable mass eccentricity of  $10 \times 10^{-6}$  in. (254 nm). For compressors operating at slower speeds, the maximum allowable mass eccentricity is

$$e_m = 0.25/f_o \quad (\text{in.}) \quad (4.3.1.2b)$$

$$e_m = 6.3/f_o \quad (\text{mm})$$

where  $f_o$  is the less than or equal to 25,000 rpm.

This permitted initial mass eccentricity equates to ISO 1940-1 balance quality Grade G0.7, which is much smaller than the G2.5 that would be applied to this type of equipment (Table 4.3.1.2, turbo compressors). As such, the dynamic force calculated from this API 617 consideration will be quite small and a larger service factor might be used to calculate a realistic design force.

API 617 also identifies a limitation on the peak-to-peak vibration amplitude  $((12,000/f_o)^{0.5} \text{ mil } [25.4(12,000/f_o)^{0.5} \mu\text{m}])$  during mechanical testing of the compressor with the equipment operating at its maximum continuous speed. Some design firms use this criterion as a mechanical test vibration limit for the mass eccentricity  $e_m$ , using Eq. (4.3.1.2a), and a service factor  $S_f$  of 2.0 to calculate the zero to peak dynamic force amplitude as

**Table 4.3.1.2—Balance quality grades for selected groups of representative rigid rotors\***

| Balance quality guide | Product of $e\omega$ , in./s (mm/s) | Rotor types—general examples                                                                                                                                                                                                                                                                          |
|-----------------------|-------------------------------------|-------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| G1600                 | 63 (1600)                           | Crankshaft/drives of rigidly mounted, large, two-cycle engines                                                                                                                                                                                                                                        |
| G630                  | 25 (630)                            | Crankshaft/drives of rigidly mounted, large, four-cycle engines                                                                                                                                                                                                                                       |
| G250                  | 10 (250)                            | Crankshaft/drives of rigidly mounted, fast, four-cylinder diesel engines                                                                                                                                                                                                                              |
| G100                  | 4 (100)                             | Crankshaft/drives of fast diesel engines with six or more cylinders                                                                                                                                                                                                                                   |
| G40                   | 1.6 (40)                            | Crankshaft/drives of elastically mounted, fast four-cycle engines (gasoline or diesel) with six or more cylinders                                                                                                                                                                                     |
| G16                   | 0.6 (16)                            | Parts of crushing machines; drive shafts (propeller shafts, cardan shafts) with special requirements; crankshaft/drives of engines with six or more cylinders under special requirements                                                                                                              |
| G6.3                  | 0.25 (6.3)                          | Parts of process plant machines; centrifuge drums, paper machinery rolls, print rolls; fans; flywheels; pump impellers; machine tool and general machinery parts; medium and large electric armatures (of electric motors having at least 3-1/4 in. [80 mm] shaft height) without special requirement |
| G2.5                  | 0.1 (2.5)                           | Gas and steam turbines, including marine main turbines; rigid turbo-generator rotors; turbo-compressors; machine tool drives; medium and large electric armatures with special requirements; turbine driven pumps                                                                                     |
| G1                    | 0.04 (1)                            | Grinding machine drives                                                                                                                                                                                                                                                                               |
| G0.4                  | 0.015 (0.4)                         | Spindles, discs, and armatures of precision grinders                                                                                                                                                                                                                                                  |

\*Excerpted from ASA/ANSI S2.19.

$$F_o = \frac{W_r f_o^{1.5}}{322,000} \quad (4.3.1.2c)$$

Units of  $F_o$  and  $W_r$  are lbf and N in inch-pound and SI systems, respectively.

**4.3.1.3 Machine unbalance load (determined from a formula)**—Rotating machine manufacturers often do not report the unbalance that remains after balancing. Consequently, formulas are frequently used to ensure that foundations are designed for some minimum unbalance, which generally includes some allowance for unbalance increases over time. One general-purpose method assumes that balancing improves with machine speed and that there is a linear relationship between the unbalanced forces and the machine speed. The zero-to-peak centrifugal force amplitude from Eq. (4.3.1.2a) and (4.3.1.2b) and a service factor of  $S_f$  of 2.5 from one such commonly used expression is

$$F_o = \frac{W_r f_o}{6000} \quad (4.3.1.3)$$

where units of  $F_o$  and  $W_r$  are lbf and N in inch-pound and SI systems, respectively.

Equations (4.3.1.1), (4.3.1.2a), (4.3.1.2c), and (4.3.1.3) appear to be very different: the exponents on the speed of rotation vary from 1 to 1.5 to 2, constants vary widely, and different variables appear. Some equations use mass, others use weight. In reality, the equations are more similar than they appear. Given the right understanding of  $Q$  as a replacement for  $e_m \omega_o$ , Eq. (4.3.1.1), (4.3.1.2a), and (4.3.1.3) take on the same character. These equations then indicate that the design force at operating speed varies linearly with both the mass of the rotating body and the operating rotational speed. Once that state is identified, Eq. (4.3.1.1) can be adjusted to reflect the actual speed of rotation, and the dynamic centrifugal force is seen to vary with the square of

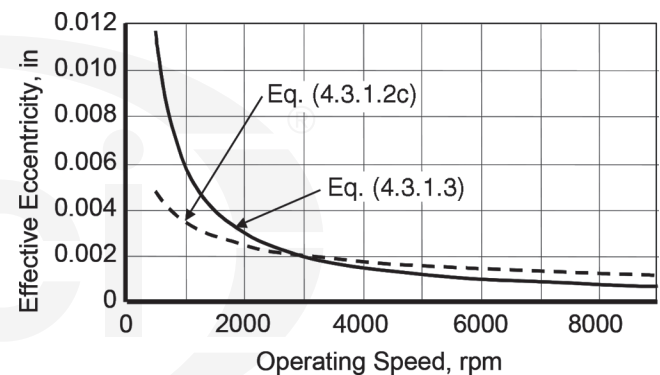


Fig. 4.3.1.3—Machine unbalanced loads—comparison of effective eccentricity.

the speed. Restating Eq. (4.3.1.2c) and (4.3.1.3) in the form of Eq. (4.3.1.1) allows for the development of an effective eccentricity implied within these equations with the comparison shown in Fig. 4.3.1.3. Equation (4.3.1.3) produces the same result as Eq. (4.3.1.2a) using  $Q = 0.25$  in./s (6.3 mm/s) and  $S_f = 2.5$ .

The centrifugal forces due to mass unbalance are considered to act at the center of gravity of the rotating part and vary harmonically at the speed of the machine in the two orthogonal directions perpendicular to the shaft axis. The forces in the two orthogonal directions are equal in magnitude and 90 degrees out of phase, and are transmitted to the foundation through the bearings. Schenck (1990) provides useful information about balance quality for various classes of machinery.

**4.3.1.4 Machine unbalance load (determined from trip vibration level and effective bearing stiffness)**—Because a rotor is often set to trip off at high vibration, it can be expected to operate continuously at any vibration level up to the trip limit. Given the effective bearing stiffness, it is possible to calculate the maximum dynamic force and moments

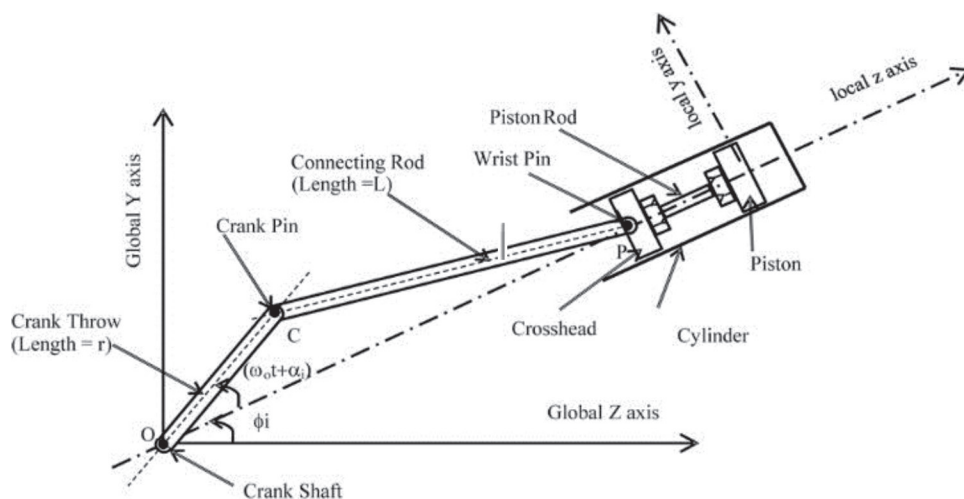


Fig. 4.3.2.1—Single crank mechanism.

$$F_o = V_{max} K_{eff} \quad (4.3.1.4)$$

To use this approach, the manufacturer should provide effective bearing stiffness from the bearing geometry and operating conditions (such as viscosity and speed) or provide the corresponding forces for the foundation design.

**4.3.1.5 Steam or gas turbine generator units**—The determination of the transmitted dynamic rotating unbalanced forces from the bearings to the foundation, for large rotating machines such as steam turbine generator units and combustion turbine generator units with complicated rotor configuration, shaft configurations, or both, can be completed by a dynamic analysis of the rotor-bearing system (the rotor/shaft power train dynamic analysis). This dynamic analysis should be completed under varying conditions of unbalance and at varying speeds. This dynamic analysis is based on the expected level of rotor/shaft unbalance that corresponds to a turbine generator rotor in operable condition but in need of balancing. The analysis can be completed by using an appropriate combination of computer programs for calculating bearing dynamic characteristics and the response of the rotor in the bearings due to the unbalance and the operating speeds. Such an analysis would usually be performed by the machine manufacturer. Results of such analyses, especially values for transmitted bearing forces, represent the best source of information for use by the engineer responsible for foundation design.

**4.3.1.6 Loads from multiple rotating machines**—If a foundation supports multiple rotating machines, the engineer should calculate the unbalanced dynamic force based on the mass, unbalance, and operating speed of each rotating component. The response to each rotating mass is then combined to determine the total response. Some practitioners, depending on the specific situation of machine size and criticality, find it advantageous to combine the unbalanced forces from each rotating component into a single resultant unbalanced force. The method of combining two dynamic forces is up to individual judgment and often involves some approximations. In some cases, loads or responses can be

added absolutely. In other cases, the loads are treated as out-of-phase so that twisting effects are increased. Often, the operating speed of the equipment should be considered. Even if operating speeds are nominally the same, the engineer should recognize that during normal operation, the speed of the machines will vary and beating effects can develop. Beating effects develop as two machines operate at close to the same speed. At one point in time, responses to the two machines are additive and motions are maximized. A short time later, the responses cancel each other and the motions are minimized. The net effect is a continual cyclic rising and falling of motion.

**4.3.2 Reciprocating machine loads**—Internal-combustion engines, piston-type compressors and pumps, some metal forming presses, steam engines, and other machinery are characterized by the rotating motion of a master crankshaft and the linear reciprocating motion of connected pistons or sliders. The motion of these components causes cyclically varying forces, often called reciprocating forces.

**4.3.2.1 Primary and secondary reciprocating loads**—The simplest type of reciprocating machine uses a single crank mechanism, as shown in Fig. 4.3.2.1. The idealization of this mechanism consists of a piston that moves within a guiding cylinder, a crank throw of length  $r$  that rotates about a crank shaft, and a connecting rod of length  $L$ . The connecting rod is attached to the piston at Point P and to the crank at Point C. The Wrist Pin P oscillates while the Crank Pin C follows a circular path. This idealized single cylinder illustrates the concept of a machine producing both primary and secondary reciprocating forces.

If the crank is assumed to rotate at a constant angular velocity  $\omega_o$ , the translational acceleration of the piston along its axis may be evaluated. If  $z_p$  is defined as the piston displacement toward the crankshaft (local z-axis), an expression can be written for  $z_p$  at any time  $t$ . Further, the velocity and acceleration can also be obtained by taking the first and second derivatives of the displacement expression with respect to time. The displacement, velocity, and acceleration expressions for the motion of the piston are as follows.

$$z_p = \left( r + \frac{r^2}{4L} \right) - r \left( \cos \omega_o t + \frac{r}{4L} \cos 2\omega_o t \right) \quad (4.3.2.1a)$$

$$\dot{z}_p = r\omega_m \left( \sin \omega_o t + \frac{r}{2L} \sin 2\omega_o t \right) \quad (4.3.2.1b)$$

$$\ddot{z}_p = r\omega_m^2 \left( \cos \omega_o t + \frac{r}{L} \cos 2\omega_o t \right) \quad (4.3.2.1c)$$

Note that the expressions contain two terms, each with a sine or cosine; the term that varies with the frequency of the rotation,  $\omega_o$ , is referred to as the primary term whereas the term that varies at twice the frequency of rotation,  $2\omega_o$ , is called the secondary term.

Similar expressions can be developed for the motion of the rotating parts of the crank along the local z-axis (parallel to piston movement) and local y-axis (perpendicular to piston movement). If any unbalance in the crankshaft is replaced by a mass concentrated at the Crank Pin C, such that the inertia forces are the same as in the original system, the following terms for motion at Point C can be written.

$$y_c = r \sin \omega_o t \quad (4.3.2.1d)$$

$$\dot{y}_c = r\omega_m \cos \omega_o t \quad (4.3.2.1e)$$

$$\ddot{y}_c = -r\omega_o^2 \sin \omega_o t \quad (4.3.2.1f)$$

$$z_c = r(1 - \cos \omega_o t) \quad (4.3.2.1g)$$

$$\dot{z}_c = r\omega_o \sin \omega_o t \quad (4.3.2.1h)$$

$$\ddot{z}_c = r\omega_o^2 \cos \omega_o t \quad (4.3.2.1i)$$

Identifying a part of the connecting rod (usually one-third of its mass) plus the piston as the reciprocating mass  $m_{rec}$  concentrated at Point P and designating the remainder of the connecting rod plus the crank as the rotating mass  $m_{rot}$  concentrated at Point C, expressions for the unbalanced forces are as follows.

Force parallel to piston movement:

$$F_z = (m_{rec} + m_{rot})r\omega_o^2 \cos \omega_o t + m_{rec} \frac{r^2 \omega_o^2}{L} \cos 2\omega_o t \quad (4.3.2.1j)$$

Force perpendicular to piston movement:

$$F_y = m_{rot} r \omega_o^2 \sin \omega_o t \quad (4.3.2.1k)$$

Note that Eq. (4.3.2.1j) consists of two terms, a primary force:

$$(m_{rec} + m_{rot})r\omega_o^2 \cos \omega_o t \quad (4.3.2.1l)$$

and a secondary force:

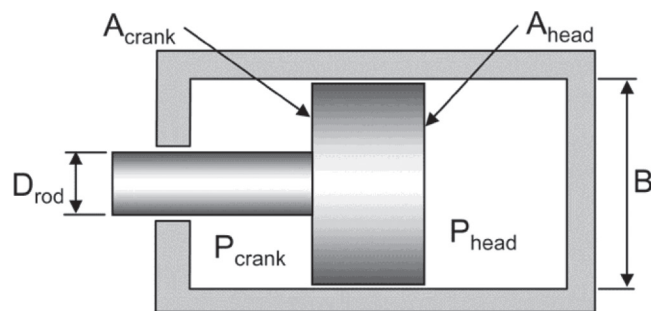


Fig. 4.3.2.2—Schematic diagram of double-acting compressor cylinder and piston.

$$m_{rec} \frac{r^2 \omega_o^2}{L} \cos 2\omega_o t \quad (4.3.2.1m)$$

whereas Eq. (4.3.2.1k) has only a primary component.

**4.3.2.2 Compressor gas loads**—A reciprocating compressor raises the pressure of a gas flow by imparting reciprocating motion on a piston within a cylinder. The piston normally compresses gas during both directions of reciprocating motion. As gas flows to and from each end, the pressure of the gas increases as it is compressed by each stroke of the piston. The increase in pressure within the cylinder creates reaction forces on the head and crank ends of the piston that alternate as gas flows to and from each end of the cylinder.

The gas force applied to the piston rod equals the instantaneous difference between the pressure force acting on the head and crank end of the piston, as shown in Fig. 4.3.2.2 for typical single crank reciprocating machines.

The following formulation can be used to estimate the maximum force acting on the piston rod of an individual double-acting cylinder.

$$F_{rod} = [(P_{head})(A_{head}) - (P_{crank})(A_{crank})]F_1 \quad (4.3.2.2a)$$

$$A_{head} = (\pi/4) B_c^2 \quad (4.3.2.2b)$$

$$A_{crank} = (\pi/4)(B_c^2 - D_{rod}^2) \quad (4.3.2.2c)$$

The head and crank end pressures vary continuously and the differential force takes both positive and negative net values during each cycle of piston motion. The normal approach is to establish the head and crank pressures using the maximum and minimum suction and discharge pressures. For design purposes, it is common to multiply Eq. (4.3.2.2a) by a factor  $F_1$  to help account for the natural tendency of gas forces to exceed the values based directly on suction and discharge pressures due to flow resistances and pulsations. Machines with good pulsation control and low external flow resistance may achieve  $F_1$  as small as 1.1. For machines with low compression ratio, high pulsations, or highly resistive flow through piping and nozzles,  $F_1$  can approach 1.5 or even higher. A reasonable working value for  $F_1$  is 1.15 to 1.2.

Preferably, the maximum rod force resulting from gas pressures is based on knowledge of the continuous variation of pres-

sure in the cylinder (measured or predicted). In a repair situation, measured cylinder pressure variation using a cylinder analyzer provides the most accurate value of gas forces. Even without cylinder pressure analysis, extreme operating values of suction and discharge pressure for each stage should be recorded before the repair and used in Eq. (4.3.2.2a).

On new compressors, the engineer should ask the machine manufacturer to provide values for maximum compressive and tensile gas loads on each cylinder rod and to recommend a value of  $F_1$  if these forces are based on suction and discharge pressures.

Gas forces act on the crankshaft with an equal and opposite reaction to the force on the cylinder. Thus, crankshaft and cylinder forces globally balance each other. Between the crankshaft and the cylinder, however, the metal compressor frame stretches or contracts in tension or compression under the action of the gas forces. The forces due to frame deflections are transmitted to the foundation through connections with the compressor frame. When acting without slippage, the frame and foundation become an integral structure and together stretch or contract under the gas loads.

The magnitude of gas force transferred into the foundation depends on the relative flexibility of the compressor frame. A very stiff frame transmits only a small fraction of the gas force while a very flexible frame transmits most or all the force. Similar comments apply to the transfer of individual cylinder inertia forces.

Based on limited comparisons using finite element analysis (Smalley 1988), the following guidelines are suggested for gas and inertia force loads transmitted to the foundation by a typical compressor

$$F_{block} = F_{rod}/F_{red} \quad (4.3.2.2d)$$

$$(F_{bolt})_{CHG} = [(F_{rod})/(N_{bolt})_{CHG}]/F_{red} \quad (4.3.2.2e)$$

$$(F_{bolt})_{frame} = [(F_{unbalance})/(N_{bolt})]/F_{red} \quad (4.3.2.2f)$$

The factor  $F_{red}$  is used to simplify a complex problem, thus avoiding the application of unrealistically high loads on the anchor bolts and the foundation block. The mechanics involved in transmitting loads are complex and cannot easily be reduced to a simple relationship between a few parameters beyond the given load equations. A detailed finite-element analysis of metal compressor frame, chock mounts, concrete block, and grout will account for the relative flexibility of the frame and its foundation in determining individual anchor bolt loads and implicitly provide a value for  $F_{red}$ . If the frame is very stiff relative to the foundation, the value for  $F_{red}$  will be higher, implying more of the transmitted loads are carried by the frame and less by the anchor bolts and foundation block. Based on experience, a value of 2 for this factor is conservatively low; however, higher values have been seen with frames designed to be especially stiff.

Simplifying this approach, Smalley and Harrell (1997) suggest using a finite element analysis to calculate forces transmitted to the anchor bolts. If a finite element analysis is not possible, the engineer should obtain from the machine

manufacturer or calculate the maximum horizontal gas force and maximum horizontal inertia force for any throw or cylinder. The mounts, anchor bolts, and blocks are then designed for

$$F_{THROW} = (\text{greater of } F_{GMAX} \text{ or } F_{IMAX})/2 \quad (4.3.2.2g)$$

**4.3.2.3 Reciprocating inertia loads for multi-cylinder machines**—As a practical matter, most reciprocating machines have more than one cylinder, and manufacturers arrange the machine components in a manner that minimizes the net unbalanced forces. For example, rotating parts such as the crankshaft can be balanced by adding or removing correcting weights. Translating parts such as pistons and those that exhibit both rotation and translation such as connecting rods can be arranged in such a way as to minimize the unbalanced forces and moments generated. Seldom, if ever, is it possible to perfectly balance reciprocating machines.

The forces generated by reciprocating mechanisms are functions of the mass, stroke, piston arrangement, connecting rod size, crank throw orientation (phase angle), and the mass and arrangement of counterweights on the crankshaft. For this reason, calculating the reciprocating forces for multi-cylinder machines can be quite complex and are therefore normally provided by the machine manufacturer. If the machine is an integral engine compressor, it can include, in one frame, cylinders oriented horizontally, vertically, or in between, all with reciprocating inertias.

Some machine manufacturers place displacement transducers and accelerometers on strategic points on the machinery. They can then measure displacements and accelerations at those points for several operational frequencies to determine the magnitude of the unbalanced forces and couples for multi-cylinder machines.

**4.3.2.4 Estimating reciprocating inertia forces from multi-cylinder machines**—In cases where the manufacturer's data are unavailable or components are being replaced, the engineer should use hand calculations to estimate the reciprocating forces from a multi-cylinder machine. One such procedure for a machine having  $n$  number of cylinders is discussed by Mandke and Troxler (1992).

**4.3.3 Impulsive machine loads**—The impulsive load generated by a forging hammer is caused by the impact of the hammer ram onto the hammer anvil. This impact process transfers the kinetic energy of the ram into kinetic energy of the entire hammer assembly. The post-impact velocity of the hammer is represented by

$$v_h = \frac{M_r}{M_h}(1 + \alpha_h)v_r \quad (4.3.3a)$$

General experience indicates that  $\alpha_h$  is approximately 60 percent for many forging hammer installations. From that point, the hammer foundation performance can be assessed as a rigid body oscillating as a single degree of freedom system with an initial velocity of  $v_h$ .

For metal-forming presses, the dynamic forces develop from two sources: the mechanical movement of the press components and material-forming process. Each of these forces is unique to the press design and application and needs to be evaluated with proper information from the press manufacturer and the owner.

The press mechanics often include rotating and reciprocating components. The dynamic forces from these individual pieces follow the rules established in earlier sections of this document for rotating and reciprocating components. Only the press manufacturer familiar with all the internal components can knowledgeably calculate the specific forces. Figure 4.3.1.3 presents a horizontal force time-history for a forging press. Similar presses can be expected to have similar characteristics; however, the specific values and timing data differ.

The press drive mechanisms include geared and direct-drive systems. Depending on the design, these drives may or may not be balanced. The press slide travels vertically through a set stroke of 1/2 in. (12 mm) to several inches (millimeters) at a given speed. Some small presses may have inclinable beds so that the slide is not moving vertically. It is often adequate to assume that the slide moves in a vertical path defined by a circularly rotating crankshaft; that is,

$$d_s(t) = \frac{S}{2} \sin(\omega_o t) \quad (4.3.3b)$$

This leads to a dynamic inertia force from the slide of

$$F_s(t) = m_d \omega_o^2 \frac{S}{2} \sin(\omega_o t) \quad (\text{lbf}) \quad (4.3.3c)$$

$$F_s(t) = m_d \omega_o^2 \frac{S}{2} \sin(\omega_o t) / 1000 \quad (\text{N})$$

This assumption is based on simple circular motions and simple linkages. Other systems may be in place to increase the press force and improve the timing. These other systems may increase the acceleration of the unbalanced weights and thus alter the magnitude and frequency components of the dynamic force transmitted to the foundation.

#### 4.4—Environmental loads

**4.4.1 Wind loads**—Loads due to wind on the surface areas of the machine, auxiliary equipment, and the support foundation are based on the design wind speed for the particular site and are normally calculated in accordance with the governing local code or standard. Wind loads rarely govern the design of machine foundations except, perhaps, when the machine is located in an enclosure that is also supported by the foundation.

When designing machine foundations and support structures, most practitioners use the wind load provisions of **ASCE/SEI 7**. The analytical procedure of **ASCE/SEI 7** provides wind pressures and forces for use in the design of the main wind-force-resisting systems and anchorage of machine components.

Most structural systems involving machines and machine foundations are relatively stiff (natural frequency in the lateral direction greater than 1 Hz). Consequently, the systems can be treated as rigid with respect to the wind gust effect factor, and simplified procedures can be used. If the machine is supported on flexible isolators and is exposed to the wind, the rigid assumption may not be reasonable and more elaborate treatment of the gust effects is necessary as described in **ASCE/SEI 7** for flexible structural systems.

Appropriate consideration of the exposure conditions and importance factors is also recommended to be consistent with the facilities' requirements.

**4.4.2 Seismic loads**—Machinery foundations located in seismically active regions are analyzed for seismic loads. These loads should be determined as equivalent static loads or dynamic loads per applicable codes and standards.

Building codes and standards such as the International Building Code (**International Code Council 2015**), **ASCE/SEI 7**, **FEMA P-750**, and **SEAOC Blue Book (2009)** contain provisions for design of nonstructural components, including dynamic machinery. The seismic loads and design requirements for nonbuilding structures, including elevated tabletop-type foundation pedestals, are also contained in the provisions of these codes. For machinery supported above grade or on more flexible elevated pedestals, seismic amplification factors are also specified. For where isolators are used (such as spring/damper/absorber configurations) to support the machine, **ASCE/SEI 7** provides criteria for seismic load calculation and design.

Depending on the weight of the machine with respect to the total weight of the machine-foundation system, the design seismic forces of the machine (nonbuilding structure) can be determined either by treating the machine weight as an integral part of the foundation weight (**ASCE/SEI 7**), or by treating the machine as a nonstructural component attached to the foundation (**ASCE/SEI 7**). For the seismic design of tabletop foundations, the foundation is usually treated as a supporting structure that should be designed in accordance with the requirements of **ASCE/SEI 7**.

For most heavy machine foundations, it is common to treat the machine weight as an integral part of the foundation for determining machine seismic load effects. In the seismic weight calculation, the full or percentage of the live load should be used per **ASCE/SEI 7**.

Horizontal and vertical components of seismic loads should be investigated in the load conditions and combinations in the following sections per **ASCE/SEI 7**.

**4.4.3 All other loads**—Environmental loads such as earth, water, rain, snow, and ice should be included in the design per applicable building codes.

#### 4.5—Load conditions

During their lives, machinery equipment support structures and foundations undergo different load conditions, including construction, testing, shutdown, maintenance, and normal and abnormal operation. For each loading condition, there can be one or more combinations of loads that apply

**Table 4.6—Load classifications for ultimate strength design**

| Design loads                                                                                                                                                                                                                                                       | Load classification |
|--------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|---------------------|
| Weight of structure, equipment, internals, insulation, and platforms<br>Fluid loads during testing and operation<br>Anchor and guide loads                                                                                                                         | Dead                |
| Platform and walkway loads<br>Materials to be temporarily stored during maintenance<br>Materials normally stored during operation such as tools and maintenance equipment<br>Vibrating equipment forces<br>Impact loads for hoist and equipment handling utilities | Live                |
| Seismic loads<br>Snow, ice, or rain loads<br>Wind loads                                                                                                                                                                                                            | Environmental       |

Notes: Thermal or self-straining loads classifications should be evaluated per requirements of ASCE/SEI 7.

Load due to lateral earth pressure, ground water pressure, or pressure of the bulk materials classifications should be evaluated per guidelines of ASCE/SEI 7.

to the structure or foundation. The following load conditions are generally considered in design:

- a) Construction condition represents the design loads that act on the structure/foundation during its construction.
- b) Testing condition represents the design loads that act on the structure/foundation while the equipment being supported is undergoing testing, such as hydro test.
- c) Normal operating condition represents the design loading during periods of normal equipment operation.
- d) Abnormal operating condition represents the design loading during periods when unusual or extreme operating loads act on the structure/foundation.
- e) Maintenance condition (empty or shutdown) represents the design loads that act on the structure when the supported equipment is at its least weight due to removal of process fluids, applicable internals, or both, as a result of maintenance or other out-of-service disruption.

#### 4.6—Load combinations

Table 4.6 shows the general classification of loads for use in determining the applicable load factors in strength design (ACI 318; ASCE/SEI 7; ASCE/SEI 37). In considering soil stresses or loading on piles, the normal approach is working stress design without load factors and with overall factors of safety identified as appropriate by geotechnical engineers. The load combinations frequently used for the various load conditions are as follows:

- a. Construction
  - i. Dead loads + construction loads
  - ii. Dead loads + construction loads + reduced wind loads + snow, ice, or rain loads
  - iii. Dead loads + construction loads + seismic loads + snow, ice, or rain loads
- b. Hydro testing
  - i. Dead loads + test loads
  - ii. Dead loads + test loads + live loads + snow, ice, or rain loads

- iii. Dead loads + test loads + reduced wind loads + snow, ice, or rain loads

- c. Normal operation

- i. Dead loads + machine operation loads + live loads (for multi bearings machine post-alignment deflection check)
- ii. Dead loads + machine operation loads + live loads + wind loads + snow, ice, or rain loads
- iii. Dead loads + machine operation loads + seismic loads + snow, ice, or rain

- d. Abnormal operation

- i. Dead loads + upset (abnormal, emergency, catastrophic) machine loads + live loads + reduced wind loads

- e. Maintenance loads

- i. Dead loads + maintenance loads + live loads + snow, ice, wind or rain loads

Environmental loads can be reduced per ASCE/SEI 37 when combined with maintenance or construction loads due to the low probability of the loads occurring at the same time.

## CHAPTER 5—IMPEDANCE OF THE SUPPORTING MEDIUM

### 5.1—Overview and use of soil impedance

The foundation dynamic response depends on the stiffness and damping characteristics of the machine-foundation-soil system. This section presents a general introduction to this subject and a summary of approaches and formulas often used to evaluate the stiffness and damping of both soil-supported and pile foundations. These stiffness and damping relationships, collectively known as impedance, are used for determining both free-vibration performance and motions of the foundation system due to the dynamic loading associated with the machine operation.

The simplest mathematical model used in dynamic analysis of machine-foundation systems is a single-degree-of-freedom representation of a rigid mass vertically supported on a single spring and damper combination (Fig. 5.1(a)). This model is applicable if the center of gravity of the machine-foundation system is directly over the center of soil resistance and the resultant of the dynamic forces (acting through a center of force [CF]) is a vertical force passing through the center of gravity. The vertical impedance ( $k_v^*$ ) of the supporting medium is necessary for this model.

The next level of complexity is a two-degree-of-freedom model commonly used when lateral dynamic forces act on the system (Fig. 5.1(b)). Because the lines of action of the applied forces and the soil resistance do not coincide, the rocking and translational motions of the system are coupled. For this model, the engineer needs to calculate the horizontal translational impedance ( $k_u^*$ ) and the rocking impedance ( $k_\psi^*$ ) of the supporting medium. These impedance values, especially the rocking terms, are usually different about different horizontal directions.

Application of the lateral dynamic forces can cause a machine-foundation system to twist about a vertical axis. To model this behavior, the engineer needs to determine

the torsional impedance ( $k_{\eta}^*$ ) of the supporting medium. As in the vertical model, if the in-plan eccentricities between the center of gravity and center of soil resistance are small, this analysis is addressed with a single-degree-of-freedom representation.

When the machine-foundation system is infinitely rigid and on the surface, the impedance ( $k_{u1}^*$ ,  $k_{u2}^*$ ,  $k_v^*$ ,  $k_{\psi1}^*$ ,  $k_{\psi2}^*$ ,  $k_{\eta}^*$ ) can be represented as six frequency-dependent spring-mass systems (three translational and three angular) and six frequency-dependent viscous or hysteretic dampers (three translational and three angular) that act at the centroid of the contact area between the foundation and the supporting medium (Fig. 5.1(c)). If the support is provided by piles or isolators, the equivalent springs and dampers act at the center of stiffness of those elements. When the foundation is embedded, additional terms can be introduced as spring and damper elements acting on the sides of the foundation. The effective location of the embedment impedance is determined rationally based on the character of the embedment (Beredugo and Novak 1972; Novak and Beredugo 1972; Novak and Sachs 1973; Novak and Sheta 1980; Lakshmanan and Minai 1981; Gazetas 1991).

The next level of complexity addresses flexible foundations. The structural model of these foundations is typically created using finite element techniques. The flexibility of the structure might be modeled with beam-type elements, two-dimensional plane stress elements, plate bending elements, three-dimensional solid elements, or some combination of these.

The supporting medium for these flexible foundations is addressed in one of three basic manners. The simplest approach uses a series of translational springs at each contact point between the structure and the soil. These spring parameters are calculated from the spring constants determined considering portions of the mat foundation small enough to be assumed rigid (Arya et al. 1979) or by some similar approximated procedure. The other two approaches use the finite element method, boundary element method, or both, for calculating the dynamic impedance matrix of the supporting medium. Such approaches are typically used for major, complex situations where a high degree of refinement is justified (for example, complex soil stratigraphy, irregular foundation geometry, or both). A detailed discussion of these techniques is outside the scope of this report.

Nevertheless, the fundamental principles and concepts discussed herein for infinitely rigid foundation also apply to flexible foundations. In summary, the main difference between flexible and rigid foundations is that flexible foundations are modeled using finite or boundary element methods, which leads to a dynamic impedance matrix rather than six dynamic impedances. This matrix will contain all degrees of freedom associated with nodes located at the boundary between the foundation and its support.

## 5.2—Basic dynamic concepts

The basic mathematical model used in the dynamic analysis of various infinitely rigid machine-foundation systems is a lumped mass with a spring and dashpot (Fig. 5.2a). This lumped mass includes the machine mass, foundation mass,

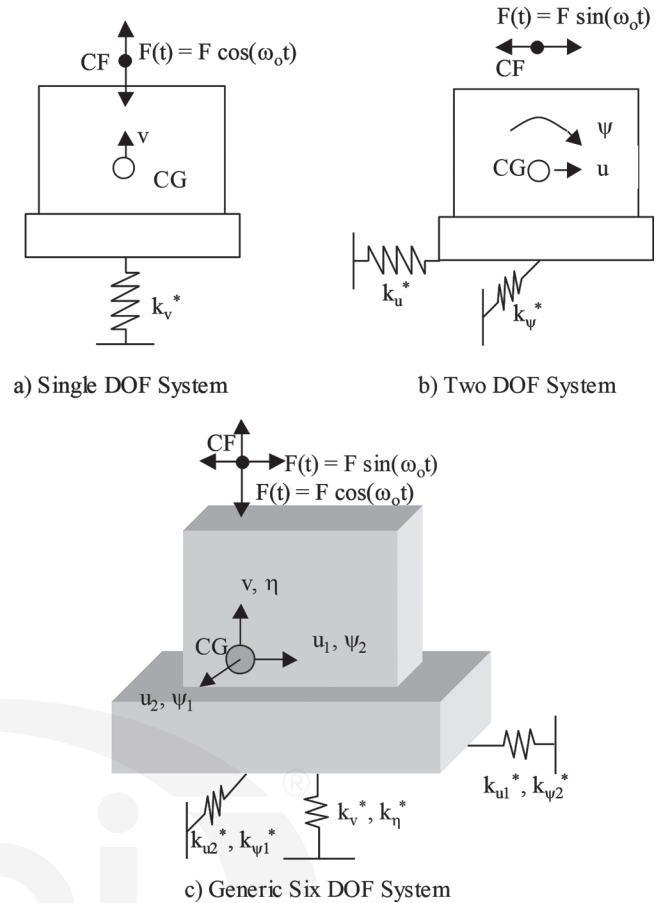


Fig. 5.1—Common dynamic models.

and often an added soil mass. A mass  $m$ , free to move only in one direction—for example, vertical—is said to have only a single degree of freedom (SDOF) (Fig. 5.1(a)). The behavior of the mass depends on the nature of both the spring and the dashpot.

The spring, presumed to be massless, represents the elasticity of the system and is characterized by the stiffness constant  $k$ . The stiffness constant is defined as the force that produces a unit displacement of the mass. For a general displacement  $u$  of the mass, the force in the spring (the restoring force) is  $ku$ .

In dynamics, the displacement varies with time  $t$ , thus,  $u = u(t)$ . Because the spring is massless, the dynamic stiffness constant  $k$  is equal to the static constant  $k_{st}$ , and, thus, the restoring force  $ku$  and  $k$  are independent of the rate or frequency at which the displacement varies.

The same concept of stiffness can be applied to a harmonically vibrating column with mass uniformly distributed along its length (Fig. 5.2b(a)). For an approximate analysis at low frequency, the distributed column mass can be replaced by a concentrated (lumped) mass  $m_s$ , attached to the top of the column, and the column itself can be considered massless (Fig. 5.2b(b)). Consequently, a static stiffness constant  $k_{st}$ , independent of frequency, can be used to describe the stiffness of this massless column. The elastic force in the column just below the mass is  $k_{st}u$  for any displacement  $u$ . Nevertheless, the total restoring force generated by the column at the

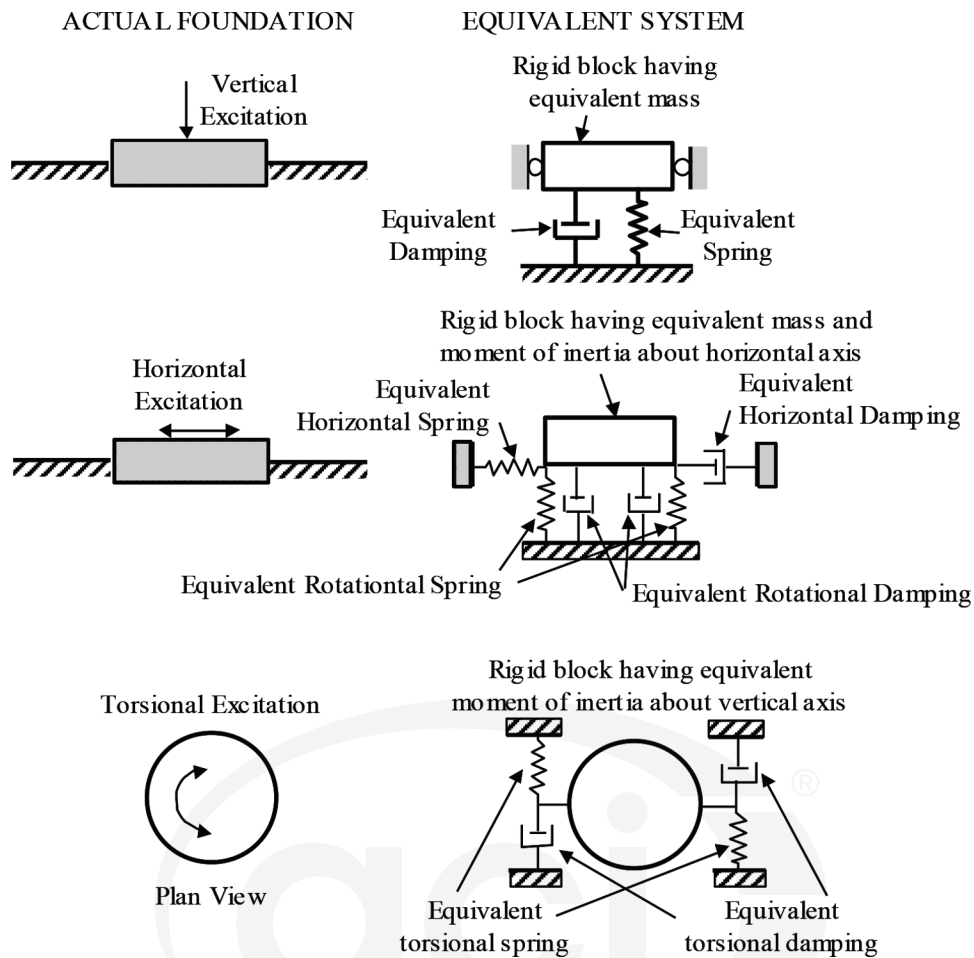


Fig. 5.2a—Lumped system for a foundation subjected to vertical, horizontal, and torsional excitation forces (Richart et al. 1970).

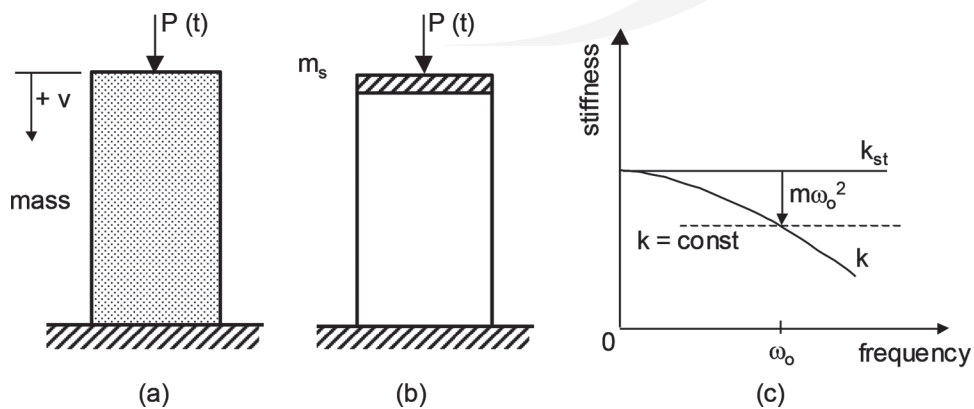


Fig. 5.2b—Effect of mass on dynamic stiffness.

top of the lumped mass is the sum of the elastic force in the column and the inertia force of the mass. If the displacement of this mass varies harmonically

$$u(t) = u_o \cos \omega t \quad (5.2a)$$

Then, the acceleration is

$$\ddot{u} = \frac{d^2 u}{dt^2} = -u_o \omega^2 \cos \omega t \quad (5.2b)$$

and the inertia force is

$$m_s \ddot{u} = -m_s u_o \omega^2 \cos \omega t \quad (5.2c)$$

In the absence of damping, the time-dependent relation of the external harmonic force applied with amplitude  $P_o$  and frequency  $\omega_o$  to the displacement  $u$  is

$$P(t) = P_o \cos \omega t = k_{st} u_o \cos \omega t - m_s u_o \omega^2 \cos \omega t \quad (5.2d)$$

and the amplitudes of force and displacement relate as

$$P_o = k_{st} u_o - m_s u_o \omega^2 \quad (5.2e)$$

The dynamic stiffness, being the constant of proportionality between the applied force and displacement, becomes

$$k = k_{st} - m_s \omega^2 \quad (5.2f)$$

Thus, with vibration of an element having distributed mass, the dynamic stiffness generally varies with frequency. At low frequency, this variation is sometimes close to parabolic, as shown in Fig. 5.2b(c). The column used in this example may represent the column of soil and, thus, a soil deposit always features stiffness parameters that are frequency dependent. The magnitude and character of this frequency effect depend on the size of the foundation geometry, vibration mode, soil layering, and other factors.

The dashpots of Fig. 5.2a are often represented as viscous dampers producing forces proportional to the vibration velocity  $\dot{u}$  of the mass. The magnitude of the damper force is

$$F_D = c \dot{u} = c \frac{du}{dt} \quad (5.2g)$$

The damping constant  $c$  is defined as the force associated with a unit velocity. During harmonic vibrations, the time-dependent viscous damping force is

$$c \dot{u} = -c u_o \omega \sin \omega t \quad (5.2h)$$

and its peak value is  $c u_o \omega$ . Equation (5.2h) shows that for a given constant  $c$  and displacement amplitude  $u_o$ , the amplitude of viscous damping force is proportional to frequency (Fig. 5.2c).

### 5.3—Calculation of dynamic foundation impedances

The calculation of dynamic impedances is fundamental for the dynamic analysis and design of machine foundations. For instance, as previously discussed, six lumped dynamic impedances—three translational and three rotational dynamic impedances—are needed to model the dynamic response of an infinitely rigid foundation. These impedances are defined and calculated as explained in the following.

**5.3.1 Dynamic impedance definition**—The response of an infinitely rigid foundation to static or dynamic load,  $P$  or  $P(\omega t)$ , arises solely from the deformation,  $U$  or  $U(\omega t + \phi)$ , of the supporting soil. In particular, the static foundation stiffness,  $k_{st} = P/U$ , is used for modeling the soil-foundation response to static loads. In an analogous manner, the

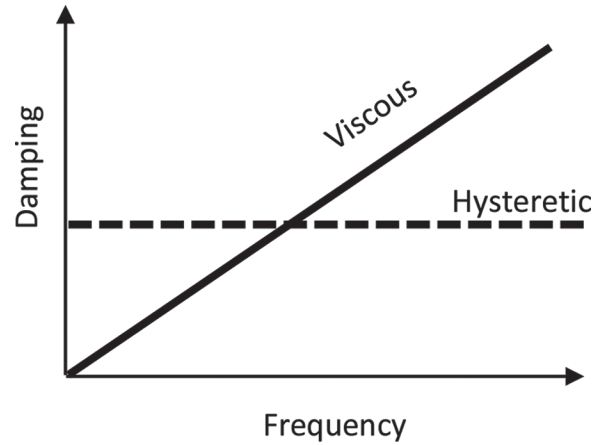


Fig. 5.2c—Comparison of viscous and hysteretic damping.

dynamic foundation impedance,  $k^*(\omega) = P(\omega t)/U(\omega t + \phi)$ , is used for modeling the soil-foundation response to a harmonic (dynamic) load of angular frequency  $\omega$  (for example, due to rotating machinery). The main difference between the static,  $k_{st}$ , and dynamic,  $k^*(\omega)$ , impedance is that in the static case, the load ( $P$ ) and displacement ( $U$ ) occur simultaneously, that is,  $P$  and  $U$  are in phase, whereas in the dynamic case, the displacement response is frequency dependent and typically lags the loading; that is, there is a phase difference between the two. Therefore, for mathematical convenience, complex notation is used to represent the dynamic impedance of a foundation, given that complex numbers encapsulate magnitude and phase information in a single entity/number. It should be emphasized, however, that the loading and response (vibration amplitude) of a machine foundation are always real; complex numbers are used just for mathematical convenience given that they greatly simplify calculations involving harmonic functions, such the loading acting in machine foundations.

**5.3.2 Dynamic impedance calculation**—The procedure used to calculate the dynamic impedances of a rigid surface foundation can be summarized in the following steps:

a) Model the foundation as massless and infinitely rigid; therefore, only the geometry of the area in contact with the soil is required. The use of a massless foundation is important because it avoids the need for recalculating the dynamic impedance every time the foundation mass changes, which often happens during the design process.

b) Apply a harmonic force or moment of frequency  $\omega$  to the rigid foundation ( $P(t) = P_o e^{i\omega t}$  or  $M(t) = M_o e^{i\omega t}$ ), as shown in Fig. 5.3.2, where  $i = \sqrt{-1}$ . Such force/moment excites the foundation, which in turn generates stress waves that propagate into the underlying soil. Where the soil is modeled as a viscoelastic material, therefore, the following properties are needed for each soil layer supporting the foundation: thickness ( $h$ ), modulus of elasticity ( $E_s$ ), Poisson's ratio ( $\nu$ ), density ( $\rho$ ), and material damping ( $\beta_m$ ).

c) Calculate the vibration response ( $U(\omega t + \phi) = U_o e^{i\omega t + i\phi}$  or  $\theta(\omega t + \phi) = \theta_o e^{i\omega t + i\phi}$ ) of the foundation under the applied harmonic excitation. It should be noted that the steady-state vibration amplitude is obtained by keeping track of the

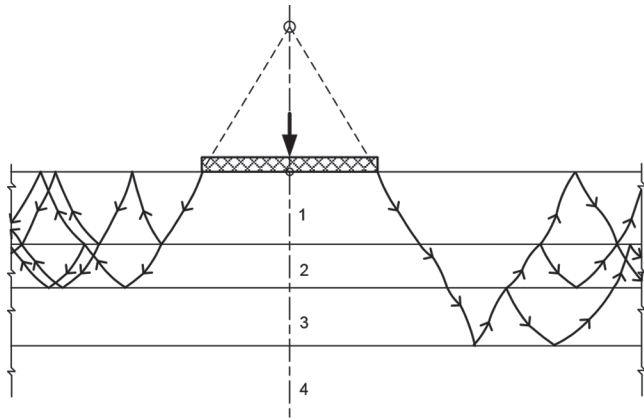


Fig. 5.3.2—Downward and upward wave propagation for surface disk in layered half-space (Wolf and Deeks 2004).

reflections and refractions that take place every time that a stress wave, generated by the foundation vibration, reaches a soil layer boundary. This is achieved by finding the different wave paths shown in Fig. 5.3.2, which can be done analytically or numerically. Note that  $\phi$  represents the phase difference (lag) of the response relative to the applied force or moment.

d) Calculate the translational/rotational dynamic impedance  $k^*(\omega)$  as the ratio between the harmonic excitation (force/moment) acting on the foundation and its corresponding vibration amplitude, as shown in Eq. (5.3.2a) and (5.3.2b). It is important to note that  $k^*(\omega)$  is a frequency-dependent quantity with a real (Re) and imaginary (Im) part.

$$k^*(\omega) = \frac{P(\omega t)}{U(\omega t + \phi)} = \frac{P_0 e^{i\omega t}}{U_0 e^{i\omega t + i\phi}} = \frac{P_0}{U_0} e^{-i\phi} \quad (5.3.2a)$$

$$k^*(\omega) = \frac{M(\omega t)}{\theta(\omega t + \phi)} = \frac{M_0 e^{i\omega t}}{\theta_0 e^{i\omega t + i\phi}} = \frac{M_0}{\theta_0} e^{-i\phi} \quad (5.3.2b)$$

e) Separate the real and imaginary parts of the dynamic impedance to streamline its use in computer codes used for structural analysis. It is customary to express the complex dynamic impedance as shown in Eq. (5.3.2c). The real and imaginary parts of the dynamic impedance are associated, by analogy to the system described in 5.2, with a dynamic (frequency-dependent) spring and dashpot as shown in Eq. (5.3.2d) and (5.3.2e).

$$k^*(\omega) = k + i\omega C \quad (5.3.2c)$$

$$k(\omega) = \text{Re}(k^*(\omega)) = \frac{P_0}{U_0} \cos \phi \quad (5.3.2d)$$

$$C(\omega) = \frac{\text{Im}(k^*(\omega))}{\omega} = -\frac{P_0}{\omega U_0} \sin \phi \quad (5.3.2e)$$

Equations (5.3.2c) and (5.3.2d) are analogous to Eq. (5.3.2b).

f) Repeat steps (b) to (e) for each frequency of interest until the range of vibration frequencies of the machine is covered.

The aforementioned approach may also be used for calculating the dynamic impedance of a single pile or pile foundation. Because, in this case, the mass and flexibility of each pile are considered, the pile impedance is a function of the pile stiffness and density per unit length. Furthermore, the concept of dynamic impedance can be extended to flexible foundations; however, in this case, it is more natural to define a dynamic impedance matrix. Details regarding the generalization of the dynamic impedance concept can be found in [Wolf and Song \(1996\)](#) and [Lysmer et al. \(1999\)](#).

## 5.4—Dynamic impedance of soil-supported foundations

The engineer can calculate the dynamic impedance of a soil-supported foundation using the steps presented in 5.3.2 in a finite or boundary element code. Many foundation impedances of practical importance, however, have been published in the literature as closed-form solutions, tables, or charts. Therefore, for application purposes, the engineer can estimate the dynamic impedance,  $k^*(\omega)$ , of a soil-supported foundation by one or a combination of the following approaches:

a) *Frequency-dependent equations*—[Veletsos and Nair \(1974\)](#), [Veletsos and Verbic \(1973\)](#), and [Veletsos and Wei \(1971\)](#) have developed appropriate equations that represent the impedance to motion offered by uniform soil conditions. They developed these formulations using an assumption of a uniform elastic or viscoelastic half-space and the related motion of a rigid or flexible foundation on this half-space. Motions can be translational or rotational. While the calculations can be manually tedious, computer implementation provides acceptable efficiency. Some of the simpler equations are presented in 5.4.1.2 of this report. More extensive equations are presented in [Gazetas \(1991\)](#) and [Sieffert and Cevaer \(1992\)](#).

b) *Constant approximation*—If the frequency range of interest is not very wide, an often-satisfactory technique is to replace the variable dynamic stiffness by the constant representative of the stiffness in the vicinity of the dominant frequency (Fig. 5.2b(c)). Another constant approximation can be to add an effective or fictitious in-phase mass of the supporting soil medium to the vibrating lumped mass of the machine and foundation and to consider stiffness to be constant and equal to the static stiffness. Nevertheless, the variation of dynamic stiffness with frequency can be represented by a parabola only in some cases, and the added mass is not the same for all vibration modes.

c) *Computer-based numerical analysis*—For complex geometry, flexible foundations, or variable soil conditions, the recommended approach is to obtain the stiffness and damping of foundations using dynamic analysis of a three- or two-dimensional continuum representing the soil medium. The continuum is modeled as an elastic or viscoelastic half-space. The half-space can be homogeneous or nonhomogeneous.